Control valves for pump protection (recirculation) service

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Abstract

*In hydraulic systems involving a centrifugal pump a control valve is usually the preferred device for converting the signals transmitted by instrumentation into a controlled diversion of the pump delivery in order to protect the pump from overheating. This type of service for a control valve falls into the “severe” category. An incorrectly specified valve can result in valve failure and damage to the pump.*
Introduction

This paper discusses the essential procedures involved in the application of control valves for the protection of pumps operating at low flow conditions. Automatic Recirculating Valves (ARC Valves), although they do not fall into the category of control valves, do play an important role in pump protection, so a reference to these is also included.

Technical terms used for pump operating conditions can vary from one reference to another. For clarification those used here are listed in Figure 1. This illustrates that the dynamic delivery head is simply the pressure that would be indicated by a pressure gauge mounted on the pump outlet. It therefore does not include the velocity head. Adding the velocity head gives the discharge head. The total static head is the static delivery head plus or minus the suction head. The total head is the discharge head with the suction head taken into account.

![Fig. 1: Pump pressure head terminology](image)

Dynamic delivery head $h_{dd} = h_{sd} + h_f$
Discharge head $hd = h_{sd} + h_v + h_f$
Total head $H = h_{ts} + h_v + h_f$

Service requirements

Referring to the pump total head characteristic in Figure 2 it will be seen that when the pump is handling its designed flow at maximum efficiency, the delivery pressure is increased to the design value and there is a small increase in fluid temperature. As the demand for flow is decreased and with the pump speed maintained at a constant value there is an increase in delivery pressure accompanied by a gradual increase in temperature until the flow falls to around 50% of maximum. At this point there is a rapid increase in temperature. This can cause damage to the pump, particularly if it produces pockets of vapour. The usual way of avoiding this is to apply a recirculating
device at the pump outlet. This maintains the flow through the pump above a predetermined minimum, regardless of the decrease in demand.

![Graph showing typical centrifugal pump performance curve including temperature rise at low flows.](image)

Fig. 2: Typical centrifugal pump performance curve including temperature rise at low flows.

The temperature rise may be calculated from these equations:

\[
\Delta T_i = \left( H \left( \frac{1-\eta}{\eta} \right) \right) \frac{g}{C}
\]

\[
\Delta T_i = \left( \frac{P}{\rho} \left( \frac{1-\eta}{\eta} \right) \right) \frac{1}{C}
\]

(see appendix for derivation of these equations)

\[
\Delta T = \Delta T_i + \Delta T_c
\]

where \( H \) = total head, m (sometimes referred to as total dynamic head)

\( P \) = total pressure, Pa (sometimes referred to as total dynamic pressure)

\( \rho \) = density of liquid \( m^3/kg \)

\( \eta \) = pump efficiency

\( C \) = specific heat of fluid \( \text{joules/kg.K} \)

\( \Delta T_i \) = rise in temperature due to pump inefficiency \( K \)

\( \Delta T_c \) = rise in temperature due to micro-compression of fluid \( K \)

\( \Delta T \) = total rise in temperature \( K \)

\( g = 9.81 \text{ m/sec}^2 \)
These equations are derived using the concept that all the heat generated is absorbed by the fluid and is equal to the energy input minus the hydraulic energy imparted to the fluid. In other words, the rise in temperature $\Delta T_i$ is principally related to the pump’s inefficiency. This concept is reasonable because under the low flow conditions the temperature rise of the fluid is too rapid for any appreciable heat, per unit mass of fluid, to be transmitted to the pump casing.

$\Delta T_C$ is the rise in temperature of the fluid associated with its infinitesimal compression under the pressure generated by the pump. This can be determined from the thermodynamic property tables for the fluid on the assumption that the “compression” of the fluid within the pump is adiabatic-isentropic. It is not attributable to the pump’s mechanical inefficiency since it is the temperature rise generated, under static conditions, by the application of pressure. The volume is considered to be almost constant and therefore there is no significant “work done”. For boiler feedwater at a temperature of 181°C at the pump inlet, the temperature rise $\Delta T_C$ will be 0.131°C per 1 MPa increase in pressure. This is insignificant compared with $\Delta T$ when the pump is operating in the low flow regime but in the proximity of the Best Efficiency Point (BEP) it can be similar to $\Delta T$ which has a low value. At all operating conditions the total rise in fluid temperature is the sum of the rise that can be attributed to the pump’s inefficiency and the rise due to the minute compression of the fluid.

The risk of damage to the pump can be avoided if the flow is always maintained above the recommended minimum value regardless of the “low to zero” demand downstream. The actual value of this minimum flow condition depends on the pump design and the safety requirements of the plant. It can be as low as 17%, but for some plants operating at very high pressures, it can be as high as 48% of the design flow. This requirement for a minimum flow can be met by inserting a minimum flow device in a line connecting the pump discharge to some point at the suction end. On power plants this connection is made either to the deaerator or the condenser (Figure 3a) whilst on other services such as oil well water injection it is made to the supply well or the bypass fluid is discharged to waste (Figure 3b).

