Valve Handbook

Theory and Practice

INCENTIVE GROUP

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INCENTIVE-GROUP

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Foreword

When designing control systems it is important that each individual component be selected with the same care. From a selection and sizing point of view, the control valve is probably the most demanding component in a control system. Improperly selected control valves constitute the single most common cause for system malfunction and failure.

In earlier years (and even today in many cases) control valves were chosen more or less at random. The normal procedure was to select a valve of the same size as that of the pipe.

Since we can calculate systems more accurately today, for example with computer calculation programs, the demand for accurately sized valves is greater.

Our intention with this handbook is to provide a technical and practical guide to the selection and sizing of control valves in heating, ventilation and air conditioning systems. It is primarily intended to be a basic documentation for TA's internal training seminars.

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Controls Division
1993

Hans Palmertz

Contents

1 Control system

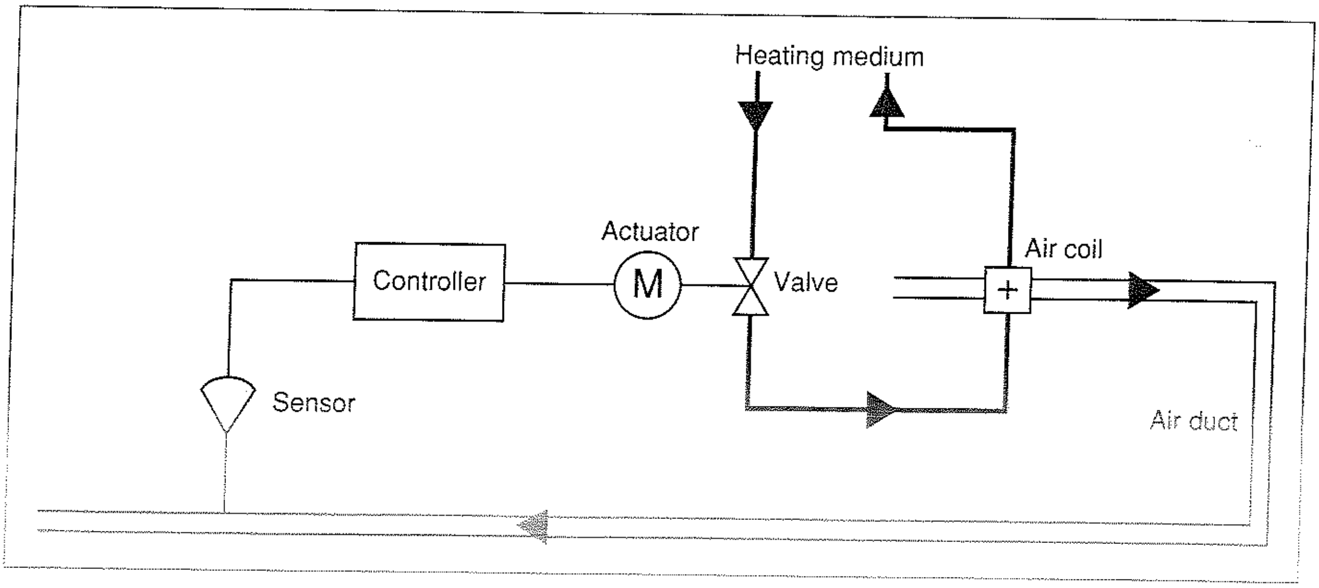
General design of control systems

Variations in the indoor climate demand that, heat energy either be supplied or removed, to maintain the desired indoor comfort. In the heating, ventilating and air conditioning (HVAC) field, water is commonly used as the heat carrier and heat exchangers are used to transfer the energy from one medium to another. This other medium may be either water, in an hydronic circuit, or air, in an air conditioning installation.

The most common control task is to maintain either a constant temperature, or a temperature that varies in a specific manner, in a space, a supply air duct, or a secondary heat distribution network (Fig. 1.1).

