CONTROL VALVE SOURCEBOOK

POWER & SEVERE SERVICE

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EMERSON™
Process Management

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Chapter 1

Control Valve Selection

Picking a control valve for a particular application used to be straightforward. Usually only one general type of valve was considered, a sliding-stem valve. Each manufacturer offered a product suitable for the job, and the choice among them depended upon such obvious matters as cost, delivery, vendor relationships and user preference.

Today, control valve selection is considerably more complex, especially for engineers with limited experience or those who have not kept up with changes in the control valve industry.

For many applications, an assortment of sliding-stem and rotary valve styles are available. Some are touted as “universal” valves for almost any size and service, while others are claimed to be optimum solutions for narrowly defined needs. Even the most knowledgeable user may wonder whether he is really getting the most for his money in the control valves he has been specifying.

Like most decisions, selection of a control valve involves a great number of variables. Presented here is an overview of the selection process. The discussion includes categorization of available valve types and a set of criteria to be considered in the selection process.

***Figure 1-1**. Modern control valve combines actuator, valve assembly and digital valve controller to provide maximum performance in a wide variety of control applications.*

General Categories of Control Valves

“Control valve” in this discussion means any power-operated valve, whether used for throttling or for on-off control. Other valve varieties such as motorized gate valves, louvers, pinch valves and self-operated regulators are not considered here.

The major valve types, sliding-stem and rotary, are further divided within Table 1-2 into a total of nine subcategories according to relative performance and cost. Despite variations found within each category, such as cage-guiding versus stem-guiding, all valves within a given subcategory may be considered very much alike in the early stages of the selection process.

Selecting a valve involves narrowing down to one of these nine subcategories and then comparing specific valves in that group.

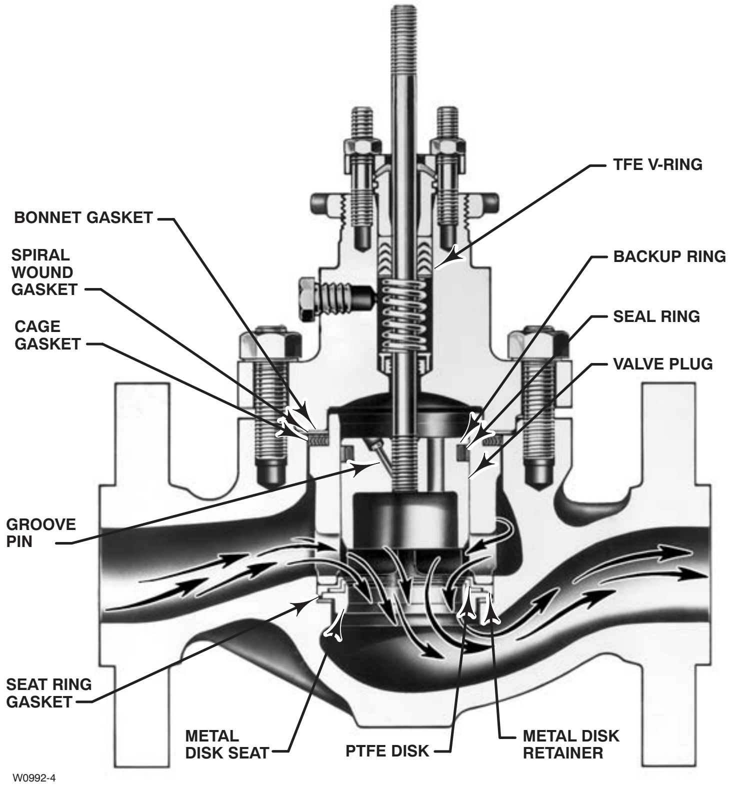

Figure 1-2. *Standard globe sliding-stem valve design is typified by the Design ET. A broad range is available in sizes, materials and end connections. The balanced plug reduces plug force and allows use of smaller actuators. These valves are the first choice for applications less than 3-inch size.*

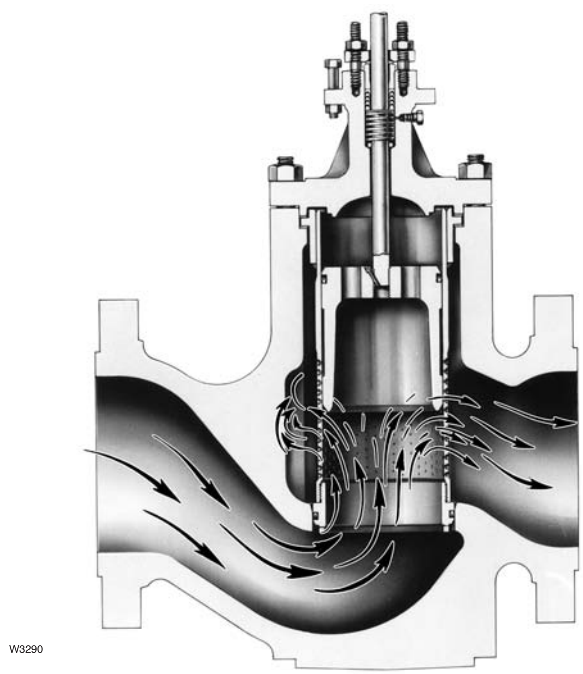

Figure 1-3. *Severe service capability in globe valves demonstrated by this large Design EWNT-2. The drilled hole cage provides attenuation of flow noise by splitting the flow into multiple passages. Spacing of the holes is carefully controlled to eliminate jet interaction and higher resultant noise levels.*

Sliding-stem Valves

The most versatile of control valves are the sliding-stem designs. Globe, angle, and Y-pattern valves can be purchased in sizes ranging from 1/2 through 36-inch. More choices of materials, end connections and control characteristics are included here than in any other product family.

Globe valves are available in cage-guided, post-guided and stem-guided designs, with flanged, screwed or weld ends. Economical cast iron as well as carbon steel, stainless steel and other high-performance body materials are available. Available valve body pressure ratings range to ANSI Class 2500 and beyond.

The globe valve's precise throttling capabilities, overall performance and general sturdiness make them a bargain despite their slight cost premium. With the globe valve, the buyer gets a rugged, dependable unit intended for long, trouble-free service.

Sliding-stem valves are ruggedly built to handle field conditions such as piping stress, vibration, and temperature changes. In sizes through 3-inch, incremental costs over rotary valves are low in contrast to the increments in benefits received.

For many extreme service conditions, sliding-stem valves are the only suitable choice. These situations include high pressures and temperatures, excessive noise, and the potential for cavitation. Due to process demands, these applications require the rugged construction of the sliding-stem design.

Barstock valves are small, economical sliding-stem valves featuring bodies machined from bar stock (Figure 1-5). Body sizes range from fractions of an inch up to 3-inch; flow capacities generally are lower than those of general purpose valves. End connections usually are flangeless (for mounting between piping flanges) or screwed.

The main advantage of the barstock valve is that far more materials are readily available in bar form than in cast form. Consequently, these valves are often used where there are special corrosion considerations.

Also, the compactness and general high-quality construction of barstock valves make them attractive for low flow rate applications. Overall, they are an economical choice when they can be used.

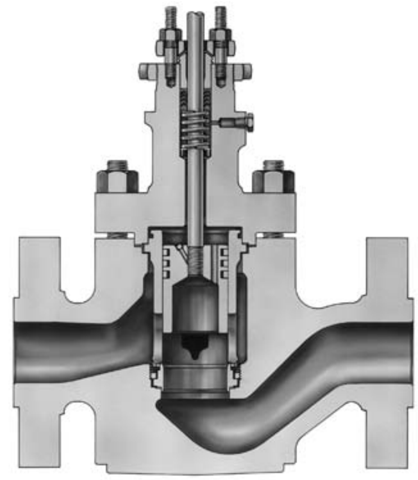

Figure 1-4. *Design EHD is typical of high-pressure globe valves. Rated at ANSI Class 2500, it provides throttling control of high-pressure steam and fluids. Anti-noise and anti-cavitation trims are available to handle flow problems.*

Figure 1-5. *Bar stock valves such as this Baumann 24000SB provide economical solutions to small flow requirements. It is capable of pressures to 1500 psi and temperatures to 450 °F. Compact spring and diaphragm actuators complement these small bodies.*

The third subcategory, which involves the lowest cost products among sliding-stem valves, utilizes "economy" bodies (Figure 1-6). These valves are used for low-pressure steam, air and water applications that are not demanding. Sizes available range from 1/2 up to 4-inch.

W7618

Figure 1-6. *This screwed end bronze body Baumann "Little Scotty" is capable of handling many utility applications. It is complemented by a wide variety of orifice sizes.*

Body materials include bronze, cast iron, steel and SST. Pressure classes generally stop at ANSI Class 300. Compared to regular sliding-stem valves, these units are very simple; their actuators are smaller, and their construction is economical. Severe service trims for noise and cavitation service are not available in these products.

Ball Valves

There are two subcategories of ball valves. The through-bore or full-ball type illustrated in Figure 1-8 is used typically for high pressure drop throttling and on-off applications in sizes to 24-inch. Full port designs exhibit high flow capacity and low susceptibility to wear by erosive streams. However, sluggish flow response in the first 20% of ball travel may make full-bore ball valves unsuitable for some throttling applications.

Another popular kind of ball valve is the segmented-ball style (Figure 1-9). This subcategory utilizes a reduced bore, and the edge of the ball segment has a contoured notch shape for better throttling control and higher rangeability. Intended primarily for modulating service, segmented-ball valves are generally higher in overall control performance than full-ball products.

W7960-2

Figure 1-7. *Compact, lightweight valve featuring a piston actuator and a FIELDVUE Digital Valve Controller.*

Segmented-ball valves with their splined shaft connections are engineered to eliminate lost motion, which is detrimental to performance. The use of heavy-duty metal and fluoroplastic sealing elements allows wide temperature and fluid applicability. Their straight-through flow design achieves high capacity with sizes ranging through 24-inch and pressure ratings to ANSI Class 600. Price is generally lower than that of globe valves.

Eccentric Plug Valves

This class of valves combines many features of sliding-stem and rotary products. Eccentric plug valves feature rotary actuation. But, unlike most rotary valves, this product utilizes a massive, rigid seat design.

Eccentric-plug valves (Figure 1-10) exhibit excellent throttling capability and resistance to erosion. Thus, many of the good aspects of both rotary and sliding-stem designs are combined.

Sizes generally range through 8-inch; pressure ratings go to ANSI Class 600. Both flanged and flangeless body styles are available in a variety of materials.

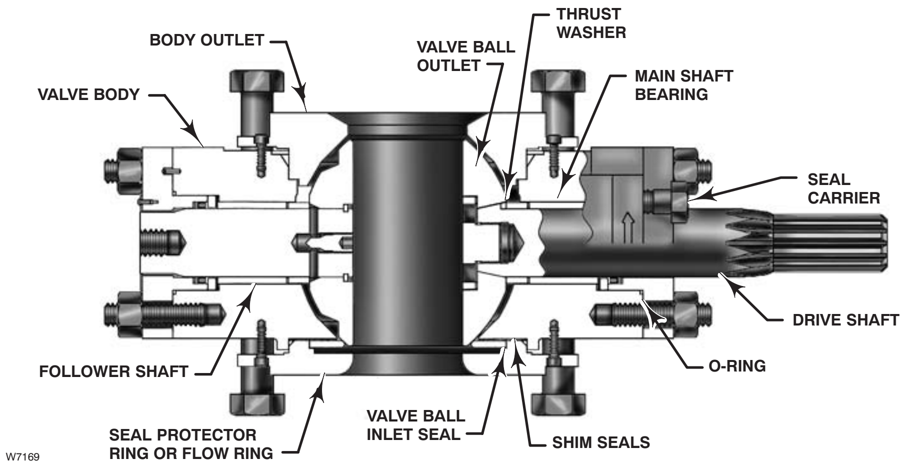

***Figure 1-8**. High-pressure ball valves feature heavy shafts and full ball designs. This Type V250 is suitable for pressure drops to 2220 psi. ANSI Class 600 and 900 bodies are available—sizes range to 24-inch.*

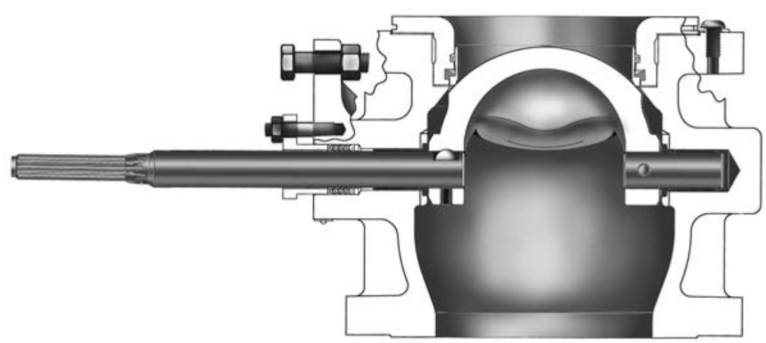

***Figure 1-9.** Applications to ANSI Class 600 can be handled by the Design V150/V200/V300 Vee-Ball. This product incorporates many features to improve throttling performance and rangeability. Tight shutoff is achieved by using either heavy-duty metal seals or composition seals.*

Butterfly Valves

Butterfly valves are divided into three subcategories: 1) swing-through, 2) lined, and 3) high-performance.

The most rudimentary is the swing-through design. Rather like a stovepipe damper, but considerably more sophisticated, the valve disk in this valve design swings close to, but clear of the body's inner wall.

Such a valve is used for throttling applications that do not require shutoff tighter than about one percent of full flow. Sizes range from 2 to 96-inch, while body materials include cast iron, carbon steel, or stainless steel. Mounting is flangeless, lugged, or welded, and body pressure ratings up to ANSI Class 2500 are common. While suited to a wide range of temperatures, the swing-through style of butterfly valve is handicapped by the lack of tight shutoff.

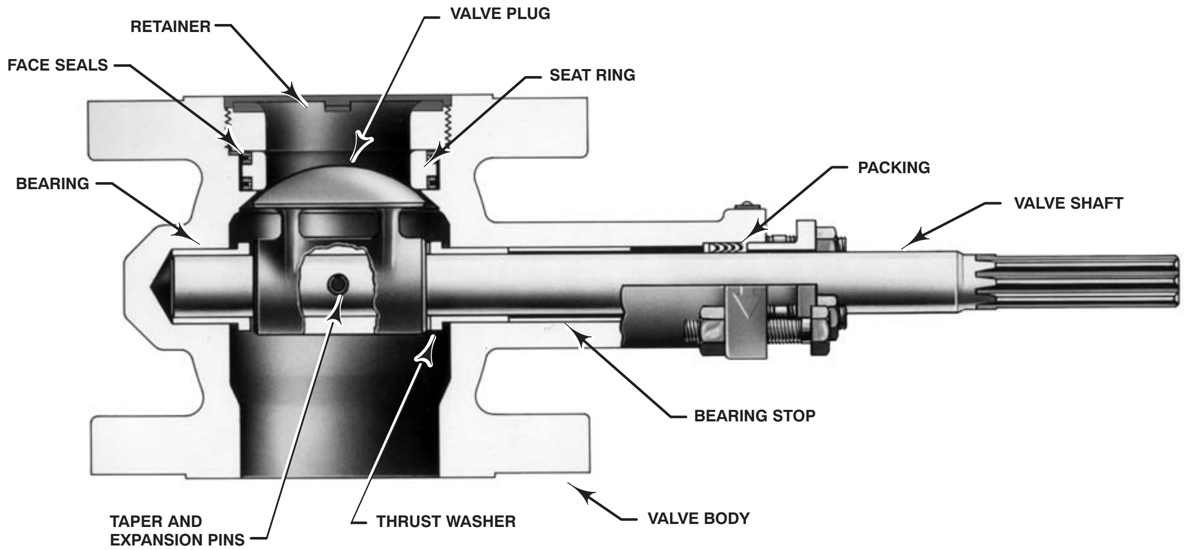

Figure 1-10. *The V500 eccentric plug valve is specially designed for severe rotary applications. Since the valve plug "cams" into the seat ring upon closure, it features tight shutoff with globe valve style seating. It also offers excellent resistance to abrasive wear and flashing induced erosion.*

Figure 1-11. *Swing-through butterfly valves provide an economical solution to high flow rate throttling applications. Leakage is higher than other designs, as no sealing is employed.*

Figure 1-12. *Lined butterfly valves offer tight shutoff but are limited to low temperature applications. The liner material keeps the process away from the metal body eliminating many corrosion problems.*

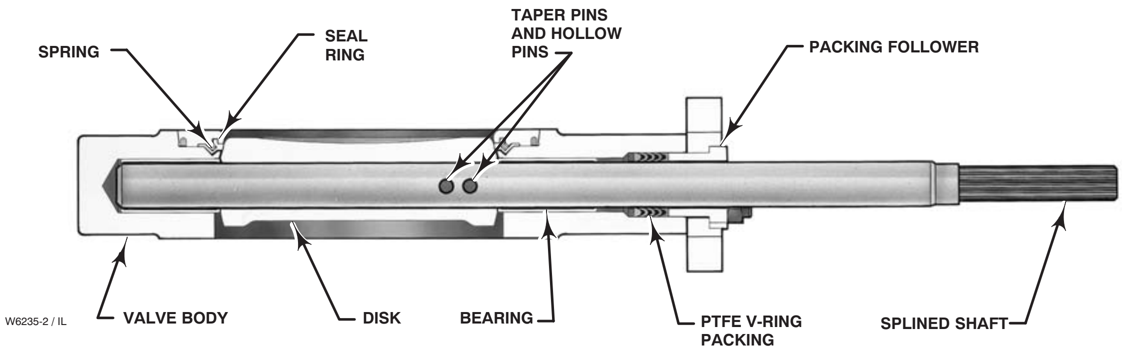

Figure 1-13. *High performance butterfly valves provide excellent performance and value. High-pressure capability, tight shutoff and excellent control are featured as standard. This Type 8560 design is made for ANSI Class 150 applications.*

The requirement for zero or low leakage gave rise to new designs such as the lined and high performance butterfly valves (BFV's).

Lined butterfly valves feature an elastomer or fluoropolymer (TFE) lining that contacts the disk to provide tight shutoff (Figure 1-12). Since this valve design depends on interference between the disc and liner for shutoff, it is more limited in pressure drop. Temperature ranges typically are limited due to the elastomer disk seal.

In addition to serving as the disk seal, the liner also protects the inner bore of the valve body from the process fluid, which allows these valves to be used in many corrosive situations. Elastomer-lined BFV's are generally the lowest price products available as control valves in medium to large sizes.

Heavy shafts and discs, full-rated bodies and sophisticated seals that allow tight shutoff against high pressures characterize high performance butterfly valves such as the one in Figure 1-13. These valves provide an excellent combination of performance features, lightweight and very reasonable pricing.

In offset disc designs, eccentric shaft mounting allows the valve disc to swing clear of its seal to minimize wear and torque. The offset disc allows uninterrupted sealing and a seal ring that can be replaced without removing the disc. High performance butterfly valves come in sizes from 2 through 72-inch, with flangeless or lugged connections, carbon steel or stainless steel bodies, and pressure ratings to ANSI Class 600. Tight metal-to-metal seals, made possible with advanced eccentric design, provide tight shutoff in applications that are too hot for elastomer lined valves. Considering this tight shutoff capability and heavy-duty construction, high performance butterfly valves can be a suitable option to many applications where sliding-stem valves are normally specified.

General Selection Criteria

Most of the considerations that guide the selection of valve type and brand are rather basic. However, there are some matters that may be overlooked by users whose familiarity is mainly limited to just one or a few valve types. Table 1-1 below provides a checklist of important criteria; each is discussed at length following the table.

Table 1-1. *Suggested General Criteria for Selecting Type and Brand of Control Valve*

Body pressure rating
High and low temperature limits
Material compatibility and durability
Inherent flow characteristic and rangeability
Maximum pressure drop (shutoff and flowing)
Noise and cavitation
End connections
Shutoff leakage
Capacity versus cost
Nature of flowing media
Dynamic performance

Pressure Ratings

Body pressure ratings ordinarily are considered according to ANSI pressure classes—the most common ones for steel and stainless steel being Classes 150, 300 and 600. (Source documents are ASME/ANSI Standards B16.34, "Steel Valves," and ANSI B16.1, "Cast Iron Pipe Flanges and Flanged Fittings.") For a given body material, each ANSI Class corresponds to a prescribed profile of maximum pressures that decrease with temperature according to the strength of the material. Each material also has a minimum and maximum service temperature based on loss of ductility or loss of strength. For most applications, the required pressure rating is dictated by the application. However, since all products are not available for all ANSI Classes, it is an important consideration for selection.

Temperature Considerations

Required temperature capabilities are also a foregone conclusion, but one that is likely to narrow valve selection possibilities. The considerations include the strength or ductility of the body material as well as relative thermal expansion of various parts.

Temperature limits also may be imposed due to disintegration of soft parts at high temperatures or loss of resiliency at low temperatures. The soft materials under consideration include various elastomers, plastics, and PTFE. They may be found in parts such as seat rings, seal or piston rings, packing, rotary shaft bearings and butterfly valve liners. Typical upper temperature limits for elastomers are in the 200-350°F range, and the general limit for PTFE is 450°F.

Temperature affects valve selection by excluding certain valves that do not have high- or low-temperature options. It also may have some effect on the valve's performance. For instance, going from PTFE to metal seals for high temperatures generally increases the shutoff leakage flow. Similarly, high temperature metal bearing sleeves in rotary valves impose more friction on the shaft than do PTFE bearings, so that the shaft cannot withstand as high a pressure-drop load at shutoff. Selection of valve packing is also based largely on service temperature.

Material Selection

The third criterion in Table 1-1, *material compatibility and durability*, is a more complex consideration. At issue may be corrosion by the process fluid, erosion by abrasive material, flashing, cavitation or simply a matter of process pressure and temperature. The piping material usually indicates the body material. However, since velocity is higher in valves other factors must be considered. When these items are included, often valve and piping materials will differ.

The trim materials, in turn, are usually a function of the body material, temperature range and qualities of the fluid. When a body material other than carbon, alloy or stainless steel is required, use of an alternate valve type such as lined or bar stock should be considered.

Flow Characteristic

The next selection criterion, *inherent flow characteristic,* refers to the pattern in which the flow at constant pressure drop changes according to valve position.

Typical characteristics are quick-opening, linear and equal-percentage. The choice of characteristic may have a strong influence on the stability or controllability of the process (see Table 1-3), since it represents the change of valve gain relative to travel.

Most control valves are carefully "characterized" by means of contours on a plug, cage, or ball element. Some valves are available in a variety of characteristics to suit the application, while others offer little or no choice. To quantitatively determine the best flow characteristic for a given application, a dynamic analysis of the control loop can be performed. In most cases, however, this is unnecessary; reference to established rules of thumb will suffice.

The accompanying drawing illustrates typical flow characteristic curves (Figure 1-14). The quick opening flow characteristic provides for maximum change in flow rate at low valve travels with a fairly linear relationship. Additional increases in valve travel give sharply reduced changes in flow rate, and when the valve plug nears the wide open position, the change in flow rate approaches zero. In a control valve, the quick opening valve plug is used primarily for on-off service; but it is also suitable for many applications where a linear valve plug would normally be specified.

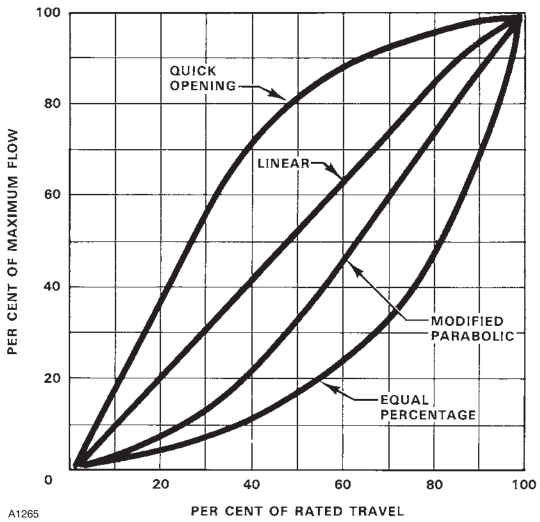

Figure 1-14. *Many control valves offer a choice of characteristic. Selection to match process requirements is guided by simple rules. Adherence to these guidelines will help assure stable operation.*

The linear flow characteristic curve shows that the flow rate is directly proportional to the valve travel. This proportional relationship produces a characteristic with a constant slope so that, with constant pressure drop, the valve gain will be the same at all flows. The linear valve plug is commonly specified for liquid level control and for certain flow control applications requiring constant gain.

In the equal percentage flow characteristic, equal increments of valve travel produce equal percentage changes in the existing flow. The change in flow rate is always proportional to the flow rate just before the change in valve plug, disc, or ball position is made. When the valve plug, disc, or ball is near its seat and the flow is small, the change in flow rate will be small; with a large flow, the change in flow rate will be large.

An equal percentage flow characteristic is used generally on pressure control applications and on other applications where a large percentage of the pressure drop is normally absorbed by the system itself with only a relatively small percentage available at the control valve. Valves with an equal percentage characteristic should also be considered where highly varying pressure drop conditions could be expected.

Rangeability

Another aspect of a valve's flow characteristic is its rangeability, which is the ratio of its maximum and minimum controllable flow rates. Exceptionally wide rangeability may be required for certain applications to handle wide load swings or a combination of start-up, normal and maximum working conditions. Generally speaking, rotary valves—especially partial ball valves—have greater rangeability than sliding-stem varieties.

Use of Positioners

A positioner is an instrument that helps improve control by accurately positioning a control valve actuator in response to a control signal. They are useful in many applications and are required with certain actuator styles in order to match actuator and instrument pressure signals or to provide operating stability. To a certain extent, a valve with one inherent flow characteristic can also be made to perform as though it had a different characteristic by using a nonlinear (i.e., characterized) positioner-actuator combination. The limitation of this approach lies in the positioner's frequency response and phase lag compared to the characteristic frequency of the process.

Although it is common practice to use a positioner on every valve application, each application should be reviewed carefully. There are certain examples of high gain processes where a positioner can hinder valve performance.

Pressure Drop

The maximum pressure drop a valve can tolerate at shutoff or when partially or fully open is an important selection criteria. Sliding-stem valves are generally superior in both regards because of the rugged nature of their moving parts. Many rotary valves are limited to pressure drops well below the body pressure rating, especially under flowing conditions, due to dynamic stresses that high velocity flow imposes on the disk or ball segment.

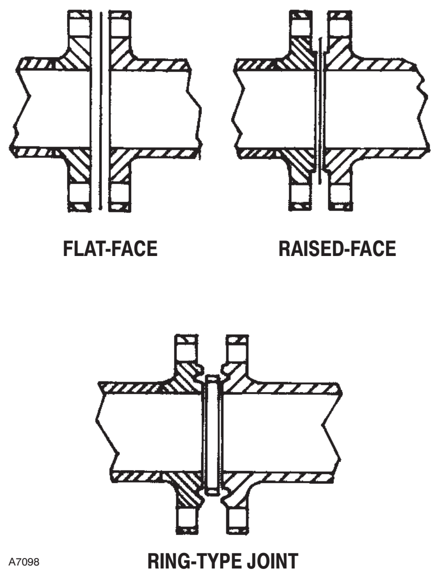

Figure 1-15. *Popular varieties of bolted flange end connections.*

Noise and Cavitation

Noise and cavitation are two considerations that often are grouped together because both result from high pressure drops and large flow rates. They are treated by special modifications to standard valves. Chapter 5 discusses noise generation and abatement, while Chapter 6 discusses the cavitation phenomenon and its impact and treatment.

End Connections

The three common methods of installing control valves in pipelines are by means of screwed pipe threads, bolted flanges, and welded end connections. At some point in the selection process, the valve's end connections must be considered with the question simply being whether the desired connection style is available in the valve being considered.

In some situations, this matter can limit the selection rather narrowly. For instance, if a piping specification calls for welded connections only, the choice usually is limited to sliding-stem valves.

Screwed end connections, popular in small control valves, offer more economy than flanged ends. The threads usually specified are tapered female NPT (National Pipe Thread) on the valve body. They form a metal-to-metal seal by wedging over the mating male threads on the pipeline ends. This connection style is usually limited to valves not larger than 2-inch, and is not recommended for elevated temperature service.

Valve maintenance might be complicated by screwed end connections if it is necessary to take the body out of the pipeline. Screwed connections require breaking a flanged joint or union connection to permit unscrewing the valve body from the pipeline.

Flanged end valves are easily removed from the piping and are suitable for use through the range of working pressures that most control valves are manufactured (Figure 1-15).

Flanged end connections can be used in a temperature range from absolute zero (–273°F) to approximately 1500°F (815°C). They are used on all valve sizes. The most common flanged end connections include flat face, raised face, and ring type joint.

Welded ends on control valves are leak tight at all pressures and temperatures and are economical in initial cost (Figure 1-16). Welded end valves are more difficult to remove from the line and are obviously limited to weldable materials. Welded ends come in two styles, socket weld and buttweld.

Shutoff Capability

Some consideration must be given to a valve's shutoff capability, which usually is rated in terms of Classes specified in ANSI/FCI70-2 (Table 1-4). In service, shutoff leakage depends on many factors, including but not limited to, pressure drop, temperature and the condition of the sealing surfaces.

Since shutoff ratings are based on standard test conditions that can be very different from service conditions, service leakage cannot be predicted accurately. However, the shutoff Class provides a good basis for comparison among valves of similar configuration. It is not uncommon for valve users to overestimate the shutoff class required.

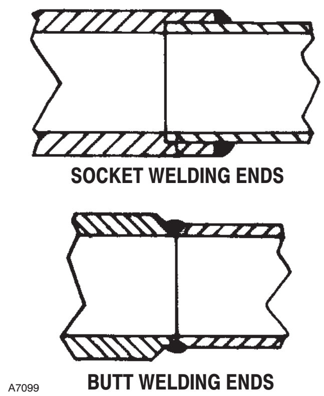

Figure 1-16. *Common welded end connections.*

Since tight shutoff valves generally cost more both in initial cost as well as in later maintenance expense, serious consideration is warranted. Tight shutoff is particularly important in high-pressure valves, considering that leakage in these applications can lead to the ultimate destruction of the trim. Special precautions in seat material selection, seat preparation and seat load are necessary to ensure success.

Flow Capacity

Finally, the criterion of capacity or size can be an overriding constraint on selection. For very large lines, sliding-stem valves are more expensive than rotary types. On the other hand, for very small flows, a suitable rotary valve may not be available.

If future plans call for significantly larger flow, then a sliding-stem valve with replaceable restricted trim may be the answer. The trim can be changed to full size trim to accommodate higher flow rates at less cost than replacing the entire valve body assembly.

Rotary style products generally have much higher maximum capacity than sliding-stem valves for a given body size. This fact makes rotary products attractive in applications where the pressure drop available is rather small. However, it is of little or no advantage in high-pressure drop applications such as pressure regulation or letdown.

Conclusion

We may simplify the process of selection as follows:

For most general applications, it makes sense both economically as well as technically to use sliding-stem valves for lower flow ranges, ball valves for intermediate capacities, and high performance butterfly valves for the very largest required flows.

For sizes less than 3-inch, general-purpose sliding-stem valves provide an exceptional value. For a minimal price premium over rotary products, they offer unparalleled performance, flexibility and service life. In 3-inch size and larger, the premium for these devices over rotary products is warranted. For severe service applications the most frequently used and often the only available product is the sliding-stem valve.

Applications ranging from 4-inch to 6-inch are best served by such transitional valve styles as the eccentric plug valve or the ball valve. These products have minimal body material and therefore lower cost. They also offer higher capacity levels than globe designs.

In sizes 8-inch and larger, applications tend to have lower pressures and pressure drops than in smaller valve sizes. This gives rise to the possibility of using high-performance butterfly valves. These valves are economical, can offer tight shut-off as well as provide good control capability, not to mention capacity benefits well beyond those of globe and high-performance rotary valves.

Special considerations may require out of the ordinary valve solutions. There are valve designs and special trims available to handle high noise applications, cavitation, high pressure, high temperature and combinations of these conditions.

After going through all the criteria for a given application, the selection process may point to several types of valves. From there on, selection becomes a matter of price versus capability as discussed here, coupled with the inevitable personal and institutional preferences. Since no single control valve package is cost-effective over the full range of applications, it is important to keep an open mind to alternative choices.

__Table 1-2.__ Major Categories and Subcategories of Control Valves, with Typical General Characteristics

Valve Style	Main Characteristics	Typical Size Range, inches	Typical Standard Body Materials	Typical Standard End Connection	Typical Pressure Ratings	Relative Flow Capacity	Relative Shutoff Capability
Regular Sliding-stem	Heavy Duty Versatile	1 to 24	Carbon Steel Cast Iron Stainless	ANSI Flanged Welded Screwed	To ANSI 2500	Moderate	Excellent
Bar Stock	Machined from Bar Stock	½ to 3	Variety of Alloys	Flangeless Screwed	To ANSI 600	Low	Excellent
Economy Sliding-stem	Light Duty Inexpensive	½ to 2	Bronze Cast Iron Carbon Steel	Screwed	To ANSI 125	Moderate	Good
Thru-Bore Ball	On-Off Service	1 to 24	Carbon Steel Stainless	Flangeless	To ANSI 900	High	Excellent
Partial Ball	Characterized for Throttling	1 to 24	Carbon Steel Stainless	Flangeless Flanged	To ANSI 600	High	Excellent
Eccentric Plug	Erosion Resistance	1 to 8	Carbon Steel Stainless	Flanged	To ANSI 600	Moderate	Excellent
Swing-Thru Butterfly	No Seal	2 to 96	Carbon Steel Cast Iron Stainless	Flangeless Lugged Welded	To ANSI 2500	High	Poor
Lined Butterfly	Elastomer or TFE Liner	2 to 96	Carbon Steel Cast Iron Stainless	Flangeless Lugged	To ANSI 300	High	Good
High Performance Butterfly	Offset Disk General Service	2 to 72	Carbon Steel Stainless	Flangeless Lugged	To ANSI 600	High	Excellent

Table 1-3. *Control Valve Characteristic Recommendations*

Liquid Level Systems

Control Valve Pressure Drop	Best Inherent Characteristic
Constant ΔP	Linear
Decreasing ΔP with increasing load, ΔP at maximum load > 20% of minimum load ΔP	Linear
Decreasing ΔP with increasing load, ΔP at maximum load < 20% of minimum load ΔP	Equal-percentage
Increasing ΔP with increasing load, ΔP at maximum load < 200% of minimum load ΔP	Linear
Increasing ΔP with increasing load, ΔP at maximum load > 200% of minimum load ΔP	Quick Opening

Pressure Control Systems

Application	Best Inherent Characteristic
Liquid Process	Equal-Percentage
Gas Process, Large Volume (Process has a receiver, Distribution System or Transmission Line Exceeding 100 ft. of Nominal Pipe Volume), Decreasing ΔP with Increasing Load, ΔP at Maximum Load > 20% of Minimum Load ΔP	Linear
Gas Process, Large Volume, Decreasing ΔP with Increasing Load, ΔP at Maximum Load < 20% of Minimum Load ΔP	Equal-Percentage
Gas Process, Small Volume, Less than 10 ft. of Pipe between Control Valve and Load Valve	Equal-Percentage

Flow Control Processes

Application		Best Inherent Characteristic	
Flow Measurement Signal to Controller	Location of Control Valve in Relation to Measuring Element	Wide Range of Flow Set Point	Small Range of Flow but Large ΔP Change at Valve with Increasing Load
Proportional to Flow	In Series	Linear	Equal-Percentage
	In Bypass*	Linear	Equal-Percentage
Proportional to Flow Squared	In Series	Linear	Equal-Percentage
	In Bypass*	Equal-Percentage	Equal-Percentage

*When control valve closes, flow rate increases in measuring element.

Table 1-4. *Control Valve Leakage Standards*

<table>
<tr><th>ANSI
B16.104-1976</th><th colspan="3">Maximum Leakage</th><th>Test Medium</th><th colspan="2">Pressure and Temperature</th></tr>
<tr><td>Class II</td><td colspan="3">0.5% valve capacity at full travel</td><td>Air</td><td colspan="2">Service ΔP or 50 psid (3.4 bar differential), whichever is lower, at 50° or 125°F (10° to 52°C)</td></tr>
<tr><td>Class III</td><td colspan="3">0.1% valve capacity at full travel</td><td>Air</td><td colspan="2">Service ΔP or 50 psid (3.4 bar differential), whichever is lower, at 50° or 125°F (10° to 52°C)</td></tr>
<tr><td>Class IV</td><td colspan="3">0.01% valve capacity at full travel</td><td>Air</td><td colspan="2">Service ΔP or 50 psid (3.4 bar differential), whichever is lower, at 50° or 125°F (10° to 52°C)</td></tr>
<tr><td>Class V</td><td colspan="3">5 x 10^{-4} mL/min/psid/inch port dia. (5 x 10^{-12} m^3/sec/Δbar/mm port dia)</td><td>Water</td><td colspan="2">Service ΔP at 50° or 125°F (10° to 52°C)</td></tr>
<tr><td>Class VI</td><td colspan="2">Nominal Port Diameter</td><td>Bubbles per Minute</td><td>mL per Minute</td><td>Test Medium</td><td>Pressure & Temperature</td></tr>
<tr><td rowspan="9"></td><td>In</td><td>mm</td><td></td><td></td><td rowspan="9">Air</td><td rowspan="9">Service ΔP or 50 psid (3.4 bar differential), whichever is lower, at 50° or 125°F (10° to 52°C)</td></tr>
<tr><td>1</td><td>25</td><td>1</td><td>0.15</td></tr>
<tr><td>1-1/2</td><td>38</td><td>2</td><td>0.30</td></tr>
<tr><td>2</td><td>51</td><td>3</td><td>0.45</td></tr>
<tr><td>2-1/2</td><td>64</td><td>4</td><td>0.60</td></tr>
<tr><td>3</td><td>76</td><td>6</td><td>0.90</td></tr>
<tr><td>4</td><td>102</td><td>11</td><td>1.70</td></tr>
<tr><td>6</td><td>152</td><td>27</td><td>4.00</td></tr>
<tr><td>8</td><td>203</td><td>45</td><td>6.75</td></tr>
</table>

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Chapter 2

Actuator Selection

The actuator is the distinguishing element that differentiates control valves from other types of valves. The first actuated valves were designed in the late 19th Century. However, today they would be better described as regulators since they operated directly from the process fluid. These “automatic valves” were the mainstay of industry through the early 1930s.

It was at this time also that the first pneumatic controllers were used. Development of valve controllers and the adaptation of standardized control signals stimulated design of the first, true, control valve actuators.

The control valve industry has evolved to fill a variety of needs and desires. Actuators are available with an array of designs, power sources and capabilities. Proper selection involves process knowledge, valve knowledge and actuator knowledge.

A control valve can perform its function only as well as the actuator can handle the static and dynamic loads placed on it by the valve. Therefore, proper selection and sizing are very important. Since the actuator can represent a significant portion of the total control valve price, careful selection of actuator and accessory options can lead to significant dollar savings.

The range of actuator types and sizes on the market today is so great that it seems the selection process might be highly complex. It is not! With a few rules in mind and knowledge of fundamental needs, the selection process can be very simple.

The following parameters are key as they quickly narrow the actuator choices:

- Power Source Availability
- Fail-safe Requirements
- Torque or Thrust Requirements
- Control Functions

Power Source Availability

The power source available at the location of a valve often can point directly to what type of actuator to choose. Typically, valve actuators are powered either by compressed air or by electricity. However, in some cases water pressure, hydraulic fluid, or even pipeline pressure can be used.

Since most plants have both electricity and compressed air readily available, the selection depends on the ease and cost of furnishing either power source to the actuator location. Reliability and maintenance requirements of the power system must also be considered. Consideration should also be given to providing backup operating power to critical plant loops.

Fail-safe Requirements

The overall reliability of power sources is quite high. However, many loops demand specific valve action should the power source ever fail. Desired action upon a signal failure may be required for safety reasons or for protection of equipment.

Fail-safe systems store energy, either mechanically in springs, pneumatically in volume tanks, or in hydraulic accumulators. When power fails, the fail-safe systems are triggered to drive the valves to the required position and to then maintain this position until returned to normal operation. In many cases the process pressure is used to ensure or enhance this action.

Actuator designs are available which allow a choice of failure mode between failing open, failing closed, or holding in the last position. Many actuator systems incorporate failure modes at no extra cost. For example, spring-and-diaphragm types are inherently fail open or closed, while electric operators typically hold their last position.

Torque or Thrust Requirements

An actuator must have sufficient thrust or torque for the prescribed application. In some cases this requirement can dictate actuator type as well as power supply requirements.

For instance, large valves requiring a high thrust may be limited to only electric or electro-hydraulic actuators due to a lack of pneumatic actuators with sufficient thrust capability. Conversely, electro-hydraulic actuators would be a poor choice for valves with very low thrust requirements.

The matching of actuator capability with valve body requirements is best left to the control valve manufacturer as there are considerable differences in frictional and fluid forces from valve to valve.

Control Functions

Knowledge of the required actuator functions will most clearly define the options available for selection. These functions include the actuator signal (pneumatic, electric, etc.), signal range, ambient temperatures, vibration levels, operating speed, frequency, and quality of control that is required.

Generally, signal types are grouped as being either:

- Two-position (on-off) or
- Analog (throttling).
- Digital

Two-position electric, electro-pneumatic, or pneumatic switches control on-off actuators. This is the simplest type of automatic control and the least restrictive in terms of selection.

Throttling actuators have considerably higher demands put on them from both a compatibility and performance standpoint. A throttling actuator receives its input from an electronic or pneumatic instrument that measures the controlled process variable. The actuator must then move the final control element in response to the instrument signal in an accurate and timely fashion to ensure effective control. The two primary additional requirements for throttling actuators are:

- Compatibility with instrument signal
- Better static and dynamic performance to ensure loop stability.

Compatibility with instrument signals is inherent in many actuator types, or it can be obtained with add-on equipment. But, the high-performance characteristics required of a good throttling actuator cannot be bolted on; instead, low hysteresis and minimal deadband must be designed into actuators.

Stroking speed, vibration, and temperature resistance must also be considered if critical to the application. For example, on liquid loops fast-stroking speeds can be detrimental due to the possibility of water hammer.

Vibration or mounting position can be a potential problem. The actuator weight, combined with the weight of the valve, may necessitate bracing.

It is essential to determine the ambient temperature and humidity that the actuator will experience. Many actuators contain either elastomeric or electronic components that can be subject to degradation by high humidity or temperature.

Economics

Evaluation of economics in actuator selection is a combination of the following:

- Cost
- Maintenance
- Reliability

A simple actuator, such as a spring-and-diaphragm, has few moving parts and is easy to service. Initial cost is low. Maintenance personnel understand and are comfortable working with them.

An actuator made specifically for a control valve eliminates the chance for a costly performance mismatch. An actuator manufactured by the valve vendor and shipped with the valve will eliminate separate mounting charges and ensure easier coordination of spare parts procurement. Interchangeable parts among varied actuators are also important to minimize spare-parts inventory.

Actuator Designs

There are many types of actuators on the market, most of which fall into four general categories:

- Spring-and-diaphragm
- Pneumatic piston
- Electric motor
- Electro-hydraulic.

Each actuator design has weaknesses, strong points and optimum uses. Most actuator designs are available for either sliding stem or rotary valve bodies. They differ only by linkages or motion translators; the basic power sources are identical.

Most rotary actuators employ linkages, gears, or crank arms to convert direct linear motion of a diaphragm or piston into the 90-degree output rotation required by rotary valves. The most important consideration for control valve actuators is the requirement for a design that limits the amount of lost motion between internal linkage and valve coupling.

Rotary actuators are now available that employ tilting pistons or diaphragms. These designs eliminate most linkage points (and resultant lost motion) and provide a safe, accurate and enclosed package.

When considering an actuator design, it is also necessary to consider the method by which it is coupled to the drive shaft of the control valve. Slotted connectors mated to milled shaft flats generally are not satisfactory if any degree of performance is required. Pinned connections, if solidly constructed, are suitable for nominal torque applications. A splined connector that mates to a splined shaft end and then is rigidly clamped to the shaft eliminates lost motion, is easy to disassemble and is capable of high torque.

Sliding stem actuators are rigidly fixed to valve stems by threaded and clamped connections. Since they don't have any linkage points, and their connections are rigid, they exhibit no lost motion and have excellent inherent control characteristics.

Spring-and-Diaphragm Actuators

The most popular and widely used control valve actuator is the pneumatic spring-and-diaphragm style. These actuators are extremely simple and offer low cost and high reliability. They normally operate over the standard signal ranges of 3 to 15 psi or 6 to 30 psi, and therefore, they are often suitable for throttling service using instrument signals directly.

Many spring-and-diaphragm designs offer either adjustable springs and/or wide spring selections to allow the actuator to be tailored to the particular application. Since they have few moving parts that may contribute to failure, they are extremely reliable. Should they ever fail, maintenance is extremely simple. Improved designs now include mechanisms to control the release of spring compression, eliminating possible personnel injury during actuator disassembly.

Use of a positioner or booster with a spring-and-diaphragm actuator can improve control, but when improperly applied, can result in very poor control. Follow the simple guidelines available for positioner applications, and look for:

- Rugged, vibration-resistant construction
- Calibration ease
- Simple, positive feedback linkages

The overwhelming advantage of the spring-and-diaphragm actuator is the inherent provision for fail-safe action. As air is loaded on the actuator casing, the diaphragm moves the valve and compresses the spring. The stored energy in the spring acts to move the valve back to its original position as air is released from the casing. Should there be a loss of signal pressure to the instrument or the actuator, the spring can move the valve to its initial (fail-safe) position.

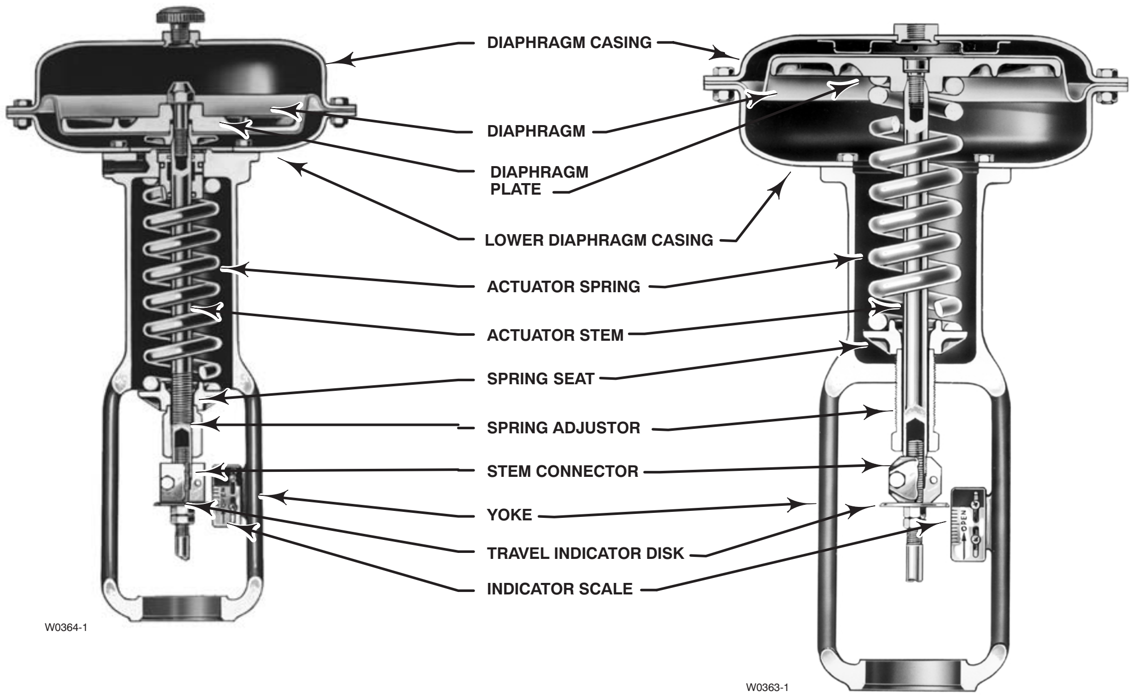


Figure 2-1. *Spring-and-diaphragm actuators offer an excellent first choice for most control valves. They are inexpensive, simple and have built-in, fail-safe action. Pictured above are cutaways of the popular Type 667 (left) and Type 657 (right) actuators.*

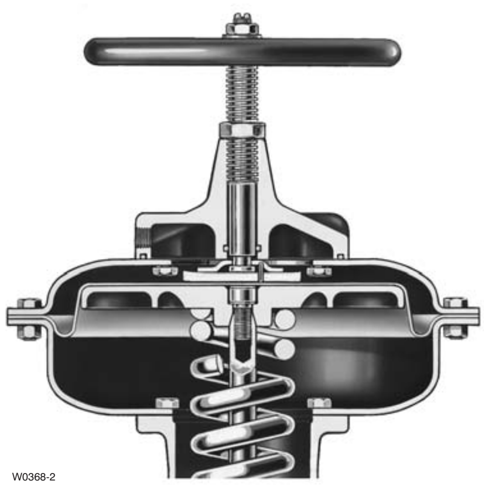


Figure 2-2. *Spring-and-diaphragm actuators can be supplied with a top-mounted handwheel. The handwheel allows manual operation and also acts as a travel stop or means of emergency operation.*

Actuators are available for either fail-open or fail-closed action. Fail-lock is also available by utilizing a pneumatic switching valve such as a Fisher Type 164A, piped as a lock valve.

The only real drawback to the spring-and-diaphragm actuator is a relatively limited output capability. Much of the thrust created by the diaphragm is taken up by the spring and thus does not result in output to the valve. Therefore, the spring-and-diaphragm actuator is used infrequently for high force requirements. It is not economical to build and use very large spring-and-diaphragm actuators because the size, weight and cost grow exponentially with each increase in output force capability.

Piston Actuators

Piston actuators are generally more compact and provide higher torque or force outputs than spring-and-diaphragm actuators. Fisher piston styles normally work with supply pressures between 50 and 150 psi and can be equipped with spring returns (however, this construction has limited application).

Piston actuators used for throttling service must be furnished with double-acting positioners, which simultaneously load and unload opposite sides of the piston. The pressure differential created across the piston causes travel toward the lower pressure side. The positioner senses the motion, and when the required position is reached, the positioner equalizes the pressure on both sides of the piston.

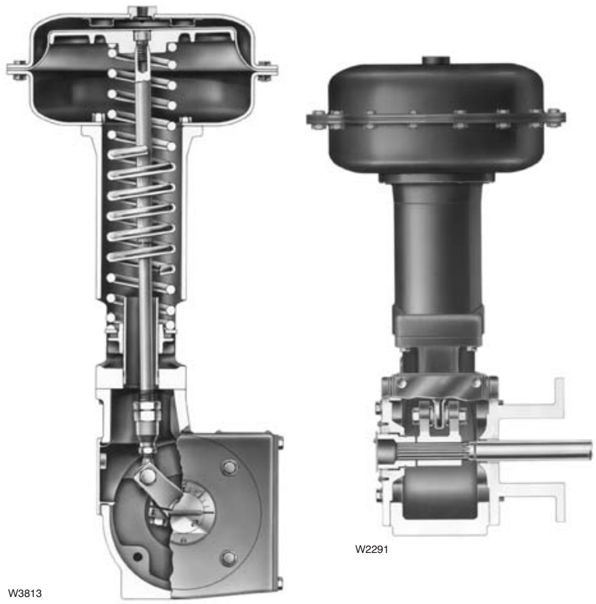


__Figure 2-3__. The Type 1052 is a spring-and-diaphragm actuator that has many features to provide precise control. The splined actuator connection features a clamped lever and single-joint linkage to help eliminate lost motion.

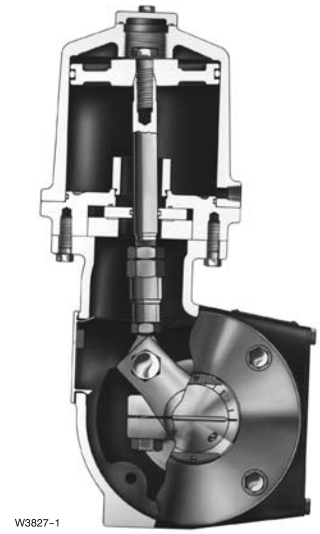


***Figure 2-4.** Double-acting piston actuators such as Type 1061 rotary actuator are a good choice when thrust requirements exceed the capability of spring-and-diaphragm actuators. Piston actuators require a higher supply pressure, but have benefits such as high stiffness and small size.*

The pneumatic piston actuator is an excellent choice when a compact unit is required to produce high torque or force. It is also easily adapted to services where high ambient temperatures are a concern.

The main disadvantages of piston actuators are high supply pressures required for positioners when used in throttling service and the lack of fail-safe systems.

There are two types of spring-return piston actuators available. The variations are subtle, but significant. It is possible to add a spring to a piston actuator and operate it much like a spring-and-diaphragm. These designs use a single-acting positioner that loads the piston chamber to move the actuator and compress the spring. As air is unloaded, the spring forces the piston back. These designs use large, high output springs that are capable of overcoming the fluid forces in the valve.

The alternative design uses a much smaller spring and relies on valve fluid forces to help provide the fail-safe action. In normal operation they act like a double action piston. In a fail-safe situation the spring initiates movement and is helped by

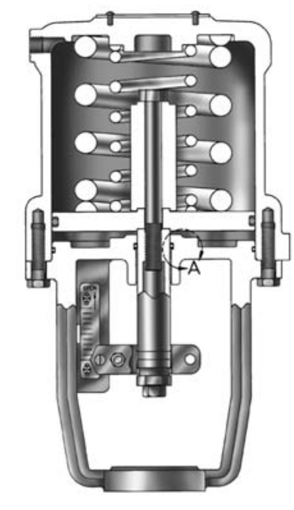


__Figure 2-5__. Spring fail-safe is present in this piston design. The Type 585C is an example of a spring-bias piston actuator. Process pressure can aid fail-safe action, or the actuator can be configured for full spring-fail closure.

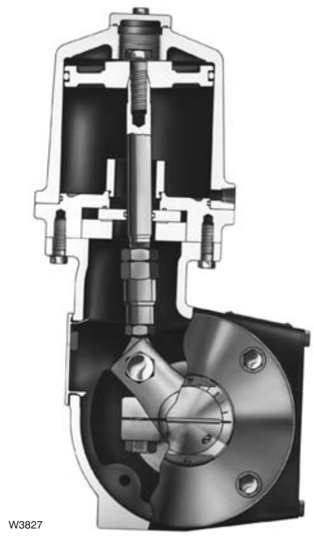


Figure 2-6. This Type 1061 actuator is a double-acting rotary piston actuator for throttling service.

unbalance forces on the valve plug. These actuators also can be sized and set up to provide full spring closure action without process assistance.

An alternative to springs is a pneumatic trip system, which often proves to be complex in design, difficult to maintain and costly. While a trip system is completely safe, any fail-safe requirement consideration should be given first to spring-and-diaphragm operators if they are feasible.

Special care should be given during the selection of throttling piston actuators to specify a design that has minimal hysteresis and deadband. As the number of linkage points in the actuator increases, so does the deadband. As the number of sliding parts increases, so does the hysteresis. An actuator with high hysteresis and deadband can be quite suitable for on-off service; however, caution is necessary when attempting to adapt this actuator to throttling service by merely bolting on a positioner.

The cost of a spring-and-diaphragm actuator is generally less than a comparable piston actuator. Part of this cost saving is a result of the ability to use instrument output air directly, thereby eliminating the need for a positioner. The inherent provision for fail-safe action in the spring-and-diaphragm actuator is also a consideration.

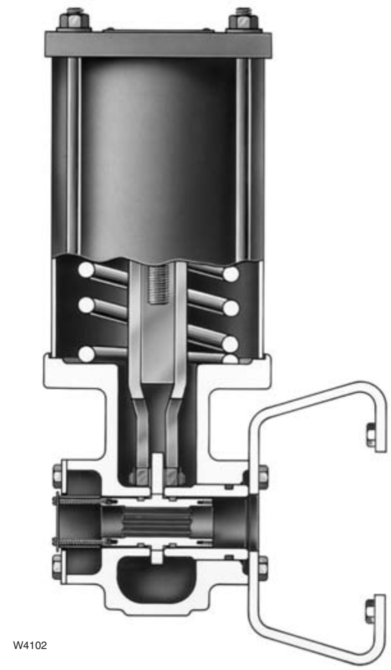


Figure 2-7. Since the requirements for accuracy and minimal lost motion are unnecessary for on-off service, cost savings can be achieved by simplifying the actuator design. The Type 1066SR incorporates spring-return capability.

Electric Actuators

Electric actuators can be applied successfully in many situations. Most electric operators consist of motors and gear trains and are available in a wide range of torque outputs, travels and capabilities. They are suited for remote mounting where no other power source is available, for use where there are specialized thrust or stiffness requirements or when highly precise control is required.

Electric operators are economical versus pneumatic actuators for applications in small size ranges only. Larger units operate slowly and weigh considerably more than pneumatic equivalents. Available fail action is typically lock in last position.

One very important consideration in choosing an electric actuator is its capability for continuous closed-loop control. In applications where frequent changes are made in control-valve position, the electric actuator must have a suitable duty cycle.

W2286

Figure 2-8. *The Type 350 is a "self-contained" electro-hydraulic actuator. The single unit incorporates a hydraulic pump and reservoir.*

High performance electric actuators using continuous rated DC motors and ball screw output devices are capable of precise control and 100% duty cycles.

Compared to other actuator designs, the electric actuator generally provides the highest output available within a given package size. Additionally, electric actuators are very stiff, that is, resistant to valve forces. This makes them an excellent choice for good throttling control of large, high-pressure valves.

Electro-hydraulic Actuators

An electro–hydraulic actuator internally pumps oil at high pressure to a piston, which in turn creates an output force. The electro-hydraulic actuator is an excellent choice for throttling due to its high stiffness, compatibility with analog signals, excellent frequency response and positioning accuracy.

Most electro-hydraulic actuators are capable of very high outputs, but they are handicapped by high initial cost, complexity and difficult maintenance. Fail-safe action on electro-hydraulic actuators can be accomplished by use of a return spring or a hydraulic accumulator and by shutdown systems.

Like electric actuators, the electro-hydraulic actuators are suitable for remote mounting where no other power source is available (such as pipelines).

Electro-hydraulic actuators from Fisher are available in two basic configurations: self-contained, and externally powered. The self-contained unit includes its own motor, pump, fluid reservoir, etc. (this type of unit can be available with spring-return fail mode.) The external power supply unit requires a separate motor, pump, reservoir, hydraulic hoses and other accessories. This type requires an accumulator to achieve a fail mode.

Actuator Sizing

The last step in the selection process is to determine the required actuator size. Fundamentally, the process of sizing is to match as closely as possible the actuator capabilities to the valve requirements.

In practice, the mating of actuator and valve requires the consideration of many factors. Valve forces must be evaluated at the critical positions of valve travel (usually open and closed) and compared to actuator output. Valve force calculation varies considerably between valve styles and manufacturers. In most cases it is necessary to consider a complex summation of forces including:

- Static fluid forces
- Dynamic fluid forces and force gradients
- Friction of seals, bearings and packing
- Seat loading

Although actuator sizing is not difficult, the great variety of designs on the market and the ready availability of vendor expertise (normally at no cost) make detailed knowledge of the procedures unnecessary.

Actuator Spring for Globe Valves

The force required to operate a globe valve includes:

- Force to overcome static unbalance of the valve plug
- Force to provide a seat load
- Force to overcome packing friction
- Additional forces required for certain specific applications or constructions

Total force required = A + B + C + D

A. Unbalance Force

The unbalance force is that resulting from fluid pressure at shutoff and in the most general sense can be expressed as:

Unbalance force = net pressure differential X net unbalance area

Frequent practice is to take the maximum upstream gauge pressure as the net pressure differential unless the process design always ensures a back pressure at the maximum inlet pressure. Net unbalance area is the port area on a single seated flow up design. Unbalance area may have to take into account the stem area depending on configuration. For balanced valves there is still a small unbalance area. This data can be obtained from the manufacturer. Typical port areas for balanced valves flow up and unbalanced valves in a flow down configuration are listed in Table 2-1.

B. Force to Provide Seat Load

Seat load, usually expressed in pounds per lineal inch or port circumference, is determined by shutoff requirements. Use the guidelines in Table 2-2 to determine the seat load required to meet the factory acceptance tests for ANSI/FCI 70–2 and IEC 534–4 leak classes II through VI.

Because of differences in the severity of service conditions, do not construe these leak classifications and corresponding leakage rates as indicators of field performance. To prolong seat life and shutoff capabilities, use a higher than recommended seat load. If tight shutoff is not a prime consideration, use a lower leak class.

C. Packing Friction

Packing friction is determined by stem size, packing type, and the amount of compressive load placed on the packing by the process or the bolting. Packing friction is not 100% repeatable in its friction characteristics. Newer live loaded packing designs can have significant friction forces especially if graphite packing is used. Table 2-3 lists typical packing friction values.

D. Additional Forces

Additional forces to consider may include bellows stiffness, unusual frictional forces resulting from seals or special seating forces for soft metal seals.

Table 2-1. *Typical Unbalance Areas of Control Valves*

Port Diameter, Inches	Unbalance Area Single-Seated Unbalanced Valves, In^2	Unbalance Area Balanced Valves, In^2
1/4	0.049	– – –
3/8	0.110	– – –
1/2	0.196	– – –
3/4	0.441	– – –
1	0.785	– – –
1 5/16	1.35	0.04
1 7/8	2.76	0.062
2 5/16	4.20	0.27
3 7/16	9.28	0.118
4 3/8	15.03	0.154
7	38.48	0.81
8	50.24	0.86

Table 2-2. *Recommended Seat Load Per Leak Class for Control Valves*

Class I	As required by customer specification, no factory leak test required
Class II	20 pounds per lineal inch of port circumference
Class III	40 pounds per lineal inch of port circumference
Class IV	Standard (Lower) Seat only—40 pounds per lineal inch of port circumference (up through a 4–3/8 inch diameter port) Standard (Lower) Seat only—80 pounds per lineal inch of port circumference (larger than 4–3/8 inch diameter port)
Class V	Metal Seat—determine pounds per lineal inch of port circumference from Figure 2-9

The manufacturer should either supply this information or take it into account when sizing an actuator.

Actuator Force Calculations

Pneumatic spring-and-diaphragm actuators provide a net force with the additional air pressure after compressing the spring in air-to-close, or with the net pre-compression of the spring in air-to-open. This may be calculated in pounds per square inch of pressure differential.

For example: Suppose 275 lbf. is required to close the valve as calculated per the process described earlier. An air–to–open actuator with 100 square inches of diaphragm area and a bench set of 6 to 15 psig is one available option. The expected operating range is 3 to 15 psig. The pre-compression can be calculated as the

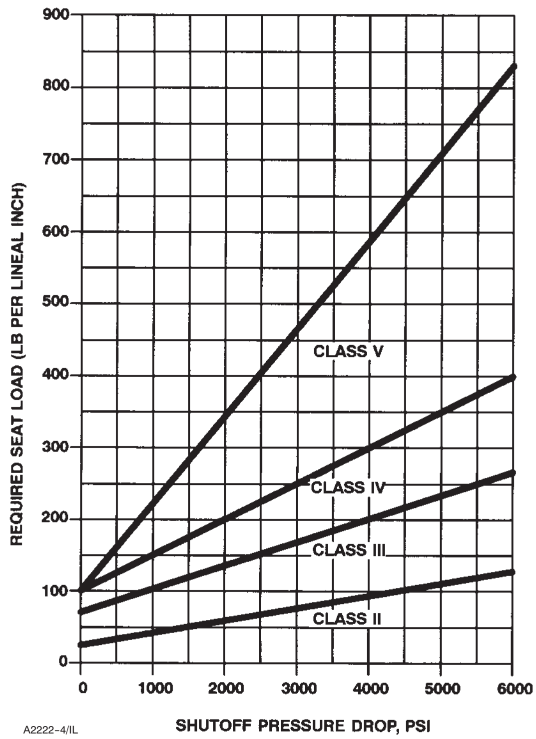


Figure 2-9. *Recommended seat load.*

difference between the lower end of the bench set (6 psig) and the beginning of the operating range (3 psig). This 3 psig is used to overcome the pre-compression so the net pre-compression force must be:

3 psig X 100 sq. in. = 300 lbf.

This exceeds the force required and is an adequate selection.

Piston actuators with springs are sized in the same manner.

The thrust from piston actuators without springs can be calculated as:

Piston Area X Minimum Supply Pressure = Minimum Available Thrust

(Be careful to maintain compatibility of units)

Table 2-3. *Typical Packing Friction Values (Lb)*

Stem Size (Inches)	ANSI Class	PTFE Packing		Graphite Ribbon/ Filament
		Single	Double	
5/16	All	20	30	– – –
3/8	125 150 250 300	38	56	– – – 125 – – – 190
	600 900 1500			250 320 380
1/2	125 150 250 300	50	75	– – – 180 – – – 230
	600 900 1500 2500			320 410 500 590
5/8	125 150 250 300 600	63	95	– – – 218 – – – 290 400
3/4	125 150 250 300	75	112.5	– – – 350 – – – 440
	600 900 1500 2500			660 880 1100 1320
1	300 600 900 1500 2500	100	150	610 850 1060 1300 1540
1–1/4	300 600 900 1500 2500	120	180	800 1100 1400 1700 2040
2	300 600 900 1500 2500	200	300	1225 1725 2250 2750 3245
Values shown are frictional forces typically encountered when using standard packing flange bolt-torquing procedures.				

In some circumstances an actuator could supply too much force and cause the stem to buckle, to bend sufficiently to cause a leak, or to damage valve internals.

The manufacturer normally takes responsibility for actuator sizing and should have methods documented to check for maximum stem loads. Manufacturers also publish data on actuator thrusts, effective diaphragm areas, and spring data.

***Table 2-4**. Typical Rotary Shaft Valve Torque Factors V-Notch Ball Valve with Composition Seal*

Valve Size, Inches	Valve Shaft Diameter, Inches	A	B	C		Maximum T_D, Lbf•In.
		Composition Bearings		60 Degrees	70 Degrees	
2	1/2		8	0.11	0.60	515
3	3/4	0.15	280	0.15	3.80	2120
4	3/4	0.10	380	1.10	18.0	2120
6	1	0.10	500	1.10	36.0	4140
8	1–1/4		750	3.80	60.0	9820
10	1–1/4		1250	3.80	125	9820
12	1–1/2	1.80	3000	11.0	143	12,000
14	1–3/4	4.00	2400	75	413	23,525
16	2	42	2800	105	578	23,525
18	2–1/8	60	2800	105	578	55,762
20	2–1/2		5200	190	1044	55,762

***Tabel 2-5.** Typical High Performance Butterfly Torque Factors for Valve with Composition Seal*

Valve Size, Inches	Shaft Diameter Inches	A	B	C			Maximum Torque, Inch–Pounds	
				60°	75°	90°	Breakout T_B	Dynamic T_D
3	1/2	0.50	136	0.8	1.8	8	280	515
4	5/8	0.91	217	3.1	4.7	25	476	225
6	3/4	1.97	403	30	24	70	965	2120
8	1	4.2	665	65	47	165	1860	4140
10	1–1/4	7.3	1012	125	90	310	3095	9820
12	1–1/2	11.4	1422	216	140	580	4670	12,000

Actuator Sizing for Rotary Valves

In selecting the most economical actuator for a rotary valve, the determining factors are the torque required to open and close the valve and the torque output of the actuator.

This method assumes the valve has been properly sized for the application and the application does not exceed pressure limitations for the valve.

Torque Equations

Rotary valve torque equals the sum of a number of torque components. To avoid confusion, a number of these have been combined, and a number of calculations have been performed in advance. Thus, the torque required for each valve type can be represented with two simple and practical equations.

Breakout Torque

$$T_B=A(\Delta P_{shutoff})+B$$

Dynamic Torque

$$T_D=C(\Delta P_{eff})$$

Specific A, B, and C factors for example rotary valve designs are included in tables 2-4 and 2-5.

Maximum Rotation

Maximum rotation is defined as the angle of valve disk or ball in the fully open position.

Normally, maximum rotation is 90 degrees. The ball or disk rotates 90 degrees from the closed position to the wide-open position.

Some of the pneumatic spring–return piston and pneumatic spring-and-diaphragm actuators are limited to 60 or 75 degrees rotation.

For pneumatic spring-and-diaphragm actuators, limiting maximum rotation allows for higher initial spring compression, resulting in more actuator breakout torque. Additionally, the effective length of each actuator lever changes with valve rotation. Published torque values, particularly for pneumatic piston actuators, reflect this changing lever length.

The Selection Process

In choosing an actuator type, the fundamental requirement is to know your application. Control signal, operating mode, power source available, thrust/torque required, and fail-safe position can make many decisions for you. Keep in mind simplicity, maintainability and lifetime costs.

Safety is another consideration that must never be overlooked. Enclosed linkages and controlled compression springs available in some designs are very important for safety reasons. Table 2-6 below lists the pros and cons of the various actuator styles.

Table 2-6 *Actuator Feature Comparison*

Actuator Type	Advantages	Disadvantages
Spring-and-Diaphragm	Lowest cost Ability to throttle without positioner Simplicity Inherent fail-safe action Low supply pressure requirement Adjustable to varying conditions Ease of maintenance	Limited output capability Larger size and weight
Pneumatic Piston	High thrust capability Compact Lightweight Adaptable to high ambient temperatures Fast stroking speed Relatively high actuator stiffness	Higher cost Fail-safe requires accessories or addition of a spring Positioner required for throttling High supply pressure requirement
Electric Motor	Compactness Very high stiffness High output capability	High cost Lack of fail-safe action Limited duty cycle Slow stroking speed
Electro-Hydraulic	High output capability High actuator stiffness Excellent throttling ability Fast stroking speed	High cost Complexity and maintenance difficulty Large size and weight Fail-safe action only with accessories

Figure 2-10. The Type 3620JP is an electro-pneumatic positioner that combines the functions of transducer and positioner into one unit. The combustion unit generally is more economical but may not be as flexible as separate units.

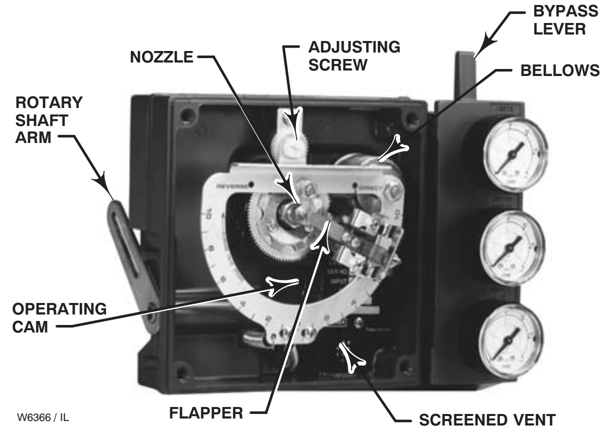


***Figure 2-11.** The standard pneumatic positioner for spring-and-diaphragm actuators is Type 3582. This time-proven design features ease for reversal and calibration as well as availability of characterizing cams to alter its input/output relationship.*

Actuator Selection Summary

- Actuator selection must be based on a balance of process requirements, valve requirements and cost.

- Simple designs such as the spring-and-diaphragm are simpler, less expensive and easier to maintain. Consider them first in most situations.

- Piston actuators offer many of the advantages of pneumatic actuators with higher thrust capability than spring-and-diaphragm styles. They are especially useful where compactness is desired or long travel is required.

- Electric and electro-hydraulic actuators provide excellent performance. They are, however, much more complex and difficult to maintain.

- Actuator sizing is not difficult, but the wide variety of actuators and valves make it difficult to master. Vendor expertise is widely available.

- Systems such as control valves are best purchased, assembled and tested by one source. Use of actuators and accessories of the same manufacture will eliminate many problems.

***Figure 2-12**. The FIELDVUE® Digital Valve Controller brings increased control accuracy and flexibility. When utilized with AMS ValveLink® software, FIELDVUE instruments provide valuable diagnostic data that helps avoid maintenance problems.*

W4908

Figure 2-13. *Electro-Pneumatic transducers are a common actuator accessory. They take a milliamp signal and produce a proportional pneumatic output. The Type 646 is compact, accurate and has low air consumption.*

W5940

Figure 2-14. *Limit switches are a common actuator accessory. This unit can accommodate up to six switches and has trip points adjustable to any point in travel.*

W4727

Figure 2-15. *On fast control loops, a positioner may not be able to react quickly enough to be of use. In these situations, performance of spring-and-diaphragm actuators can be improved by use of pneumatic boosters such as the Type 2625.*

Chapter 3

Liquid Valve Sizing

Valve Sizing

Standardization activities for control valve sizing can be traced back to the early 1960s when a trade association, the Fluids Control Institute, published sizing equations for use with both compressible and incompressible fluids. The range of service conditions that could be accommodated accurately by these equations was quite narrow, and the standard did not achieve a high degree of acceptance. In 1967, the ISA established a committee to develop and publish standard equations. The efforts of this committee culminated in a valve sizing procedure that has achieved the status of American National Standard. Later, a committee of the International Electrotechnical Commission (IEC) used the ISA works as a basis to formulate international standards for sizing control valves. (Some information in this introductory material has been extracted from ANSI/ISA S75.01 standard with the permission of the publisher, the ISA.) Except for some slight differences in nomenclature and procedures, the ISA and IEC standards have been harmonized. ANSI/ISA Standard S75.01 is harmonized with IEC Standards 534-2-1 and 534-2-2. (IEC Publications 534-2, Sections One and Two for incompressible and compressible fluids, respectively.)

In the following sections, the nomenclature and procedures are explained, and sample problems are solved to illustrate their use.

Sizing Valves for Liquids

Following is a step-by-step procedure for the sizing of control valves for liquid flow using the IEC procedure. Each of these steps is important and must be considered during any valve sizing procedure. Steps 3 and 4 concern the determination of certain sizing factors that may or may not be required in the sizing equation depending on the service conditions of the sizing problem. If one, two, or all three of these sizing factors are to be included in the equation for a particular sizing problem, refer to the appropriate factor determination section(s) located in the text after the sixth step.

1. Specify the variables required to size the valve as follows:

- Desired design
- Process fluid (water, oil, etc.), and
- Appropriate service conditions q or w, P_1, P_2 or ΔP, T_1, G_f, P_v, P_c, and υ

The ability to recognize which terms are appropriate for a specific sizing procedure can only be acquired through experience with different valve sizing problems. If any of the above terms appears to be new or unfamiliar, refer to the Abbreviations and Terminology Table 3-1 for a complete definition.

2. Determine the equation constant, N.

N is a numerical constant contained in each of the flow equations to provide a means for using different systems of units. Values for these various constants and their applicable units are given in the Equation Constants table 3-2.

Use N_1, if sizing the valve for a flow rate in volumetric units (gpm or m^3/h).

Use N_6 if sizing the valve for a flow rate in mass units (lb/h or kg/h).

3. Determine F_p, the piping geometry factor.

F_p is a correction factor that accounts for pressure losses due to piping fittings such as reducers, elbows, or tees that might be attached directly to the inlet and outlet connections of the control valve to be sized. If such fittings are attached to the valve, the F_p factor must be considered in the sizing procedure. If, however, no fittings are attached to the valve, F_p has a value of 1.0 and simply drops out of the sizing equation.

Table 3-1. *Abbreviations and Terminology*

Symbol		Symbol	
C_v	Valve sizing coefficient	P_1	Upstream absolute static pressure
d	Nominal valve size	P_2	Downstream absolute static pressure
D	Internal diameter of the piping	P_c	Absolute thermodynamic critical pressure
F_d	Valve style modifier, dimensionless	P_v	Vapor pressure absolute of liquid at inlet temperature
F_F	Liquid critical pressure ratio factor, dimensionless	ΔP	Pressure drop (P_1-P_2) across the valve
F_k	Ratio of specific heats factor, dimensionless	$\Delta P_{max(L)}$	Maximum allowable liquid sizing pressure drop
F_L	Rated liquid pressure recovery factor, dimensionless	$\Delta P_{max(LP)}$	Maximum allowable sizing pressure drop with attached fittings
F_{LP}	Combined liquid pressure recovery factor and piping geometry factor of valve with attached fittings (when there are no attached fittings, F_{LP} equals F_L), dimensionless	q	Volume rate of flow
F_P	Piping geometry factor, dimensionless	q_{max}	Maximum flow rate (choked flow conditions) at given upstream conditions
G_f	Liquid specific gravity (ratio of density of liquid at flowing temperature to density of water at 60°F), dimensionless	T_1	Absolute upstream temperature (degree K or degree R)
G_g	Gas specific gravity (ratio of density of flowing gas to density of air with both at standard conditions[1], i.e., ratio of molecular weight of gas to molecular weight of air), dimensionless	w	Mass rate of flow
k	Ratio of specific heats, dimensionless	x	Ratio of pressure drop to upstream absolute static pressure ($\Delta P/P_1$), dimensionless
K	Head loss coefficient of a device, dimensionless	x_T	Rated pressure drop ratio factor, dimensionless
M	Molecular weight, dimensionless	Y	Expansion factor (ratio of flow coefficient for a gas to that for a liquid at the same Reynolds number), dimensionless
N	Numerical constant	Z	Compressibility factor, dimensionless
		γ_1	Specific weight at inlet conditions
		υ	Kinematic viscosity, centistokes

1. Standard conditions are defined as 60°F (15.5°C) and 14.7 psia (101.3kPa).

Table 3-2. Equation Constants[1]

		N	w	q	p[2]	γ	T	d, D
N_1		0.0865	- - -	m^3/h	kPa	- - -	- - -	- - -
		0.865	- - -	m^3/h	bar	- - -	- - -	- - -
		1.00	- - -	gpm	psia	- - -	- - -	- - -
N_2		0.00214	- - -	- - -	- - -	- - -	- - -	mm
		890	- - -	- - -	- - -	- - -	- - -	inch
N_5		0.00241	- - -	- - -	- - -	- - -	- - -	mm
		1000	- - -	- - -	- - -	- - -	- - -	inch
N_6		2.73	kg/h	- - -	kPa	kg/m^3	- - -	- - -
		27.3	kg/h	- - -	bar	kg/m^3	- - -	- - -
		63.3	lb/h	- - -	psia	lb/ft^3	- - -	- - -
N_7[3]	Normal Conditions $T_N = 0°C$	3.94	- - -	m^3/h	kPa	- - -	deg K	- - -
		394	- - -	m^3/h	bar	- - -	deg K	- - -
	Standard Conditions $T_s = 15.5°C$	4.17	- - -	m^3/h	kPa	- - -	deg K	- - -
		417	- - -	m^3/h	bar	- - -	deg K	- - -
	Standard Conditions $T_s = 60°F$	1360	- - -	scfh	psia	- - -	deg R	- - -
N_8		0.948	kg/h	- - -	kPa	- - -	deg K	- - -
		94.8	kg/h	- - -	bar	- - -	deg K	- - -
		19.3	lb/h	- - -	psia	- - -	deg R	- - -
N_9[3]	Normal Conditions $T_N = 0°C$	21.2	- - -	m^3/h	kPa	- - -	deg K	- - -
		2120	- - -	m^3/h	bar	- - -	deg K	- - -
	Standard Conditions Ts = 15.5°C	22.4	- - -	m^3/h	kPa	- - -	deg K	- - -
		2240	- - -	m^3/h	bar	- - -	deg K	- - -
	Standard Conditions $T_S = 60°F$	7320	- - -	scfh	psia	- - -	deg R	- - -

1. Many of the equations used in these sizing procedures contain a numerical constant, N, along with a numerical subscript. These numerical constants provide a means for using different units in the equations. Values for the various constants and the applicable units are given in the above table. For example, if the flow rate is given in U.S. gpm and the pressures are psia, N_1 has a value of 1.00. If the flow rate is m^3/hr and the pressures are kPa, the N_1 constant becomes 0.0865.
2. All pressures are absolute.
3. Pressure base is 101.3 kPa (1.013 bar)(14.7 psia).

For rotary valves with reducers (swaged installations), and other valve designs and fitting styles, determine the F_p factors by using the procedure for Determining F_p, the Piping Geometry Factor, page 3-4.

4. Determine q_{max} (the maximum flow rate at given upstream conditions) or ΔP_{max} (the allowable sizing pressure drop).

The maximum or limiting flow rate (q_{max}), commonly called choked flow, is manifested by no additional increase in flow rate with increasing pressure differential with fixed upstream conditions. In liquids, choking occurs as a result of vaporization of the liquid when the static pressure within the valve drops below the vapor pressure of the liquid.

The IEC standard requires the calculation of an allowable sizing pressure drop (ΔP_{max}), to account for the possibility of choked flow conditions within the valve. The calculated ΔP_{max} value is compared with the actual pressure drop specified in the service conditions, and the lesser of these two values is used in the sizing equation. If it is desired to use ΔP_{max} to account for the possibility of choked flow conditions, it can be calculated using the procedure for determining q_{max}, the Maximum Flow Rate, or ΔP_{max}, the Allowable Sizing Pressure Drop. If it can be recognized that choked flow conditions will not develop within the valve, ΔP_{max} need not be calculated.

5. Solve for required C_v, using the appropriate equation:

- For volumetric flow rate units—

$$C_v = \frac{q}{N_1 F_p \sqrt{\frac{P_1 - P_2}{G_f}}}$$

- For mass flow rate units—

$$C_v = \frac{w}{N_6 F_p \sqrt{(P_1 - P_2)\gamma}}$$

In addition to C_v, two other flow coefficients, K_v and A_v, are used, particularly outside of North America. The following relationships exist:

$$K_v = (0.865)(C_v)$$

$$A_v = (2.40 \times 10^{-5})(C_v)$$

6. Select the valve size using the appropriate flow coefficient table and the calculated C_v value.

Determining Piping Geometry Factor (F_p)

Determine an F_p factor if any fittings such as reducers, elbows, or tees will be directly attached to the inlet and outlet connections of the control valve that is to be sized. When possible, it is recommended that F_p factors be determined experimentally by using the specified valve in actual tests.

Calculate the F_p factor using the following equation.

$$Fp = \left[1 + \frac{\Sigma K}{N_2}\left(\frac{C_v}{d^2}\right)^2\right]^{-1/2}$$

where,

N_2 = Numerical constant found in the Equation Constants table

d = Assumed nominal valve size

C_v = Valve sizing coefficient at 100-percent travel for the assumed valve size

In the above equation, the ΣK term is the algebraic sum of the velocity head loss coefficients of all of the fittings that are attached to the control valve.

$$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2}$$

where,

K_1 = Resistance coefficient of upstream fittings

K_2 = Resistance coefficient of downstream fittings

K_{B1} = Inlet Bernoulli coefficient

K_{B2} = Outlet Bernoulli coefficient

The Bernoulli coefficients, K_{B1} and K_{B2}, are used only when the diameter of the piping approaching the valve is different from the diameter of the piping leaving the valve, whereby:

$$K_{B1} \text{ or } K_{B2} = 1 - \left(\frac{d}{D}\right)^4$$

where,

d = Nominal valve size

D = Internal diameter of piping

If the inlet and outlet piping are of equal size, then the Bernoulli coefficients are also equal, $K_{B1} = K_{B2}$, and therefore they are dropped from the equation.