It will be appreciated that when this bypass device is operating it is handling very severe conditions due to the pressure drop being the maximum that the pump can generate. Variable speed pumps can alleviate this to some extent but plant design requirements usually insist on the bypass device being capable of operating with the pump running at maximum speed and with the demand below the pump’s safe operating value. The recirculating device, in some high pressure systems, can be called upon to reduce an inlet pressure, for example, of $52.0 \times 10^3$ kPa to an outlet pressure of $0.5 \times 10^3$ kPa and there are many examples of pressure reductions from $32.0 \times 10^3$ kPa to $0.5 \times 10^3$ kPa. These conditions can lead to the onset, within the device, of cavitation, impact erosion and vibration, so the design must incorporate features to avoid these problems or minimise their effects.

The requirement for pump protection at low loads occurs wherever a high pressure pump is operating, but the most frequent examples are boiler feedwater systems and injection systems for oil wells.
Fig. 3a: Diagram of a power station feedwater system

1 = Main feedwater control valve
2 = Feed water pump
3 = HP Heaters
4 = Feedwater start-up control valve
5 = Feed water pump recirculating valve
6 = Deaerator
7 = Condenser
8 = Boiler
9 = Main feedwater control valve & start-up control valve alternative installation

Fig. 3b: Oil well water injection system

1 = water well
2 = submerged pump
3 = to dump
4 = recirculating valves=
5 = oil/gas well
6 = injection pump
The operating principles of different recirculating systems

The operating principles of ARC valves (Figure 4) are fundamentally different from those of control valves. These are discussed in the “Mechanical Valves” section. There are three different systems for the provision of low flow protection by means of recirculation. These are:

1) Continuous recirculation.

This is achieved by using a fixed orifice as the pump bypass (Figure 5a). This method is usually found only on older power stations. Recirculation is operating continuously without any regard for the flow through the pump. It does protect the pump from low flow problems but it is wasteful of energy. This system is not of great importance so it will receive only a brief mention.

2) On/Off recirculation

This system achieves recirculation using a control valve operating in the on/off mode (Figure 5b). The instrumentation commands the valve to open whenever the pump flow falls below the minimum value. Above this value, the recirculating valve is closed. This system offers the opportunity to enhance the instrument flow measurement by adding pressure and temperature to the signal controlling the recirculating valve. Adequate protection for the pump is achieved but when recirculating the valve is fully open under all conditions. This means it is for much of its time bypassing more fluid than is necessary and is thus wasting energy. The on/off was used extensively on old power stations for the protection of boiler feedwater pumps and is still used to a limited extent. It is accomplished using an instrument controlled on/off valve located in the pump bypass line. It will be described in some detail.

3) Modulating recirculation

The most efficient and flexible system for pump protection is one that modulates the recirculating flow in response to the constant measurement of the pump variables – delivery, pressure, flow, temperature – individually or in any combination. Recirculation is not limited to the two values- zero and maximum. It is, at any time, just sufficient to supplement the demand on the pump in order to maintain the pump’s minimum flow requirement. With this system the recirculation is never more than necessary. It is for this reason that it is more efficient than the “continuous” and the “on/off” systems.

A further advantage of the fully modulating system is its ability to satisfy the NPSH requirements. A design characteristic of power plants is the available net positive suction head (NPSH). The boiler feed pump also has a required NPSH which is indicated on the manufacturers characteristic curve. To prevent cavitation at the pump, the plant NPSH must be above the NPSH required by the pump. The modulating recirculating valve under the control of the transducer signals allows the required minimum flow through the pump and at the same time ensures that the pump NPSH is below the plant available NPSH.

The modulating recirculation system, is very important, and so the selection of the control valves for this system will be fully described.
Recirculating valves and associated equipment

a) Continuous system

Although this system normally employs a pressure reduction device, rather than a control valve in the recirculation line, it should receive some attention here. The device is usually an orifice or a number of orifices in series and, except in special cases, it has a fixed Cv (flow capacity).

\[ Q = C_v \sqrt{\frac{\Delta p}{G}} \]

(see appendix for definition of Cv)

Continuous recirculating systems are not prone to appreciable variations in bypass flow so it is only necessary to ensure that the pressure reduction device is capable of passing this flow at the pressure drop dictated by the pump delivery pressure and the pressure in the receiving vessel (suction vessel, deaerator etc). In rare cases where there is a significant variation in bypass flow and for reasons of initial economy the continuous system is used, pressure reduction devices are available with a variable Cv. The variation is achieved by the movement of a plunger operated manually. This varies the orifice area, but the rangeability is limited. The device with a variable Cv could quite well be a manually operated control valve.

The pressure drop from the pump delivery pressure to the recirculation receiving vessel can be sufficiently high as to cause cavitation and flashing at a single orifice reduction device (ref 1), but correctly sized multiple orifice devices which break down the pressure drop into manageable increments can resolve this problem.

The disadvantages of the continuous bypass system are highlighted in the earlier section describing the different recirculating systems. Its advantages are simplicity and capital economy.

b) On / Off system

The control valve used in the recirculation line in this system must be capable of continuous trouble-free operation over long periods with the pressure drop as set by the pump delivery pressure and the receiving vessel pressure. This can be very high requiring the valve to be equipped with a special multistage trim with critical surfaces protected by erosion resistant materials. The severity of the service is sometimes reduced on the valve by including downstream one or more orifice plates (used as a pressure reduction device). During the initial period of valve opening, and also immediately prior to full closure, the pressure drop across the recirculating valve is at a maximum. Extremely fast valve movement is therefore recommended to hold these intervals to a minimum.