The heat output of the heat exchanger is controlled by a control valve, which is a component of the control circuit. If the temperature is to be maintained constant, the supplied heat must equal the heat loss. If the heat supplied exceeds the heat loss, the temperature will increase.

Fig. 1.1 Typical control system. Fundamental diagram for control of supply air temperature in an air conditioning system



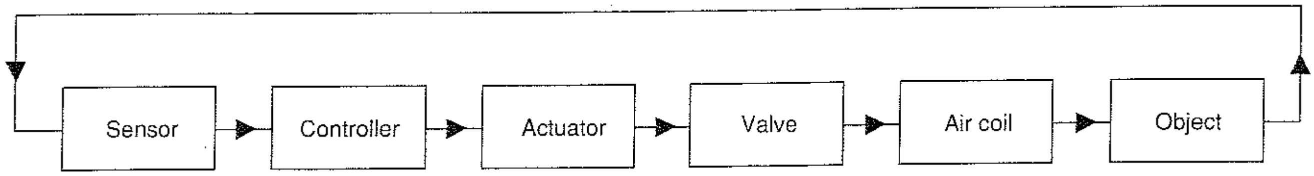


Fig. 1.2 Block diagram for the control system shown in Fig. 1

The control system includes a sensor, which measures the supply air temperature (actual value). The signal from the sensor is sent to a controller, which compares the actual value with the desired value, the set point. If these two signals are not identical, the controller will make the necessary correction, by increasing or decreasing the heat energy*.

The control device can be a control valve, which controls the flow to a radiator system or a coil.

If the energy source in a building is a boiler, the boiler can be connected directly to the radiator or air conditioning circuit. If the energy source is a district heating system, where the temperature and pressure are unsuitable for the secondary circuit, a heat exchanger must be connected between the district heating network and the radiator or air conditioning circuit. If, for instance, the pressure in the district heating network is 1.6 MPa (16 bar) and the temperature is 120°C, the primary circuit cannot be connected directly to the radiator circuit.

* A more comprehensive text on basic control theory is given in the Control Handbook, HVAC Systems

Heat exchangers

Heat exchangers are designed to exchange heat between:

- water – water
- steam – water
- oil/gas – water
- water – air
- steam – air
- oil/gas – air

By design, air coils normally are *tube heat exchangers*. Water passes through the tubes of the coil. The air to be heated or cooled flows past the outside of the coil.

The *plate heat exchanger* normally used, e.g. in district heating systems, is made up of thin corrugated plates, typically of stainless steel. The plates are clamped together by means of heavy end plates. The primary water flows on one side of the plate and the secondary water on the other.

There is a distinction between *preheating coils* and *reheating coils*. Preheating coils can be exposed to outdoor air. Precautions must be taken to ensure that these coils cannot freeze. Reheating coils, however, are normally exposed to preheated air only and therefore are not subject to freezing risk.

Hot water coils which are designed to be connected directly to a district heating network or to substations, are normally controlled by a two-way valve, which has been designed to operate at the high static pressures, differential pressures and temperatures prevailing in district heating networks.

Steam coils are fed with steam and are controlled by two-way valves, which have been designed to operate at critical pressure drops.