The most commonly used fitting in control valve installations is the short-length concentric reducer. The equations for this fitting are as follows:

- For an inlet reducer—

$$K_1 = 0.5\left(1 - \frac{d^2}{D^2}\right)^2$$

- For an outlet reducer—

$$K_2 = 1.0\left(1 - \frac{d^2}{D^2}\right)^2$$

- For a valve installed between identical reducers—

$$K_1 + K_2 = 1.5\left(1 - \frac{d^2}{D^2}\right)^2$$

Determining Maximum Flow Rate (q_{max})

Determine either q_{max} or ΔP_{max} if it is possible for choked flow to develop within the control valve that is to be sized. The values can be determined by using the following procedures.

$$q_{max} = N_1 F_L C_v \sqrt{\frac{P_1 - F_F\ P_v}{G_f}}$$

Values for F_F, the liquid critical pressure ratio factor, can be obtained from figure 3-1, or from the following equation:

$$F_F = 0.96 - 0.28\sqrt{\frac{P_v}{P_c}}$$

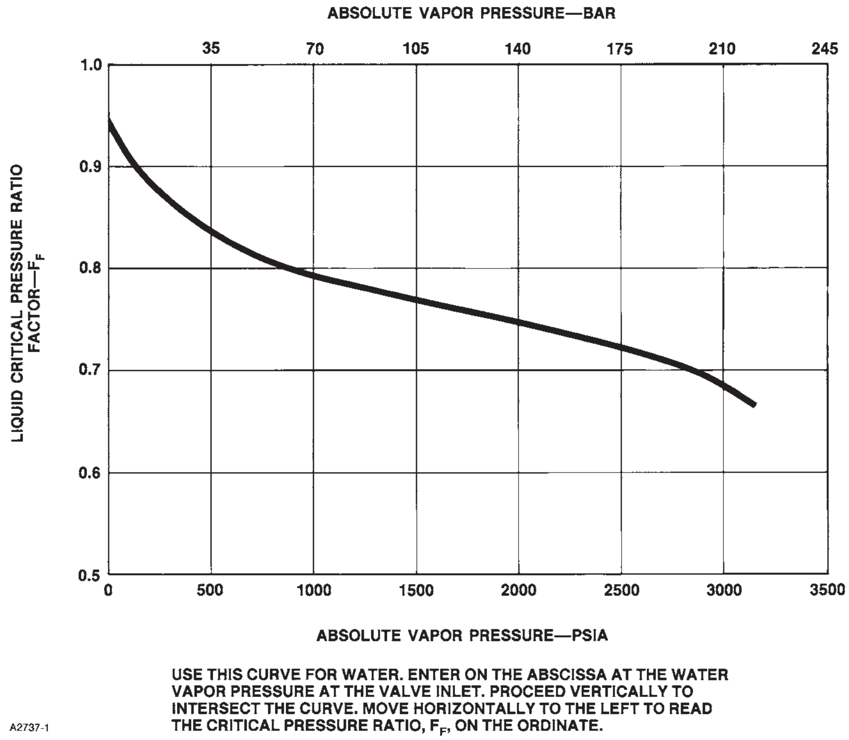


Figure 3-1. *Liquid critical pressure ratio factor for water.*

Values of F_L, the recovery factor for rotary valves installed without fittings attached, can be found in published coefficient tables. If the given valve is to be installed with fittings such as reducer attached to it, F_L in the equation must be replaced by the quotient F_{LP}/F_p, where:

$$F_{LP} = \left[\frac{K_1}{N_2}\left(\frac{C_v}{d^2}\right)^2 + \frac{1}{F_L{}^2}\right]^{-1/2}$$

and

$K_1 = K_1 + K_{B1}$

where,

K_1 = Resistance coefficient of upstream fittings

K_{B1} = Inlet Bernoulli coefficient

(See the procedure for Determining F_p, the Piping Geometry Factor, for definitions of the other constants and coefficients used in the above equations.)

Determining Allowable Sizing Pressure Drop (ΔP_{max})

ΔP_{max} (the allowable sizing pressure drop) can be determined from the following relationships:

For valves installed without fittings—

$$\Delta P_{max(L)} = F_L{}^2\left(P_1 - F_F\ P_v\right)$$

For valves installed with fittings attached—

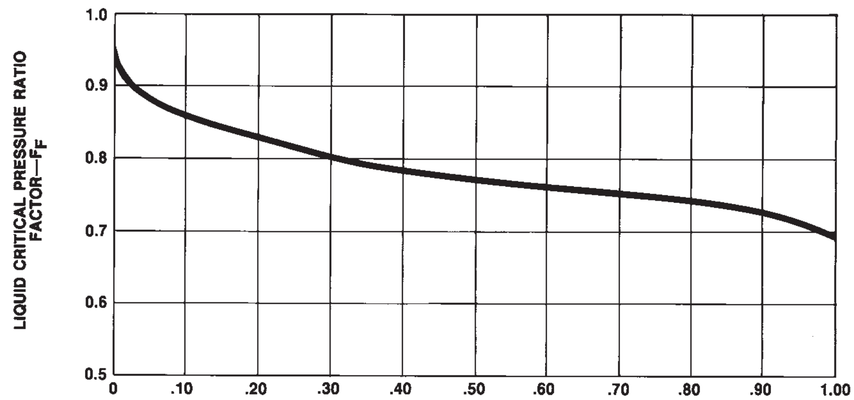
$$\Delta P_{max(LP)} = \left(\frac{F_{LP}}{F_P}\right)^2\left(P_1 - F_F\ P_V\right)$$

where,

P_1 = Upstream absolute static pressure

P_2= Downstream absolute static pressure

P_v = Absolute vapor pressure at inlet temperature



$$\frac{\text{ABSOLUTE VAPOR PRESSURE}}{\text{ABSOLUTE THERMODYNAMIC CRITICAL PRESSURE}} - \frac{P_v}{P_c}$$

USE THIS CURVE FOR LIQUIDS OTHER THAN WATER. DETERMINE THE VAPOR PRESSURE/CRITICAL PRESSURE RATIO BY DIVIDING THE LIQUID VAPOR PRESSURE AT THE VALVE INLET BY THE CRITICAL PRESSURE OF THE LIQUID. ENTER ON THE ABSCISSA AT THE RATIO JUST CALCULATED AND PROCEED VERTICALLY TO INTERSECT THE CURVE. MOVE HORIZONTALLY TO THE LEFT AND READ THE CRITICAL PRESSURE RATIO, F_F, ON THE ORDINATE.

Figure 3-2. *Liquid critical pressure ratio factor for liquids other than water.*

Values of F_F, the liquid critical pressure ratio factor, can be obtained from figure 3-1 or from the following equation:

$$F_F = 0.96 - 0.28\sqrt{\frac{P_v}{P_c}}$$

An explanation of how to calculate values of F_{LP}, the recovery factor for valves installed with fittings attached, is presented in the preceding procedure Determining q_{max} (the Maximum Flow Rate).

Once the ΔP_{max} value has been obtained from the appropriate equation, it should be compared with the actual service pressure differential ($\Delta P = P_1 - P_2$). If ΔP_{max} is less than ΔP, this is an indication that choked flow conditions will exist under the service conditions specified. If choked flow conditions do exist ($\Delta P_{max} < P_1 - P_2$), then step 5 of the procedure for Sizing Valves for Liquids must be modified by replacing the actual service pressure differential ($P_1 - P_2$) in the appropriate valve sizing equation with the calculated ΔP_{max} value.

Note

Once it is known that choked flow conditions will develop within the specified valve design (ΔP_{max} is calculated to be less than ΔP), a further distinction can be made to determine whether the choked flow is caused by cavitation or flashing. The choked flow conditions are caused by flashing if the outlet pressure of the given valve is less than the vapor pressure of the flowing liquid. The choked flow conditions are caused by cavitation if the outlet pressure of the valve is greater than the vapor pressure of the flowing liquid.

Liquid Sizing Sample Problem

Assume an installation that, at initial plant start-up, will not be operating at maximum design capability. The lines are sized for the ultimate system capacity, but there is a desire to install a control valve now which is sized only for currently anticipated requirements. The line size is 8-inches, and an ANSI Class 300 globe valve with an equal percentage cage has been specified. Standard concentric reducers will be used to install the valve into the line. Determine the appropriate valve size.

1. Specify the necessary variables required to size the valve:

- Desired valve design—ANSI Class 300 globe valve with equal percentage cage and an assumed valve size of 3-inches.

- Process fluid—liquid propane
- Service conditions—q = 800 gpm

P_1 = 300 psig = 314.7 psia

P_2 = 275 psig = 289.7 psia

ΔP = 25 psi

T_1 = 70°F

G_f = 0.50

P_v = 124.3 psia

P_c = 616.3 psia

2. Use an N_1 value of 1.0 from the Equation Constants table.

3. Determine F_p, the piping geometry factor.

Because it is proposed to install a 3-inch valve in an 8-inch line, it will be necessary to determine the piping geometry factor, F_p, which corrects for losses caused by fittings attached to the valve.

$$F_p = \left[1 + \frac{\Sigma K}{N_2}\left(\frac{C_v}{d^2}\right)^2\right]^{-1/2}$$

where,

N_2 = 890, from the Equation Constants table

d = 3 inches, from step 1

Cv = 121, from the flow coefficient table for an ANSI Class 300, 3-inch globe valve with equal percentage cage

To compute ΣK for a valve installed between identical concentric reducers:

$$\Sigma K = K_1 + K_2$$

$$= 1.5\left(1 - \frac{d^2}{D^2}\right)^2$$

$$= 1.5\left(1 - \frac{(3)^2}{(8)^2}\right)^2$$

$$= 1.11$$

where,

D = 8 inches, the internal diameter of the piping so,

$$F_p = \left[1 + \frac{1.11}{890}\left(\frac{121}{3^2}\right)^2\right]^{-1/2}$$

$$= 0.90$$

4. Determine ΔP_{max} (the Allowable Sizing Pressure Drop.)

Based on the small required pressure drop, the flow will not be choked ($\Delta P_{max} > \Delta P$).

5. Solve for C_v, using the appropriate equation.

$$C_v = \frac{q}{N_1 F_P \sqrt{\frac{P_1 - P_2}{G_f}}}$$

$$= \frac{800}{(1.0)(0.90)\sqrt{\frac{25}{0.5}}}$$

$$= 125.7$$

6. Select the valve size using the flow coefficient table and the calculated C_v value.

The required C_v of 125.7 exceeds the capacity of the assumed valve, which has a C_v of 121. Although for this example it may be obvious that the next larger size (4 inches) would be the correct valve size, this may not always be true, and a repeat of the above procedure should be carried out.

Assuming a 4-inch valve, C_v = 203. This value was determined from the flow coefficient table for an ANSI Class 300, 4-inch globe valve with an equal percentage cage.

Recalculate the required C_v using an assumed C_v value of 203 in the F_p calculation.

where,

$$\Sigma K = K_1 + K_2$$

$$= 1.5\left(1 - \frac{d^2}{D^2}\right)^2$$

$$= 1.5\left(1 - \frac{16}{64}\right)^2$$

$$= 0.84$$

and

$$F_p = \left[1.0 + \frac{\Sigma K}{N_2}\left(\frac{C_v}{d^2}\right)^2\right]^{-1/2}$$

$$= \left[1.0 + \frac{0.84}{890}\left(\frac{203}{4^2}\right)^2\right]^{-1/2}$$

$$= 0.93$$

and

$$C_v = \frac{q}{N_1 F_p \sqrt{\frac{P_1 - P_2}{G_f}}}$$

$$= \frac{800}{(1.0)(0.93)\sqrt{\frac{25}{0.5}}}$$

$$= 121.7$$

This solution indicates only that the 4-inch valve is large enough to satisfy the service conditions given. There may be cases, however, where a more accurate prediction of the C_v is required. In such cases, the required C_v should be redetermined using a new F_p value based on the C_v value obtained above. In this example, C_v is 121.7, which leads to the following result:

$$F_p = \left[1.0 + \frac{\Sigma K}{N_2}\left(\frac{C_v}{d^2}\right)^2\right]^{-1/2}$$

$$= \left[1.0 + \frac{0.84}{890}\left(\frac{121.7}{4^2}\right)^2\right]^{-1/2}$$

$$= 0.97$$

The required C_v then becomes:

$$C_v = \frac{q}{N_1 F_p \sqrt{\frac{P_1 - P_2}{G_f}}}$$

$$= \frac{800}{(1.0)(0.97)\sqrt{\frac{25}{0.5}}}$$

$$= 116.2$$

Because this newly determined C_v is very close to the C_v used initially for this recalculation (116.2 versus 121.7), the valve sizing procedure is complete, and the conclusion is that a 4-inch valve opened to about 75-percent of total travel should be adequate for the required specifications.

Chapter 4

Gas and Steam Valve Sizing

Sizing Valves for Compressible Fluids

Following is a six-step procedure for the sizing of control valves for compressible flow using the ISA standardized procedure. Each of these steps is important and must be considered during any valve sizing procedure. Steps 3 and 4 concern the determination of certain sizing factors that may or may not be required in the sizing equation depending on the service conditions of the sizing problem. If it is necessary for one or both of these sizing factors to be included in the sizing equation for a particular sizing problem, refer to the appropriate factor determination section(s), which is referenced and located in the following text.

1. Specify the necessary variables required to size the valve as follows:

- Desired valve design (e.g. balanced globe with linear cage)

- Process fluid (air, natural gas, steam, etc.) and

- Appropriate service conditions— q, or w, P_1, P_2 or ΔP, T_1, G_g, M, k, Z, and γ_1

The ability to recognize which terms are appropriate for a specific sizing procedure can only be acquired through experience with different valve sizing problems. If any of the above terms appear to be new or unfamiliar, refer to the Abbreviations and Terminology table 3-1 in chapter 3 for a complete definition.

2. Determine the equation constant, N.

N is a numerical constant contained in each of the flow equations to provide a means for using different systems of units. Values for these various constants and their applicable units are given in the Equation Constants Table 3-2 in Chapter 3.

Use either N_7 or N_9 if sizing the valve for a flow rate in volumetric units (scfh or m^3/h). Which of the two constants to use depends upon the specified service conditions. N_7 can be used only if the specific gravity, G_g, of the following gas has been specified along with the other required service conditions. N_9 can be used only if the molecular weight, M, of the gas has been specified.

Use either N_6 or N_8 if sizing the valve for a flow rate in mass units (lb/h or kg/h). Which of the two constants to use depends upon the specified service conditions. N_6 can be used only if the specific weight, γ_1, of the flowing gas has been specified along with the other required service conditions. N_8 can be used only if the molecular weight, M, of the gas has been specified.

3. Determine F_p, the piping geometry factor.

F_p is a correction factor that accounts for any pressure losses due to piping fittings such as reducers, elbows, or tees that might be attached directly to the inlet and outlet connections of the control valves to be sized. If such fittings are attached to the valve, the F_p factor must be considered in the sizing procedure. If, however, no fittings are attached to the valve, F_p has a value of 1.0 and simply drops out of the sizing equation.

Also, for rotary valves with reducers and other valve designs and fitting styles, determine the F_p factors by using the procedure for Determining F_p the Piping Geometry Factor, which is located on page 3-4 in chapter 3.

4. Determine Y, the expansion factor, as follows:

$$Y = 1 - \frac{x}{3F_k\ x_T}$$

where,

F_k = k/1.4, the ratio of specific heats factor

k = Ratio of specific heats

x = $\Delta P/P_1$, the pressure drop ratio

x_T = The pressure drop ratio factor for valves installed without attached fittings. More definitively, x_T is the pressure drop ratio required to produce critical, or maximum, flow through the valve when F_k = 1.0

If the control valve to be installed has fittings such as reducers or elbows attached to it, then their effect is accounted for in the expansion factor equation by replacing the x_T term with a new factor x_{TP}. A procedure for determining the x_{TP} factor is described in the following section for Determining x_{TP}, the Pressure Drop Ratio Factor.

Note

Conditions of critical pressure drop are realized when the value of x becomes equal to or exceeds the appropriate value of the product of either Fk x_T or F_k x_{TP} at which point:

$$y = 1 - \frac{x}{3F_k\ x_T} = 1 - 1/3 = 0.667$$

Although in actual service, pressure drop ratios can, and often will, exceed the indicated critical values, this is the point where critical flow conditions develop. Thus, for a constant P_1, decreasing P_2 (i.e., increasing ΔP) will not result in an increase in the flow rate through the valve. Values of x, therefore, greater than the product of either $F_k x_T$ or $F_k x_{TP}$ must never be substituted in the expression for Y. This means that Y can never be less than 0.667. This same limit on values of x also applies to the flow equations that are introduced in the next section.

5. Solve for the required C_v using the appropriate equation:

For volumetric flow rate units—

- If the specific gravity, G_g, of the gas has been specified:

$$C_v = \frac{q}{N_7\ F_p\ P_1\ Y \sqrt{\frac{x}{G_g\ T_1\ Z}}}$$

- If the molecular weight, M, of the gas has been specified:

$$C_v = \frac{q}{N_9\ F_p\ P_1\ Y \sqrt{\frac{x}{M\ T_1\ Z}}}$$

For mass flow rate units—

- If the specific weight, γ_1, of the gas has been specified:

$$C_v = \frac{w}{N_6 F_p Y \sqrt{x\ P_1\ \gamma_1}}$$

- If the molecular weight, M, of the gas has been specified:

$$C_v = \frac{w}{N_8\ F_p\ P_1\ Y \sqrt{\frac{x\ M}{T_1\ Z}}}$$

In addition to C_v, two other flow coefficients, K_v and A_v, are used, particularly outside of North America. The following relationships exist:

$$K_v = (0.865)(C_v)$$

$$A_v = \left(2.40\ X\ 10^{-5}\right)(C_v)$$

6. Select the valve size using the appropriate flow coefficient table and the calculated C_v value.

Determining x_{TP}, the Pressure Drop Ratio Factor

If the control valve is to be installed with attached fittings such as reducers or elbows, then their effect is accounted for in the expansion factor equation by replacing the x_T term with a new factor, x_{TP}.

$$x_{TP} = \frac{x_T}{F_p^2}\left[1 + \frac{x_T K_i}{N_5}\left(\frac{C_v}{d^2}\right)^2\right]^{-1}$$

where,

N_5 = Numerical constant found in the Equation Constants table

d = Assumed nominal valve size

C_v = Valve sizing coefficient from flow coefficient table at 100 percent travel for the assumed valve size

F_p = Piping geometry factor

x_T = Pressure drop ratio for valves installed without fittings attached. x_T values are included in the flow coefficient tables

In the above equation, K_i, is the inlet head loss coefficient, which is defined as:

$$K_i = K_1 + K_{B1}$$

where,

K_1 = Resistance coefficient of upstream fittings (see the procedure for Determining F_p, the Piping Geometry Factor, which is contained in the section for Sizing Valves for Liquids).

K_{B1} = Inlet Bernoulli coefficient (see the procedure for Determining F_p, the piping Geometry factor, which is contained in the section for Sizing Valves for Liquids.)

Compressible Fluid Sizing Sample Problem No. 1

Determine the size and percent opening for a Fisher Design V250 ball valve operating with the following service conditions. Assume that the valve and line size are equal.

1. Specify the necessary variables required to size the valve:

- Desired valve design—Design V250 valve
- Process fluid—Natural gas
- Service conditions—

P_1 = 200 psig = 214.7 psia

P_2 = 50 psig = 64.7 psia

ΔP = 150 psi

$x = \Delta P/P_1 = 150/214.7 = 0.70$

T_1 = 60°F = 520°R

M = 17.38

G_g = 0.60

k = 1.31

q = 6.0 x 10^6 scfh

2. Determine the appropriate equation constant, N, from the Equation Constants Table 3-2 in Chapter 3.

Because both G_g and M have been given in the service conditions, it is possible to use an equation containing either N_7 or N_9. In either case, the end result will be the same. Assume that the equation containing G_g has been arbitrarily selected for this problem. Therefore N_7 = 1360.

3. Determine F_p, the piping geometry factor. Since valve and line size are assumed equal, F_p = 1.0.

4. Determine Y, the expansion factor.

$$F_k = \frac{k}{1.40}$$

$$= \frac{1.31}{1.40}$$

$$= 0.94$$

It is assumed that an 8-inch Design V250 valve will be adequate for the specified service conditions. From the flow coefficient table 4-1, x_T for an 8-inch Design V250 valve at 100-percent travel is 0.137.

x = 0.70 (This was calculated in step 1.)

Since conditions of critical pressure drop are realized when the calculated value of x becomes equal to or exceeds the appropriate value of $F_k x_T$, these values should be compared.

$$F_k x_T = (0.94)\ (0.137)$$

$$= 0.129$$

Because the pressure drop ratio, x = 0.70 exceeds the calculated critical value, $F_k x_T = 0.129$, choked flow conditions are indicated. Therefore, Y = 0.667, and $x = F_K X_T = 0.129$.

5. Solve for required C_V using the appropriate equation.

$$C_v = \frac{q}{N_7\ F_p\ P_1\ Y \sqrt{\frac{x}{G_g\ T_1\ Z}}}$$

The compressibility factor, Z, can be assumed to be 1.0 for the gas pressure and temperature given and $F_p = 1$ because valve size and line size are equal.

So,

$$C_v = \frac{6.0\ \times\ 10^6}{(1360)(1.0)(214.7)(0.667)\sqrt{\frac{0.129}{(0.6)(520)(1.0)}}} = 1515$$

6. Select the valve size using the flow coefficient table and the calculated C_V value.

The above result indicates that the valve is adequately sized (rated $C_V = 2190$). To determine the percent valve opening, note that the required C_V occurs at approximately 83 degrees for the 8-inch Design V250 valve. Note also that, at 83 degrees opening, the x_T value is 0.252, which is substantially different from the rated value of 0.137 used initially in the problem. The next step is to rework the problem using the x_T value for 83 degrees travel.

The $F_k\, x_T$ product must now be recalculated.

$$x = F_k\ x_T$$

$$= (0.94)\ (0.252)$$

$$= 0.237$$

The required C_V now becomes:

$$C_v = \frac{q}{N_7\ F_p\ P_1\ Y \sqrt{\frac{x}{G_g\ T_1\ Z}}}$$

$$= \frac{6.0\ \times\ 10^6}{(1360)(1.0)(214.7)(0.667)\sqrt{\frac{0.237}{(0.6)(520)(1.0)}}}$$

$$= 1118$$

The reason that the required C_V has dropped so dramatically is attributable solely to the difference in the x_T values at rated and 83 degrees travel. A C_V of 1118 occurs between 75 and 80 degrees travel.

The appropriate flow coefficient table indicates that x_T is higher at 75 degrees travel than at 80 degrees travel. Therefore, if the problem were to be reworked using a higher x_T value, this should result in a further decline in the calculated required C_V.

Reworking the problem using the x_T value corresponding to 78 degrees travel (i.e., x_T = 0.328) leaves:

$$x = F_k\ x_T$$

$$= (0.94)\ (0.328)$$

$$= 0.308$$

and,

$$C_v = \frac{q}{N_7\ F_p\ P_1\ Y \sqrt{\frac{x}{G_g\ T_1\ Z}}}$$

$$= \frac{6.0\ \times\ 10^6}{(1360)(1.0)(214.7)(0.667)\sqrt{\frac{0.308}{(0.6)(520)(1.0)}}}$$

$$= 980$$

The above C_V of 980 is quite close to the 75 degree travel C_V. The problem could be reworked further to obtain a more precise predicted opening; however, for the service conditions given, an 8-inch Design V250 valve installed in an 8-inch line will be approximately 75 degrees open.

Compressible Fluid Sizing Sample Problem No. 2

Assume steam is to be supplied to a process designed to operate at 250 psig. The supply source is a header maintained at 500 psig and 500°F. A 6-inch line from the steam main to the process is being planned. Also, make the assumption that if the required valve size is less than 6 inches, it will be installed using concentric reducers. Determine the appropriate Design ED valve with a linear cage.

1. Specify the necessary variables required to size the valve:

a. Desired valve design—ANSI Class 300 Design ED valve with a linear cage. Assume valve size is 4 inches.

b. Process fluid—superheated steam

c. Service conditions—

w = 125,000 lb/h

P_1 = 500 psig = 514.7 psia

P_2 = 250 psig = 264.7 psia

ΔP = 250 psi

x = $\Delta P/P_1$ = 250/514.7 = 0.49

T_1 = 500°F

γ_1 = 1.0434 lb/ft^3 (from Properties of Saturated Steam table)

k= 1.28 (from Properties of Saturated Steam table)

2. Determine the appropriate equation constant, N, from the Equation Constants Table 3-2 in Chapter 3.

Because the specified flow rate is in mass units, (lb/h), and the specific weight of the steam is also specified, the only sizing equation that can be used is that which contains the N_6 constant. Therefore,

$$N_6 = 63.3$$

3. Determine F_p, the piping geometry factor.

$$F_p = \left[1 + \frac{\Sigma K}{N_2}\left(\frac{C_v}{d^2}\right)^2\right]^{-1/2}$$

where,

N_2 = 890, determined from the Equation Constants table

d = 4 in.

C_v = 236, which is the value listed in the flow coefficient Table 4-2 for a 4-inch Design ED valve at 100-percent total travel.

and

$$\Sigma K = K_1 + K_2$$

$$= 1.5\left(1 - \frac{d^2}{D^2}\right)^2$$

$$= 1.5\left(1 - \frac{4^2}{6^2}\right)^2$$

$$= 0.463$$

Finally:

$$F_p = \left[1 + \frac{0.463}{890}\left(\frac{(1.0)(236)}{(4)^2}\right)^2\right]^{-1/2}$$

$$= 0.95$$

4. Determine Y, the expansion factor.

$$Y = 1 - \frac{x}{3F_k\ x_{TP}}$$

where,

$$F_k = \frac{k}{1.40}$$

$$= \frac{1,28}{1.40}$$

$$= 0.91$$

x = 0.49 (As calculated in step 1.)

Because the 4-inch valve is to be installed in a 6-inch line, the x_T term must be replaced by x_{TP}.

$$x_{TP} = \frac{x_T}{F_p{}^2}\left[1 + \frac{x_T\ K_i}{N_5}\left(\frac{C_v}{d^2}\right)^2\right]^{-1}$$

where,

N_5 = 1000, from the Equation Constants table

d = 4 inches

F_p = 0.95, determined in step 3

x_T = 0.688, a value determined from the appropriate listing in the flow coefficient table

C_v = 236, from step 3

and

$$K_i = K_1 + K_{B1}$$

$$= 0.5\left(1 - \frac{d^2}{D^2}\right)^2 + \left[1 - \left(\frac{d}{D}\right)^4\right]$$

$$= 0.5\left(1 - \frac{4^2}{6^2}\right)^2 + \left[1 - \left(\frac{4}{6}\right)^4\right]$$

$$= 0.96$$

where D = 6 inches

so:

$$X_{TP} = \frac{0.69}{0.95^2}\left[1 \frac{(0.69)(0.96)}{1000}\left(\frac{236}{4^2}\right)^2\right]^{-1} = 0.67$$

Finally:

$$Y = 1 - \frac{x}{3\ F_k\ x_{TP}}$$

$$= 1 - \frac{0.49}{(3)\ (0.91)\ (0.67)}$$

$$= 0.73$$

5. Solve for required C_v using the appropriate equation.

$$C_v = \frac{w}{N_6\ F_P\ Y \sqrt{x\ P_1\ \gamma_1}}$$

$$C_v = \frac{125,000}{(63.3)(0.95)(0.73)\sqrt{(0.49)(514.7)(1.0434)}}$$

$$= 176$$

6. Select the valve size using flow coefficient tables and the calculated C_v value.

Refer to the flow coefficient Table 4-2 for Design ED valves with linear cage. Because the assumed 4-inch valve has a C_v of 236 at 100-percent travel and the next smaller size (3 inches) has a C_v of only 148, it can be surmised that the assumed size is correct. In the event that the calculated required C_v had been small enough to have been handled by the next smaller size, or if it had been larger than the rated C_v for the assumed size, it would have been necessary to rework the problem again using values for the new assumed size.

Table 4-1. *Representative Sizing Coefficients for Rotary Shaft Valves*

Valve Size (inches)	Valve Style	Degrees of Valve Opening	C_v	F_L	X_T	F_D
1	V–Notch Ball Valve	60	15.6	0.86	0.53	
		90	34.0	0.86	0.42	
1 1/2	V–Notch Ball Valve	60	28.5	0.85	0.50	
		90	77.3	0.74	0.27	
2	V–Notch Ball Valve	60	59.2	0.81	0.53	
		90	132	0.77	0.41	
	High Performance Butterfly Valve	60	58.9	0.76	0.50	0.49
		90	80.2	0.71	0.44	0.70
3	V–Notch Ball Valve	60	120	0.80	0.50	0.92
		90	321	0.74	0.30	0.99
	High Performance Butterfly Valve	60	115	0.81	0.46	0.49
		90	237	0.64	0.28	0.70
4	V–Notch Ball Valve	60	195	0.80	0.52	0.92
		90	596	0.62	0.22	0.99
	High Performance Butterfly Valve	60	270	0.69	0.32	0.49
		90	499	0.53	0.19	0.70
6	V–Notch Ball Valve	60	340	0.80	0.52	0.91
		90	1100	0.58	0.20	0.99
	High Performance Butterfly Valve	60	664	0.66	0.33	0.49
		90	1260	0.55	0.20	0.70
8	V–Notch Ball Valve	60	518	0.82	0.54	0.91
		90	1820	0.54	0.18	0.99
	High Performance Butterfly Valve	60	1160	0.66	0.31	0.49
		90	2180	0.48	0.19	0.70
10	V–Notch Ball Valve	60	1000	0.80	0.47	0.91
		90	3000	0.56	0.19	0.99
	High Performance Butterfly Valve	60	1670	0.66	0.38	0.49
		90	3600	0.48	0.17	0.70
12	V–Notch Ball Valve	60	1530	0.78	0.49	0.92
		90	3980	0.63	0.25	0.99
	High Performance Butterfly Valve	60	2500			0.49
		90	5400			0.70
16	V–Notch Ball Valve	60	2380	0.80	0.45	0.92
		90	8270	0.37	0.13	1.00
	High Performance Butterfly Valve	60	3870	0.69	0.40	
		90	8600	0.52	0.23	

Table 4-2. *Representative Sizing Coefficients for Design ED Single-Ported Globe Style Valve Bodies*

Valve Size (inches)	Valve Plug Style	Flow Characteristic	Port Dia. (in.)	Rated Travel (in.)	C_V	F_L	X_T	F_D
1/2	Post Guided	Equal Percentage	0.38	0.50	2.41	0.90	0.54	0.61
3/4	Post Guided	Equal Percentage	0.56	0.50	5.92	0.84	0.61	0.61
1	Micro Form™	Equal Percentage	3/8	3/4	3.07	0.89	0.66	0.72
			1/2	3/4	4.91	0.93	0.80	0.67
			3/4	3/4	8.84	0.97	0.92	0.62
	Cage Guided	Linear	1 5/16	3/4	20.6	0.84	0.64	0.34
		Equal Percentage	1 5/16	3/4	17.2	0.88	0.67	0.38
1 1/2	Micro-Form™	Equal Percentage	3/8	3/4	3.20	0.84	0.65	0.72
			1/2	3/4	5.18	0.91	0.71	0.67
			3/4	3/4	10.2	0.92	0.80	0.62
	Cage Guided	Linear	1 7/8	3/4	39.2	0.82	0.66	0.34
		Equal Percentage	1 7/8	3/4	35.8	0.84	0.68	0.38
2	Cage Guided	Linear	2 5/16	1 1/8	72.9	0.77	0.64	0.33
		Equal Percentage	2 5/16	1 1/8	59.7	0.85	0.69	0.31
3	Cage Guided	Linear	3 7/16	1 1/2	148	0.82	0.62	0.30
		Equal Percentage			136	0.82	0.68	0.32
4	Cage Guided	Linear	4 3/8	2	236	0.82	0.69	0.28
		Equal Percentage			224	0.82	0.72	0.28
6	Cage Guided	Linear	7	2	433	0.84	0.74	0.28
		Equal Percentage			394	0.85	0.78	0.26
8	Cage Guided	Linear	8	3	846	0.87	0.81	0.31
		Equal Percentage			818	0.86	0.81	0.26

Chapter 5

Control Valve Noise

Noise Control

In closed systems (not vented to atmosphere), any noise produced in the process becomes airborne only by transmission through the valves and adjacent piping that contain the flowstream. The sound field in the flowstream forces these solid boundaries to vibrate. The vibrations cause disturbances in the ambient atmosphere that propagate as sound waves.

Noise control employs either source treatment, path treatment, or both. Source treatment, preventing or attenuating noise at its source, is the most desirable approach, if economically and physically feasible.

Recommended cage-style source treatment approaches are depicted in Figure 5-1. The upper view shows a cage with many narrow parallel slots designed to minimize turbulence and provide a favorable velocity distribution in the expansion area. This economical approach to quiet valve design can provide 15 to 20 dBA noise reduction with little or no decrease in flow capacity.

The lower view in Figure 5-1 illustrates flow through a two-stage, cage-style trim designed for optimum noise attenuation where pressure drop ratios ($\Delta P/P_1$) are high.

This trim design can reduce the valve noise as much as 40 dBA by utilizing a combination of several noise reduction strategies:

- Unique passage shape that reduces the conversion of total stream power generated by the valve into noise power.
- Multistage pressure reduction that divides the stream power between stages and further reduces the acoustic conversion efficiency.

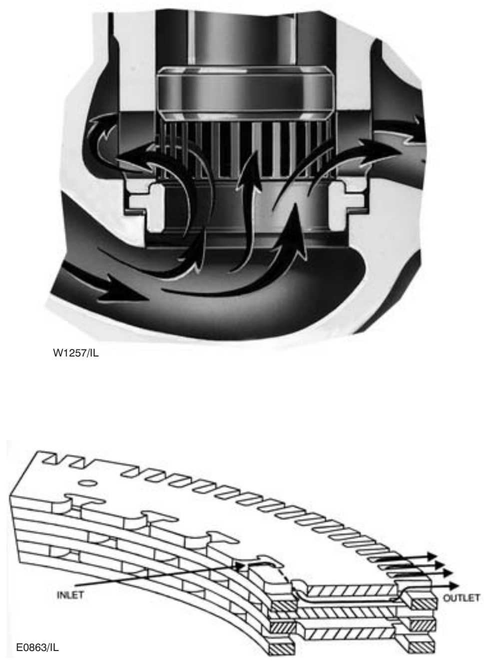


Figure 5-1. *Valve trim designs for reducing aerodynamic noise*

- Frequency spectrum shifting that reduces acoustic energy in the audible range by capitalizing on the transmission loss of the piping.
- Exit jet independence that avoids noise regeneration due to jet coalescence.
- Velocity management is accomplished with expanding areas to accommodate the expanding gas.
- Complementary body designs that avoid flow impingement on the body wall and secondary noise sources.

For control valve applications operating at high pressure ratios ($\Delta P/P_1 > 0.8$) the series restriction

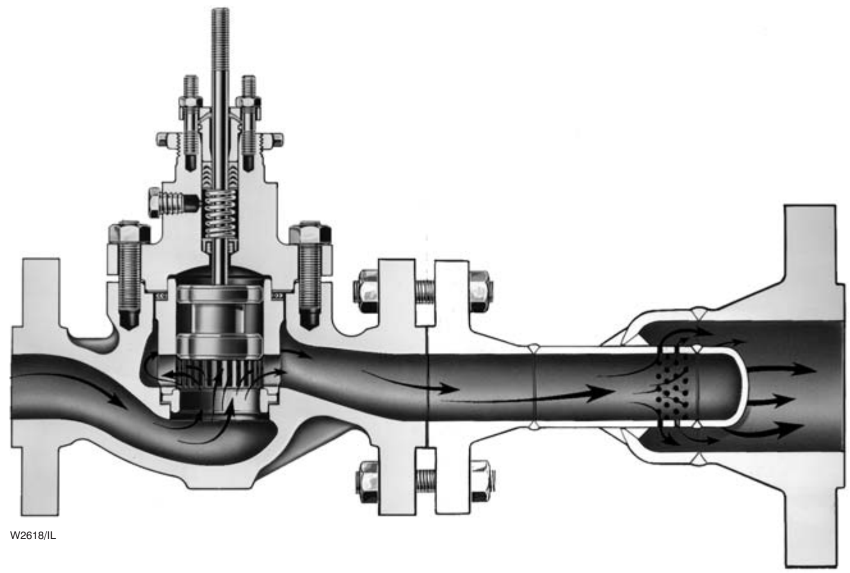


Figure 5-2*. Valve and inline diffuser combination*

approach, splitting the total pressure drop between the control valve and a fixed restriction (diffuser) downstream of the valve can be effective in minimizing noise. To optimize the effectiveness of a diffuser, it must be designed (special shape and sizing) for each given installation so that the noise levels generated by the valve and diffuser are equal. Figure 5-2 shows a typical installation.

Control systems venting to atmosphere are generally very noisy because of the high pressure ratios and high exit velocities involved. Dividing the total pressure drop between the actual vent and an upstream control valve, by means of a vent diffuser, quiets both the valve and the vent. A properly sized vent diffuser and valve combination, such as that shown in Figure 5-3, can reduce the overall system noise level as much as 40 dBA.

Source treatment for noise problems associated with control valves handling liquid is directed primarily at eliminating or minimizing cavitation. Because flow conditions that will produce cavitation can be accurately predicted, valve noise resulting from cavitation can be eliminated by application of appropriate limits to the service conditions at the valve by use of break-down orifices, valves in series, etc. Another approach to source treatment is using special valve trim that uses the series restriction concept to eliminate cavitation as shown in Figure 5-4.

A second approach to aerodynamic noise control is that of path treatment. The fluid stream is an excellent noise transmission path. Path treatment

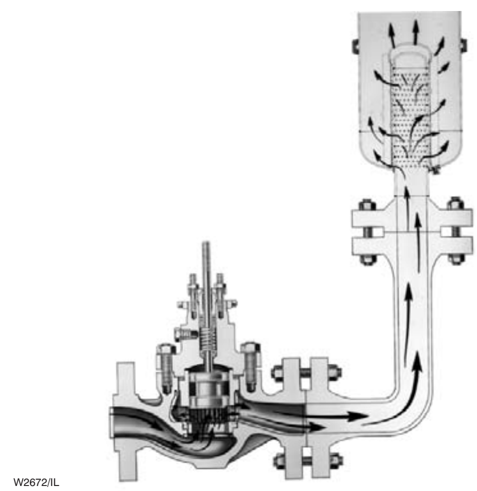
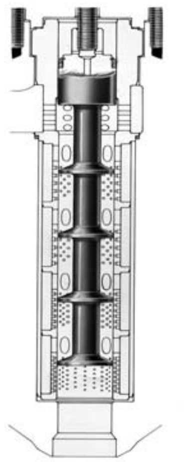


Figure 5-3. *Valve and vent diffuser combination*

consists of increasing the impedance of the transmission path to reduce the acoustic energy communicated to the receiver.

Dissipation of acoustic energy by use of acoustical absorbent materials is one of the most effective methods of path treatment. Whenever possible the acoustical material should be located in the flow stream either at or immediately downstream of the noise source. In gas systems, inline silencers effectively dissipate the noise within the fluid stream and attenuate the noise level transmitted to the solid boundaries. Where high mass flow rates and/or high pressure ratios across the valve exist, inline silencers, such as that shown in



W2673/IL

Figure 5-4. *Special valve design to eliminate cavitation*

W1304/IL

Figure 5-5. *Typical in-line silencer*

Figure 5-5, are often the most realistic and economical approach to noise control. Use of absorption-type inline silencers can provide almost any degree of attenuation desired. However, economic considerations generally limit the insertion loss to approximately 25 dBA.

Noise that cannot be eliminated within the boundaries of the flow stream must be eliminated by external treatment. This approach to the abatement of control valve noise suggests the use of heavy walled piping, acoustical insulation of the exposed solid boundaries of the fluid stream, use of insulated boxes, buildings, etc., to isolate the noise source.

Path treatment such as heavy wall pipe or external acoustical insulation can be an economical and effective technique for localized noise abatement. However, noise is propagated for long distances via the fluid stream and the effectiveness of the heavy wall pipe or external insulation ends where the treatment ends.

Aerodynamic Noise Prediction

Industry leaders use the International Electrotechnical Commission standard *IEC 534-8-3: Industrial-process control valves—Part 8: Noise Considerations—Section 3: Control valve aerodynamic noise prediction* method. This method consists of a mix of thermodynamic and aerodynamic theory and some empirical information. The design of the method allows a noise prediction for a valve based only on the measurable geometry of the valve and the service conditions applied to the valve. There is no need for specific empirical data for each valve design and size. Because of this pure analytical approach to valve noise prediction the IEC method allows an objective evaluation of alternatives.

The method defines five basic steps to a noise prediction:

1. Calculate the total stream power in the process at the vena contracta.
The noise of interest is generated by the valve in and downstream of the vena contracta. If the total power dissipated by throttling at the vena contracta can be calculated, then the fraction that is noise power can be determined. Since power is the time rate of energy, a form of the familiar equation for calculating kinetic energy can be used. The kinetic energy equation is $1/2\ mv^2$ where m is mass and v is velocity. If the mass flow rate is substituted for the mass term, then the equation calculates the power. The velocity is the vena contracta velocity and is calculated with the energy equation of the First Law of Thermodynamics.

2. Determine the fraction of total power that is acoustic power.
The method considers the process conditions applied across the valve to determine the particular noise generating mechanism in the valve. There are five defined regimes dependent on the relationship of the vena contracta pressure and the downstream pressure. For each of these regimes an acoustic efficiency is defined and calculated. This acoustic efficiency establishes the fraction of the total stream power, as calculated in Step 1, which is noise power. In designing a quiet valve, lower acoustic efficiency is one of the goals.

3. Convert acoustic power to sound pressure.
The final goal of the IEC prediction method is determination of the sound pressure level at a reference point outside the valve where human hearing is a concern. Step 2 delivers acoustic power, which is not directly measurable. Acoustic or sound pressure is measurable and therefore

has become the default expression for noise in most situations. Converting from acoustic power to the sound pressure uses basic acoustic theory.

4. Account for the transmission loss of the pipewall and restate the sound pressure at the outside surface of the pipe.

Steps 1 through 3 are involved with the noise generation process inside the pipe. There are times when this is the area of interest, but the noise levels on the outside of the pipe are the prime requirement. The method must account for the change in the noise as the reference location moves from inside the pipe to outside the pipe. The pipe wall has physical characteristics, due to its material, size, and shape, that define how well the noise will transmit through the pipe. The fluid-borne noise inside the pipe must interact with the inside pipe wall to cause the pipe wall to vibrate, then the vibration must transmit through the pipe wall to the outside pipe wall, and there the outside pipe wall must interact with the atmosphere to generate sound waves. These three steps of noise transmission are dependent on the noise frequency. The method represents the frequency of the valve noise by determining the peak frequency of the valve noise spectrum. The method also determines the pipe transmission loss as a function of frequency. The method then compares the internal noise spectrum and the transmission-loss spectrum to determine how much the external sound pressure will be attenuated by the pipe wall.

5. Account for distance and calculate the sound pressure level at the observer's location.

Step 4 delivers the external sound pressure level at the outside surface of the pipe wall. Again, basic acoustic theory is applied to calculate the sound pressure level at the observer's location. Sound power is constant for any given situation, but the associated sound pressure level varies with the area the power is spread over. As the observer moves farther away from the pipe wall, the total area the sound power is spread over increases. This causes the sound pressure level to decrease.

Hydrodynamic

Noticeable hydrodynamic noise is usually associated with cavitation. The traditional description of the sound is as if rocks were flowing inside the pipe. This association of hydrodynamic noise with cavitation is reflected in the various prediction methods available today. The methods account for one noise characteristic for liquids in non-choked flow situations and another characteristic in choked, cavitating flow situations.

There are a variety of situations where the fluid is a two-phase mixture. These include liquid-gas two-phase fluids at the inlet of the valve, flashing fluids, and fluids that demonstrate out-gassing due to throttling. Noise prediction methods for these cases are not yet well established. Test results and field surveys of installed multi-phase systems indicate these noise levels do not contribute to overall plant noise levels or exceed worker exposure levels.

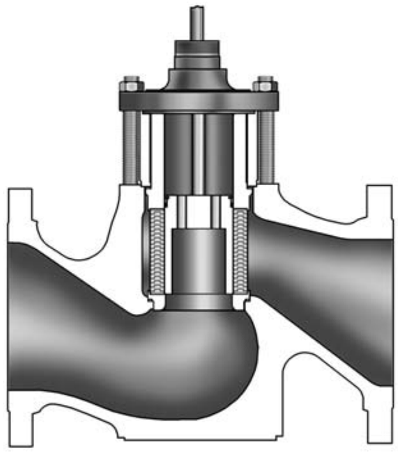


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Figure 5-6. *Globe-style valve with noise abatement cage for aerodynamic flow*

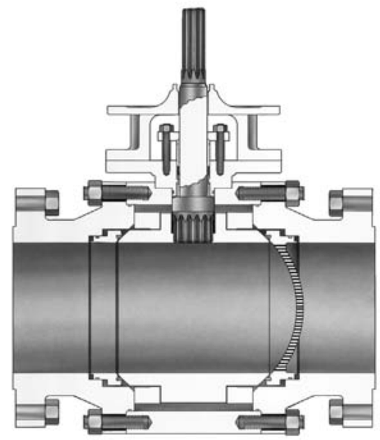

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Figure 5-7. *Ball-style valve with attenuator to reduce hydrodynamic noise*

Noise Summary

The amount of noise that will be generated by a proposed control valve installation can be quickly and reasonably predicted by use of industry standard methods. These methods are available in computer software for ease of use. Such sizing and noise prediction tools help in the proper selection of noise reduction equipment such as shown in Figures 5-6 and 5-7. Process facility requirements for low environmental impact will continue to drive the need for quieter control valves. The prediction technologies and valve designs that deliver this are always being improved. For the latest in either equipment or prediction technology, contact the valve manufacturer's representative.

Chapter 6

Control Valve Cavitation and Flashing

Cavitation and a related issue, flashing, are hydrodynamic flow phenomena that began to receive recognition as technical engineering problems in the early 1900s. At that time, observations led to the conclusion that vaporization of water in the vicinity of high-speed propellers was responsible for a decrease in their effectiveness. This liquid vaporization was defined later as cavitation.

Cavitation and flashing are a concern to the control valve industry not only because they decrease flow capability through control valves, but also because they create noise, vibration, and material damage.

This chapter will define cavitation and flashing and will discuss the negative effects of both flow phenomena. Control valves designed to prevent cavitation and flashing damage will be discussed along with other considerations in choosing control valves for cavitating and flashing services.

Cavitation and Flashing

Cavitation and flashing are purely liquid flow phenomena. Gases and vapors can neither cavitate nor flash. While there are three recognized types of cavitation, the most significant type, as applied to control valves, is vaporous cavitation.

Vaporous cavitation consists of explosive vapor cavity growth within a liquid resulting from local liquid vaporization. Cavity formation is followed by rapid cavity collapse due to vapor re-condensation. The phase change sequence is caused by local pressure fluctuations in the liquid.

To understand the conditions in a control valve that can cause cavitation or flashing, consider the simple restriction shown in Figure 6-1. In this figure the pressure of the liquid, P, is plotted as a function of the distance, x, through the simple restriction shown at the top of the figure. Figure 6-2 is a plot of the velocity, V, of the liquid versus the distance, x, as it relates to the pressure profile.

There is a point called the vena contracta where the flow area of the liquid will be at a minimum. The vena contracta will be downstream some distance of the minimum flow area. This distance will vary with pressure conditions and actual type of restriction.

Figure 6-2 shows that at the vena contracta point, the pressure will hit a minimum that will be defined as the vena contracta pressure, P_{VC}. After the vena contracta pressure occurs, the liquid pressure will recover to the downstream pressure. Note that the downstream pressure is less than the upstream pressure because some of the available energy is converted into heat.

To maintain a constant flow rate through the restriction, the flow velocity must increase to offset the effect of the decreased cross-sectional flow area. As the energy due to fluid velocity (kinetic energy) increases, other forms of fluid energy must be offset accordingly, which accounts for the pressure drop of the liquid through the restriction.

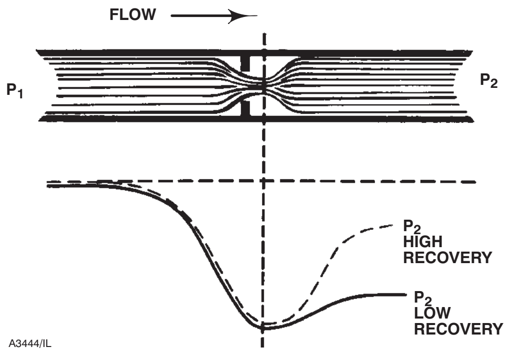

Figure 6-1. *Pressure profile of flow through a restriction*

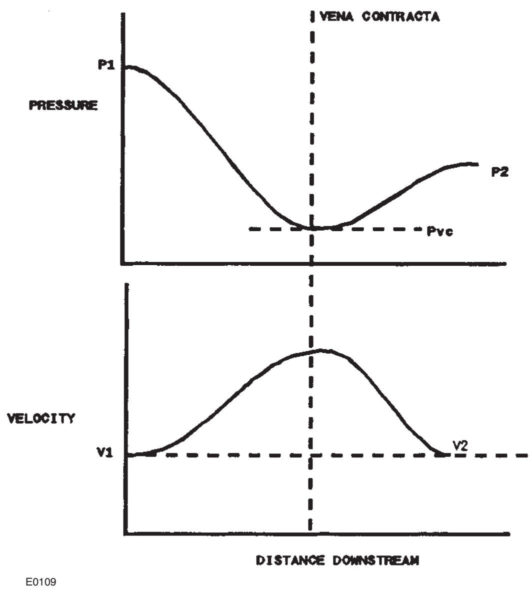

Figure 6-2. *Pressure versus velocity curves illustrate that the highest flow rate occurs at the vena contracta.*

This relationship can be shown in equation form by looking at Bernoulli's Equation (equation 6-1). Bernoulli's Equation between point P_1 and the vena contracta may be written as:

$$\frac{\rho V_1^{\ 2}}{2g_c} + P_1 = \frac{\rho V_{vc}^{\ 2}}{2g_c} + P_{vc} \qquad (6\text{-}1)$$

An increase in the fluid velocity (kinetic energy) must be offset by a decrease in the static pressure.

To clearly define flashing and cavitation and also the difference between the two, it is best to look at the pressure profile (Figure 6-1) of flow through a fixed restriction in relationship to the vapor pressure of the liquid. Figure 6-3 shows this relationship and graphically defines the difference between cavitation and flashing.

A liquid that cavitates or flashes behaves in the same manner from the inlet to the vena contracta. As the pressure falls from the inlet pressure, a point is reached where the local fluid pressure is equal to the vapor pressure. When the fluid pressure falls below the vapor pressure, the fluid becomes unstable, and it begins to turn to vapor. This phase change from a liquid to a vapor shows up as bubbles in the flow stream and is very similar to the bubbles that form in a boiling pan of water.

When the water hits a temperature (212°F, 100°C) where its vapor pressure is equal to the atmospheric pressure, bubbles will form. Boiling occurs when the vapor pressure increases to equal atmospheric pressure.

Conversely, in cavitation, the velocity change at the restriction lowers the fluid pressure to equal the vapor pressure. Cavitation is a much faster transition than boiling and produces much more vigorous results.

Flashing is another liquid phenomenon that is related to cavitation. In fact, the onset of flashing is identical to that of cavitation. It is only when the fluid passes the restriction and its pressure recovers that the two phenomena separate.

A fluid is said to flash when the downstream pressure of that fluid is less than its vapor pressure. The vapor bubbles that are formed when the pressure falls below the vapor pressure continue to grow, and eventually the liquid changes, or flashes to a vapor. It should be noted that flashing is determined by the vapor pressure of the liquid and the downstream pressure. Therefore, it is a system phenomenon, and no control valve can prevent flashing unless the system conditions (P_2, P_v) are changed.

Cavitation occurs when the pressure downstream of the valve rises back above the vapor pressure. The bubble can no longer exist as a vapor, and it immediately turns back to a liquid. Since the vapor bubble mass had a larger volume than the equivalent liquid mass, the bubble is said to implode. Note that cavitation is a liquid-vapor-liquid phase change. Unlike flashing, a correctly chosen control valve can prevent cavitation. This will be covered in detail in the upcoming sections.

water.

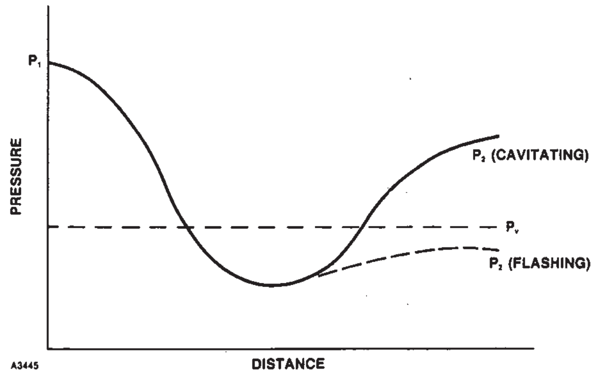

Figure 6-3. *Pressure recovery above the vapor pressure of the liquid results in cavitation. Remaining below the vapor pressure incure flashing.*

Bubble Cycle

The bubble cycle refers to the phase changes from a liquid to a vapor and back to a liquid that occur when a liquid cavitates. The behavior of these bubbles has a direct bearing on the degree of the negative side effects that will result. There are four primary events in the bubble cycle: nucleation, growth, collapse, and rebound.

In order for a liquid to cavitate at or near the vapor pressure of the liquid, it is necessary to have a place for the cavity to form. Often this is a small bubble of an entrained noncondensible gas in the liquid. These nuclei must be of a certain minimum size in order to explosively grow or cavitate. This process of initiating bubble formation is known as nucleation.

Once the bubble is formed, it proceeds across the reduced pressure region and grows in response to the continually decreasing pressure and increasing liquid vaporization. This portion of the bubble cycle is known as the growth portion. Eventually, the pressure recovery halts the growth of the bubble and forces it to collapse which is the third event in the bubble cycle.

Under certain circumstances, several growth and collapse cycles can occur in a series of rebounds.

There are four primary negative side effects of cavitation: excessive noise, excessive vibration, material damage, and deterioration of flow effectiveness. The flow through a restriction is

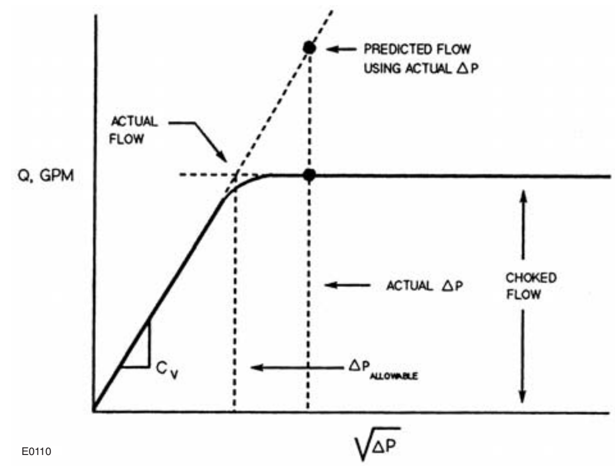

Figure 6-4. *The $\Delta P_{allowable}$ equation will predict the occurrence of fully choked flow*

normally proportional to the square root of the pressure drop. The constant of proportionality is the liquid-flow coefficient of the restriction, C_V, divided by the square root of the specific gravity, G.

Therefore,

$$Q = C_v \sqrt{\frac{P_1 - P_2}{G}} \qquad (6\text{-}2)$$

This relationship suggests that increasing the pressure differential across the restriction can continually increase the flow rate. However, in practice the relationship begins to break down when a sufficient amount of the vapor phase (produced in the cavitation process) is formed. Less of a flow increase is realized for the same pressure differential increase (at a given P_1), until finally the flow remains constant despite an increase in the pressure drop. Figure 6-4 portrays this choking phenomenon graphically.

The exact mechanisms of liquid choking are not fully confirmed, although there are parallels between it and critical flow in gas applications. In gas flows, the flow chokes when the flow velocity is equal to the acoustic wave speed (sonic velocity).

For pure liquids (incompressible fluids) the acoustic wave speed is very high. In liquids that partially vaporize, however, the fluid is actually a two-phase mixture and typically has a very low acoustic wave speed (actually lower than that of a pure gas). Therefore, it is possible for the mixture velocity to become equal to the sonic velocity and choke the flow.

Material Damage

Cavitation damage is usually the most troublesome negative side effect plaguing the control valve industry. It does not take many examples of such damage to fully demonstrate the destructive capabilities of cavitation.

Typically, cavitation damage is characterized by a very irregular, rough surface. The phrase “cinderlike appearance” is used frequently to describe cavitation damage. It is discernible from other types of flow damage such as erosion and flashing damage, which are usually very smooth and shiny in appearance. This next section will deal with cavitation damage, although most of the comments also can apply to flashing damage. Figure 6-6 illustrates these differences.

While the results of cavitation damage are all too familiar, the events and mechanisms of the cavitation damage process are not known or understood completely in spite of extensive study over the years. There is general agreement, however, on a number of aspects of the process and a consistency in certain observations.

Cavitation damage has been consistently observed to be associated with the collapse stage of the bubble dynamics. Furthermore, this damage consists of two primary events or phases: (1) an attack on a material surface as a result of cavitation in the liquid, and (2) the response or reaction of the material to the attack. Any factor that influences either of these events will have some sort of final effect on the overall damage characteristics.

The attack stage of the damage process has been attributed to various mechanisms, but none of them account for all the observed results. It appears that this attack involves two factors that interact in a reinforcing manner: (1) mechanical attack and (2) chemical attack.

There is evidence indicating the almost universal presence of a mechanical attack component which can occur in either of two forms:

1. *Erosion resulting from high-velocity microjets impinging on the material surface.*

2. *Material deformation and failure resulting from shock waves impinging on the material surface.*

In the first type of mechanical attack, a small, high-velocity liquid jet is formed during the asymmetrical collapse of a vapor bubble. If orientation and proximity of the jets is proper, a

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Figure 6-6. *The top plug shows the characteristic rough texture of cavitation damage, which differs greatly from the polished appearance of damage due to flashing (lower photo). The two damage mechanisms vary greatly.*

damaging attack occurs on the metal surface, as shown in Figure 6-7. This is the most probable form of mechanical attack, and high-speed cinematography, liquid drop impingement comparisons, and various analytical studies support its presence.

The second type of mechanical attack—shock wave impingement—does not appear to be as dominant. Analytical estimations of vapor bubble collapse pressures do not suggest that the shock waves are on a damaging order of magnitude—at

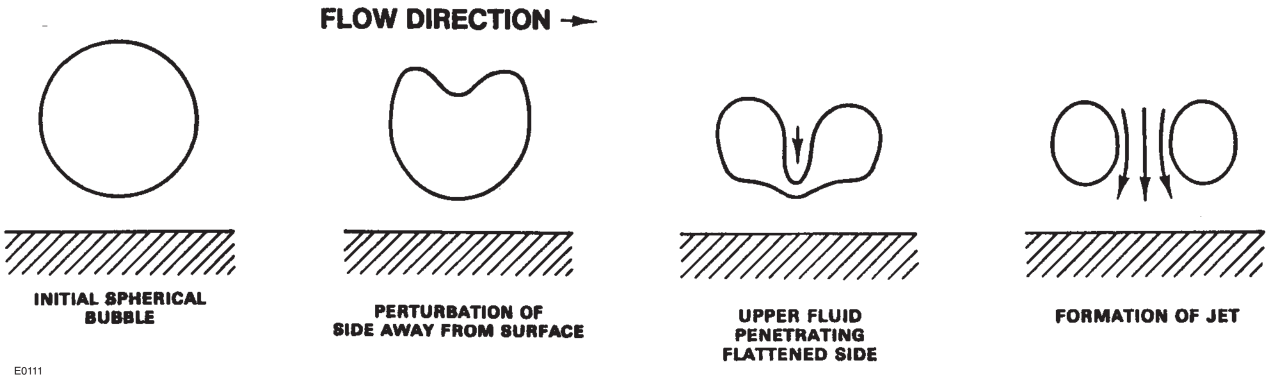

Figure 6-7. *The implosion of cavitation vapor cavities is rapid, asymmetric and very energetic. The mechanics of collapse give rise to high velocity liquid jets, which impinge on metallic surfaces. Ultimately, the metal fatigues and breaks away in small pieces.*

least during the initial collapse. Experimental studies bear this out. They also reveal that resulting collapse pressures increase in magnitude with subsequent rebound collapses and become potentially damaging.

The other primary component of attack, chemical attack, is perhaps more significant since it interacts with the mechanical component, rather than acting by itself. After a period of mechanical attack many of the protective coatings of a material, (films, oxides, etc.) are physically removed, making the base material more vulnerable to chemical attack.

Just as a number of variables have an effect on the behavior of individual cavities, so too there are influences affecting the degree and extent of material damage. The principal influences include such variables as air content, pressure, velocity, and temperature.

Air content impacts cavitation damage primarily through its effect on cavity mechanics, as previously discussed. Again, two opposing trends are evident on increasing the amount of air. Adding air supplies more entrained air nuclei, which in turn produce more cavities that can increase the total damage. After a point, however, continued increases in air content disrupt the mechanical attack component and effectively reduce the total damage.

Pressure effects also exhibit two opposing trends. Given a fixed inlet pressure P_1, decreasing the backpressure P_2 tends to increase the number of cavities formed, which creates a worse situation. However, a lower backpressure also creates a lower collapse pressure differential (P_2 - P_v), resulting in a decrease in the intensity of the cavitation.

An additional pressure effect unrelated to the above concerns the location of damage. As the backpressure is changed, the pressure required to collapse the cavities moves upstream or downstream, depending on whether the pressure is increased or decreased, respectively. In addition to a change in the severity of the total damage, there may be an accompanying change in the physical location of the damage when pressure conditions are altered.

It should now be apparent that the cavitation and flashing damage process is a complex function of:

1. *Intensity and degree of cavitation (cavitation attack)*
2. *Material of construction (material response)*
3. *Time of exposure*

While the above-mentioned influences have been observed, they remain to be quantified. Often, experience is the best teacher when it comes to trying to quantify cavitation damage.

Noise

Although the noise associated with a cavitating liquid can be quite high, it is usually of secondary concern when compared to the material damage that can exist. Therefore, high intensity cavitation should be prevented to decrease the chance of material damage. If cavitation is prevented, the noise associated with the liquid flow will be less than 90 dBA.

For a flashing liquid, studies and experience have shown that the noise level associated with the valve will be less than 85 dBA, regardless of the pressure drop involved to create the flashing.

Hardware Choices for Flashing Applications

It was stated previously that flashing is a liquid flow phenomenon that is defined by the system, and not by the valve design. Therefore, since flashing cannot be prevented by the control valve, all that can be done is to prevent flashing damage.

There are three main factors that affect the amount of flashing damage in a control valve.

- Valve Design
- Materials of Construction
- System Design

Valve Design

While valve design has no bearing on whether flashing does or does not occur, it can have a large impact on the intensity of flashing damage. Generally, there are two valve designs that are more resistant to flashing damage than the generally accepted globe body.

An angle valve with standard trim in the flow down direction and with a downstream liner is perhaps the best solution to preventing flashing damage. Figure 6-8 shows a typical angle valve for flashing service.

The reason this construction is an excellent choice is that flashing damage occurs when high velocity vapor bubbles impinge on the surface of a valve. An angle valve reduces the impingement by directing flow into the center of the downstream pipe, not into the valve body. If damage does occur, the downstream liner can be replaced much more economically than the valve body.

A rotary plug style of valve is also an excellent choice for medium to low pressure flashing applications. This valve can be installed with the plug facing the downstream side of the body (Figure 6-9) so when flashing occurs, it does so downstream of the valve. In some cases, a spool piece of sacrificial pipe is used to absorb the flashing damage.

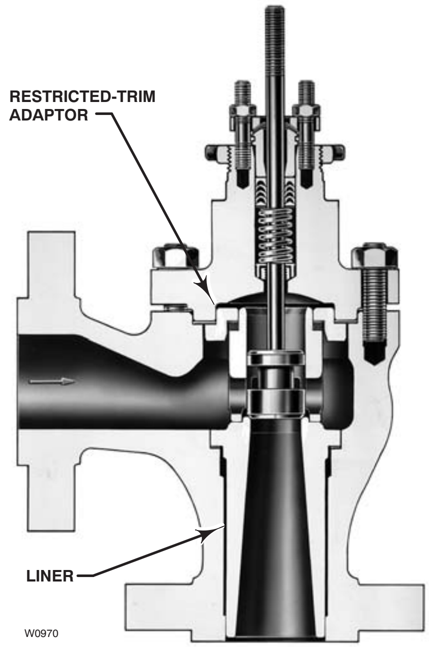

Figure 6-8. *Design EAS valve with outlet liner is used for flashing service. The liner resists erosion and protects the body.*

Materials of Construction

There are several factors which determine the performance of a given material in a particular flashing and/or cavitating situation, including the materials' toughness, hardness, and its corrosion-resistance in the application environment. Within a given material family (for example, the 400-series stainless steels), hardness is a fairly accurate method for ranking materials. However, when comparing materials from different families, hardness doesn't correlate with overall resistance to damage. For example, cobalt-chromium-tungsten based Alloy 6 has much more resistance to cavitation and flashing than either hardened type 410 or 17-4 stainless steels, even though they all exhibit roughly the same hardness. In fact, Alloy 6 equals or exceeds the performance of many materials with hardnesses of 60 HRC and higher. The superior performance of Alloy 6 is attributed to a built-in "energy-absorbing" mechanism shared by a number of cobalt-base alloys. However, Alloy 6 is not without its weaknesses; in many amine-treated feedwater applications, Alloy 6 suffers accelerated attack by an erosion-corrosion mechanism (see "Alloy 6 Corrosion" in Chapter 10 for more information).

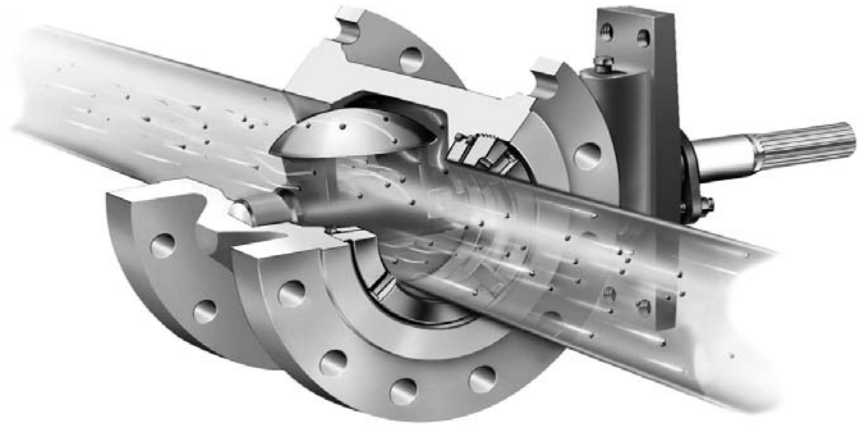

Figure 6-9. *Rotary plug valves, such as the Design V500 (reverse flow trim direction, trim level 3) have excellent erosion resistance and perform well in flashing service.*

Materials commonly used for flashing and cavitating services are Alloy 6 (solid and overlays), nickel-chromium-boron alloys (solid and overlays), hardened 440C stainless steel, hardened 17-4 stainless steel, and hardened 410/416 stainless steel,

Since the standard materials used in valve bodies are relatively soft, selection for caviation and flashing resistance must rely on factors other than hardness. In general, as the chromium and molybdenum contents increase, the resistance to damage by both cavitation and flashing increases. Thus the chromium-molybdenum alloy steels have better resistance than the carbon steels, and the stainless steels have even better resistance than the chromium-molybdenum alloy steels.

In the past, ASME SA217 grade C5 was the most commonly specified chromium-molybdenum alloy steel. However, because of the poor casting, welding, and manufacturing characteristics of C5, ASME SA217 grade WC9 has become a more popular alternative. Experience indicates that WC9 performs on par with C5 in cavitation and flashing services despite its lower chromium content (2-1/4% vs. 5%). This is apparently because its higher molybdenum content (1% vs 1/2%) makes up for the lower chromium content.

ASTM A217 grade C12A is becoming more common in the power industry. This material has excellent high-temperature properties, and is usually used at temperature exceeding 1000°F (538°C). Its higher chromium and molybdenum contents (9% Cr, 1% Mo) would indicate excellent cavitation resistance.

While angle bodies are a better choice for flashing applications than globe bodies, they are also a more economical choice in most cases. The reason is that carbon steel bodies can be used in an angle valve with an optional hardened downstream liner, (17-4PH SST, Alloy 6,etc.) since only the downstream portion of the valve will experience the flashing liquid. See Figure 6-8. If a globe valve is used, it is better to use a chromium-molybdenum alloy steel body since the flashing will occur within the body itself.

System Design

This section discusses system design where it is assumed that flashing will occur. The optimum position of the valve in a flashing service can have a great impact on the success of that valve installation.

Figure 6-10 shows the same application with the exception of the location of the control valve. These figures are fairly representative of a valve that controls flow to a condenser.

In the top illustration, the flashing will occur in the downstream pipe between the control valve and the tank. Any damage that occurs will do so in that area.

In the bottom illustration, the flashing will occur downstream of the valve within the tank.

Since the tank has a very large volume compared to the pipe, high velocity fluid impingement on a material surface will not occur since there is essentially no material surface. This system design will help prevent flashing damage.

Hardware Choices for Cavitating Applications

The design of a control valve greatly affects the ability of a valve to control cavitation. This section discusses the theories behind each type of trim design that is used for cavitation control and also reviews each type of trim that Fisher Controls uses to control cavitation.

The design theories or ideas behind the various trim designs include:

- Tortuous path
- Pressure drop staging
- Expanding flow area
- Drilled hole design
- Characterized cage
- Separation of seating and throttling locations
- Cavitation control in lieu of prevention

Tortuous Paths

Providing a tortuous path for the fluid through the trim is one way to lower the amount of pressure recovery of that trim. Although this tortuous path can be in the form of drilled holes or axial or radial flow passages, the effect of each design is essentially the same. The use of a tortuous path design concept is utilized in virtually every cavitation control style of hardware.

Pressure Drop Staging

This approach to damage control routes flow through several restrictions in series, as opposed to a single restriction. Each restriction dissipates a certain amount of available energy and presents a lower inlet pressure to the next stage.

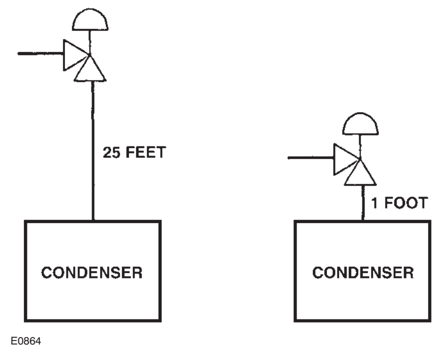

***Figure 6-10**. Location of a control valve can often be changed to lengthen its life or allow use of less expensive products. Mounting a heater drain valve near the condenser is a good example.*

A well-designed pressure-staging device will be able to take a large pressure differential, yet maintain the vena contracta pressure above the vapor pressure of the liquid, which prevents the liquid from cavitating.

For the same pressure differential then, the vena contracta pressure in conventional trim will be lower than for the staged trim, and the liquid will be more prone to cavitate.

Trims that dissipate available energy have an additional advantage. If the design pressure differential is exceeded and cavitation does occur, the intensity will be less. This is because the pressure that causes the collapse of cavities (i.e., the recovered pressure) will be less.

Expanding Flow Areas

The expanding flow area concept of damage control is closely related to the pressure drop staging concept. Figure 6-11 shows a pressure versus distance curve for flow through a series of fixed restrictions where the area of each succeeding restriction is larger than the previous. Notice that the first restriction takes the bulk of the pressure drop and the pressure drop through successive sections decreases.

In the last restriction, where cavitation is most likely to occur, the pressure drop is only a small percentage of the total drop, and the pressure recovery is substantially lowered.

The expanding flow area concept requires fewer pressure drop stages to provide the same cavitation protection as the equal area concept. Since the pressure drop of the last stage is rather low compared to the total pressure drop, if

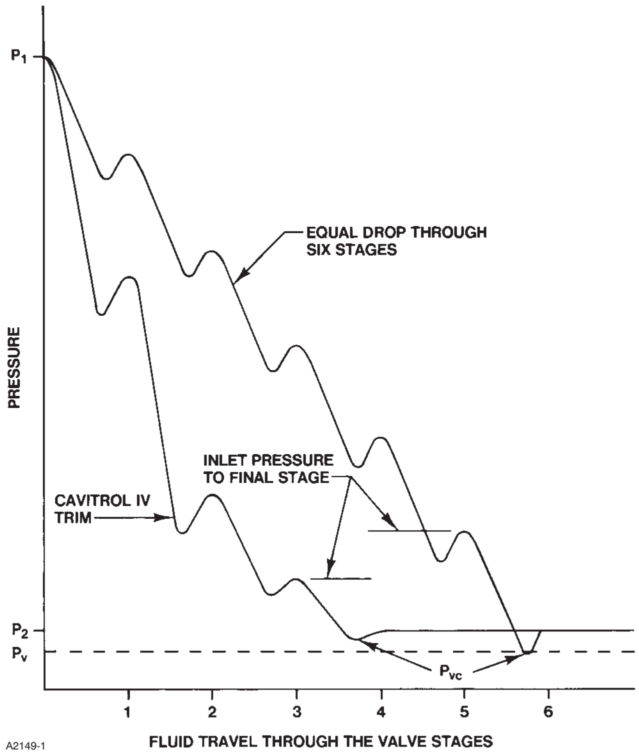

Figure 6-11. *In Cavitrol trim, the pressure drop is staged in two or more unequal steps. Staging is accomplished by increasing the flow area from stage to stage. This stepped reduction allows full pressure drop without the vena contracta pressure falling below the vapor pressure of the liquid.*

cavitation does occur, the intensity and cavitation damage will be much less.

Drilled Hole Design

Fisher Controls uses drilled hole cages to provide a tortuous path, pressure drop staging, and expanding flow area in its Cavitrol® line of cavitation control trims. The design of each particular drilled hole has a significant impact on the overall pressure recovery of the valve design.

Figure 6-12 shows a cross section of three types of drilled holes that could be used in a cavitation control cage. The thin plate design is a very inefficient flow device, but it does provide a high F_L^2 and therefore a low pressure recovery. The thick plate design provides an efficient design, but also provides a high pressure recovery as denoted by a low F_L^2 value.

The Cavitrol trim hole design is a balance between the thick plate and the thin plate hole designs.

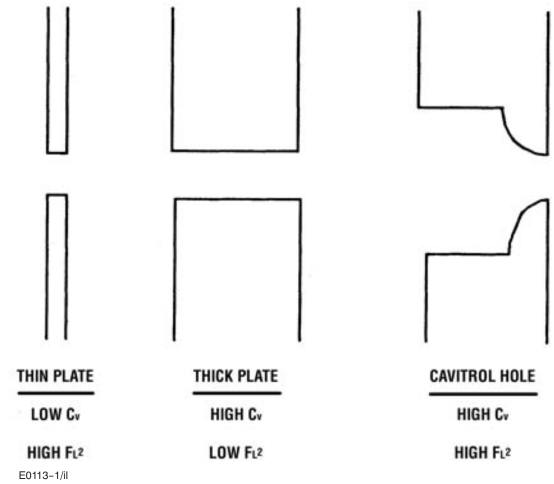

Figure 6-12. *By combining the geometric effects of thick plates and thin plates, it is possible to design a flow passage that optimizes capacity and recovery coefficient values. These carefully designed passages are used exclusively in Cavitrol cages.*

It provides a relatively high flow efficiency while still maintaining a high F_L^2, which results in a low pressure recovery. This design represents the optimal choice between capacity and cavitation control.

Other benefits of this type of drilled hole design is that the vena contracta point is further from the exit of the hole when compared to a straight through drilled hole. Therefore, if pressure recovery above the vapor pressure occurs (cavitation), it will do so further away from the external wall of the cage, and the amount of damage will be smaller.

One disadvantage of cavitation control trims is the potential for flow passages to become plugged. The flowing media often times contains small particulate such as sand that can plug the passages, restricting or totally stopping flow through the valve. If this potential exists, the particles must be removed from the flow stream, usually by filtration, or an alternative approach to cavitation should be taken.

An alternative is to use a trim that is designed to allow the particulate to pass, but still control cavitation. The Fisher Dirty Service Trim (DST) has been designed to allow particles up to ¾" to be passed and to control cavitation up to pressure drops of 4000 psi. This trim has been used extensively in feed pump recirculation, feedwater startup, and numerous recycle applications.

Characterized Cages

The characterized cage design theory has evolved from the fact that "capacity is inversely related to a design's ability to prevent cavitation." In those applications where the pressure drop decreases as the flow rate increases, characterized cages can be used to optimize cavitation prevention and capacity.

For a Cavitrol III characterized trim design, as the travel increases, the cage design changes. It begins as a pressure-staging design and then to a straight-through hole design. Consequently, the cavitation control ability of this trim design is greatest at low travels and decreases with increasing valve plug travel.

Care should be taken to employ characterized cages only in applications where the pressure drop decreases as travel increases. For applications such as boiler feed pump recirculation where the pressure drop across the valve is relatively constant, a characterized cage should not be used.

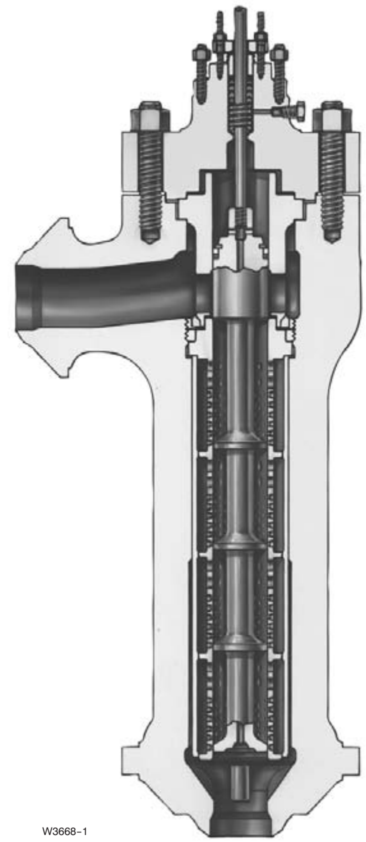

Figure 6-13. *Cavitrol IV trim provides cavitation protection at pressures to 6500 psi. It uses expanding flow areas to affect a four-stage pressure drop. All significant pressure drop is taken downstream of the shutoff seating surface.*

Separate Seating and Throttling Locations

In a modern power plant, most cavitating applications require a control valve to not only provide cavitation control, but also provide tight shutoff. The best way to accomplish this is to separate the throttling location from the seating location as shown in Figure 6-13. The seating surface of the plug is upstream of the throttling location, and the upper cage is designed such that it takes very little pressure drop. The seating surface experiences relatively low flow velocities since velocity is inversely related to pressure. A recent technological advancement has been to implement the use of a softer seating material relative to the material of the plug. This allows for a slight deformation of the seating material, which provides much better plug/seat contact and hence greatly enhanced shutoff capability. Valves utilizing this soft seating material are capable of providing Class VI shutoff.

Cavitation Control Hardware Alternatives

The previous section dealt with the theories behind modern types of cavitation control hardware. This section presents alternatives to the sometimes-costly cavitation hardware. Guidelines are also presented to help determine when cavitation control hardware is required or when other alternatives can be employed.

System Design

Correct liquid system design is the most economical way to prevent the damaging effects that are caused by cavitation without applying cavitation prevention control valves. Unfortunately, even the best system design is likely to need cavitation type control valves, but by applying certain design features, the complexity of these control valves may be simplified.

The most common and oldest method of designing a liquid flow system where large pressure drops must occur is to use a standard trim control valve with a downstream backpressure device. Although these devices come in various sizes, shapes, and designs, they all perform the same function of lowering the pressure drop across the control valve by raising its downstream pressure.

Since the downstream pressure of the valve is increased, the vena contracta pressure is increased. If the backpressure device is sized correctly, the vena contract pressure will not fall below the vapor pressure, and cavitation will not occur.

While this is a simple and cost effective way to prevent cavitation damage in the control valve, there are several serious considerations to look at before using a downstream backpressure device.

- A larger valve may be required to pass the required flow since the pressure drop is lowered.

- Although cavitation may not occur at the control valve, it may occur at the backpressure device.

- The backpressure device can only be sized for one condition. If other conditions exist, the backpressure provided may allow cavitation to occur.

- If the backpressure device becomes worn, the backpressure will decrease and cavitation in the valve may occur.

Another disadvantage that is rarely mentioned occurs when a valve is opened against a high upstream pressure. Until the flow reaches the backpressure device and stabilizes, the valve will experience the entire pressure drop of the system. Although this may only occur for a short period of time, a potential for damage exists.

In the instance of rotary valves, air injection also can be used to minimize the effects of cavitation in a system. With this method, air is injected upstream of the vena contracta. The dispersed air acts as a buffer when the vapor bubbles implode so that the intensity of the cavitation is lowered. Unfortunately, the location of the vena contracta, the amount of air to be injected, etc. are sometimes hard to quantify.

Since air is being injected into the system, this method of cavitation control is usually used on large valves dumping to a tank or pond or where solids in the system prevent the use of other cavitation control devices.

Cavitation is an interesting but destructive phenomenon. Preventing cavitation is the most acceptable way of limiting potential for damage. Proper application of available products based on sizing equations and field experience will provide long term success.

Cavitation Control Summary

- Cavitation is a phenomenon arising from a liquid-vapor-liquid phase change. This phase change occurs when liquids are throttled through control valves.

- Cavitation must be controlled because it potentially damages valves and piping and creates noise and vibration in piping systems.

- Flashing is a related liquid phenomenon arising from similar circumstances. With flashing, however, vapor bubbles remain in the fluid stream rather than collapsing.

- Flashing is a system situation that cannot be prevented by valve selection. Use of designs and materials that resist erosion will generally solve problems related to flashing.

- Either system changes or valve hardware changes can solve cavitation. Prevention and control by use of specialized valve designs is most effective. Material substitutions are helpful, but limited in long term effectiveness.

- Valve recovery coefficients are used along with system and fluid parameters to predict the presence of cavitation and choked flow. Experience gives guidance on sizing and how to avoid problems.

Chapter 7

Steam Conditioning

Introduction

Power producers have an ever-increasing need to improve efficiency, flexibility, and responsiveness in their production operations. Changes resulting from deregulation, privatization, environmental factors, and economics are combining to alter the face of power production worldwide. These factors are affecting the operation of existing power plants and the design of future plants resulting in a myriad of changes in the designs and operating modes of future, as well as existing, power plants.

Competing in today's power market requires heavy emphasis on the ability to throttle back operations during non-peak hours in order to minimize losses associated with power prices falling with demand. These changes are implemented in the form of increased cyclical operation, daily start and stop, and faster ramp rates to assure full load operation at daily peak hours.

Advanced combined cycle plants are now designed with requirements including operating temperatures up to 1500°F, noise limitation in urbanized areas, life extension programs, cogeneration, and the sale of export steam to independent customers. These requirements have to be understood, evaluated, and implemented individually, with a minimum of cost and a maximum of operational flexibility to assure profitable operation.

Great strides have been made to improve heat rates and increase operational thermal efficiency by the precise and coordinated control of the temperature, pressure, and quality of the steam. Most of the steam produced in power and process plants, today, is not at the required conditions for all applications. Thus, some degree of conditioning is warranted in either control of pressure and/or temperature, to protect downstream equipment, or desuperheating, to enhance heat transfer. Therefore, the sizing, selection, and application of the proper desuperheating or steam conditioning systems are critical to the optimum performance of the installation.

Thermodynamics of Steam

Highly superheated steam, (i.e. 900 - 1100°F) is usually generated to do mechanical work such as drive turbines. As the dry steam is expanded through each turbine stage, increasing amounts of thermal energy is transformed into kinetic energy and turns the turbine rotor at the specified speed. In the process, heat is transferred and work is accomplished. The spent steam exits the turbine at greatly reduced pressure and temperature in accordance with the first law of thermodynamics.

This extremely hot vapor may appear to be an excellent source for heat transfer, but in reality it is just the opposite. Utilization of superheated steam for heat transfer processes is very inefficient. It is only when superheated steam temperatures are lowered to values closer to saturation that its heat transfer properties are significantly improved. Analysis has shown that the resultant increase in efficiency will very quickly pay for the additional desuperheating equipment that is required.