The on/off recirculating valve is sometimes used as a feed pump start-up valve. With the main feedwater valve closed, the recirculating valve is opened and this condition is maintained throughout the warm-up time. The feedwater valve is then slowly opened and the plant comes on line. At the time when the pump increases to some specified value which for low to medium pressure pumps is probably between 10% and 25% of maximum, the recirculating valve is closed and remains closed until the
control system require it to open because the flow demand from the pump has fallen below the safe level.

The recirculating valve usually receives its control signal from an instrument measuring the pump delivery pressure or flow but it can also be controlled by a signal originating from the measurement of the temperature of the fluid at the pump outlet. In more sophisticated control systems a combination of these three measurements may be used. (refer to Figures 5a, 5b, 5c and 5d)

![Diagram 4](image)

1 = Pump  
2 = ARC valve  
3 = To process or to boiler  
4 = Back pressure device  
5 = Supply  

Fig. 4: Pump automatic recirculating system based on a combined non-return and automatic recirculating valve (ARC valve)

![Diagram 5a](image)

1 = Pump  
2 = Non-return valve  
3 = To process or to boiler  
4 = Recirculating valve  
5 = Back pressure device  
6 = Supply  
7 = ARC valve  

Fig. 5a: Pump automatic recirculating system based on pressure measurement  
* For on/off control, A = pressure switch or on/off controller  
* For modulating control, A = modulating pressure transmitter/controller
Fig. 5b: Pump automatic recirculating system based on flow measurement
* For on/off control, A = flow switch or on/off controller
* For modulating control, A = modulating flow transmitter/controller

1 = Pump
2 = Non-return valve
3 = To process or to boiler
4 = Recirculating valve
5 = Back pressure device
6 = Supply
7 = ARC valve

Fig. 5c: Pump automatic recirculating system based on temperature measurement
* For on/off control, A = temperature switch or on/off controller
* For modulating control, A = modulating temperature transmitter/controller

1 = Pump
2 = Non-return valve
3 = To process or to boiler
4 = Recirculating valve
5 = Back pressure device
6 = Supply
7 = ARC valve

Fig. 5d: Pump automatic recirculating system based on multivariable measurement.
* For modulating control based on any combination of flow, pressure and temperature measurement, A = transmitters communicating with an optimising controller.
c) Mechanical on/off and partially modulating automatic recirculating valves (ARC valves)

An often used alternative to the control valve in recirculating applications is the automatic recirculating valve (ARC Valve) which unlike the control valve is installed in line with the pump discharge piping.

The ARC valve is a combined non-return and recirculating valve and requires no external means of control or operation. (see Figure 4). It is sometimes described as a leak-off non-return valve. With a reduction in flow, the plug or the vane of the non return valve closes under the influence of gravity assisted by a spring in some designs, and in so doing it starts to open the recirculating valve. When the flow falls below a minimum value the non-return valve closes completely and opens the recirculating valve. The link between the non return plug or vane and the recirculating valve is self contained and mechanical. This type of leak-off non return valve is simple, requiring no instrumentation or external operating medium but it does not have the flexibility of adjustment available with the instrument controlled systems. The elementary designs of ARC valves have an on/off action with the disadvantage of being wasteful of energy. The design in Figure 6 is more advanced and has a degree of modulation achieved by the characterised flow/lift relationship of the main non-return valve plug. A characterised flow/lift relationship is also imparted to the recirculating valve by means of shaped porting in the hollow sliding stem which is the variable control orifice of the recirculating valve. These features mitigate the concerns over the wastage of energy, as found in a 100% on/off system. The small size of the recirculating valve, due to its being constricted within the body of the non-return valve, and its dependence solely on a mechanical operating force provided by the movement of the non-return valve plug, imposes limitations on the capacity of the recirculating valve, possibly restricting the size of pump that can be protected by ARCs, This low pressure design is limited by the maximum pressure drop that the recirculating valve can handle without cavitation and vibration. Recent more advanced designs, described in the following paragraphs have extended the range of service conditions. Figure 8 is a graphic representation of the limiting service conditions for ARCs but these frontiers are being expanded continuously with the introduction of improved designs.
Fig. 6: ARC valve for low to intermediate pressure services. The recirculating valve is operated directly by the movement of the non-return valve disc.

Figure 7 indicates a more advanced design of an ARC valve. It embodies all the features of the valve in Figure 6 but the movement of the non-return valve plug or disc operates the plug of the recirculating valve by means of a lever which through its mechanical advantage is able to exert a greater operating force. This along with the multiple staging and balancing of the recirculating valve enables it to handle high pressure drops without cavitation or instability. The balancing of the recirculating valve plug may be accomplished by seal balancing as in Figure 7 or by the use of a pilot valve (design not shown) requiring only minimum movement to equate pressures at each end of the plug.
Fig. 7: ARC valve for high pressure services. (Courtesy of HBE Engineering Inc.). The seal balanced recirculating valve is operated though a lever by movement of the non-return valve disc.