In order to understand why desuperheating is so essential for optimization of heat transfer and efficiency, we must examine the thermodynamic relationship of temperature and the enthalpy of water. Figure 7-1 illustrates the changes of state that occur in water over a range of temperatures, at constant pressure, and relates them to the enthalpy or thermal energy of the fluid.

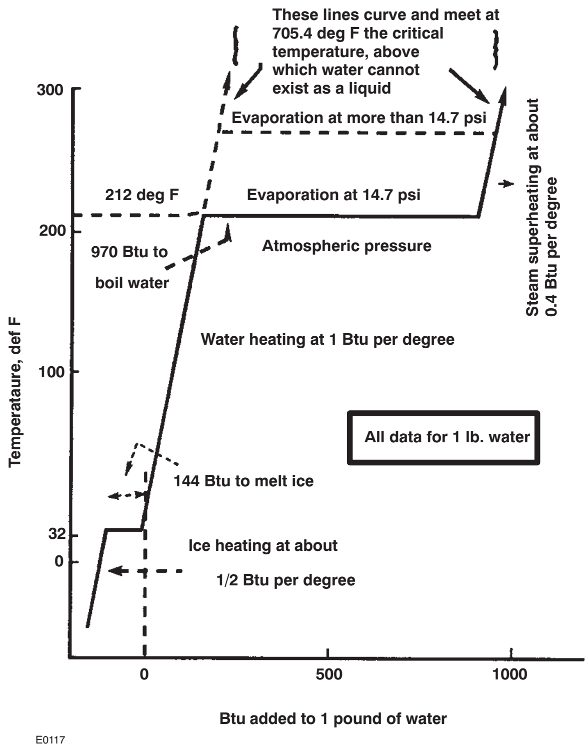

Figure 7-1. Temperature enthalpy diagram for water. Note that the greatest amount of thermal energy input is used to vaporize the water. Maximum efficiency in heat transfer requires operation at near saturation temperature to recover this energy.

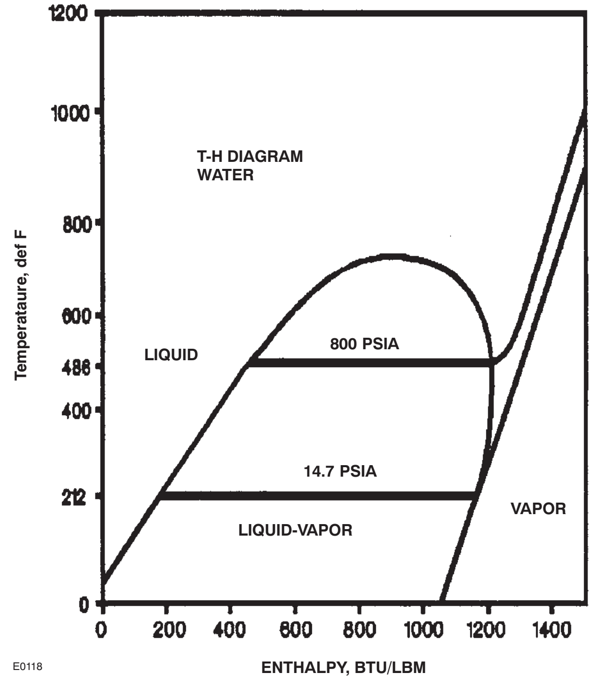

Figure 7-2. Temperature enthalpy diagram for water showing that saturation temperature varies with pressure. By choosing an appropriate pressure, both correct system temperature and thermal efficiency can be accommodated.

In the lower left portion of the graph, the water is frozen, at atmospheric pressure and below 32°F. At this point, heat is being rejected from the water as it maintains its solid state. As heat is gradually added, the ice begins to change. Addition of heat to the ice raises the temperature and slows the rate of heat rejection. It requires approximately 1/2 BTU of thermal energy to be added to a pound of ice to raise its temperature 1°F. Upon reaching 32°F, the addition of more heat does not immediately result in an increase in temperature. Additional heat at this point begins to melt the ice and results in a transformation of state from a solid to a liquid. A total of 144 BTUs is required to melt one pound of ice and change it to water at 32°F.

Once the phase change from a solid to a liquid is complete, the addition of more heat energy to the water will again raise its temperature. One BTU of heat is required to raise the temperature of one pound of water by 1°F. This relationship remains proportionate until the boiling point (212°F) is reached. At this point, the further addition of heat energy will not increase the temperature of the water. This is called the saturated liquid stage.

The water begins once again to change state, in this case from water to steam. The complete evaporation of the water requires the addition of 970 BTUs per pound. This is referred to as the latent heat of vaporization and is different at each individual pressure level. During the vaporization process, the liquid and vapor states co-exist at constant temperature and pressure. Once all the water, or liquid phase, has been eliminated, we now have one pound of steam at 212°F. This is called the saturated vapor stage. The addition of further thermal energy to the steam will now again increase the temperature. This process is known as superheating. To superheat one pound of steam 1°F requires the addition of approximately .4 BTUs of thermal energy.

The potential thermal energy release resulting from a steam temperature change differs significantly depending on temperature and superheat condition. It is much more efficient, on a mass basis, to cool by addition of ice rather than by the addition of cold fluids. Similarly, it is more efficient to heat with steam at temperatures near the saturation temperature rather than in the superheated region. In the saturated region, much more heat is liberated per degree of temperature change than in the superheated range because production of condensate liberates the enthalpy of evaporation, the major component of the total thermal energy content. The temperature-enthalpy diagram in Figure 7-2 is generalized to show the thermodynamic relationship at various pressures.

The graph in Figure 7-2 illustrates three distinct phases (i.e., liquid, vapor, and liquid-vapor) and how enthalpy relates to temperature in each phase at constant pressure. The rounded section in the middle of the graph is called the "steam dome" and encompasses the two-phase, liquid-vapor region. The left boundary of the steam dome is called the saturated liquid line. The right boundary line is the saturated vapor line. The two boundaries meet at a point at the top of the dome called the critical point. Above this point, 3206 psi and 705°F, liquid water will flash directly to dry steam without undergoing a two-phase coexistence. When conditions exceed this critical point they are considered to be existing in the supercritical state.

In the lower left side of the graph, the saturated liquid line intersects the temperature axis at 32°F. At this point we have water and a defined enthalpy of 0 BTU/LB. As heat is added to the system, the temperature and enthalpy rise and we progress up the saturated liquid line. Water boils at 212°F at 14.7 psia. Thus, at 212°F and 180 BTU/LB, we note a deviation from the saturated liquid line. The water has begun to boil and enter a new phase; Liquid-Vapor.

As long as the liquid stays in contact with the vapor, the temperature will remain constant as more heat is added. At 1150 BTU/LB (at 14.7 psi) we break through to the saturated vapor line. Thus, after inputting 970 BTU/LB all of the water has been vaporized and enters the pure vapor state. As more heat is added, the temperature rises very quickly as the steam becomes superheated.

Why Desuperheat?

Desuperheating, or attemperation as it is sometimes called, is most often used to:

- Improve the efficiency of thermal transfer in heat exchangers
- Reduce or control superheated steam temperatures that might otherwise be harmful to equipment, process or product
- Control temperature and flow with load variation

As mentioned earlier, dry superheated steam is ideally suited for mechanical work. It can be readily converted to kinetic energy to drive turbines, compressors and fans. However, as the steep temperature-enthalpy line slope would indicate, the amount of heat output per unit of temperature drop is very small. A heat exchanger using superheated steam would have to be extremely large, use great quantities of steam, or take tremendous temperature drops. A 10°F drop in temperature liberates only 4.7 BTU per pound.

If this same steam had been desuperheated to near saturation the thermal capabilities would be greatly enhanced. The same 10°F drop in temperature would result in the release of over 976 BTU of heat. This illustrates the obvious advantages of desuperheating when thermal processes are involved. Only by desuperheating the superheated steam is it possible to economically retrieve the energy associated with vaporization. By changing steam pressure, the saturation temperature can be moved to match the temperature needs of the process and still gain the thermal benefits of operating near saturation.

The previous discussion centered on why we superheat steam (to do mechanical work) and when it should be desuperheated back to saturation (to heat.) There are many situations when saturated steam suddenly and unintentionally acquires more superheat than the downstream process was designed to accommodate. This "unintentional" superheat produces the same thermal inefficiencies mentioned previously. In this case, we are talking about the sudden expansion and temperature change associated with a pressure reducing valve. Take the following steam header conditions for example:

Conditions: $P1 = 165$ psia
$T1 = 370$°F
Enthalpy = 1198.9 BTU/LB

Saturation temperature at 165 psia is 366°F. Therefore, the steam has only 4°F of superheat and would be excellent for heat transfer. Assume that another thermal process requires some steam, but at 45 psia rather than 165 psia. The simple solution is to install a pressure reducing valve. Since throttling devices, such as valves and orifices, are isenthalpic (constant enthalpy processes) the total heat content of the steam will not change as flow passes through the restriction.

After the valve, the steam will have the following conditions:

Conditions: P2 = 45 psia
Enthalpy = 1198.9 BTU/LB

Referencing a set of steam tables, we see that at the above conditions the steam temperature is 328°F giving the impression that it has cooled. However, from the steam tables we see that the saturation temperature for 45 psia steam has also dropped to 274°F. The net result is that our steam now has 54°F of superheat (328°F - 274°F). Use of this steam for heat transfer could be uneconomical and return on investment on a desuperheater would be most favorable.

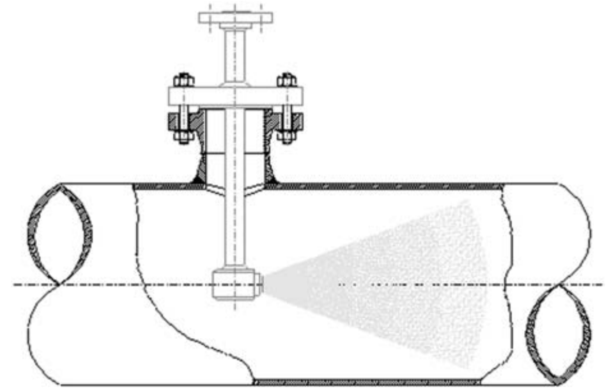

Figure 7-3. *Insertion style desuperheater injects a controlled amount of cooling water into superheated steam flow.*

Desuperheating

In this section, we will briefly discuss the process of desuperheating. The need to desuperheat is usually performed simply to control the steam temperature, or heat content, of the flowing vapor media. Depending on the process downstream of the main steam source, a desuperheater will be utilized to transform the steam into a medium that is more efficient for heat transfer or just more conducive for interaction with its surrounding components. One means of accomplishing this is with a direct contact heat transfer mechanism. This can easily be achieved by the use of a single spray injection nozzle that, when properly placed, diffuses a calculated quantity of liquid into the turbulent flow stream. Vaporization of the liquid phase proceeds while mass, momentum, and energy transfer occurs and the resultant vapor exits the process at the desired temperature or heat content level.

Desuperheaters

A desuperheater is a device that injects a controlled amount of cooling water into a superheated steam flow in an effort to reduce or control steam temperature (Figure 7-3). Desuperheaters come in various physical configurations and spray types that optimize performance within specified control and installation parameters. Selection should also always include attention to those details that would provide the most economic solution without sacrificing required performance.

The success of a particular desuperheater station can rest on a number of physical, thermal, and geometric factors. Some of the factors are quite obvious and others are more obscure, but they all have a varying impact on the performance of the equipment and the system that it is installed in. Considerable research has been conducted into the characteristics of desuperheaters and the transformation of spraywater to vapor. The findings are of considerable interest to both design and process engineers. In the next several sections, we will discuss these findings and how they relate to the desuperheating system as a whole.

The most important factor is the selection of the correct desuperheater type for the respective application. Units come in all shapes and sizes and use various energy transfer and mechanical techniques to achieve the desired performance criteria and optimize the utilization of the system environment. These design criteria include:

- Mechanically Atomized – Fixed & Variable Geometry Spray Orifice
- Geometrically Enhanced
- Externally Energized

The mechanically atomized style of desuperheater is the most popular and simplistic style that provides nominal performance over a wide range of flow and conditions. These models are of the internally energized variety, (Figures 7-4A & 7-4B). The atomization and injection of the spray water is initiated by the pressure differential between the spraywater and the steam. The

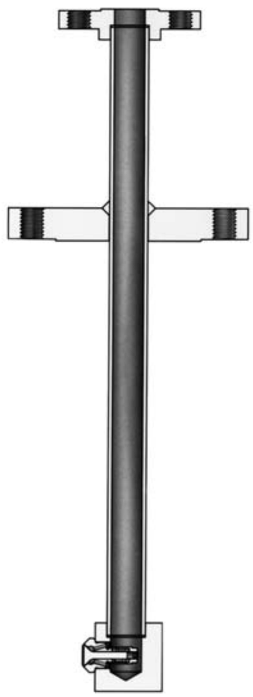

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Figure 7-4A. *Design DMA/AF desuperheater utilizes variable-geometry, back-pressure activated spray nozzles.*

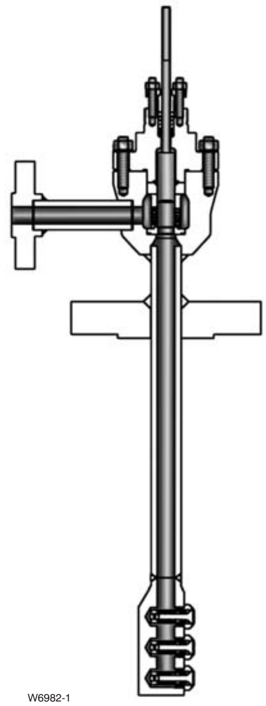

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Figure 7-4B. *Design DVG/AF variable-geometry, mechanically atomized, self-contained desuperheater for moderate to high flow variation.*

Design DMA, fixed geometry spray orifice, units are the simplest and by design have a constant area flow path. These units are highly dependent on the pressure differential and thus provide levels of performance that are commensurate with the magnitude of the difference. Obviously, the larger the water/steam differential the better the unit will perform (i.e., penetration velocity, flow variation) and droplet size. Since the equipment turndown is usually limited to 4:1, it is best suited for near steady load applications.

An upgrade from the fixed geometry unit is the DMA/AF variable geometry nozzle desuperheater. Here the actual flow geometry of the unit is varied to maintain an optimum differential across the discharge orifice. As a result of this change, the level of flow variation is greatly enhanced and so is the performance. Equipment turndowns can exceed 40:1, thus making this style a good choice for medium to high load change applications. This unit can be used with an external control valve or be supplied with an integral flow regulating trim and actuating system in the form of the Design DVG/AF desuperheater.

Another form of mechanically atomized desuperheater is the Design DVI, Geometric Enhanced style, (Figure 7-4C). Here, the unit is

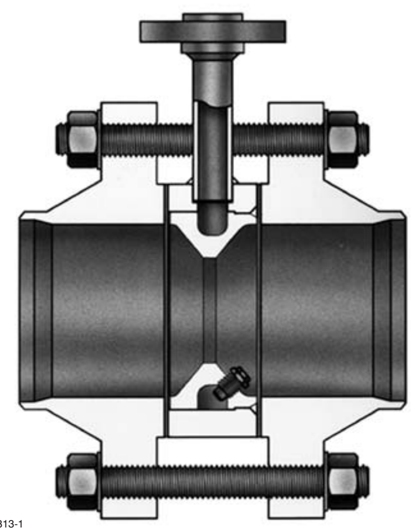

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Figure 7-4C. *Design DVI desuperheater injects spraywater in the outlet of the venturi section, assuring excellent mixing and rapid atomization.*

supplied a high pressure recovery flow restriction that alters flow geometry and helps to keep the level of turbulence and kinetic energy at a high level during all phases of the units operation due to an increased velocity at the point of spray water injection. This increased level of surrounding energy helps to impart energy transfer to the droplets and assists in break-up, mixing, and vaporization. This style is best suited for medium turndown applications typically around 15:1.

The last group of desuperheater units utilizes an external energy source for the atomization of the spraywater. The most common medium is a high pressure steam source. In this case, the high levels of kinetic energy are provided by a critical pressure reduction in the desuperheater sprayhead. The critical drop is used to ablate and shear the water into a fine mist of small droplets, which is ideal for vaporization, as shown in Figure 7-4D. This type of system can provide a very high degree of flow variation without requiring a high pressure water supply. Applications requiring turndown ranges greater then 40:1 utilize this type of equipment for best performance. In addition to an external spraywater control valve, the system will also require an atomizing steam shut-off valve (Figure 7-5).

Other factors that have a large amount of impact on the performance of a desuperheating system include:

- Installation Orientation
- Spray Water Temperature
- Spray Water Quantity
- Pipeline Size
- Equipment vs. System Turndown

Installation orientation is often overlooked, but a critical factor in the performance of the system. Correct placement of the desuperheater can have more impact on the operation than the style of the unit itself. For most units, the optimum orientation is in a vertical pipeline with the flow direction up. This flow direction is ideal, as the natural flow direction of the injected water tends to be in the counter direction due to effect of gravity. The role of gravity in this orientation will suspend the droplets in the flow longer while they are being evaporated, thus shortening the required downstream distance or efficient mixing.
Other orientation factors that are of concern include downstream pipefittings, elbows, and any other type of pipeline obstruction that can provide a point for water impingement or fallout.

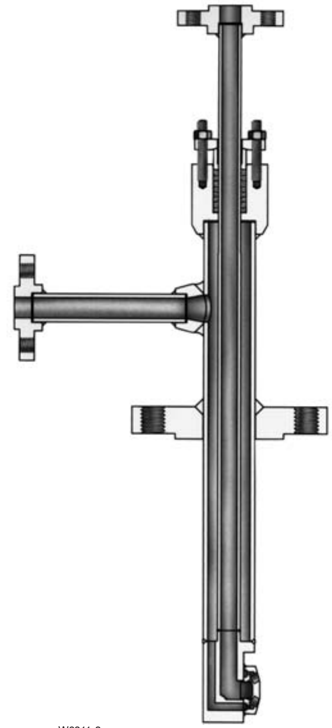

Figure 7-4D. *Design DSA desuperheater uses high-pressure steam for rapid and complete atomization of spraywater in low-velocity steam lines.*

Spraywater temperature can have an great impact on the desuperheater performance. While it goes against logical convention, hotter water is better for cooling. As the temperature increases and moves closer to saturation, its flow and thermal characteristics are improved and impact most significantly the following:

- Surface Tension
- Drop Size Distribution
- Latent Heat of Vaporization
- Vaporization Rate

Improvement in all these areas will act to improve the overall performance of the system, as the spraywater will evaporate and mix with the steam at a faster rate.

The quantity of water to be injected will, as with any mass flow calculation, have a directly proportionate affect on the time for vaporization.

The heat transfer process is time dependent; thus, the quantity of spray water will increase the time for complete vaporization and thermal stability.

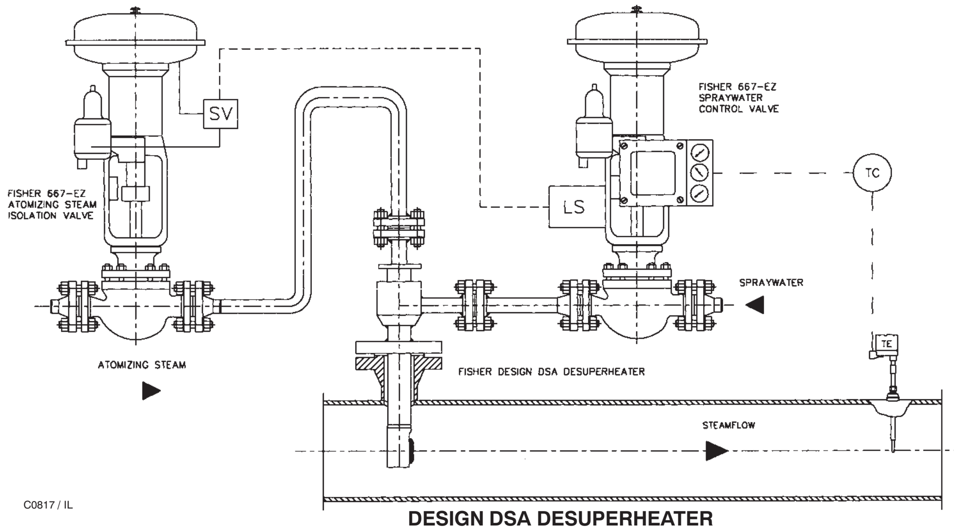

Figure 7-5. *Design DSA desuperheater utilizes two external control valves: a spraywater unit and an atomizing steam valve.*

Another concern for proper system performance is pipeline size. Pipe size should be determined in an effort to balance the velocity of the steam flow. Steam traveling at a fast rate will require longer distances to effectively cool, as heat transfer is a function of time. Steam traveling at low velocity will not have enough momentum to suspend water droplets long enough for evaporation. As a result, water will fall out of the steam flow to collect along the bottom of the pipe, and it will not cool the steam effectively. Ideal velocity is typically in the range of 250 ft/sec to 300 ft/sec.

As the pipeline size increases to limit steam velocity, more attention must be paid to the penetration velocity of the spray and the coverage in the flow stream. Experience shows that single point injection type desuperheaters will have insufficient nozzle energy to disperse throughout the entire cross-sectional flow area of the pipeline. As a result, the spray pattern collapses and thermal stratification occurs (i.e., sub-cooled center core within a superheated outer jacket.)

This condition normally is eliminated after the flow stream undergoes several direction changes, although this is not always possible within the limits of the control system or process. Proper placement of high-energy Design TBX-T (Figure 7-6), multi-nozzle steam coolers in the larger pipelines will normally prevent thermal stratification.

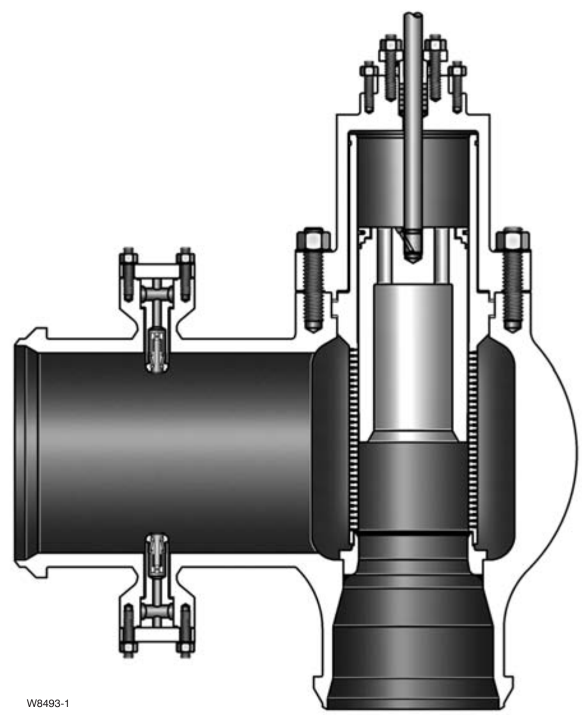

Figure 7-6. *Design TBX-T utilizes an external spraywater manifold with multiple nozzles for moderate to large volume applications.*

The most overly used and misunderstood word in the field of desuperheating is “turndown.” When applied to a final control element, such as a valve, it is a simple ratio of the maximum to minimum controllable flow rate. Turndown is sometimes used interchangeably with rangeability; however, the exact meaning differs considerably when it comes to actual performance comparisons.

Since a desuperheater is not a final control element, its performance is linked directly to its system environment. Therefore, the actual turndown is more a function of system parameters rather than based on the equipment's empirical flow variations. Once this is understood, it is obvious that even a good desuperheater cannot overcome the limitations of a poorly designed system.

A final design parameter for an insertion type desuperheater is its ability to withstand high levels of thermal cycling. Due to the nature of operation of today's plants, desuperheaters should be designed to operate under daily cycling environments. Exposure to frequent daily cycling can lead to thermal fatigue and desuperheater failure if the unit isn't designed for the operation. Design upgrades for this application consist of thermal liners to reduce thermal loads and structural optimization to reduce induced vibration at stress sensitive welds.

To summarize the requirements to size a desuperheater correctly, the following system and operating information is required:

- Minimum & Maximum Steam Flow
- Steam Pressure & Temperatures
- Cooling Water Pressure & Temperature
- Required System Turndown Ratio
- Pipe Size & System Layout
- Planned Mode of Operation

Steam Conditioning Valves

Steam conditioning valves represent state-of-the-art control of steam pressure and temperature by integrally combining both functions within one control element unit. These valves address the need for better control of steam conditions brought on by increased energy costs and more rigorous plant operation. Steam conditioning valves also provide better temperature control, improved noise abatement, and require fewer piping and installation restrictions than the equivalent desuperheater and pressure reduction station.

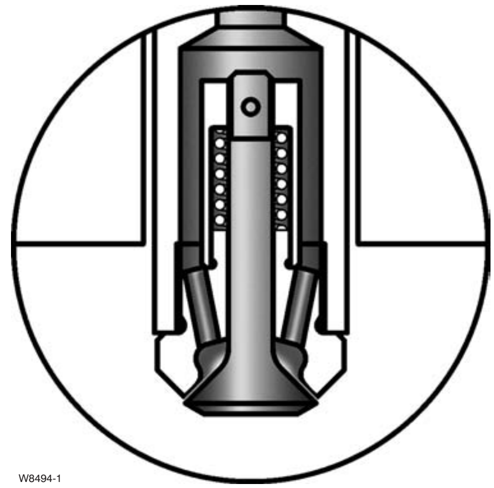

Figure 7-7. *Detail of Type AF Spray Nozzle.*

Steam conditioning valve designs can vary considerably, as do the applications they are required to handle. Each has particular characteristics or options that yield efficient operation over a wide range of conditions and customer specified requirements.

The TBX steam-conditioning valve (Figure 7-6) combines pressure and temperature control in a single valve. Finite element analysis (FEA) and computational fluid dynamic (CFD) tools were used in its development to optimize the valve's operating performance and overall reliability. The rugged design of the TBX proves capable of handling full mainstream pressure drops, while its flow-up configuration in conjunction with Whisper Trim technology prevents the generation of excessive noise and vibration.

The simplified trim configuration used in the TBX accommodates rapid changes in temperature, as experienced during a turbine trip. The cage is casehardened for maximum life and is allowed to expand during thermally induced excursions. The valve plug is continuously guided and utilizes cobalt-based overlays both as guide bands and to provide tight, metal-to-metal shutoff against the seat.

The TBX incorporates a spraywater manifold downstream of its pressure reduction stage. The manifold features variable geometry, backpressure activated Type AF nozzles that maximize mixing and quick vaporization of the spraywater.

Type AF nozzle (Figure 7-7) was developed originally for condenser dump systems in which the downstream steam pressure can fall below the saturation level. In these instances, the spraywater may flash and significantly change the flow characteristic and capacity of the associated nozzle at a critical point in the operation.

Spring loading of the valve plug within the AF nozzle prevents any such changes by forcing the plug to close when flashing occurs. With flashing, the compressibility of the fluid changes, and the nozzle spring will force closure and re-pressurization of the fluid leg. Once this is done, the fluid will regain its liquid properties and reestablish flow to the condenser.

The TBX injects the spray water towards the center of the pipeline and away from the pipe wall. The number of injection points varies by application. With high differentials in steam pressure, the outlet size of the valve increases drastically to accommodate the larger specific volumes. Correspondingly, an increased number of nozzles are arranged around the circumference of the outlet, making for a more even and complete distribution of the spray water.

The simplified trim arrangement in the TBX permits extending its use to higher pressure classes (through ANSI Class 2500) and operating temperatures. Its balanced plug configuration provides Class V shutoff and a linear flow characteristic.

The TBX typically uses high-performance, pneumatic piston actuators in combination with FIELDVUE Digital Valve Controllers to achieve full stroke in less than two seconds while maintaining highly accurate step response. The FIELDVUE instruments along with AMS ValveLink® software provide a self-diagnostic capability that gives answers about valve performance. The current valve/actuator signature (seat load, friction, etc.) can be compared against previously stored signatures to identify performance changes before they cause process control problems.

When piping dictates, the TBX valve can be provided as separate components, allowing pressure control in the valve body and temperature reduction in a downstream steam cooler. The steam cooler (Figure 7- 8) is equipped with a water supply manifold (multiple manifolds are possible also.) The manifold provides cooling water flow to a number of individual spray nozzles that are installed in the pipe wall of the cooler section. The result is a fine spray injected radially into the high turbulence of the axial steam flow.

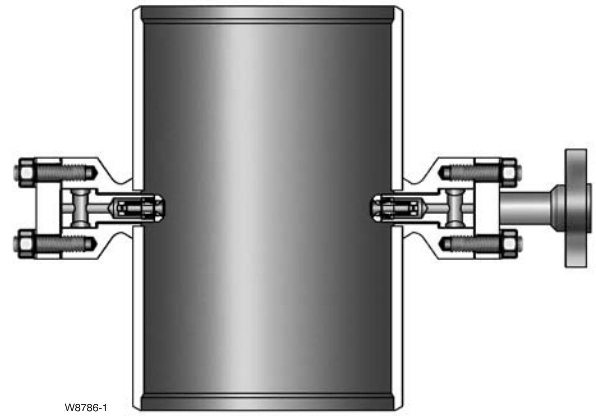

Figure 7-8. *Design TBX-T Cooler.*

Installation Guidelines

Installation of desuperheaters and steam conditioning valves is key to long term success and performance. It is best to install desuperheaters in a straight run of horizontal or vertical pipe. Installation in elbows is also possible, but it can affect system turndown and thermal stratification due to momentum caused changes in the velocity profile.

Momentum forces the majority of the steam flow to the outside surfaces of the bend. This results in a low velocity void on the inside of the elbow. If high turndowns are not required, this installation is satisfactory since the voids would rarely be below minimum velocity at maximum flow. As the flow is reduced, however, these areas may lose their ability to perform as required to desuperheat the steam.

Other installation parameters that are always of interest to the piping designer are how much straight run of pipe is required and where the temperature sensor should be located. Both are thermally derived questions and require thermally derived answers. It is desirable to have the thermal sensor as close as possible to the desuperheater in order to reduce the signal lag time. It is also desirable not to have any piping components (e.g., elbows or tees) that would detract from the thermal process.

The following equations provide guidelines for designing a proper system. These equations relate to time required for complete vaporization and mixing.

Downstream Straight Pipe Requirements (SPR):

SPR (ft) = 0.1 Sec. x Maximum Steam Velocity (ft/sec)

Downstream Temperature Sensor Distance (TS):

15% Spraywater or less:

TS (ft) = 0.2 Sec. x Maximum Steam Velocity (ft/sec)

Greater than 15% Spraywater:

TS (ft) = 0.3 Sec. x Maximum Steam Velocity (ft/sec)

Temperature control is not limited to receiving a signal from a downstream temperature sensor. Another valid alternative is feed-forward control. Feedforward control is accomplished using an algorithm that is characterized specifically to the valve installed in the application. The algorithm is programmed into the distributed control system to provide an accurate calculation of the spray water that is required to reduce the steam enthalpy and temperature to the desired outlet set point. The algorithm requires input of upstream temperature and pressure as well as the position of the valve. Upstream and spraywater enthalpies are then determined using an inherent steam table within the DCS. The total spraywater required is calculated from a heat balance using the final enthalpy into the condenser. This method of temperature control is a practical solution for applications that do not have enough downstream pipe distance for accurate measurement by a temperature sensor.

Turbine Bypass Systems

The most severe and critical application of any steam conditioning installation is that of the turbine bypass.

The concept of the turbine bypass has been around for a long time; however, its application and importance has broadened significantly in recent years. Steam turbine bypass systems have become essential to today's power plant performance, availability, responsiveness, and major component protection.

The following will concentrate on the general application of bypass systems as used in fossil fueled utility power plants. The closed water/steam heat cycle of such typical units may be comprised, but not limited to, sub- or super-critical pressures, to single, double, or triple reheat sections and to condensation at or near ambient temperatures. The steam generating principles where such bypass systems are employed include natural or assisted circulation drum boilers, combined circulation boilers, and once-through boilers. The turbine may be of single or double shaft design and operated either at fixed inlet pressure or on sliding pressure.

Bypass System Benefits

Just how beneficial a bypass system proves to be depends upon many factors (e.g., plant size, mode of operation, age of existing components, size of the condenser, main fuel type, control philosophy, etc.) However, the main benefits for the application of a comprehensive bypass system in the 25-100% size range are:

- **The matching of steam and heavy turbine metal component temperatures during the startup and shutdown phase.** This has proven to be of major economic significance in terms of fuel savings and the thermal protection of critical heavy wall boiler and turbine components. By limiting temperature differentials during turbine admission, the effects of thermal fatigue are minimized and longevity of components maximized. This is especially important for life extension programs were the role and justification of the bypass system may be centered solely on this aspect.

- **The ability to avoid a boiler trip following a full load rejection.** A boiler (HRSG) / turbine unit with a bypass can withstand a complete system load rejection and remain available for rapid reloading after the disturbance has been removed. This important advantage for system flexibility and operating efficiency can make the difference between a more costly and time consuming warm start and a hot start.

- **Reduction in solid-particle erosion of turbine components.** The loss of material from the boiler tubing and internals is most prevalent during commissioning startup and after the unit has been shutdown for an expended period of time. Thermal transients assist in the dislodging of scale, oxides, and weldments within the boiler circuit to form an abrasive steam flow that, over time, could accelerate the wear of sensitive turbine blades and seriously affect operating efficiencies and maintenance costs. Damage can be reduced or eliminated by routing the steam through the bypass system.

- **Independent operation of the boiler and turbine set.** The ability to operate the boiler without the turbine, at any load up to the limit of the bypass capacity, can be surprisingly useful for operational or testing purposes. For example, all boiler controls and firing systems can be tested and fine-tuned independent of the turbine operation. This significantly reduces both cost and time relating to initial commissioning of the plant, retrofitting and checking equipment performance, and system troubleshooting.

General System Description

A complete and comprehensive turbine bypass system can be comprised of many inter-linked and coordinated components. These include the bypass valves, spray water control valves, control system, and the actuation and positioning system. For this discussion, we will center our attention on the bypass valves themselves.

The bypass system incorporates the dual operating function of steam conditioning valves (i.e., for the controlled reduction of both pressure and temperature.) The bypass valve incorporates the latest technology in pressure-reducing/low noise trim to handle the flow and reduction of pressure energy to acceptable levels. However, since steam throttling in a control valve is an isenthalpic process, desuperheating is required to control the discharge temperature and enthalpy levels. As a result, the valves are equipped with a special spraywater injection system that produces a finely atomized and evenly distributed water interface for rapid vaporization and steam temperature control.

The bypass system can be supplied with one or two control inputs depending on the role it plays in the control scheme. If the valve is used solely for startup and shutdown, it will receive a single modulating control signal to position the trim as a function of the startup and shutdown curves for the respective unit. If the valve must also act to relieve pressure during a turbine trip or load rejection, an additional discrete input is included that will ramp open the valve quickly to a predetermined position, before reverting to a modulating configuration in accordance with the boiler control requirements. Fast positioning speed and resultant alternate flow path are critical to counteract the pressure build-up resulting from the isolation of the boiler piping circuit when the turbine valves close in this trip situation.

High Pressure Bypass

During startup, shutdown, or on turbine trip, the HP bypass system directs steam from the superheater outlet to the cold reheat line, thereby bypassing the HP turbine section (Figure 7-9). The major advantages of such an action have been generally outlined above. However, more specific duties are:

1. Pressure & temperature controlled bypass of the HP turbine section.
2. Controlled main steam pressure build-up in the boiler.
3. Cooling of the reheat section of the boiler.
4. Prevention of the opening of spring-loaded HP safety valves during minor disturbances.
5. Avoidance of condensate loss and noise from blowing safety valves.
6. Protection of the boiler against exceeding design pressures.

The failure mode of the HP bypass system is very dependent on local design codes and the performance scenario for the system. If it is designed as a safety bypass system and replaces the standard safety relief valve function, the valves must always fail in the open position. However, if the standard safety relief valves are in place, the valve is normally required to fail closed, especially in over-temperature situations on drum boilers.

Control of the HP bypass is normally initiated via feedback input signals from the main steam pressure and the cold reheat temperature. The ratio of steam to spraywater is normally inversely proportional to the respective valve position, especially during startup and shutdown. This is because startup conditions normally require large valve Cvs, due to the large specific volumes associated with low pressures at high temperatures, even though flow is greatly reduced.

During trip conditions, the opposite is true, and large quantities of spraywater are required at lower valve openings. For this situation, special control algorithms usually are incorporated into the control system to provide independent feedforward control. This is especially important during a trip sequence where time of response is critical to maintain system integrity, performance, and component protection.

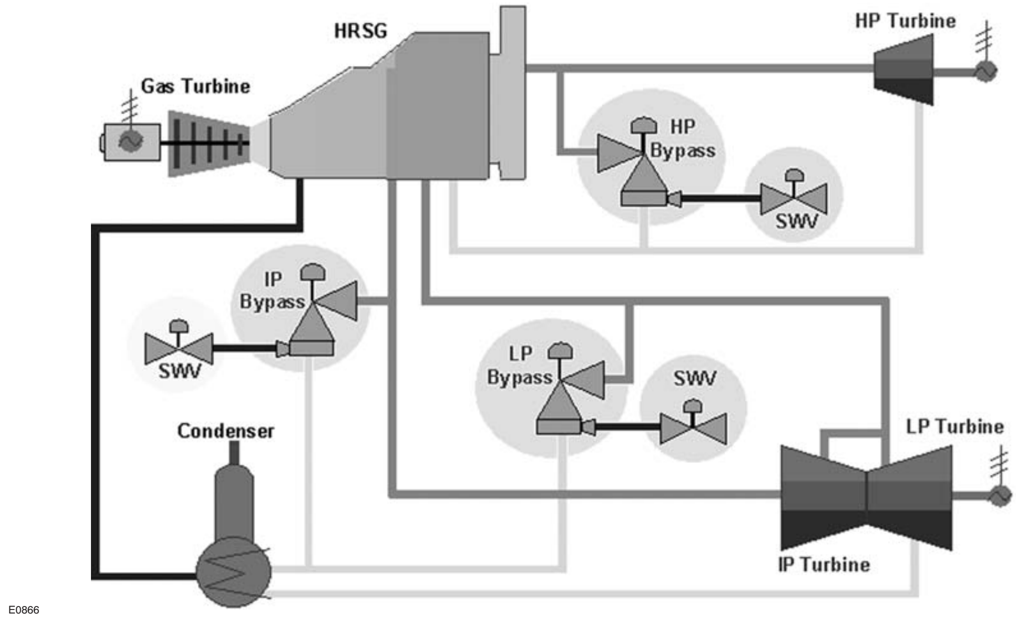

Figure 7-9.

Spraywater for cooling is normally obtained from the boiler feed pump discharge and is regulated by an external spraywater control valve that is properly sized to handle the required flow and pressure drop.

Hot-Reheat & Low Pressure Bypass

During startup, shutdown, or on turbine trip, the HRH and LP bypass systems direct steam from the hot reheat line to the condenser, thus bypassing the IP and LP turbine sections (Figure 7-9). The major advantages of such an action have been generally outlined above. However, more specific duties are:

1. Pressure & temperature controlled bypassing of the IP and LP turbines.
2. Controlling pressure build-up in the boiler reheat section.
3. Prevention of condensate losses during load. trips and minor disturbances.
4. Protecting the condenser against excessive pressure, temperature, and enthalpy excursions during bypass operation.

In contrast to the HP bypass, the HRH and LP bypass valves only have closed as a failure mode. While it is important to control the hot reheat pressure, it is even more critical to protect the condenser against damage from uncontrolled or improper admission of steam. The condenser manufacturer interfaces specific condenser control permissives with these bypass control systems. If any of these permissives is not met or is exceeded during bypass operation, the valve is quickly shut. These permissives include, but are not limited to:

1. Condensate level high
2. Condenser temperature high
3. Condenser pressure high
4. Spraywater pressure low
5. Loss of coolant

Another added challenge of the HRH and LP bypass system is to properly control the amount of backpressure on the bypass valves. A condenser or condenser duct which is downstream of these bypass valves typically operates at a vacuum in the range of 1 - 3 psia. Given this scenario, it is crucial to create backpressure in order to maintain a desired velocity within reasonable pipe sizes.

A second challenge to this application is to create these desired conditions while minimizing the noise generated by this process. Dumping high velocity steam into a low pressure, thin wall condenser/turbine exhaust duct requires careful evaluation in order to assure steam jets do not converge. Hole spacing within the sparger and sparger placement within the duct are critical for maintaining low noise levels.

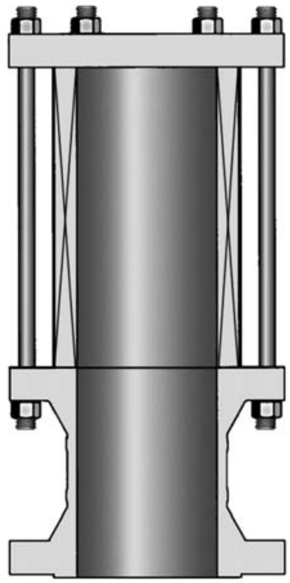

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Figure 7-10. *Design TBX WhisperFlo Sparger.*

A typical bypass to condenser installation requires a steam conditioning valve to control pressure and temperature, a spraywater valve to regulate the water supply, and a downstream TBX sparger to create backpressure. Low noise WhisperFlo trim alternatives are also available for the TBX sparger (Figure 7-10).

Control of the HRH and LP bypass valves normally is initiated via feedback input signals from the hot reheat steam pressure and the specified condenser inlet temperature/enthalpy. The steam entering the condenser must be controlled specifically to guard against excessive thermal expansion of the tubing and shell. As in the case of the HP bypass, the ratio of steam to spray water is normally inversely proportional, especially during startup and shutdown. In addition, the dual role of the HRH and LP bypass system in controlling the thermal admission parameters to the condenser normally results in the requirement for a prescribed amount of over-spray.

This situation is compounded by the close proximity of these valves to the condenser. This makes any kind of feedback temperature control almost impossible considering the quantity of spray water to be vaporized and the short distance available to measure the process. It is highly recommended that feedforward control algorithms be incorporated into the control system to provide independent feedforward control for the spraywater admission.

Spraywater for cooling is normally obtained from the condensate boost pump discharge and is regulated by a properly sized external spraywater control valve.

Bypass Size

A comprehensive bypass system includes HP bypass, HRH bypass and LP bypass valves. However, they may or may not be sized for the same capacity. There are many variables that can influence the required size of each bypass system.

Bypasses for once-through boiler plants are generally designed for 100% of full-load steam to suit startup and part-load operation. If conventional safety valves are omitted, 100% bypass capacity is essential.

Bypass capacity for drum boiler plants involve several different issues. Some argue that 100% capacity bypasses are worthwhile, but experience has proven that bypasses with capacities of between 25 - 70% normally are sufficient to handle most operating and trip conditions.

For temperature matching in a drum plant during hot startup only, it may be possible to use a bypass of only 30% when firing with oil and 40-50% for coal. Overall, these values are considered the lowest practical load for the boiler under automatic control.

On bypass applications requiring the control of a full turbine trip, the values increase to 40% on gas and oil-fired drum units and up to 70% for coal. In selecting the bypass capacity, it is important to consider all control systems and plant components and their ability to turn down instantaneously from full to auxiliary load.

Note also that if the high pressure bypass capacity exceeds approximately 50%, and the low pressure bypass passes all the steam to the condenser, then condenser duty during bypass operation is more severe than during normal, full-load turbine operation. This fact may limit bypass capacity, especially on systems being retrofit to existing plants.

Starts, Trips, Load Rejection, Two-Shift Operation

The worth of a turbine bypass and the flexibility, added efficiency, and responsiveness are never more apparent than during starts, trips, or load rejections. Modern bypass systems operate during:

- Cold starts
- Warm starts
- Hot starts
- Load rejection
- Quick turbine shutdown
- Two-shift operation

Bypass valves and systems that are designed correctly have noteworthy advantages for these individual modes. They are detailed as follows:

Cold Starts A cold start typically occurs after the unit has been down for over a week. Preheating of the system is required as first stage and reheat temperatures are normally below 200°F. The bypass system permits involvement of the furnace, superheaters, and reheater very early in the steam/water cycle.

This is important in the production of steam purity before the turbine start. Steam flow through the superheater and reheater enhances the tube cooling effect, thereby allowing greater latitude in gas and steam temperatures. During the startup, thermal stresses are controlled while achieving the fastest possible loading rate. Depending on the size of the bypass system, the unit can typically be brought on line in 4.5 - 9 hours.

Warm Starts A warm start is indicative of a weekend shutdown. In this case, the HP turbine casing is usually above 450°F. As with the cold start, the steam temperature can be controlled to permit the matching of steam and metal temperatures under all operating conditions. Expected startup time is between 2.5 - 5 hours.

Hot Starts A hot start is usually associated with a minor disturbance that created a unit trip. The bypass allows the boiler to remain on line until the disturbance is cleared and the unit can be reloaded in the shortest possible time, which is usually between 1 - 2 hours.

Load Rejection/Quick Restart During load rejection, the bypass system provides the necessary control and flow path for unit runback to minimum load and for the establishment of a definitive course of action (i.e., complete shutdown or quick restart.) All systems are protected, and a minimum of condensate is lost.

Two-Shift Operation Two-shift operation may become necessary if a utility grid has a number of large base-loaded units, which are not as maneuverable as the smaller fossil fueled units used for peaking purposes. This would require that the smaller units be shutdown every night and restarted every morning, which is a very material-life consuming means of operating. Once again, the bypass system provides a means for the efficient and timely matching of steam and metal temperatures. This allows the efficient startup of the units every morning without thermally stressing the components, yet it increases unit efficiency and availability.

Chapter 7 — Steam Conditioning Summary

The implementation of a properly designed turbine bypass system can be beneficial and instrumental in the pursuit of increased efficiency, flexibility, and responsiveness in the utility power plant. Component life can be extended as the ability to regulate temperatures between the steam and turbine metal is enhanced. Commissioning time and cost can be reduced through independent boiler and turbine operation. The magnitude of return on investment hinges on the specific application mode, style or service of plant, and equipment supplied. While not discussed here, this logic applies as well to combined cycle plants, cogeneration facilities, and industrial power facilities.

Short Notes:

- A desuperheater is a device that sprays a precisely controlled amount of water into a steam line to modify steam temperature.

- System parameters and required turndown are the most influential parameters in desuperheater selection.

- Desuperheating is done primarily to improve efficiency of thermal transfer devices and to provide temperature protection for process, product and equipment.

- Another reason to desuperheat is to control the "unintentional superheat" created by pressure reduction valves.

- Proper installation is key to best performance. Guidelines for piping geometry and placement of downstream temperature sensors are available.

- Steam conditioning is the process of combining pressure reduction and desuperheating into a single control element.

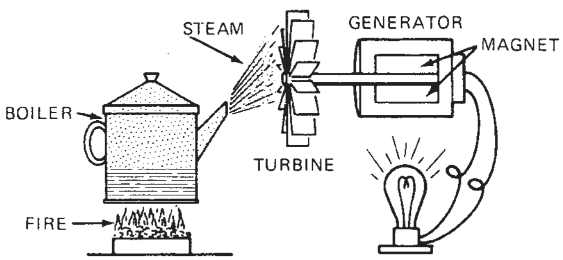
- Turbine bypass systems are beneficial and instrumental for achieving high efficiency, flexibility, and responsiveness in today's power plants.

Chapter 8

Power Plant Primer

A steam power plant is a means for converting the potential chemical energy of fuel into electrical energy. In its simplest form, it consists of a boiler and a turbine driving an electric generator.

The boiler is a device for turning water into steam. The steam jet issuing from the spout spins the fan (turbine) and also the generator. In the sketch the very simplest kind of boiler and turbine are shown. The boiler is a tea kettle, and the turbine is nothing more than a little windmill. Actual turbines are more complicated but the principle is the same.



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A word about the generator shown in the sketch. To most people the process of generating electricity is very mysterious, yet the actual process is easy to understand. As shown, the generator consists of a little bar magnet spinning inside a stationary coil or wire. This may seem an absurdly simple affair, yet that is exactly what a real generator consists of -- a magnet rotating inside of a coil of wire. As the magnetic field issuing from the ends of the magnet moves across the turns of wire in the stationary coil an electric current is set up in the wire. By winding a large number of turns of wire into a ring or doughnut, the current set up in each turn is added to the current set up in the other turns of wire, and so a much more powerful current is produced.

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This is all you need to know about an electric generator now--just think of it as a rapidly rotating magnet inside of a coil of wire. This produces an electric current in the wire. Later, we will elaborate on this simple description.

You may wonder, if a power plant is basically as simple as this, why we build the complex plants we see described in POWER ENGINEERING Magazine? The answer is quite simple: the plant shown in the sketch is not very efficient. Indeed, its efficiency is close to zero. Since we want to get as much power as possible out of a given quantity of fuel, it is necessary to make our plants as efficient as possible.

Until the early 1920s, the electric power plants of the nation used over three pounds of good coal to produce a kilowatt-hour of electricity. Today, the national average is less than one pound of coal per kilowatt-hour. In other words, plants built at that time used three times as much coal to produce a kilowatt-hour as we use today. In 1985, the electric utilities of this country produced over 1401 billion kilowatt hours by means of coal-fired plants. This required the burning of 693 million tons of coal. If, however, we had had to produce this amount of electrical energy by means of the type of plants we had 70 years ago, we would have needed three times the coal, or over two billion tons. In 1985, coal-fired plants supplied 64% of total steam generated electricity. Oil-fired plants supplied 5%, gas-fired 13% and nuclear 18%.

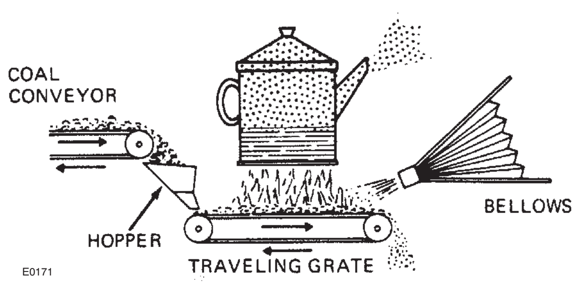
The reason for the great decrease in the consumption of coal lies in the gradual improvement of our power systems, both with respect to the individual pieces of equipment and in the system as a whole. Just how do we go about improving the system shown in the sketch?

Boiler Components

Looking at it again, it is obvious that it can be broken down into several divisions.

First, there is the fire under the boiler. This involves not only the fuel itself, but also the method of placing the fuel under the boiler and the arrangement for burning it properly.

So let us extend the diagram to look like this:



Here, we see a belt conveyor transporting coal to the furnace where it is burned on a traveling grate stoker. A bellows supplies air for combustion.

Remember, when you burn coal you are really promoting a chemical reaction--a chain reaction. When coal is heated to a high enough temperature in the presence of air, the carbon in the coal combines with the oxygen of the air to form either carbon dioxide (CO_2) or carbon monoxide (CO). These, of course, are both gases. Which gas is formed depends upon the quantity of oxygen present. The CO means that the coal is only partially burned, indeed, the CO can be combined with more oxygen to form CO_2.

In burning coal we do not want CO because that means the coal is only partially burned: there is still energy left in the gas, energy that we can recover if we can burn it to CO_2. It is desirable, in the operation of our boiler furnaces, always to get as much CO_2 as we can because in that way we get all the heat out of the fuel.

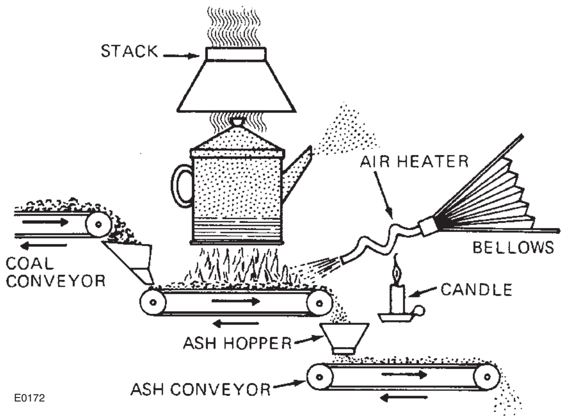
We can obtain the CO_2 by supplying more air to the fuel as it is burning. But we do not want to supply too much air, because if we do we will be supplying more oxygen than is actually needed to combine with the carbon, and this excess oxygen will play no part in the combustion process. Not only will it play no part but it will actually detract from the efficiency by absorbing heat that otherwise could be used to heat the water in the boiler. In actual practice it is not possible to supply exactly the required amount of air, so somewhat more than enough is supplied. This is commonly referred to as excess air.

So, in the process of combustion, we are dealing with chemistry. It involves knowledge of the composition of the coal, its physical condition, its behavior under various conditions of temperature, moisture, etc. Actually, the combustion of coal is a very complex process requiring a good knowledge of both physics and chemistry. In a large plant it involves a major problem in materials handling--fuel, ashes, air and flue gas. Remember, to burn coal, you have to supply about 11 pounds of dry air for each pound of dry coal used. Because of widely varying coal compositions, and allowing for excess air and moisture, the actual amount required is usually somewhat more than this.

So far, we have merely mentioned ashes and flue gas. These have to be removed continuously. In the days of hand firing the removal of ash was simple, though laborious. The fireman merely raked the ash out of the ashpit and carried it away in wheelbarrows. Today, in large plants, the removal of ash is a complicated process requiring rather elaborate equipment. So, we must add ash removal equipment to our diagram; also a chimney for the removal of flue gas.

Furthermore, the process of combustion is stimulated by heat; indeed, the process will not start until the fuel is brought to the kindling temperature. Everything must be done, therefore, to maintain a high temperature in the furnace. This makes it desirable to heat the air for combustion before it is delivered to the furnace. This aids combustion and increases the efficiency.

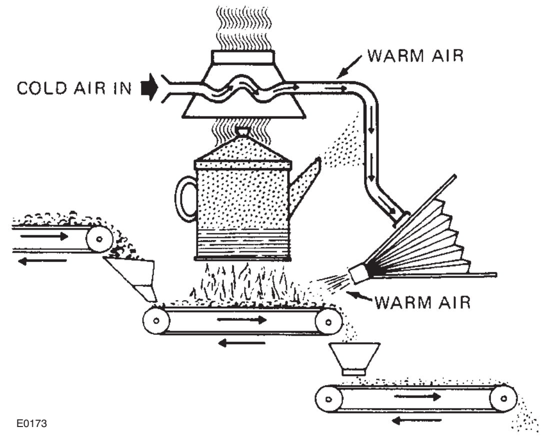
With these facts in mind, let us redraw our diagram to incorporate these improvements. This way:



As you see, it is becoming more complicated. Now, we have a chimney or stack to remove the gas of combustion and a heater to heat the air from the bellows before it is blown into the furnace. Also, we have put in an ash conveyor.

Now, if you are of an analytical mind, you will see that it takes additional heat to heat the air. We have shown a candle. Since candles cost money, it is obvious that we are not going to save much money that way. Why can't we use some of the heat from the fire under the boiler itself to heat the air? Maybe that would be cheaper than buying candles?

Well, it is, and moreover, there is heat going to waste up the stack. You know from experience that if you hold your hand above a tea kettle on a stove that there is a lot of heat being wasted. So, let us put a heating coil in the stack so the bellows will be blowing warm air into the fire as shown in the sketch.



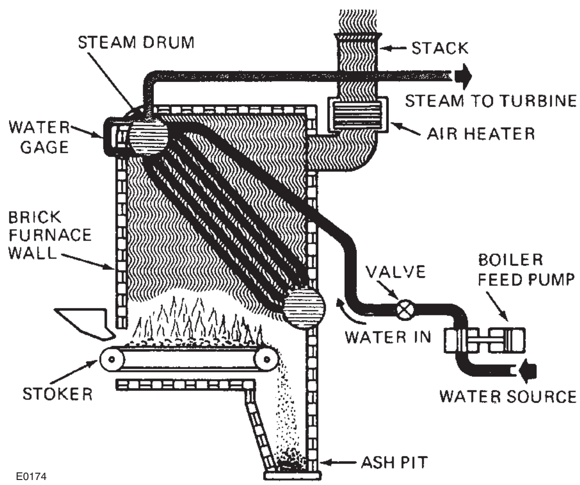
All this probably seems absurdly simple, and it is. The reason for explaining it in this way, however, is to show, by means of the simplest kind of equipment, how engineers go about improving the efficiency of any system. Step by step, adding something here, saving something there, establishing closer supervision over everything gradually improves the effectiveness and the efficiency of almost any kind of system. These are the kinds of things engineers are concerned with. Nearly all of them are more or less complex and require a great deal of specific as well as general knowledge.

In the example just described, for example, just how much surface should the heater in the stack have to heat the air to a certain temperature? How hot should the air be for best combustion, how much air should be supplied, how much power will it take to run the bellows, what happens to the flue gases if too much heat is extracted from the gases, what happens to the stack? None of these is a foolish question.

Consider the last question, for example, that of cooling the flue gases too much. What happens? Well, there is always a certain amount of water vapor in the flue gases; from the air and from the hydrogen in the fuel. As the temperature of the gases is lowered, there comes a time when the saturation point is reached and the moisture condenses. If, at the same time, there happens to be any sulfur in the gas (and there usually is), sulfuric acid will be formed, resulting in the spread of a thin but extremely corrosive layer of liquid on the inside surfaces of the flues.

This single example, then, shows what the engineer runs into when he begins to refine the simple system we began with. He may add something to improve it but he may find that the improvement is not an unmixed blessing; it may also have deleterious effects. These he must guard against.

But let's get on with our power system. The tea kettle representing the boiler is not a very efficient generator of steam. Let's see if we cannot design a better one.

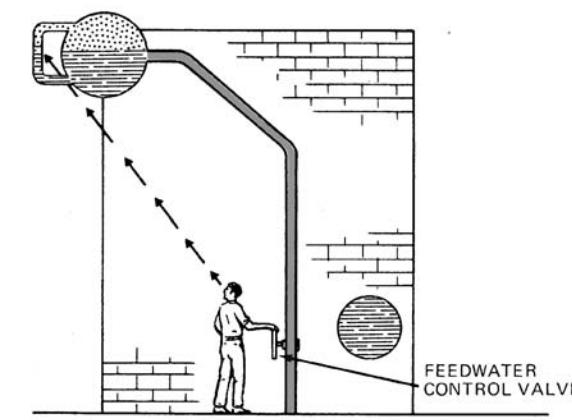


Look at the above sketch. Here we have a boiler consisting of two steel drums connected by a number of steel tubes, and arranged in a furnace so that the hot gases have to pass through the bank of tubes on their way to the stack. The total surface of the tubes is large, making it possible to absorb a great deal of heat. The steam bubbles formed in the tubes rise to the upper drum (called the steam drum) where the steam collects before it flows into the pipe leading to the turbine.

This is the basic principle of the modern water tube boiler. Note that a boiler feed pump has been included in the diagram. Since steam flows out of the boiler it obviously becomes necessary to replenish the water that is evaporated. For this reason a boiler feed pump is necessary. This pump must operate at a pressure high enough so that it can overcome the pressure in the boiler.

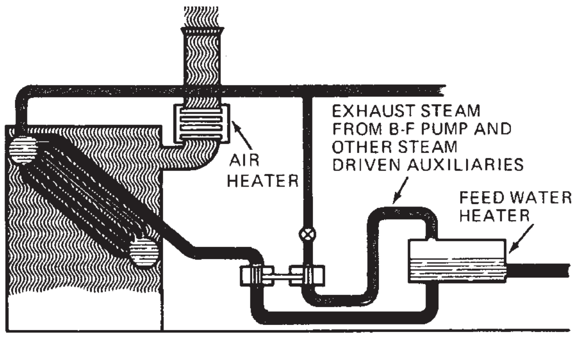
In the operation of any boiler, even a tea kettle, it is essential always to keep enough water in the boiler If it should run dry the metal would become red hot, soften, and rupture. At the same time it should not be filled to a point where there is no room for the steam to collect. To check the water level a water gage is fitted to the steam drum. This shows the water level in the boiler at all times.

Now, it is obvious that if the amount of steam leaving the boiler was always the same as the amount of water entering, the water level would remain the same. This state of affairs seldom occurs. Variations in load causes variations in steam flow; variations in the fuel supply and air supply cause variations in the rate of combustion, which in turn results in variations in the rate of evaporation, and all of these cause changes in the water level. This makes it necessary for the operator to maintain a continual watch on the water level. If it drops, he increases the water supply; if it rises, he decreases it.



This constant vigilance on the part of operators was a problem, so feedwater regulators were developed which control the flow automatically as the water in the boiler drum rises and falls. These are very helpful even with small boilers, but in the case of modern high-pressure boilers they are almost imperative. A large high-pressure boiler, evaporating in the neighborhood of a million pounds of water per hour, would run dry in about 90 seconds if the water supply was suddenly cut off.

Now, getting back to our water supply to the boiler, so far nothing has been said about the temperature of the water being delivered to the boiler. It should be obvious, however, that it would not be wise to pump cold water into a boiler since that would decrease the temperature of the water already in the boiler. While this reduces the rate at which steam was being made, it might also set up strains in the boiler by virtue of the great temperature difference. Therefore, it becomes expedient to heat the water before pumping it into the boiler; so it is run through a feedwater heater.



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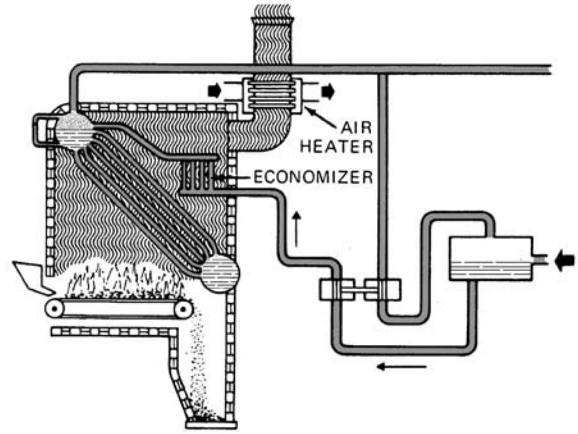
A separate fuel-fired furnace could heat the feedwater heater, but, as in the case of the air heater, it would be much more economical if it could be heated by heat that would otherwise be wasted.

Suppose, for example, that the boiler feed pump was a steam driven pump, and we could use the exhaust steam from this pump. This would cost virtually nothing. So, we add a heater, as shown at left.

After the steam has done its work driving the pump, it is delivered to the feedwater heater, which, as shown here, is nothing more than a large tank in which the steam mixes directly with the water to be heated. It is called an open heater As will be shown later, there are other kinds of feedwater heaters called closed heaters in which the steam and water do not mix.

So far, so good; we have saved a little by using the heat in exhaust steam which otherwise would have been wasted. Let's look a bit further, however; maybe we can save some more heat somewhere else. In the boiler, for example, remember that we picked up some heat in the flue gases by means of the air heater Have we got all of it?

An air heater may or may not remove all the heat we want to remove from the flue gases. But if we want to, we can modify the boiler and install one section of tubes to heat the feedwater before it is delivered to the boiler drum. Look at this sketch:



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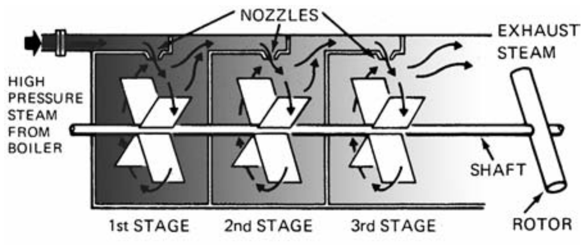
Here we have added a separate bank of tubes through which the feedwater passes before it goes into the boiler drum. This bank of tubes is placed in the path of the gases traveling towards the air heater and the stack. Most of the heat in these gases has been absorbed in the boiler tubes, but not all of it. The gases probably still have a temperature of about 600 degrees Fahrenheit. By making them travel through this added tube bank, still more of the heat will be absorbed.

Consequently, the economy of the boiler as a whole will be increased. Hence, this bank of tubes is known as an economizer. With this arrangement, the water is first heated to a temperature of about 212°F in the feedwater heater by the exhaust steam from the feed pump, and then, in the economizer, the temperature is further raised to a point not very far below the temperature of the water in the boiler

Steam Turbine and Condenser

Our boiler - or steam generator, as it is called - has now become quite complex, so before we do anything more to it let's hook it up to the rest of the system.

As already mentioned, a turbine is essentially a windmill; not a simple fan like the one shown but a more complex one with many hundreds of blades, some stationary and some rotating. These blades are arranged in groups or stages, so that the steam is compelled to pass successively through the various stages. Here is a very simple diagram.



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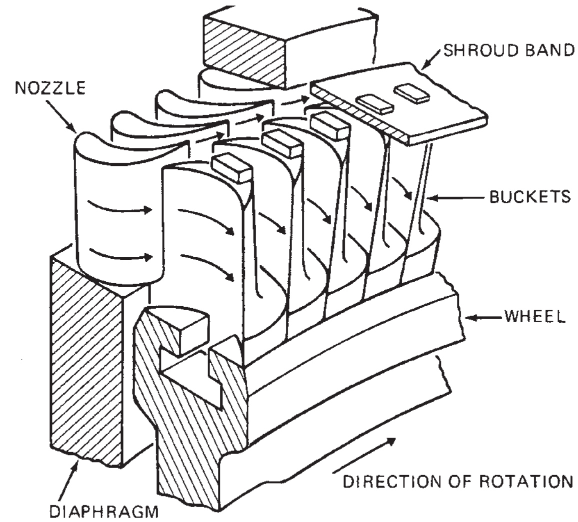
In this diagram, three fans are shown mounted on a common shaft, each one in a separate compartment.

Steam issuing from the nozzle in the first stage pushes against the fan blades and causes the entire rotor assembly to turn. In turning the blades in the first stage the steam gives up some of its energy, resulting in a drop in pressure. Thus, at a slightly lower pressure, it enters the second stage nozzle and again it gives up some of its energy in turning the second stage of this rotor. After passing through the third stage in this way, practically all of the energy of the steam has been given up to the rotor, and it leaves the turbine as exhaust steam.

This arrangement, obviously, provides a much more efficient means for spinning the rotor shaft than the simple little fan shown in the first diagram. Of course, it is still merely a diagram-no turbine would ever be built this way. The only step remaining to make this elemental turbine into a commercial machine is to introduce multiple nozzles of proper design and change the shape of the inefficient paddles to an efficient blade having curved entrances and exits. Here is a detailed drawing of the nozzles and blades (they are also called buckets) of a modern turbine.



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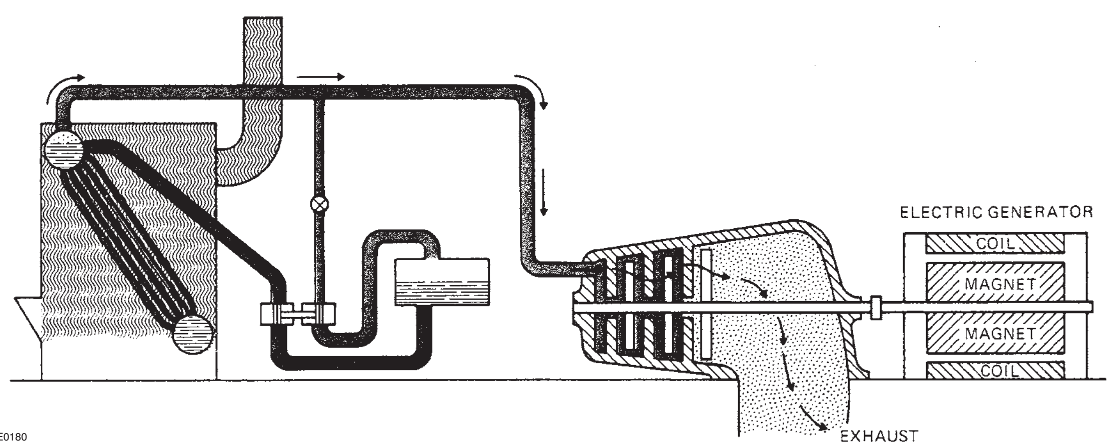
Here you see steam from the boiler being fed into the turbine by means of the connecting piping and after passing through the various stages of the turbine the steam exhausts through an opening in the bottom of the turbine. The steam has given up its energy to the turbine rotor and this in turn spins the generator rotor The generator rotor, remember, is simply a magnet.

But what shall we do with the exhaust steam? Is it of any use? Well, if you measure the temperature of the exhaust steam right at the point of the exhaust opening, you will find that it has a temperature of 212°F. This is the temperature of steam at atmospheric pressure.

Obviously, we can use it to heat the water in the feedwater heater in the same way that we used the exhaust steam from the boiler feed pump. However, you would find that there is far more



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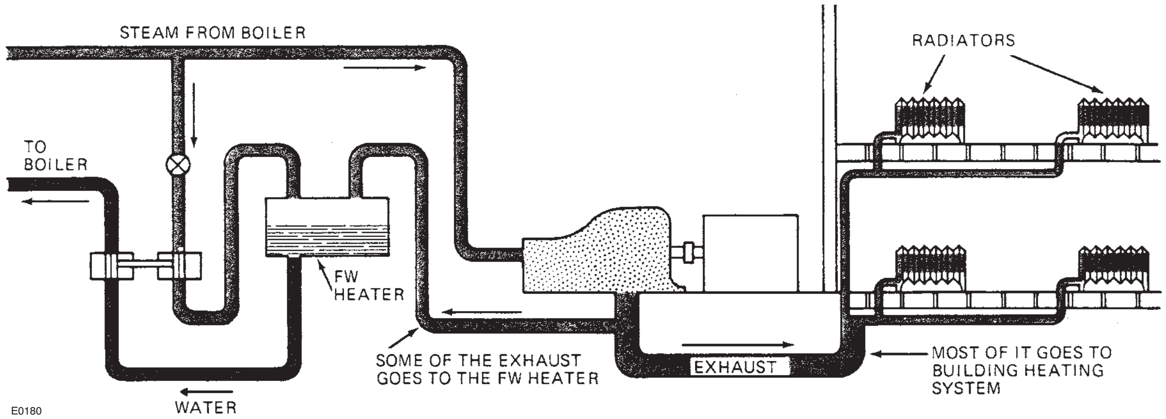


exhaust steam coming from the turbine than you could use in the feedwater heater Remember, practically the entire boiler output passes through the turbine and out into the exhaust.

If you want, you can pipe this exhaust steam to radiators and use it to heat houses and buildings in winter.

Note that part of the exhaust steam goes to the feedwater heater Most of it, however, goes to the building heating system.

Now, this is fine if you have a building you want to heat or if you need steam heat for other purposes such as cooking, heating stills in food or chemical plants or any of scores of different purposes in industry. In practice, that is how a great deal of exhaust steam is used. Indeed, this is one of the reasons it pays the owners of an industrial plant, a paper mill for example, or a textile mill or a food products plant, to have their own power plant. They can use practically all of the exhaust steam from the turbine for heating and other purposes.

If they did not have the turbine, they still would need a boiler to generate steam. By first running the steam through a turbine they can get the electric power so produced for a very low cost.

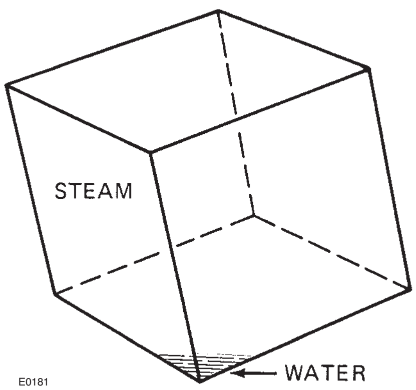
Suppose, however, there is no building to be heated or no factory process to use up the exhaust steam; what then? Take a public utility plant for example. A public utility plant is designed solely to generate electricity to sell. Such plants usually are far away from buildings where the exhaust steam might be used for heating. Should the exhaust steam be permitted to be wasted to the atmosphere?

No. There is a much better way of disposing of this exhaust steam. That is by using it to create a vacuum at the exhaust end of the turbine.

A vacuum? Why a vacuum? What good would that do? Well, remember that the turbine is surrounded on all sides by the atmosphere which, at sea level, exerts a pressure of about 15 lb. per square inch (psi). In order to get out of the exhaust opening, the steam has to push against this 15 psi pressure, and this requires work, just as it does to push the turbine blades around. But suppose, by some means we could remove the atmosphere from around the exhaust opening so that the steam issuing from the exhaust opening would encounter no resistance whatever. You would find that you could develop more power in the turbine; indeed, it would be equivalent to an increase in steam pressure.

To understand how we can accomplish this (removal of the atmospheric pressure) it is necessary to know a few facts about steam. Steam, remember, is evaporated water When water is heated to a temperature of 212°F at atmospheric pressure it turns into steam. If the water is enclosed in a tightly closed vessel such as a boiler, the temperature at which the water turns into steam will be higher. In any case. Also, the volume of the steam produced will be very much larger than the volume of the water from which it was produced. At atmospheric pressure, for example, a pound of steam occupies a volume of 26 cubic feet.

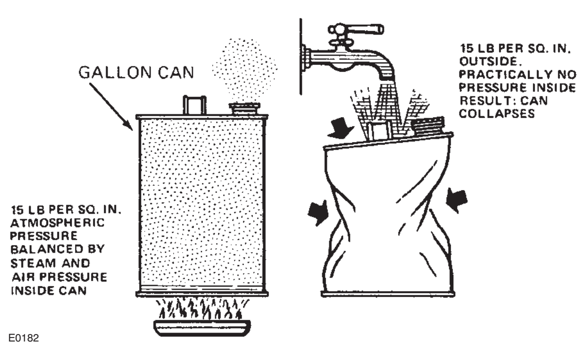
Suppose, now, that you had a pound of steam at atmospheric pressure in a closed vessel with a volume of exactly 26 cu ft. This vessel would be a trifle less than 3 feet on a side, assuming it to be a cube. It would be full of steam. There would be no air. If you suddenly placed this vessel on a large block of ice, or cooled it by spraying cold water on it, what would happen? The steam would condense-it would turn back into water--into one pound of water This pound of water, however, would occupy only 1/60th of a cubic foot. It would look about like this.



This is very little water Most of the interior is now occupied by nothing-99.93% of the total volume. This means a vacuum.

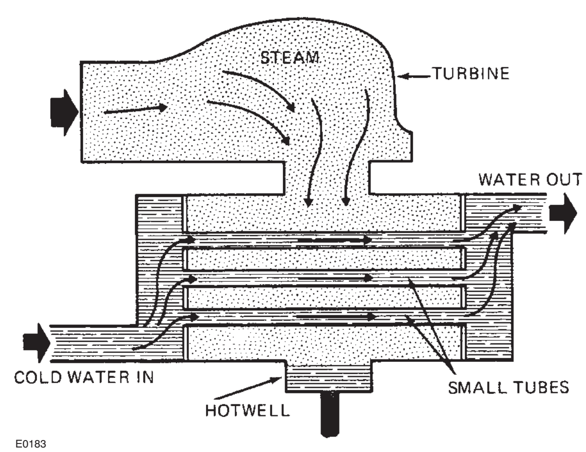
The total surface of this cube has an area of 7776 sq. in. Since each square inch has 15 lb. of atmosphere pressing down on it (and with nothing inside to counteract it) the total atmospheric pressure on the cube is now 7776 x 15 or about 116,640 lb.

If you want to see whether this is really true, try it sometime. Take an ordinary rectangular gallon can with a screw cap closure, pour in about a half-inch of water, and bring the water to a boil by placing it on a gas burner for a few minutes. Do this with the screw cap off. Then, when the water is boiling vigorously, screw the cap on and quickly place the can under a stream of cold water. The can will crumple up like so much paper.



This spectacular experiment is one that anybody can do at home but it is extremely convincing in demonstrating the production of a vacuum by the condensation of steam.

Remember, we wanted to create a vacuum at the exhaust end of the turbine. Now that we know how to create a vacuum, just how can we apply the principle to our turbine? Well, suppose we attach a large hollow vessel to the exhaust opening of the turbine and install a bank of small tubes in the vessel through which we can pump cold water.

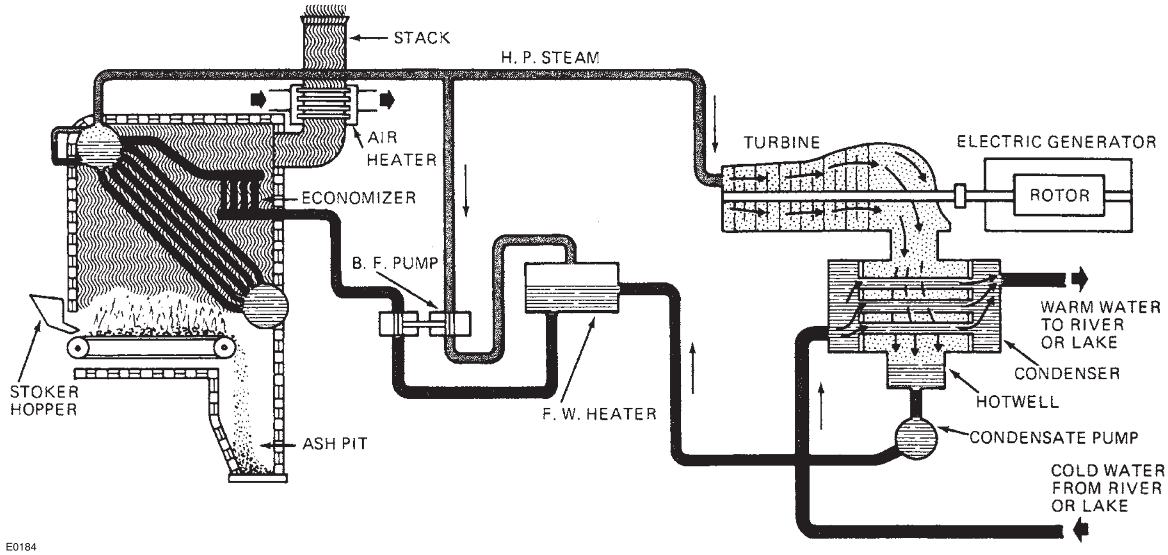


With such an arrangement, the steam issuing from the turbine will come in contact with the cold tubes and thereby turn back into water. This will create a vacuum in the vessel just as it did in the case of the gallon can. The vessel now, however, is made of heavy steel, capable of withstanding the pressure of the atmosphere, and will not collapse.

Since cold water continues to flow through the tubes, the process is a continuous one. There will be a steady conversion of steam into water, and a steady state of vacuum will exist inside the vessel.

Technically, such a vessel is called a condenser. Its purpose is twofold: first, to create a vacuum at the turbine exhaust; and second, to recover the condensate (the condensed steam) so that it can be used over again in the boiler. Since this condensate is really distilled water, it is very pure, and therefore highly desirable for use as boiler feedwater.

So, we build another smaller chamber at the bottom of the condenser to provide a place where the condensate can collect and from which it can be pumped back to the boiler, or rather, first to the boiler feedwater heater. This reservoir is called the hotwell since the water that collects in it is fairly warm.



Now, our system looks like the drawing above.

Steam produced in the boiler flows through the main steam header to the turbine. In the turbine it passes, successively, through the various stages, losing pressure at each stage and giving up its energy to the blades on the rotor. This turns the electric generator and produces electricity. Emerging from the exhaust opening at the bottom of the turbine, the steam enters the condenser where it condenses on the tubes through which the cool circulating water flows.

Condensation of the steam creates a vacuum which reduces the back pressure which otherwise would impede the flow of steam to a considerable extent. The condensed steam collects in the hotwell of the condenser and is drawn off by the condensate pump that pumps the water into the feedwater heater. Here the water is further heated by the exhaust steam from the boiler feed pump (or other steam-driven auxiliaries) and then is pumped back into the boiler by the boiler feedpump. The latter, it will be noted, is run by steam from the main high-pressure header.

We now have a completely closed system. All the water that is turned into steam in the boiler is condensed back into water in the condenser and pumped back into the boiler again. Of course, there are slight losses at various points in the system, leakage through pump bearings, steam leakage through valve packing, etc. To make up for this loss a small quantity of raw water has to be pumped into the system. This is known as makeup water or simply makeup. The idea is to keep the amount of makeup as low as possible.

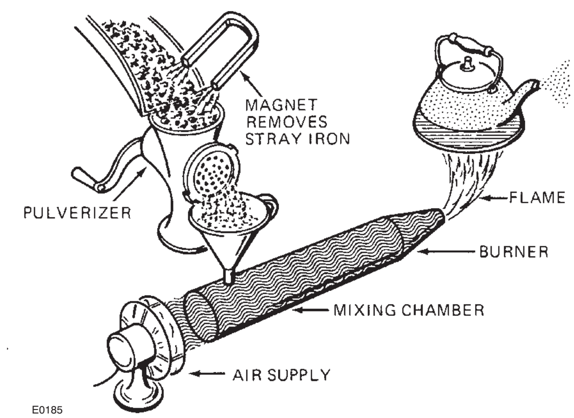
There is only one additional requirement to make a working system. Wherever there is a vacuum, there will be some air leakage into the system, and this air must be removed or it will gradually build up and destroy the vacuum. If the vacuum required is not too high, a combination condensate-air removal pump can be used. For higher vacuums, separate air removal equipment is needed.

Condenser air pumps separate the air from the water vapor and return the water to the condensate system. They generally are of the steam ejector type. Even these are fitted with heat exchangers to recover the heat that might otherwise be lost in extracting the air from the condenser.

If you are interested only in the basic principle of a power plant, this is all you need to know. True, many more refinements can be added which will further improve the efficiency but the system shown in the last diagram would work.

If this explanation, so far, has made sense to you and if it has been of interest, maybe you would like to go a little further and learn something about actual plants as they are built today.

The boiler plant shown in the diagrams is fired by a chain-grate stoker. Most people are reasonably familiar with a stoker. Power plant boilers are also fired by oil or pulverized coal. Where pulverized coal is used, the coal is first passed through a pulverizer which grinds the coal to the consistency of flour. Then, by means of a fan, the powdered coal is blown into the furnace where it burns very much as a gas flame.



Most of the large coal-fired plants today are fired by pulverized coal. One reason for this is that the control of pulverized coal firing is much more flexible than stoker firing. With stoker firing there is always a bed of coal on the grate which contains a considerable amount of heat.

Even if the coal supply were cut off completely, the coal on the grate would continue to burn for an appreciable length of time. With pulverized coal there is no such reservoir of heat and if the coal supply is cut off combustion ceases instantly. The same is true of oil or gas. Also, the maximum rating of stokers is limited. With pulverized coal firing, very much larger boilers can be built.

The use of pulverized coal instead of stokers does not change the basic principle of operation as far as the power plant as a whole is concerned; it merely involves a different type of equipment.

Efficiency

Now, what about steam pressure? In one of the foregoing paragraphs we said something about efficiency being related to steam pressure. Is this true?

Yes, but largely because steam pressure is related to temperature. The higher the pressure of steam, the higher its temperature. At atmospheric pressure, that is, 15 psi absolute pressure, steam has a temperature of 212°F. At 500 psi absolute, the temperature of steam is 449°F.

The efficiency of a turbine, or any other kind of heat engine such as a steam engine or a gas engine, does not depend upon the nature of the working medium-steam, compressed air, ammonia, etc. Instead, it relies upon the quantity and the absolute temperature of the heat received and the heat rejected.

This means merely that a turbine supplied with steam at 400°F and exhausting it at 212°F is more efficient than one receiving the steam at 300^{0}F and exhausting at 212°F.

Also, a turbine receiving steam at 400°F and exhausting at 212°F is less efficient than one receiving the steam at the same temperature (400) but exhausting it at 100^{0}F. This not only explains the value of the condenser but also that of high steam pressure. Without the condenser, the lowest temperature at which steam can be exhausted is 212°F, since that is the temperature of steam at atmospheric pressure. By means of the condenser, however, a vacuum can be created so that the steam will exhaust at a pressure of, say, 10 pounds below atmospheric pressure (i.e., at 5 psi.) Modern turbines exhaust at about 2 psia, or even lower. At 5 psi absolute, the steam temperature would be 162°F.