The thermal leak-off non-return valve is a variation of this design. Instead of the recirculating valve operating under the influence of the movement of the non-return plug or disc, it is equipped with a solenoid which is connected to a thermal transducer sensing the temperature of the feed water downstream of the pump. At a preset temperature, indicating that the pump is overheating due to a lack of demand for feed water, the recirculating valve opens. This thermal design has more flexibility than the purely mechanical version since the temperature at which the leak-off feature operates can be adjusted, but its operation is on/off, which is wasteful of energy.

An alternative to the thermal ARC valve is a similar design but instead of the recirculating valve being operated by the temperature of the pump discharge it is operated by the discharge flow. The recirculating valve solenoid, instead of being connected to a thermal transducer, is connected to a flow transducer which can be set to open the valve at a minimum flow in the pump outlet line.

In these self-contained valves the leak-off or recirculating valve has to be limited in its physical size. The best designs embody features to handle the high pressure drops (25.0 x 10^3 kPa ) including the most difficult cases where the outlet pressure may be below the vapour pressure of the feedwater. To handle this without premature
mechanical and material failure requires some form of multistage anticavitation trim and the use of erosion resistant materials but the limitations of size of these recirculating valves presents difficulties and the service conditions within their range is more limited than those within the capabilities of a separate control valve.

![Graph showing service limits of ARC valves and control valves.](image)

**Fig. 8:** Representation of approximate maximum service limits of automatic recirculation non-return valves. (With design developments these limits may change. The manufacturer should be consulted.)

There is no clear-cut boundary between those services that can be managed by an automatic recirculating valve (ARC valve) and a control valve working with the necessary instrumentation. ARC valves are usually limited to pump delivery pressures not exceeding $9.75 \times 10^3$ kPa but some high pressure designs extend this upper limit to $25.0 \times 10^3$ kPa.

Figure 8, which has been produced from practical experience and manufacturers’ data, provides a very approximate guide to the limitations of ARC valves, but the decision on which type of valve to use depends mostly on the service and operating requirements. Some of the advantages of each valve are listed below:

- [List of advantages]
Advantages of ARC valves

a) Self contained requiring no external control equipment
b) Easy installation
c) Requires no external source of power
d) Combines in one unit the main delivery non-return valve, the flow sensing element (movement of non-return valve stem), the recirculating valve and in some cases the additional pressure reduction device in the recirculating line.

Advantages of the Control Valve with associated instrumentation.

a) Increased flexibility – the instruments allow adjustments to be made to suit changing conditions and different performance requirements.
b) The recirculation flow can be controlled by measurement of the pump discharge flow, by measurement of the pump discharge pressure, by measurement of the pump temperature, or by any combination of these.
c) Very high pressure drops from the pump delivery pressure to the recirculated fluid receiving vessel impose severe working conditions on the recirculating valve. There are control valves designed exclusively for these services incorporating features that can not be accommodated in the limited size of the ARC recirculating valves.

Fully modulating control recirculating valves

For complete modulating control, there is no substitute for an instrument operated control valve independent of the non-return valve. The conditions under which this valve has to operate are seldom uncomplicated due, for the most part, to the pressure drop being of a magnitude that requires careful consideration of the design and the materials of construction. The valve trim components are the most critical. Materials noted for their resistance to impact erosion, such as Stellite 6, stainless 420 (hardened), Alloy 20 and Inconel, are often specified. The valve must be capable of providing tight shut-off for long periods and, at other times, it must be capable of operating in partially open positions for flow control with stability. The type of control valve illustrated in Figure 9 meets the requirements for most modulating recirculating services but, for the highest pressures, purpose designed valves capable of breaking down the pressure drop into numerous stages are essential. Pressure drops from the pump delivery to the deaerator or condenser, as experienced in power stations, may be in the region of \(35 \times 10^3\) kPa, but in other industrial systems (oil well injection) the pressure drop between the pump discharge and the outflow (efflux system) may be as high as \(52.0 \times 10^3\) kPa.

The valve illustrated in Figure 10 is one form of such a valve. Depending on the pressure reduction required, it may have any number of discrete stages from 5 to 13. Between these stages, there are turns in the fluid path which increase the actual number of pressure reduction stages to 8 and 24 respectively. These stages are so designed as to constrain approximately 90% of the pressure reduction to occur in the first group of stages. For example, with a valve having 7 discrete stages, 90% of the
reduction would be taken by the first 3 stages. This reduces the risk of cavitation because these stages are ejecting their high velocity jets into high pressure fluid, at pressures too high to allow cavitation to develop. The low pressure drop across the final stages ensures low trim exit velocities. Additionally all the plug heads will be designed so that they approach the closure of their flow paths at the same instant thus reducing the pressure drop across the seating surfaces of the final closing member (the main plug) which secures tight shut-off.