The thing that is important in the operation of a turbine or any other kind of heat engine, then, is the temperature range through which the heat energy falls in its passage through the engine. The thermal efficiency of the engine depends upon this temperature range. This can be explained very simply by imagining a perfect engine, one in which there are no heat or friction losses of any kind. Of course, such a machine could never be built.

Assuming that we had such an engine, however, let us connect it to a source of steam having a temperature of 400°F. Also, assume that the engine exhausts against atmospheric pressure. The exhaust steam then would have a temperature of 212°F.

Now the thermal efficiency of such a perfect engine is easily figured by means of a very simple equation. Here it is:

$$E = (T_1 - T_2 / T_1) \times 100$$

In this expression, E stands for efficiency in percent, T_1 is the absolute temperature of the steam entering the engine, and T_2 is the absolute temperature of the steam leaving the engine. So, with 400°F (860 absolute)* for the entering steam and 212°F for the exhaust, the efficiency of this theoretical engine is:

100 x (860–672/860) = 188/860 = 21.8 percent

Instead of letting the steam from this engine exhaust against atmospheric pressure, suppose we attach a condenser to the exhaust opening to produce a backpressure of 5 psia. As explained previously, at this pressure the temperature of steam is 162°F (162 + 460 = 622 abs).

Now the equation will give us an efficiency of

100 x (860 – 622)/860 = 238/860 = 27.5 percent

So, by the addition of the condenser, we have raised the efficiency of the engine from 21.8 to 27.5%.

This simple example, then, shows why power engineers have been striving, not only for higher and higher steam temperatures, but also for lower exhaust temperatures. The greater the range between the temperature of the steam entering and leaving the turbine, the higher will be the efficiency of the turbine.

Of course, there is no such thing as a perfect heat engine, and in practice the efficiencies obtained are much lower than those considered in this example for similar temperature ranges. By using much higher temperatures, higher actual efficiencies can be obtained. The most efficient conventional steam power plant in operation has an overall efficiency of about 40%.

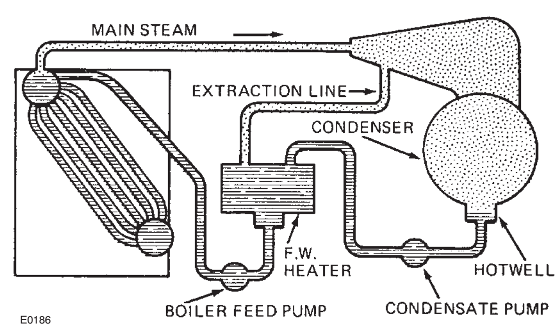
It should be perfectly clear also, that when a condenser is added to a system, large quantities of cooling water must be pumped through it. Also, the condensate has to be pumped out of the condenser This requires power, and this added power has to be subtracted from that developed by the turbine when the efficiency of the system as a whole is considered. We do not get something for nothing in this world.

And this is as it should be; it makes the power engineer's job interesting regardless of which branch of the field he may specialize in. Take this quest for the attainment of higher steam temperatures and lower exhaust temperatures, for example. This has led the power engineer into all sorts of complex things involving chemistry, physics, metallurgy, techniques of manufacture and construction, and, of course, economics. High steam temperatures and pressures together with systems of higher and higher capacity necessitate alloy steels capable of withstanding the high temperatures and pressures. High rates of evaporation in boilers together with high pressures and temperatures make elaborate feedwater treating system necessary, involving constant, close chemical control. High pressures and temperatures also affect the character of the piping arrangements, valves, fittings, and methods of insulation as well as many other things.

With the use of high pressures and temperatures, the simple method of heating the feedwater by exhaust stem from the boiler feed pump or other steam driven auxiliaries will no longer suffice. This is because the feedwater must be heated to a far higher temperature before it enters the boilers.

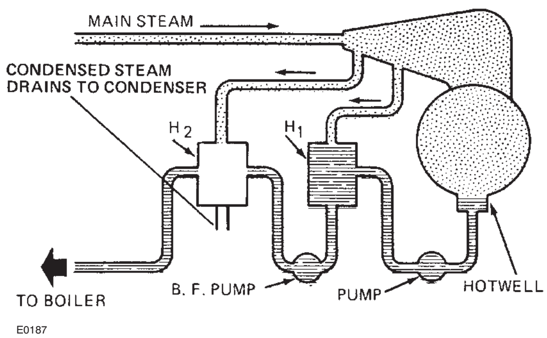
Modern boilers operating at steam pressures of 2500 or 3500 psi require higher feedwater temperatures to avoid severe strains. The use of higher temperatures and more heaters also improves plant efficiency, as explained below.

So we have the problem of increasing the feedwater temperature. What is the best way of doing it? Of course, one way of doing it would be to use high-pressure, high-temperature live steam from the main header and feed it into a special feedwater heater capable of withstanding the high pressure. This, however, would be wasteful. The steam in the main header is the most valuable commodity we have in the system, and every pound of it should be delivered to the main turbine where it can do the most good.



Another method would be to supply the heat required for feedwater heating from a separately fired water heater, but this would be even less efficient than taking live steam from the boiler itself.

Suppose, however, that feedwater is heated by steam extracted from an intermediate stage of the main turbine. Then power will be developed by the steam used for feedwater heating. Moreover, this additional power generated by the extracted steam will be produced at a very high efficiency, and this in turn, will increase the average efficiency at which the total power output of the system is generated.



Now, why is this so? Why does this extracted steam produce power at a much lower fuel cost than that of power produced by steam flowing to the condenser?

The answer to this question lies deep in thermodynamic theory, but for practical purposes it can be explained as follows.

In even the best modern condensing turbine power plants, approximately two-thirds of the heat present in the steam at the turbine inlet is still in the steam at the exhaust. Even if it were possible to have a 100% efficient turbine, the amount of heat thrown away at the exhaust would not be greatly reduced. Actually, the cooling water flowing through the condenser carries this heat away.

This means, then, that even at best, less than one-third of the heat in the fuel can be turned into power in a straight condensing cycle. However, if we extract a portion of the steam from the turbine before it reaches the condenser and use it to heat the feedwater, none of the heat in this steam will be wasted. The reason is that it all will be absorbed in the boiler feedwater Thus it decreases, heat unit for heat unit, the heat that must be supplied to the boiler

Putting it another way, of the steam flowing from the throttle to the condenser, over two-thirds of the heat will be thrown away. Of the steam flowing from the throttle to an extraction opening, no heat will be wasted. It follows directly, then, that the more power that can be generated by extracted steam, the higher will be the average plant efficiency. Of course, the amount of steam that can be used for feedwater heating is determined by the amount of heat needed to raise the temperature of the boiler feedwater to the required level. This places an upper limit on the gain in efficiency that can be made in this way.

As in the previous diagrams, steam from the boiler enters the turbine and flows through the turbine stages to generate power. Most of the steam passes through the entire turbine and exhausts into the condenser. A part of the steam, however, is extracted from an intermediate stage of the turbine at a pressure and temperature higher than that at the exhaust.

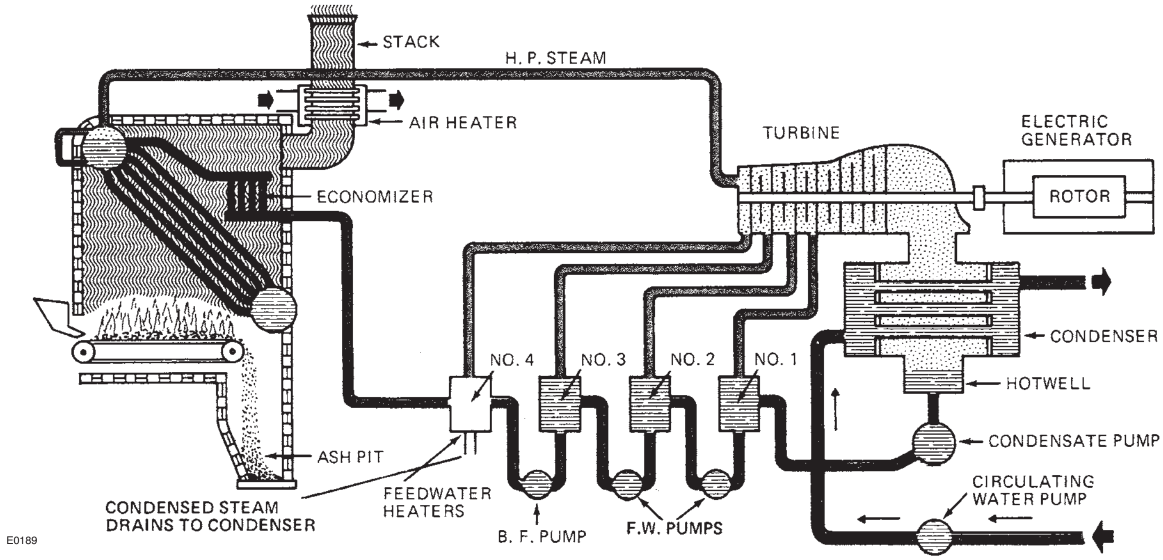
Here is the way in which we can take advantage of this scheme.

In order that the temperature of the feedwater be high enough, it is necessary in this case to extract the steam at a point in the turbine where the temperature is somewhere in the neighborhood of the temperature of the water in the boiler. This, of course, limits the amount of power that can be obtained from this extracted steam. It would be better if an extraction point could be located closer to the exhaust end of the turbine in the manner shown below.

In heating water or any other substance to a higher temperature, however, only a portion of the total heat necessary need be at the highest temperature. Much of the total heat is used in heating the water through a lower range of temperatures. With this fact in mind, it becomes possible to heat the feedwater successively in two or more stages.

Here we have two feedwater heaters: a low-pressure, low-temperature open heater receiving extracted steam from a stage close to the condenser; and another one receiving steam from an extraction point closer to the throttle.

Water from the hotwell is first pumped to heater H_1 where it is raised to a relatively low temperature. Then it is pumped through the second heater where it is raised to a temperature close to the temperature of water in the boiler. This is a closed heater in that the feedwater passes through tubes, and the steam condenses on the outside of the tubes.



With this arrangement, it will be obvious that more power can be obtained from the total amount of extracted steam. In other words, by the use of two heaters instead of one, we have increased the efficiency of the system.

So the question arises, if two heaters are better than one, why would not three be better than two, or four better than three? The answer, of course, is that each additional stage of extraction improves the thermal efficiency.

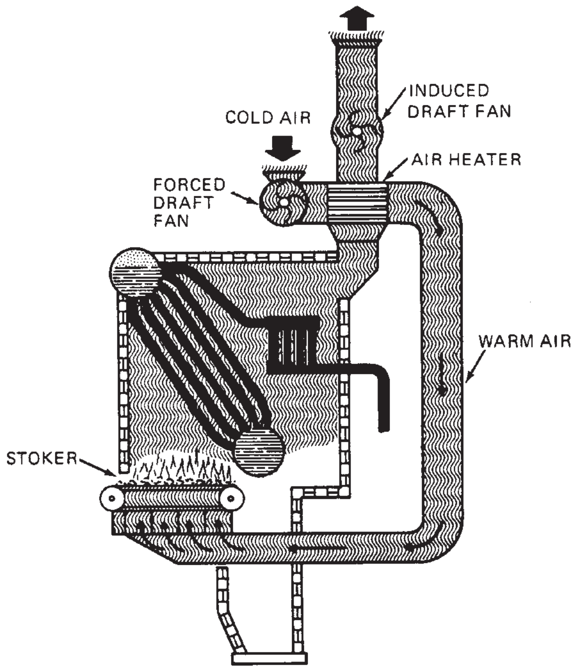
However, in this instance, as in all engineering projects, there is a point of diminishing returns beyond which the further addition of heaters becomes uneconomical. Theoretically, maximum efficiency would be obtained by means of an infinite number of extraction points and feedwater heaters. Actually, four or five stages are commonly used and some of the most modern stations use seven or eight.

The method of heating the boiler feedwater in this fashion is known as regenerative feedwater heating. It is used in all modern steam power generating systems. With such systems a total of 20 to 30% of the throttle steam may be withdrawn from the turbine at various points and used to heat the feedwater.

In the diagram below we have incorporated four stages of feedwater heating in our system with three open heaters indicated and one closed heater. At each heater the water is raised to a higher temperature. Since the pressure in each open heater is higher than in the one preceding, a pump is necessary between successive heaters. Finally, after the water passes through the last open heater, the boiler feed pump pumps it through the closed heater and delivers it to the economizer section of the boiler.

It is very evident from this diagram that a steam power plant can become quite complex when we try to take advantage of all the methods available to increase its efficiency. Indeed, the diagram, as shown, is still far from complete.

While an air heater is shown, the fans and connection to the air heater have been left out so as to keep the diagram simple. Actually, the air is forced through the air heater by means of a fan, and after being heated it is forced into the furnace; in this manner:



This shows how the forced draft fan forces cold air through the air heater and how the warm air is forced through ducts (which must be insulated) into a plenum chamber underneath the stoker. Here it passes through the coal bed and thus supports combustion. A portion of the warm air is also admitted above the fuel bed.

The air heater, it will be noted, is placed directly above the outlet of the furnace. Since it is built of a great many tubes, it introduces a certain amount of resistance to the flow of the hot gases of combustion.

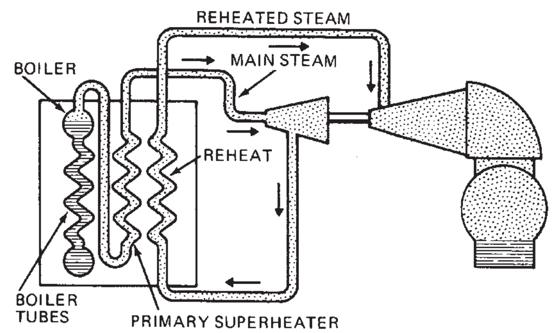
In older plants where air heaters were not used, a tall stack or chimney usually produced sufficient draft to pull the gases out of the furnace; but where air heaters are used, it has become general practice to place an exhaust fan in the passage to the stack.

Such fans are known as induced draft fans. In some cases, the forced draft fan is made powerful enough to maintain flow through the boiler without the need for an induced draft fan.

Since induced draft fans have to operate at comparatively high temperatures and handle all the gases of combustion, they often are very large. Motors as large as 5000 hp, or even larger, may be used, and steam turbine drives may be used for the largest plants.

The plant as we have now designed it is a fairly good power plant but it lacks several important elements. One of these is a superheater. The purpose of a superheater is to heat steam above the temperature at which it is produced in the boiler.

In practice, the superheater is merely an arrangement of alloy steel tubes placed in the gas path through the boiler. After the steam collects in the boiler drum, it passes through the superheater tubes and is thus heated to a temperature higher than that associated with the pressure at which it is produced.



For example, saturated steam at 1000 psi absolute pressure has a temperature of 556°F (remember steam at atmospheric pressure has a temperature of 212°F). Now, by passing the 1000 psi steam through the superheater, it can easily be heated 200 degrees higher, or 756°F. The pressure will remain the same.

Such superheated steam has two advantages over steam that is not superheated. First, it increases the thermal range of the steam cycle, and hence the efficiency; and second, being drier, it is less likely to condense in the lower stages of the turbine.

In large turbines, the formation of drops of water on the blades near the exhaust end of the turbine can be quite damaging. By using superheated steam, however, this condensation can be minimized to a point where it is harmless. Hence, all modern fossil-fired power plants use super-heated steam. Indeed in the most modern plants, in addition to the superheater, the steam is reheated in a second superheater after it has passed through a portion of the turbine. With such an arrangement, steam collected in the steam drum of the boiler passes first through the primary superheater, then through the first few stages of the turbine, then back to the boiler where it is reheated in the reheat superheater. Finally it is sent back to the lower pressure stages of the turbine and so to the condenser.

As a rule, the reheated steam has a temperature only a little below that of the primary steam, but the pressure is considerably lower. For example, the primary steam may have a pressure of 1400 psi and a temperature of 1050°F. After this has passed through a number of stages in the turbine, it is extracted at, say, 200 psi where its temperature will be around 400°F. In the reheat section of the boiler this steam will then be heated to a temperature of 1000°F. Its pressure, however, will still be nearly 200 psi with just a small pressure drop in the piping.

So now we have superheat, reheat, and regenerative feedwater heating in our system (diagram above), and it is becoming very efficient, as steam cycles go. With systems of this kind, thermal efficiencies of about 32% can be obtained (i.e., 32% of the heat energy in the fuel will be converted into electricity.)

You may wonder why this figure is so low. The reason, as implied earlier in this treatise, lies in the fact that most of the heat is carried away by the condenser cooling water. Unfortunately, this low efficiency is a consequence of the second law of thermodynamics, and there is practically nothing that engineers can do about it.

If the temperature of the steam at the exhaust of the turbine could be brought down to absolute zero, we could recover almost all of the heat energy in the fuel. But we live in a world where the ambient temperature is some 490°F above absolute zero, and there is no economical way by which we can reduce the temperature of the exit steam below ambient temperature. Of course, we could do it by refrigeration, but the refrigeration system would require power. We would gain nothing, and indeed, we would lose efficiency.

We have now incorporated into our system about all the known methods for improving the efficiency. Further improvement can be made in any system by going to still higher pressures and temperatures, by refinements in the condensing system, and by the further reduction of heat losses wherever they occur. The latter can be achieved by improved equipment, insulation, and recovery of minor wastes, but the gains to be expected by these means are relatively small.

In large power stations, however, each increment, each fraction of a percent in the overall efficiency is worthwhile because such stations use millions of tons of coal a year. Even small gains in thermal efficiency reflect large savings in fuel costs.

As indicated at the beginning of this discussion, modern steam-electric generating stations use less than one third the amount of coal for the same kilowatt output than they did in early years. This savings has been brought about by continual refinements such as described in these pages.

It is obvious, then, that in the operation of a power station, every effort has to be made to keep the temperatures, pressures, and the vacuum in the condenser at their optimum values at all times. A change in any one of these values affects the efficiency of the system as a whole.

In the condenser, for example, the vacuum must be maintained at its highest value by the use of the coldest available circulating water. Since the circulating water usually is drawn from rivers or lakes, the temperature varies throughout the year, warming up in summer and becoming cooler in winter. This means, generally, that a better vacuum can be maintained in the winter than in the summer, and a somewhat greater power production is possible.

One element in power station operation that has received an ever-increasing amount of attention in recent years is that having to do with the delivery of pure feedwater to the boiler. Modern high-pressure boilers evaporate several million pounds of water per hour, and they do this 24 hours a day, 365 days a year. With such tremendous rates of evaporation, it is obvious that if the water delivered to such boilers contained even small amounts of scale-forming materials, the internal heating surfaces of the boilers would soon become so coated with scale that overheating and subsequent failure would result.

Because of this, elaborate systems of chemical feedwater treatment have been developed which not only reduce the scale-forming materials to practically zero but also reduce the oxygen content to reduce corrosion.

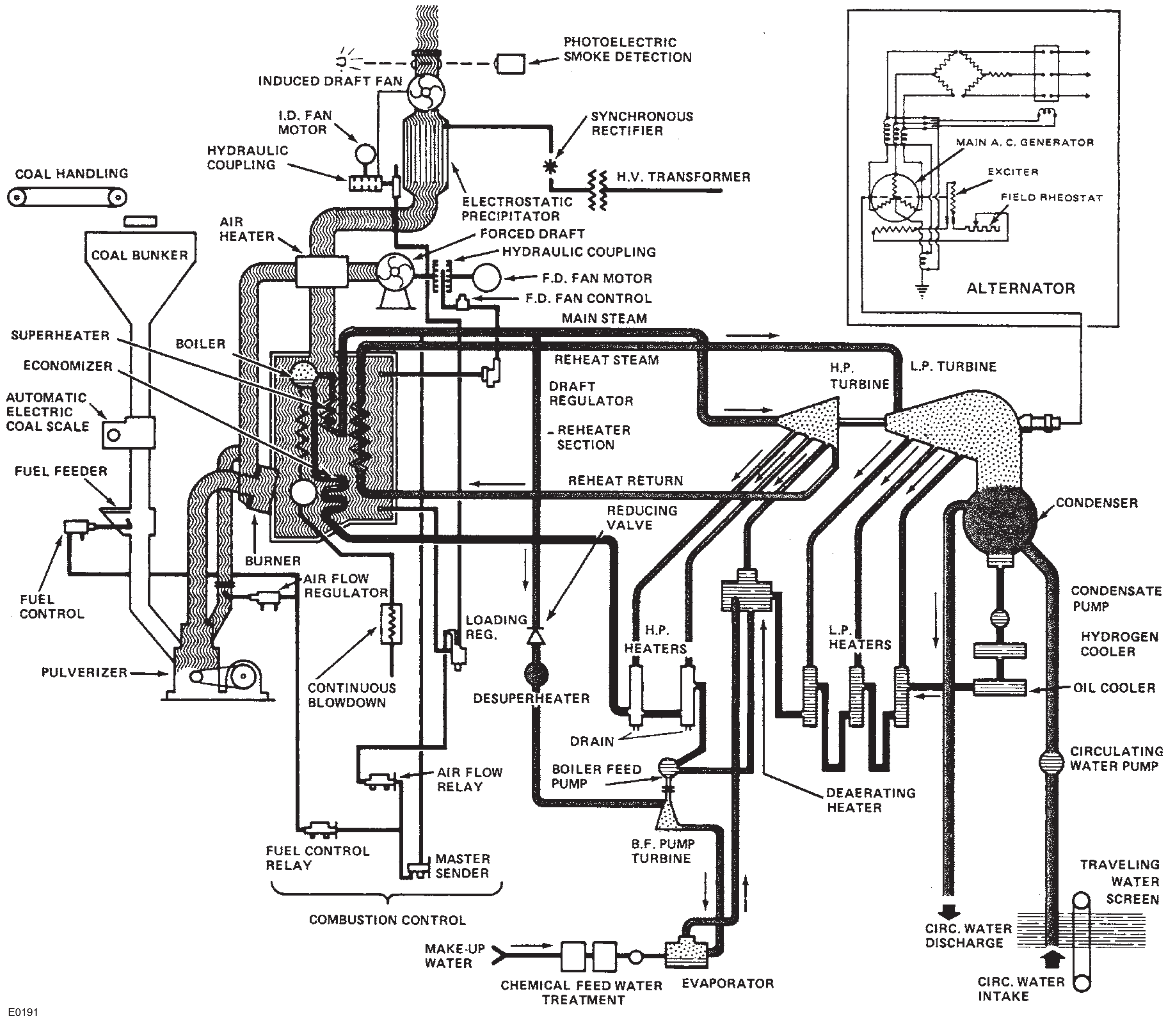
Oxygen in hot water is an extremely corrosive agent. Sometimes these feedwater treatment systems involve evaporators in which raw water is evaporated by steam extracted from the turbine and then condensed. Other systems use ion exchange type demineralizers or chemical treatment which precipitates the scale-forming materials in the form of sludge before the water enters the boiler.

It should be obvious that the amount of water that needs to be added to the system while in operation is small since all the steam flowing through the turbine is condensed and returned to the boiler. There are small losses, however, which have to be replaced by what is known as makeup water. This makeup is the water that has to be treated.

Another factor that has become important in the operation of large power stations is the emission of dust and flash from the stacks. In plants where many thousands of tons of coal are burned each day, the emission of fine ash from the stacks can become a serious nuisance. In most communities, ordinances have been enacted requiring power stations to reduce the emission of flash and dust to a minimum. This has not been an easy problem to solve, and many millions of dollars have been spent in developing equipment for collecting such dust before it enters the stacks.

A number of different methods are in use. In one type of system, the flue gases are passed between electrically charged plates. The intense electrostatic field charges the dust particles, causing them to be attracted to the plates. After the dust has accumulated to a certain thickness on the plates, it is scraped or knocked off. In other systems, the flue gas is whirled through cyclone separators or passed through sprays of water. Whatever system is used, it is expensive and involves high orders of engineering skill in its design.

All new coal-fired boilers will probably also have to be equipped with flue gas desulfurization systems to remove sulfur oxides from the flue gases. This is another highly complex system that increases operating problems.



The Complete Plant

So far we have been concerned only with the steam end of the power system, and have mentioned the electric generator only in passing. This is due to the fact that most of the economies that can be achieved in the operation of a steam-electric generating station lie on the steam side.

The modern electric generator is an extremely reliable and efficient machine. Modern generators have efficiencies as high as 99%, so the additional gains that can be made in the improvement of generators is small.

As pointed out at the beginning of this booklet, an electric generator consists basically of a magnet spinning inside a group of coils of wire. As the rotating magnetic field cuts the convolutions of the stationary coils, electric currents are set up in the coils. By properly connecting them, currents of almost any required voltage can be produced.

Modern generators produce current anywhere between 13,000 and 26,000 volts. Where the electricity has to be transmitted over long distances, the generator voltage is stepped up by means of transformers. A transformer is a sort of electrical lever by means of which voltages can be stepped up or reduced to any desired value. Transformers are the most efficient large machines man has yet devised; some of the larger units have efficiencies higher than 99%.

Now, in the light of all this, let us finally redraw the diagram of our power plant (page 14) to incorporate everything we have mentioned. As you see, it has become a highly complex affair involving many branches of science-physics, chemistry, metallurgy, thermodynamics, hydraulics, structural engineering and electricity.

Complex as it may seem, this diagram is still only the simplest of schematic diagrams; an actual power plant layout has in it countless other small and large devices and subsystems not shown or only indicated on this drawing. For example, so far in this treatise we have said nothing about the combustion control system. It is indicated on this diagram merely in bare detail, showing only the basic devices.

Actually the combustion control system in a large plant would require a large separate drawing, since it is very important and the operation of the station would be virtually impossible without it. In its simplest form, it consists of a device that is sensitive to slight variations in steam pressure. This device, in turn, controls a variety of relays and actuating mechanisms that automatically control the flow of fuel, air, and water to the boiler in accordance with the varying load conditions.

The diagram also incorporates reheat and both high-and low-pressure turbines. After the steam has passed through the high-pressure turbine, it is returned to the boiler, and after being heated again, it is delivered to the low-pressure turbine.

Also, there are two sets of water heaters, three low-pressure heaters taking extraction steam from the low-pressure turbine, and two high-pressure heaters using extraction steam from the high-pressure turbine.

Between the two sets of heaters is a deaerating heater in which the oxygen in the feedwater is boiled off. Oxygen in boiler water at high pressures is extremely corrosive and must be removed before the water is delivered to the boiler. This is done by the deaerating heater which, in effect, is merely a large tank of water boiling under atmospheric pressure.

The necessity for proper feedwater treatment has been mentioned already. On the diagram, the feedwater treating system is merely indicated. It consists of a chemical treating system feeding into the plant system through an evaporator. All the makeup into the plant system passes through the evaporator . Thus the makeup water (the water which has to be added to the system continually to replenish water lost by leakage and blowdown) enters the system in the form of vapor, which is delivered, as shown, to the deaerating feedwater heater. In the evaporator, the incoming water is heated by exhaust steam from the boiler feed pump turbine or other steam driven auxiliaries. As already mentioned, a demineralizer may be used instead of an evaporator.

Note that the boiler feed pump turbine receives steam from the main steam header through a reducing valve and a desuperheater. It is not usual to use superheated steam for small auxiliary turbines for several reasons.

First, the metals that must be used with superheated steam are very expensive; and second, high thermal efficiency is not so important in an auxiliary turbine since the heat in the exhaust is returned to the system.

It becomes expedient, therefore to first reduce the high-pressure steam from the main steam system to a lower pressure by means of a reducing valve, and then to desuperheat the steam by spraying water into it. In this way, low-pressure, saturated steam is delivered to the auxiliary turbines.

The diagram shows a symbol labeled “continuous blow-down.” Because of the continual recycling of the water through the boiler and because of slight leakage in the system, the boiler water tends to increase its concentration of impurities-scale-forming salts. To keep this concentration to a minimum, it is necessary to blow down the boiler periodically or continuously.

In small power plants, this is done periodically by the operator by merely opening a blowoff valve for a few seconds and blowing out the water in the lowest part of the boiler where the concentration is highest.

In large plants, the amount of heat lost by such blowdown practice tends to be rather high, so continuous blowdown systems are used. With such systems a small amount of water is withdrawn continuously and run through a heat exchanger in which the heat from the blowdown is transferred to the incoming feedwater.

These are some of the thousand and one details of a modern power plant that make it the complex thing it is. It is not the intent here to consider all these details but merely to point out that they exist. Little has been said about control except brief mention of the combustion control. The latter, however, constitutes only one element of the station control as a whole. Today, most power stations are controlled from a single control room where quantities from all parts of the plant are measured, indicated, recorded and integrated.

The modern generating station is one of the most completely automated systems man has devised; it has to be because it would be virtually impossible for operators to watch and accurately control all the varying quantities involved in the operation of the plant. Indeed, it has become impossible to keep track of all the pressures, temperatures, liquid levels and speeds of all the various machines without some type of automatic monitoring or supervisory system for recording and announcing all the hundreds of items involved.

The protective features of the station are complex and involved. In case of trouble due to failure of a piece of apparatus or an electrical fault on the external electrical system, events happen rapidly.

If the load on a large generator suddenly dropped and the turbine governors failed to act, the machine, normally rotating at 3600 rpm, would suddenly increase its speed and would explode from centrifugal force within just a few seconds. The 40-ton rotor of a modern electric generator spinning at 3600 rpm has a rotational energy of 650 million foot-pounds. This is approximately the same kinetic energy that a 40-ton jet airliner would have at a speed of 500 miles per hour.

On the boiler side, if the feed-water supply failed, the boiler producing, say, a million pounds of steam per hour, would run dry in 90 seconds.

Since these various types of equipment are valued in terms of millions of dollars, it is obvious that every possible measure must be taken to insure their protection.

There are complex instruments that measure not only the speed of a turbogenerator to a fraction of a revolution, but also the degree of shaft eccentricity and the vibration characteristics of the machine. An expansion indicator shows the axial expansion of the turbine casing.

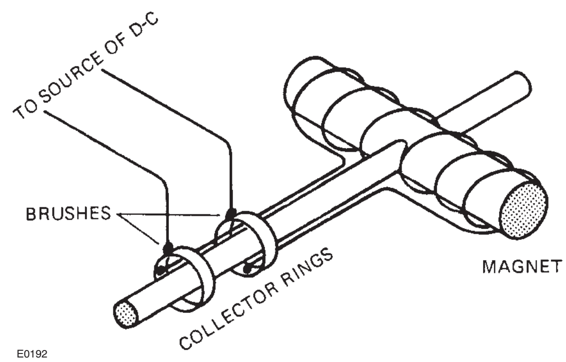
The large diagram shows the generator somewhat differently than the early diagrams; here it is shown symbolically. Also, it is associated with an exciter, a device which, so far, has not been mentioned. Its purpose is to supply the magnetizing current for the rotating magnet.

In an actual electric generator, the rotating magnet is not simply a permanent magnet such as is indicated in the earlier diagrams. A permanent magnet would not provide a strong enough magnetic field, so an electromagnet is used.

An electromagnet is magnetized by electricity flowing through coils wound around the magnet structure. In the case of a large generator a very powerful direct current is sent through the coils on the rotor. These windings are known as the field coils, since they produce the magnetic field in the generator.

To supply the field coils with current makes it necessary to provide an arrangement of sliding contacts to conduct the current from the stationary to the rotating part of the machine. Also, it becomes necessary to provide a separate source of current to excite the field coils.

Basically, here is how it is done, though in an actual generator the shape of the rotor is quite different from the simple bar magnet shown in this diagram.



Two collector rings are mounted on the main generator shaft as shown. These rings are connected to the ends of the field winding. Stationary brushes mounted on the collector rings conduct the exciting current from the source to the windings on the rotating member of the generator.

A separate small direct-current generator mounted on the same shaft as the main rotor usually supplies the direct current needed to excite the field winding. The small dc generator is called the exciter because it furnishes the excitation current for the main generator.

The exciter circuit also provides a means of controlling the voltage of the main generator. By varying the excitation current by means of the field rheostat, the voltage of the generator can be controlled between normal operating limits. The steam-turbine governor, however, controls the speed of the generator, and this control is exceedingly close.

In fact, the generator speed is so closely controlled that it serves as our time standard. Most of us these days measure our time by means of synchronous electric clocks. These clocks depend upon the constant speed of the generator.

Means are provided to check this speed in relation to accurate independent clocks at the National Bureau of Standards. Temperature recorders measure the temperature deep in the interiors ofthe windings of the generator; differential relays guard against internal electrical failure of the machine. Elaborate hydrogen cooling systems are provided to remove heat from the generator, and by means of hydrogen coolers in the condensate system, this heat is returned to the feedwater. No heat is lost that possibly can be saved. Even the heat produced in the bearings is returned to the feedwater by means of the oil coolers in the condensate system.

From all this, it should be clear that a modern power plant engineer has to be a specialist of a very high order. Although the system operates automatically, it still needs the guidance of engineers who are thoroughly conversant not only with details of the equipment but also with the basic principles upon which it operates. Power engineering is a field calling for the highest engineering talent in both design and operation. It provides an occupation that is at once stimulating and challenging, and at the same time affords steady employment in a rapidly growing field with practically unlimited opportunity.

Also, despite this rather long treatise on power plant design and operation, it should be obvious that it is, basically, still only a primer.

Chapter 9A

Conventional Power

The demands placed on control valves within power generating systems vary dramatically. Some valves handle only low-pressure water while in others extreme service conditions create high intensity noise or highly destructive erosion and cavitation. Service conditions often tell only part of the story... application experience and application specific products are a must in order to be successful. This section will talk about the valve solutions available for the key applications common to fossil fuel power stations and most industrial boilers.

Power plants can be broken down into two main types. The first style utilizes a drum style boiler and steam drum that acts as a separator between the steam and the boiling feedwater. These boilers are referred to as "subcritical" since the pressure in the boiler is less than the critical pressure of water. These boilers generally operate with pressures from 2400 to 2800 psi. Drum style boilers are used in most commercial and industrial power plants because of their simplicity, load swinging ability and relative ease of start-up.

The second style of power plant utilizes a once-through boiler. The once-through boiler is designed such that all the feedwater is turned to steam as it journeys through the boiler tubes, and therefore, a steam drum or separator is not required. In this style of boiler, the steam pressure leaving the boiler is generally over 3206 psia, the critical pressure of water. Most once-through boilers are "supercritical", operating at pressures to the turbine of around 3500 psi. However, there are some subcritical once-through boilers in operation.

Large, once-through supercritical boilers are most often base load plants since they have higher efficiencies than drum style boilers. They are handicapped, however, by difficulty of start-up and poor ability to vary plant output.

Shown on the following pages are typical schematics of boiler systems and major control valves. Each valve shown is considered to be a severe service valve that sees cavitation, flashing, noise, or a combination of high pressure and low flow rates. This section will look closely at the tough and specialized valve requirements for these applications. Use of the best available valve technology and application expertise tames these difficult applications and provides longer valve life and higher reliability.

For ease of discussion, the material has been further broken down into five subsystems of the basic Power Cycle. These subsystems include the condensate system, the feedwater system, the main steam system, the heater drain system, and the scrubber and ash handling system. The key severe service valves for each system are noted below.

The Condensate System

- Condensate Recirculation Valve
- Deaerator Level Control Valve

The Feedwater System

- Boiler Feedwater Recirculation Valve
- Boiler Feedwater Startup Valve
- Boiler Feedwater Regulator Valve

The Main Steam System

- Superheater Attemperator Spray Valve
- Reheater Attemperator Spray Valve
- Turbine Bypass Valves
- Deaerator Pegging Steam Valve
- Soot Blower Valve

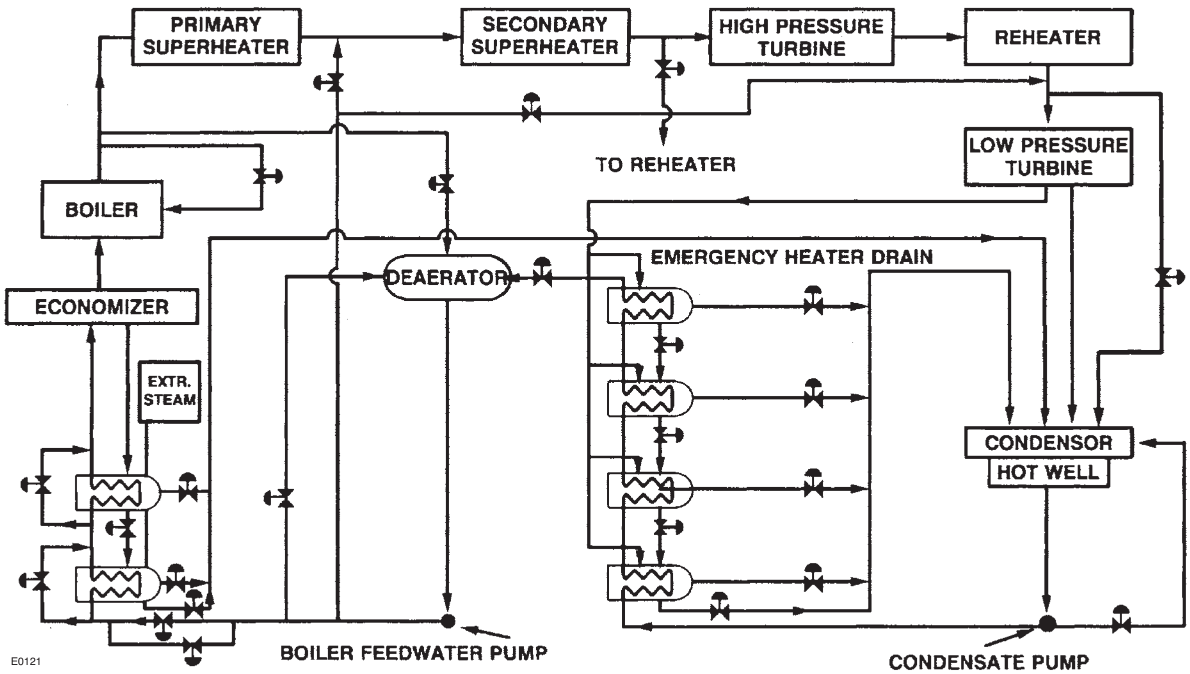


Figure 9A-1. *The steam cycle of a fossil power plant has control valves in 11 key severe service applications. Valves shown in this diagram will be discussed in detail by application.*

The Heater Drain Systems

- High Pressure Heater Drain Valves
- Low Pressure Heater Drain Valves

Scrubber Slurry Handling and Ash Handling Systems

- Scrubber Effluent Valves
- Chemical Service Valves
- Ash Removal Valves

Condensate system

The condensate system spans the portion of the plant from the hotwell of the condenser to the deaerator. The condenser acts as a heat exchanger that condenses the steam from the turbine. The low pressure, low temperature steam passes its heat to the cooling water, is condensed, and collects in the hotwell. The effect of the volume decrease caused by condensation creates a vacuum inside the condenser, which increases the efficiency of the turbine by lowering its outlet pressure. The condensate is pumped from the hotwell through the low-pressure heaters, and into the deaerator. There are two severe service valves in this system. They include the condensate recirculation valve and the deaerator level control valve. Figure 9A-2 shows them schematically.

Condensate recirculation valve. As with most large centrifugal pumps, the condensate pump must have a minimum amount of flow at all times to prevent it from overheating and to avoid cavitation. Therefore, a recirculation line runs from the pump outlet back to the condenser. When recirculation is necessary, the 300 to 600 psig, 100-150°F condensate is dumped to the condenser, which is very close to atmospheric pressure or at a vacuum. The condensate recirculation valve must absorb the entire pressure drop. At the high-pressure drop experienced, the condensate will be cavitating in a standard valve.

The service conditions derived from plant data sheets often will indicate that flashing is occurring [Application Ratio (AR) = $\Delta P/P1Pv>1$]. But experience shows that this is always a cavitating application. It appears to be flashing because often the effects of pipe friction, elevation and condensate sparger backpressure are ignored

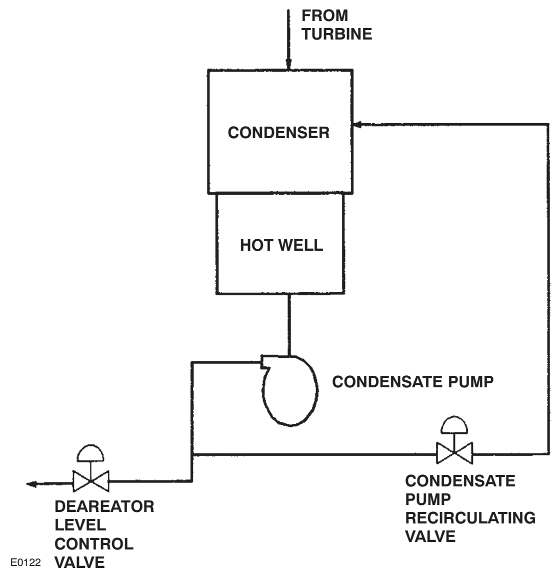


***Figure 9A-2**. The condensate system employs two key control valves. The condensate recirculation valve requires cavitation protection. The deaerator level control valve is distinguished by its need for high rangeability.*

giving very low P_2 values. Secondly, there is a tendency to use the worst case service conditions when in actuality the valve may never operate in that mode. The maximum temperature given will be at full load when no recirculation is required. So, if the sizing is done using this high temperature, the vapor pressure will be too high and the result will indicate flashing.

Control valves in this application need tight shutoff (ANSI Class V) to combat seat damage when the valve is closed. Also, since unwanted recirculation or leakage is lost energy, the best possible shutoff will improve efficiency.

Typical service conditions are:

P_1 = 400-600 psig

ΔP = 400-600 psig

T = 100-150°F

Typically a 3 or 4-inch ANSI Class 600 carbon steel Design ET or EWT with Cavitrol III, two-stage trim is ideal for this application. This combination will give cavitation protection along with the tight shutoff needed. Although the pressure conditions are low and indicate that one-stage trim could be used, experience shows that one-stage trim does not yield satisfactory results. Therefore, the two-stage

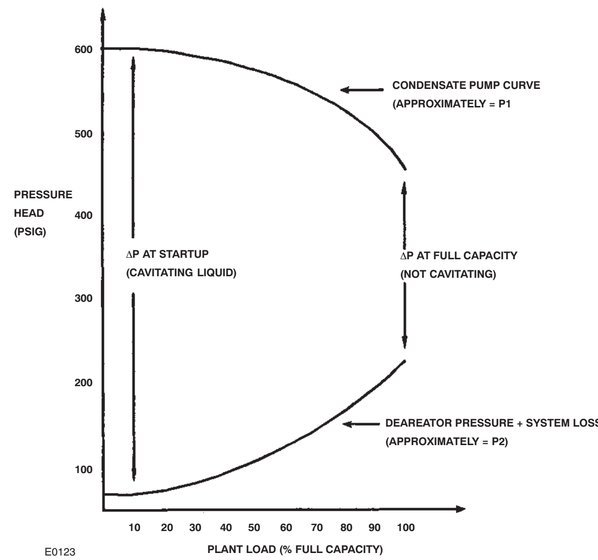


***Figure 9A-3.** A decrease in pressure drop across the DALC Valve is experienced as load increases. At startup, cavitation may occur, thus protection is indicated. At higher loads, the pressure drop decreases, calling for high flow capacity.*

anti-cavitation trim should always be used. Use of standard valves with materials to combat flashing as service conditions indicate is a mis-application that will lead to premature failure.

Deaerator level control valve. The DALC valve is another tough application with very mild looking service conditions. The deaerator level valve handles the same fluid as the condensate recirculation valve, but as plant startup proceeds it must control a widely varying range of flows and temperatures. The purpose of this valve, as the name implies, is to maintain a correct level in the deaerator. The deaerator is an open, contact feedwater heater whose main function is to drive air and other non-condensable gases from the feedwater. The deaerator level control valve performs its function by varying condensate flow into the deaerating vessel in response to load and a level signal. The service conditions confronting this valve vary directly with the plant load. During start-up the pumping load is small, and the valve inlet pressure is high. (Figure 9A-3) The outlet pressure is low because the deaerator pressure has not built up yet. The resulting high-pressure drop presents a need for cavitation protection. The valve Cv required is fairly low. As the plant load and flow increase, the condensate pump can no longer maintain the same pressure head. This results in a lower inlet pressure to the valve. At the same time, line pressure losses to the deaerator are building which back pressure the DALC valve and raise its outlet pressure. The combination of higher flow requirements and lower valve pressure drop creates a need for much more capacity. Cavitation is no longer a concern.

The primary challenge of the deaerator level control valve application involves sizing and selection to fit the specific plant conditions in startup, intermediate load and full load, long term operation. The DALC valve should have cavitation protection for the low flows at start-up and high capacity for low resistance at high flows and loads. Although this valve is open at all times when the plant is up and running, tight shutoff capability may be required.

Typical service conditions are:

P_1 = 300-600 psig

ΔP = 100-600 psid

T = 100-150°F

There are several valve possibilities for this application depending on the Cv requirements and preference. The first option is a globe valve with Cavitrol III one- or two-stage “characterized” trim. Characterized trim cages have a combination of staged drilled anti-cavitation holes at low travel for low flow cavitation protection. Straight through holes are used at high travel for unrestricted flow capability. The typical Cv required produces valve selections ranging from a medium sized ET to a large EWT design. Cavitrol III two-stage characterized trim will satisfy this application.

The second option is a Vee-Ball® valve with an attenuator. This combination gives excellent rangeability, adequate cavitation protection, and is generally the lowest cost solution. However, this design may not be suitable for a cycling plant due to a low flow limitation.

The last option is a parallel valve set consisting of a small globe control valve with Cavitrol III two-stage trim and a large valve with standard trim. Sizing of the small valve is critical so that cavitation is not present when the large valve is opened.

Figure 9A-4. *The Design ET with Cavitrol trim is recommended for condensate pump recirculation applications. Its drilled hole trim gives protection from cavitation damage. Tight shutoff is also featured.*

Feedwater System

The feedwater system encompasses that portion of the plant from the outlet of the deaerator to the boiler inlet. It includes the feedwater pumps, the high pressure heaters, the economizer, and finally the boiler. In this section of the plant, the feed-water pressure is raised to the full outlet pressure of the boiler feedpump, and it is also heated to prepare it for entry into the boiler.

There are three important valves associated with this subsystem--the boiler feedpump recirculation valve, the feedwater regulator start-up valve, and the feedwater regulator. The latter two are generally unique to drum-style boilers and, depending on the boiler manufacturer, the A/E, or the utility, may or may not be control valves.

CAUTION

Alloy 6 corrosion problems often occur in feedwater systems. Typically, boiler feedwater is treated with ammonia, hydrazine or amine derivatives at the deaerator to eliminate excess oxygen. These chemicals attack the protective oxide films on Alloy 6 and allow initiation of an erosion/corrosion process that degrades Alloy 6 overlays. This phenomenon is responsible for many valve trim failures previously attributed to poor design or maintenance. Selection of solid 400 Series stainless steel trim or use of Colmonoy® overlays will eliminate the problem.

Feedwater recirculation valve. The boiler feedpump recirculation valve typically has the most severe service conditions of any control valve in a power plant. In order to protect the feedpump, there must be a recirculation system. The BFPR valve takes feedwater from the boiler feedpump discharge and recirculates it to the deaerator. Depending on the size and type of plant, the valve may be reducing an inlet pressure of as high as 5500 psig to outlet pressures of about 150 psig. These extremely high pressure drops cause very high energy cavitation that will destroy a standard trim control valve in a very short time. In some plants, the recirculation line runs to the condenser, which provides an even higher pressure drop for the valve to handle.

There are three basic methods of providing feedpump recirculation. Each method employs a little different philosophy.

1. The modulating type of BFPR valve provides the most efficient method of recirculating. For each boiler feedpump, the manufacturer publishes a pump curve on which the required Net Positive Suction Head (NPSH) is given. For each power plant, there is a certain Net Positive Suction Head available as a function of the actual plant design and layout. To prevent cavitation from occurring in the pump, the NPSH available must be above the NPSH required. Therefore, the BFPR valve modulates to provide the minimum amount of feedwater to the feedpump to insure that the NPSH available is above NPSH required.

***Figure 9A-5**. The Design EWNT-1 is typical of globe valves used for DALC service. Flow resistance is minimal at high loads and cavitation protection is provided for low load operation and startup.*

2. An on/off method of recirculation provides a constant amount of feedwater flow to the feedpump, up to a specified plant load, at which time the BFPR valve will close. This on/off valve service is less efficient than the modulating recirculation, because some energy is wasted since feedwater is recirculated at higher plant loads.

3. Continuous recirculation might be found in an older power plant. This method recirculates a certain amount of feedwater regardless of plant load, feedwater pressure or feedwater temperature. Since most feedpumps only require recirculation at low load conditions, this method is the most inefficient since excess recirculation is lost pumping energy. Also, since a certain amount

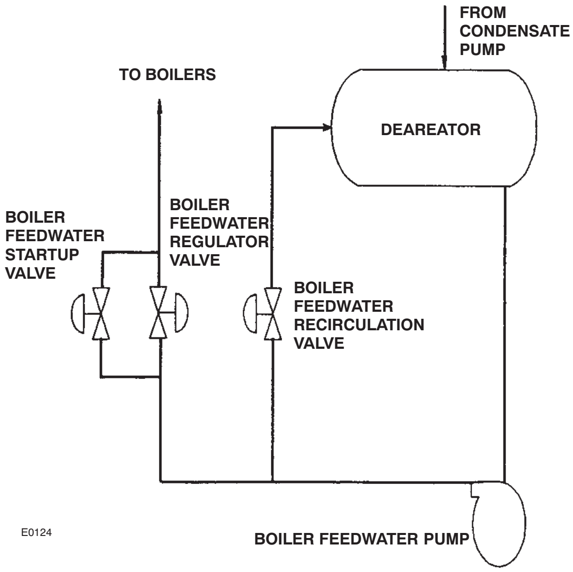


Figure 9A-6. *The feedwater system includes three critical service valves: the feedwater recirculation valve, the feedwater regulator, and the feedwater startup valve. While the valves handle the same fluid, the pressure drop and severity of each application vary greatly.*

of feedwater is always being recirculated, the feedpump will never obtain its maximum rated output capacity.

In older plants, before development of reliable cavitation control trims, the cavitation damage in the control valve was reduced by using a backpressuring device downstream such as orifice plates or a system of capillary tubing. The valves in this type of system were exclusively on/off. This method cut damage by lessening the pressure drop, but did not solve the cavitation problem. The main problem with this type of recirculation valve combination occurs when the valve plug pulls off its seat. For a certain amount of time, the valve will not experience any backpressure and high-energy cavitation occurs. Also, since the backpressure plates are only designed for one set of service conditions, any other conditions will not provide the necessary backpressure to keep the valve from cavitating. If on/off transitions are quick and flow ranges are narrow, it works. Be cautious, however.

As power plant efficiency became a more prevalent concern, need arose for a valve trim that would prevent cavitation altogether and be capable of throttling. Cavitation abatement trim has developed fully and there is a Cavitrol trim to handle every cavitating situation.

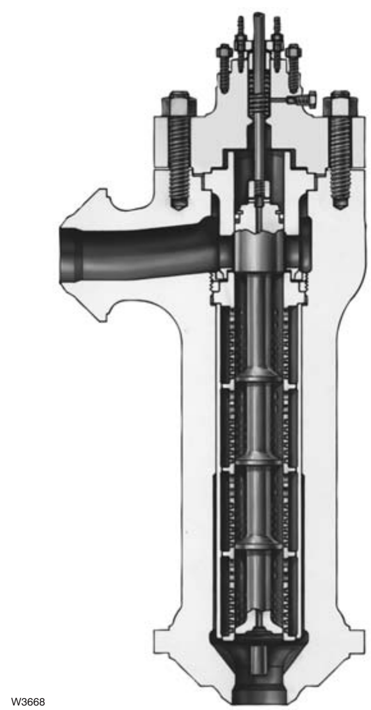


Figure 9A-7*. The CAV 4 is designed for boiler feedpump recirculation use with pressure drops to 6500 psi. It has four-stage, anti-cavitation trim, which is designed to take the pressure drop away from seating surfaces.*

Typical service conditions are:

Drum-Style

P_1 = 2800-3200 psig

ΔP = 2700-3100 psid

T = 250-400°F

Once-Through

P_1 = 3700-4500 psig

ΔP = 3800-4400 psid

T = 350-450°F

Fisher's recommendation for the BFPR valve depends on the pressure drops involved. For drops less than 3000 psi, an appropriate size and pressure class HPT or EHT with two- or three-stage Cavitrol III trim may be used. Characterized trim should not be used since the pressure drop is relatively constant regardless of flow.

For pressure drops up to 4000 psi the Cavitrol III four-stage trim installed in an appropriate size HPT or EHT can be used.

For pressure drops greater than 4000 psi, a CAV 4 with Cavitrol IV trim is used. Typically, a 2-inch or a 4-inch valve is needed. The Design CAV 4 is the most proven valve on the market for these extreme conditions.

At these high-pressure drops, the quality and care of seating surfaces is very important. Tight shutoff can be provided by use of metal seats, which are "soft" and easily deformed and by the use of Tight Shutoff Trim (TSO). These soft seats avoid energy loss and provide longer trim life due to the elimination of wire-draw effects.

For dirty feedwater applications, the Dirty Service Trim (DST) has been used to allow the passage of particulate up to three-quarter inch in diameter and still provide cavitation protection up to pressure drops of 4000 psi.

In a drum style boiler, when the drum is being filled at startup, the feedwater flow control system must absorb the full pressure difference between the feedpump pressure and the unpressurized drum. The high-pressure drops present in this situation offer a high potential for cavitation. Initial flow rates are usually very small. As the drum comes up to pressure, the pressure drop across the feedwater flow control system decreases and the flow increases. Once startup is complete the feedwater flow regulator experiences very small pressure drops. For this reason, feedwater flow control is generally handled with a parallel piped two-valve system. A small cavitation protected valve handles startup and a high capacity feedwater regulator handles higher loads. Due to pressure conditions, both of these valves are high-pressure globe style valves.

Figure 9A-8. *The Design EHD is recommended for feedwater regulator service. It has high capacity and good rangeability with modified equal percentage characteristic.*

Feedwater start-up valve. The start-up valve will have much the same service conditions as the boiler feedpump recirculation valve. That is, high inlet pressure and full drop. This valve needs anti-cavitation trim and tight shutoff. Unlike the boiler feedpump recirculation valve, the feedwater

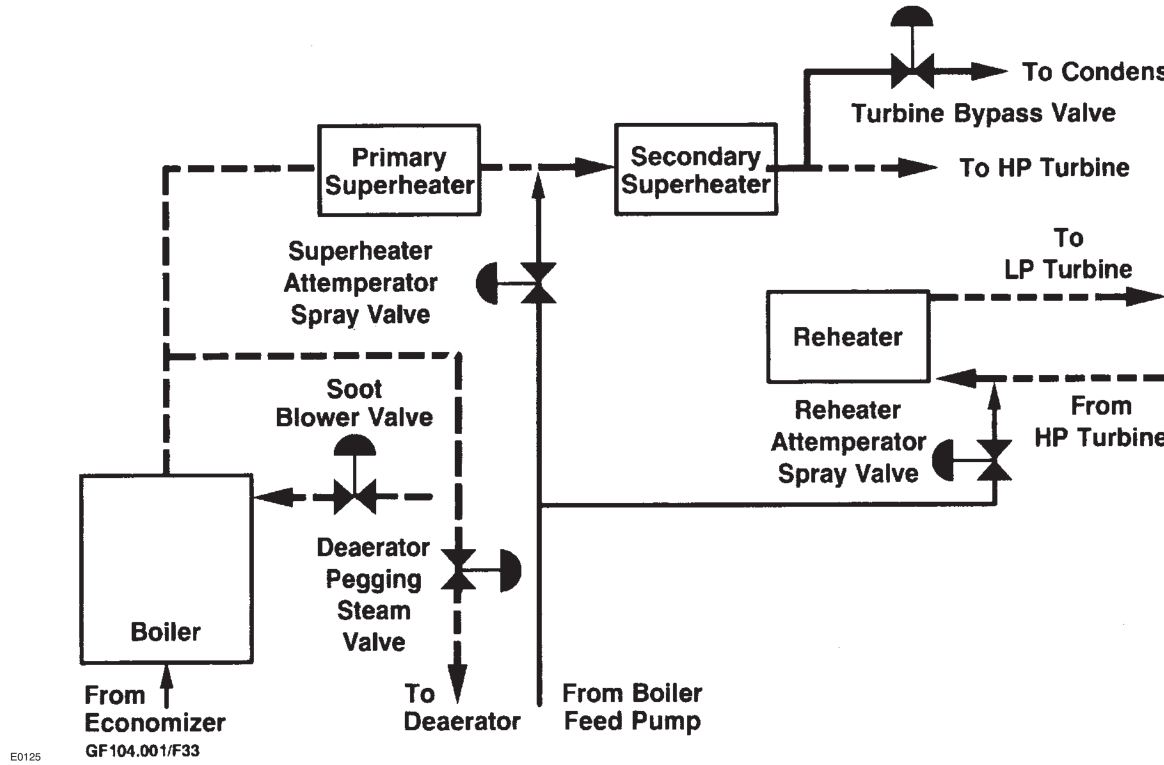


Figure 9A-9. *The main steam system features five critical service control valves.*

start-up valve generally does not need the same amount of cavitation protection at all valve travels since pressure drop changes with travel. Therefore, a Cavitrol III characterized cage may be used.

Typical service conditions are:

P_1 = 2800-3200 psig

ΔP=2700-3100 psid

T = 300-400°F

The normal recommendation is a small HPT or EHT with Cavitrol III three-stage trim. Trim materials should be chosen to avoid use of Alloy 6.

Feedwater regulator valve. Depending on the A/E, utility and boiler manufacturer, the feedwater regulator valve will either be a large motor operated valve or a large, high capacity control valve. In plants with constant speed, motor operated feedpumps, this application usually requires a control valve, and it controls feedwater flow. Those plants with steam driven, variable speed pumps use the pump to control flow and use the valve to control drum pressure and maintain spraywater differential. The valve in this case is usually a motor operated valve, but it can be a control valve.

Typical service conditions are:

P_1 = 2800-3200 psig

ΔP = 100-600 psid

T = 300-400°F

Fisher designed the large, high capacity EH to handle this application (Figure 9A-8). The modified equal percentage characteristic gives the high capacity and large turndown ratio required. Depending on the specific utility, tight shutoff may or may not be required. Again, avoid use of Alloy 6 trim components.

In many smaller plants, A/E's and utilities are looking to combine the Feedwater Startup and Regulator application into one control valve. Depending on the actual service conditions, a Design HP or EH with characterized Cavitrol III trim can often provide the low travel cavitation protection along with the high capacity requirement. Sizing is critical, however.

Main Steam System

The main steam system covers the portion of the steam cycle from the boiler outlet to the condenser. It includes the path from the superheaters, into the high-pressure turbine, through the reheater, and into the low-pressure turbine. Finally, after all the potential is extracted from the steam, it is dumped into the condenser to start the whole process over again.

Five important and challenging valve applications are contained in the main steam system. Included are spray water control valves for both the superheater and reheater attemperators. Turbine bypass systems for both high and low pressure turbines will be discussed as well as steam auxiliary valves for deaerator pegging steam pressure reduction and soot blower control valves. All of these applications involve combinations of difficult service conditions covering high pressure, high temperature, cavitation and noise.

The type of boiler drum style or once-through units has a major effect on the control valves in the main steam system. Once-through supercritical boilers are most often larger units that require larger valves. They also operate with higher main steam pressures. In addition to the valves mentioned there are several valves unique to a once-through boiler. Supercritical start-up system valves and sliding pressure control are in the main steam system but are described and discussed separately in Chapter 9B.

Superheater attemperator spray valve. To gain maximum efficiency from the steam and at the same time not damage the delicate blades of the turbine, the temperature of the steam must be carefully controlled. Between the primary and secondary superheater in the main steam system there is a desuperheater, or attemperator, in the steam line. The water spray from the attemperator lowers and regulates the temperature of the steam going into the secondary superheater. This valve controlling the water spray flow is called the superheater attemperator spray valve. The water is taken from an auxiliary line off the feedwater pump. Its pressure is slightly higher than the steam line pressure; therefore, the pressure drop through the valve is low.

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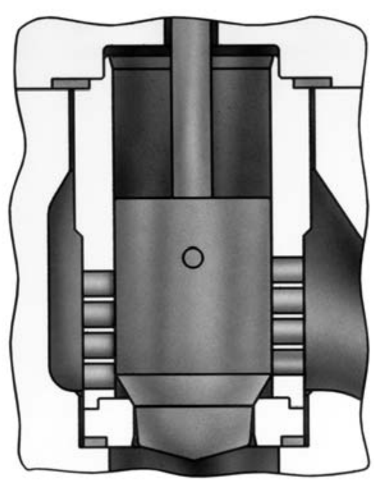

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Figure 9A-10A & 10B. *Small Design HPS valves with Micro-Form trim are commonly used for superheater attemperator spray valves. Tight shutoff is required.*

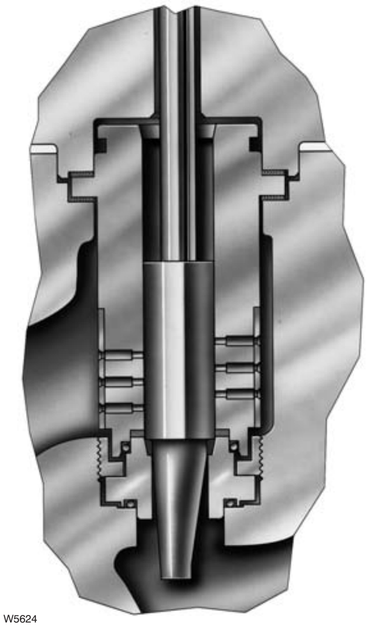

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Figure 9A-11. *The high pressure drops encountered in reheat attemperator spray applications require special trim design. The Micro-Flat plug combined with Cavitrol two-stage trim is recommended.*

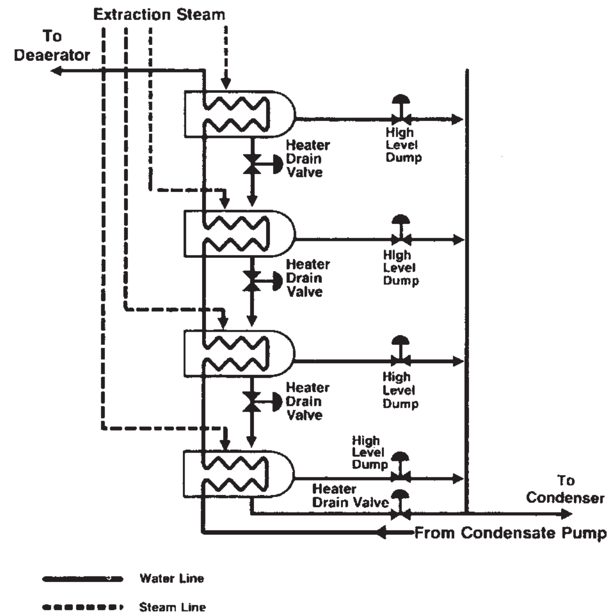

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Figure 9A-12.

Typical service conditions are:

Drum-Style Unit

P_1 = 2800-3200 psig

ΔP = 100-200 psid

T = 300-400°F

Once-Through Unit

P_1 = 3600-3900 psig

ΔP = 100-200 psid

T = 300-400°F

Application requirements include high pressure, good control performance and tight shutoff. The valves most often used in this application are a ANSI Class 2500 Design HPS or EHS with Micro-Form trim. The Micro-Form trim is used since the flow rates are fairly small, and it gives an excellent control characteristic. Tight shutoff is required to eliminate unwanted spraywater flow and to allow for the best possible temperature control of the steam.

Reheater attemperator spray valve. The reheater attemperator spray valve serves a function similar to the superheater attemperator spray valve. In this case, the attemperator is located in the reheat line before the steam enters the low-pressure turbine. The steam is at a much lower pressure in the reheat section, so the spray water valve experiences much higher pressure drops and is likely to cavitate. This is a particularly demanding application because very low flows must be controlled in the presence of cavitation. Tight shutoff is also required.

Typical service conditions are:

Drum-Style Unit

P_1 = 2800-3200 psig

ΔP = 2300-2600 psid

T= 300-400°F

Once-Through Unit

P_1 =3600-3900 psig

ΔP = 2300-3600 psid

T = 300-400°F

Two trims have been designed for and perform well on this application. A unique Fisher trim used in the Design HP and EH is the first of these. The trim is called Micro-Flat trim. It is designed to accommodate low flow and can throttle to a Cv as low as 0.01. This trim is limited to a maximum pressure drop of 1000 psi. It utilizes a plug and seat ring with a relatively low recovery coefficient, but does not prevent cavitation. Rather it isolates the cavitation, preventing it from causing seat damage. By putting this trim in a carbon steel angle body and also by using a liner, body damage can be prevented. Assuming a chrome-moly or stainless steel body is used, a liner is not required.

For applications where very high pressure drops and higher flow rates are encountered, the Micro-Flat trim is used with a Cavitrol III cage. With the combination of the Micro-Flat plug and the Cavitrol III cage, the maximum allowable pressure drop is dependent on the selected cage. This combination gives a high recovery coefficient and eliminates cavitation at the high flow condition while preventing seat damage at the low flow condition. The resulting trim provides long life, good throttling performance and eliminates problems related to cavitation.

Turbine bypass valves. Use of sophisticated turbine bypass systems is increasing. These systems save energy, allow quicker startup, cope better with load rejection and help eliminate solid particles in steam.

As more large generating facilities swing their output to meet varying demand, they place additional requirements on the turbine bypass valve, which formerly served only as an on/off device. The bypass function now requires the valve to quickly change load on the turbine during start-up, emergency or cleanup situations. Desuperheating functions to control steam temperature often are necessary to protect piping, reheaters and the condenser. Valve noise and the damage it can cause are major concerns because of the extended periods that turbine bypass and steam conditioning valves may operate.

High-pressure turbine bypass. A high-pressure turbine bypass system provides an alternate flow path for high-pressure, high-temperature steam. Flow passes around the turbine and back to the reheat section. This bypass system permits stable operation of the boiler when the turbine trips off line or during startup operations. Steam flowing through the high pressure bypass valve (TBX-T) is cooled to a temperature slightly above the H.P. turbine exhaust temperature by spraying feedwater into the outlet of the bypass valves. This flow is then combined with high-pressure exhaust steam and passes through the reheater. The control system must provide the logic to open the valve quickly and then modulate with feedback control to predetermined pressure and temperature set points. Opening speeds of less than two seconds are typical. In operation, the high-pressure bypass provides the same expansion, pressure reduction and cooling which would have happened in the high-pressure turbine. It protects the reheat section of the boiler and quickly unloads the turbine without requiring boiler trip.

Typical service conditions are:

Drum-Boilers

P_1 =2800 psig

ΔP = 2400-2800 psid

T = 1000°F

Once-Through Boilers

P_1 = 4500 psig

ΔP = 3000-4500 psid

T = 1000°F

Low-pressure turbine bypass. The low-pressure bypass system provides a flow path around the low-pressure turbine and controls pressure and temperature whenever the high-pressure bypass system is operating. The low-pressure bypass valve takes steam from the reheat section of the boiler and conditions the steam to be accepted into the condenser. High pressure and temperature drops are taken by valve throttling and by addition of large amounts of desuperheating spray. Temperature control is not critical except to protect the condenser. This protection is accomplished by deliberate overspray and by quick closing capability. The Type TBX-T usually handles low-pressure bypass or condenser dump applications.

Product requirements for both high and low pressure bypass valves include high pressure and temperature, noise attenuation, tight shutoff and desuperheating.

Typical service conditions are:

P_1 =500 psig

ΔP= 300 psig

T=1000°F

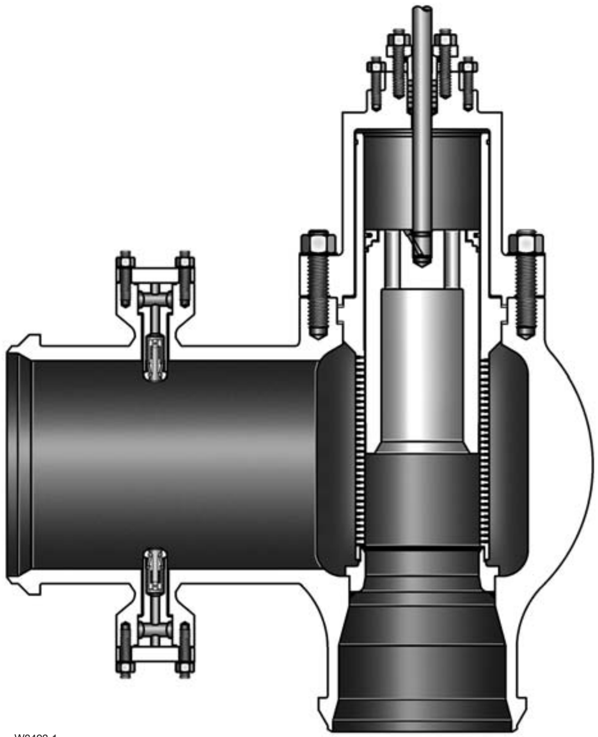

Figure 9A-13. *Type TBX-T turbine by-pass valve. Spraywater is introduced from multiple nozzles in the radial steam flow to provide efficient mixing.*

Figure 9A-14*. Type TBX-T features maximum desuperheating spraywater capacity to ensure condensor protection.*

For these applications, either an EHD or HPD with Class V shutoff utilizing a C-Seal can be used. Steam cooling can be accomplished through use of a cooling chamber downstream or desuperheaters. Noise trim is not always required because of the infrequent use of the valves, but the noise should always be kept below 100 dBA to prevent trim damage.

Deaerator pegging steam valve. The deaerator is a contact heater (a heater where steam and feedwater are mixed) that is especially designed to remove the non-condensable gases from the feedwater. These non-condensable gases include oxygen and carbon dioxide that attack and corrode piping and boiler tubes. Generally hot steam is mixed with the feedwater entering the deaerator bringing it to saturation temperature, thus liberating any dissolved non-condensable gases.

For drum-style boilers, the steam that is supplied to the deaerator is usually taken from an auxiliary line off the main steam line. For once-through boilers, steam is taken from the flash tank or the final high-pressure heater. Thus, it is at a lower pressure than a comparable drum-style unit.

The steam is usually at a higher pressure than needed and, therefore, must be reduced. When steam is subject to a substantial pressure drop across a control valve, there are inevitably noise problems to consider. The deaerator pegging steam valve is considered a severe service valve because the noise levels will probably be of concern and some sort of noise abatement equipment should be specified.

Typical service conditions are:

Drum-Boilers

P_1 =2400 psig

ΔP = 2300-2400 psid

T = 650°F

Once-Through Boilers

P_1 = 200-400 psig

ΔP = 200-350 psid

T = 650°F

Product requirements must be able to handle high pressure, high-pressure drop and high temperature. Long life in a throttling application is required. Because of the wide variety of service conditions seen by valves in this service, there is no 'typical' valve for the application. For selection purposes, start with a standard trim valve. If high noise levels are indicated they should be treated using Whisper Trim I®, Whisper Trim I plus a diffuser, Whisper Trim III or WhisperFlo. Experimentation may be necessary to come up with the best combination of noise abatement versus money spent.

Soot blower valves. The efficiency of a fossil-fuel boiler is highly dependent on the heat transfer effectiveness of the boiler tubes. These tubes are fairly delicate, and hot spots due to soot buildup cannot be tolerated as a leak could result. A cleaning process for the boiler tubes is needed even while the boiler is in operation. This process is called soot blowing and is accomplished one of two ways. Some plants spray steam bled off their main or secondary steam lines to blow the carbon deposits off the tubes. Others use air blowers for the same purpose.

The applications that are most difficult for control valves are the soot blowers that use steam. Product requirements include a high pressure class rating due to the pressures and temperatures as well as needing tight shutoff so the valves don't leak valuable steam. Soot blower valves are often operated numerous times during the day. This leads to temperature cycling of the valve, especially if a block valve isolates the soot blower valve from the steam line. In addition to temperature cycles, large pressure drops create high noise levels and can cause excessive wear and vibration to occur in the trim. Use of trim specially designed for this service will provide much improved performance. The combination of cyclic conditions, high noise levels and frequent operation is likely to create problems otherwise.

Typical service conditions for the steam soot blower valves are:

P_1 = 2400-3600 psig

ΔP = 2800 psid

T = 650-950°F

For most steam service applications the Design HPS or EHS should be used. They provide high pressure and temperature capability along with the tight shutoff that is required. From Fisher's experience, the soot blower valve should be supplied with a drilled hole cage in the flow up direction and an oversized valve stem connection. This combination, known as soot blower trim, will minimize wear and vibration and maximize life. Usually, chrome-moly body materials will be used due to the high temperatures involved. For soot blower valves that utilize air, the valve selection could range from an **easy-e**® globe valve to a Vee-Ball rotary valve, depending on the pressure and temperatures for the specific application.

Heater Drain System

There are two sets of feedwater heater systems in a typical power plant. The low-pressure system heats the condensate coming from the condensate pump so that it is near saturation when it enters the deaerator. The other set, called high-pressure heaters, heat the feedwater coming from the boiler feedwater pump so that it is near saturation at the higher pressure conditions when it enters the boiler. Both systems work in the same way with the exception of the heating media. In the low-pressure heaters, exhaust steam from the low-pressure turbine is used while the high-pressure heaters use extraction steam from the reheat section.

Feedwater heaters are shell and tube heat exchangers. Steam is introduced, cooled and ultimately condensed back to liquid. In the process it passes its heat to the feedwater. The level of condensate in the heaters must be closely controlled for best system efficiency, so the drain system is fairly elaborate.

A potential problem is that the condensate in the bottom of the heater is at saturation. When the condensate is drained to the condenser it loses pressure and flashes. The flashing fluid causes erosion damage to the control valve and associated piping. The important thing in choosing control valves in this application is to use a low resistance valve to keep the velocities as low as possible and keep flashing damage to a minimum. Material selection is also critical.

For the high-pressure heater drain system, a globe valve is probably required to handle the static pressures involved. The primary recommendation is to use a WC9 alloy steel Design EW with hardened trim. The materials resist erosive attack, and oversized end connections slow fluid velocity in the body. When an ANSI Class 1500 valve is required, the recommendation is an HP or EH. An angle body will eliminate body erosion concerns due to its configuration.

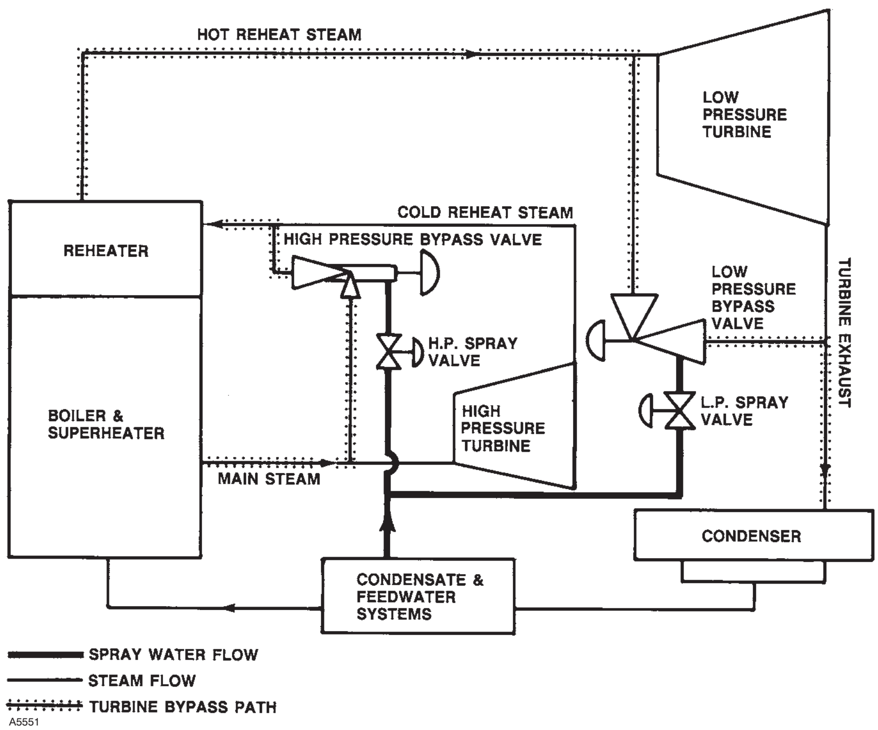

Figure 9A-15. *Schematic of typical turbine by-pass system featuring both high pressure and low pressure by-pass application.*

Regardless of pressure class, material erosion is a common problem in flashing service valves such as feedwater heater drains. This wear is caused by impingement of high velocity liquid droplets on the trim and body surfaces. Chrome-moly steels are often used to improve valve body erosion resistance. While C5 has commonly been used, it has largely been phased out by use of WC9 which is more resistant to erosion and is much better from a manufacturing viewpoint.

Erosion resistance is a function of both molybdenum and chromium content. While WC9 has lower chromium content, its much higher molybdenum content more than compensates. WC9 is more erosion resistant than C5 and also is much easier to cast, weld repair and manufacture reliably.

There are several recommendations for the low-pressure heater drain system. If globe valves are required, the WC9 Alloy EW with hardened trim is the answer. A low pressure EA also can eliminate problems associated with body erosion due to its angle configuration. Because of the low resistance, high recovery characteristics of rotary valves, it has been Fisher's experience that the rotary line is an excellent choice for this service.

Fisher has had excellent results utilizing the Design V500, level 3 trim, reverse flow, for heater drain applications. The reverse flow direction keeps flashing from damaging the control valve. Reduce velocities by using a line size valve.

Scrubber Slurry

Federal regulations require a sulfur dioxide removal system on every utility boiler, even when it is burning the lowest sulfur coal available. Because of these regulations, many power plants have added sulfur dioxide removal systems.

Sulfur dioxide (SO_2) is a gas formed when the sulfur in the fuel is burned. All but a small percentage of the sulfur is converted to sulfur dioxide during the combustion process. Wet scrubbers using lime or limestone, dry scrubbers, fluidized-bed combustion and reducing the sulfur content of the fuel are methods of reducing the amount of this pollutant released to the atmosphere.

There are many designs for wet scrubbing systems, such as lime, dual-alkali, and double-loop processes, the last of which produces a commercial by-product, gypsum. The single-loop

Figure 9A-16. *The Design V500 is recommended for erosive and flashing applications found on low pressure heater drains and on slurries of ash or scrubber by-products.*

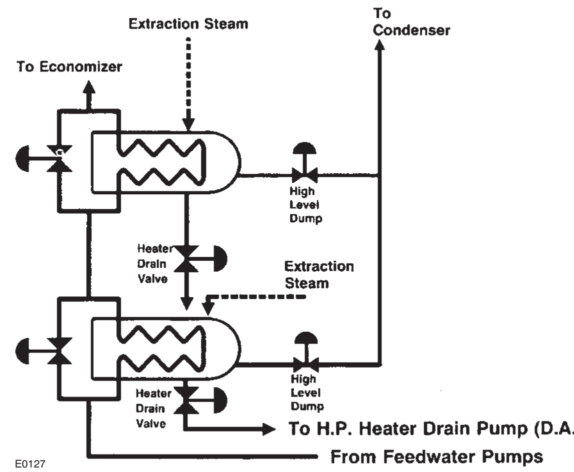

Figure 9A-17. *The high pressure heater drain system provides a difficult combination of high pressure and flashing fluid. Use of oversized end globe valves is generally recommended.*

wet limestone process is the system most commonly used in today's plants and will be discussed here.

Wet scrubbing, by its name, means that a wet product is produced after the sulfur dioxide has been removed from the flue gas. Today, the wet limestone throwaway type of flue-gas desulfurization (FGD) system is most commonly used. The system is based on a closed-loop waste disposal system. Solids are removed from the FGD system in the form of sludge.

The sulfur dioxide removed in the scrubber is converted to calcium sulfite ($CaSO_2$) and calcium sulfate ($CaSO_4$). These are the waste products that must be continuously removed and sent to disposal.

A magnesium-oxide wet-scrubbing system is often used for sulfur recovery. Magnesia solutions or slurries are used to remove up to 95% of the sulfur oxides and 99% of the particulates from the flue gas. The scrubbing effluent is regenerated to recover MgO for reuse. A saleable sulfur product is produced. Erosive and corrosive slurries are to be handled similar to the system described above, requiring similar valve applications outlined below.

Despite the ultimate disposal method or the end use of the scrubber's effluent, piping is used to move the sludge from the scrubber. A control valve is used in this sludge disposal pipeline to continuously throttle a 30% or more solids slurry containing calcium sulfate/calcium sulfite and ash, which is very erosive and corrosive. It is located immediately following the scrubber and controls the scrubber effluent dumping into settling ponds, maintaining the scrubber level so it is not flooded. Typical valve service conditions are: 150-350 gpm and 50 psi pressure drop.

Boot valves have been commonly used in this service, using the boot as a sacrificial part to be replaced periodically with wear. This can become a frequent, difficult, expensive, and messy maintenance job, in some cases as often as every three months.

The Design V500 has proved to be a very effective valve for this application with level 3 trim materials; 316 SST body and solid Alloy 6 valve plug, seat ring, retainer; and sealed bearing with Viton o-rings and a Nitronic-50 shaft. The higher initial cost of the V500 can be paid for quickly compared to the alternative of using a boot valve requiring frequent boot changeouts.

Sulfuric acid disposal may also require a throttling or on/off control valve. A chemical service valve, such as the 585C-CP with Alloy 20 body and trim parts is a suitable selection for 10-20% solution sulfuric acid service. Ninety-eight percent sulfuric acid solutions can be adequately handled utilizing more standard 316 SST and nickel based alloys such as Hastelloy® C.

When choosing and specifying alloy materials it is important to note the quality of the vendor's material process technology. Seemingly small differences in raw material processing can make the difference between long life and quick failure. Nickel base cast parts should at a minimum be specified from AOD material.

Ash Handling

Ash removal valves are primarily on/off operated and are subjected to a highly erosive service. A variety of valves are used and replaced frequently due to the severity of material destruction occurring in this application.

The Design V500 can be very effective in ash slurry flow applications, including ash disposal. This application can also be highly erosive and corrosive, requiring frequent valve maintenance and changeouts. The V500 with level 3 trim and sealed bearings will provide extended service life.

Balls and seals in remote operated ball valves wear out frequently, and the cost of maintenance becomes high enough to justify considering a valve capable of severe services. Again, the V500 with level 3 trim and sealed bearings has been applied successfully.

Chapter 9B

Sliding Pressure Control

The need for cyclic operation of base load plants during the past two decades has become more critical for several reasons. Forecasts of electric power demand for the previous decade greatly exceeded actual demand, so many utilities had excess capacity. Also, many nuclear plants that were under construction were completed and are being used to meet base load requirements. Finally, fluctuations in fuel prices for oil and gas fired units, the traditional choices for cyclic or peak load plants, may make these units less cost effective. These three factors make it necessary to have dependable, efficient variable output from plants which were originally designed for base load operation.

In a turbine-generator, the electrical power output is dependent on the pressure of steam entering the turbine. First, the boiler is fired and brought up to a constant discharge pressure. The turbine is equipped with several valves, known as turbine control valves, which regulate the turbine inlet pressure. As load decreases, the valves move closed to reduce turbine inlet pressure. All valves may move closed equal amounts in unison (full arc admission) or they may close sequentially (partial arc admission). This is known as constant pressure operation, since the inlet pressure to the turbine control valves is essentially constant.

Constant pressure operation has two adverse effects when large load changes occur. First, the turbine will experience temperature fluctuations that cause fatigue and reduce its life. Second, the net thermal efficiency or heat rate of the turbine drops at lower loads.

Sliding pressure operation is designed to eliminate these effects. In general, it consists of adding pressure-reducing valves upstream of the turbine control valves. These valves reduce the discharge pressure from the boiler and allow the turbine control valves to remain fully open. Temperature variations are reduced and the net thermal efficiency is improved.

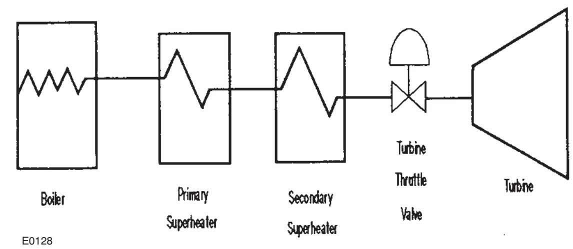

Figure 9B-1. *Once-through main steam system -- constant pressure control*

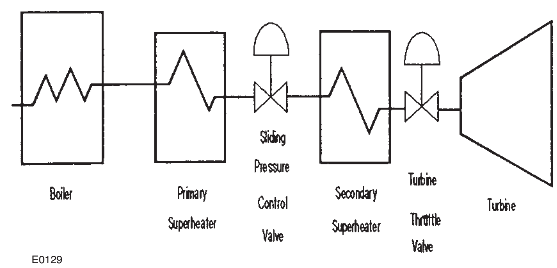

Figure 9B-2. *Once-through main steam system -- sliding pressure control*

Figure 9B-1 shows the typical main steam system of a supercritical, once-through fossil unit from the boiler to the turbine. In order to cycle the plant, the inlet pressure to the turbine must be varied. In constant pressure operation, turbine throttle valves control the inlet pressure to the turbine, which is in proportion to plant load.

For units with sliding pressure control, a large valve or series of large valves are installed between the primary superheater and the secondary superheater (Figure 9B-2). Although the turbine throttle valves are still in the system,

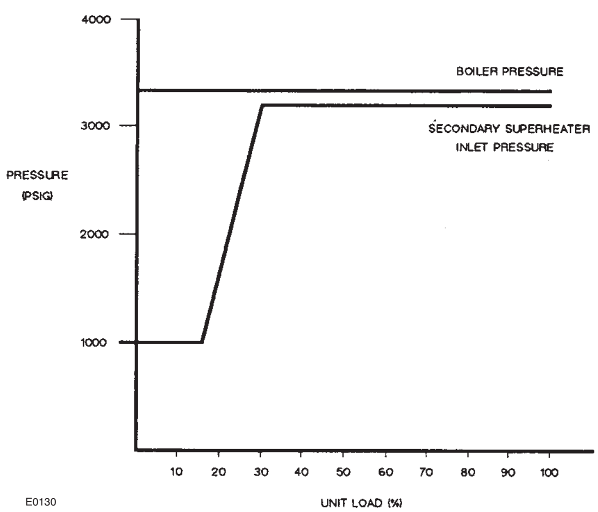

Figure 9B-3. *Constant pressure control*

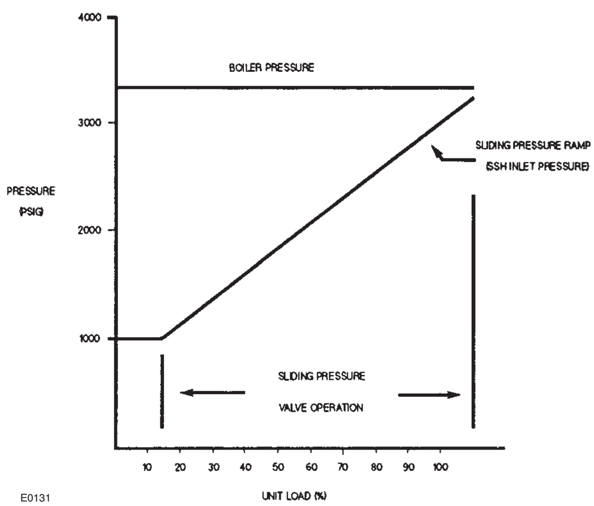

Figure 9B-4. *100% sliding pressure control*

they are held wide open, and the sliding pressure control valve varies plant load (turbine pressure.)

Figures 9B-3 and 9B-4 show the relationship between boiler pressure and secondary superheater inlet pressure for constant pressure control and sliding pressure control respectively. The combustion control system is set up to bring the boiler to a pre-established discharge header pressure, where it is maintained throughout operation. Under constant pressure operation, a set of small startup valves controls the secondary superheater inlet pressure and therefore, turbine

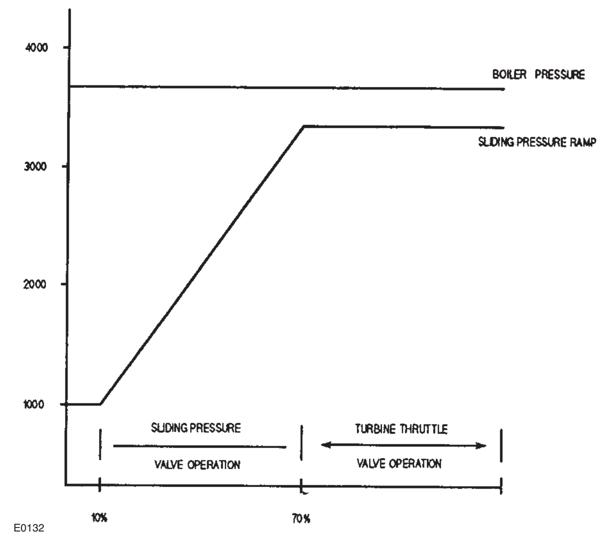

Figure 9B-5. *70% sliding pressure control*

pressure, up to 25% of maximum plant load. Above 25% load, the turbine throttle valves control the turbine pressure. 100% sliding pressure control (Figure 9B-4) means that the sliding pressure control valve is sized to control the plant load from a minimum load through 100% plant load. If the sliding pressure control valve is sized for 70% of the plant load, the unit is said to have 70% sliding pressure control (Figure 9B-5). From 70 to 100% plant load, the turbine throttle valves are used to control plant load and the benefits of sliding pressure control are lost. In summary, the sizing of the sliding pressure control valve determines the amount of sliding pressure control that a plant has available.

The thermodynamics of a control valve must be reviewed in order to understand the benefits of sliding pressure control. Thermodynamic laws state that for a system which does no work, the enthalpy of the system is constant. Since no work is done in a control valve, the enthalpy upstream of a valve must equal the enthalpy downstream of the valve. In equation form:

h1 = h2

h = enthalpy, BTU/lb

Since enthalpy is a function of pressure and temperature, steam tables can be used to show that as steam pressure is reduced, steam temperature is also reduced.

In constant pressure operation, the pressure reduction and consequently, temperature reduction, occur at the inlet to the turbine. This temperature loss reduces the efficiency of the turbine. Additionally, this uncontrolled inlet steam temperature places increased stress and potential wet steam erosion problems in the latter stages of the turbine.

Under sliding pressure control, the pressure drop and corresponding temperature drop occurs upstream of the secondary superheater. The steam then travels into the secondary superheater where the temperature is elevated to its full load limit of around 1005 degrees F. As a result, sliding pressure control allows constant temperature steam to enter the turbine at all plant loads. This increases turbine efficiency and reduces thermal cycle effects on the turbine.

Sliding Pressure Operation also provides the following benefits:

- Full ARC admission of steam to the turbine.
- Lower turbine thermal stresses.
- Faster load changes because of reduced temperature differentials.
- Improved overall plant heat rate.
- Lower minimum load capabilities.

Sliding pressure control is generally associated with larger supercritical pressure units, although drum style subcritical units can operate under sliding pressure control. The main reasons that supercritical units are better candidates for sliding pressure control than drum style units can be obtained from looking at the inherent differences between the two boiler types. Supercritical, once-through units are typically larger, harder to start-up, and more difficult to cycle than drum style units. Therefore, the benefits of quick load changes are of greater magnitude in a supercritical unit. The efficiency that sliding pressure control provides is of far greater concern to the utility for the larger supercritical units than the smaller drum units. Therefore, sliding pressure control will be discussed only as it concerns supercritical boilers in the upcoming sections.

In order to look at the control valves required for sliding pressure control, a basic understanding of the once-through start-up systems for various boiler manufacturers must be obtained. Start-up systems for Babcock & Wilcox, Combustion Engineering, and Foster Wheeler boilers will be discussed in the following sections. Although the benefits of sliding pressure control versus constant pressure control are many, the function of the valves in the start-up system are very similar. Generally, only the sizing of the valves is different between the two modes of operation.

The supercritical boiler is required to have a minimum flow inside the furnace waterwalls to prevent overheating of the boiler tubes. This flow must be established before firing of the boiler. A bypass system that is integral with the main steam, condensate, and feedwater systems is required so that the minimum design flow can be maintained at start-up and at times when the required minimum flow exceeds the turbine steam demand.

The bypass system performs the following additional functions:

- Reduces the pressure and temperature of the steam leaving the boiler during start-up to conditions that are suitable for the flash tank, condenser, and auxiliary equipment.
- Provides a way of recovering heat from the feedwater that flows to the bypass system by utilizing the feedwater heaters.
- Provides a means for conditioning the water during start-up without delaying boiler and turbine warming operations.
- Protects the secondary superheater against thermal shock from water during start-up.
- Provides a means for relieving excess pressure in the boiler during a load trip.

As previously mentioned, bypass arrangements differ considerably in detail, but are similar in their general design. The slight differences are required to accommodate the needs of the turbine as well as the supercritical boiler. The following sections describe each type of bypass system and sliding pressure control system in detail.

Sliding Pressure Control & Supercritical Start-up Systems for Babcock & Wilcox Boilers.

Babcock and Wilcox (B&W) has the largest number of supercritical boilers operating in the United States. Approximately 79 B&W Universal Pressure® Boilers are available for operation with the units ranging up to 1300 MW in size. During

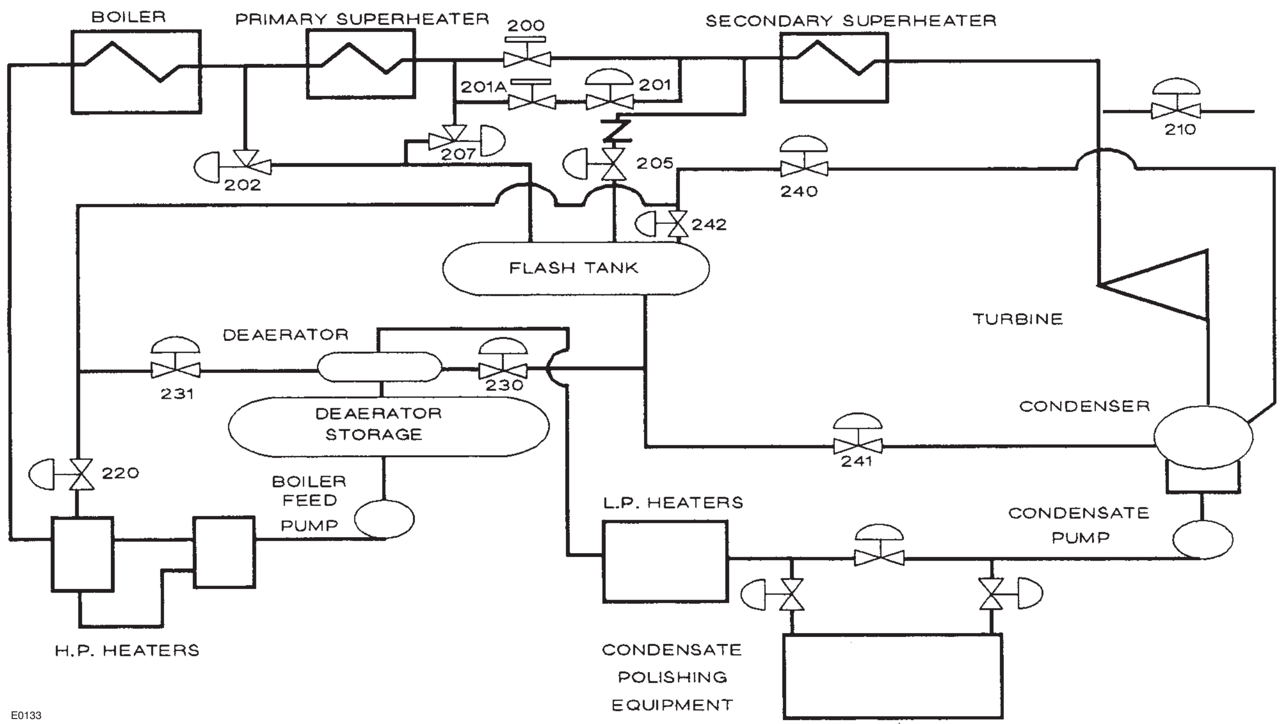

Figure 9B-6. *Babcock and Wilcox universal pressure boiler bypass system*

the course of a plant start-up, the Universal Pressure Boiler bypass system is operated in various modes. The following sequence describes the startup of a B&W once-through boiler. Figure 9B-6 shows a typical B&W Universal Pressure Bypass System.

Cold water clean-up mode. The cold water clean-up mode is used to adjust the water chemistry of the feed-water to predetermined values prior to firing the boiler. The boiler feedpumps are used to circulate approximately 25% of full load flow through the condensate polishing system for the fastest possible clean up of the cold water. This mode is used until the water cation conductivity is less than one micromho at the inlet to the economizer. The flash tank serves as a moisture separator during start-up to keep water from being carried through to the turbine, much like a steam drum in a conventional subcritical boiler.

Figure 9B-7 shows the arrangement of the bypass system in this mode. The following typical operating sequence is used for the cold clean-up mode.

1. Approximately 25% of full load flow is established.

2. BW-201, pressure reducing valve; BW-201A, pressure reducing stop valve; BW-200, high pressure superheater stop valve; and BW-207 secondary superheater (SSH) bypass valve are closed.

3. BW-202, primary superheater (PSH) bypass valve is set to maintain 600 psig at the outlet of the primary superheater.

4. BW-241, flash tank level control valve, is open dumping all the flow to the condenser and condensate polishing system. The flash tank will be flooded and pressurized during this mode to allow BW-241 to pass total start-up flow.

5. BW-242, flash tank steam stop valve, is kept closed until the flash tank level starts to fall below the flash tank center line. It then will open until the flash tank level hits a predetermined high level point.

6. Circulation is maintained in this manner until the fluid cation conductivity entering the economizer and at the entrance to BW-202 is below one micromho.

7. No firing of the boiler is done in this mode.

When sliding pressure control is employed, BW-202 valves are eliminated and the BW-207 valve is used to complete the function of the

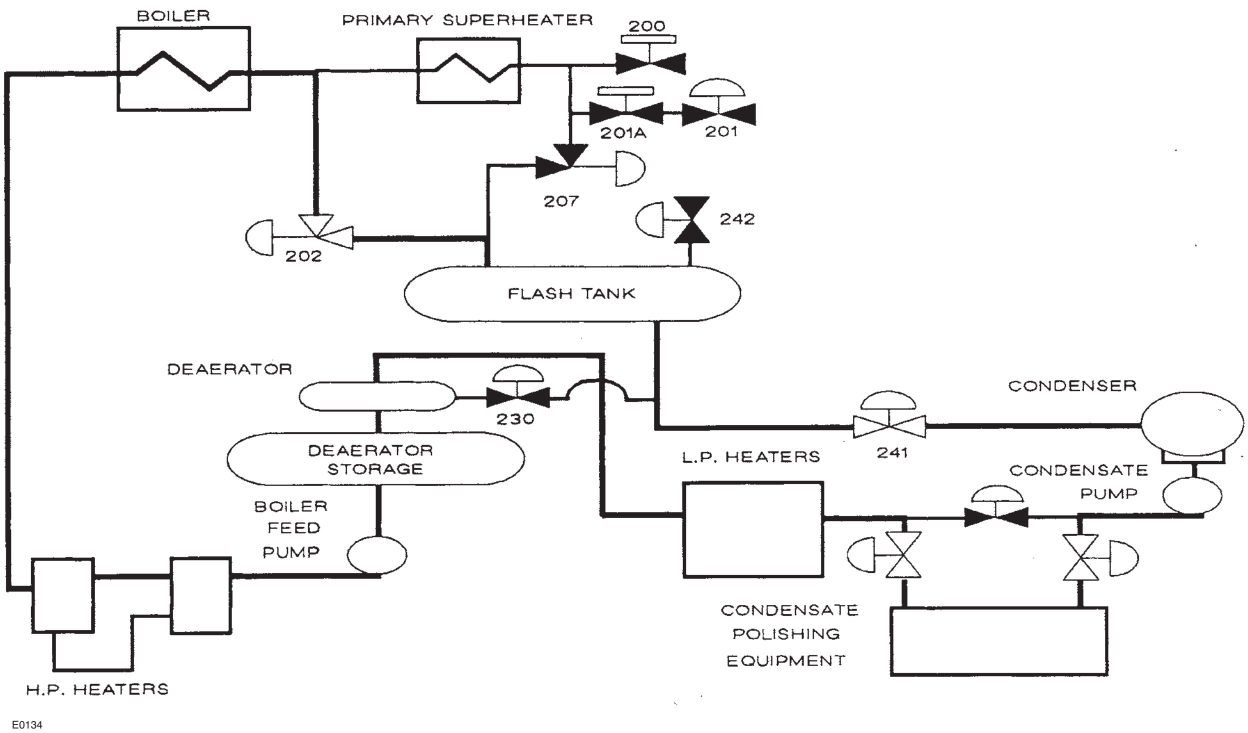

Figure 9B-7. *Cold water clean-up mode – constant pressure control*

BW-202. Figure 9B-8 shows a sliding pressure control unit in the cold water clean-up mode.

Initial firing mode. The initial firing mode begins when firing is initiated in the boiler and it is completed when steam is allowed to enter the secondary superheater.

Figure 9B-9 shows the bypass system in the initial firing mode. The following operating sequence is followed in this mode of operation.

1. Firing is initiated in the boiler.

2. All flash tank drain flow will be transferred from the condenser to the deaerator. BW-241 is held closed and BW-230, Deaerator Water Pegging Control Valve, is held open until the deaerator is pegged at its full load operating pressure, which is typically around 145 psig. After the deaerator is pegged, BW-230 will limit flow to the deaerator to maintain its pressure at setpoint.

3. BW-241 will control flash tank about its normal level after the deaerator is pegged.

4. When the fluid temperature at the primary superheater inlet exceeds 300 degrees F the primary superheater outlet pressure setpoint will be ramped automatically from 600 to 3650 psig so the unit will operate under supercritical conditions.

5. BW-202 operates to maintain 3650 psig at the primary superheater outlet.

6. At a fluid temperature of 300°F, BW-207 opens to a minimum position. As the fluid temperature leaving the primary superheater increases, BW-207 operates to maintain a programmed primary superheater outlet temperature.

7. BW-220, high pressure heater steam control valve, and BW-240, flash tank overpressure control valve, will open to limit the flash tank pressure at its setpoint of 500 psig.

8. During this period the secondary superheater will be boiling out to remove all the water.

9. The flash tank pressure increases as firing is continued.

The initial firing mode for a unit with sliding pressure control is the same as for constant pressure control except that the BW-207 valve replaces the BW-202 valve. Figure 9B-10 shows the initial firing mode for a sliding pressure control unit.

Initial turbine roll mode. Initial turbine roll begins with the opening of the BW-205, low pressure superheater non-return valve. The turbine is rolled and once synchronization is established, the unit load can be increased by using a ramp function.

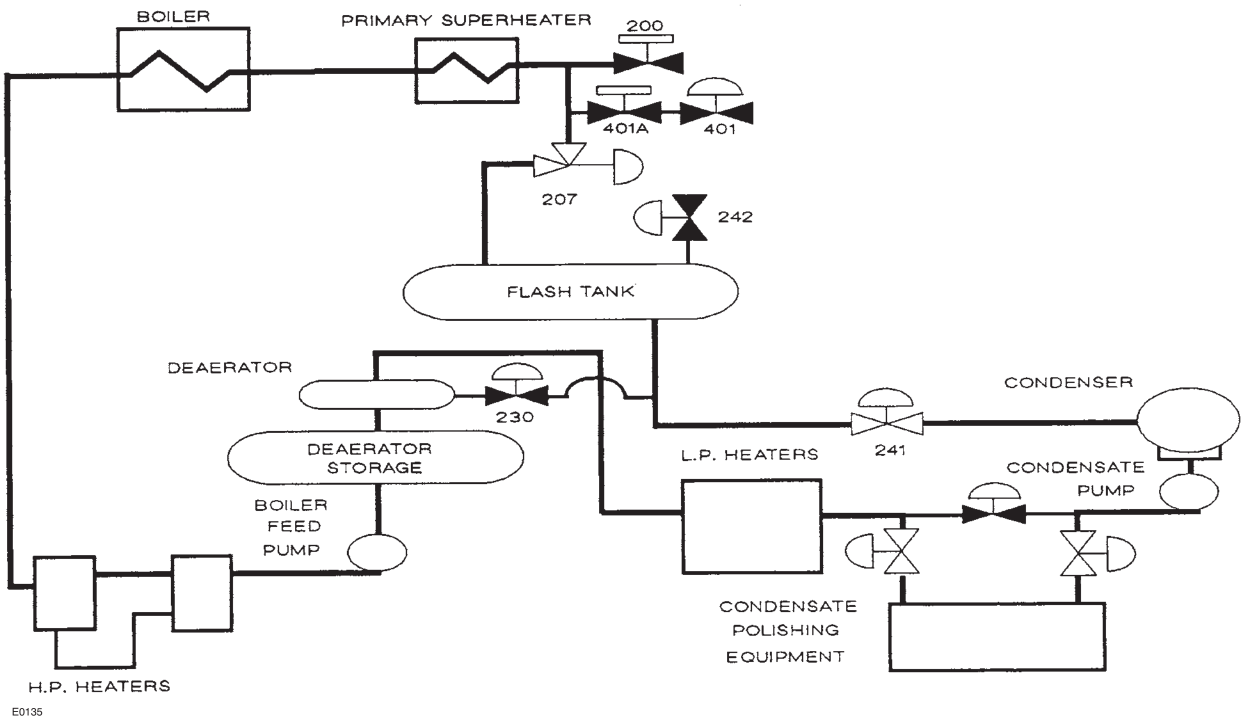

Figure 9B-8. *Cold water clean-up mode – sliding pressure control*

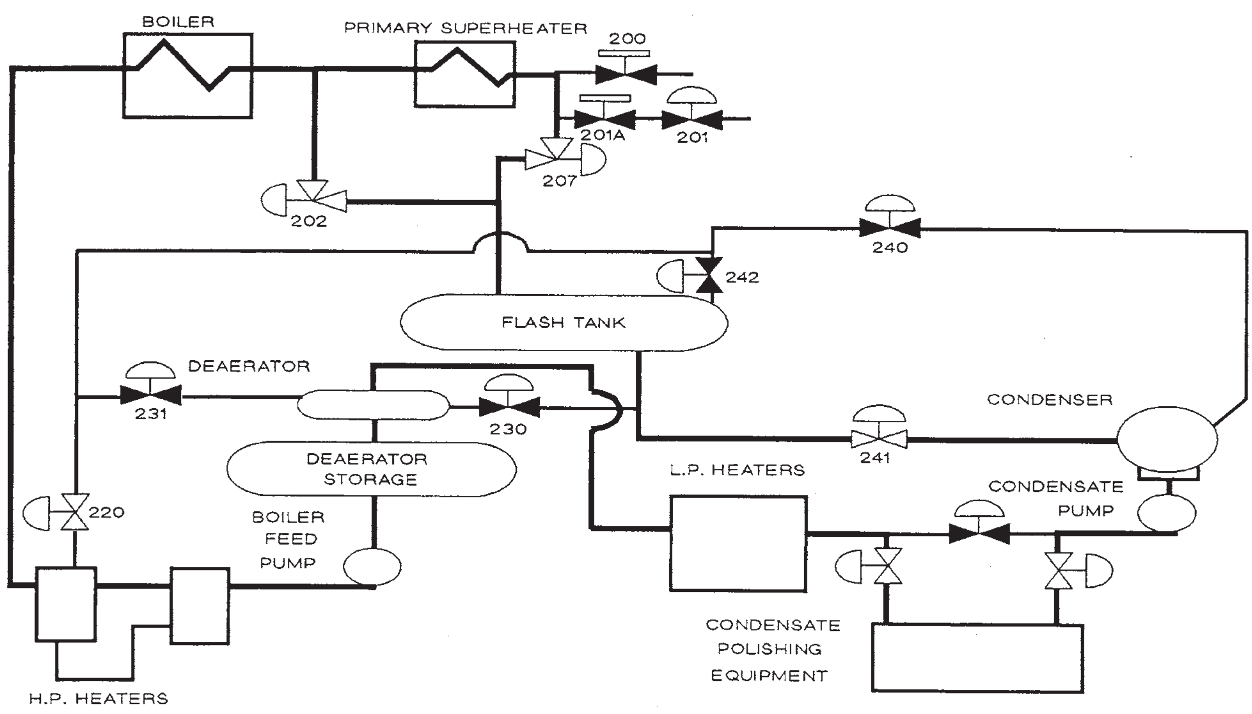

Figure 9B-9. *Initial firing mode – constant pressure control*

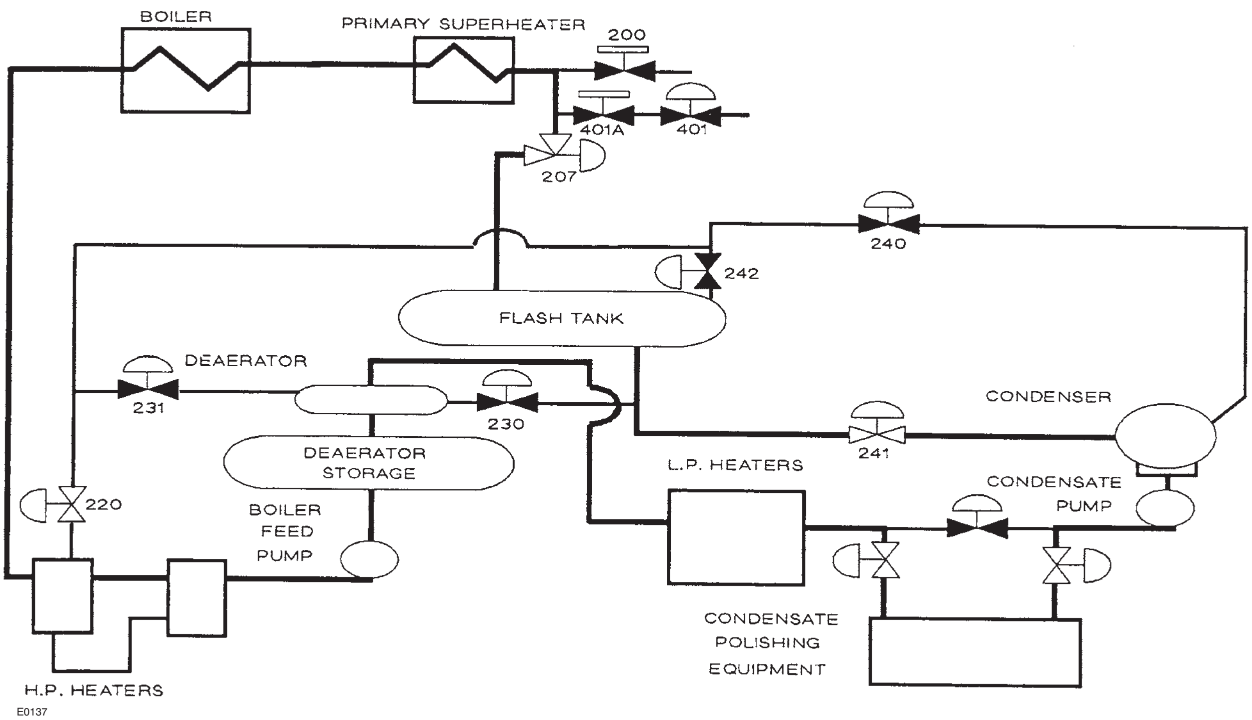

Figure 9B-10. *Initial firing mode – sliding pressure control*

Figure 9B-11 shows this mode of operation as the following sequence occurs.

1. At a flash tank pressure of 300 psig, BW-205, low-pressure superheater non-return valve, will open.

2. BW-210, turbine above seat drain valve is opened to pass approximately 2% of full load flow through the secondary superheater to warm the main steam lines. The flash tank pressure will continue to increase, as firing is continued, to its setpoint of 500 psig.

3. At the flash tank pressure of 500 psig, the turbine can be rolled. Approximately 2% of full load flow is required to roll the turbine. BW-210 should be kept open to pass an additional 2% of flow to the drains.

4. The firing rate should be adjusted until the gas temperature is approximately 50 degrees above the desired steam temperature to the turbine.

5. After the turbine steam requirements are met, BW-220 will open to limit the flash tank pressure at its setpoint of 500 psig.

6. BW-241 is still maintaining flash tank level. BW-230 is maintaining deaerator pressure.

7. When the capacity of BW-220 is exhausted, the steam pressure entering the turbine should be increased to 1000 psig. This will increase the flash tank setpoint to 1000 psig.

8. BW-220 and BW-240 are automatically set to hold the flash tank at its pressure setpoint of 1000 psig.

9. At a flash tank pressure of 1000 psig, the turbine can be synchronized and loaded. The unit load is ramped to approximately 7% load.

10. BW-210 is closed after the turbine is synchronized.

Figure 9B-12 shows the operation of a sliding pressure control unit in the initial turbine roll mode. As in both previous modes of operation, the only difference between normal operation and sliding pressure operation is in the use of the BW-207 valve instead of both the BW-202 and BW-207 valves.

Transfer to once-through operation mode. Up to this point in the start-up sequence, valves BW-201, BW201S, and BW-200 have been closed and all the flow has been through the BW-202 and BW-207 valves to the flash tank. During the transfer to once-through operation mode, this is reversed and the flash tank is isolated from the main steam system. Once this mode is completed, the flash tank is no longer used, and the primary job of the bypass system is complete. This mode in the start-up sequence is also where the main differences in constant pressure control and sliding pressure control become evident.

Figure 9B-13 shows a boiler in the transfer to once-through operation mode, but before the flash tank is taken out of service. Figure 9B-14 shows the boiler after the flash tank is taken out of service and at a minimum load operation that will be continued up to full load. Both these figures apply to a boiler operating under constant pressure control. The following operations sequence occurs in the transfer to once-through operation mode:

1. When the load on the turbine reaches approximately 7%, BW-201, pressure-reducing valve, BW-201S, pressure reducing stop valve, will begin to open. This will allow steam to flow directly to the secondary superheater rather than to the flash tank.

2. Pressurization of the secondary superheater occurs as BW-201 is opened. When the secondary superheater inlet pressure exceeds the flash tank pressure, BW-205 will close.

3. BW-201 will continue to be opened at a predetermined rate to allow the load on the turbine to increase to approximately 25% of full load.

4. As BW-201 opens, BW-207 will close to control the primary superheater outlet temperature at setpoint.

5. The flash tank drain flow to the deaerator will decrease as BW-201 is opened. The deaerator pressure will decay as the flash tank drains decrease. When the deaerator pressure decays below 25 psig, BW-231 will open to hold the deaerator pressure at 25 psig with flash tank steam. (Refer to Figure 9B-14)

6. The flow to the secondary superheater is through BW-201, until its full capacity is exhausted, which is typically around 25% of full load. The high-pressure superheater stop valve, BW-200, will then be fully opened to achieve full pressurization of the secondary superheater.

7. As BW-200 and BW-201 are opened, BW-202 will close to hold the primary superheater outlet pressure at 3650 psig.

8. As the flow to the flash tank decreases, the heaters and deaerator will be pressurized by steam from turbine extraction points.

9. As the load on the turbine reaches 25%, valves BW-202 and BW-207 will close and their opening setpoint will be 4250 psig. These valves will now act as relief valves during a unit trip.

10. BW-260, flash tank warming line non-return valve, will be opened to pass a small amount of steam from the deaerator back to the flash tank. This is required in order to keep the flash tank warm in case valves BW-202 and BW-207 open for overpressure relief. (BW-260 bypasses valve BW-231 and is not shown.)

11. During this time, BW-241 operates to maintain the flash tank level.

When sliding pressure control is utilized, the above sequence is followed up through Step 5. The main difference between constant pressure control and 100% sliding pressure control is that under 100% sliding pressure control, there is no BW-200 valve. All the steam is controlled by the sliding pressure control valve, BW-401. The BW-401 valve is then used to ramp the pressure to the turbine, which defines the plant load (Figure 9B-15).

When other than 100% sliding pressure control is used, such as 70% SPC (Figure 9B-5) the BW-201 is called the sliding pressure control valve and the BW-200 valve is still used. The BW-201 valve is sized to handle 70% of main steam load. Above this point, BW-200 is fully opened and the turbine throttle valves control load.

In summary, the capacity of the BW-201 valve determines the amount of sliding pressure control capability of a unit. If it is sized to handle 100% of the steam flow, it is called a BW-401 valve.

One of the secondary benefits of the B&W sliding pressure control system is obtained by adding two control valves.

The furnace bypass control valves, BW-263 and BW-264, are used in conjunction with lower furnace modifications that allow the minimum load of the unit to be reduced to 12% from the conventional 25%. Typically, 25% load is necessary to prevent ‘pseudo film boiling’ which causes sudden reductions in heat transfer and the potential of tube failure. With the addition of BW-263 and BW-264, two passes are made through the lower furnace which doubles the velocity which reduces the possibility of ‘pseudo film boiling.” The disadvantage is an increased pressure drop in the system created by the increased velocity. This increased pressure drop is reduced by the addition of furnace bypass control valves BW-263 and BW-264, and corresponding mixing bottles (Figure 9B-16).

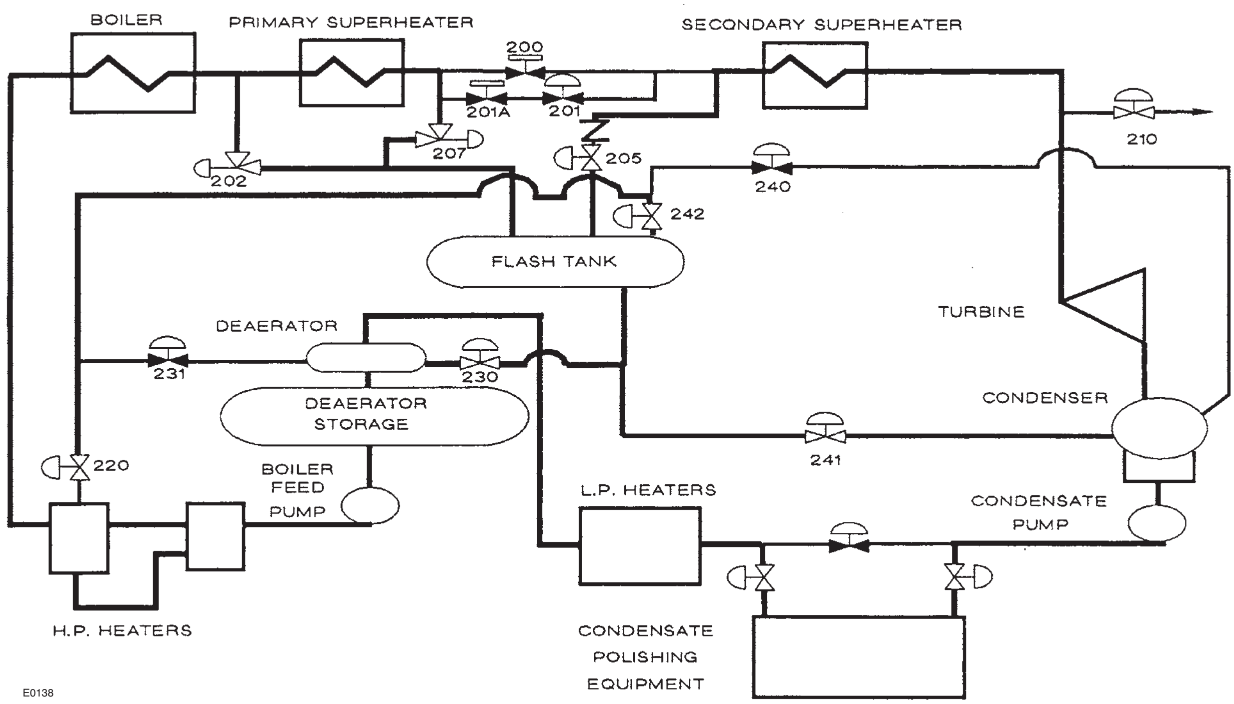

Figure 9B-11. *Initial turbine roll mode – constant pressure control*

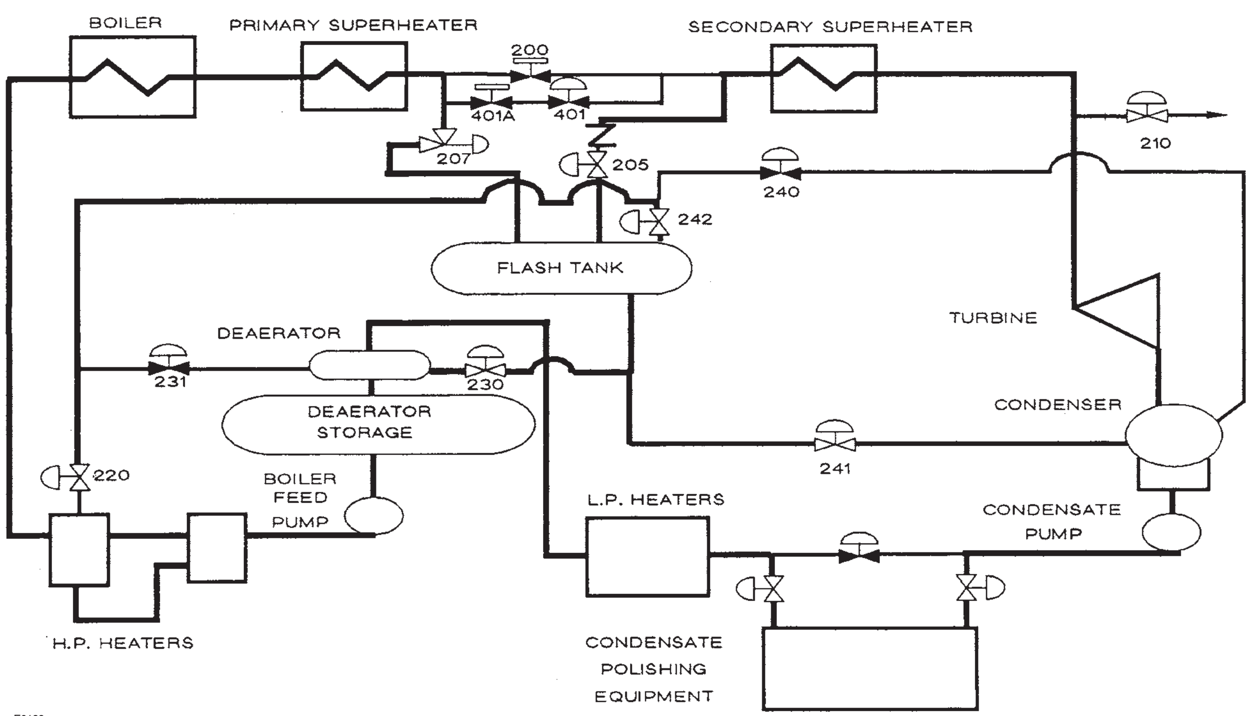

Figure 9B-12. *Initial turbine roll mode – sliding pressure control*

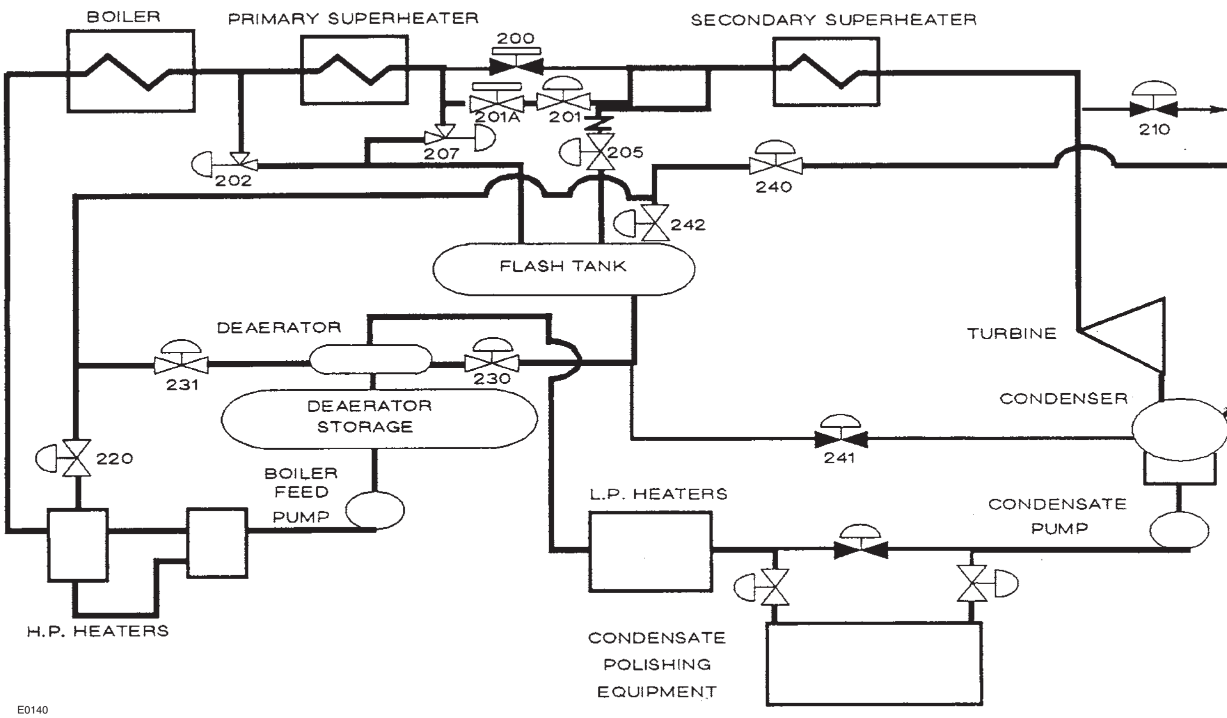

Figure 9B-13. *Transfer to once-through operation mode with the flash tank in service – constant pressure control*

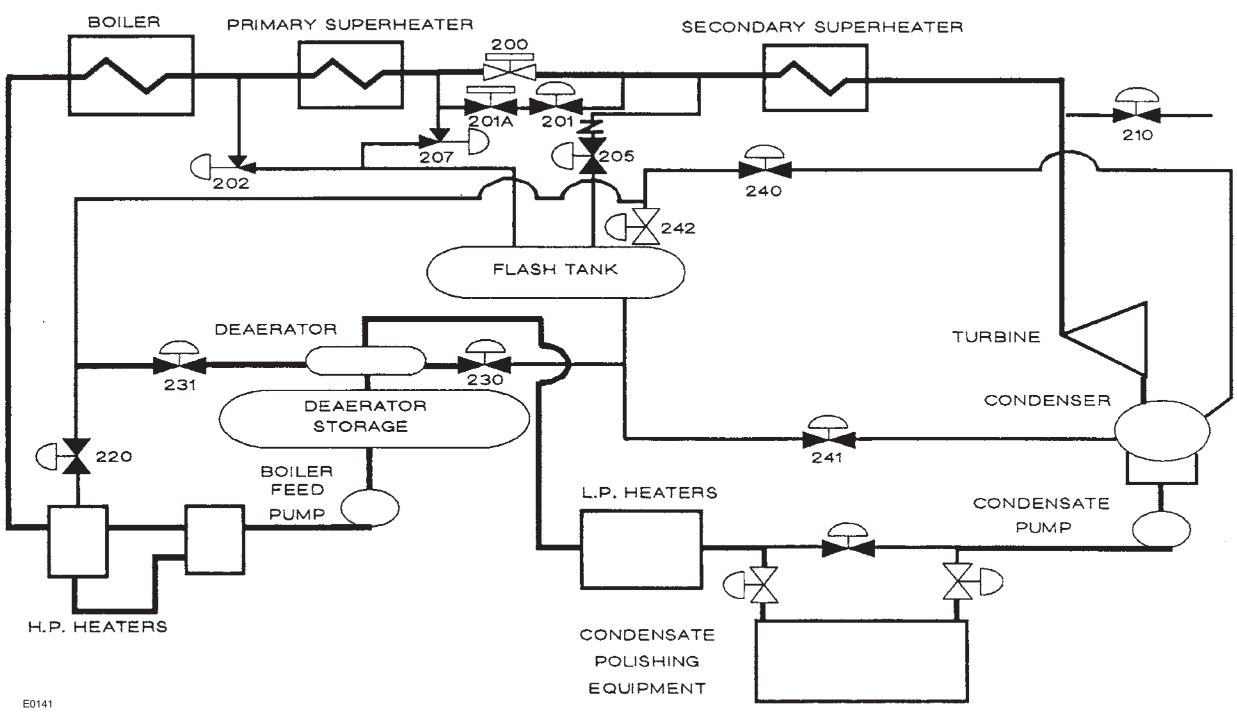

Figure 9B-14. *Transfer to once-through operation mode after the flash tank is out of service –sliding pressure control*

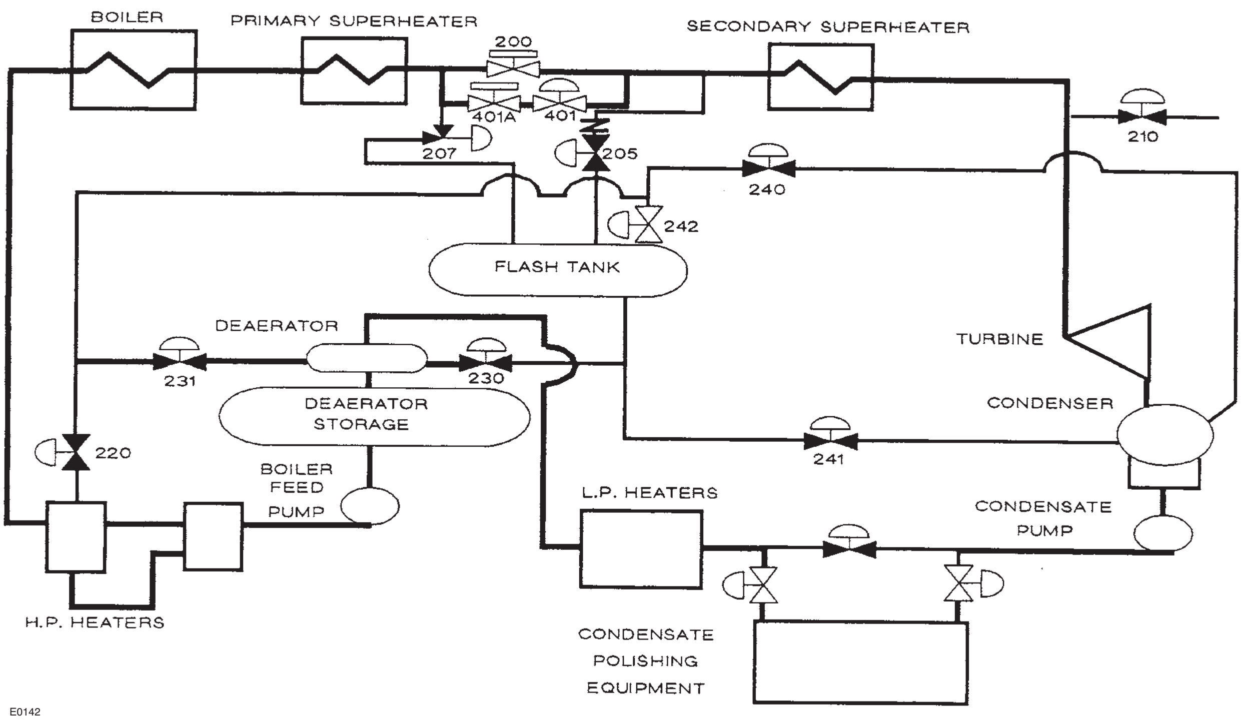

Figure 9B-15. *Sliding pressure control system – 12-1/2% to 100% plant load*

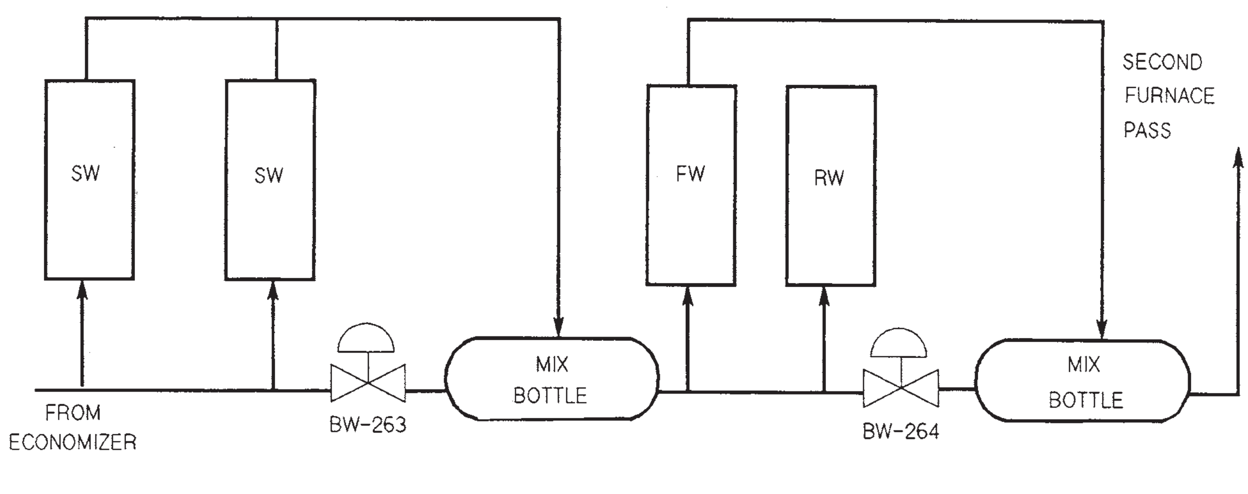

Figure 9B-16. Furnace bypass control valves

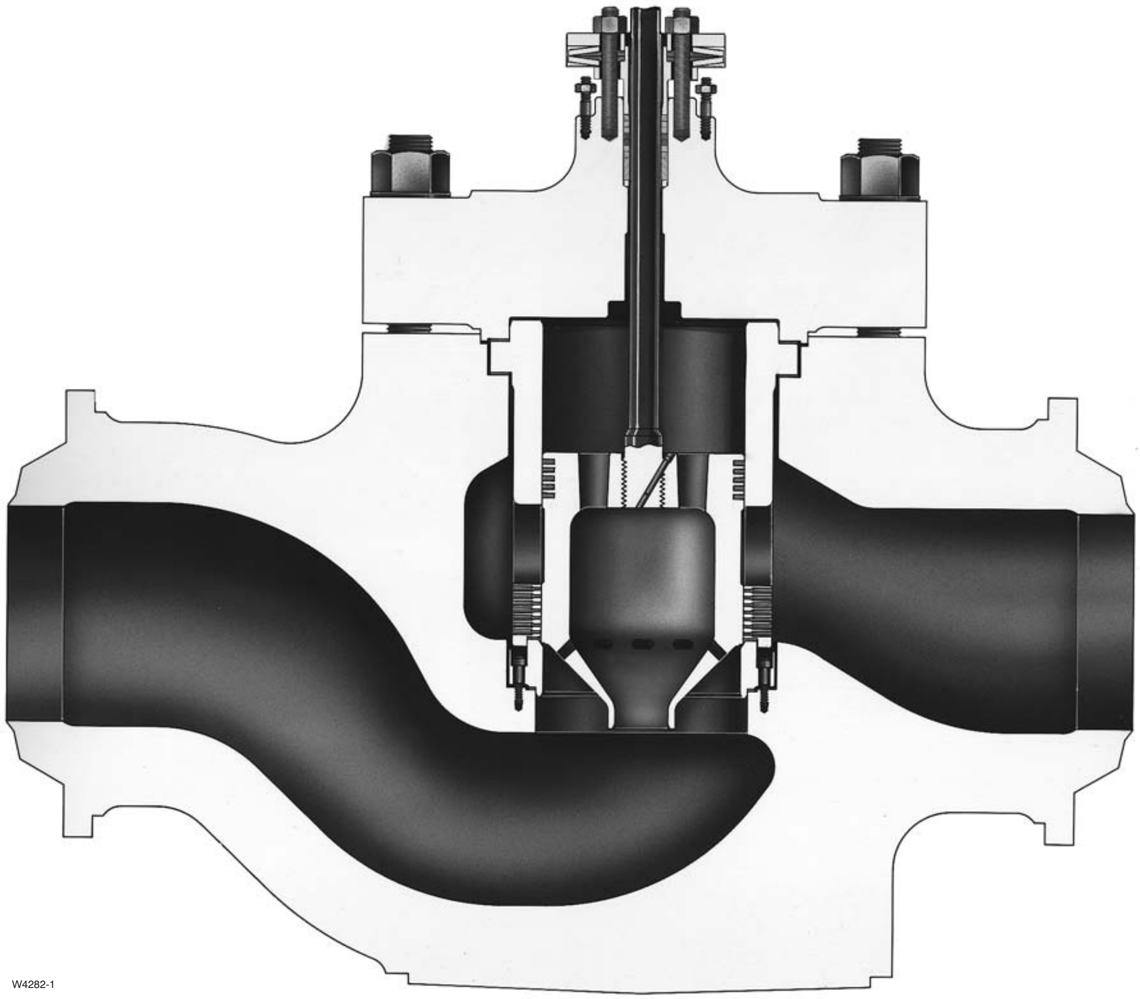

Figure 9B-17. *20-inch Design EHD for sliding pressure control*

BW-401 100% sliding pressure control valve. The BW-401, sliding pressure control valves perform the most critical functions in the sliding pressure control system. The unit availability and unit load is fully dependent on the successful operation of these valves. Since the BW-401 valves control the unit load from 12% to 100%, the trim design must have low travel noise abatement trim along with high top end capacity. These valves are typically large globe valves ranging in size from 8 to 20-inch depending on plant size and the number of valves. A special installed linear characteristic of drilled holes at the low end and milled windows at the high end are used to match the boiler flow characteristics. Typical service conditions for the BW-401 valves are as follows:

P_1 = 3965 psig

P_2 = 3940 psig

ΔP = 25 psid

T = 845°F

Q = Maximum boiler flow divided by the number of valves

The Fisher Design EHD with characterized Whisper Trim III meets the needs of this demanding service. Figure 9B-17 shows the Fisher 20-inch Design EHD that was designed specifically for sliding pressure control service. The coned plug tip eliminates damaging vibration and state-of-the-art materials are used to meet the high pressures and temperatures of supercritical units.

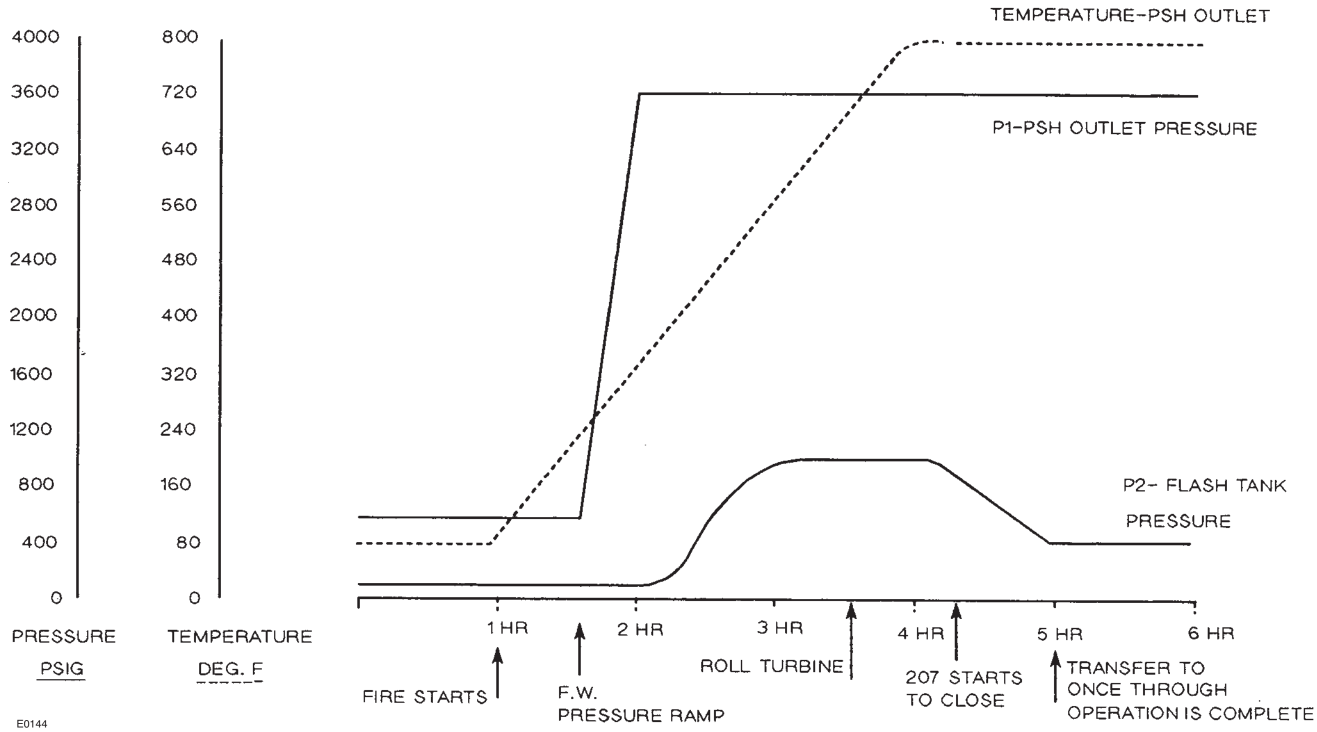

Figure 9B-18. *Typical service conditions for a BW-207, secondary superheater bypass valve*

The actuator is also very important in the design of a sliding pressure control valve assembly. It is Fisher Controls' experience that an electric actuator is best suited for the BW-401 service. The requirements for this service are as follows:

1. *Very precise positioning accuracy (0.5%).*
2. *Stiffness of operation to eliminate any position variances due to friction (packing, piston rings, etc).*
3. *Reliable positioner throttling capabilities.*
4. *Accurate stem position feedback.*
5. *Continuous throttling capability instead of on/off or multi-position service.*
6. *The actuator must fail in last position.*

An electric ball screw actuator with LVDT feedback meets all of these requirements.

BW-201 pressure reducing valve and sliding pressure control valve. It was previously mentioned that a normal operation unit and a sliding pressure unit's control operation varies mainly in the sizing of the BW-201 valve. Under normal operation, the BW-201 valve is used only to throttle the unit from approximately 12% to 25% of full load. The BW-200 valve is then opened and the turbine throttle valves control the unit. Sliding pressure control can be accomplished by sizing the BW-201 valves for 60% load (60% SPC), 70% load (70% SPC), etc. Typically, the sizing of the valve is accomplished by choosing a valve to provide approximately 25 psi pressure drop at the required flow rate. Above that flow rate, the valve capacity is being exceeded and useless pressure drop is introduced into the system. The following typical service conditions illustrate the sizing for constant pressure operation and 70% sliding pressure control.

Constant Pressure Operation

P_1 = 3650 psig

P_2 = 3625 psig

ΔP = 25 psid

T = 845°F

Q = 25% of maximum boiler flow divided by the number of valves

70% Sliding Pressure Control Operation

P_1 =3960 psig

P_2 = 3940 psig

ΔP = 25 psid

T = 845°F

Q = 70% of maximum boiler flow divided by the number of valves

The valve and actuator choice for the BW-201 service is a specially designed globe valve similar to the BW-401. The only difference is that the BW-201 service requires smaller size control valves.

BW-202 primary superheater bypass valve and BW-207 secondary superheater bypass valve. These bypass valves are discussed together since they operate at approximately the same service conditions (the BW-207 valve flow will be approximately 45°F hotter than the BW-202 valve). These valves are used to bring the unit from start-up through 25% load at which point the flow is transferred from the flash tank to once-through operation. These valves see a variety of services: cold water where cavitation occurs, hot water where flashing occurs, and superheated steam where noise abatement trim is required. Figure 9B-18 shows typical pressure and temperatures for a BW-207 valve. Although the BW-202 valve is not used in sliding pressure control systems, the sizing of the BW-202 and BW-207 valves are nearly the same as evidenced below.

Constant Pressure Operation

BW-202 Valve

Condition 1	Condition 2
P_1 =600 psig	P_1 = 3500 psig
P_2 = 100 psig	P_2 = 1000 psig
ΔP = 500 psid	ΔP = 2500 psid
T = 250°F	T = 800°F
Q = 10% of maximum flow per valve (usually 2 valves)	Q= 10% of maximum boiler flow per valve

BW-207 Valve

P_1 =3500 psig

P_2 = 1000 psig

ΔP = 2500 psid

T = 845°F

Q = 10% of maximum boiler flow (usually 1 valve)

Figure 9B-19. *Design CAV4 for applications where cavitation is a problem, such as superheater bypass.*

Sliding Pressure Control

BW-207 Valve

Condition 1	Condition 2
P_1 =600 psig	P_1 = 3500 psig
P_2 = 100 psig	P_2 = 1000 psig
ΔP = 500 psid	ΔP = 2500 psid
T = 250°F	T = 845°F
Q = 20% of maximum boiler flow	Q= 20% of maximum boiler flow

The BW-202 and BW-207 valves experience the most severe service conditions of any valve in the start-up system. The Fisher Design CAV4 is designed to handle the severe service posed by the BW-207 and BW-202 valves. This valve offers a four-stage pressure reduction, which prevents cavitation damage while also providing noise attenuation at the superheated steam condition. The angle valve configuration prevents the flashing condition from damaging the valve body. The stem balanced design with special soft metal seat or the C-seal design provide ANSI/FCI Class V shutoff for extended periods of time which prevents seat damage.

Globe valves using Cavitrol III four-stage trim can also be used to provide cavitation and noise protection for either the BW-202 or BW-207 service. The valve bodies are constructed using WC9 to combat problems associated with flashing and high temperatures. The valves will still provide tight shutoff using the C-seal trim design.

BW-240 flash tank pressure control valve. The BW-240 valve is typically a large, ANSI Class 600 valve such as the Fisher Design EUD or EWD. Depending on the particular plant, noise abatement trim may be required. Typical service conditions are as follows:

P_1 =1000 psig

P_2 = 0 psig

ΔP = 1000 psid

T = 550°F

BW-241 flash tank level control valve. The BW-241 valve will experience high pressure, flashing water as it controls the level of the flash tank. Generally, a large ANSI Class 600 Design EUD or EWD in WC9 material is used. An angle valve can also be used which will further minimize the damaging effects of flashing.

P_1 = 500 psig

P_2 = 0 psig

ΔP = 500 psid

T = 350°F

Q = 25% of maximum boiler load

BW-205 low pressure superheater non-return valve. The function of the BW-205 valve is to allow the initial steam to enter the secondary superheater and hence, the turbine. This valve also acts as a check valve to keep the main steam from entering the flash tank during once-through operation. It also has a secondary function of creating a pressure differential between the flash tank and the secondary superheater outlet, which is necessary for the operation of the SW-218, Secondary Superheater Attemperator Valve. Typical service conditions are:

P_1 = 1000 psig

P_2 = 900 psig

ΔP = 100 psid

T = 550°F

This valve will also be a large, high pressure valve such as the Fisher Design EHD.

BW-230 deaerator steam pegging control valve. The BW-230 valve allows flash tank drain flow to pass to the deaerator (DA) which maintains the DA at its setpoint. Flashing will occur in this valve. A small Fisher Design ES in WC9 material is used. Typical service conditions are:

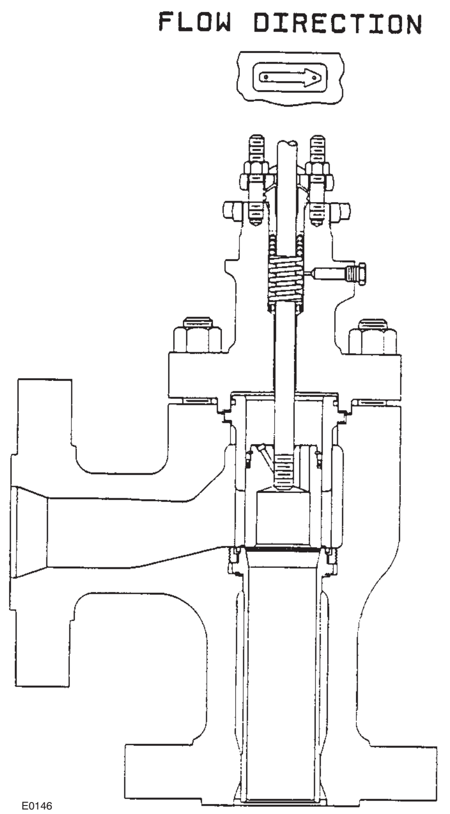

Figure 9B-20. *Design EHAT with optional liner*

P_1 = 500 psig

P_2 = 25 psig

ΔP = 475 psid

T = 350°F

BW-263 and 264 furnace 1st bypass control valves. The BW-263 valve is used along with the BW-264 valve to double the first pass fluid velocity in the furnace which reduces the unit minimum load from 25% to 12%. Both valves perform the same function, but the service conditions are slightly different as follows:

BW-263 Valve	BW-264 Valve
P_1 = 4300 psig	P_1 =4200psig
P_2 = 2800 psig	P2 = 4100 psig
ΔP = 1500 psid	ΔP = 100 psid
T = 600°F	T = 700°F

The BW-263 and BW-264 valves are generally large, high pressure ANSI Class 2500 valves. The

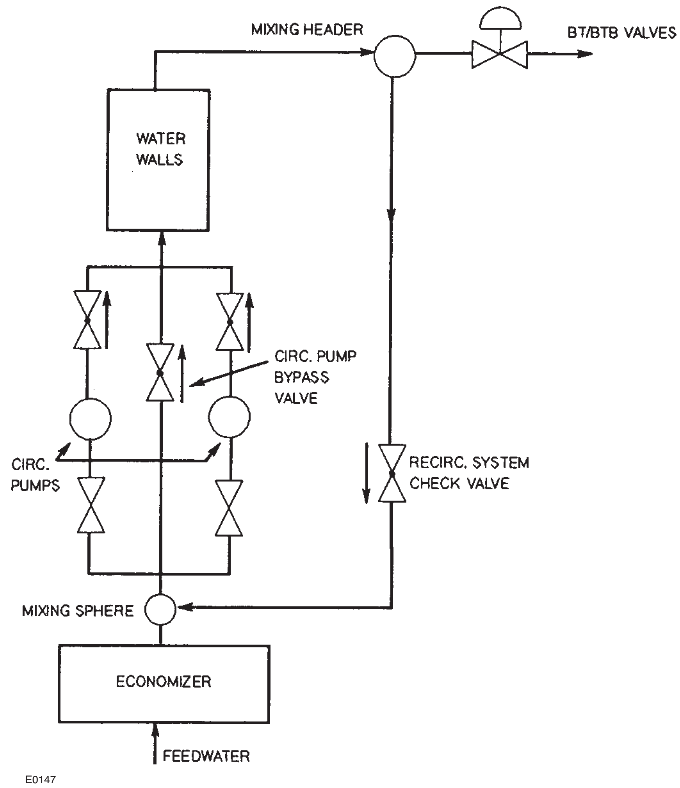

***Figure 9B-21**. CE integral recirculation system*

Fisher Design EHD with standard trim is an excellent choice for these services.

BW-218 secondary superheater attemperator control valve. The BW-218 valve permits steam from the flash tank to be introduced directly to the secondary superheater outlet steam, which reduces the steam temperature. A wide range of steam temperatures may be obtained by using this valve which allows for a good match between turbine metal temperature and steam temperature. Sizing of the BW-218 valve is essentially the same as the BW-205 valve:

P_1 = 1000 psig
P_2 = 900 psig
ΔP = 100 psid
T = 550°F

A valve such as the Fisher Design EHS or HPS provides the low flow control that is required to maintain exact steam temperature.

BW-219 reheat steam attemperator control valve. The BW-219 valve controls steam temperature at the outlet of the reheat section of the boiler by using main steam from upstream of the secondary superheater. Reheat attemperation is needed to 35% load due to the natural tendency of high reheat steam temperatures at low loads. Typical service conditions for the BW-219 valve are as follows:

Condition 1	Condition 2
P_1 =3950 psig	P_1 = 1000 psig
P_2 = 3500 psig	P_2 = 450 psig
ΔP = 450 psid	ΔP = 550 psid
T = 830°F	T = 550°F

The same valve design as the BW-218 valve can be used in the BW-219 valve service. The Fisher Design EHS or HPS with Micro-Form trim provides the necessary low flow control.

Sliding Pressure Control & Supercritical Start-up Systems for Combustion Engineering Boilers.

The Combustion Engineering Combined Circulation® supercritical boiler incorporates a start-up system that has many of the same features and benefits as the Babcock and Wilcox design. The main unique feature, which differentiates the CE unit from the B&W unit, is the integral recirculation system in the boiler that separates waterwall protection from low flow requirements. This recirculation system is shown in Figure 9B-21. Integral recirculation allows for lower minimum flow of approximately 10% of full boiler load flow which not only minimizes heat rejection during start-up but allows the transfer from the bypass system to once-through operation to take place without a sudden drop in steam temperature. For the CE unit, the transfer from recirculation to once-through operation occurs without operator intervention. Increasing water wall pressure drop that is due to increased flow causes a reversal of the pressure differential across the check valve in the recirculation line. Once a check valve is closed the operator has the option of leaving the recirculation pumps in service or shutting them down.

GUIDELINE SUMMARY
B & W Once-through Supercritical Boilers

Valve Description	Critical Parameters	Fisher Recommendations
BW-401 100% Sliding Pressure Control Valve	High Pressure High Temperature Low Travel Noise Rangeability	Large, High Pressure, Design EHD with an Installed Linear Characteristic (Whisper Trim III Characterized)
BW-201 Pressure Reducing Valve, Constant Pressure Control or Sliding Pressure Control Valve	High Pressure High Temperature Low Travel Noise Rangeability	Large, High Pressure, Design EHD with an Installed Linear Characteristic (Whisper Trim III Characterized)
BW-202 & BW-207 Superheater Bypass Valve	Cavitation Flashing Noise Tight Shutoff	Design CAV 4 or Cavitrol III 4-Stage Trim
BW-240 Flash Tank Pressure Control Valve	High Temperature Noise	Large Design EWD or EUD with Whisper Trim III or WhisperFlo Trim
BW-241 Flash Tank Level Control Valve	Flashing	Large Design EUD or EWD in Chrome Moly Material or EAD in Carbon Steel with a Liner
BW-205 Low Pressure Superheater Non-Return Valve	High Pressure High Temperature	Large Design EHD
BW-230 Deaerator Pegging Valve	Flashing Rangeability	Design ES
BW-263 Furnace First Bypass	High Pressure Supercritical Liquid	Large Design EHD
BW-264 Furnace Second Bypass	High Pressure Supercritical Liquid	Large Design EHD
BW-218 Secondary Superheater Attemperator Control	High Pressure High Temperature Low Flow Rates	Design EH or HPS with Micro-Form Trim
BW-219 Reheat Steam Attemperator Control	High Pressure High Temperature Low Flow Rates	Design EH or HPS with Micro-Form Trim

The concept of separating waterwall protection requirements from plant cycle requirements is also used in CE's sliding pressure units as shown in Figure 9B-22. The other main difference between the CE unit and the B&W unit is that the mainsteam control valves (boiler throttle and boiler throttle bypass valves) are located upstream of the initial superheater (primary superheater) instead of between the superheaters. Although the location of these valves is different for the two bypass designs, the function of the valves is essentially the same. The following can be used as a guide for comparing the function of the B&W to the CE bypass system valves.

CE Valves	B&W Valves
BE	BW-202, BW-207
BTB	BW-201
BT	BW-400 or BW-401
SA	BW-205
WD	BW-241
SP	BW-240
IS	BW-218
IR	BW-219

Some other differences between the bypass systems of Combustion Engineering boilers and Babcock and Wilcox boilers is from a terminology standpoint. CE uses initial and final superheaters and also start-up separator compared to B&Ws primary and secondary superheater and flash tank. Figure 9B-23 shows a schematic of the CE bypass system.

Figure 9B-24 through Figure 9B-29 shows a start-up sequence of a CE combined circulation unit. The BE, boiler extraction valve, is used to maintain the waterwall furnace pressure at its setpoint of approximately 3500 psi. The BTB, boiler throttle bypass valve, and the BT, boiler throttle valve, are used to control the throttle pressure through 30% load for constant pressure operation or 80% load for sliding pressure operation. Figures 30 and 31 show the sequencing of the BE, BTB, and BT valves for constant pressure operation and sliding pressure operation respectively. The furnace wall outlet pressure is considered the inlet pressure for all the valves and the initial superheater inlet pressure is considered the downstream pressure for the BT and BTB valves. The throttle pressure before turbine synchronization is the downstream pressure for the BE valves. Comparing the constant pressure operation to the sliding pressure operation the only difference is the sizing of the BT valves. For sliding pressure control, the valves are sized to handle approximately 80% of the maximum steam flow of the boiler with approximately 40 psi pressure drop compared to handling 30% maximum plant steam flow load with constant pressure operation. Figure 9B-32 further defines the BE/BTB transfer point and the BTB/BT sequencing of operation.

BT boiler throttle valve. The BT valves are similar to the BW-401 or the BW-201 valve in the function of the valve. Under constant pressure operation the BT valves are sized to handle approximately 30% plant load which corresponds in operation to the BW-201 valve. Above 30%, the valves would be wide open and will take an increasing pressure drop as the flow rate increases. For sliding pressure control the BT valves are sized to handle approximately 80% plant load. Above 80% the turbine throttle valves are used to control the plant load and essentially this would be considered an 80% sliding pressure control system. Typical service conditions for the BT valves are as follows:

Constant pressure operation:

P_1 = 3500 psig

P_2 = 3460 psig

ΔP = 40 psid

T = 800°F

Q = 30% of maximum steam flow divided by the number of valves

Sliding pressure control operation:

P_1 = 3900 psig

P_2 = 3860 psig

ΔP = 40 psid

T = 800°F

Q = 80% of maximum steam flow divided by the number of valves

For the BT valve service, a design such as the 8”-14” Design EHD is the preferred solution. The hung cage design and state-of-the-art materials negates the effects of thermal cycles that could otherwise lead to early fatigue of the valve.