A valve using the same principle of pressure reduction in stages but with a very different trim design is illustrated in Figures 11 and 12. This is generally known as a "labyrinth" valve because the trim consists of a large number of stacked discs, each disc carrying a large number of small grooves forming tortuous paths with many right angled bends through which the fluid is constrained to flow. In its passage through these labyrinths the fluid changes direction many times and with each change of direction there is a reduction in pressure. The number of turns in a disc may vary from 4 to 44 depending on the pressure reduction required. The size of the passages is varied from inlet to outlet to ensure that most of the pressure drop is taken by the first series of turns with the final turns taking a low drop and thus ensuring low trim exit velocities. The valve 

High pressure drop liquid flow, as explained in many published articles on the subject (ref 1) has the capability of damaging valves through the impact of jets and the implosion of cavitating bubbles eroding the trim materials. Excessive jet velocities within the valve may lead to vibration which can damage valve components and pipework. The recirculating valve must incorporate features designed to prevent the onset of these problems.

These two types of valve for high pressure pump recirculation have been used successfully with pumps delivering at pressures ranging from $28.0 \times 10^3$ kPa to $48.1 \times 10^3$ kPa.

The valve is shown with 3 discrete stages. These include 4 interstage turns, making a total of 7 stages of pressure reduction. A valve of this type would be suitable for medium pressure modulating control service. If combined with a 2 stage pressure reduction device downstream it could operate with a maximum pressure drop of $12.20 \times 10^3$ kPa.

For higher pressure services it could be equipped with 4 discrete stages making a total of 10 stages when the 6 interstage turns are included. Equipped with a single stage, the valve is also suitable for low pressure services in either modulating or on/off systems. In cases of pressures at the higher end of the range, one or more orifices may be installed in the downstream line.

Fig. 9: Control valve for pump recirculation control (courtesy Koso Kent Introl Ltd)
Fig. 10: High pressure multistage control valve equipped with 9 discrete stages (1) including 7 interstage turns making 16 pressure reduction stages. The valve also has 3 exit baffles (2). (Courtesy Koso Kent Introl Ltd.)

Fig. 11: Recirculating control valve with a multistage labyrinth trim for high pressure services. (courtesy Koso Kent Introl Ltd.)
1 = Labyrinth trim
2 = exit diffuser
Fig. 12: Detail of discs from a labyrinth trim showing the tortuous flow paths. The trim is made up of many of these discs stacked together. The Cv can be varied by varying the number of discs.
1 = detail of flow paths
2 = stacked discs

The importance of “tight shut-off”

Whether it be of conventional or special design the recirculating valve will remain closed for 95% of its time. With high inlet pressures and very low outlet pressures, it is essential that tight shut-off is achieved. Any leakage at these high pressure drops will rapidly erode the seating surfaces and the leakage flow will increase. It is therefore advisable to hold the pressure drop across the recirculating valve to the lowest possible value. This valve may be connected to the condenser or to the deaerator. The highest pressure drop is experienced when it is connected to the condenser rather than the deaerator, which explains why in modern plants it is usually connected to the deaerator.

Operation of the plant for a long period on a very low load requires the recirculating valve to operate over this period in the almost closed position resulting in erosive damage to the seating surfaces. This can be avoided by arranging the control signal to move the valve quickly from, say, 10% open to fully closed.

ANSI/FCI 70/2 – 2006 Class V leakage requirements (ref 7) are achievable with the designs described using hard metallic materials such as ASTM A461 Gr 630 17-4PH, ASTM A276 440C, Stellite 6, Inconel and Tungsten Carbide. The most damaging working condition for the valve seating surfaces is when the control signal commands the valve to operate in the almost closed position, such as 3% to 5% open. In this operating regime practically all the pressure drop will be taken across the first seat in the multistage trim. The valves with discrete stages and the labyrinth valve in Figures 9, 10 and 12 have built-in seat protection. This incorporates a protective shroud or band around the seating surfaces that transfers the flow control surfaces in low lift
operation to a non-critical area of the plug and control guide. It also includes a dead-band in the flow lift characteristic that reduces the flow to an infinitesimal amount well before the seating surfaces make contact with each other. Where recirculating valves do not have this feature, it is established practice to build into the control system an on/off characteristic that opens or closes the valve rapidly between 0 to 10% open.

The following examples are based on the technical specifications of actual pump recirculation valves as supplied for power station boiler feed and oil well water injection services.

**Examples of Pump Recirculation Valves.**

**Boiler Feed Pump Recirculation Valve**

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<thead>
<tr>
<th>Service conditions</th>
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<tbody>
<tr>
<td>Fluid</td>
<td>Water</td>
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<tr>
<td>Temperature</td>
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<tr>
<td>Flow rate</td>
<td>100,200 kg/hr</td>
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<tr>
<td>Inlet pressure</td>
<td>14.1 x 10³ kPa a</td>
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<td>Outlet pressure</td>
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<td>Pressure drop</td>
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<td>Vapour pressure</td>
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<td>Critical pressure</td>
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<td>Viscosity</td>
<td>1.0 centipoise</td>
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</table>

Calculated values

- Cv 10.36 (Kv 8.96)
- Valve operating position 75.5% of rated travel
- Pressure recovery factor $F_L$ 0.96 (ref 4)
- Cavitation coefficient for process $\sigma$ 1.033
- Coefficient of incipient cavitation for valve $\sigma_i$ 1.010
- Calculated noise level 70dBA
- Inlet/Outlet velocity 5.9 m/sec
- Pipework correction factor $F_p$ 1.0 (ref 4)
- Power conversion level 0.013 mW distributed through 12 stages
- Cavitation – not encountered because $\sigma$ for process is greater than $\sigma_i$ for valve. (ref 3)

The valve specified for this service is:

- Globe style valve with multi discrete stages.
- Size inlet/outlet 80mm/80mm
- Body material ASTM A217 WC6  Body rating ANSI 1,500
- Flanged ANSI 1,500 RTJ
- Trim Multistage 7 discrete stages (12 pressure reduction stages including interstage turns)
- Balanced plug
- Trim material Combination of 17.4 PH stainless, Tungsten
Carbide and 316L stainless with Weartech 50 facing.
Design Cv 20 US units (Kv 17.30)
Flow/ Lift characteristic equal percentage
Leakage in closed position within ANSI Class IV

Actuator:
Spring opposed diaphragm size 150
Spring range 40 to 120 kPa g
Air supply 280 kPa g
Rated travel 40mm (1.5 inches)
Air to close action
In event of air failure the valve locks.
Valve positioner ABB type TZID C120
Time for valve movement 9 secs to open / 7 secs to close.

Oil Well Water Injection Pump Recirculating Valve.

Service conditions

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Water</th>
<th>Temperature 25 °C</th>
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<tbody>
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<td>Flow rate</td>
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<td>Inlet pressure</td>
<td>48.1 x 10^3 kPa a</td>
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<tr>
<td>Outlet pressure</td>
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<tr>
<td>Pressure drop</td>
<td>47.5 x 10^3 kPa</td>
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<tr>
<td>Specific gravity</td>
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<td>Vapour pressure</td>
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<td>Critical pressure</td>
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<tr>
<td>Viscosity</td>
<td>1.0 centipoise</td>
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</tr>
</tbody>
</table>

Calculated values

Cv 46.51 (Kv 40.23)
Valve operating position 86.04 %
Pressure recovery factor $F_L$ 0.988 (ref 4)
Cavitation coefficient for process $\sigma$ 1.012
Coefficient of incipient cavitation for valve $\sigma_i$ 1.007
Calculated noise level 71 dBA
Inlet / Outlet velocity 21.5 m/sec / 7.5 m/sec
Pipework correction factor 0.984
Power conversion level 0.353 Mw
Cavitation not encountered because $\sigma$ for the process is greater than $\sigma_i$ for the valve. (ref 3)

The valve specified for this service is:

Angle style valve with multi discrete stages
Size inlet 120mm outlet 203mm
Body material  Ferr. 255 – 3SF Duplex  NACE MR-01-75
Body connections to customer specified design
Trim  Multistage with 9 discrete stages–15 pressure reduction stages
- including interstage turns. ( with exit baffle unit)
Trim material  Tungsten Carbide
Balanced Plug
Design Cv 58 US Units  (Kv 51.7)
Flow/Lift characteristic  equal percentage
Leakage in closed position  within  ANSI Class V

Actuator : -

Spring opposed diaphragm size 300
Spring range  20 to 100 kPa g
Air supply  280 kPa g
Rated travel   57mm
Air to close action
Valve opens in event of air failure
Valve positioner  ABB TZIM DIR AL
Time for valve movement  10 secs to open / 8 secs to close

An alternative design also specified for this service is : -

Angle style valve with labyrinth stacked disc trim
Size     inlet 120mm   outlet 203mm
Body material  Ferr. 255 – 3SF Duplex  NACE MR-01-75
Body connections to customer specified design
Trim  Labyrinth - 56 discs, 24 turns in each disc.( + exit baffle unit)
Trim material  Inconel
Balanced Plug
Design Cv 58 US Units  (Kv 51.7)
Flow/Lift characteristic  linear
Leakage in closed position  within  ANSI Class V

Actuator
Double acting piston.
Air supply  280 kPa g
Rated travel   177 mm
Air to close action
Valve opens in event of air failure
Valve positioner – double acting
Time for valve movement  12 secs to open / 10 secs to close.
Conclusions

The major limiting factors governing the choice between an automatic recirculating valve (ARC) and a control valve for pomp protection service are their capabilities of handling the high pressure drop from the pump delivery to the condenser or deaerator and the maximum required rate of flow. ARC valves have a proven record on service conditions within the manufacturers’ recommendations and they have the added advantage of being simple to install. However, if flexibility of operating parameters and continuous monitoring of the plant performance are required, the choice must be a control valve with its associated instrumentation and, of course, regardless of flexibility requirements in operation, control valves can be the only choice when pressures and flow rates exceed the ARC valves’ design limits.

Whichever valve type is being considered careful sizing and attention to the design requirements demanded by the service conditions are paramount.