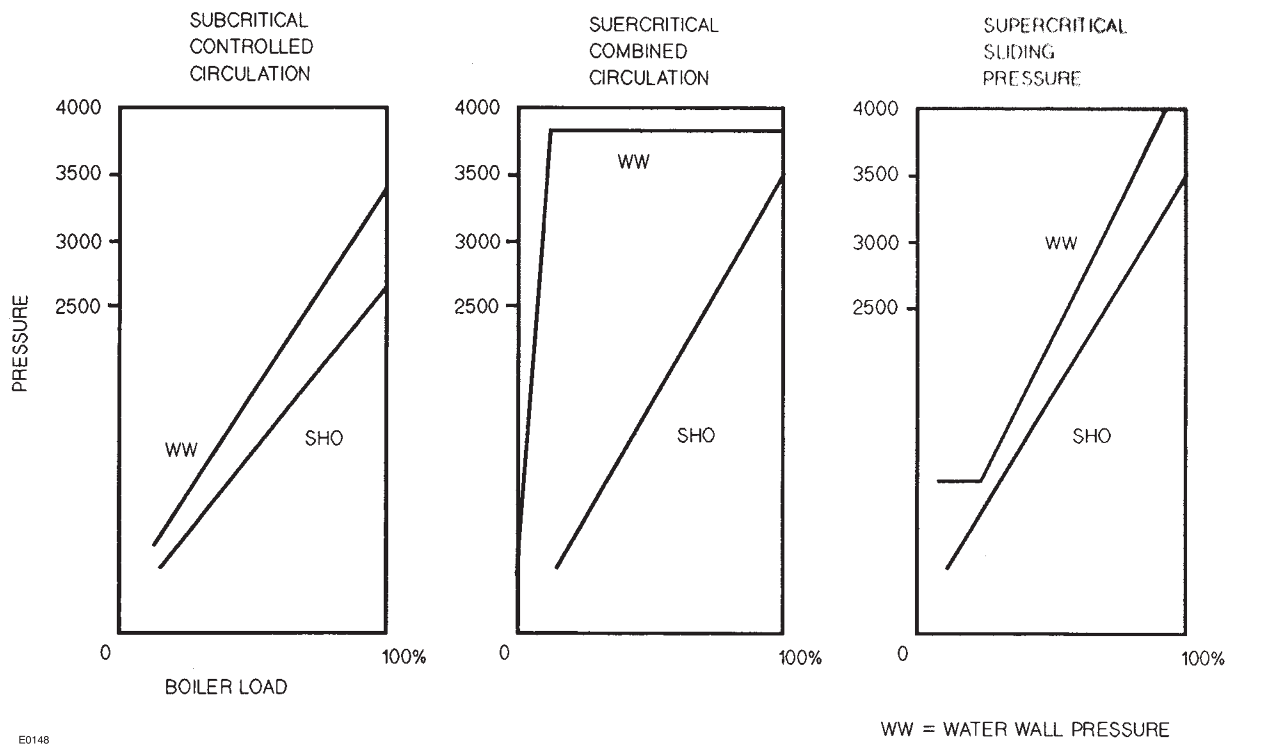

Figure 9B-22. *Sliding pressure operation*

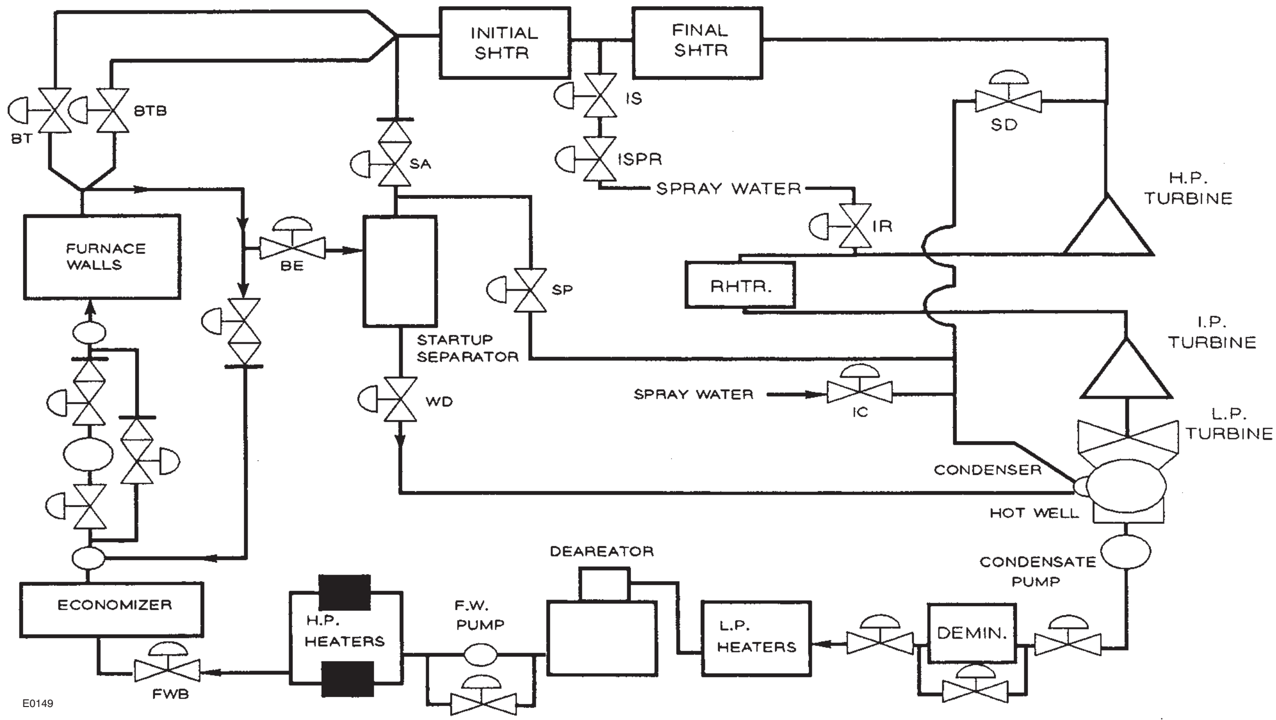

Figure 9B-23. *CE boiler bypass system*

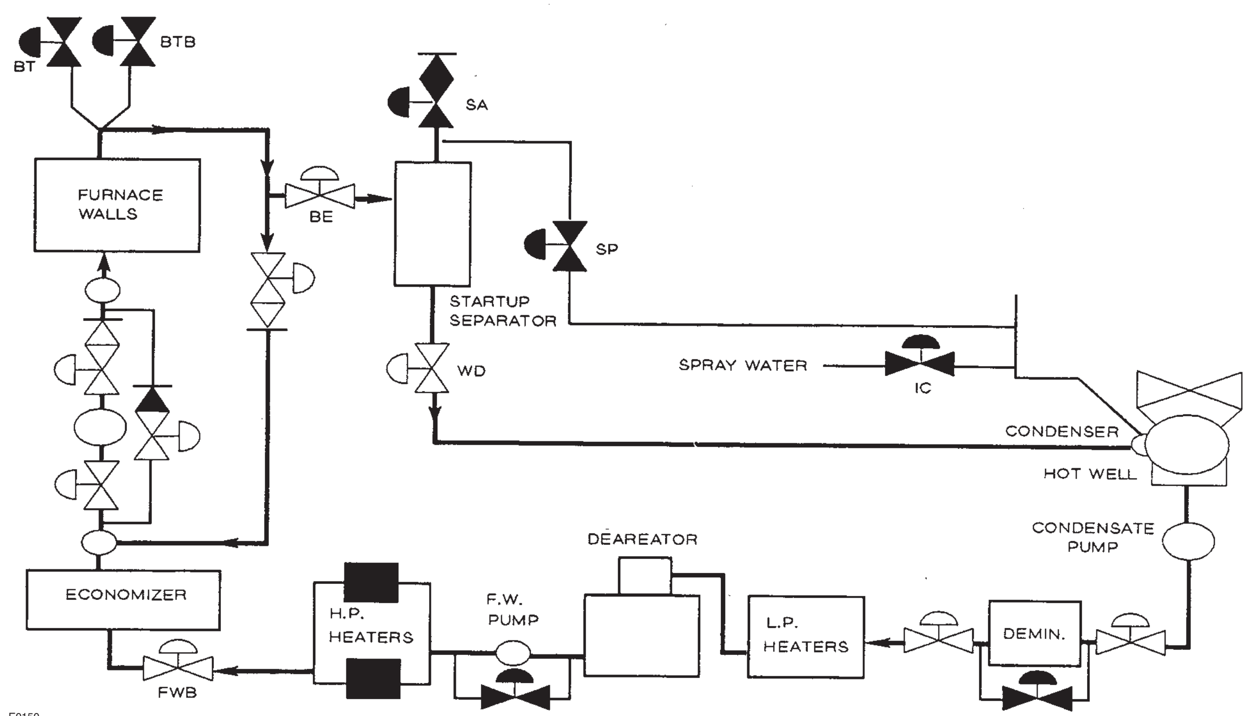

Figure 9B-24. *Establishing flow preliminary to firing*

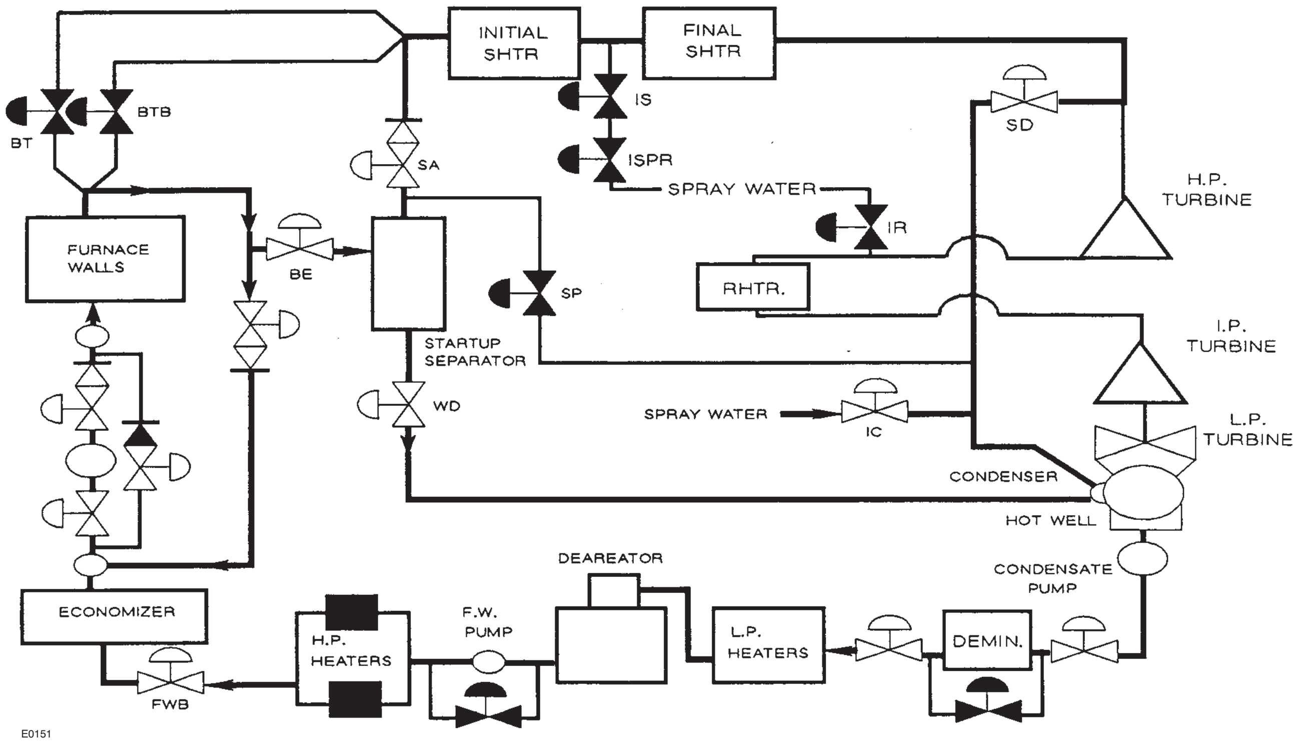

Figure 9B-25. *Lighting off and warming up*

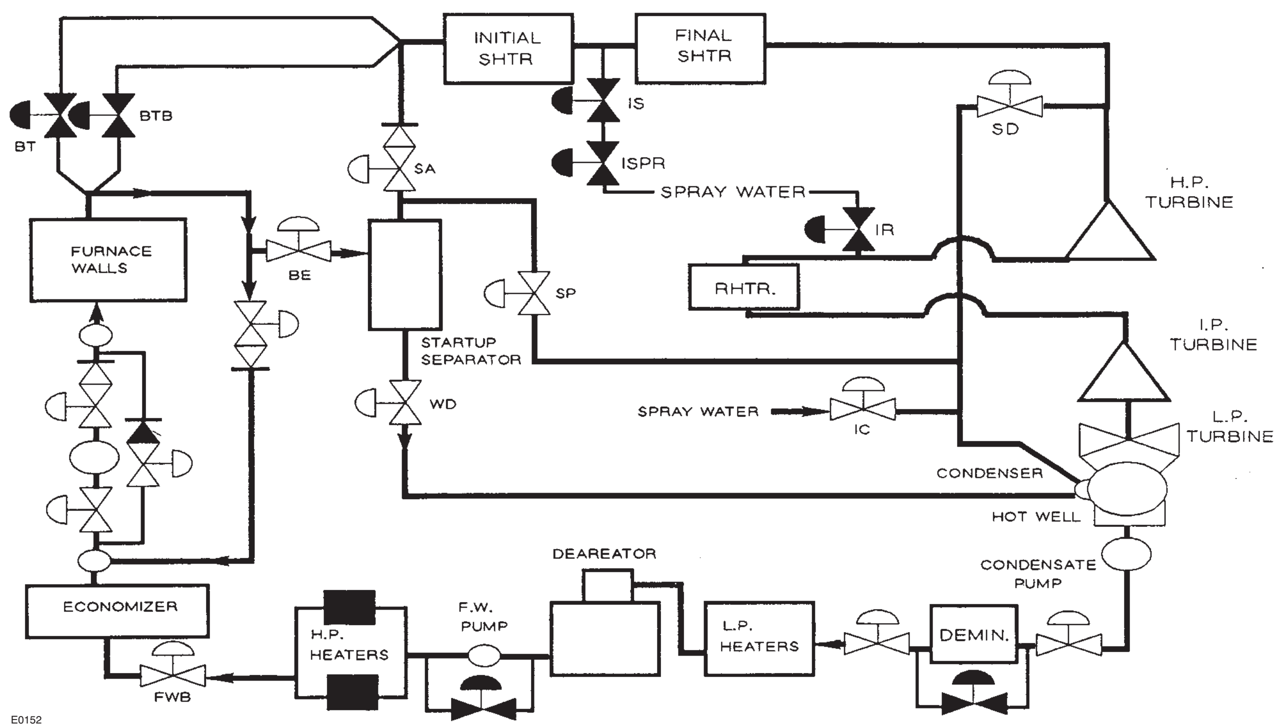

Figure 9B-26. *Turbine rolling and synchronizing*

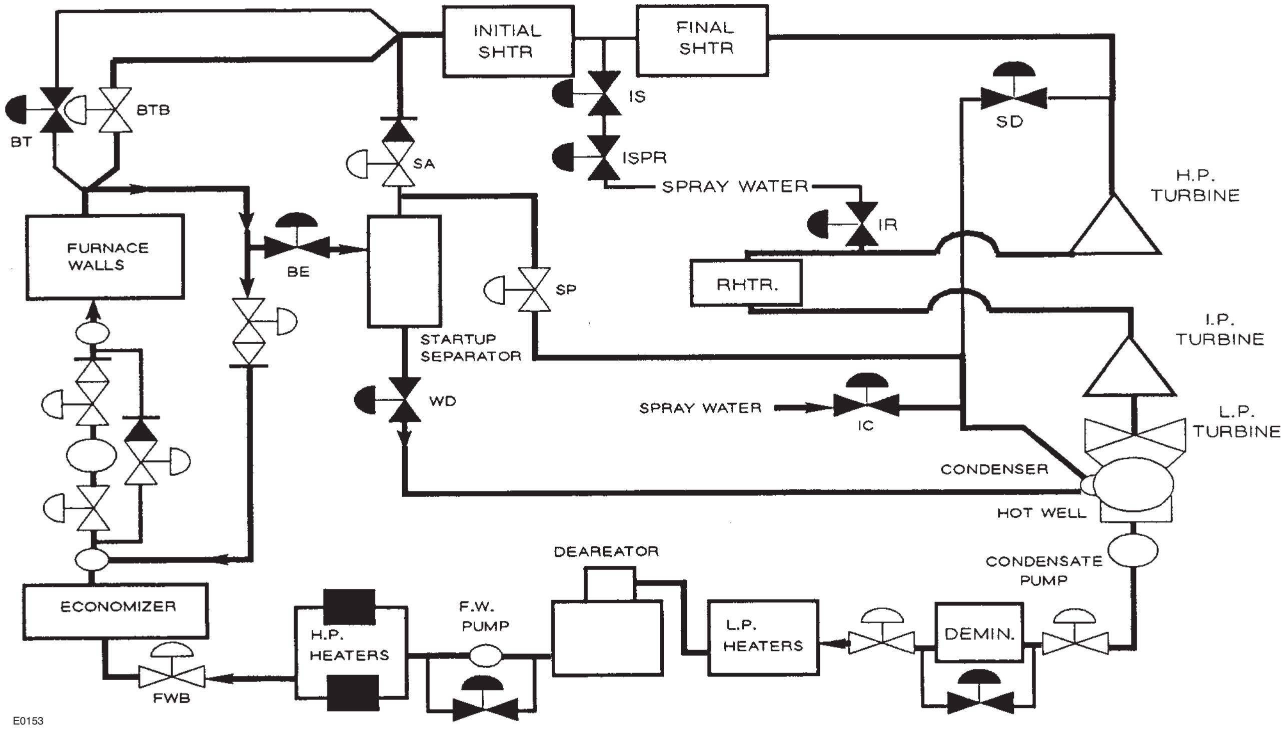

__Figure 9B-27__. Initial turbine loading (synchronizing to 10%)

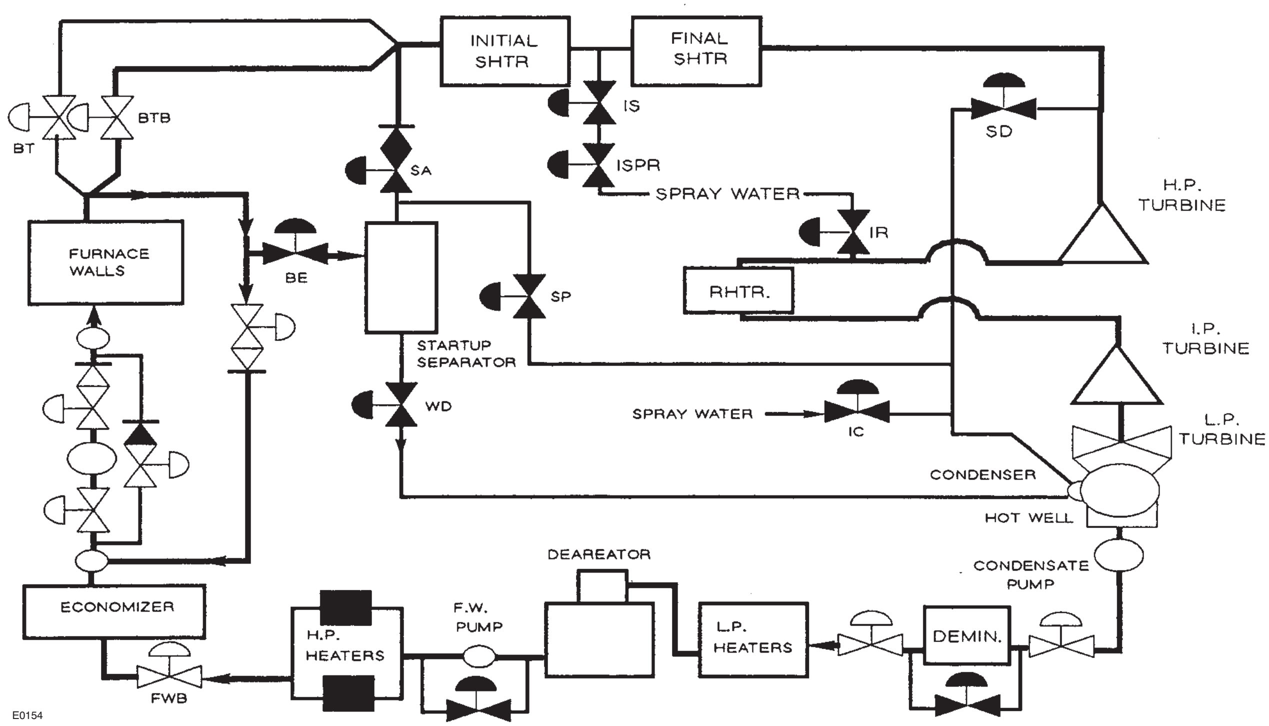

***Figure 9B-28.** Initial turbine loading (10 – 20% load)*

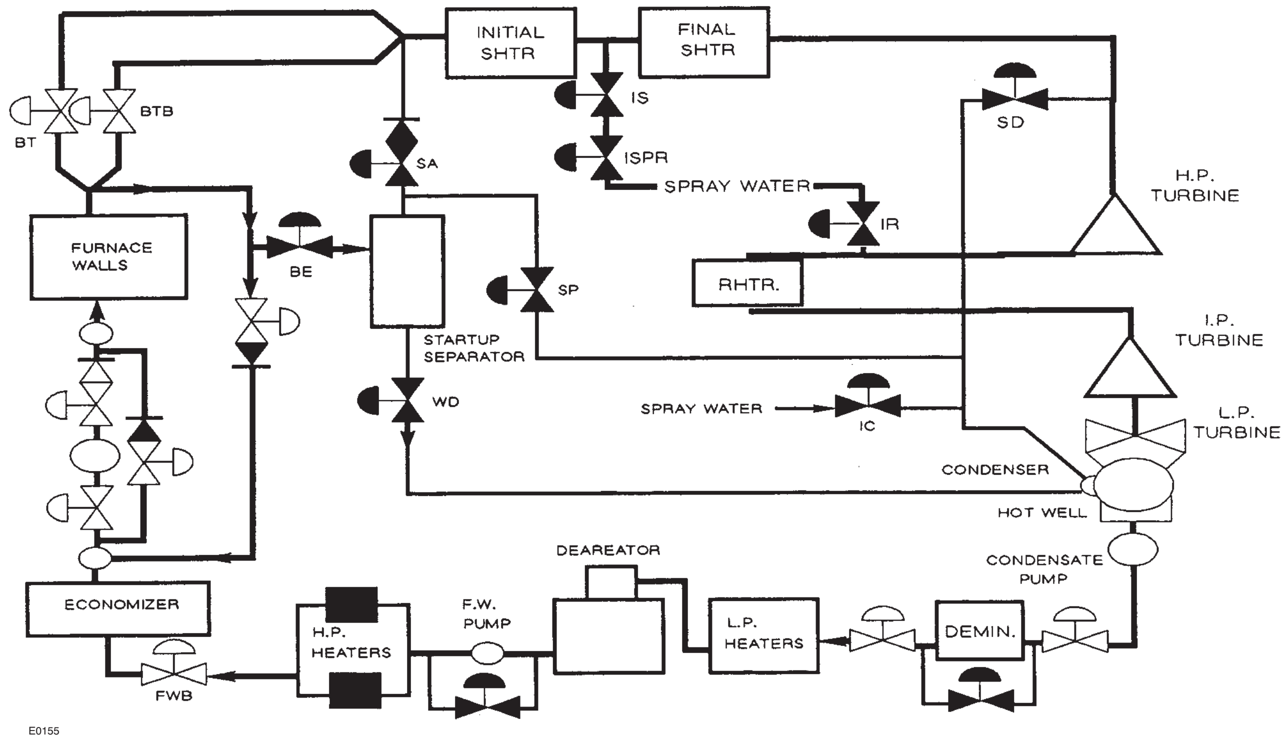

Figure 9B-29. *Turbine loading above 20%*

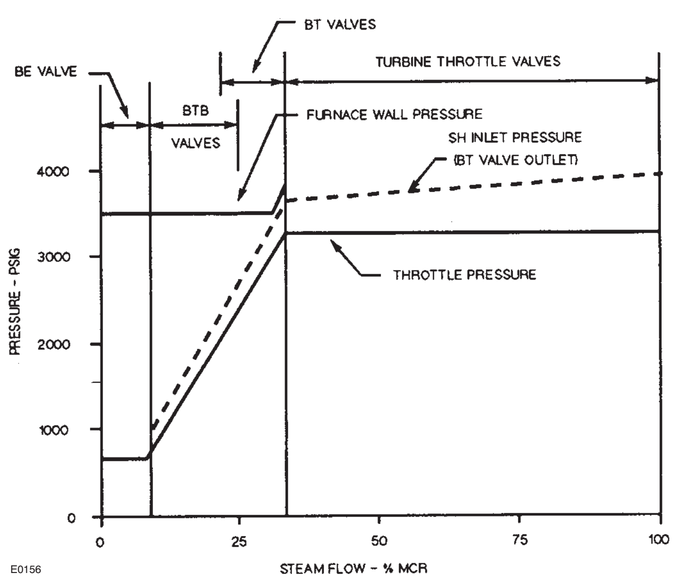

Figure 9B-30. *Constant pressure operation*

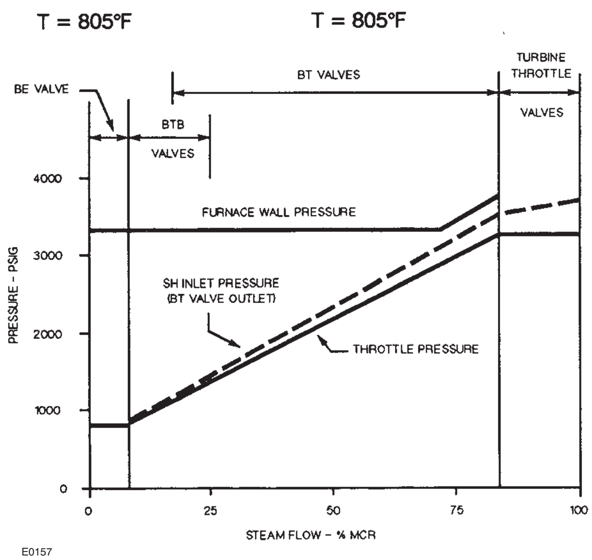

Figure 9B-31. *Sliding pressure operation*

BTB boiler throttle bypass valve. Boiler throttle bypass valves are similar to the BW-201 valve in the constant pressure operation. These valves are used during the initial turbine loading to maintain the furnace wall pressure setpoint of 3500 psig. The BTB valves are used in sequence with the BT valves to bring the unit on-line. These BTB valves are generally smaller valves but of the same design as the BT valves. Generally, a middle size Design EHD with WhisperTrim III style of trim is required due to the noise attenuation requirement of these valves. Typical service conditions are as follows:

<u>Condition 1</u>	<u>Condition 2</u>
P_1 =3500 psig	P_1 = 3500 psig
P_2 = 1000 psig	P_2 = 2500 psig
ΔP = 2500 psid	ΔP = 1500 psid
T = 805°F	T = 805°F

BE boiler extraction valve. The BE valve is used to control the furnace wall pressure at approximately 3500 psig during start-up. This valve passes water and eventually steam from the furnace walls to the start-up separator, which is at significantly lower pressure than the furnace pressure. The BE valve will see cold water where cavitation occurs, hot water where flashing occurs, and then high temperature, high pressure steam with very high pressure drops. In this condition the BE valve must provide some noise attenuation. The BE valve also acts as a safety relief valve for the furnace in case of a unit trip. Typical service conditions for the BE valve are as follows:

<u>Condition 1</u>	<u>Condition 2</u>
P_1 =3500 psig	P_1 = 3900 psig
P_2 = 750 psig	P_2 = 1000 psig
ΔP = 2750 psid	ΔP = 2900 psid
T = 200°F	T = 805°F

FWB feedwater bypass valve. The FWB valve is used to control feedwater flow during start-up and at low loads. It is generally used up to approximately 20% plant load. Above that point, feedwater control is transferred from the FWB valve to the boiler feedpump speed control. The FWB closes on a unit trip to prevent feedwater from continuing to flow to the boiler. Typical service conditions for the FWB valve are as follows:

P_1 = 3700 psig
P_2 = 3400 psig
ΔP = 300 psid
T = 200-500°F

The Fisher Design HPT or EHT is an excellent choice for the FWB service. Typically this is a large valve with standard trim in equal percentage or modified equal percentage characteristic.

SP spillover valve. The SP valve is similar to the BW-240 valve in that it limits pressure in the start-up separator to approximately 950 psig. Above that point the SP valve automatically opens. Typical service conditions for the SP valve are as follows:

P_1 = 950 psig
P_2 = 0 psig
ΔP = 950 psid
T = 950°F

The SP valve will typically be a medium to large size, high-pressure valve such as the Fisher Design EUD. Noise attenuation trim will probably be required due to the high-pressure drop conditions of this valve.

WD water drain valve. The WD valve is used to maintain the level in the start-up separator during

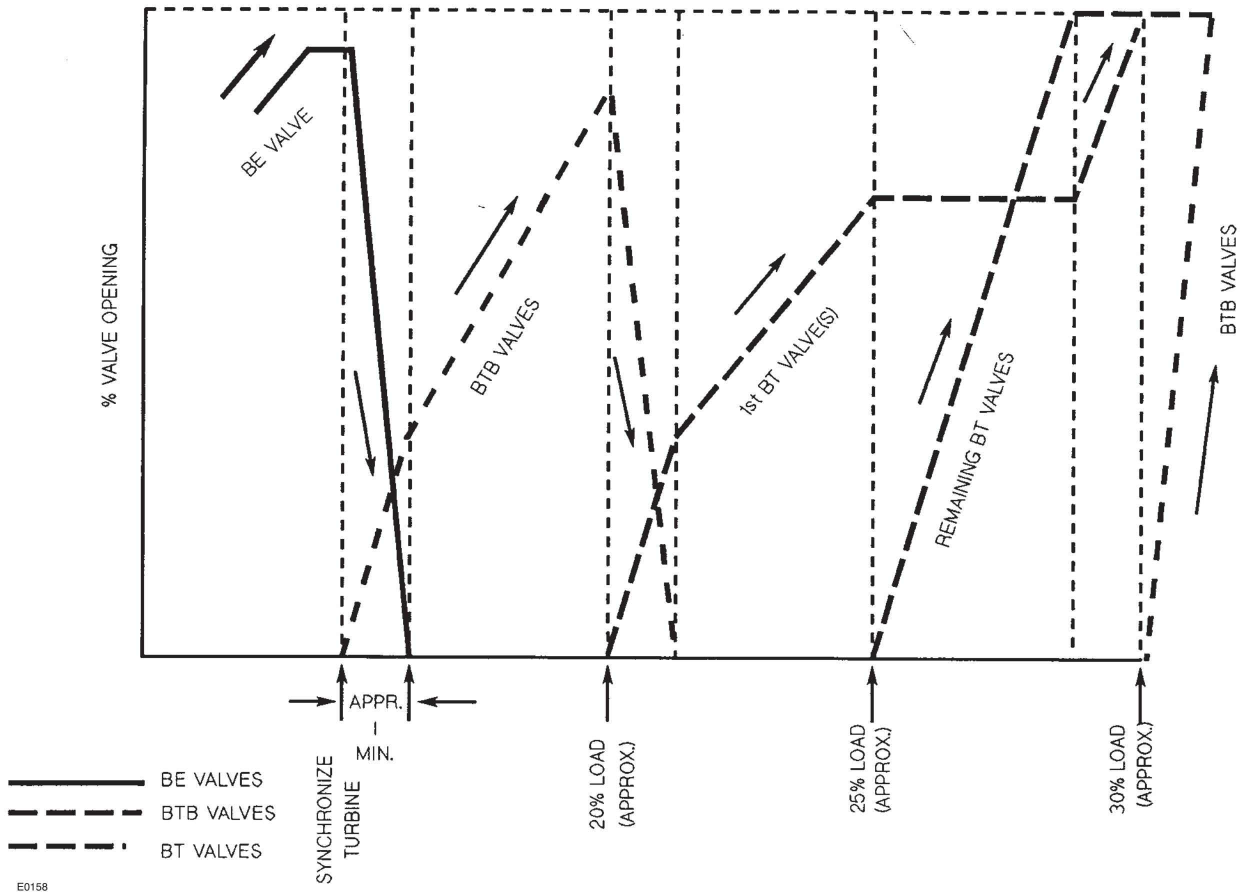

Figure 9B-32. *BE/BTB transfer and BTB/BT sequencing*

start-up. It automatically opens on high level in the separator. The WD valve passes the water from the start-up separator to the condenser, which is at or near a vacuum. Therefore, flashing will occur across the WD valve. Either a globe valve in WC9 Chrome-Moly body material or an angle valve in Carbon Steel body material with a liner should be used to minimize the damage caused by flashing in this service. Typical service conditions for this valve are:

P_1 = 35 psig

P_2 = 0 psig

ΔP = 35 psid

T = 225°F

SA steam admission valve. The steam admission valve is similar to the BW-205 valve and is used to pass steam from the start-up separator to the inlet of the initial superheater during start-up. The SA valve is open when start-up pressure reaches a predetermined value, typically 100 psig. Once the valve is open and transfer to once-through operation is complete, the SA valve acts as a standard check valve to prevent flow through the BT and BTB valves from returning to the start-up separator. Typical service conditions for the SA valve are:

P_1 = 100 psig

P_2 = 0 psig

ΔP = 100 psid

T = 950°F

Typically, the SA valve is a large, high pressure flow valve such as the Design EHD.

IS, IR, IC superheat, reheat and condenser spray valves. The IS, IR, and IC valves are used to control steam temperature at various locations in the main-steam system. The IS valve controls steam temperature at the outlet of the initial superheater, the IR valve controls steam temperature at the inlet to the reheater and the IC valve controls steam temperature at the inlet to the condenser. Although the pressures for each one of these valves may be different, the problem associated with each of these valves is the same. The valve is called on to throttle at very low lifts, so valves such as the Fisher Design HPS or EHS Series are used in this service. Figure 9B-34 shows an HPS with special micro-form trim for low flow control. Typical service conditions of these valves are:

IS Valve:

P_1 = 4000 psig

P_2 = 3900 psig

ΔP = 100 psid

T = 500°F

IR Valve:

P_1 = 2000 psig

P_2 = 1850 psig

ΔP = 150 psid

T = 400°F

IC Valve:

P_1 = 330 psig

P_2 = 250 psig

ΔP = 100 psid

T = 120°F

ISPR superheater injection water pressure-reducing valve. The ISPR valve is used to control pressure at the inlet of the IS valve. This valve also regulates the pressure drop across the IS valve. In case of a unit trip, this valve automatically closes to reduce the chance of water getting in the main steam line and creating water hammer. Typical size and conditions for the ISPR valve are:

P_1 = 4000 psig

P_2 = 3900 psig

ΔP = 50 psid

T = 500°F

The ISPR valve is typically a small, high-pressure valve with standard trim. The Fisher Design HPS with standard trim meets the requirements of this service.

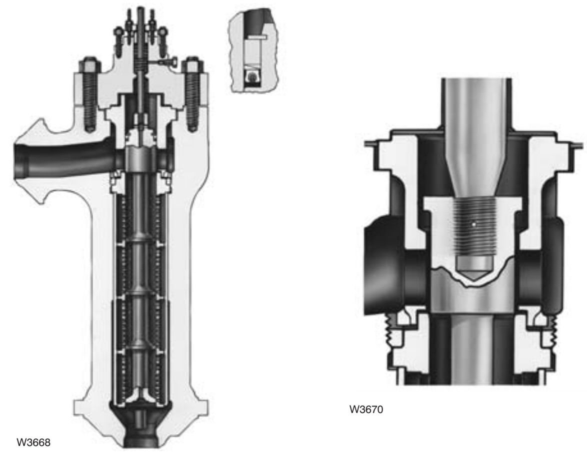

Figure 9B-33. *Stem-balanced Design CAV4 with Cavitrol IV trim*

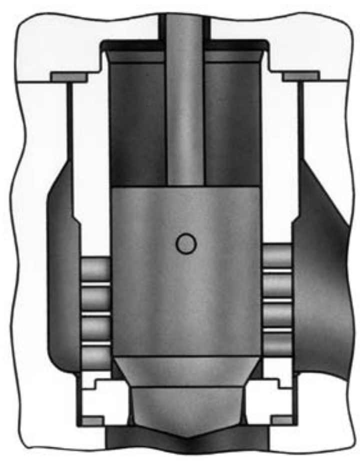

Figure 9B-34. *HPS with Micro-Form trim*

SD steam drain valve. The SD valve is used to warm the main steam lines and to depressurize the superheater during start-up. The SD valve opens when the start-up separator reaches 100 psig. Prior to the SD opening, the SA valve must be open. The SD valve is located downstream of the final superheater and upstream of the high-pressure turbine. This valve dumps high-pressure steam from this location down to the condenser. Typical service conditions for the SD valve are:

P_1 = 3500 psig

P_2 = 0 psig

ΔP = 3500 psid

T = 907°F

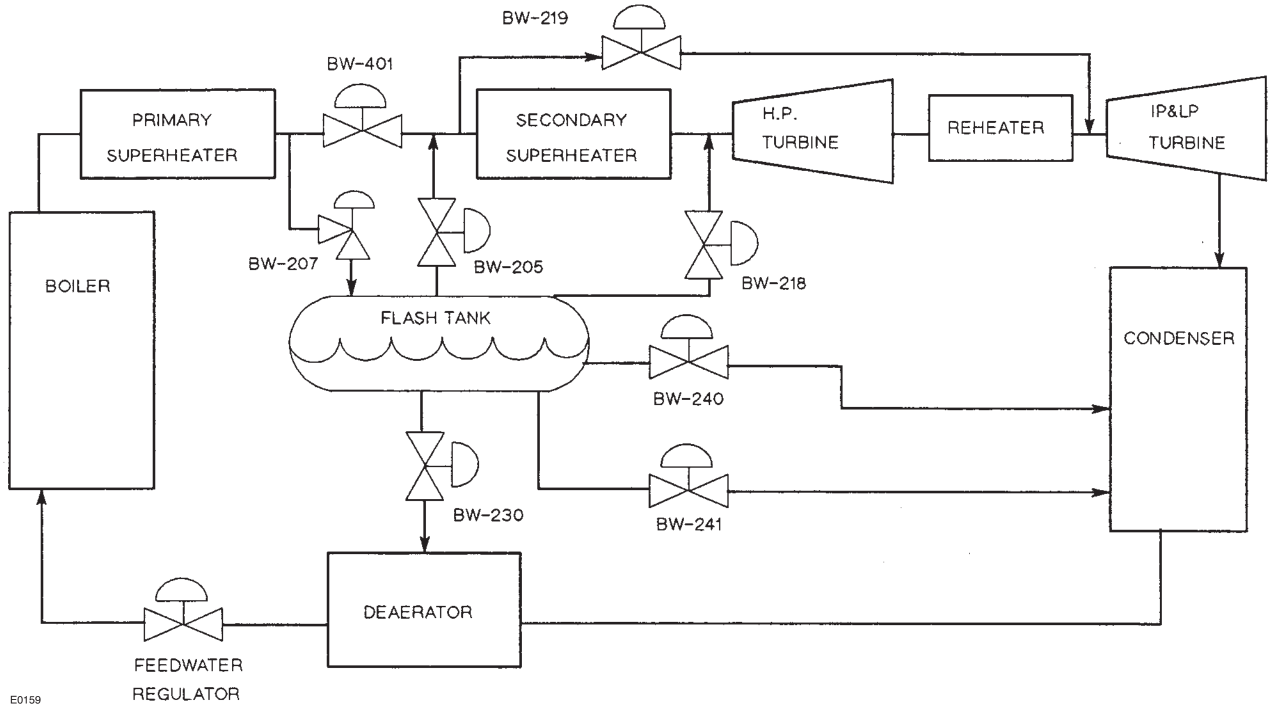

Figure 9B-35. *Flash tank sliding pressure system*

The very high pressure and temperatures of this valve along with the very high pressure drop makes the SD valve somewhat of a unique valve. To accommodate the high noise levels that are obtained by the high-pressure valves, the SD valve has to provide some sort of noise attenuation. The Fisher Design EHD with Whisper Trim III is used to accomplish this task. The EHD can also be used with an in-line diffuser, which will further aid the noise attenuation required by this valve.

Sliding Pressure Control & Supercritical Start-up Systems for Foster Wheeler Boilers

Several systems can be used to implement sliding pressure operation. One, using a flash tank between the primary and secondary superheaters is typical of Babcock and Wilcox installations. (Figure 9B-35.) The flash tank is used as a moisture separator during startup, and when superheated steam is available from the primary superheater, the flash tank is blocked out of the loop. Valves installed between the primary and secondary superheater perform pressure reduction of the superheated steam.

The second system uses an integral separator installed between the furnace outlet and the primary superheater. (Figure 9B-36) The integral separator also acts as a moisture separator during start-up.

However, unlike the flash tank, the integral separator is rated for the full design pressure and temperature of the boiler, and is kept in the loop during once-through operation. Pressure reduction of the superheated steam occurs at valves installed just upstream of the integral separators. By keeping the separators in line, Foster Wheeler avoids some of the controls and extra valves required in the Babcock and Wilcox system to isolate the flash tank.

Foster Wheeler's research has led to conclusions that the optimum sliding pressure configuration is one in which the pressure reducing valves control turbine inlet pressure up to 60% load, and the turbine throttle valves control turbine inlet pressure from 60% to 100% load. The strong points are that temperature variations are reduced and efficiency is increased at low loads, but at full load, the unit can respond quickly to load changes. This mode of operation is called hybrid sliding pressure and is the mode used by the Integral Separator Start-up System (ISSS)®.

GUIDELINE SUMMARY
CE Once-through Supercritical Boilers

Valve Description	Critical Parameters	Fisher Recommendations
BT Boiler Throttle Valve	High Pressure High Temperature Rangeability Tight Shutoff	Large, High Pressure, Design EHD
BTB Boiler Throttle Bypass Valve	High Pressure High Temperature Noise Tight Shutoff	Medium to Large Design EHD with Whisper Trim III
BE Boiler Extraction Valve	Cavitation Flashing Noise Tight Shutoff	Design CAV 4 with Cavitrol IV trim or Cavitrol III / 4-stage Trim
FWB Feedwater Bypass Valve	High Pressure	Large, High Pressure, Design HPT or EHT with Modified Equal-Percentage Characteristic
SP Spillover Valve	High Pressure High Temperature Noise	Large Design EUD or EWD with Whisper Trim III or WhisperFlo Trim
WD Water Drain Valve	Flashing	Design ED in Chrome-Moly Body Material
SA Steam Admission Valve	High Temperature	Design EHD
IS Superheat Injection Water Spray Valve	High Pressure Low Flow Rates	Design HPS or EHS with Micro-Form Trim
IR Reheat Injection Water Spray Valve	High Pressure Low Flow Rates	Design HPS or EHS with Micro-Form Trim
IC Condenser Injection Water Spray Valve	Low Pressure Low Flow Rates	Design EHS with Micro-Form Trim
ISPR Superheat Injection Water Pressure Reducing Valve	High Pressure	Design HPS
SD Steam Drain Valve	High Pressure High Temperature Noise	Fisher Design EHD with Whisper Trim III and/or Diffuser

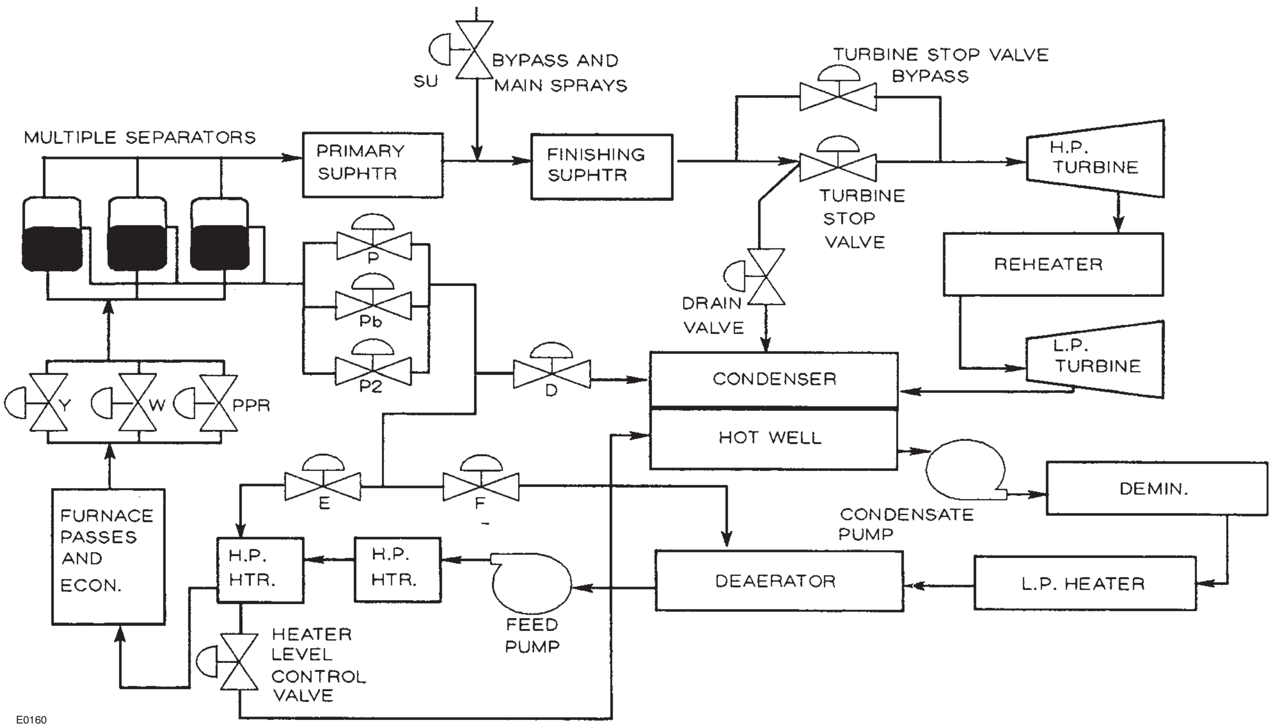

Figure 9B-36. *Integral separator startup system*

The operation of the ISSS can be separated into five stages.

1. *Flushing*
2. *Light off to 8% load*
3. *Ramp boiler, 8-25% load*
4. *Ramp boiler, 25-60% load*
5. *Full pressure operation, 60-100% load*

Flushing. When the flushing cycle (Figure 9B-37) is initiated, the Primary Pressure Reduction (PPR) valves are opened. Valves D, E, F, W, and Y are closed. P1, P2, and Pb are on automatic and control water level in the separators. The feedpump is started and as feedwater flow begins, valve D opens and routes water to the condenser until the water quality is correct. Valve E then opens and routes water to the high-pressure feedwater heater. Feedwater flow is increased to 15% of full load flow, and circulation continues until the conductivity of the water is below 1.0 micromho. At this point, the PPR valves maintain boiler pressure at 1000 psig.

Light off to 8% load. Firing (Figure 9B-38 & 39) begins after the water chemistry is correct. As the unit is fired, the boiler pressure and temperature increase. At 15% flow, the boiler pressure will reach 1200 psig at 450°F. When the inlet temperature of the PPR valves reaches 450°F, the feedwater flow is ramped to 25% of full load flow. The boiler pressure will begin to rise and the W valves open to regulate boiler pressure to 3550 psig. Valves E and F open, and along with valve D, begin to throttle. Valve E holds the high-pressure heater outlet temperature constant. Valve F controls the deaerator pressure. Valve D passes any excess fluid to the condenser. The firing rate is increased to raise the separator pressure to 500 psig. The turbine stop valve is opened to warm and roll the turbine. The initial turbine load is applied, and turbine inlet pressure is maintained at 500 psig up to 8% load.

Valves D, E, and F will normally require ANSI 600 special or ANSI 900 Class ratings. The service is usually flashing, so WC9 bodies are recommended. Typical service conditions are as follows:

Valves D, E, F

P_1 = 100-275 psig

ΔP = 100-275 psid

T = 212-350°F

Ramp boiler, 8-25% load. To ramp to 25% load, the firing rate is increased and the turbine control valves are held at 60% open. Valves P1, P2 and Pb (Figure 9B-40) control separator pressure and, therefore, control turbine inlet pressure. These valves move closed which ramps the turbine inlet pressure to 1400 psig at 25% load. The W valves open as necessary to control boiler pressure to 3550 psig. The SU valve sprays water which attemperates the steam between the primary and secondary superheater. At 25% load, steam output equals water input and full once-through operation is obtained. Valves P1, P2, and Pb are fully closed and the start-up system is bypassed. The W valves are almost fully opened.

The fluid at the inlet of valves P1, P2, and Pb is normally hot condensate, near its saturation temperature. As these valves close, the upstream pressure will increase, the upstream temperature will increase, and the flow rate will decrease. These valves are usually closed sequentially, with P2 closing first, followed by P and Pb Flashing will occur, so WC9 bodies are recommended. These valves must handle the full pressure-temperature rating of the boiler, so ANSI Class 2500 bodies are normally used. Typical service conditions are as follows:

P_1 = 500-1000 psig

ΔP = 250-900 psid

T = 212-550°F

The W valves will also see severe service during this period. Initially, the valves will flow warm condensate that may cavitate or flash. As temperature increases, the incoming fluid will become a two-phase water-steam mixture. Finally, above 25% load, the valve will flow superheated steam. Typical service conditions are:

P_1 = 3550 psig

ΔP = 3500-2000 psid

T = 700-850°F

Typical valves used in this service are high temperature CAV4 or Cavitrol III four-stage trim. By properly selecting the capacity of the W valves, the need for PPR valves may be eliminated, as both the CAV4 and Cavitrol III four-stage trim are capable of throttling at very low capacities.

Ramp boiler, 25-60% load. As load increases (Figure 9B-41), the firing rate is increased and the Y valves open, gradually increasing the turbine inlet pressure. The Y valves continue to open until 60% load is reached and the valves are about 85% open. Turbine inlet pressure is 3500 psig.

As the Y valves open, the inlet pressure will be about 3550 psig, and the outlet pressure will be 1400 psig.

The valve will be flowing superheated steam. As the valve opens, the pressure drop will fall to 50 psid. Typical service conditions are:

P_1 = 3550 psig

ΔP = 2150-50 psid

T = 700-850°F

Noise control trim is required at low travels due to the high pressure drop. At full travel, high capacity is needed to pass the required flow with minimal pressure drop. This is usually done using a specially characterized Whisper Trim III cage. An ANSI Class 2500, WC9 EHD body is used because of the high pressures and temperatures.

Full pressure operation, 60-100% load. For full pressure operation, the Y valves are fully opened. The separator pressure increases almost to full boiler pressure, and the turbine control valves throttle to control turbine inlet pressure.

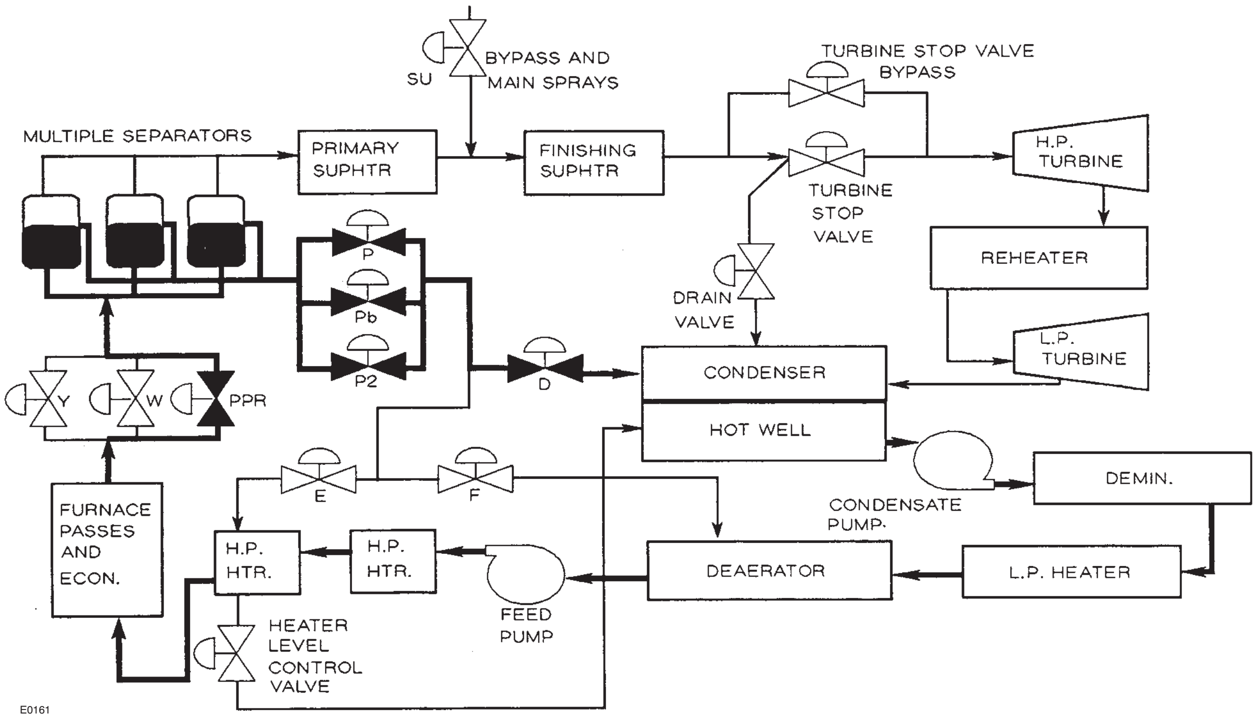

***Figure 9B-37**. Flushing*

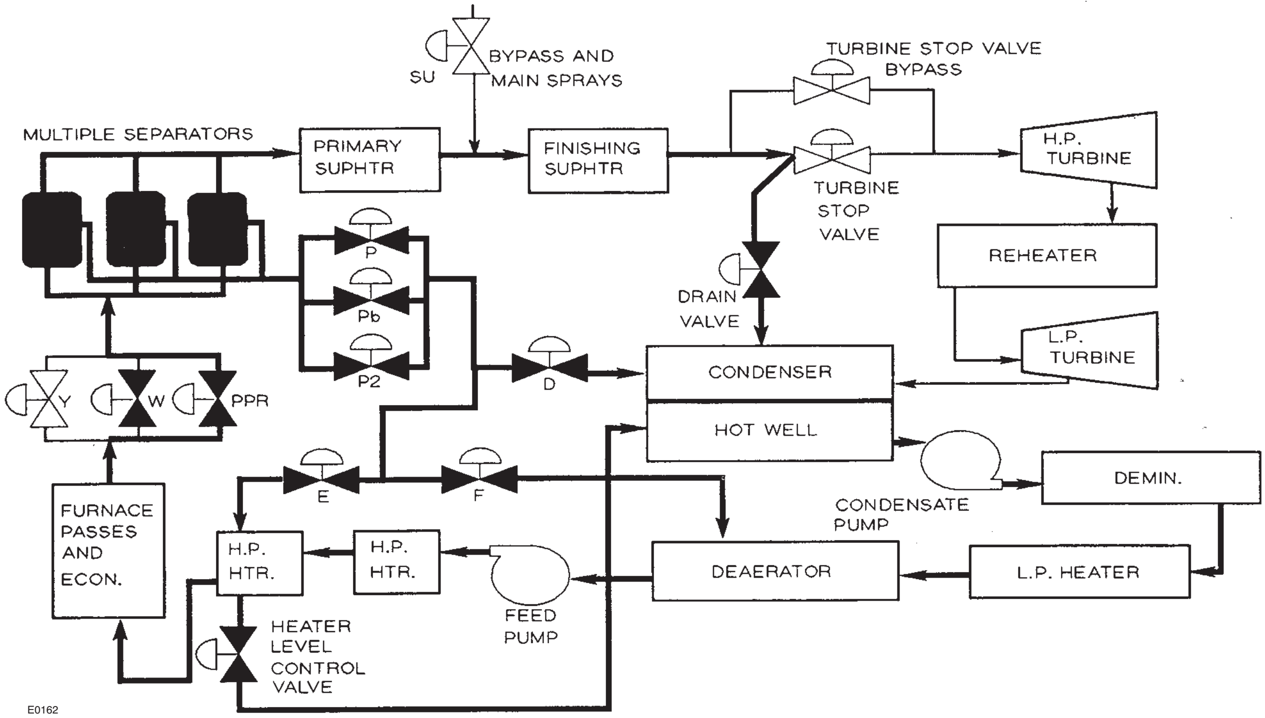

***Figure 9B-38.** Light off*

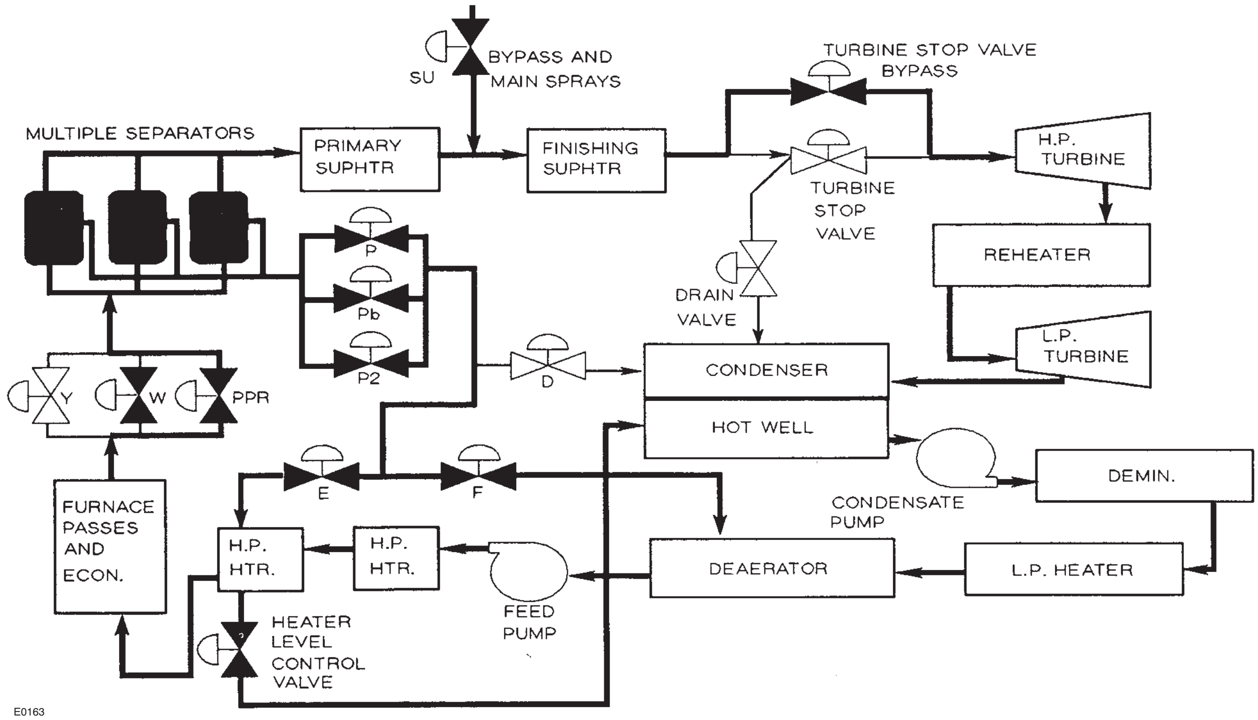

Figure 9B-39. *89% load*

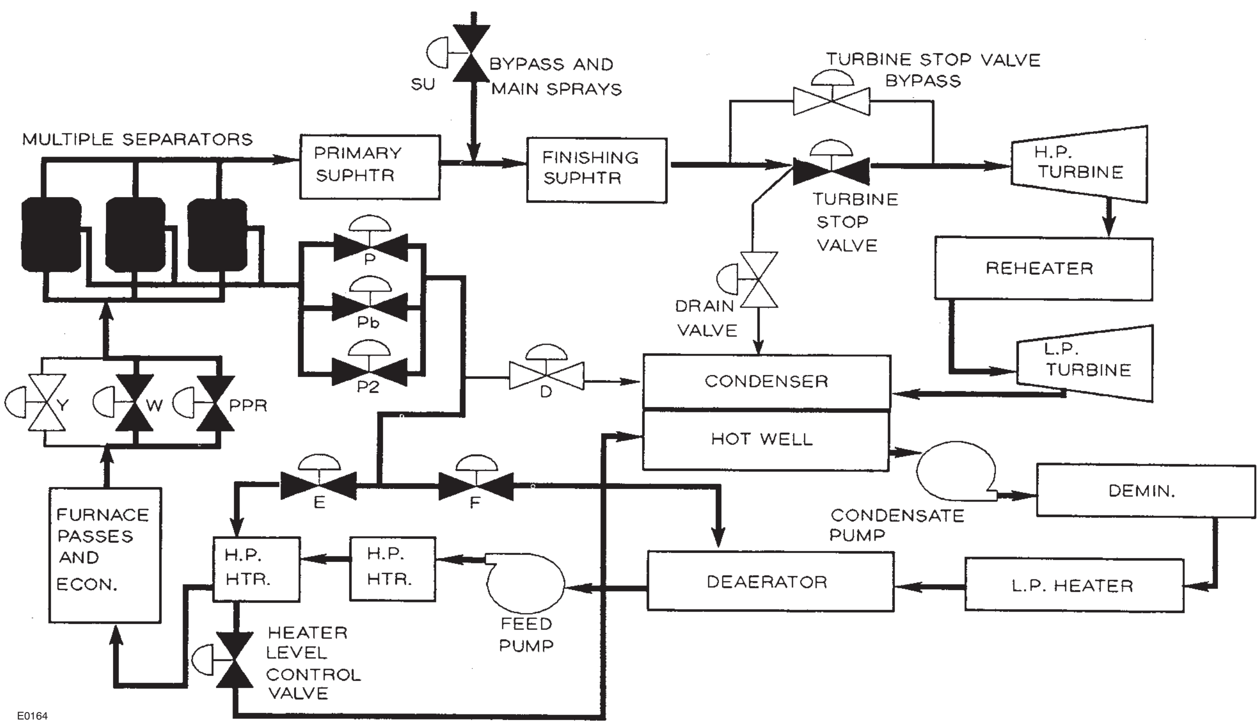

***Figure 9B-40**. Ramp boiler (8 – 25% load)*

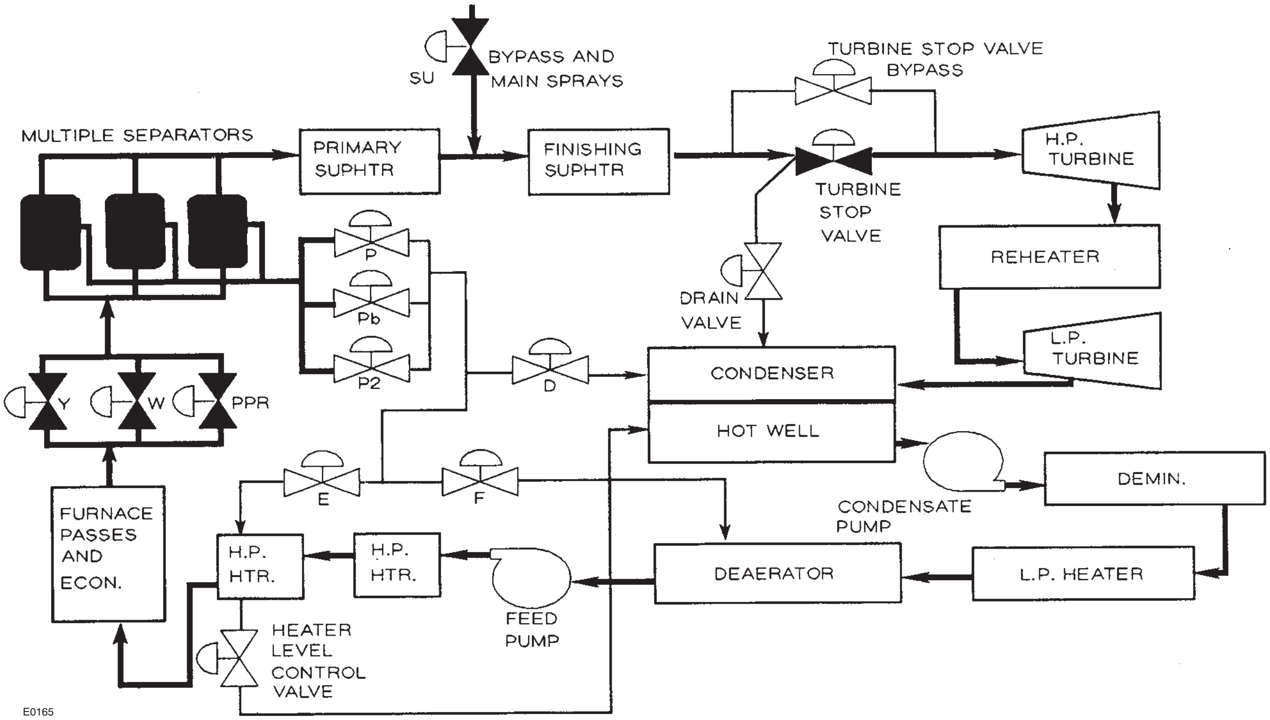

Figure 9B-41. *Ramp boiler (25 – 60% load)*

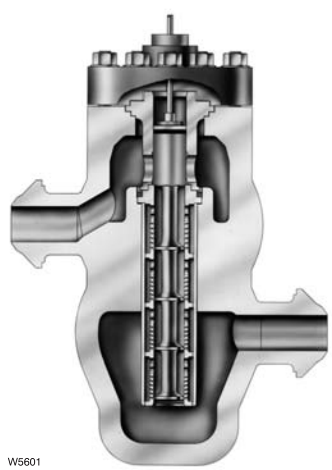

***Figure 9B-42**. Cavitrol IV trim is available in a variety of body designs to fit piping and a variety of service conditions. This valve is ideally suited to services such as the BE application on combustion engineering.*

Chapter 9C

Geothermal Power

The term *geothermal* applies to an enormous underground heat reserve that offers the promise of a clean, renewable source of energy. Geothermal uses today range from direct heating applications, such as spas, greenhouses and the like, to electric power generation.

While direct heating occurred in ancient times, the first commercial use of geothermal electricity dates back only to the early 1900s. In 1913, a 250-kWe geothermal power plant began operation in Larderello, Tuscany, Italy.

Today, geothermal energy is used to generate in excess of 8,000 MW in 21 countries. The United States is the largest producer with approximately 2200 MW of capacity currently, with the majority of its geothermal plants located in California and Nevada.

Geothermal energy is categorized as being either hydrothermal, geo-pressured, hot dry rock, or magma. Other than heat pumps that capture the heat contained in shallow soil, most existing applications utilize hydrothermal resources, which consist of some combination of hot water and steam that is located/contained in permeable rock. In some instances the resource is readily accessible, while others require the drilling of wells, at times a mile or more deep.

If the geothermal resource (water or steam) temperature exceeds 90°C, then it can be used to generate power. Geothermal power production tends to be environmentally clean, and although steam plants may release some gases, these amounts are usually small and easily mitigated.

The amount of geothermal energy contained within our planet is immense and has a lifetime measured in billions of years. However, use of a near-surface source can result in a temporary decrease in the amount of available energy.

Reinjecting geothermal fluid that remains after steam is extracted can help preserve the fluid volume of the reservoir, but it does not regenerate the heat content. Studies suggest that the recovery period of a geothermal resource used for electricity generation is several hundred years. Balancing that against the typical 50-year life span of a generating facility suggests that the periodic construction of a new plant at a new site would allow reservoir recovery.

Converting Steam and Hot Water to Electricity

Three power plant technologies are utilized today to convert hydrothermal fluid energy to electricity. The type of conversion depends on the state of the fluid (whether steam or water) and its temperature:

- **Dry Steam Power Plants**. Hydrothermal fluids (primarily steam) are fed to the turbine. For geothermal resources rich in steam (such as at The Geysers in California) the steam can be used directly. Dry steam plants emit only excess steam and very minor amounts of gases.

- **Flash Steam Power Plants**. Hydrothermal fluids above 400°F (200°C) are sprayed into a tank held at a much lower pressure than the fluid, causing some of the fluid to rapidly vaporize or “flash” to steam. (Figure 9C-1) The flashed steam then drives a turbine. Only excess steam and trace gases are emitted.

- **Binary-Cycle Power Plants.** Geothermal fluid below 400°F and a secondary (binary) fluid with a much lower boiling point than water pass through a heat exchanger. (Figure 9C-2) Heat from the geothermal fluid causes the secondary

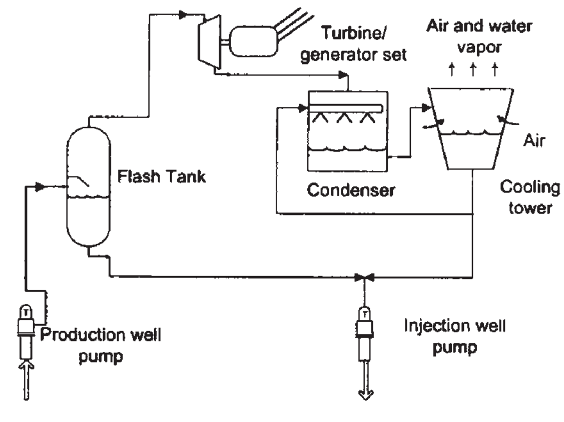

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Figure 9C-1. *Schematic of Flash Steam Power Plant*

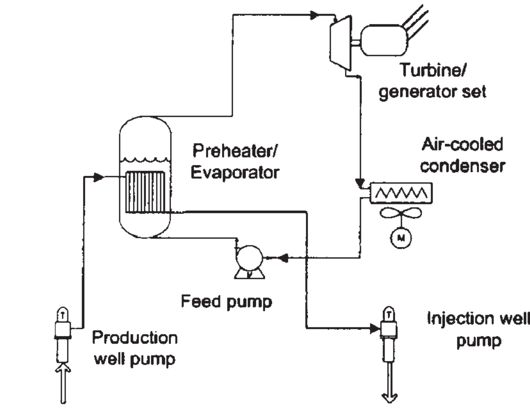

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Figure 9C-2. *Schematic of Binary Cycle Power Plant*

fluid (e.g., isobutane or isopentane) to flash to steam, which then drives the turbine. This is a closed-loop system, with virtually no emission to atmosphere. Since moderate-temperature water is the most common geothermal resource, it is likely that future geothermal power plants will be binary-cycle.

Severe Service Considerations

Geothermal processes can involve harsh fluids that are corrosive, saturated with dissolved minerals, and that contain gases and solids. These conditions range in severity based upon the geothermal fluid's composition, and they need to be considered when selecting and sizing control valves.

- Carbon dioxide corrodes steam and condensate lines.

- Hydrogen sulfide gas is corrosive and lethal at high concentrations.

- Sulfates themselves are not a corrosion factor, but when they combine with the water's calcium, calcium sulfate scale forms. This scale adheres to the inside of pipes and heat exchanger tubing.

- Chlorides increase the corrosiveness of water.

- Ammonia reacts with copper and zinc alloys.

- Oxygen accelerates the corrosion of water lines, heat exchange equipment, boilers, return lines, and the like.

Valve Options

How does one combat this corrosion? Valve material selection and knowing geothermal process conditions are crucial.

Most geothermal fluids contain very high levels of dissolved minerals, especially chlorides. Due to these high levels, corrosion occurs on carbon steel and chloride stress corrosion cracking (SCC) on 300 series stainless steels. Other grades of stainless steel bodies typically are utilized in this service situation because they handle corrosion better by a considerable factor.

What valves should be used in "dirty" service? A superior candidate is the V500, which is known for its ability to fight off the effects of erosive slurries and other tough-to-control abrasive fluids. It is available with a ceramic plug, seat and seat retainer for unsurpassed erosion resistance.

Depending upon the composition of the geothermal brine, one option is to use CV500 eplug™ valves with Alloy 6 chrome oxide coating on internal components for the more severe, dirty steam conditions. CV500 valves also have been used on wellhead applications where a minimum pressure drop across the valve is preferred.

Vee-Ball® valves typically are used both on re-injection level tanks where there is a high

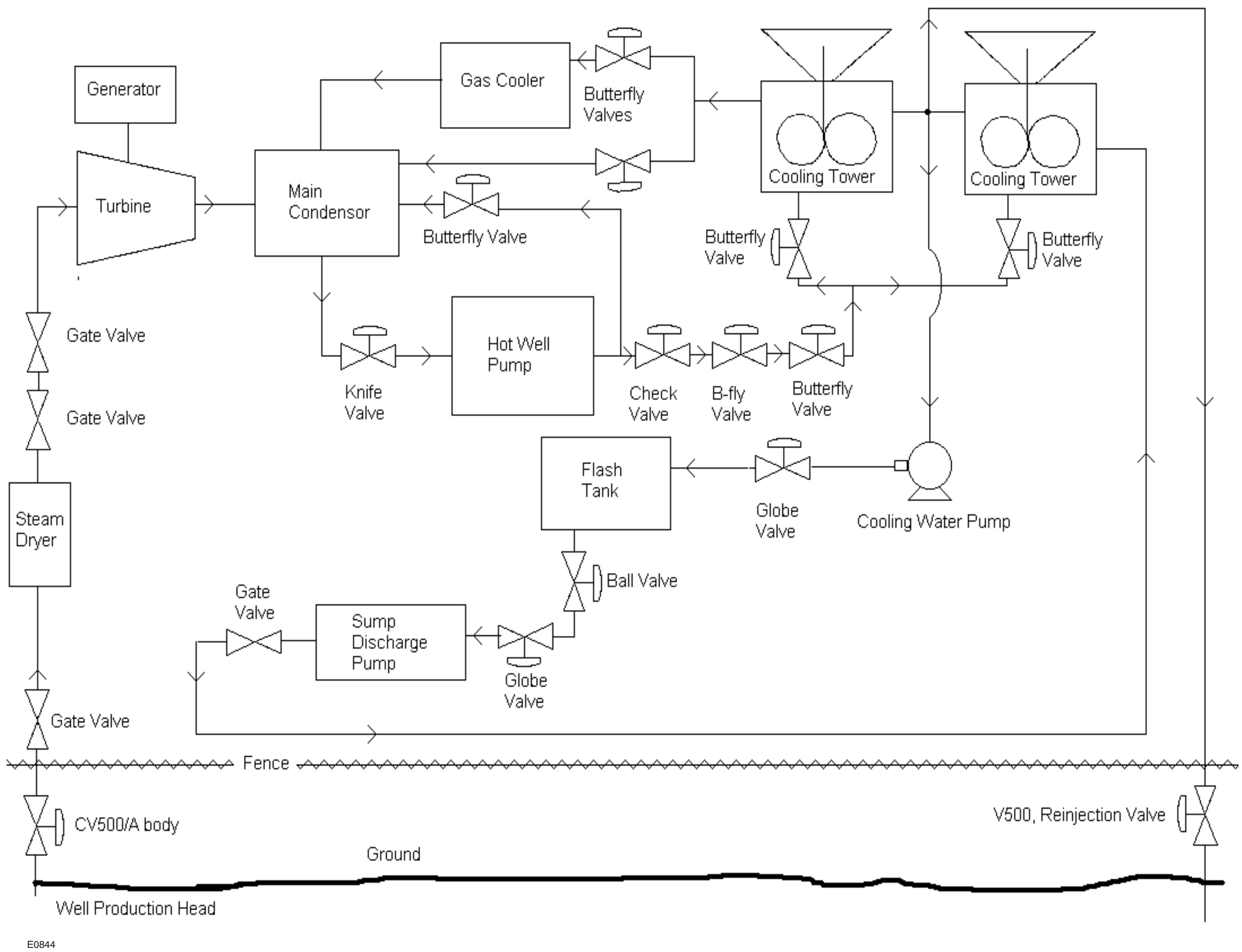

Figure 9C-3. *Location of Frequently Referenced Valves in a Flash Steam Cycle Plant*

pressure drop and before the separator. Another valve that can be used before the separator (a.k.a. production valve) is a high-performance butterfly valve such as the Posi-Seal A41. Taking into account service conditions, the butterfly valve should incorporate a metal disc seal or an elastomer disc seal with a metal backup (firesafe construction.)

What kinds of bearings should be used? First, know where the valve is being placed in the process. If on the “dirty” side or “outside the fence” of the plant, then sealed metal bearings are an option. Metal bearing materials include 316SST, Alloy 6 and 440C. Conversely, if the valve is on the “clean” side or “inside the fence”, PEEK elastomer-lined bearings may be used.

Geothermal Industry Future

Geothermal electricity has proven to be clean, reliable, and an economical alternative to fossil fuels. Continued research and technology development in exploration, drilling and energy conservation offer a promise of increased geothermal power production, making this power source a future major energy contributor in many countries around the world.

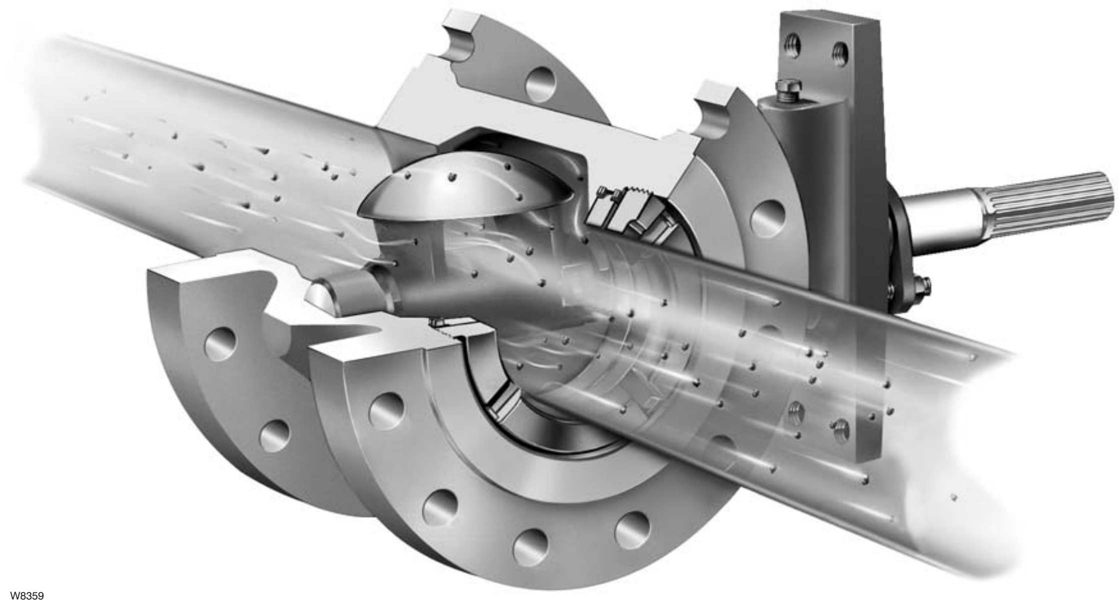

Figure 9C-4*. The V500 ēplug valve offers a reverse flow advantage in that highest velocity flow is isolated in the port or outlet area where the seat and retainer can be protected by utilizing erosion-resistant materials.*

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Chapter 9D

Combined Cycle, Cogen, Simple Cycle

Introduction

This section provides an insight into the operation, maintenance, components and functions of Simple Cycle, Combined Cycle and Co-Generation power plants.

Safety Precautions

Power plants are no more dangerous than any other industrial location. This is quite a statement when you stop to consider that in a power plant high pressure steam, high voltage electricity, burning fuel and many other hazards are all around. Safety Programs are taken very seriously in the power industry. A large amount of time and money are expended each year to make their environment as safe as possible. As a guest it is important to know the safety precautions of the plant you are visiting. This will help protect you and those in your party from any potentially unsafe situations. Additionally, this demonstrates to the plant personnel that you are considerate of them and respect their safety program.

Cost to Operate

Since fuel costs typically account for 60% to 70% of the operating costs for any electric generating unit, it has the largest single effect on the total cost of operation. Other factors that contribute to operating cost include: Maintenance costs, personnel (support, administrative, operations and maintenance) costs and cost of capital investment, to name a few. Plant type, construction, fuel type, location and operation as peaking or base load units are other factors that determine the total cost of operation.

Unit Efficiency by Type	
Simple cycle	<30%
Combined cycle	50%
Conventional ST without reheat	30%
Conventional ST with reheat	35%
Capital Investment per KW	
Simple cycle	$250
Combined cycle	$500
Conventional coal fired	>$1000
Conventional oil or gas fired	>$600
Availability in Percent	
Simple cycle gas fired	88% - 95%
simple cycle oil fired	80% - 90%
Combined cycle gas fired	85% - 92%
Combined cycle oil fired	80% - 88%
Conventional coal fired	80% - 85%
Conventional oil or gas fired	85% - 90%

Typically, one might expect that a cost comparison between an oil-fired simple cycle plant and a coal-fired conventional unit with reheat would reveal that the simple cycle unit was more economical when both are operated as peaking units of less than 1000 hours each year. In this instance, even though the coal-fired unit is more efficient and the fuel is less costly, the high cost of operation and maintenance for the coal fired unit outweigh those considerations.

A comparison between a gas-fired combined cycle plant and a conventional coal-fired base loaded unit with reheat would reveal that the total operational cost for the combined cycle unit would probably be slightly more, but compare favorably with that of the coal fired unit. Again, the lower costs for maintenance and support of the combined cycle unit weigh heavily in its favor.

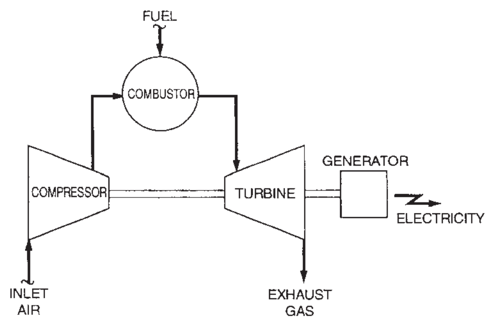


Figure 9D-1. *Gas Turbine - Simple Cycle (From Power Plant Engineering @ Black & Veatch)*

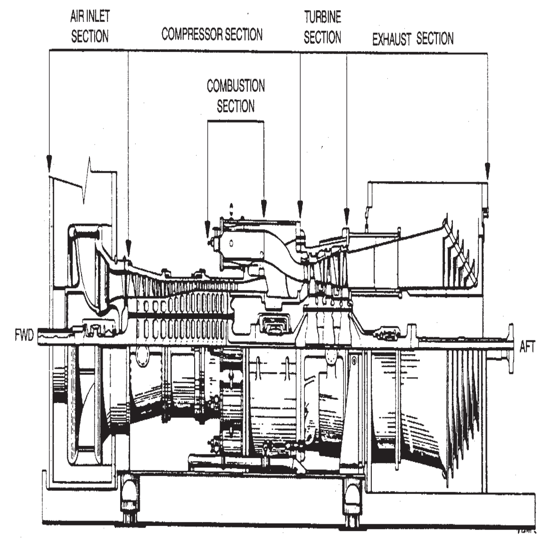


Figure 9D-2. *Major Sections of the MS-7000 Gas Turbine (From General Electric. Used with Permission)*

Plant Types

The following is a discussion of Simple Cycle, Combined Cycle and Co-Generation power generation units.

Simple Cycle. Simple cycle units (Figure 9D-1) consist only of a gas turbine, generator and the auxiliary equipment necessary to support their operation.

The gas turbine is essentially an internal combustion engine and is sometimes referred to as a combustion turbine. In this cycle, the hot gases from the combustion process are used to convert the chemical energy in the fuel to mechanical energy. Figure 9D-2 illustrates the basic gas turbine thermal cycle. Air enters the gas turbine compressor, where its pressure is increased to several times atmospheric (greater than 100 psig). This compression also elevates the temperature of the air to several hundred degrees above ambient (greater than 800F in some turbine types). Typically 30% to 50% of the power produced by a gas turbine's turbine section is required to drive its own compressor.

The compressed, heated air enters the multiple combustion chambers where it is mixed with fuel, and the mixture is ignited. The combustion process further heats the air (greater than 2000° F). The combustion gases are expanded through the turbine section of the gas turbine. In expanding through the turbine section, the thermal energy in the gases is converted to mechanical energy, which is used to drive the compressor and electrical generator. The temperature of the exhaust gas exiting the turbine is typically 900 to 1300 degrees F. As stated above, greater than 50% of the power produced in the turbine section of the gas turbine may be used to power the compressor section with the remainder available to drive the generator.

Gas turbines in commercial use today are either derived from aircraft engine designs or are specifically designed as industrial gas turbines. Typically the aircraft design turbines are based on jet engine designs and are used for smaller (less than 20 MW) generation and industrial applications. Because of their aircraft origins, these turbines may be used in small, lightweight power generation packages.

The industrial turbines were derived from the aircraft engine technologies, but these turbines are much more rugged in construction and are capable of using lower grades of distillate fuel oils. Some industrial gas turbine designs are presently capable of producing over 200 MW with machines of even greater capacities in the design stages. The industrial machines are less expensive per kW and have usually had better emissions levels than their aero design cousins.

Because of the lower efficiency and higher bus bar cost (usually greater than $80/MW) associated with these units, they are usually used primarily as peaking units. Peaking units run only when necessary to provide power during peak demand periods or when economic situations justify their operation. Typically the capacity factors for these machines may be less than 10%. One attractive feature of simple cycle units is that they are modular in construction and may be placed in service in a relatively short period. Later, the extra efficiency of a combined cycle can be had by the addition of a Heat Recovery Steam Generator and Steam Turbine.

Advantages of Simple Cycle units include:

- Use of clean fuel sources such as natural gas and relatively low sulfur distillate fuels (No. 2 fuel oil)

- Lower installed costs (approximately $300/kW verses $1000/kW for a conventional coal fired plant)

- Very short on-site construction periods (as little as six months)

- Because of their compact, modular construction they occupy a very small foot print

- Fast start-ups and ramp rates

- Excellent cycling capability

Disadvantages of Simple Cycle plants are:

- Lower efficiency

- Natural gas and fuel oil are higher cost fuels and when coupled with their low efficiency ratings these plants are expensive to operate.

- Natural gas supplies are not always assured, especially during peak heating and cooling season

Manning. Operational and maintenance manning requirements for Simple Cycle plants are usually very low. In fact, some installations are completely remote controlled and have no permanent operational or maintenance located at the site.

Even when they are site manned, since they are usually operated as peaking facilities, they may only be manned during peak load periods of each weekday (one shift). A one-shift operation would only require six to 10 personnel.

Application Discussion

Fuel Control. Prior to connecting to the electrical grid the fuel control system is responsible for maintaining the turbine at the correct rotational speed (rpm), by positioning the fuel valve. Once the unit is connected to the grid control of the load on the unit's generator is essentially a function of the amount of fuel supplied to the turbine. To increase the load, the turbine fuel valve admits more fuel, and decreasing the fuel supply reduces the generator output. In this mode the control system controls the fuel supply to the turbine so that it can supply the mechanical energy needed to maintain the desired electrical energy output form the generator.

The fuel control system also has to be capable of reacting to sudden and drastic changes in load on the turbine such as during a full or partial load rejection. A load rejection occurs when the generator output breaker opens abruptly. When the breaker opens the load on the unit is instantaneously reduced and the turbine will overspeed and trip if the fuel supply is not quickly adjusted to maintain the turbine speed at less than the overspeed trip value.

Most gas turbines will use either natural gas or fuel oil, typically #2 or #6. The type of fuel used will depend upon the environmental requirements of the area where the plant is installed. Most units today are supplied with some manner of emissions control. In natural gas fired systems, this is accomplished through staged fuel control that lowers the overall flame temperature that in turn limits the formation of harmful emissions.

This same technique is used when firing fuel oil, but must be accompanied by either water or steam injection. Water and steam injection is used to atomize and warm the oil droplets that will lead to a lower flame temperature and hence, lower emission levels.

Figure 9D-3 shows a typical staged combustion layout. Each stage will feed a separate portion of the nozzle that will admit a certain amount of fuel to that stage of the combustion process.

Service conditions for a typical fuel gas and fuel oil system can be found below. Valve selection will vary depending on the unit type and size. Many older installations used ball valves for fuel control, but issues with performance due to 'sloppy' linkages caused the industry to change to sliding stem valves. The other reason was the increased need for emissions control.

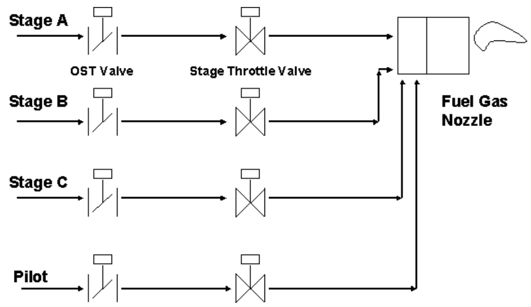


Figure 9D-3. *Fuel Gas Staged Combustion*

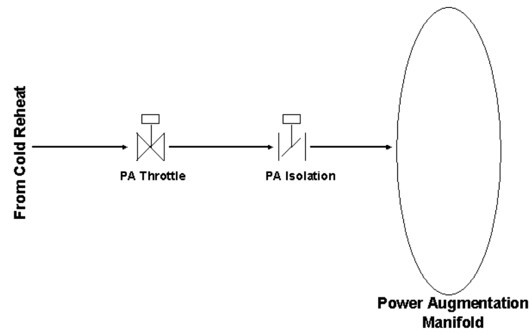


Figure 9D-4. *Power Augmentation*

	Fuel Gas	Fuel Oil
Inlet pressure (psig)	600	1500
Pressure drop (psid)	105 - 550	105 - 700
Inlet temperature (F)	100	100
Valve type and size	2” - 4” ET/ES	2” - 3” ET/ES
Actuation	Electric	Electric

The other valves noted in Figure 9D-3 are called the OST valves. These valves protect the turbine from overspeed when the fuel control system fails to react quickly enough to prevent an overspeed condition. Modern gas turbines use multiple electronic sensors to detect the turbine speed and trip the turbine when an overspeed condition exists. The ultimate result of the activation of this device is the closure of the fuel valves and isolation of the fuel system from the combustion chamber of the gas turbine. A line size butterfly valve with tight shutoff is used for this application.

Power Augmentation. To increase the mass flow through the turbine, which increases the turbine output and efficiency, steam is injected into the turbine blades of the gas turbine. This is typically used in a combined cycle configuration where the steam is pulled off the cold reheat line, but can also be used in single cycle configurations if there is an available outside steam source.

The allowable load on the turbine blades limits power augmentation. Typically, injection rates can reach as high as two pounds per unit of fired fuel. When selecting a control valve, this typically means a range of 30,000 up to 150,000 pounds per hour.

A control valve that can handle high temperatures and noise and still provide tight shutoff is required. In some cases, as noted in Figure 9D-4, an isolation valve is used in combination with the power augmentation control valve. This eliminates the need for tight shutoff at the control valve.

However, the isolation valve must be able to provide tight shutoff in either direction. This is due to the possibility of combustion gases backing up in the steam line during operations where power augmentation is not required. This can lead to premature failure of the control valve components due to the high combustion gas temperatures.

The follow recommendations have been made for the power augmentation throttle and isolation valves.

	Throttle Valve	Isolation Valve
Inlet pressure (psig)	550 - 650	550 - 650
Pressure drop (psid)	50 - 200	50 - 200: 1000 - 1500
Inlet temp. (F)	550 - 680	550 - 680: 800 - 1075
Valve type	6” - 8” ED WhisperFlo	Line size butterfly
Actuation	Pneumatic - spring return	Pneumatic - spring return

Water and Steam Injection. The primary function of the injection system is to lower the flame temperature when firing fuel oils. By lowering the flame temperature, the formation of NOx emissions is limited (Figure 9D-5). This also results is higher electrical output due to the increased mass flow through the turbine similar to power augmentation.

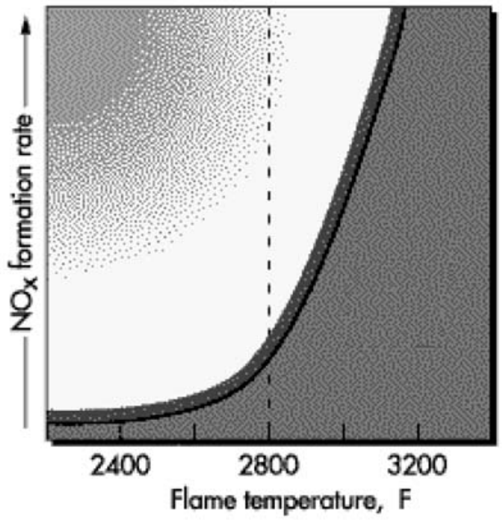


Figure 9D-5. *NO_x Formation vs. Temperature*

The steam injection application is very similar to the Power Augmentation application in terms of product selection. A steam control valve can be used to provide both control and tight shutoff or an isolation valve may be supplied in conjunction with the throttle valve. Steam is injected into the fuel nozzles to cool the flame temperature at the source of combustion. The product selection guidelines are below.