The author thanks Dr J.T. Turner for reading through the proofs and offering suggestions for improvements to the paper.
APPENDIX

Flow Coefficients - Cv and Kv

These are the valve flow coefficients used to simplify the sizing equations. (see ref 4). The only difference between the two is in the units used for the flow conditions. The definition for Cv is the flow in US gallons per minute of water at 60°F, with a pressure drop across the valve of 1 psi (6,895 pa). The units of Cv are therefore US gallons/minute. The definition for Kv is the flow in cubic metres of water at 10.6°C with a pressure drop across the valve of 1 bar (1 \times 10^5 pa). The units of Kv are therefore cubic metres/hour. Using these coefficients the basic sizing equations for liquid turbulent flow are:

\[
Q = C_v \sqrt{\frac{\Delta p}{G}} \quad \text{where } Q = \text{US gpm, } \Delta p = \text{psi, } G = \text{specific gravity}
\]

\[
Q = \frac{C_v}{1.156} \sqrt{\frac{\Delta p}{G}} \quad \text{where } Q = \text{m}^3/\text{hr}, \Delta p = \text{bar, } G = \text{specific gravity}
\]

\[
Q = \frac{C_v}{0.0865} \sqrt{\frac{\Delta p}{G}} \quad \text{where } Q = \text{m}^3/\text{hr}, \Delta p = \text{kPa, } G = \text{specific gravity}
\]

\[
Q = K_v \sqrt{\frac{\Delta p}{G}} \quad \text{where } Q = \text{m}^3/\text{hr}, \Delta p = \text{bar, } G = \text{specific gravity}
\]

\[
Q = \frac{K_v}{10} \sqrt{\frac{\Delta p}{G}} \quad \text{where } Q = \text{m}^3/\text{hr}, \Delta p = \text{kPa, } G = \text{specific gravity}
\]

It will be seen from references (4) and (8) that these flow coefficients, although based on liquid flow, are used in sizing equations for gases and vapours, made possible by the introduction of an expansion factor. The procedures for testing valves to allocate Cv and Kv values will be found in reference (5). Information on valve sizing for conditions of critical flow will be found in references (4) and (9).

Pressure recovery Factor \( F_L \)

This factor allows the pressure recovery, that usually takes place between the vena contracta and the valve outlet, to be recognised when sizing valves for liquid flow conditions when the pressure drop is in excess of the critical value. For normal flow conditions the pressure recovery is included in the valve Cv and Kv. (see ref 4)

\[
F_L = \sqrt{\frac{\Delta p}{\Delta p_{VC}}} \quad \text{where } \Delta p = \text{pressure drop across the valve} \quad \Delta p_{VC} = \text{pressure drop across the vena contracta.}
\]

A complete explanation of \( F_L \) and its use in sizing calculations will be found in IEC standard 60534-2-1 (ref 4).

Flow/Lift Characteristic
A control valve, assuming there is a constant pressure drop across it, has a predetermined relationship between the fluid flowing through it and the amount of the valve opening, usually termed the “lift”. This relationship, or characteristic, depends on the design of the valve plug and seat, known collectively to valve engineers as the “trim”. The most important feature is the way in which the orifice area varies with the valve lift. The two most frequently used characteristics in process control are “linear” and “equal percentage”. As its name implies, the linear characteristic gives a linear relationship between flow and lift, whilst the equal percentage gives an exponential relationship such that at any point in the valve lift, a further incremental increase in the lift gives an equal percentage increase in flow. (see ref 6).

**Cavitation Coefficient \( \sigma \)**

The onset of cavitation in a control valve can be predicted by the use of the cavitation coefficient \( \sigma \).

\[
\sigma = \frac{P_1 - P_y}{P_1 - P_2}
\]

Various stages of cavitation are indicated by different values of \( \sigma \), the higher the value the greater the risk of cavitation. Valve manufacturers must determine the value at which their valve may be affected by a level of cavitation which will impair normal performance. The manufacturer’s safe limit is usually represented by the symbol \( \sigma_{mr} \). Details of test methods to determine the values of \( \sigma \) and its application to valve calculations are given in ISA RP75.23 (ref 3).

**Derivation of equation for the calculation of the temperature rise of a liquid with the pump running at low flow conditions.**

The equation is based on the assumption that the difference between the shaft input power and the fluid output power minus the fluid input power is converted into an increase in temperature of the fluid. This is to say that the temperature rise is due to pump inefficiency. This precept ignores any increase in temperature of the pump casing but as the fluid is flowing rapidly the rise in temperature of the casing per mass/sec of fluid is very small.

The mechanical efficiency of a pump is defined as :

\[
\eta = \frac{\text{useful output}}{\text{useful input}} = \frac{N_U}{N}
\]

The total head \( H = h_d - h_s \) where \( h_d \) = discharge head m
\( h_s \) = suction head m
\( H \) = total head m

The total pressure head \( P = H \rho g \)
\[ \rho = \text{density \ kg/m}^3 \]
\[ g = 9.8 \text{ m/sec}^2 \]
\[ P = \text{Pascal} = \text{Newton/m}^2 \]
\[ m = \text{mass flow \ kg/sec} \]

\[ N_U = m \frac{g}{\rho} \frac{P}{g} = m \frac{g}{\rho} H \quad \text{----- (1)} \]

\[ N_U = \text{useful output power, watts} \]
\[ N = \text{useful input power, \ watts} \]

It is a fair assumption that the rise in liquid temperature is attributable to the inefficiency of the pump through turbulence and friction and also to the physical phenomena of the temperature rise accompanying the minute compression of the liquid which can be obtained from thermal property tables and has nothing to do with the pump’s inefficiency. The rise due to pump inefficiency is represented by \( T_i \) and the rise due to compression by \( T_C \).