	Throttle Valve	Isolation Valve
Inlet pressure (psig)	550 - 650	550 - 650
Pressure drop (psid)	50 - 200	50 - 200 1000 - 1500
Inlet temp. (F)	550 - 680	550 - 680 800 - 1075
Valve type	6" - 8" ED WhisperFlo	Line size butterfly
Actuation	Pneumatic - spring return	Pneumatic - spring return

Water injection uses the same idea as steam injection, but is more common in single cycle configurations. This eliminates the need for an offsite steam source and can be readily used on peaking units. It is very uncommon to find two identical units in this configuration. The amount of water injection depends upon the emission requirements, the temperature of the water and the ambient temperature. These factors lead to different valve selections for each site location.

A valve with low flow and anti-cavitation capabilities is usually required. The CAVIII or CAVIII/Micro-Flat™ can be used to meet the requirements of this application. In cases where very low flows and cavitation control are required, the multistage Micro-Flat can be used. A set of typical service conditions can be found below.

	Fuel oil water injection
Inlet pressure (psig)	1450
Pressure drop (psid)	250 - 1300
Inlet temp. (F)	60
Valve selection	1" - 2" ET with Cavitrol III/2-Stage or CAVIII/MicroFlat
Actuation	Pneumatic

Turbine Lube Oil. The turbine lube oil system is used to provide lubrication to both the gas turbine and gas turbine generator. This system consists of reservoirs, pumps, coolers, heaters, filters, piping and valves. A cooler rejects the heat that is absorbed from the equipment with two coolers typically being provided.

There are two main valves in the turbine lube oil skid. These are the bearing pressure regulator and the cooler temperature control valve. Turbine and generator bearing pressure is controlled via the bearing pressure regulator. Both valves have different functions, but rely on one another to provide lube oil at the correct pressure and temperature to ensure proper sealing between the shafts and bearings.

	Pressure Regulator	Temperature Regulator
Inlet pressure (psig)	150	150
Pressure drop (psid)	20 - 50	20 - 50
Inlet temp. (F)	180	200
Valve selection	4" ED or 4" V150	4" ED
Actuation	Pneumatic	Pneumatic

Additional Applications

There are many other applications in a simple cycle configuration such as turbine cooling, compressor wash, atomizing air and drain systems that have not been covered. These applications are very similar to associated applications seen in fossil fired power plants.

Combined Cycle. The Combined Cycle Power Plant, (Figure 9D-6) in its most basic form, consists of a Gas Turbine (Combustion Turbine), a Heat Recovery Steam Generator (HRSG), Steam Turbine and the auxiliary equipment required for power production. This is essentially a fossil fuel power plant, where natural gas or fuel oil is combusted to provide the thermal energy necessary to drive a gas turbine and generator.

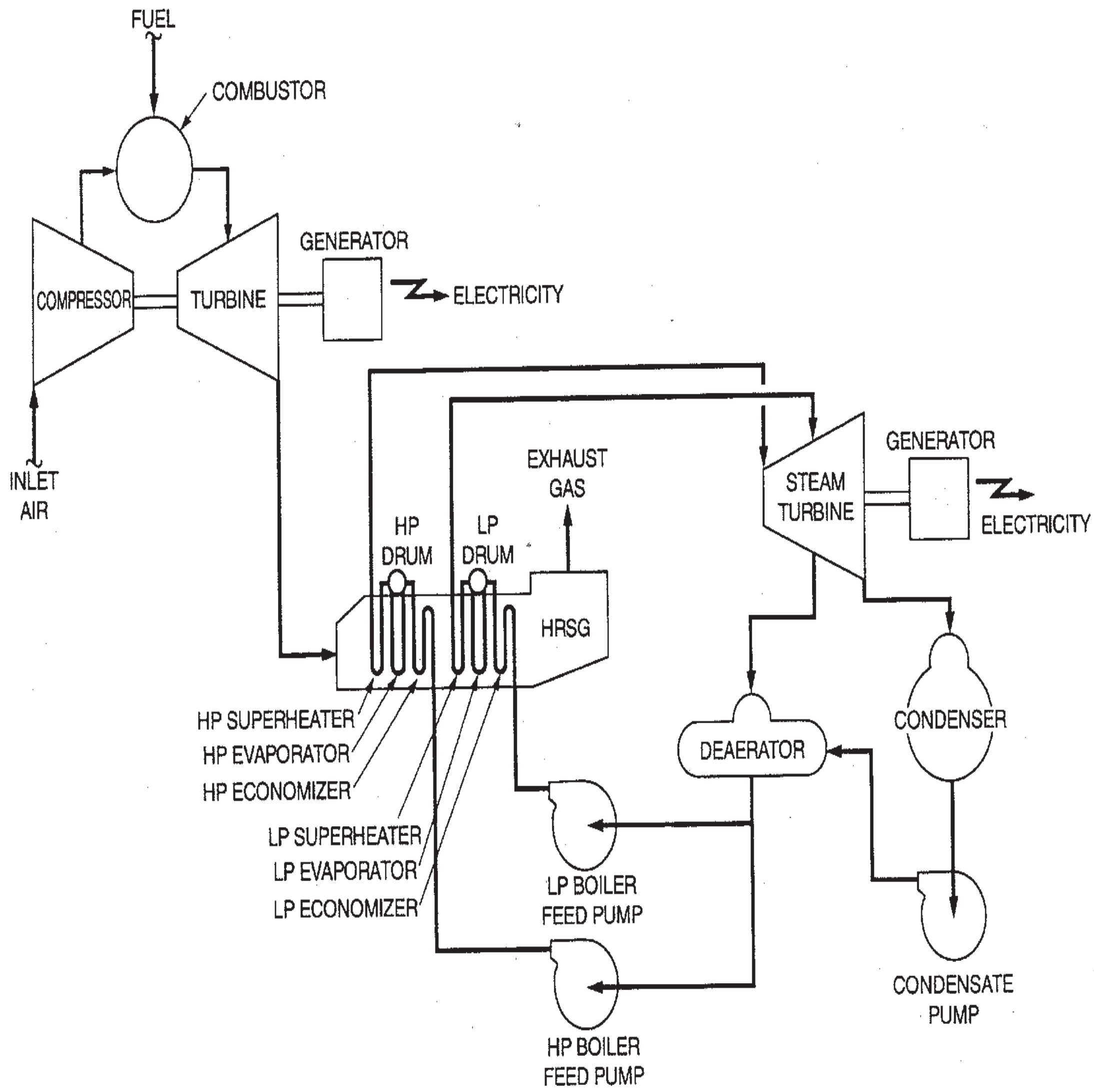


Figure 9D-6. *Gas Turbine Generation Cycles - Combined Cycle (from Power Plant Engineering @ Black & Veatch)*

The exhaust heat from the Gas Turbine is ducted through a gas to water heat exchanger known as a Heat Recovery Steam Generator (HRSG). The exhaust gases heat the water and produce steam at pressures and temperatures sufficient to operate a Steam Turbine Generator.

The HRSG is similar to the conventional fossil fired boilers, but uses finned tubes (Figure 9D-7) to maximize the surface area and permit more efficient heat transfer from the hot exhaust gases to the water in the tubes.

Normally, all the heat required to generate steam is furnished by the hot exhaust from the gas turbine. However, in some cases, the HRSG is supplied with a natural gas or fuel oil fired duct burner. This supplemental heat is used to generate additional steam for either increased power output or cogeneration. Figure 9D-8 gives a sectional view of a HRSG with duct firing.

HRSGs vary widely in construction. Smaller, less complicated HRSGs may consist only of a high-pressure steam drum, accompanying steam generation tubes and possibly a superheater. As the needs for unit efficiency increase, low-pressure drum, intermediate pressure drum, high, intermediate and low-pressure economizers, deaerators and superheaters are added. The tube sections vary so that the hottest gases pass over

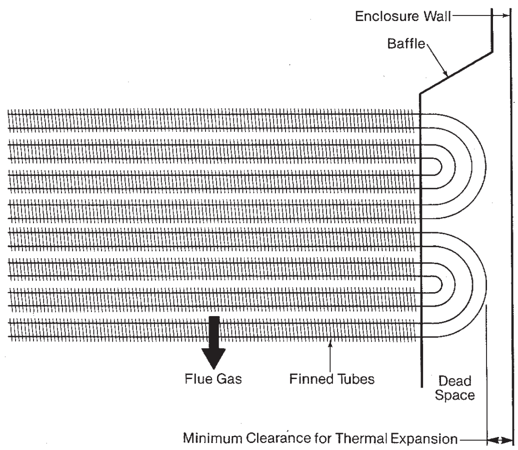


Figure 9D-7. *Finned Tubes (From Steam, Its Generation and Use @ Babcock & Wilcox)*

the high-pressure superheat stages first. After some of the heat is given up, reheat steam is produced and eventually, the warm gases are used in the evaporator and preheat sections through the boiler.

The Steam Turbines (ST) used with Combined Cycle Units can have considerable differences from those employed with conventional power units. For one thing, combined cycle STs are usually smaller in construction than their conventional unit counterparts. Typically combined cycle STs vary from 10 to 200 mWs in size, with the majority being less than 100 MWs.

Besides being smaller, combined cycle STs are sometimes constructed without reheat capability (when their associated HRSG cannot supply reheated steam). Typically, all conventional unit steam turbines are manufactured with reheat capability because of the increased efficiency benefits that reheat offers.

One other major difference is that most steam turbines used in combined cycle applications have little if any steam extraction points. This is because most combined cycle installations do not employ shell type feedwater heaters like those typically used in conventional plants. Instead, combined cycle plants rely almost exclusively on the finned tube economizer(s) installed in the HRSG to increase the feedwater temperature to acceptable levels.

Modern industrial gas turbine engines are the most efficient producers of electrical power when used in the combined cycle configuration. Advances in the gas turbine metallurgy and improved internal cooling techniques have resulted in machines that operate at higher more efficient temperatures. The industrial machines also tend to have higher exhaust temperatures, which may be used to produce more steam in the HRSG and provide more efficient superheat and reheat steam to the steam turbine.

In the simple cycle mode of operation, large industrial turbines are capable of proceeding from start-up to full load in from 10 to 30 minutes. The smaller aircraft turbine versions are capable of proceeding to full load in less than 5 minutes. Temperature heat-up rates for the HRSG and the steam turbine usually restrict the rate that the gas turbine can be ramped up when operating in the combined cycle mode.

Many times the gas turbine are fitted with a "Diverter or Bypass Damper" on the gas turbine exhaust duct. The purpose of the diverter damper is to permit the gas turbine to be run in the simple cycle mode by diverting the hot gases from the gas turbine through a "Bypass Stack" instead of through the HRSG. This is important during periods when the HRSG or ST are not available due to maintenance or other restrictions. Diverter dampers are also sometimes installed on new installation simple cycle units. This permits later HRSG installations without interfering with the gas turbine operations.

Combined cycle installations have efficiency ratings of over 50% (based on the Lower Heating Value), a considerable improvement over the 35% normally achievable with conventional fossil plants. In addition to their high efficiency combined cycle units enjoy several advantages such as:

- Lower emissions due to their use of clean fuel sources such as natural gas and relatively low sulfur distillate fuels (No. 2 fuel oil)
- Lower installed costs (approximately $500/kW verses $1000/kW for a conventional coal fired plant)
- Short construction periods
- Because of their compact, modular construction they occupy less real estate per kW
- Fast start-ups and ramp rates
- Excellent cycling capability

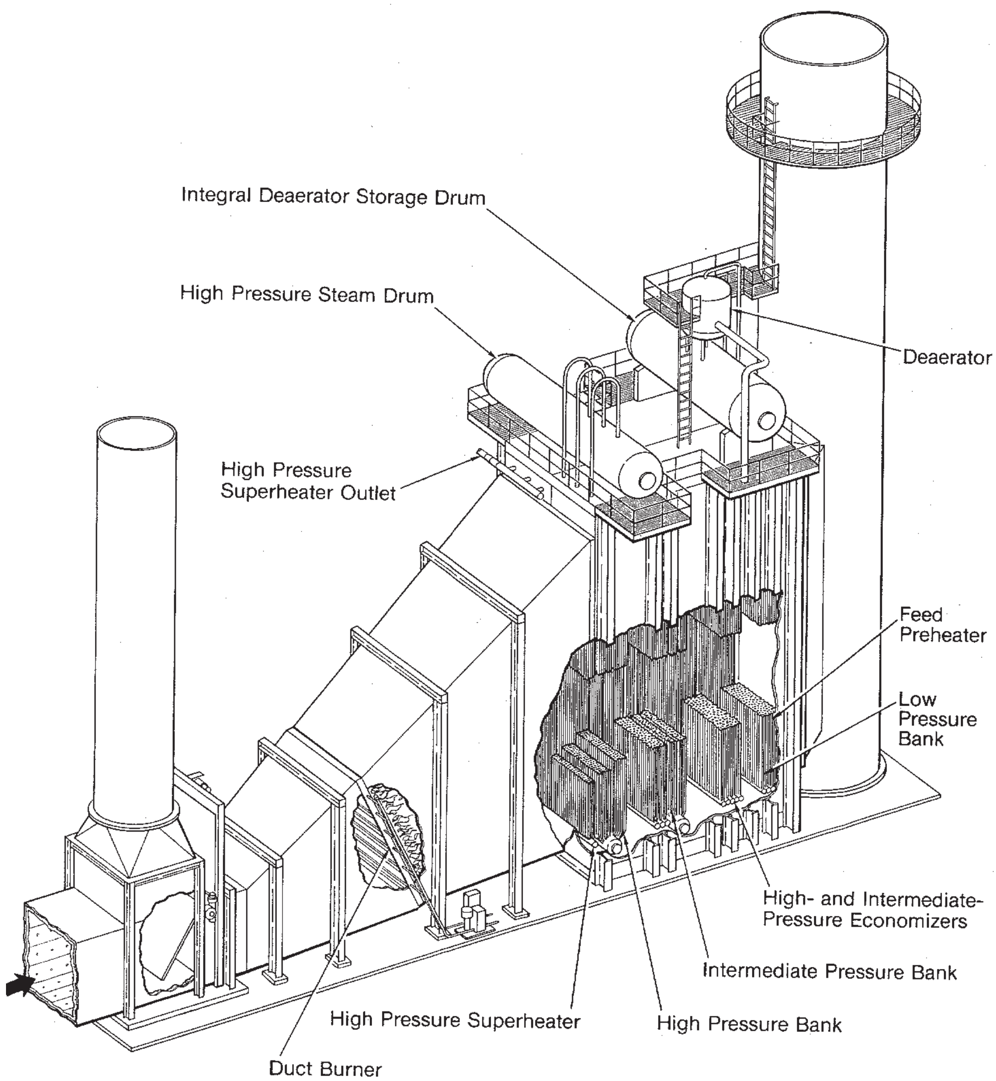


Figure 9D-8. *Heat recovery steam generator with duct burner (from Combustion Fossil Power @ ABB)*

Disadvantages of gas turbines and combined cycle plants are:

- Natural gas and oil are higher cost fuels, so even though these plants are more efficient, they may not be as economical when fuel costs are considered, especially fuel oil.

- Natural gas supplies are not always assured, especially during peak heating and cooling season

The popularity of combined cycle units increased significantly during the 1990s. This increase in popularity was brought about by a number of factors that include:

Environmental – Because of environmental concerns it has become increasingly difficult to site permit and build conventional power plants. This is especially true for base loaded coal fired units.

Construction Costs – Lower cost to build per kW (about half the cost to construct verses a conventional coal fired unit).

Cleaner Fuels – The use of natural gas and distillate fuel oils has several benefits in addition to the reduction of emissions to the atmosphere. Because of the lower sulfur content of the fuels, scrubbers and other emissions control equipment, which are expensive to own, operate and maintain are not required. Coal storage and handling equipment (another maintenance intensive system) is not required.

Manning – Operational and maintenance manning requirements for Combined Cycle plants are usually less than 50% of that for a conventional coal fired unit of the same capacity. The type of fuel in use at the site affects manning levels at combined cycle units. Typically the use of distillate fuels requires more personnel to operate and maintain the unit than natural gas does. This is to some degree caused by the extra tanks, pumps, filters, etc. to handle and store the distillate fuel. However, another factor in the extra manning requirement is due to the effects that the distillate fuel has on the various components it comes in contact with and the higher temperatures the gas turbine is exposed to when operating with the distillate fuel.

Typically a 400 MW combined cycle installation may require from 10 to 20 operational personnel. The maintenance personnel requirements, for the same size plant, may be 10 to 20 personnel for a natural gas fired plant and 15 to 30 personnel for an installation that has dual fuel capability (uses both natural gas and distillate fuel) or for one that uses distillate fuel only. A typical example of the manning for a combined cycle plant would be a dual fueled 320 MW plant that has 25 total management, operational and maintenance personnel. Future improvements in plant automation will further reduce or eliminate the need for on-site manning at combined cycle installations.

Co-Generation. Co-generation can be defined as the production of steam for export to an outside manufacturing or other industrial activity while also generating electrical power. In other words the co-generating unit produces two products, steam and electricity for outside consumption.

The overall efficiency of co-generators can be over 70%. This is due to the higher efficiency that can be achieved by the direct use of steam as an energy source, without the waste of the condensing cycle normally associated with the conversion of steam to electrical power. (The typical efficiency of a modern conventional condensing cycle electrical unit is approximately 35%. The typical efficiency of a modern combined cycle generating unit is approximately 50%. The efficiency of steam as an energy source for heating only can approach 90%.)

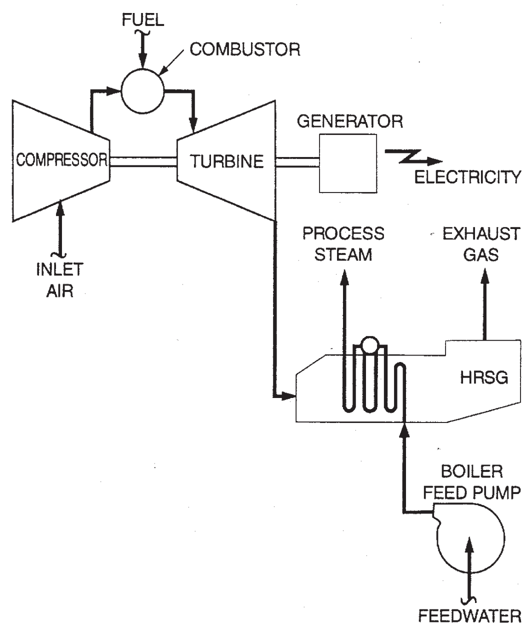


Figure 9D-9. *Gas turbine generation cycles - co-generation (from Power Plant Engineering @ Black & Veatch)*

Co-generators, because of their high efficiency and use of waste heat to replace conventional fossil fuels have found favor with environmental groups and governmental agencies.

All types of electrical plants are capable of being co-generators, but the overwhelming number of co-generators is of the combined cycle configuration.

Figure 9D-9 illustrates one possible configuration for a co-generation plant. In Figure 9D-9 all the steam being produced is exported for off-site consumption. Other configurations will export only a portion of the steam with the remainder used to generate electricity. Another variation would be that all the steam is initially routed to a steam turbine for the generation of electricity. Steam is then extracted from the steam turbine for off-site use.

Operational considerations for co-generators are different than for traditional generators. The co-generator must also consider the steam needs of the steam customer. In many cases, a conventional auxiliary boiler or other methods may be employed to supply steam when it is not economical to operate the entire co-generation facility.

Application Discussion

The applications in a combined cycle and cogeneration power plant are very similar to that in a conventional fossil fuel plant. The main difference is that all heating is done within the HRSG instead of external heaters. This reduces not only the space requirements, but equipment requirements as well.

The main systems are discussed in Chapter 9A, but a brief description of the major combined cycle systems will be discussed below.

Feedwater System

Note

Alloy 6 corrosion problems often occur in feedwater systems. Typically, boiler feedwater is treated with ammonia, hydrazine or amine derivatives at the deaerator to eliminate excess oxygen. These chemicals attack the protective oxide films on Alloy 6 and allow initiation of an erosion/corrosion process that degrades Alloy 6 overlays. This phenomenon is responsible for many valve trim failures previously attributed to poor design or maintenance. Selection of solid 400 series stainless steel trim or use of Colmonoy overlays will eliminate the problem.

The feedwater system in the combined cycle power plant encompasses the portion of the plant from the outlet of the preheater section at the rear of the HRSG. It includes the feedwater pump, the economizer and vaporizing sections of the HRSG and the feedwater control valves.

Unlike a conventional power plant, there are typically two main applications to deal with in the combined cycle power plant. These are the feedwater regulator and feedwater recirc valves. However, unlike a conventional plant, the combined cycle plant has separate feedwater control valves for each drum of the HRSG.

There are usually one or two feedwater pumps that feed each section of the boiler. Variable staged pumps can provide low pressure, intermediate pressure and high pressure water for each boiler section. Some plant designs separate the low pressure pump from the intermediate/high pressure pump.

The feedpump recirculation valve is nearly identical to conventional designs. The valve is intended to maintain the net positive suction head on the feedwater pump to keep cavitation from occurring in the pump stages. Valve selection depends upon the inlet pressure to the valve. Service conditions and recommend valve solutions can be found in the table below.

	Pressures <3000 psig	Pressures >3000 psig
Inlet pressure (psig)	2400 - 3000	3000 - 3500
Pressure drop (psid)	2400 - 3000	3000 - 3500
Inlet temp. (F)	220 -275	220 - 275
Valve selection	2” - 3” HP/EH with Cavitrol trim or DST	2” - 4” HP/EH with Cavitrol trim or DST or CAV4
Actuation	Pneumatic	Pneumatic

Unlike conventional power plants, the startup regulator and main feedwater regulator are typically combined into one valve. Characterizing the trim, which provides cavitation protection as the valve first opens to address the startup condition followed by standard holes as the travel increases to provide the necessary capacity, does this. This eliminates the need for additional piping and additional valves.

In most applications, these valves are installed at ground level so that the pressure drop is taken prior to the water passing through the economizer section of the drum. However, in some designs the valve is mounted next to the steam drum to prevent steam from forming in the economizer section of the boiler.

Product selection can vary on the IP and HP level control valves. With mounting the valve at the drum comes the possibilities of variances during the startup condition. The valve must still address cavitation during initial startup, but be able to withstand flashing as the temperature increases. An angle valve with cavitation protection is recommended.

	Standard Configuration	After Economizer Configuration
Inlet pressure (psig)	1500 - 3500	2600 - 3500
Pressure drop (psid)	200 - 3350	200 - 3500
Inlet temp. (F)	220 - 278	220 - 600
Valve selection	3” - 6” ET/HPT/EHT with characterized Cavitrol trim	3” - 4” EHA with characterized Cavitrol trim
Actuation	Pneumatic	Pneumatic

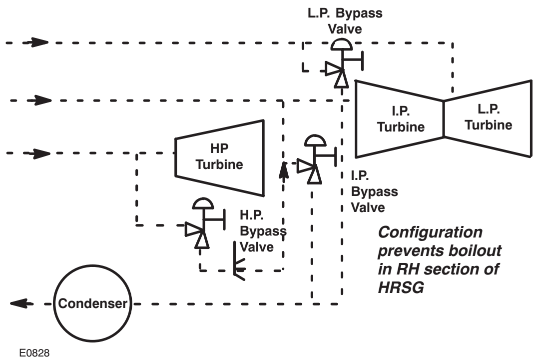


Figure 9D-10. *With reheat Turbine bypass system*

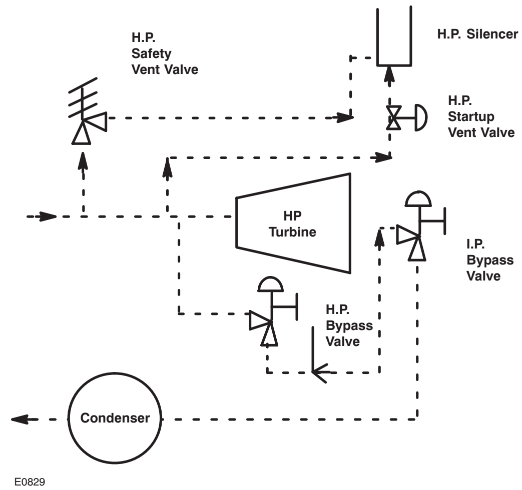


Figure 9D-11. *HP vent startup system*

Turbine Bypass System

The Turbine Bypass System (TBS) is probably the most key system in the combined cycle power plant. TBS valves operate under severe conditions during startup and shutdown. During initial firing they are required to bypass steam either to atmosphere or to the condenser. During plant shutdown, they must also be able to bypass full steam load to the condenser to protect the steam turbine components.

Figure 9D-10 shows a conventional turbine bypass system with reheat capability. This system desuperheats the HP steam to match that of the cold reheat steam prior to being admitted to the reheater section. Once the steam passes through the reheater, it is then bypassed around the turbine to the condenser. The LP Bypass Valve is designed to bypass all low pressure steam to the condenser. The function of the LP turbine bypass valves does not change.

Sizing of the TBS valves varies depending upon plant size. As can be seen above, the valve sizes will also depend upon the type of plant and whether the unit has reheat. The valves typically supplied for these applications include pressure reduction, noise control and steam conditioning. These valves are addressed by the Fisher steam conditioning product line, which is discussed in Chapter 7.

Startup Vent

As steam is first formed in the HRSG, it must be bypassed around the steam turbine. One of the main problems with bypassing steam directly through the turbine bypass valves is that this causes a major shock to the valve and piping systems. Therefore, before steam is admitted to the bypass valves, it is typically bypassed directly to atmosphere in order to warm the bypass valves prior to opening.

The startup vent valves are required to operate during initial plant startup, but their role extends far beyond the startup cycle. Once the valves are closed, they are required to maintain tight shutoff for long durations. This is because any lost steam yields less efficiency and less power production. Therefore, it is important that these valves be able to handle the high pressure drop requirements, but also maintain long term shutoff. Figure 9D-11 shows the schematic for the HP bypass system.

In these applications, a high-pressure valve with C-seal is provided in combination with a vent silencer. As control valve technology has improved, so have the offerings in the vent diffuser line. Fisher now offers the WhisperFlo vent diffuser (Figure 9D-12) that incorporates the proven WhisperFlo technology. The WhisperFlo vent diffuser can provide an additional 10 dBA of noise attenuation compared with today's offerings as well as being approximately 35% the size and weight of an equivalent drilled hole device.

Service conditions are the same as that seen for the turbine bypass valves with the exception of the mass flow rate. The startup vent valves typically see 10% to 20% of the maximum flow that can pass through the bypass valves.

Figure 9D-12. *WhisperFlo vent diffuser*

	HP Vent	IP Vent	LP Vent
Inlet pressure (psig)	1880	550	100
Pressure drop (psid)	1880	550	100
Inlet temp (F)	1065	750	600
Flowrate (lb/hr)	7,000 - 250,000	10,000 - 75,000	3,500 - 20,000
Valve selection	6” EH with WhisperFlo, C-seal and 6014/6015 WhisperFlo vent diffuser	8” - 12” EW/EU with WhisperFlo, C-seal and 6014/6015 WhisperFlo vent diffuser	4” - 8” ED with WhisperFlo, C-seal and 6014/6015 WhisperFlo vent diffuser
Actuation	Pneumatic	Pneumatic	Pneumatic

	HP Bypass	IP / HRH Bypass	LP Bypass
Inlet pressure (psig)	1880	1100	100
Pressure drop (psid)	1300	1010	60
Inlet temp (F)	1065	1035	600
Outlet temp (F)	600	345	300
Flow (lb/hr)	70,000 - 500,000	100,000 - 500,000	35,000 - 100,000
Valve	8” - 12” EH or TBX depending on configuration	24” - 36” TBX or FB with attached cooler	14” - 24” TBX or EW/FB with attached cooler
Actuator	Pneumatic or hydraulic	Pneumatic or hydraulic	Pneumatic or hydraulic

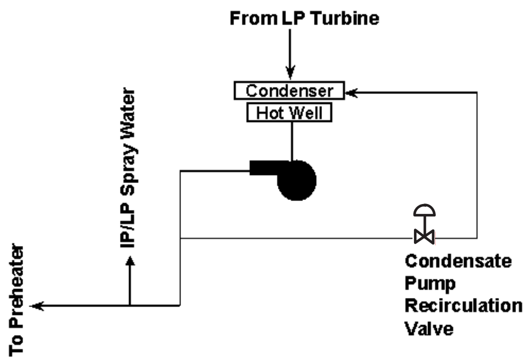


Figure 9D-13. *Condensate system*

Condensate System

The condensate system has the same function in a combined cycle plant as it does in a conventional power plant. Its role is to take condensate from the hotwell below the condenser and send this through the preheat system at the rear of the HRSG. The condensate will then flow into the deaerator, which separates out all compressible gases.

The condensate recirc valve is used to maintain the net positive suction head on the condensate pump to protect it from cavitation. The recirculation line runs from the outlet of the condensate pump back to the hot well, which is very close to atmospheric pressure or vacuum (Figure 9D-13). The service conditions will often indicate that the valve will experience flashing. However, the effects of pipe friction, elevation and condensate sparger backpressure are ignored. This increases the outlet pressure seen at the valve, which leads to the formation of cavitation, not flashing.

An EW with Cavitrol III/2-stage trim is typically used for this application. This will provide the necessary cavitation protection and address any issues with noise and vibration.

	Condensate Recirc
Inlet pressure (psig)	600
Pressure drop (psid)	500 - 550
Inlet temp (F)	150
Valve selection	3” - 4” EWT with Cavitrol III/2-stage
Actuation	Pneumatic

This is a very brief discussion of the applications that exist in simple cycle, combined cycle and cogeneration type power plants. As can be seen by the current demand at the turbine and boiler OEMs these types of power plants will continue to be the mainstay of the power generation market until a more cost effective and efficient solution can be developed. This may be filled by fuel cells, coal gasification or circulating fluidized bed boilers. Until that day comes, the natural gas fired technology will reign supreme.

Chapter 10

Materials Guidelines

The selection of materials for control valve components is a very complex process. Control valves are required to function with precision in some very extreme environments. A number of factors must be considered to insure that a material will perform properly in service.

These factors fall primarily into two categories: 1) the material's suitability to function mechanically; and 2) the material's compatibility with the environment.

These constraints conflict in many instances, making it difficult or impossible to satisfy all considerations with a single material. In these cases, the best compromise must be identified.

Components in control valves are subjected to a variety of conditions that warrant consideration when selecting materials of construction. Among the material properties that must be considered are strength, wear resistance, and toughness.

Mechanical and Physical Properties

When selecting materials, the mechanical and physical properties that must be considered vary depending upon the component. Obviously, the properties that are important in the selection of a body material are different from those used in the selection of a cage material. Some of the properties that must be considered when selecting valve materials are described in the following.

Yield Strength

The yield strength of a material is the stress required to cause a permanent deformation of 0.2%. This parameter is utilized heavily in the design of pressure containing parts as well as other parts which carry loads, such as valve stems, cages, seat rings, and bolting.

Hardness

Hardness is a material's resistance to penetration or indentation, and it is one of the most difficult material properties to understand. In metals it usually is measured by loading an indenter into the material and measuring either the depth of penetration or the surface area of the indentation. The deeper the penetration or the greater the surface area of the indentation, the lower the hardness. The hardness as measured in this manner is a function of a number of other mechanical and physical properties, such as yield strength, work hardening rate and elastic modulus. Hardness is often used to estimate a material's sliding wear resistance and resistance to erosion or abrasion.

Toughness

Toughness is a material's resistance to fracture. Toughness has traditionally been measured using impact tests, such as the Charpy and Izod tests. These both measure the number of foot-pounds required to fracture a sample with a pre-existing stress riser. Recently, the science of fracture mechanics has introduced new methods for both determining a material's resistance to fracture and evaluating a structure's susceptibility to fracture in the presence of defects.

The measure of toughness in the fracture mechanics realm is called fracture toughness, and it is a measure of the stress at the tip of a sharp crack that is sufficient to cause catastrophic failure in a particular material.

Fracture toughness values, and even Charpy and Izod impact toughness values, are difficult to find for many of the materials used in control valves. In most cases, they are even more difficult to correlate with operating conditions in a control valve.

In many cases, the other mechanical properties are examined to give an indication of toughness. In general, a tough material displays a higher percent elongation and/or percent reduction in area than does a brittle material. Also, a tougher material displays a greater difference in yield strength and ultimate tensile strength (or, a greater work hardening rate) than does a brittle material. And finally, austenitic materials (such as 300 series stainless steels and nickel-base alloys) generally have much greater toughness than ferritic materials (such as carbon and alloy steels and 400 series stainless steels.)

Wear Properties

Wear is a term used in conjunction with a number of mechanisms involving material removal or damage. The most important specific wear categories encountered in control valves are sliding wear, erosion, and cavitation damage.

Sliding wear refers to the damage caused when two mating parts move relative to one another. Sliding wear actually encompasses a number of different mechanisms. The two mechanisms most often encountered in metallic components of control valves are adhesive wear and oxidative wear.

Adhesive wear (usually called “galling”) occurs when the frictional heat and contact pressure between asperities on the surface of the two parts are sufficiently high to cause localized welding. The relative motion of the parts causes repeated welding and fracture of these localized areas, causing material transfer between the parts. The surface of both parts becomes rough, which in most cases aggravates the situation. The roughness of the parts reduces mechanical efficiency and can even cause seizing of the parts.

Oxidative wear is similar to adhesive wear, except that the frictional heat causes oxidation of the asperities. Oxidative wear generally produces a fine, powdery wear product, which may or may not cause abrasive damage to the metallic parts. Whether sliding wear is adhesive or oxidative in nature depends on a number of factors, including the wear couple materials, the contact pressure, and the environment. Galling is more likely to occur in inert atmospheres, whereas oxidative wear is more likely in atmospheres that are reactive toward the metal alloys involved.

In general, sliding wear resistance is optimized by observing the following set of guidelines:

- Use a wear couple with dissimilar elemental composition. This makes welding of the materials at the wear interface less likely.

- Use materials having different surface hardness.

- Use lubricants where possible. Lubricants reduce frictional heating and interfere with welding of the materials at the interface.

In most cases, other factors (such as corrosion, erosion, or strength considerations) prevent all of these guidelines from being followed. In these instances, the component materials providing the best combination of properties must be determined.

Erosion is mechanical damage caused by either high-velocity fluid impingement or impact by abrasive particles in the flow medium. A common form of erosion in high-pressure control valves is “wire drawing”, which is localized erosion damage to seating surfaces or cage holes.

Cavitation damage is caused by the shock waves generated when vapor bubbles implode during pressure recovery. Both erosion and cavitation damage can be minimized by material selection, although in most cases the use of an appropriate valve and/or trim style is more effective.

Corrosion

Corrosion is a major consideration in most of the process control industries. However, the power industry is involved predominantly with systems that are relatively inert. Most feedwater and steam systems are non-corrosive to standard control valve materials, and for this reason selection is usually based upon other consideration such as wear resistance and mechanical properties. Resistance to erosion-corrosion in certain treated feedwater and oxidation resistance at high temperatures are about the only corrosion related factors to consider in the vast majority of applications.

Many materials achieve their corrosion resistance through the development of a "passive layer" on the surface. Most common are the corrosion resistant materials containing chromium and/or molybdenum. In these alloys, the chromium and/or molybdenum will combine with oxygen at the surface to form a tough, adherent oxide layer which is resistant to attack in many environments. Under high-velocity conditions in certain environments, this passive layer is "washed off" of the surface, allowing the material to corrode, a phenomenon called "erosion-corrosion". This problem can be prevented only by limiting the use of susceptible alloy/environment combinations to low-velocity applications.

Effects of Inlet and Differential Pressure

The inlet pressure and differential pressure (pressure drop) have little influence on valve materials from a chemical standpoint, but they do play a role in determining not only the valve and trim style, but also the materials of construction.

The pressure drop influences material selection for both the body and the trim components. In flashing applications, body erosion can occur in the cavity below the seat ring. Special materials are used often to avoid body erosion problems even when they are not required because of pressure-temperature considerations. If the pressure drop indicates high-velocity erosion and/or cavitation may occur, different trim materials will often be selected than would be used in a non-erosive or non-cavitating situation.

Effects of Elevated Temperature on Metallurgical Stability

Most metal alloys have structures that are metastable in nature, and when they are placed into an elevated temperature environment, they tend to transform to their stable structures. The reactions that occur affect a number of properties that are important in control valve applications.

For example, carbon steel materials that are used for valve bodies possess a two-phase microstructure consisting of ferrite (essentially pure iron) and iron carbides. Prolonged exposure to temperatures in excess of 800°F causes the carbides to decompose into iron and graphite reducing both the strength and toughness of the material, a phenomenon known as "graphitization." Steel alloys with chromium and/or molybdenum are utilized above 800°F because their carbide phases are more stable.

Metallurgical stability problems occurring at high temperatures affect other materials and account for upper service temperature limits in many cases. Some examples of materials limited by elevated temperature stability problems include 17-4PH and related precipitation hardenable stainless steels (which become brittle), cold worked 300 series stainless steels (which lose their cold-worked effects), and duplex stainless steels (which become brittle).

Effects of Elevated Temperature on Yield Strength

Yield strength in metal alloys is a strong function of defects in their crystalline structure. These defects are formed purposely through heat treatment, cold working, etc., to strengthen materials. Elevated temperatures decrease the effectiveness of these mechanisms, effectively lowering the yield strength. Each material has its own yield strength vs. temperature profile that is dependent upon composition and material condition.

Creep at Elevated Temperature

At highly elevated temperatures, a phenomenon called creep comes into play. To explain creep, it is first necessary to explain elasticity. A typical metallic material at ambient temperature displays elastic behavior, i.e., stress is linearly proportional to strain, and the relationship is not time-dependent. A specimen loaded to a certain stress will strain a certain amount, and that strain will remain constant until the load is changed.

However, if the temperature is high enough, the behavior becomes non-elastic. The strain will slowly increase with time (hence the name "creep"). In some applications, creep becomes a significant factor in the design of a workable control valve. The temperature required to cause creep is dependent upon material composition and material condition. Creep data is usually presented in graphical or tabular form displaying the stress to cause a certain amount of permanent deformation as a function of temperature. At temperatures where creep is active, yield strength becomes irrelevant.

Effects of Elevated Temperature on Elastic Modulus

In metallic materials, stress (S = load divided by area) is proportional to strain (e = change in length divided by initial length). The elastic modulus (E) related stress and strain by the equation:

$S = E (e)$

The elastic modulus is basically a measure of the "stiffness" or "spring rate" of the material, and is only dependent upon composition and temperature. The elastic modulus decreases with increasing temperature, which means that the material becomes less "stiff". This can affect a number of components in control valves.

For example, assume a bonnet bolt is torqued to provide a particular load. This load actually corresponds to a given amount of strain in the bolt at the torque limit. The valve is subsequently placed into service at an elevated temperature, which causes a reduction in the elastic modulus of the bolt material.

Since the strain remains constant (assuming that all parts in the assembly have the same thermal expansion coefficients), the stress in the bolt (and thus the load) is reduced by the same proportion. Each material has its own elastic modulus vs. temperature profile that can be used to help optimize the material selection for control valve components.

Coefficient of Thermal Expansion

When metallic materials are heated, they expand in a predictable and repeatable manner. Each alloy has its own characteristic thermal expansion coefficient vs temperature curve which can be used to predict its growth as it is heated. In general, related materials have similar thermal expansion properties, and can be grouped for general discussion purposes. The carbon steels, alloy steels, and 400 series stainless steels have fairly low thermal expansion coefficients, whereas the 300 series stainless steels have very high expansion rates. The nickel alloys fall in between.

When selecting materials for a valve that will be used at low or elevated temperatures, thermal expansion differences must be taken into account. Differential thermal expansion between plugs and cages can cause binding or excessive looseness at operating temperature. Likewise, differential thermal expansion in a body-bonnet-cage-seat ring system can cause loss of gasket load, resulting in leakage. Differences in thermal expansion rates must be either eliminated (by selection of like materials) or accounted for (by proper dimensioning of parts) when a valve is to be used at temperatures significantly above ambient.

Materials of Construction

This section describes the materials commonly used within the power industry for various components in control valves.

Materials for valve bodies and bonnets must meet a number of requirements:

- They must lend themselves to manufacture by casting to accommodate the irregular shapes that bodies and bonnets tend to have.

- They must be reliable materials; i.e., they must have known strength properties, and should be produced and sold under adequate codes and standards to ensure their integrity.

- They must have adequate mechanical properties while at operating temperature.

- They must be resistant to corrosion, oxidation, and other adverse effects in the environment(s) where they will be utilized so they will retain their integrity.

Most Fisher valves are designed and manufactured per ASME/ANSI B16.34, entitled <u>Valves - Flanged and Buttwelding End</u>. This code specifies standard dimensions, markings, and other design practices for valves in a series of classes, ensuring that designs will fit into standard piping and will be produced according to sound design practices.

The ASME/ANSI code also specifies allowable pressures as a function of temperature for the series of classes. Since the ASME/ANSI approved materials have different strength properties, pressure limits within a particular class vary according to the material.

When a valve is being selected for a particular working pressure/temperature combination, it usually turns out that the requirements can be satisfied with several materials selected from different classes. A decision must then be made whether to buy the lower strength material in a higher ANSI class, or buy the higher strength material in a lower ANSI class. This decision is usually based upon economics, but in some cases other factors such as line stresses and transient conditions can affect the decision.

Carbon Steel Bodies and Bonnets

The standard material for valve bodies and bonnets is ASME SA216 Grade WCC. This is a carbon steel material that is easily cast, welded and machined. Fisher recently decided to switch to WCC from WCB (which has been a standard valve material for many years) due to the ANSI pressure-temperature rating advantages of WCC over WCB and the increasing popularity of WCC among both customers and vendors. Transition is still underway and thus, both materials may be used.

Carbon steel is used for a large majority applications due to its low cost and reliable performance in general applications. Its use is strongly recommended over any other material if possible because if its standard availability and low cost.

Alloy Steel Bodies and Bonnets

When higher temperatures and/or pressures are involved, alloy steels are often specified for bodies and bonnets. There are a large number of alloy steel materials which Fisher and other valve manufacturers have supplied over the years for these applications. Most are steels with chromium and/or molybdenum added to enhance their resistance to tempering and graphitization at elevated temperatures. The chromium and molybdenum additions also increase their resistance to erosion in flashing applications. Among the more popular materials are ASME SA 217 grades C5, WC9, and WC6.

In the past, C5 (5% Cr, 1/2% Mo) was commonly specified for applications requiring chromium-molybdenum steels. However, this material is difficult to cast, and tends crack when welded. If casting defects are uncovered during ddddd, weld repair is very difficult, and bodies must sometimes be scrapped and re-ordered due to proliferation of cracking.

For this reason, Fisher has standardized on WC9 (2-1/4% Cr, 1% Mo) as its standard chromium-molybdenum steel material. WC9 is preferred by the foundries, and is much easier to work with in manufacturing than C5. Use of WC9 as a substitute for C5 has been successful in all applications for several years. Our experience has shown little difference in erosion resistance.

Stainless Steel Bodies and Bonnets

The Fisher standard for stainless steel bodies and bonnets is CF8M, which is the cast version of 316. With its nominal 19% Cr, 10% Ni, 2% Mo composition, CF8M is a relatively low-cost material with good high temperature properties and excellent resistance to corrosion in a wide variety of environments.

CF8M's high chromium and molybdenum contents give it even better resistance to erosion in flashing applications than WC9 material, and it is sometimes used when the chromium-molybdenum steels provide inadequate performance. It is also used in many applications for its high temperature pressure-temperature rating. When used for temperatures above 1000°F, it is specified with a minimum carbon content of 0.04% per ASME Boiler and Pressure Vessel Code requirements to ensure adequate high-temperature strength.

Selection of Body/Bonnet Material

Comparing pressure-temperature ratings in ASME/ANSI B16.34 is much simpler when the ratings are presented in graphical form.

The first discovery is that the ANSI Class 150 ratings for WCB, WCC, WC9, and C5 are identical over their common temperature ranges, and that CF8M is only rated slightly lower at temperatures below 550°F.

The second discovery is that the rating plots for these materials in all other classes have the same shapes, and all that changes is the y-axis scale for the allowable pressures.

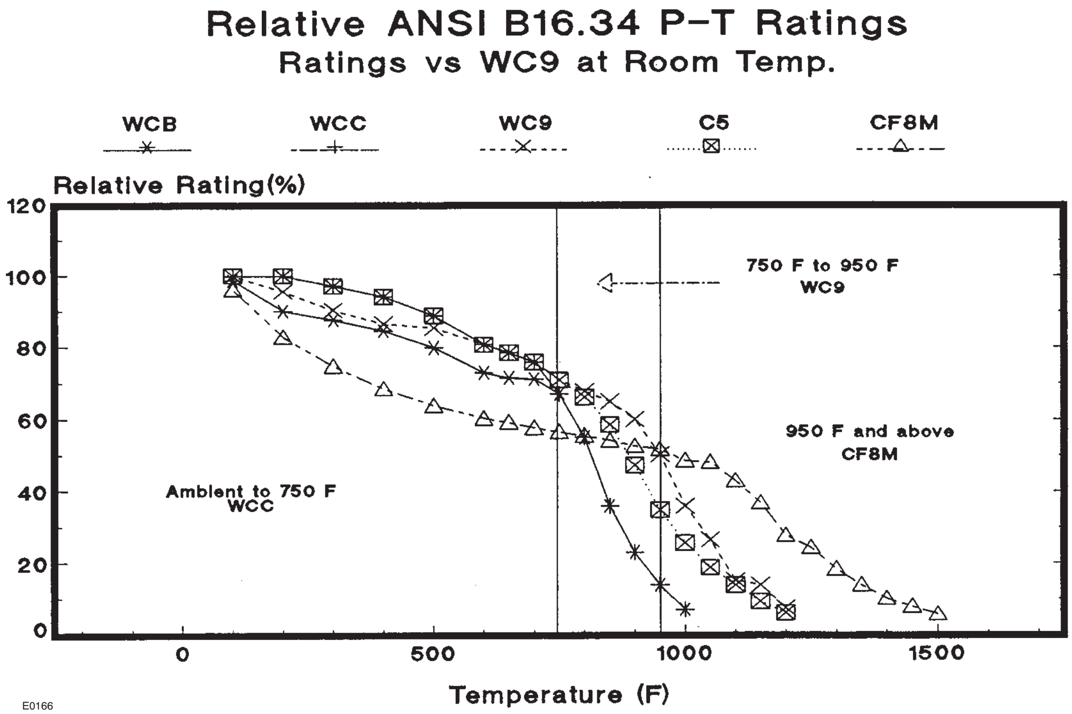


Figure 10-1.

Figure 10-1 is a plot of the pressure-temperature ratings for WCB, WCC, WC8, C5, and CF8M where the allowable pressure has been normalized to 100%. This curve is representative of the relative pressure-temperature ratings of these materials for ANSI Classes 300 through 4500. Note that if the material providing the maximum allowable pressure at any temperature is determined from the plot, three material regimes can be established. From ambient temperature to 700°F, WCC (and C5) have the highest P-T ratings of this group of materials. From 700-950°F, WC9 has the highest ratings, and from 950°F up, CF8M has the highest ratings.

Trim Parts

Valve trim components have much different material requirements than valve bodies and bonnets. They are not pressure retaining, so they are not directly safety related. However, since the trim components provide the flow control, they are very important with respect to the overall performance of the valve. In general, trim materials must have excellent resistance to corrosion by the process fluid in order to maintain adequate flow control and mechanical stability. Each individual component must possess certain other characteristics depending upon the valve design and application. The following is a breakdown of the specific properties necessary in the various trim components.

Plugs

Valve plugs provide throttling control and shutoff in globe valves, and are directly impinged by the flow stream. The seating surface on the valve plug must be capable of withstanding the seat loads required for shutoff. It must also withstand the erosive forces generated by fluid jets from Cavitrol® trim or during low-lift throttling. In cage-guided valves, the plug guide surfaces must resist galling and excessive wear when sliding against the cage material, and must withstand the erosive action caused by clearance flow between the plug and cage.

Cages

Valve cages can serve a number of functions in the control valve depending upon the valve design. In most control valves, the cage provides plug guidance and is involved in flow control. In some of these designs, the cage transfers a portion of the bonnet bolt loading to the seat ring to hold it in place and maintain gasket loading. In Cavitrol® designs, the cage provides a portion of the pressure drop the fluid experiences within the valve. Thus, the cage may be required to withstand axial compressive loading due to the bonnet bolt load and radial compressive loading due to the pressure differential across it. It must provide good sliding wear properties in combination with the plug material. In some designs, it must provide good erosion resistance.

Seat Rings

Seat rings work with the valve plug to provide shutoff. They must be able to withstand the seat loads required for shutoff as well as the erosive forces caused by the high fluid velocities that can be encountered during low-lift throttling. In some designs, the seat ring is an integral part of the cage. In certain seat ring designs, the seat ring flange must be strong enough to withstand the bending loads imposed on it by the seat ring retainer.

Stems

The valve stem connects the valve plug with the valve operator through the packing box in the bonnet. The valve stem must be resistant to both general corrosion and pitting so that leakage and/or damage to the packing will not occur. The stem has to be strong enough to sustain the loads required to provide good shutoff without bucking or yielding. In certain designs, the stem must provide good sliding wear properties in combination with one or more guide bushings and with the valve packing.

Common Trim Materials

There are a large number of material combinations that have been used for valve trim through the years. Fisher has several combinations that are considered to be somewhat standard, and these have proven to provide excellent performance in a very wide range of applications.

Standard Trim Combination for Cage-Guided Globe Valve

Most Fisher globe valves used in the power industry are cage guided. The standard valve trim (usually called Trim 1) consists of:

Cage	17-4PH or CB7Cu-1
Plug	416 HT HRC 38 min
Seat Ring	416 HT HRC 38 min
Stem	316 Condition B

This material combination provides good performance at a minimum cost. The 17-4/CB7Cu-1 cage material has high strength properties, and displays good wear resistance combined with the 416 plug for most general applications.

The 416 plug and seat ring have good erosion resistance, and are easily machined and finished to provide shutoff. The stem material is a Fisher standard in nearly all product lines, and provides excellent corrosion and pitting resistance in most environments.

The 17-4PH or CB7Cu-1 material is usually supplied in the H900 condition, which provides the highest strength and hardness. However, the H1075 condition is being used in all Cavitrol® cages because of its superior resistance to stress corrosion cracking. This is necessary to avoid cracking of the outer sleeves which are shrink-fit around the main portion of the cage. The use of the H1075 condition has caused no decrease in wear performance even though the hardness is lower.

Trim for High Temperature/Pressure

When higher temperatures and/or pressures are involved, the standard trim combination may not be acceptable due to inadequate yield strength, creep resistance, or sliding wear resistance. Fisher utilizes a number of trim material combinations for these applications depending upon the valve style and the actual operating conditions.

One of the most common trim combinations consists of a 316 or CF8M chromium coated cage, a 316/CoCr-A plug, and a 316/CoCr-A or solid Alloy 6 seat ring. The chromium coating used on these cages is a proprietary material that provides galling resistance up to 1100°F. Normal chromium plating fails above approximately 500°F. The combination of this coated cage and a CoCr-A hardfaced plug provides excellent high temperature mechanical performance, and the CoCr-A hardfaced or solid Alloy 6 seating surfaces provide excellent erosion resistance in most applications.

The Super 12% chromium stainless steels are a newer family of specialty 400 series stainless steels. They contain small percentages of alloying elements such as tungsten, vanadium, etc., which serve to increase their resistance to tempering at elevated temperatures. These materials are sometimes used in plugs and cages (usually chromium coasted) for high temperature steam applications. Availability of these materials has been poor.

Alloy 6 Corrosion

Cobalt-chromium Alloy 6 has been a mainstay valve trim hardsurfacing material for many years. It has been used successfully for most applications considered either non-corrosive or slightly corrosive. Until recently, boiler feedwater was classified as one of those "non-corrosive" applications. However, over the last ten years, more and more failures of Alloy 6 have been reported in valves which are controlling feedwater in electric utility power plants as well as in cogeneration power plants.

Alloy 6, commonly called Stellite 6 (Stellite is a registered trademark of Stoody Deloro Stellite) is available in several forms, all of which have separate grade designations and slight variations of composition. Briefly, the cast material designation is UNS R30006, and the hardsurfacing material is designated as AWS CoCr-A. The wrought material designation is Alloy 6B, but it is commonly called Haynes Alloy 6B (Haynes is a registered trademark of Haynes International.)

Alloy 6 is a cobalt-chromium-tungsten alloy with approximately 1% carbon. The material consists of a soft matrix of cobalt-chromium-tungsten solid solution surrounding a small percentage of hard, brittle chromium carbides. In the case of the wrought material, the carbides are dispersed uniformly throughout the network.

It is the carbide phase that provides the hardness of approximately HRC 40, but research has shown that it is the cobalt-chromium-tungsten matrix that is responsible for the excellent wear and cavitation resistance that Alloy 6 demonstrates. This is due to a phase transformation (change in crystal structure) that Alloy 6 undergoes when it is highly stressed, such as in a wear or cavitation implosion situation. This phase transformation requires energy, so some of the energy that would cause damage is absorbed instead, thereby reducing the level of damage.

Alloy 6, like most corrosion resistant alloys containing chromium as an alloying element, obtains its corrosion resistance from the formation of a stable, chromium-containing oxide passive layer. The oxide layer protects the underlying material from further reaction with the environment. In the presence of certain chemicals, this passive layer can be weakened until it fails to protect the material from corrosion.

This loss of corrosion resistance is most often noticed in boiler feedwater applications where the water is treated with hydrazine or some other amine derivative. It occurs exclusively in regions where the flow velocity is high, indicating that the failure mode is actually erosion-corrosion. There are at least two possible explanations for this phenomenon:

1. The presence of the amines weakens the oxide passive layer so that it erodes easily. The passive layer is repeatedly eroded away, rebuilt, eroded away, rebuilt, etc., resulting in accelerated corrosion.

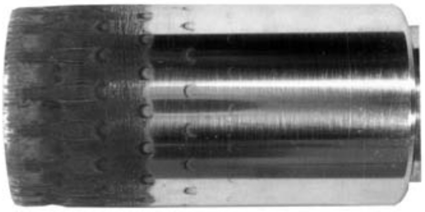
2. The presence of amines prevents the oxide passive layer from reforming after it has eroded away initialy, thus leaving the alloy unprotected from corrosion. Further erosion-corrosion then occurs at accelerated rates.

There may be other possible mechanisms for this type of failure, but the point remains that the alloy is attacked at higher rates than would be expected in an equivalent water application without the presence of the amine compounds.

Studies of the returned parts have demonstrated that the attack definitely correlates to the presence of Alloy 6. Figure 10-2 shows a valve plug with Alloy 6 seat and guide surfaces that have suffered erosion-corrosion damage. Figure 10-3 shows how the plug was sectioned for the examination. It highlights a damaged area that appears as a light colored band adjacent to the outer diameter of the plug.

Figure 10-4 shows the sectioned area after removal from the plug. The circle shows the location where the photograph in Figure 10-5 was taken. The "A" and "B" planes are identified on both.

The "A" surface was metallographic polished and etched to aid in identification of the base material (S31600 SST) and the hardsurfaced layer (CoCr-A). The scanning electron microscope (SEM) photomicrograph of polished and etched face (plane "A") in Figure 10-5 clearly shows the interface between the S31600 and the CoCr-A (identified by the lower arrow). The material to the right of the interface is CoCr-A, and to the left is the S31600 base material.

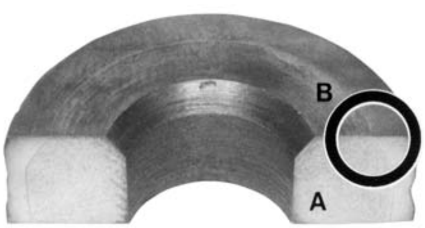


W5703

Figure 10‐2. Side view of valve plug with CoCr‐A hardsurfaced seat and guides showing erosion‐corrosion damage.

W5702

Figure 10‐3. End view of the same plug shown in Figure 10‐2. Note the light band adjacent to the O.D. and the outside of the plug where the erosion‐corrosion damage has occurred. This view also shows how the sample was removed for further evaluation.

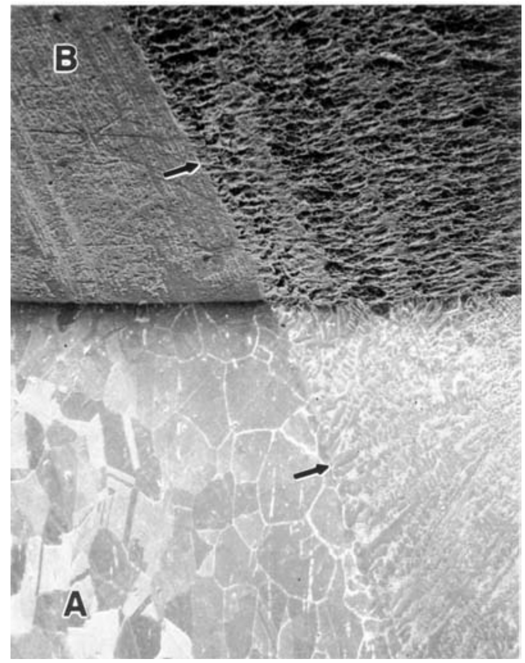


W5701

Figure 10‐4. Photograph of the sample, removed per Figure 10‐3. This photograph was taken after metallographic polishing and etching of face "A", which is a cross‐section perpendicular to plane "B", the bottom of the plug. The circle identifies the region shown in Figure 10‐5.

Note that the interface between the damaged (right) and the undamaged (left) regions in the unpolished face (plane "B"), identified by the upper arrow, corresponds with the interface between the two materials in the polished and etched plane. It is clear that the damage is occurring exclusively in the hard CoCr-A material on the right. The soft S31600 material on the left is unchanged.

Most failures have occurred in feedwater regulator valves. Failures have included special and



W5700

Figure 10‐5. Scanning electron microscope photograph of the region shown in the circle in Figure 10‐4. Note the microstructures of the S31600 (left‐hand side) and the CoCr‐A (right‐hand side) in plane "A". Also note that the interface between the two materials (lower arrow) coincided with the interface between the damaged and undamaged areas on plane "B" (upper arrow). This indicates that the CoCr‐A material is being preferentially damaged. Original Magnification: 100X.

standard trim valves operating at temperatures as low as 300°F and pressure drops as low as 100 psi. Similar failures have also been experienced with tungsten carbide trim (which utilizes cobalt binders) in hydrazine-treated feedwater as well as in ammonia applications (ammonia is chemically related to hydrazine.)

There are no specific guidelines dictating hydrazine content, temperature, or velocity limits which are safe for use with these materials. It is best to simply avoid the use of cobalt-containing alloys in feedwater service unless the feedwater is known to be compatible with Alloy 6.

In most cases, the attack is minimized through the use of hardened stainless steel trim materials such as S41600 (Type 416 SST), S41000 (Type 410 SST), S42000 (Type 420 SST), S44004 (Type 440C SST), and or S17400 (17-4 PH SST). In some severely erosive applications, the use of nickel-chromium-boron hardsurfacing materials such as NiCr-C (commonly called Colmonoy 6, a trademark of Wall Colomonoy), have been necessary.

There are instances where CoCr-A hardfacing or Alloy 6 castings must be avoided. The use of hydrazine for removing oxygen from feedwater has become very popular in recent years, and along with its increased use Fisher has experienced an increased number of returns of CoCr-A hardsurfaced and solid Alloy 6 valve trim parts.

Most failures have occurred in boiler feedwater regulation valves utilizing standard trim types and operating at temperatures as low as 300°F and pressure drops as low as 100 psi. It is suspected that these failures actually are caused by erosion-corrosion, and that the elimination of free oxygen by the hydrazine inhibits the re-passivation of the cobalt-containing phase.

Similar failures have been experienced also with tungsten carbide trim (which utilizes cobalt binders) in these types of treated feedwaters as well as in certain ammonia applications (ammonia is related to hydrazine chemically). There are no specific guidelines dictating hydrazine contents or velocity limits that are safe for use with these materials. It is simply to avoid the use of cobalt-containing alloys in feedwater service.

The Super 12% chromium stainless steels are a newer family of specialty 400 Series stainless steels. They contain small percentages of alloying elements such as tungsten, vanadium, etc., which serve to increase their resistance to tempering at elevated temperatures. These materials sometimes are used in plugs and cages (usually chromium coated) for high temperature steam applications. Availability of these materials has been poor.

Cavitrol IV and EHAT Plugs

The Cavitrol IV and EHAT valve plug designs introduce a need for dimensional stability that is not present in standard valve plugs. These plugs are very long, contain slender sections, and are very susceptible to warping during high temperature processing. For this reason, standard plugs are currently made from 440C with hardened seats and guides. Some plugs are constructed from 316 with CoCr-A or NiCr-C hardfaced seats and guides, but these materials are only utilized when absolutely necessary.

Bolting

Bolting is a confusing subject because of the many different grades and the fact that the grade designations are different than those used for other products. While there are a large number of materials with slightly different compositions, heat treatments, and resulting mechanical characteristics, most needs can be met with just a few grades of material.

The most common bolting materials used in Fisher products are ASME SA193 Grades B7, B8M, and B16. The ASME Boiler and Pressure Vessel Code Sections VIII allowable stresses (required per ASME/ANSI B16.34) are the main criteria used to determine bolting materials for most applications. Figure 10-6 shows the allowable stresses for these three bolt materials as a function of temperature.

B7 Bolting

ASME SA193 Grade B7 bolting is actually a 4140 or similar alloy steel that has been heat treated to provide certain mechanical properties. Grade B7 is the standard bolting material supplied with Fisher products, offering excellent strength over a large temperature range, thermal expansion rates closely matching those of WCC and WC9, excellent availability, and reasonable cost. Grade B7 can be used to 1000°F, although above 700°F its allowable stress is lower than for Grade B16.

Grade B16

ASME SA193 Grade B16 is a 4140 material with additional vanadium and extra molybdenum to give it superior high temperature properties. It also matches the thermal expansion properties of WCC and WC9. It is mainly used for temperatures above 700°F in conjunction with alloy steel bodies and bonnets.

B8M Bolting

ASME SA193 Grade B8M bolting is 316 stainless steel. B8M is used for high-or-low-temperature applications or to match the thermal expansion characteristics of a CF8M body and bonnet. B8M is available in two strength levels, annealed and strain-hardened, which are identified as Class 1 and Class 2, respectively.

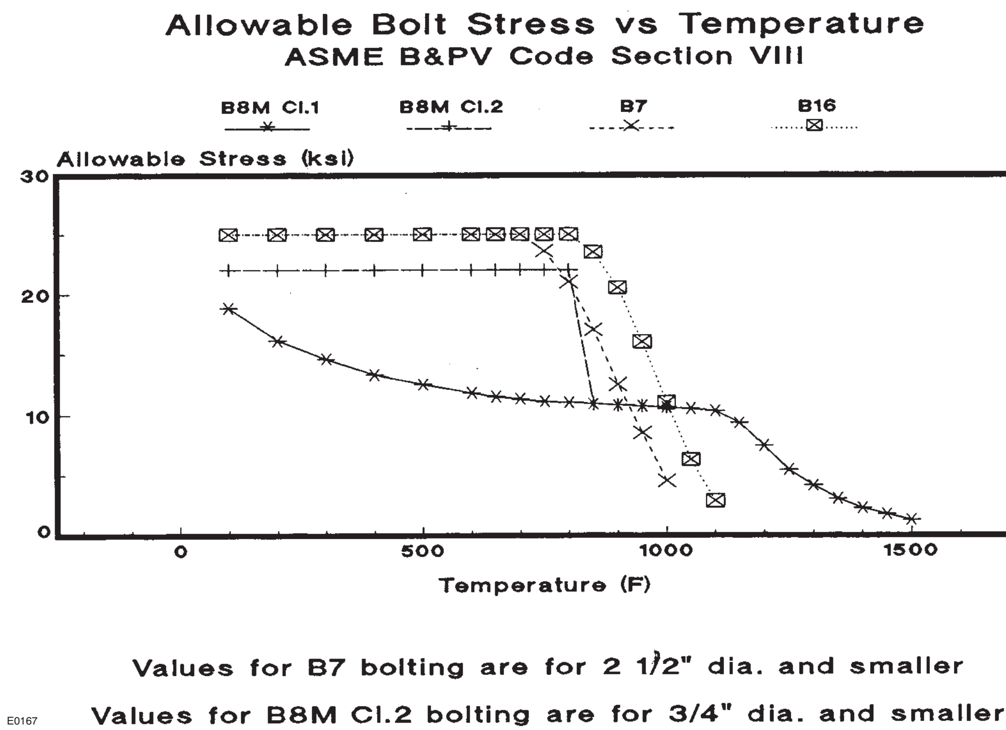


Figure 10-6.

Allowable stress levels are listed for B8M Class 1 bolting up to 1500°F, and above 1000°F its allowable stress values are greater than for B16.

B8M Class 2 bolting has higher allowable stress values up to 800°F due to the strain hardening. However, above 800°F, annealing reduces the strain-hardening effects. For this reason, the allowable stresses for B8M Class 2 are equal to those for B8M Class 1 from 850°F to 1000°F, and its use is not permitted above 1000°F.

Materials Standardization

Due to the number of companies in different industries served by Fisher Controls, a wide variety of materials are specified by customers for both pressure retaining and non-pressure retaining parts. In a large number of these instances, the materials specified are not commonly utilized in Fisher products, or are not standard materials for the particular product being ordered. Sometimes the use of these materials is necessary, but in many cases a material that is standard to that Fisher product could be utilized more economically and with a shorter leadtime. For example, the following are lists of ANSI approved casting materials:

<u>Low-Temperature Use</u>

LCB

<u>LCC</u>

LC3

<u>CF8M</u>

<u>Intermediate-Temperature Use</u>

WCB

<u>WCC</u>

<u>Elevated-Temperature Use</u>

WC1	C5
WC4	C12
WC5	CF8
WC6	<u>CF8M</u>
<u>WC9</u>	

The materials are listed accordingly to increased capability in their particular category of use. They are also listed in order by price, if it is assumed that alloying element content is the sole factor in determining price. Based upon performance and price, applications exist for which each alloy in the list would be the most economical valve body material.

It would be uneconomical for a company with a widely varied product line to stock all of its valve bodies in all of these materials. For each Fisher product line, standard valve body materials are offered.

When customers order non-standard body materials, then special castings needed from the foundry, which increases costs and delays delivery of the finished product. The end result is that the customer is often unsatisfied with the delivery performance, and the non-standard material actually costs more than a superior standard material.

The best approach to the selection of materials is to utilize standard materials offered where possible. The underlined materials in the above list are the proposed universal standard materials for Fisher globe and rotary valve bodies. Note that these materials cover the entire range of low-, intermediate-, and high-temperature applications. Utilizing these materials will result in better delivery and, in the long run, better value.

This philosophy for utilizing a few standard materials extends to trim as well as valve bodies. Most applications can be handled by one of the standard trim offerings. Trim materials vary much more throughout the product line than do valve body materials, since many of the valve designs have been targeted to particular applications. However, certain standard trim materials are used in the majority of products. The following is a listing of the proposed standard materials for globe and rotary valve components.

Globe Valves

Bodies/Bonnets

LCC

WCC or WCB

WC9

CF8M

Plugs

S41600

S44004

S31600/CoCr-A

Cages

S17400 (or CB7Cu-1)

S31600 (or CF8M)/(CoCr-A)

S31600 (or CF8M) ENC

R30006

Seat Ring

S41600

S44004

S31600 (or CF8M)/CoCr-A)

R30006

Stems

S31600 Condition B

S31603 Strain Hardened

Bolting

B7

B16

B8M

Rotary Valves

Bodies / Bonnets

LCC

WCC or WCB

WC9

CF8M

Balls / Discs

WCB / CrPlate

CR8M or CG8M / CrPlate

CF8M or CG8M / CoCr-A

Alloy 6

Seal RIngs

316

316 (or CF8M) / CoCr-A

Alloy 6

Shafts

316

17-4PH H1075

Nitronic 50

Bolting

B7

B16

B8M

When specifying valve materials, use the Fisher catalogs to determine the standard materials of construction, and try to use one of the standard combinations if possible. Using a standard construction can save money and improve lead-time, even though it might seem to be a premium choice compared with some non-standard material(s) that would perform satisfactorily.

Standardized Metallic Materials Designations

Valves used in the process control industry contain a wide variety of metallic materials due to the broad range of chemicals, temperatures, and pressures involved. Over the years, these materials have come to be known by many designations, including popular trade names, American Iron and Steel Institute (AISI) designations, American Society for Testing and Materials (ASTM) designations, etc.

The lack of a standard for material designations has caused a great deal of confusion between customers and suppliers throughout industry. In the recent past, several standards organizations and trade associations have made an effort to standardize alloy designations in order to alleviate this problem. The following sections describe the UNS and ACI designation systems, which are two of the most common designation systems being utilized.

Unified Numbering System (UNS)

The most widely accepted designation system is the Unified Numbering System developed by the Society of Automotive Engineers (SAE) and ASTM. This system provides a uniform method of designating metallic materials. The UNS system divides metals and alloys into 18 series.

Within each series the material designation starts with a single alpha character, which in many cases is suggestive of the family of metals it identifies (i.e., “A” for aluminum alloys, “C” for copper alloys, “N” for nickel alloys, “S” for stainless steels, etc.).

Following the alpha character are five numeric digits, which likewise often suggest alloys within the family of metals (e.g., A92024 for 2024 aluminum, C36000 for copper alloy 360, S31600 for type 316 stainless steel, N04400 for Monel® 400, N10276 for Hastelloy® C276, G10180 for 1018 steel, etc.).

The publication “Metals and Alloys in the Unified Numbering System” is a complete listing of all UNS numbers assigned to date. In addition, it provides a cross-reference for UNS numbers, UNS material descriptions, tradenames, compositions, and specifications. It can be obtained from the Society of Automotive Engineers, Inc., 400 Commonwealth Drive, Warrendale, PA 15096.

ACI Designations

A former division of the Steel Founder’s Society called the Alloy Casting Institute (ACI) developed a system for designating stainless and heat resisting casting alloys.

Casting designations per this system begin with either a “C” (for corrosion resistant materials) or a “H” (for heat resistant materials).

The second letter in the designation ranges from “A” to “Z” depending upon the nickel, and to a lesser degree, the chromium content.

For example, a corrosion resistant material with 12% chromium and no nickel begins with “CA” (example, CA15). An alloy with 100% nickel begins with “CZ” (example, CZ100). The following numeric digits indicate the maximum carbon content (percent x 100) of the alloy.

Additional letters following the numeric digits indicate the presence of supplementary alloying elements.

Two groups of materials have designations that do not follow the above scheme. Nickel-copper materials use “M” as the first letter (examples are M35-1 and M25S). Nickel-molybdenum materials utilize “N” as the beginning letter (such as N7M and N12MV).

Although ACI no longer exists, the system has been adopted by ASTM, and designations for new alloys are assigned by the appropriate ASTM committees. UNS numbers have also been assigned to many of these alloys; however, they are less systematic than the ACI designations and are therefore much more difficult to utilize.

Industry Trends

A survey of codes and standards commonly referenced in the process control industry (in particular, ANSI, ASME, and ASTM) indicates that no single designation system is being used for metallic materials. However, a comparison of today’s codes and standards with those from several years ago reveals trends indicating the direction being taken with regard to material designations. For the most part, a combination of three designation systems is being used:

• UNS numbers are favored over other systems for nearly all wrought products, including AISI carbon and alloy steels, stainless steels, nickel alloys, copper alloys, aluminum alloys, etc. UNS numbers are also favored for cast aluminum and copper alloys.

• ACI designations are used for all cast stainless and heat resisting steels and cast nickel-base alloys.

• ASTM/ASME designations have been retained for many special carbon and alloy steel products. In addition, cast iron materials have retained their ASTM grade designations. The fact that UNS numbers have not been adopted for these materials within the codes and standards is probably due to the difficulty in relating their UNS numbers with the commonly known designations.

Fisher Standard Designation System

Fisher Controls has documented a standard designation system for use within all of its internal and advertising literature. The designation system basically matches what ANSI, ASME, and ASTM use for designations at this time, and is based predominantly on the UNS, ACI, and ASME/ASTM designation systems.

In the cases where the Fisher system does not currently match the ANSI, ASME, and ASTM codes/standards, Fisher has attempted to anticipate which designations will be utilized in future revisions of those codes/standards based upon current trends.

Fisher Controls’ Technical Monograph 35 Standardizing Metallic Material Designations provides a complete discussion of the Fisher standard designation system. It also includes tables allowing the user to cross-index standard designations with other designations for a large number of common materials.

Chapter 11

Packing Materials and Systems

This chapter provides information and insight to help in selection of packing systems. Application guidelines for each packing selection have been developed to aid in the packing selection process. These guidelines were developed based on test results of various packing systems conducted in Fisher's technical center. Exceeding these guidelines may result in reduced service life.

The ratings for these systems do not affect the valve pressure/temperature class rating. Traditional valve selection entails selecting a valve design based on pressure and temperature capabilities, flow characteristics, and material compatibility. An additional factor—packing selection—is now a critical factor in the valve selection process.

Proper packing selection is being driven in the United States by the Clean Air Act Amendments, subsequent Environmental Protection Agency (EPA) regulations, and our customers' increasing concern for improved packing performance (less maintenance and longer service life.)

In recent years, packing selection has been primarily based on process temperature; that is, PTFE was selected for temperatures below 450°F (232°C), and graphite was selected for temperatures above 450°F (232°C). Considerations now include the effect of packing friction on process control, hysteresis, seal quality, and cycle life.

Given the variety of process applications and installation conditions, these variables are difficult to quantify. A relative packing performance comparison can be made that provides an engineered approach to the packing selection process.

Clarification of trade names is required for proper understanding of the Table 11-1 shown in this publication. From the perspective of Fisher, ENVIRO-SEAL® packing is defined as an advanced system using a "compact", live-load spring design. From a user perspective, ENVIRO-SEAL packing is most typically thought of as an emission-reducing packing system; however, as shown in the following chart and graphs, it is highly useful in non-environmental applications as well.

Although other packing materials and combinations of packing materials are available, the recommendations for nearly all applications are listed in Table 11-1. Simply select the appropriate packing based on pressure, temperature, leakage requirements, service life required, etc. Use the charts for additional guidance.

The packing selection guidelines in Table 11-1 present two categories of service conditions. The first category defines those packing systems designed for environmental or fugitive emission applications where 500 parts per million volume (ppmv) seal performance is required.

The second category defines application guidelines for non-environmental services. Depending on category requirements, different pressure/temperature guidelines have been established for the packing systems.

In addition, a given packing design has certain characteristics concerning seal performance, service life, and friction. The pressure and temperature guidelines and the relative comparison of these characteristics are defined in Table 11-1 and the supporting charts.

Table 11-1. Packing Selection Guidelines for Sliding-Stem Valves

Packing System	Maximum Pressure & Temperature Limits for 500 PPM Service[1]		Application Guideline for Nonenvironmental Service[1]		Seal Performance Index	Service Life Index	Packing Friction[2]
	Customary U.S.	Metric	Customary U.S.	Metric			
Single PTFE V-Ring	300 psi 0 to 200°F	20.7 bar –18 to 93°C	See figure 11-1 –50 to 450°F	See figure 11-1 –46 to 232°C	Better	Long	Very low
ENVIRO-SEAL PTFE	See figure 2 –50 to 450°F	See figure 2 –46 to 232°C	See figure 11-1 –50 to 450°F	See figure 11-1 –46 to 232°C	Superior	Very long	Low
ENVIRO-SEAL Duplex	750 psi –50 to 450°F	51.7 bar –46 to 232°C	See figure 11-1 –50 to 450°F	See figure 11-1 –46 to 232°C	Superior	Very long	Low
KALREZ® with PTFE (KVSP 400)[3]	350 psig 40 to 400°F	24.1 bar 4 to 204	See figure 11-1 –40 to 400°F	See figure 11-1 –40 to 204°C	Superior	Long	Low
ENVIRO-SEAL Graphite ULF	1500 psi 20 to 600°F	103 bar –7 to 315°C	3000 psi –325 to 700°F	207 bar –198 to 371°C	Superior	Very long	High
HIGH-SEAL Graphite with PTFE	1500 psi 20 to 600°F	103 bar –7 to 315°C	4200 psi[4] –325 to 700°F	290 bar[4] –198 to 317°C	Superior	Very long	High
HIGH-SEAL Graphite	– – – – – –	– – – – – –	4200 psi[4] –325 to 1200°F[5]	290 bar[4] –198 to 649°C[5]	Better	Very long	Very high
Braided Graphite Filament	– – – – – –	– – – – – –	1500 psi –325 to 1000°F[5]	103 bar –198 to 538°C[5]	Acceptable	Acceptable	High

1. The values shown are only guidelines. These guidelines can be exceeded, but shortened packing life or increased leakage might result. The temperature ratings apply to the actual packing temperature, not to the process temperature.
2. See Catalog 14 for actual friction values.
3. The KALREZ pressure/temperature limits referenced in this bulletin are for Fisher Controls valve applications only. DuPont may claim higher limits.
4. Except for the 3/8-inch (9.5 mm) stem, 1600 psi (110 bar).
5. Except for oxidizing service, –325 to 700°F (–198 to 371°C).

Keep in mind that for non-environmental applications these are guidelines and are not intended as the ultimate limit. Exceeding the pressure guidelines for packing may be required; however, depending on how much the guidelines are exceeded, a reduction in total life span can be expected.

With flexible graphite packing this same logic applies to temperature. The recommended pressure / temperature limits for environmental applications using Fisher ENVIRO-SEAL packing cannot be exceeded without compromising sealing integrity.

Packing Set Possibilities

Single PTFE V-Ring Packing (Sliding-Stem and Rotary Valves)

The single PTFE V-ring arrangement uses a coil spring, as in the **easy-e** valves, and meets the EPA 500 ppmv criteria, assuming that the pressure does not exceed 300 psi (20.7 bar) and the temperature is between 0°F and 200°F (-18 and 93°C). This packing arrangement offers very good seal performance with the lowest packing friction and usually the lowest cost.

ENVIRO-SEAL PTFE Packing (Sliding-Stem and Rotary Valves)

The ENVIRO-SEAL PTFE arrangement is suitable for environmental applications on services up to 750 psi and 450°F (51.7 bar and 233°F). The sealing capability is superior. The ENVIRO-SEAL PTFE packing system is designed to operate at nearly a constant high stress. This design approach results in an increase in stem friction. Typically, the slight increase in stem friction does not cause problems with actuator sizing or process control, such as hysteresis. It is always recommended that actuator sizing be verified whenever you select a different packing material.

KALREZ with PTFE V-Ring Packing (Sliding-Stem Only)

KALREZ with PTFE V-ring (KVSP-400) is limited to 350 psi pressure and 400°F (24.1 bar and 204°C) applications. KALREZ with PTFE V-Ring is currently available for **easy-e** valves. KALREZ packing is also available in KVSP-500 (500°F; 260°C service). The KVSP-500 series uses another type of DuPont V-ring material called ZYMAXX, which is a carbon fiber-reinforced TFE.

Note that KALREZ packing arrangements require that a controlled low stress be applied to the packing in order to seal properly and also to ensure a longer life. This is achieved by using springs that deliver the required low force.

KALREZ packing arrangements seal effectively, due to the resilient properties of the KALREZ material and the shape of the V-rings, which concentrate the applied stress on the perimeter of the packing rings. Laboratory testing has shown excellent seal performance when the packing arrangement was loaded in this manner. However, in additional tests where high stress was applied to the packing arrangement, the V-rings were damaged and material extruded out of the packing arrangement. This test result confirmed the need for a controlled low stress when using the KALREZ arrangements.

In contrast, the ENVIRO-SEAL PTFE packing system is designed to operate at high stress (approximately 10 times the KALREZ stress). This gives the ENVIRO-SEAL PTFE packing system the ability to tolerate less-than-perfect conditions and continue to seal reliably. For example, in changing installed valves to ENVIRO-SEAL PTFE packing, minor imperfections in the stem finish or packing bore can be tolerated because of the high stress design of the packing system. KALREZ with PTFE V-ring might have difficulty sealing in this situation.

The KALREZ with PTFE V-ring packing arrangement is basically the same packing concept as the **easy-e** single PTFE V-ring packing, except two of the PTFE V-rings have been replaced with two KALREZ V-rings. The KALREZ V-rings can be identified by their black color.

The total cost (initial and on-going maintenance costs) will be less for the ENVIRO-SEAL PTFE packing systems than for KALREZ packing arrangements. On reviewing the pricing, you will notice that KALREZ packing is markedly higher in price than ENVIRO-SEAL PTFE. With KALREZ packing, each time the packing is replaced when maintaining the packing, the KALREZ packing cost will be repeated. For ENVIRO-SEAL packing systems, the replacement packing is only slightly higher than a set of single PTFE V-ring packing.

Flexible Graphite Packing Material (Sliding-Stem Only)

It is important to understand some of the issues associated with the base material that is common to all high temperature graphite packing systems. The single most important issue is the potential for galvanic corrosion in valves in storage that have graphite packing system installed.

Fisher uses only the highest grades of base flexible graphite material that contain a non-metallic, inorganic, passivating inhibitor for galvanic corrosion resistance. All Fisher graphite packing systems (with the exception of nuclear applications) also include sacrificial zinc washers to prevent galvanic corrosion.

However, even with these preventive measures, under humid conditions, in the presence of chlorides, with air or oxygen present, there is potential for galvanic corrosion of the valve stem. Because there are numerous variables associated with galvanic corrosion it is near impossible to predict when it may occur.

The best policy is to remove the packing if the valve is to be stored for a long period. This, however, may create a safety issue if someone were to use the valve not realizing that there was no packing in the valve. For this exact reason all Fisher valves are supplied with the packing installed.

ENVIRO-SEAL Graphite ULF (Sliding-Stem and Rotary Valves)

ENVIRO-SEAL Graphite ULF packing is designed for environmental applications from 300 to 600°F (149 to 316°C), or for those applications where fire safety is a concern and graphite packing is desired. The ENVIRO-SEAL Graphite ULF arrangement can also be used with higher pressures up to 1500 psi (103 bar) and still maintain the 500 ppmv EPA leakage criteria.

ENVIRO-SEAL Graphite ULF packing systems were developed by utilizing the benefits of both PTFE and graphite components. These special packing systems provide the firesafe capability of graphite packing along with reduced friction advantages of PTFE packing. Thus, ENVIRO-SEAL Graphite ULF packing systems provide a lower friction, lower emission, fire-tested solution for applications with process temperatures up to 316°C (600°F). ENVIRO-SEAL Graphite ULF packing systems were tested in accordance with API Standard 589, “Fire Test Evaluation of Valve Stem Packing”, second edition. See Figure 11-1 for applications guidelines for 500 ppm service. Figure 11-4 shows the ENVIRO-SEAL Graphite ULF packing system.

HIGH-SEAL™ Graphite with PTFE for Environmental Applications (Sliding-Stem Only)

The HIGH-SEAL packing system features live-loading and unique packing ring arrangements for consistent, long-term sealing performance. It is offered without PTFE washers for nuclear service where PTFE is prohibited and with PTFE washers for all other applications.

The primary advantage of the HIGH-SEAL packing system arrangement is the use of large Belleville springs, which can be calibrated with a load scale. The load scale provides a visual indication of packing load, making it easier to determine when additional torque might be required.

HIGH-SEAL packing systems have a disadvantage, from an installation perspective. This disadvantage is that the actuator cannot be removed from the valve without removing the Belleville springs.

The HIGH-SEAL graphite packing system provides a service life that is very long compared to other graphite packing arrangements.

Non-Environmental Services

For the non-environmental pressure/temperature guidelines, using ENVIRO-SEAL and HIGH-SEAL packing systems can extend service life.

As service pressures and temperatures increase, the ENVIRO-SEAL packing systems provide a significant step change in performance, compared to the traditional single PTFE V-ring or braided graphite filament arrangements. For example, an ANSI Class 1500 (Design HP valve) application at 2000 psi and 200°F (137 bar and 93°C) frequently required packing maintenance. The HIGH-SEAL packing system with PTFE washers was installed, significantly reducing maintenance requirements. This system is rated to 420 psi and 700°F (290 bar and 371°C). The ENVIRO-SEAL packing system or compact Belleville spring arrangement is rated to 3000 psi and 700°F (20.7 bar and 371°C).