The power required to produce \( T_C \) is:

\[ m.\Delta T_C \text{ watts} \quad C = \text{specific heat of fluid} = 4187 \text{ joules/kg.K} \]
\[ \Delta T_C = \text{rise in temp. due to compression. K} \]

If \( \Delta T_i \) is the rise in temperature of the fluid due to the pump’s mechanical inefficiency (turbulence and friction).

\[ m.\Delta T_i \text{ watts} \]

if \( \Delta T \) is the total rise in temperature \( \Delta T = \Delta T_i + \Delta T_C \)

\[ \Delta T = \frac{N - N_U}{m.C} + \Delta T_C \quad \text{----- K} \]

from equ (1)

\[ m = \frac{N_U}{g.H} \quad \Delta T = \frac{N - N_U}{N_U} \times \frac{H.g}{C} + \Delta T_C \quad \text{----- K} \]

since \( \eta = \frac{N_U}{N} \)

\[ \Delta T = \left( \frac{1}{\eta} - 1 \right) \frac{H.g}{C} + \Delta T_C \quad \text{----- K} \]
Nomenclature

\( h_{sd} = \) static delivery head \( \quad \) \( \text{------ m} \)
\( h_{dd} = \) dynamic delivery head \( \quad \) \( \text{------ m} \)
\( h_s = \) suction head \( \quad \) \( \text{------- m} \)
\( h_t = \) total static head \( \quad \) \( \text{------- m} \)
\( h_v = \) velocity head \( \quad \) \( \text{------- m} \)
\( h_f = \) friction head \( \quad \) \( \text{------- m} \)
\( v = \) fluid velocity \( \quad \) \( \text{------- m} \)
\( d = \) pipe inside diameter \( \quad \) \( \text{------- m} \)
\( g = 9.81 \) \( \quad \) \( \text{---------- m/sec}^2 \)
\( f = \) friction factor for D’Arcy equation
\( h_d = \) discharge head \( \quad \) \( \text{------- m} \)
\( H = \) total head \( \quad \) \( \text{------- m} \)
\( C = \) specific heat of fluid \( \quad \) \( \text{-------- joule / kg.K} \)

\( \sigma_{mr} = \) manufacturer’s cavitation coefficient for valve \( \quad \) \( \sigma_{mr} = \frac{P_1 - P_v}{\Delta P_{mr}} \)

\( \sigma_i = \) incipient cavitation coefficient \( \quad \) \( \sigma_i = \frac{P_1 - P_v}{\Delta P_i} \)

\( \sigma = \) cavitation coefficient for service \( \quad \) \( \sigma = \frac{P_1 - P_v}{P_1 - P_2} \)

\( P_1 = \) valve inlet pressure \( \quad \) \( \text{------ kPa} \)
\( P_2 = \) valve outlet pressure \( \quad \) \( \text{------ kPa} \)
\( P_v = \) vapour pressure of liquid at valve outlet temperature \( \quad \) \( \text{------ kPa} \)
\( \Delta P_{mr} = \) manufacturer’s recommended valve pressure drop to avoid \( \quad \) \( \sigma_v \) falling below the safe value. \( \quad \) \( \text{------ kPa} \) (see ref. 3)
\( \Delta P_i = \) pressure drop producing incipient cavitation \( \quad \) \( \text{------ kPa} \)
References


2) ANSI/FCI Standard 70/2-2006 “Classification of seat leakages”

3) ISA Recommended Practice RP.75.23 – 1995 “Evaluation of control valve cavitation.”

4) IEC Standard – 60534-2-1 “Control valve sizing equations”

5) IEC Standard - 60534-2-3 “Control valve flow capacity test Procedures.”

6) IEC Standard - 60534-2-4 “Control valve inherent flow characteristics and rangeability.”

7) ANSI/FCI Standard 70-2 “Control valve seat leakage.”

8) E.W.Singleton – “The derivation of the IEC/ISA control valve Sizing equations.” Published by Kent Introl Jan 2002


10) Karassik, Messina, Cooper - “Pump Handbook”, Published by McGraw Hill.
Fig. 13: Multistage fully modulating recirculating valve with air operated piston actuator. (courtesy Koso Kent Introl Ltd.)

About the Author

Ed Singleton received his training in the mid 40s with the Blakeborough Company and the Hammel-Dahl company in the USA. He decided to form his own company in 1967 to specialise in the design and manufacture of control valves for the difficult services presented by the emerging North Sea oil and gas industries. The company was “Introl”. Expansion was rapid and in 1973 it joined forces with the George Kent Group, which was then the largest British manufacturer of process control instrumentation. In the following 30 years the Kent Group went through a number of changes in ownership and in 2005 this culminated in a restructuring of the group resulting in “Kent Introl” joining its present owner, the Koso Group. Ed was managing director of the Introl Company from its foundation in 1967 until 1990 when he was retained as a consultant. He was for many years a member of British and IEC standards committees and in 1995 he was awarded the BVAMA Certificate of Merit for contributions to control valve technology.