Table 11- 1 shows that the braided graphite filament arrangement is limited to 1500 psi (103 bar). Due to its construction, braided graphite filament cannot sustain high packing stress levels. Over time, the braided graphite filament will break down and compress, and sealing force will be lost. Due to these characteristics, braided graphite filament will not provide the required seal performance for environmental services. The braided graphite filament arrangement is also susceptible to stick-slip action, which might cause process control deviations.

Rotary Valve Considerations

The basic principles of sealing a rotary shaft are the same as sealing a sliding stem. However, experience shows that sealing the shaft on a rotary valve is less difficult than on sliding stem applications. The rotary motion does not tend to remove material from the packing box as does the sliding stem motion. Also, many of the rotary valve applications tend to be lower pressure applications than sliding stem valve services. As with sliding stem valves, the basic material recommendation for rotary valve packing is either PTFE or flexible graphite.

Single Packing Arrangements

Used in most applications for a durable, yet economical package.

Double Packing Arrangements

Where an especially tight seal is required, double packing is available. Consisting of two stacks of packing and a spacer, double packing arrangements have more packing on the actuator side if a lubricating connection is used.

Leak-Off Packing Arrangements

Certain rotary valve applications require that any leakage from the inner set of packing be detected and channeled away before it can leak through the outer set to the atmosphere. Leak-off packing arrangements are designed for this purpose.

Some standard packing arrangements, such as double PTFE V-ring, that have packing on either side of a lubricator connection can also be used for leak-off applications. However, standard packing arrangements are not specifically designed for leak-off applications and do not have a suitable ratio of packing on either side of the lubricator.

Purged Packing Arrangement

Certain rotary valve applications require that a lantern gland and a tapped pipe connection be provided inboard of the packing itself to permit purging of any accumulated deposits around the valve shaft and bushing. This arrangement is recommended for handling fluids with entrained solids and where fluids condense or solidify with loss of temperature.

Oxygen Service

Because liquid or gaseous oxygen supports the combustion of most lubricants or foreign material, special precautions must be taken when cleaning and handling oxygen service equipment. Be sure to specify oxygen service when ordering packing or other equipment for this service so the packing will be appropriately degreased.

Process Fluid Compatibility and Temperature Ranges

Fisher's published literature lists packing types that are used with various process fluids. However, some process fluids may not be compatible with the packing throughout the entire temperature range. If in doubt about the compatibility of Fisher's packing material with a process fluid of a specific concentration and temperature, consult your Fisher Representative sales office.

Published temperature ranges refer to temperatures in the packing box. When using any packing at low temperatures, frost allowed to form on the shaft can damage the packing. For a given process temperature, the packing box temperature depends on several factors. Contact the Fisher sales representative for information on packing box temperatures.

Conclusion

It is important to recognize that many parameters affect seal performance and service life. Even if the optimum design is selected, other factors such as stem finish, packing bore finish, and installation practices will have an effect on performance.

The knowledge gained by Fisher as a result of the ENVIRO-SEAL and HIGH-SEAL packing development program has provided an opportunity to help you "engineer" the packing selection. Proper selection has a bottom line result; that is, increased service life and reduced maintenance.

If you need a more detailed engineering discussion of the design principles affecting packing seal design, contact your Fisher sales office or sales representative. The sales representative can provide you a copy of TM-38, Control Valve Packing Systems.

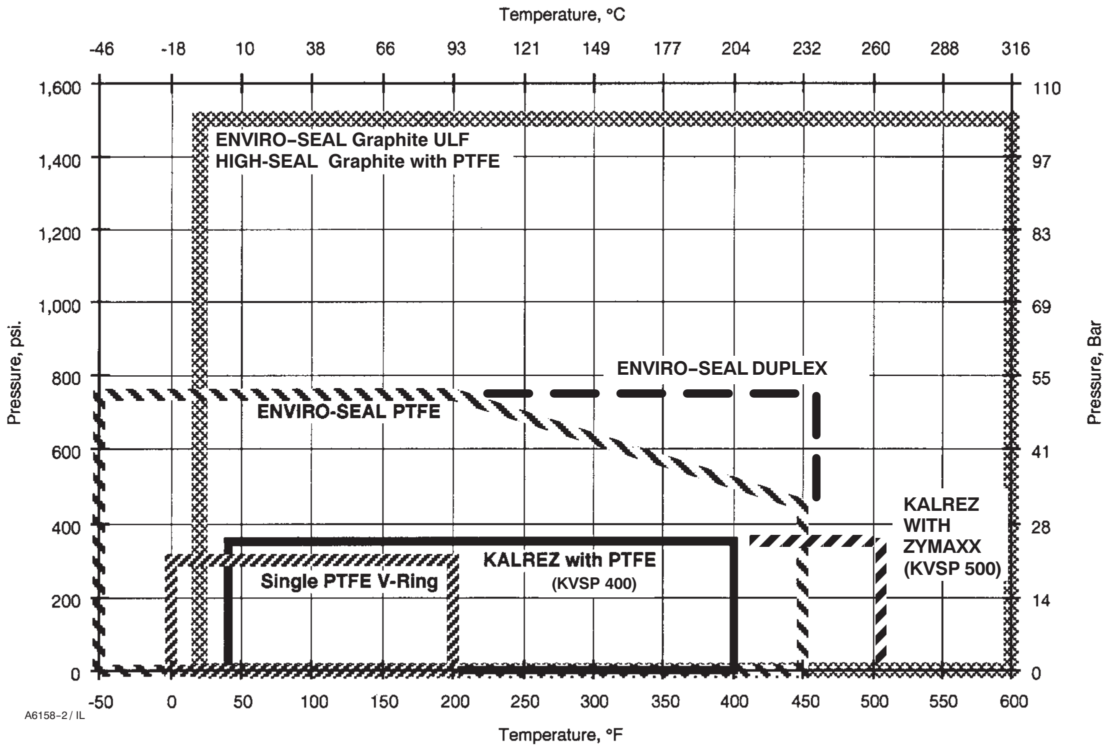


***Figure 11-1**. Application guidelines chart for 500 PPM service*

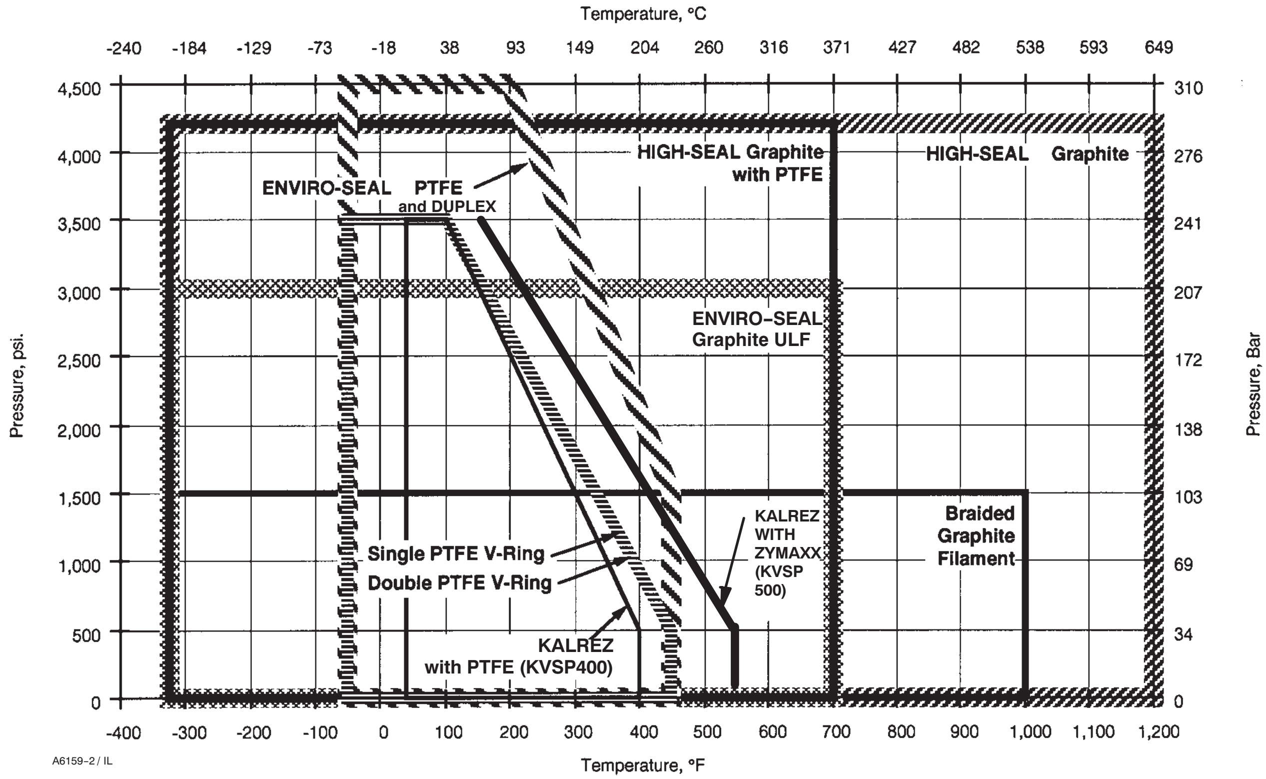


Figure 11-2. *Application guidelines chart for non-environmental service*

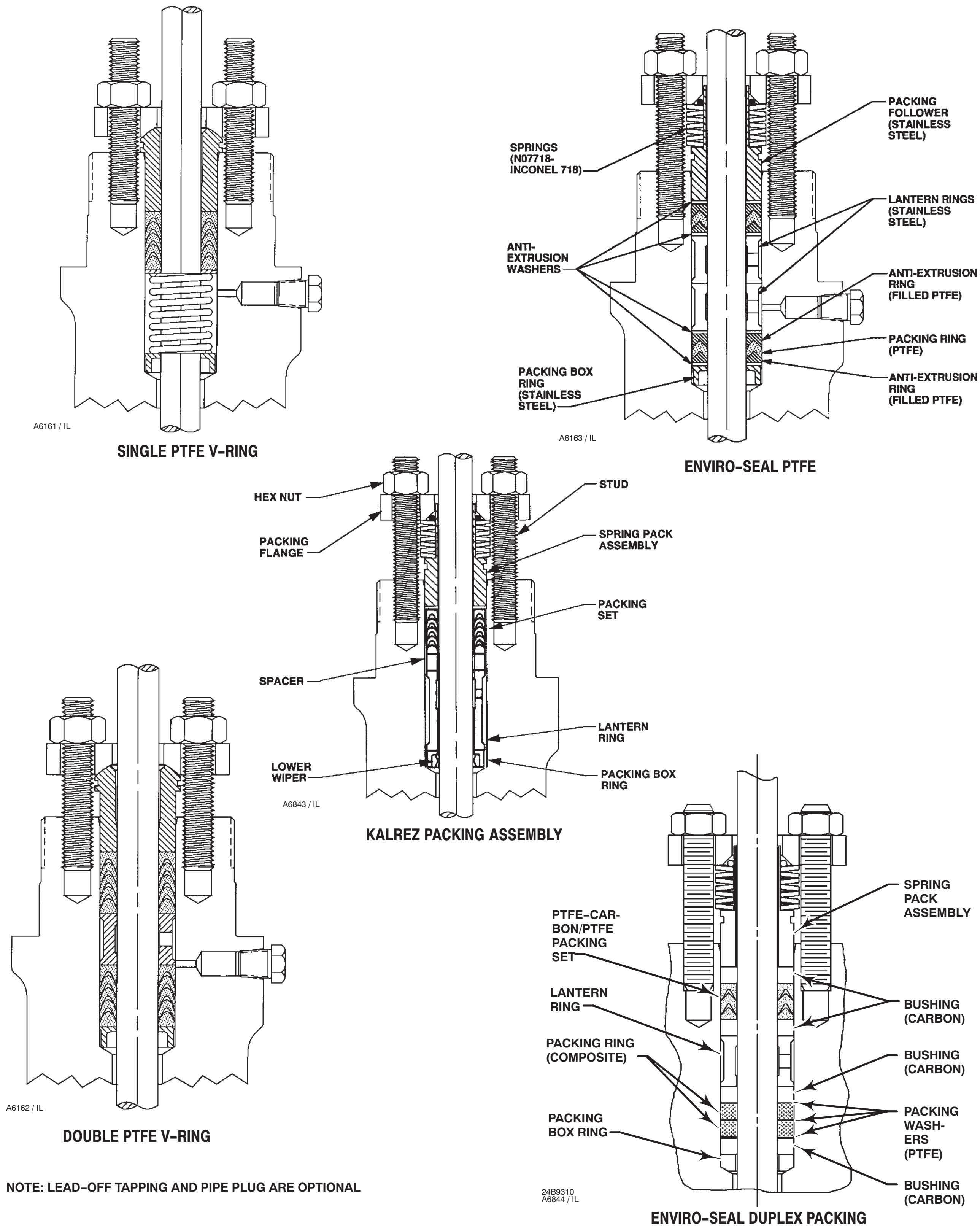
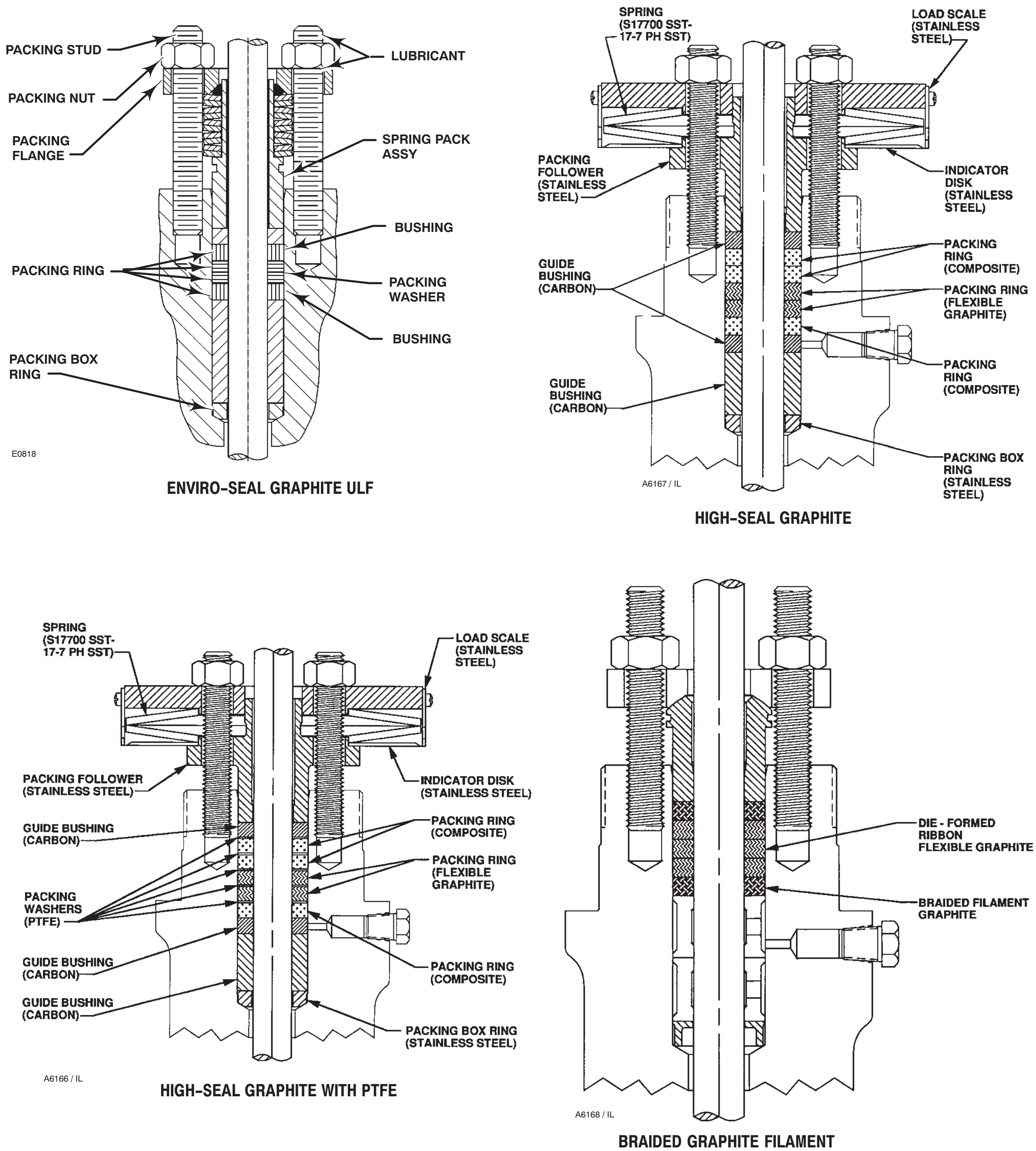


Figure 11-3. *Typical packing examples for sliding-stem valves*



NOTE: LEAD–OFF TAPPING AND PIPE PLUG ARE OPTIONAL

Figure 11-4. *Typical packing examples for sliding-stem valves*

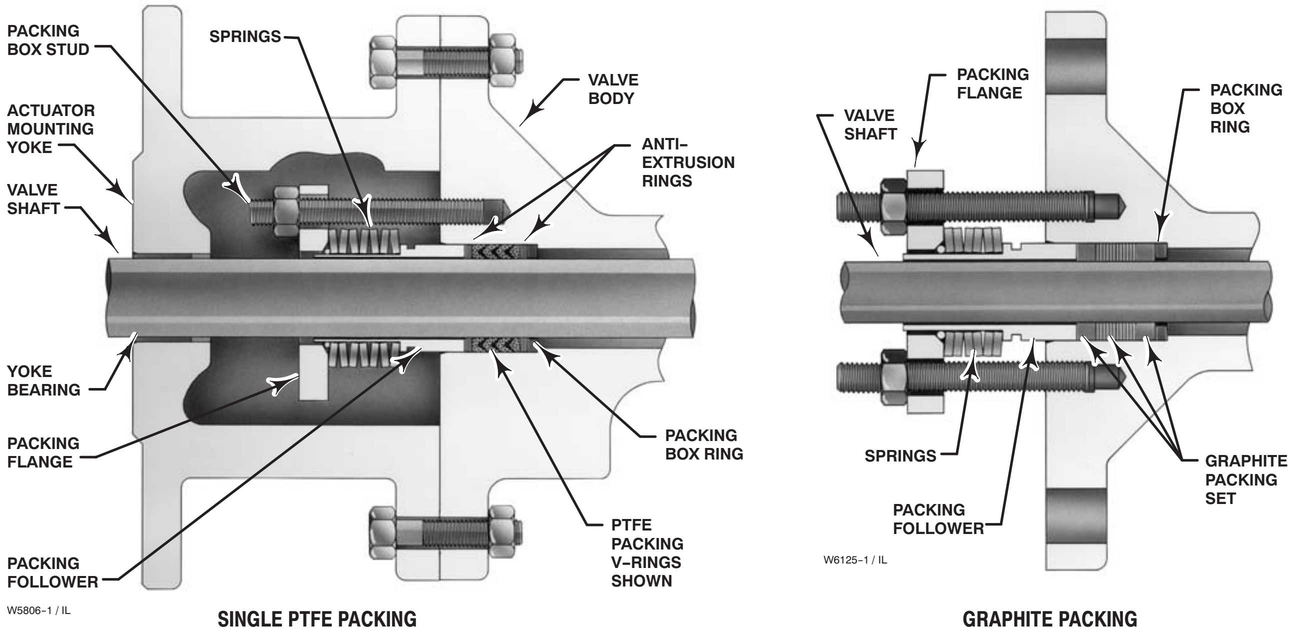


Figure 11-5. *Typical ENVIRO-SEAL® packing arrangements for rotary valves*

Chapter 12

Codes and Standard Overview

CODES AND STANDARDS OVERVIEW

The design, manufacture and use of control valves in power plants is governed by a variety of codes and standards. These documents provide for safe design and operation as well as consistency of product to facilitate plant construction and procurement.

This section summarizes Fisher's position regarding conformance to the most common codes and standards used to specify control valves for fossil power plant applications. The comments are divided into six groups: design standards, dimensional standards, performance testing, non-destructive examination, welding, and painting/cleaning. These comments apply to guide preparation of purchase specifications, as well as provide an awareness of situations where pricing adjustments are required. In many cases a minor change in specification can have significant commercial ramifications and only minor technical benefit.

Design standards. The boiler proper includes superheaters, economizers, reheaters, steam drums, water drums and other pressure parts connected directly to the boiler without intervening valves. The ASME Boiler and Pressure Vessel Code (BPVC) has administrative jurisdiction and technical responsibility for the boiler proper. Fisher does not manufacture equipment that is installed in the boiler proper.

Boiler external piping is that piping which begins where the boiler proper terminates. This termination is considered to be:

- The first circumferential joining for welding end connections;

or

- the face of the first flange in bolted flanged connections;

or

- the first threaded joint in a threaded connection.

Boiler external piping extends up to and including the valves required by the ASME BPVC. This may include water drum, superheater, reheater, and economizer header drain and vent valves, steam drum vent valves, and steam drum level indicators. The ASME BPVC has administrative jurisdiction, while the ASME Section Committee B 31.1 has technical responsibility. This means that design and construction rules are contained in ANSI/ASME B 31.1, but that ASME code certification, data forms, code symbol stamping and/or inspections by authorized inspectors are per ASME BPVC Section I when required.

The remainder of the power plant piping (non-boiler external piping) and is covered by ANSI/ASME B 31.1, Power Plant Piping.

ASME Boiler and Pressure Vessel Code – Section I. While control valves are not included in the boiler proper, the boiler external piping may include control equipment. (A common example is a steam drum level controller.) Design and construction of these devices must comply with the requirements of ANSI/ASME B 31.1 and also comply with the quality assurance requirements of ASME BPVC Section 1. This implies that vendors must provide inspection, data reports and stamping, which many are not authorized to provide. However, the ASME BPVC Section I waives these requirements for certain parts that already comply with an ANSI product standard or manufacturer's standard and which comply with certain other requirements for material, welding and radiography and heat treatment documentation. Under these conditions, Fisher may comply with ASME BPVC Section I without providing code stamping. The Comments to ANSI/ASME B 31.1 later contain more information.

ANSI/ASME B 31.1, Power Piping Code. Control valves and other equipment may be supplied per ANSI/ASME B 31.1 to meet requirements for either boiler external piping or non-boiler external piping. In most cases, this code will be applied to both valves and level controls.

1. Valves

This code references ANSI B 16.34 as an applicable design standard for valves. To comply with ANSI/ASME B 31.1, Fisher builds valves per ANSI B 16.34 and provides some additional marking requirements per ANSI B 16.34 and ANSI B 16.5. The code prohibits the use of ungasketed, screwed bonnets on source valves in steam service over 250 psig.

2. Level Controls

Standard cage-style level controls often require modifications before complying with ANSI/ASME B 31.1. All branch welds (such as the side connection saddle welds) must have a fillet weld added. Torque tube retainer flanges must conform to code dimensions.

On both valves and level controls, fabrication welds (including valve body to reducer welds) may require radiographic or liquid penetrant/magnetic particle examination. This requirement depends on nominal pipe size, wall thickness at the weld, design pressure and design temperature.

ANSI B 16.34. This standard covers pressure-temperature ratings, dimensions, materials, nondestructive examination requirements, testing and marking of cast, forged, and fabricated flanged and buttweld end, and wafer or flangeless valves.

Pressure-temperature ratings provided in the code are divided into four groups as follows:

1. Standard Class

These are the normal ANSI Classes 150 through 4500 pressure-temperature ratings. Most standard products fall in these standard classes. Pressure temperature ratings are published for a variety of materials.

2. Intermediate Standard Class

These ratings fall <u>between</u> standard class ratings and are achieved by designing the valve body and bonnet with extra wall thickness and by designing the body-to-bonnet bolting to handle higher loads. NDE is not required. Only BWE valves may carry intermediate ratings. The intermediate ratings many times allow use of less expensive products in high duty applications.

3. Special Class

These ratings are typically <u>higher</u> than standard class ratings and are obtained by ultrasonic or radiographic testing of the body and bonnet castings. Any BWE globe or angle valves may be given a special class rating. See ANSI B 16.34 for these ratings.

4. Intermediate Special Class

These ratings require both the nondestructive examination of the body and bonnet (as in special class) and the extra wall thickness/bolt strength (as in intermediate class). These ratings fall between the special class ratings, and may be applied only to BWE valves that have intermediate ratings. Intermediate special class ratings for applicable products are published in vendor literature.

Special Class, Intermediate Standard Class and Intermediate Special Class ratings all require pricing considerations.

Valves built to comply with B 16.34 must also meet marking requirements. To meet these requirements, Fisher applies one nameplate with valve body information and one with actuator information. The full nameplate requirements are met only when ANSI B 16.34 compliance is specified in writing by the customer.

ANSI B 16.5. This standard covers the design of flanges and flanged fittings and also establishes flanged fitting ratings. Although the current edition of this standard is not a valve design standard, earlier issues (before 1973) were applied to valve design. Design responsibility was transferred to ANSI B 16.34 in 1973 for buttweld end valves and in 1977 for flanged end valves.

ANSI B 16.5 may be applied to valves several ways:

1. As a dimensional/design standard for the flanged ends of valves. Literature will commonly say "Mates with ANSI XXX flanges."

2. To designate the pressure-temperature rating of the valve. The bulletin will commonly say, *"Pressures consistent with the applicable ANSI flange rating."*

3. As a valve design standard. This is not common now that ANSI B 16.34 covers valve design, but many older valves or older specifications may reference ANSI B 16.5 as the design specification.

MSS SP-66. This standard was published as a valve design standard prior to ANSI B 16.34. Conformance to ANSI B 16.34 should generally be specified in lieu of MSS SP-66. "Special inspections" per MSS SP-66 to increase the pressure-temperature ratings are now replaced by ANSI B 16.34 special class ratings.

MSS SP-67. This standard covers design and test performance requirements for butterfly valves and divides them into three leak classes. In most situations, these leak classes have been superseded for control valve usage by the ANSI/ISA B 16.104 Standards.

Type I: Tight shutoff valve. No leakage allowed.

Type II: Low leakage valve. Leakage within tolerances is allowed in the closed position. Type II valves are not subjected to a seat test unless required by the purchaser. When a test is required, the valve is to be subjected to a hydrostatic or air seat test at the rated shutoff pressure, and the leakage must not exceed the leak rate specified by the purchaser.

Type III: Nominal leakage valve. No seat leak test required.

MSS SP-67 also defines face-to-face dimensions for certain butterfly valves.

ASME Boiler and Pressure Vessel Code – Section VIII. This code covers requirements for pressure vessels. It is not used for valve design, although some design calculations for diffusers and actuators are based on Section VIII.

ANSI B 16.10. This standard defines face-to-face dimensions for gate, plug, check, ball and control valves. Control valves covered include Class 125 and 250 cast iron through 8-inch size, and Class 150, 300, 400 and 600 steel flanged valves through 8-inch size. Face-to-face dimensions for large valves and high-pressure valves will vary by manufacturer as necessary to suit the constraints of each design. Socket weld valves are covered by ANSI B 16.11.

ANSI B 16.37. This standard covers hydrostatic testing of control valves. Test pressures are 1.5 times the cold working pressure given in ANSI B 16.34. Fisher complies with ANSI B 16.37 on those products whose pressure shell is rated per ANSI Class B 16.34 (i.e., with ANSI 150, 300...etc. ratings). Testing is completed when specified in full compliance with methods prescribed. As standard, Fisher product is hydro tested by component using ANSI B 16.37 pressures and procedures. This component hydrotest is followed by an aerostatic test after assembly to confirm gasket joint integrity. This procedure allows us to ensure the integrity of valve parts and joints and contributes to more efficient manufacture.

ANSI/FCI 70-2 (formerly ANSI B 16.104). This standard defines seat leak classes and testing procedures. Fisher complies with this standard on those valves which are given ANSI leak rates (i.e., ANSI Class III, IV, V, etc.). The standard prescribes test procedures for each leak class as well as allowable leak rates.

MSS SP-61. This standard covers pressure testing of steel valves but is not intended for use with control valves. It includes shell tests and seat closure tests. The seat closure tests are specifically for shutoff or isolation valves (this includes on/off drain and block valves).

With regard to control valves the seat leak procedure is not adequate to recognize leak rates of different trim styles and sizes which may lead to over- or under-specification of the leak rate. For control valve seat leakage tests, refer to ISA SS75.19 and ANSI/FCI 70-2.

SNT-TC-1A. This standard defines qualification requirements for personnel who perform non-destructive examination. All personnel doing NDE should be qualified per SNT-TC-1A.

ANSI B 16.34. This standard allows increased pressure-temperature ratings for valves that are non-destructively examined (special class). Radiography or with customer acceptance, ultrasonic testing, is performed on certain areas of the body and bonnet. Consequently, this standard includes test procedures and acceptance criteria for diographic, ultrasonic, magnetic particle and liquid penetrant examinations. Use of B 16.34 is recommended in lieu of comparable MSS standards due to broader acceptance.

MSS SP-55. This standard covers visual examination of castings.

MSS SP-54. This standard covers radiographic examination of castings. Radiographic examination per ANSI B 16.34 or ASTM E94 should be proposed, however, due to broader acceptance of the standard.

MSS SP-53. This standard covers magnetic particle examination. Again, magnetic particle examination per ANSI B 16.34 should be proposed.

ASME Boiler and Pressure Vessel Code – Section V. Section V contains requirements and methods for nondestructive testing and describes procedures for various types of testing. This section is applicable only when it is specifically referenced and required by other ASME BPVC sections or other design specifications.

ASME Boiler and Pressure Vessel Code – Section IX. This standard defines requirements for qualification of welders and welding procedures. Welders and welding procedures should all comply with ASME Section IX. Noncompliance will violate other code and standard requirements.

SSPC-SP5, SP6, SSPC-SP10. These standards define requirements for blast cleaning metal surfaces. Most vendor procedures will comply with either SSPC-SP6 or SSPC-SP10. Requirements for special blast cleaning will often be coupled with special painting requirements.

Chapter 13

Piping and Installation Guidelines

This section provides guidelines for the installation of control valves and includes the topics of piping arrangement, valve capacity, line size versus valve size, velocity limitations, and control valve orientation. These guidelines answer many common questions about installing control valves and help to identify and avoid potential installation problems.

Control Valve Piping

Control valves are selected and specified after careful consideration of process conditions and performance requirements. Common sense suggests that to ensure the integrity of the valve selection process, a similar degree of engineering should be given to where and how control valves are installed within the piping system.

For example, an indication of how adjacent piping may affect the capacity of a control valve is given by the following actual test data.

When the control valve size is reduced from the line size by 2:1 ratio and the adjacent block valves are made the same size as the control valve, the capacity of a typical 2-inch control valve will be reduced by approximately 5%.

As the valve size is increased, while keeping a 2:1 ratio of line size to valve size, the reduction in capacity approaches 10%.

Capacity reduction becomes less if the block valves are made the same as line size and are moved outside the reducers.

If other fittings are incorporated, such as elbows between the block valves and reducers or elbows between the reducers and control valve, reduction in capacity may easily be as much as 15%.

General recommendations for control valve installation are:

- Avoid arrangements that produce a nozzle effect into the inlet of a control valve. This will alter the valve characteristic adversely.

- Avoid close-coupled inlet block valves reduced from line size, as it will affect characteristic and capacity.

- If a block and bypass arrangement must be used around a control valve, the control valve is best placed in a straight-through section. Avoid tortuous manifolds that are sometimes used to provide accessibility to a control valve.

- Avoid sharp turns close to the valve inlet. Straight-line sections into and out of the control valve should simulate the piping that was used originally to establish the flow capacity and characteristic test data. As a general comment, it is best if there are 10 to 20 pipe diameters of straight pipe run upstream of the valve and five to 10 pipe diameters downstream of the valve.

Line Size Versus Valve Size

The consideration of line size versus valve size is related primarily to the strength of the valve in relation to the strength of the adjacent piping. Common rules of thumb used in this regard are:

1. Valve size not less than 1/2 pipe size.

2. Valve size not less than two nominal sizes below the line size.

Either of these rules may be used with adequate assurance of safety. However, it should be noted that they do not yield the same result (i.e., assume a 20-inch line, which in Rule 1 would allow a 10-inch valve, but in Rule 2 would allow a 16-inch valve.)

Velocity Limitations in Control Valves

Questions regarding fluid velocity limitations may arise when valve and line sizes are being determined. Limitations vary between liquids and gases because the limits arise from different physical phenomena.

For vapors and gases, there is no limitation in velocity except when noise is a consideration. In this case, the velocity should be kept below the manufacturer's recommended velocity limit. For Fisher, these limits are 0.3 Mach for Whisper Trim III and 0.5 Mach for Whisper Trim I.

Limitations on liquid service have been a concern for many years. Fisher developed velocity guidelines for use with top-and-bottom guided valves. These guidelines were established to eliminate guide wear, vertical instability, body erosion and noise.

On the other hand, cage-style valves offer massive guiding at the point where pressure drop occurs. The cage shields the valve plug from direct fluid impingement and helps to prolong guide surface life. These features make it possible to operate cage-style valves at maximum rated inlet pressure and full pressure drop with no danger of premature guide failure.

Vertical instability, often called buffeting, was associated with many double-ported, top-and-bottom-guided valves. The concern for vertical instability has been eliminated through the application of cage-style valves with their single-seated, balanced trims. These valves have been designed not only to balance static forces, but also to minimize dynamic ones.

Body erosion is considered by many to be the primary reason for imposing fluid velocity limits. In erosive slurry applications this is certainly a major consideration; however, in "clean" fluid there is no correlation between velocity and body erosion.

Virtually all reported cases of body damage attributed to body erosion have been traceable to known trouble sources such as cavitation, flashing, wet steam erosion, or erosion from abrasive fines in the fluid. It is therefore believed that irrespective of valve style or material, velocity of clean fluids need not be limited simply to prevent body erosion.

The noise contribution of a liquid passing through a control valve is usually considered inconsequential, provided that no adverse condition such as cavitation or severe vibration exists.

In general, control valves should be sized strictly on the basis of capacity requirements. Instances where velocity considerations dictate valve size are few and far between, particularly with cage-style valves and their superior guiding. Those designs that are limited have published pressure drop limitations.

Actuator and Body Orientation

The primary considerations of actuator and body orientation are:

- Trim wear
- Packing reliability
- Maintenance

The ideal situation with regard to all of these considerations is that the actuator be oriented in the vertical plane and on top.

Although Fisher sliding stem control valve assemblies are not position sensitive, the recommended installation is with the actuator vertical and on top. In this orientation, gravity does not produce increased wear effects in the same manner as it would with the valve assembly at some angle.

Mounting the valve with the actuator underneath can allow all rust, weld slag, corrosion, etc. in the flow line to fall into the packing box area of the valve. This may shorten packing and valve stem life since any of these entrained materials can be pulled into the packing by stem motion combined with gravity. With the actuator under the valve, valve disassembly and reassembly will be very difficult as well.

Bracing the actuator when it is installed at an angle will not offset the effects of gravity. One must remember that bracing the actuator will only provide support to the extended structure. It will NOT impact the effects of gravity on control valve internal parts such as valve plug, actuator diaphragm head, etc.

These internal components will move due to gravity, within their allowed tolerance, resulting in increased wear in a localized area. Wear is directly related to the weight of the parts, and it is safe to say that the larger the actuator/valve assembly, the more likely there will be an increased wear rate.

Fisher ENVIRO-SEAL® packing is the best selection for any sliding stem control valve from a long-term performance standpoint. With any sliding stem control valve that will be installed in a position other than vertical, with actuator on top, it is highly recommended that ENVIRO-SEAL packing be used. This is because the ENVIRO-SEAL packing arrangements contain bushing materials that support side load so that the soft sealing portions of the packing system are not required to provide side load support.

If the actuator must be mounted horizontally, it is recommended that hardened guides and seats be used in the trim to reduce potential increased wear.

Typically, increased wear will occur between:

- Plug and cage (on a cage-guided valve)
- Post and bushing (on a post-guided valve)
- Plug seat and seat ring (on all globe valves)
- Valve stem packing

This increased wear eventually will cause deformation of the trim parts and is likely to increase seat leakage, especially on valves used on high pressure drop water or steam service.

Orientation of the actuator in a position other than vertical and on top will also complicate assembly and disassembly of an installed valve.

Installation

Two main areas of concern appear during valve installation:

- Welding procedures
- System flushing

Most valves used in power applications have buttweld or socket weld ends. Welding procedures can create unusually high temperatures at the valve, and these temperatures may damage trim parts. After welding, the slag and debris are removed from the pipe and valve by flushing. The system flush may also damage trim parts.

Welding Procedures

The welding process can be divided into three stages:

1. *Preheating*
2. *Welding*
3. *Post-weld heat treating*

Piping codes specify that preheat be used, which involves heating the two pieces to be welded prior to beginning the weld. The heating may be performed with heating pads or with a torch in the area of the butt weld or socket weld ends.

Depending on the material and joint thickness, the temperature may range upward to 300-500° F. This temperature is not damaging for most metal parts, but elastomeric and plastic parts may have lower temperature limits. These parts may include o-rings, seal rings, backup rings, packing rings and gaskets.

There is usually little danger in the packing box area as the bonnet and body will dissipate heat. Inside the body, however, temperatures may be closer to the preheat temperature.

Caution should be used whenever the preheat temperature is close to or higher than the temperature limits for elastomeric or plastic parts in a valve. When in doubt, remove these components.

In the welding process (such as around a socket weld end), as the welder begins, the temperature of the first puddle of molten metal will be very high. As the process continues, the first puddle will continue to cool until the welder reaches the starting point. The temperature of that first puddle, just as the first pass is completed, is called the interpass temperature. The interpass temperature must be maintained below a maximum, which is dependent upon the material being welded. As an example, chrome-moly steels and stainless steel interpass temperatures are usually in the range of 300-500° F, so the same cautions apply as with preheating.

Postweld heat-treating generates some very special problems. When chrome-moly steel is welded, stresses are created which can lead to cracking of the welded area. To avoid these stresses, the welded area is heated to a very high temperature and held there for a period of time. Temperatures for post weld heat treat may range up to about 1400° F, with hold times lasting several hours. Although the heat is usually applied to only the welded area, it conducts back into the valve body. Potentially, this may damage elastomeric and plastic parts, but metal parts also may be damaged or warp. Shrunk-fit pieces and threaded connections may loosen. In general, if PWHT is to be performed, all trim parts should be removed.

System Flushing

Flushing and chemical cleaning also generate some special problems. Flushing involves the flowing of water or steam through a system to remove solid particles (such as weld slag) prior to operation of the system.

Valves are essentially restrictions in the flowstream, especially when equipped with any drilled hole or small orifice trim, i.e. Whisper Trim, Cavitrol® Trim, Micro-Flat or standard drilled hole trim, etc. Solid particles may lodge in the valve trim, which can lead to plugging. There is also the possibility of scoring between the plug and cage and nicking of the seating surface.

To protect valve trim, Fisher recommends use of a special set of flushing trim. This special flushing trim normally consists of the following sacrificial parts: cage or cage flange, seat ring, double-nutted stem to seal the bonnet stem hole, and gaskets. This sacrificial trim allows the flushing fluid and any solid particles to pass freely through the valve body and protects the seating and gasket surfaces from damage.

Chemical cleaning may cause similar problems. Some cleaning solutions can etch seating or gasket surfaces, especially if the solutions are not neutralized after the cleaning is complete and if the solutions are allowed to sit in the valve body. In these cases, flushing trim should be used to protect the seating and gasket surfaces and the real trim. Since many different acids or cleaning solutions can be used for chemical cleaning, Fisher recommends that the real trim not be exposed to the chemical cleaning solution.

Once it has been determined that a valve must be disassembled because of welding requirements or because flushing trim must be installed, make sure that an extra set of packing and gaskets is available. Gaskets should always be replaced when the valve has been disassembled. It is recommended that the packing also be replaced due to potential damage from removing the valve stem from the packing box. Failure to replace these parts may lead to leaks and can result in erosion of the gasket surfaces, the packing box bore, or the valve stem.

Chapter 14

Control Valve Performance & Diagnostics

In today's dynamic business environment, manufacturers are under extreme economic pressures. Market globalization is resulting in intense pressures to reduce manufacturing costs to compete with lower wages and raw material costs of emerging countries. Competition exists between international companies to provide the highest quality products and to maximize plant throughputs with fewer resources, all the while meeting ever-changing customer needs and being in full compliance with public and regulatory policies.

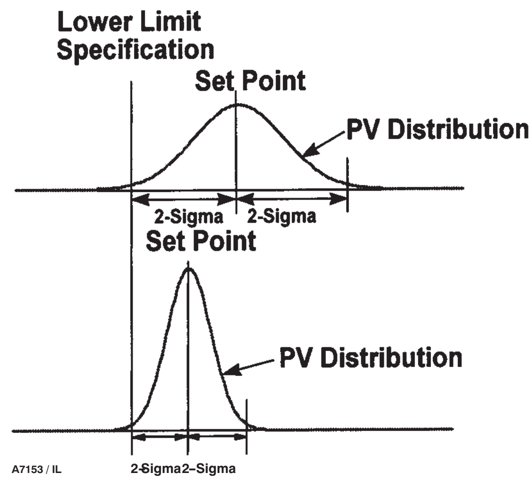


Figure 14-1. *Process variability*

Process Variability

To deliver acceptable returns to their shareholders, international industry leaders are realizing they must reduce raw material and scrap costs while increasing productivity. Reducing process variability in the manufacturing processes through the application of process control technology is recognized as an effective method to improve financial returns and meet global competitive pressures.

The basic objective of a company is to make a profit through the production of a quality product. A quality product conforms to a set of specifications. Any deviation from the established specification means lost profit due to excessive material use, reprocessing costs, or wasted product. Thus, a large financial impact is obtained through improving process control. Reducing process variability through better process control allows optimization of the process and the production of products correctly, the first time.

The non-uniformity inherent in raw materials and production processes is the common cause of process variability both above and below the set point. A process that is in control, with only the common causes of variation present, typically follows a bell-shaped normal distribution (Figure 14-1).

A statistically derived band of values on this distribution, called the +/-2 sigma band, describes the spread of process variable deviations from the set point. This band is the variability of the process. Process Variability is a precise measure of tightness of control and is expressed as a percentage of the set point.

If a product must meet a certain lower-limit specification, for example, the set point needs to be established at a 2 sigma value above this lower limit. Doing so will ensure that all the product produced at values to the right of the lower limit will meet the quality specification.

The problem, however, is that money and resources are being wasted by making a large percentage of the product to a level much greater

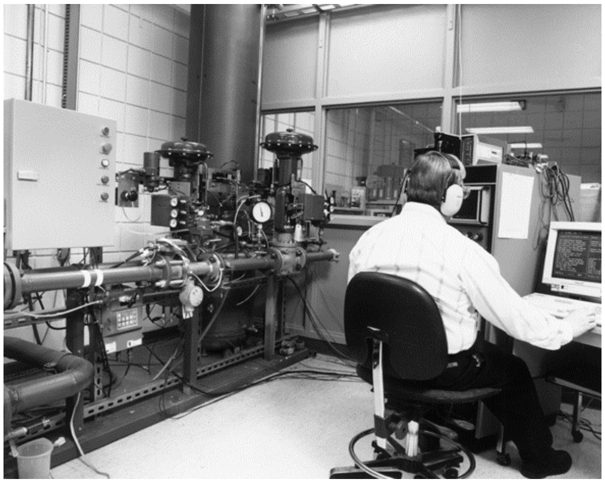


Figure 14-2. *Performance test loop*

than required by the specification (see upper distribution in Figure 14-1).

The most desirable solution is to reduce the spread of the deviation about the set point by going to a control valve that can produce a smaller sigma (see lower distribution in Figure 14-1).

Reducing process variability is a key to achieving business goals. Most companies realize this, and it is not uncommon for them to spend hundreds of thousands of dollars on instrumentation in efforts to reduce process variability.

Unfortunately, the control valve is often overlooked because its impact on dynamic performance is not realized. Extensive studies of control loops indicate as many as 80% of the loops did not do an adequate job of reducing process variability. Furthermore, the control valve was found to be a major contributor to this problem for a variety of reasons.

To verify performance, control valve manufacturers must test their products under dynamic process conditions. These are typically performed in a flow lab in actual closed-loop control (Figure 14-2). Evaluating control valve assemblies under closed-loop conditions provides the only true measure of variability performance. Closed-loop performance data proves significant reductions in process variability can be achieved by choosing the right control valve for the application.

The ability of control valves to reduce process variability depends upon many factors. More than one isolated parameter must be considered. Research within the industry has found the particular design features of the final control element, including the valve, actuator, and positioner, are very important in achieving good process control under dynamic conditions. Most importantly, the control valve assembly must be optimized or developed as a unit. Valve components not designed as complete assemblies typically do not yield the best dynamic performance. Some of the most important design considerations include:

- Dead band
- Actuator/positioner design
- Valve response time
- Valve type and sizing

Each of these design features will be considered in this chapter to provide insight into what constitutes a superior valve design.

Dead Band

Dead band is a major contributor to excess process variability, and control valve assemblies can be a primary source of dead band in an instrumentation loop due to a variety of causes such as friction, backlash, shaft wind-up, relay or spool valve dead zone, etc.

Dead band is a general phenomenon where a range or band of controller output values fails to produce a change in the measured process variable when the input signal reverses direction. When a load disturbance occurs, the process variable deviates from the set point. This deviation initiates a corrective action through the controller and back through the process. However, an initial change in controller output cannot produce a corresponding corrective change in the process variable. Only when the controller output has changed enough to progress through the dead band does a corresponding change in the process variable occur.

Dead band has many causes, but friction and backlash in the control valve, along with shaft wind-up in rotary valves, and relay dead zone are some of the more common forms. Because most control actions for regulatory control consist of small changes (1% or less), a control valve with excessive dead band might not even respond to many of these small changes. A well-engineered valve should respond to signals of 1% or less to provide effective reduction in process variability. However, it is not uncommon for some valves to exhibit dead band as great as 5% or more. In a recent plant audit, 30% of the valves had dead bands in excess of 4%. Over 65% of the loops audited had dead bands greater than 2%.

Figure 14-3 shows just how dramatic the combined effects of dead band can be. This diagram represents an open-loop test of three different control valves under normal process conditions. The valves are subjected to a series of step inputs which range from 0.5% to 10%. Step tests under flowing conditions such as these are essential because they allow the performance of the entire valve assembly to be evaluated, rather than just the valve actuator assembly as would be the case under most bench test conditions.

Some performance tests on a valve assembly compare only the actuator stem travel versus the input signal. This is misleading because it ignores the performance of the valve itself.

It is critical to measure dynamic performance of a valve under flowing conditions so the change in process variable can be compared to the change in valve assembly input signal. It matters little if only the valve stem changes in response to a change in valve input because if there is no corresponding change in the controlled variable, there will be no correction to the process variable.

In all three valve tests (Figure 14-3), the actuator stem motion changes fairly faithfully in response to the input signal changes. On the other hand, there is a dramatic difference in each of these valve's ability to change the flow in response to an input signal change.

For Valve A, the process variable (flow rate) responds well to input signals as low as 0.5%. Valve B requires input signal changes as great as 5% before it begins responding faithfully to each of the input signal steps. Valve C is considerably worse, requiring signal changes as great as 10% before it begins to respond faithfully to each of the input signal steps. The ability of either Valve B or C to improve process variability is very poor.

Friction is a major cause of dead band in control valves. Rotary valves are often very susceptible to friction caused by the high seat loads required to obtain shut-off with some seal designs. Because of the high seal friction and poor drive train stiffness, the valve shaft winds up and does not translate motion to the control element. As a result, an improperly designed rotary valve can

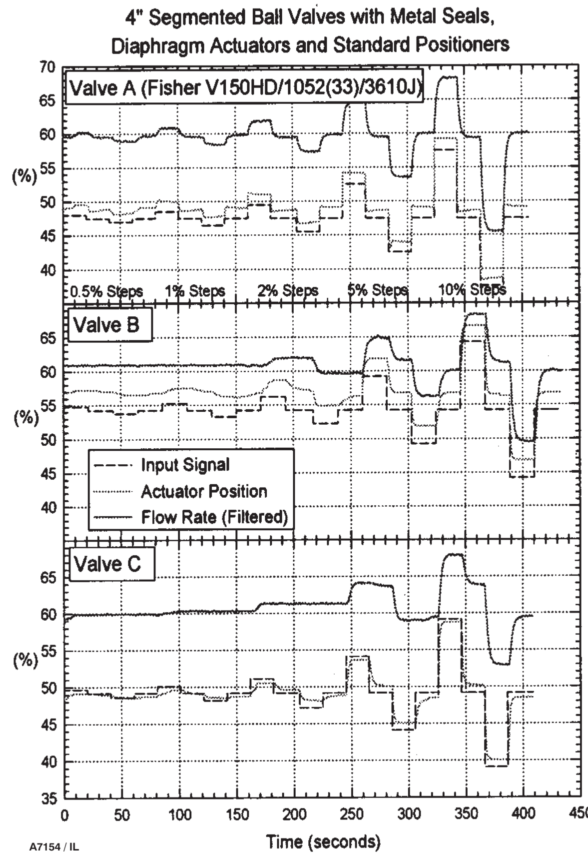


Figure 14-3. *Open-loop step test of three valve constructions reveals dramatic differences in ability to change the flow rate.*

exhibit significant dead band that clearly has a detrimental effect on process variability.

Manufacturers usually lubricate rotary valve seals during manufacture, but after only a few hundred cycles this lubrication wears off. In addition, pressure-induced loads also cause seal wear. As a result, the valve friction can increase by 400% or more for some valve designs. This illustrates the misleading performance conclusions that can result from evaluating products using bench type data before the torque has stabilized. Valves B and C (Figure 14-3) show the devastating effect these higher friction torque factors can have on a valve's performance.

Packing friction is the primary source of friction in sliding stem valves. In these types of valves, the measured friction can vary significantly between valve styles and packing arrangements.

Actuator style also has a profound impact on control valve assembly friction. Generally, spring-and-diaphragm actuators contribute less friction to the control valve assembly than piston actuators. An additional advantage of spring-and-diaphragm actuators is that their frictional characteristics are more uniform with age. Piston actuator friction probably will increase significantly as guide surfaces and the O-rings wear with use, lubrication fails, and the elastomer degrades. Thus, to ensure continued good performance, maintenance is required more often for piston actuators than for spring-and-diaphragm actuators. If that maintenance is not performed, process variability can suffer dramatically without the operator's knowledge.

Backlash is the name given to slack or looseness of a mechanical connection. This slack results in a discontinuity of motion when the device changes direction. Backlash commonly occurs in gear drives of various configurations. Rack-and-pinion actuators are particularly prone to dead band due to backlash. Some valve shaft connections also exhibit dead band effects. Spline connections generally have much less dead band than keyed shafts or double-D designs.

While friction can be reduced significantly through good valve design, it is a difficult phenomenon to eliminate entirely. A well-engineered control valve should be able to virtually eliminate dead band due to backlash and shaft wind-up.

For best performance in reducing process variability, the total dead band for the entire valve assembly should be 1% or less. Ideally, it should be as low as 0.25%.

Actuator-Positioner Design

Actuator and positioner design must be considered together. The combination of these two pieces of equipment greatly affects the static performance (dead band), as well as the dynamic response of the control valve assembly and the overall air consumption of the valve instrumentation.

Positioners are used with the majority of control valve applications specified today. Positioners allow for precise positioning accuracy and faster response to process upsets when used with a conventional digital control system. With the increasing emphasis upon economic performance of process control, positioners should be considered for every valve application where process optimization is important.

The most important characteristic of a good positioner for process variability reduction is that it be a high gain device. Positioner gain is composed of two parts: the static gain and the dynamic gain.

Static gain is related to the sensitivity of the device to the detection of small (0.125% or less) changes of the input signal. Unless the device is sensitive to these small signal changes, it cannot respond to minor upsets in the process variable. This high static gain of the positioner is obtained through a preamplifier, similar in function to the preamplifier contained in high fidelity sound systems. In many pneumatic positioners, a nozzle-flapper or similar device serves as this high static gain preamplifier.

Once a change in the process variable has been detected, the positioner then must rapidly supply a large volume of air to the actuator to make the valve respond promptly. The ability to do this comes from the high dynamic gain of the positioner. Although the positioner preamplifier can have high static gain, it typically has little ability to supply the power needed. Thus, the preamplifier function must be supplemented by a high dynamic gain power amplifier that supplies the required air flow as rapidly as needed. A relay or a spool valve typically provides this power amplifier function.

Spool valve positioners are relatively popular because of their simplicity. Unfortunately, many spool valve positioners achieve this simplicity by omitting the high gain preamplifier from the design. The input stage of these positioners is often a low static gain transducer module that changes the input signal (electric or pneumatic) into movement of the spool valve. This type of device generally has low sensitivity to small signal changes. The result is increased dead time and longer overall response time of the control valve assembly.

Some manufacturers attempt to compensate for the lower performance of these devices by using spool valves with enlarged ports and reduced overlap of the ports. This increases the dynamic power gain of the device, which helps performance to some extent if it is well matched to the actuator. However, it also dramatically increases the air consumption of these high gain spool valves. Many high gain spool valve positioners have static instrument air consumption five times greater than typical high performance two-stage positioners.

Typical two-stage positioners use pneumatic relays at the power amplifier stage. Relays are preferred because they can provide high power gain that gives excellent dynamic performance with minimal steady-state air consumption. In addition, they are less subject to fluid contamination.

Positioner designs are changing dramatically, with microprocessor devices becoming increasingly popular. These microprocessor-based positioners provide dynamic performance equal to the best conventional two-stage pneumatic positioners. They also provide valve monitoring and diagnostic capabilities to help ensure that initial good performance does not degrade with use.

In summary, high-performance positioners with both high static and dynamic gain provide the best overall process variability performance for any given valve assembly.

Valve Response Time

For optimum control of many processes, it is important that the valve reach a specific position quickly. A quick response to small signal changes (1% or less) is one of the most important factors in providing optimum process control. In automatic, regulatory control, the bulk of the signal changes received from the controller are for small changes in position. If a control valve assembly can quickly respond to these small changes, process variability will be improved.

Valve response time is measured by a parameter called T_{63} (Tee-63). T_{63} is the time measured from initiation of the input signal change to when the output reaches 63% of the corresponding change. It includes both the valve assembly dead time, which is a static time, and the dynamic time of the valve assembly. The dynamic time is a measure of how long the actuator takes to get to the 63% point once it starts moving.

Dead band, whether it comes from friction in the valve body and actuator or from the positioner, can significantly affect the dead time of the valve assembly. It is important to keep the dead time as small as possible. Generally, dead time should be no more than one-third of the overall valve response time. However, the relative relationship between the dead time and the process time constant is critical. If the valve assembly is in a fast loop where the process time constant approaches the dead time, the dead time can dramatically affect loop performance. On these fast loops, it is critical to select control equipment with dead time as small as possible.

Also, from a loop tuning point of view, it is important that the dead time be relatively consistent in both stroking directions of the valve. Some valve assembly designs can have dead times that are three to five times longer in one stroking direction than the other. This type of behavior is typically induced by the asymmetric behavior of the positioner design, and it can severely limit the ability to tune the loop for best overall performance.

Once the dead time has passed and the valve begins to respond, the remainder of the valve response time comes from the dynamic time of the valve assembly. This dynamic time will be determined primarily by the dynamic characteristics of the positioner and actuator combination. These two components must be carefully matched to minimize the total valve response time.

In a pneumatic valve assembly, for example, the positioner must have a high dynamic gain to minimize the dynamic time of the valve assembly. This dynamic gain comes mainly from the power amplifier stage in the positioner. In other words, the faster the positioner relay or spool valve can supply a large volume of air to the actuator, the faster the valve response time will be.

However, this high dynamic gain power amplifier will have little effect on the dead time unless it has some intentional dead band designed into it to reduce static air consumption. Of course, the design of the actuator significantly affects the dynamic time. For example, the greater the volume of the actuator air chamber to be filled, the slower the valve response time.

At first, it might appear that the solution would be to minimize the actuator volume and maximize the positioner dynamic power gain. It is really not that easy. This can be a dangerous combination of factors from a stability point of view. Recognizing that the positioner/actuator combination is its own feedback loop, it is possible to make the positioner/actuator loop gain too high for the actuator design being used, causing the valve assembly to go into an unstable oscillation. In addition, reducing the actuator volume has an adverse affect on the thrust-to-friction ratio, which increases the valve assembly dead band resulting in increased dead time.

If the overall thrust-to-friction ratio is not adequate for a given application, one option is to increase the thrust capability of the actuator by using the next size actuator or by increasing the pressure to the actuator. This higher thrust-to-friction ratio reduces dead band, which should help to reduce the dead time of the assembly. However, both of these alternatives mean that a greater volume of air needs to be supplied to the actuator. The tradeoff is a possible detrimental effect on the valve response time through increased dynamic time.

One way to reduce the actuator air chamber volume is to use a piston actuator rather than a spring-and-diaphragm actuator, but this is not a panacea. Piston actuators usually have higher thrust capability than spring-and-diaphragm actuators, but they also have higher friction, which can contribute to problems with valve response time.

It is usually necessary to use a higher air pressure with a piston actuator than with a diaphragm actuator, because the piston typically has a smaller area. This means that a larger volume of air needs to be supplied with its attendant ill effects on dynamic time. In addition, piston actuators, with their greater number of guide surfaces, tend to have higher friction due to inherent difficulties in alignment, as well as friction from the O-ring.

These friction problems also tend to increase over time. Regardless of how good the O-rings are initially, these elastomeric materials will degrade with time due to wear and other environmental conditions. Likewise, wear on the guide surfaces will increase the friction, and depletion of the lubrication will occur. These friction problems result in a greater piston actuator dead band, which will increase the valve response time through increased dead time.

Instrument supply pressure can also have a significant impact on dynamic performance of the valve assembly. For example, it can dramatically affect the positioner gain, as well as overall air consumption.

Fixed-gain positioners generally have been optimized for a particular supply pressure. This gain, however, can vary by a factor of two or more over a small range of supply pressures. For example, a positioner that has been optimized for a supply pressure of 20 psig might find its gain cut in half when the supply pressure is boosted to 35 psig.

Supply pressure also affects the volume of air delivered to the actuator, which in turn determines stroking speed. It is also directly linked to air consumption. Again, high-gain spool valve positioners can consume up to five times the amount of air required by more efficient high-performance, two-stage positioners that use relays for the power amplification stage.

To minimize the valve assembly dead time, minimize the dead band of the valve assembly, whether it comes from friction in the valve seal design, packing friction, shaft wind-up, actuator, or positioner design. As indicated, friction is a major cause of dead band in control valves.

On rotary valve styles, shaft wind-up can also contribute significantly to dead band. Actuator style also has a profound impact on control valve assembly friction. Spring-and-diaphragm actuators generally contribute less friction to the control valve assembly than piston actuators over an extended time. As mentioned, this is caused by the increasing friction from the piston O-ring, misalignment problems, and failed lubrication.

Having a positioner design with a high static gain preamplifier can make a significant difference in reducing dead band. This can also make a significant improvement in the valve assembly resolution. Valve assemblies with dead band and resolution of 1% or less are no longer adequate for many process variability reduction needs. Many processes require the valve assembly to have dead band and resolution as low as 0.25%, especially where the valve assembly is installed in a fast process loop.

One of the surprising things to come out of many industry studies on valve response time has been the change in thinking about spring-and-diaphragm versus piston actuators. It has long been a misconception that piston actuators are faster than spring-and-diaphragm actuators. Research has shown this to be untrue for small signal changes.

This mistaken belief arose from many years of experience with testing valves for stroking time. A stroking time test is normally conducted by subjecting the valve assembly to a 100% step change in the input signal and measuring the time it takes the valve assembly to complete its full stroke in either direction.

Although piston-actuated valves usually have faster stroking times than most spring-and-diaphragm actuated valves, this test does not indicate valve performance in an actual process control situation. In normal process control applications, the valve is rarely required to stroke through its full operating range. Typically, the valve is only required to respond within a range of 0.25% to 2% change in valve position.

Table 14-1. *Valve Response Time*

		$T_{d\ (v0.2)}$			$T_{63\ (v0.6)}$		
Valve Action	**Step Size %**	**Valve A**	**Valve B**	**Valve C**	**Valve A**	**Valve B**	**Valve C**
Opening	2	0.25	5.61	4.40	0.34	7.74	5.49
Closing	-2	0.50	0.46	NR	0.74	1.67	NR
Opening	5	0.16	1.14	5.58	0.26	2.31	7.06
Closing	-5	0.22	1.04	2.16	0.42	2.00	3.90
Opening	10	0.19	0.42	0.69	0.33	1.14	1.63
Closing	-10	0.23	0.41	0.53	0.46	1.14	1.25
NR = No Response							

Extensive testing of valves has shown that spring-and-diaphragm valve assemblies consistently outperform piston-actuated valves on small signal changes, which are more representative of regulatory process control applications. Higher friction in the piston actuator is one factor that plays a role in making them less responsive to small signals than spring-and-diaphragm actuators.

Selecting the proper valve, actuator, positioner combination is not easy. It is not simply a matter of finding a combination that is physically compatible. Good engineering judgment must go into the practice of valve assembly sizing and selection to achieve the best dynamic performance from the loop.

Table 14-1 shows the dramatic differences in dead time and overall T_{63} response time caused by differences in valve assembly design.

Valve Type And Characterization

The style of valve used and the sizing of the valve can have a large impact on the performance of the control valve assembly in the system. While a valve must be of sufficient size to pass the required flow under all possible contingencies, a valve that is too large for the application is a detriment to process optimization.

Flow capacity of the valve inherent characteristic of the valve. The inherent characteristic is the relationship between the valve flow capacity and the valve travel when the differential pressure drop across the valve is held constant.

Typically, these characteristics are plotted on a curve where the horizontal axis is labeled in percent travel although the vertical axis is labeled as percent flow (or C_V). Since valve flow is a function of both the valve travel and the pressure drop across the valve, it is traditional to conduct inherent valve characteristic tests at a constant pressure drop. This is not a normal situation in practice, but it provides a systematic way of comparing one valve characteristic design to another.

Under the specific conditions of constant pressure drop, the valve flow becomes only a function of the valve travel and the inherent design of the valve trim. These characteristics are called the inherent flow characteristic of the valve. Typical valve characteristics are named linear, equal percentage, and quick opening.

The ratio of the incremental change in valve flow (output) to the corresponding increment of valve travel (input) which caused the flow change is defined as the valve gain; that is,

Inherent Valve Gain = (change in flow)/(change in travel) = slope of the inherent characteristic curve

The linear characteristic has a constant inherent valve gain throughout its range, and the quick-opening characteristic has an inherent valve gain that is the greatest at the lower end of the travel range. The greatest inherent valve gain for the equal percentage valve is at the largest valve opening.

Inherent valve characteristic is a function of the valve flow passage geometry and does not change as long as the pressure drop is held constant. Many valve designs, particularly rotary ball valves, butterfly valves, and eccentric plug valves, have inherent characteristics, which cannot be easily changed; however, most globe valves have a selection of valve cages or plugs that can be interchanged to modify the inherent flow characteristic.

Knowledge of the inherent valve characteristic is useful, but the more important characteristic for purposes of process optimization is the installed flow characteristic of the entire process, including the valve and all other equipment in the loop. The installed flow characteristic is defined as the relationship between the flow through the valve

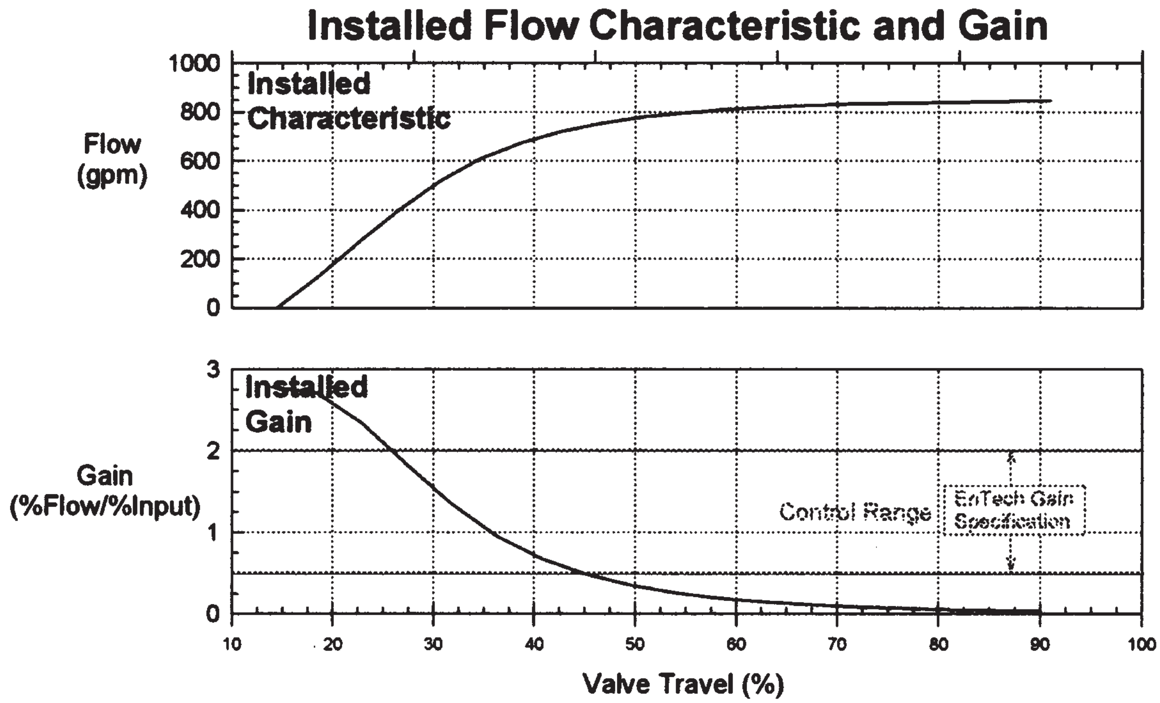


Figure 14-4. *Installed flow characteristic and gain*

and the valve assembly input when the valve is installed in a specific system, and the pressure drop across the valve is allowed to change naturally, rather than being held constant.

An illustration of such an installed flow characteristic is shown in the upper curve of Figure 14-4. The flow in this figure is related to the more familiar valve travel rather than valve assembly input.

Installed gain, shown in the lower curve of Figure 14-4, is a plot of the slope of the upper curve at each point. Installed flow characteristic curves such as this can be obtained under laboratory conditions by placing the entire loop in operation at some nominal set point and with no load disturbances. The loop is placed in manual operation, and the flow is then measured and recorded as the input to the control valve assembly is manually driven through its full travel range. A plot of the results is the installed flow characteristic curve shown in the upper part of the figure. The slope of this flow curve is then evaluated at each point on the curve and plotted as the installed gain as shown in the lower part.

Field measurements of the installed process gain can also be made at a single operating point using open-loop step tests (Figure 14-3). The installed process gain at any operating condition is simply the ratio of the percent change in output (flow) to the percent change in valve assembly input signal.

The reason for characterizing inherent valve gain through various valve trim designs is to provide compensation for other gain changes in the control loop. The end goal is to maintain a loop gain, which is reasonably uniform over the entire operating range, to maintain a relatively linear installed flow characteristic for the process. Because of the way it is measured, as defined above, the installed flow characteristic and installed gain represented in Figure 14-4 are really the installed gain and flow characteristic for the entire process.

Typically, the gain of the unit being controlled changes with flow. For example, the gain of a pressure vessel tends to decrease with throughput. In this case, the process control engineer would then likely want to use an equal percentage valve that has an increasing gain with flow. Ideally, these two inverse relationships should balance out to provide a more linear installed flow characteristic for the entire process.

Theoretically, a loop has been tuned for optimum performance at some set point flow condition. As the flow varies about that set point, it is desirable to keep the loop gain as constant as possible to maintain optimum performance. If the loop gain change due to the inherent valve characteristic does not exactly compensate for the changing gain of the unit being controlled, then there will be a variation in the loop gain due to variation in the installed process gain. As a result, process optimization becomes more difficult. There is also a danger that the loop gain might change enough to cause instability, limit cycling, or other dynamic difficulties.

Loop gain should not vary more than a 4-to-1 ratio; otherwise, the dynamic performance of the loop suffers unacceptably. There is nothing magic about this specific ratio; it is simply one which many control practitioners agree produces an acceptable range of gain margins in most process control loops.

This guideline forms the basis for the following EnTech gain limit specification (From *Control Valve Dynamic Specification*, Version 2.1, March 1994, EnTech Control Inc., Toronto, Ontario, Canada):

> Loop Process Gain = 1.0 (% of transmitter span)/(% controller output)
>
> Nominal Range: 0.5 - 2.0 (Note 4-to-1 ratio)

Note that this definition of the loop process includes all the devices in the loop configuration except the controller. In other words, the product of the gains of such devices as the control valve assembly, the heat exchanger, pressure vessel, or other system being controlled, the pump, the transmitter, etc. is the process gain.

Because the valve is part of the loop process as defined here, it is important to select a valve style and size that will produce an installed flow characteristic that is sufficiently linear to stay within the specified gain limits over the operating range of the system. If too much gain variation occurs in the control valve itself, it leaves less flexibility in adjusting the controller. It is good practice to keep as much of the loop gain in the controller as possible.

Although the 4-to-1 ratio of gain change in the loop is widely accepted, not everyone agrees with the 0.5 to 2.0 gain limits. Some industry experts have made a case for using loop process gain limits from 0.2 to 0.8, which is still a 4-to-1 ratio. The potential danger inherent in using this reduced gain range is that the low end of the gain range could result in large valve swings during normal operation. It is good operating practice to keep valve swings below about 5%. However, there is also a danger in letting the gain get too large. The loop can become oscillatory or even unstable if the loop gain gets too high at some point in the travel. To ensure good dynamic performance and loop stability over a wide range of operating conditions, industry experts recommend that loop equipment be engineered so the process gain remains within the range of 0.5 to 2.0.

Process optimization requires a valve style and size be chosen that will keep the process gain within the selected gain limit range over the widest possible set of operating conditions. Because minimizing process variability is so dependent on maintaining a uniform installed gain, the range over which a valve can operate within the acceptable gain specification limits is known as the control range of the valve.

The control range of a valve varies dramatically with valve style. Figure 14-5 shows a line-size butterfly valve compared to a line-size globe valve. The globe valve has a much wider control range than the butterfly valve. Other valve styles, such as V-notch ball valves and eccentric plug valves generally fall somewhere between these two ranges.

Because butterfly valves typically have the narrowest control range, they are generally best suited for fixed-load applications. In addition, they must be carefully sized for optimal performance at fixed loads.

If the inherent characteristic of a valve could be selected to exactly compensate for the system gain change with flow, one would expect the installed process gain (lower curve) to be essentially a straight line at a value of 1.0.

Unfortunately, such a precise gain match is seldom possible due to the logistical limitations of providing an infinite variety of inherent valve trim characteristics. In addition, some valve styles, such as butterfly and ball valves, do not offer trim alternatives that allow easy change of the inherent valve characteristic.

This condition can be alleviated by changing the inherent characteristics of the valve assembly with nonlinear cams in the feedback mechanism of the positioner. The nonlinear feedback cam changes the relationship between the valve input signal and the valve stem position to achieve a desired inherent valve characteristic for the entire valve assembly, rather than simply relying upon a change in the design of the valve trim.

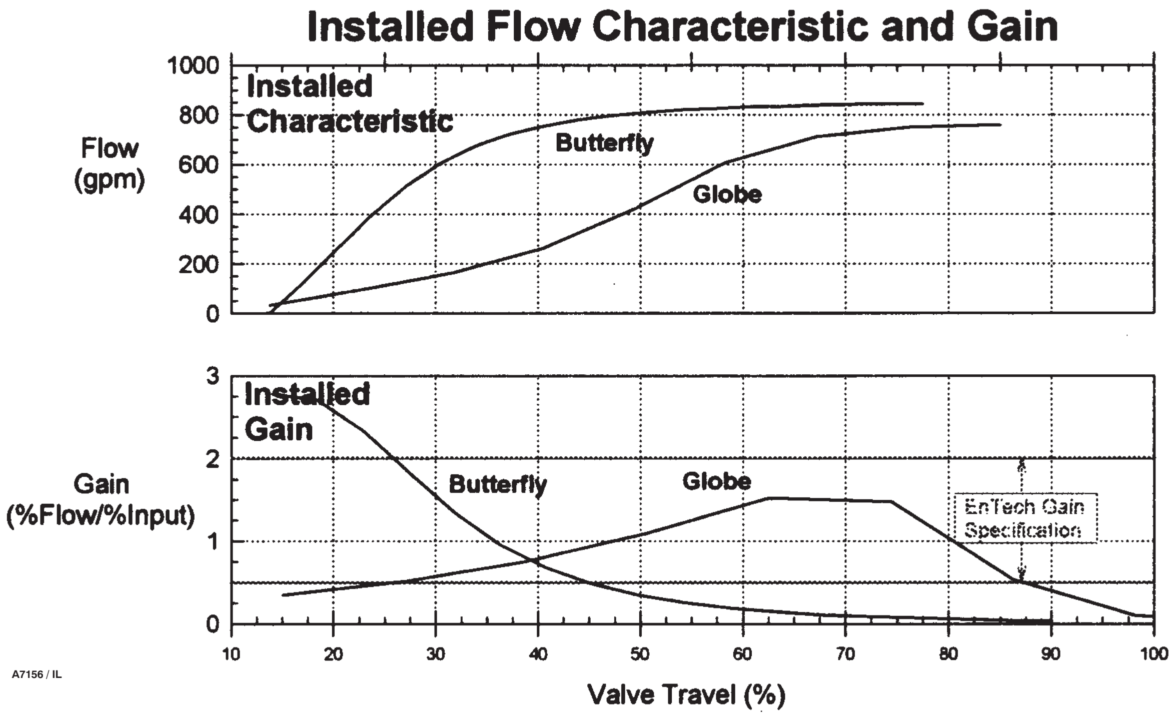


Figure 14-5. *Effect of valve style on control range*

Although the use of positioner cams does affect modifying the valve characteristic and can sometimes be useful, the effect of using characterized cams is limited in most cases. This is because the cam also dramatically changes the positioner loop gain, which severely limits the dynamic response of the positioner. Using cams to characterize the valve is usually not as effective as characterizing the valve trim, but it is always better than no characterization at all, which is often the only other choice with rotary valves.

Some electronic devices attempt to produce valve characterization by electronically shaping the I/P positioner input signal ahead of the positioner loop. This technique recalibrates the valve input signal by taking the linear 4-20 mA controller signal and using a pre-programmed table of values to produce the valve input required to achieve the desired valve characteristic. This technique is sometimes referred to as forward path or set point characterization.

Because this characterization occurs outside the positioner feedback loop, this type of forward path or set point characterization has an advantage over characterized positioner cams. It avoids the problem of changes in the positioner loop gain. This method, however, also has its dynamic limitations. For example, there can be places in a valve range where a 1.0% process signal change might be narrowed through this characterization process to only a 0.1% signal change to the valve (that is, in the flat regions of the characterizing curve). Many control valves are unable to respond to signal changes this small.

The best process performance occurs when the required flow characteristic is obtained through changes in the valve trim rather than through use of cams or other methods. Proper selection of a control valve designed to produce a reasonably linear installed flow characteristic over the operating range of the system is a critical step in ensuring optimum process performance.

Valve Sizing

Oversizing of valves sometimes occurs when trying to optimize process performance through a reduction of process variability. This results from using line-size valves, especially with high-capacity rotary valves, as well as the conservative addition of multiple safety factors at different stages in the process design.

Oversizing the valve hurts process variability in two ways. First, the oversized valve puts too much gain in the valve, leaving less flexibility in adjusting the controller. Best performance results when most loop gain comes from the controller.

Notice in the gain curve of Figure 14-4, the process gain gets quite high in the region below about 25% valve travel. If the valve is oversized, making it more likely to operate in or near this region, this high gain can likely mean that the controller gain will need to be reduced to avoid instability problems with the loop. This, of course, will mean a penalty of increased process variability.

The second way oversized valves hurt process variability is that an oversized valve is likely to operate more frequently at lower valve openings where seal friction can be greater, particularly in rotary valves. Because an oversized valve produces a disproportionately large flow change for a given increment of valve travel, this phenomenon can greatly exaggerate the process variability associated with dead band due to friction.

Regardless of its actual inherent valve characteristic, a severely oversized valve tends to act more like a quick-opening valve, which results in high installed process gain in the lower lift regions (Figure 14-4). In addition, when the valve is oversized, the valve tends to reach system capacity at relatively low travel, making the flow curve flatten out at higher valve travels (Figure 14-4). For valve travels above about 50 degrees, this valve has become totally ineffective for control purposes because the process gain is approaching zero and the valve must undergo wide changes in travel with very little resulting changes in flow. Consequently, there is little hope of achieving acceptable process variability in this region.

The valve shown in Figure 14-4 is totally misapplied because it has such a narrow control range (approximately 25 degrees to 45 degrees). This situation came about because a line-sized butterfly valve was chosen, primarily due to its low cost, and no consideration was given to the lost profit that results from sacrificing process variability through poor dynamic performance of the control valve.

Unfortunately, this situation is often repeated. Process control studies show that, for some industries, the majority of valves currently in process control loops are oversized for the application. While it might seem counter-intuitive, it often makes economic sense to select a control valve for present conditions and then replace the valve when conditions change.

When selecting a valve, it is important to consider the valve style, inherent characteristic, and valve size that will provide the broadest possible control range for the application.

Economic Results

Consideration of the factors discussed in this chapter can have a dramatic impact on the economic results of an operating plant. More and more control valve users focus on dynamic performance parameters such as dead band, response times, and installed gain (under actual process load conditions) as a means to improve process-loop performance. Although it is possible to measure many of these dynamic performance parameters in an open-loop situation, the impact these parameters have becomes clear when closed-loop performance is measured. The closed-loop test results shown in Figure 14-6 demonstrate the ability of three different rotary valves to reduce process variability over different tuning conditions.

This diagram plots process variability as a percent of the set point variable versus the closed-loop time constant, which is a measure of loop tuning. The horizontal line labeled Manual, shows how much variability is inherent in the loop when no attempt is made to control it (open-loop). The line sloping downward to the left marked Minimum Variability represents the calculated dynamic performance of an ideal valve assembly (one with no non-linearities). All real valve assemblies should normally fall somewhere between these two conditions.

Not all valves provide the same dynamic performance even though they all theoretically meet static performance purchase specifications and are considered to be equivalent valves (Figure 14-6). Valve A does a good job of following the trend of the minimum variability line over a wide range of controller tunings. This valve shows excellent dynamic performance with minimum variability. In contrast, Valves B and C fare less well and increase in variability as the system is tuned more aggressively for decreasing closed-loop time constants.

All three valve designs are capable of controlling the process and reducing the variability, but two designs do it less well. Consider what would

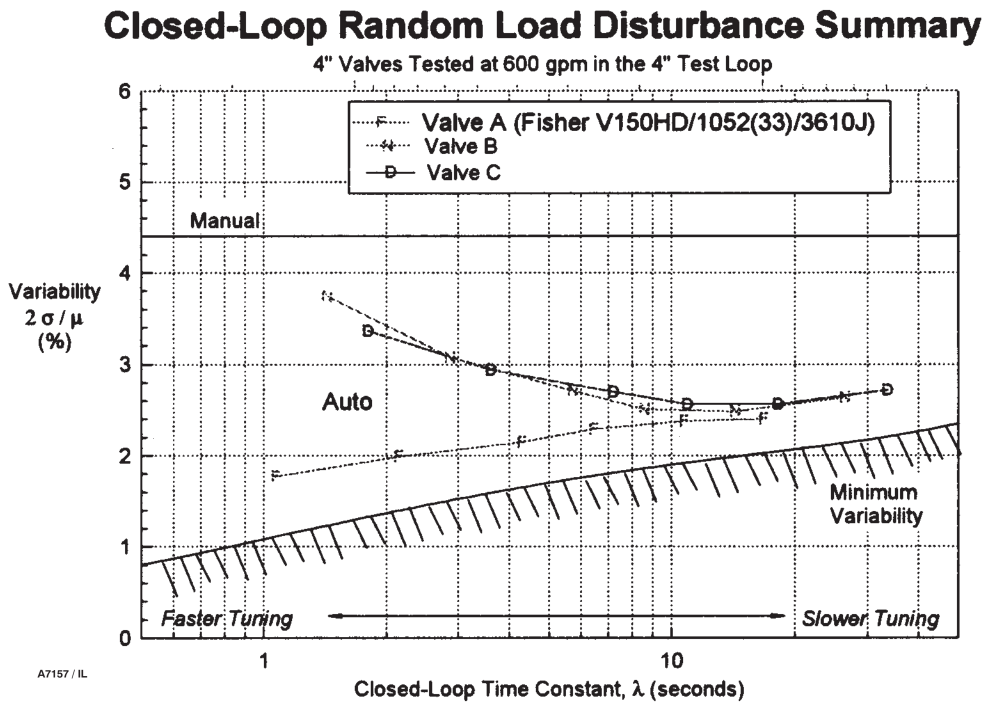


Figure 14-6. *Closed-loop performance*

happen if the poorer performing Valve B was replaced with the best performing Valve A, and the system was tuned to a 2.0 second closed-loop time constant.

The test data shows this would result in a 1.4% improvement in process variability. This might not seem like much, but the results over a time can be impressive. A valve that can provide this much improvement every minute of every day can save significant dollars over a single year.

By maintaining closer adherence to the set point, it is possible to achieve a reduction in raw materials by moving the set point closer to the lower specification limit. This 1.4% improvement in this example converts to a raw material savings of 12,096 U.S. gallons per day. Assuming a material cost of US $0.25 per gallon, the best valve would contribute an additional US $3,024 per day directly to profits. This adds up to an impressive US $1,103,760 per year.

The excellent performance of the better valve in this example provides strong evidence that a superior control valve assembly can have a profound economic impact. This example is only one way a control valve can increase profits through tighter control. Decreased energy costs, increased throughput, less reprocessing cost for out-of-spec product, and so on are all ways a good control valve can increase economic results through tighter control. While the initial cost might be higher for the best control valve, the few extra dollars spent on a well-engineered control valve can dramatically increase the return on investment. Often the extra initial cost of the valve can be paid for in a matter of days.

As a result of studies such as these, the process industries have become increasingly aware that control valve assemblies play an important role in loop/unit/plant performance. They have also realized that traditional methods of specifying a valve assembly are no longer adequate to ensure the benefits of process optimization. While important, such static performance indicators as flow capacity, leakage, materials compatibility, and bench performance data are not sufficiently adequate to deal with the dynamic characteristics of process control loops.

Summary

The control valve assembly plays an extremely important role in producing the best possible performance from the control loop. Process optimization means optimizing the entire process, not just the control algorithms used in the control room equipment. The valve is called the final control element because the control valve assembly is where process control is implemented. It makes no sense to install an elaborate process control strategy and hardware instrumentation system capable of achieving 0.5% or better process control and then to implement that control strategy with a 5% or worse control valve. Audits performed on thousands of process control loops have provided strong proof that the final control element plays a significant role in achieving true process optimization. Profitability increases when a control valve has been properly engineered for its application.

Control valves are sophisticated, high-tech products and should not be treated as a commodity. Although traditional valve specifications play an important role, valve specifications must also address real dynamic performance characteristics if true process optimization is to be achieved. It is imperative that these specifications include such parameters as dead band, dead time, response time, etc.

Finally, process optimization begins and ends with optimization of the entire loop. Parts of the loop cannot be treated individually to achieve coordinated loop performance. Likewise, performance of any part of the loop cannot be evaluated in isolation. Isolated tests under non-loaded, bench-type conditions will not provide performance information that is obtained from testing the hardware under actual process conditions.

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