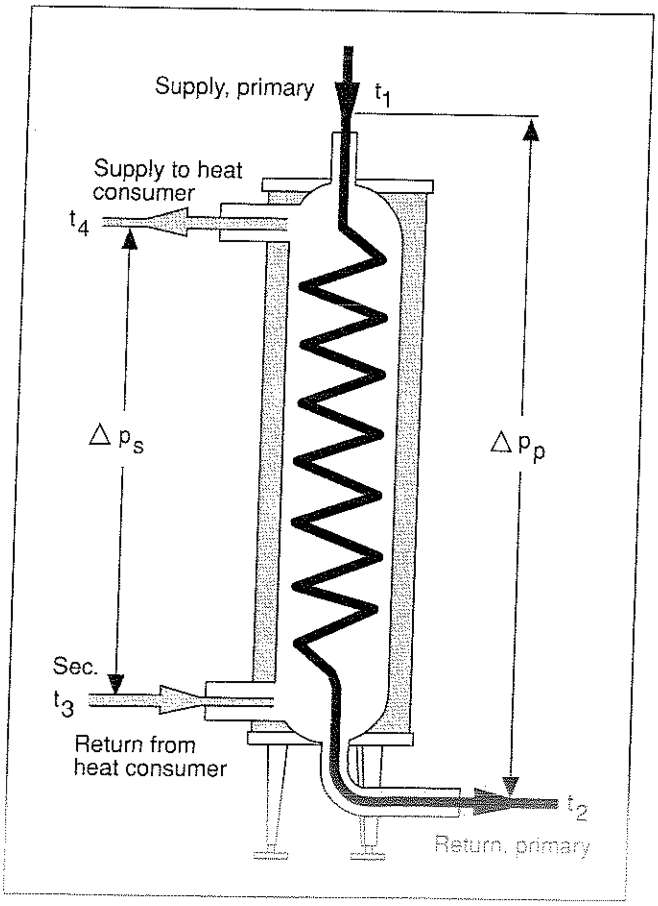


Fig 1.3 Heat Exchanger

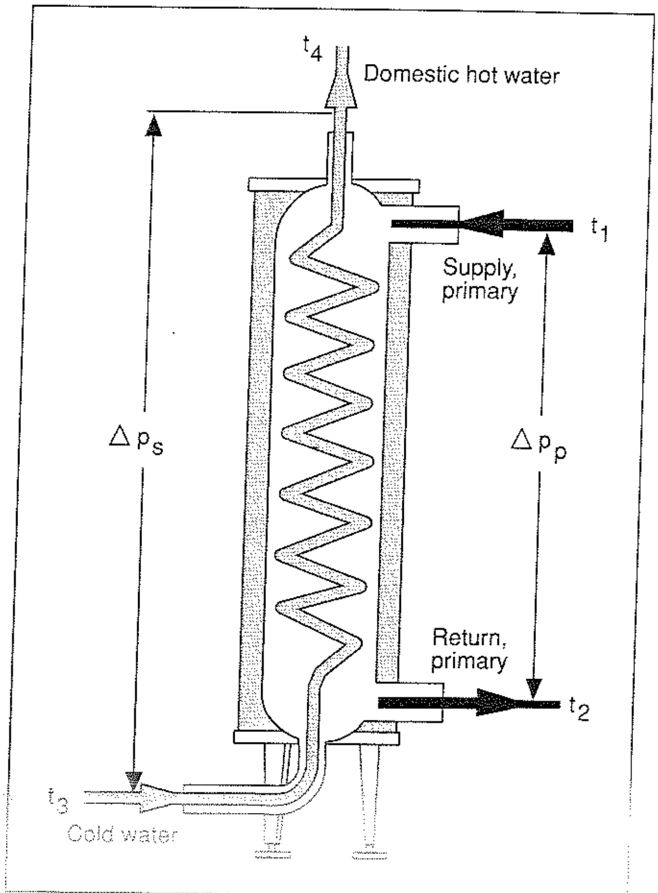


Fig 1.4 Water heater

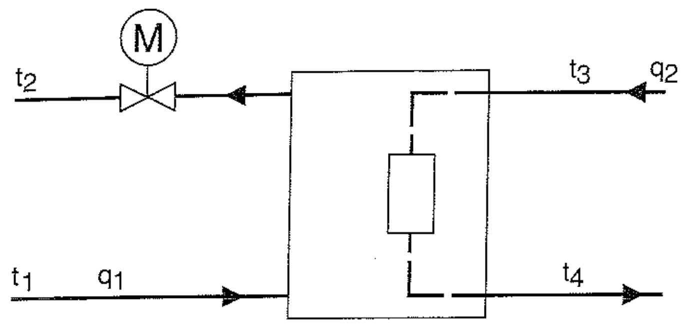


Fig 1.5 Water heater

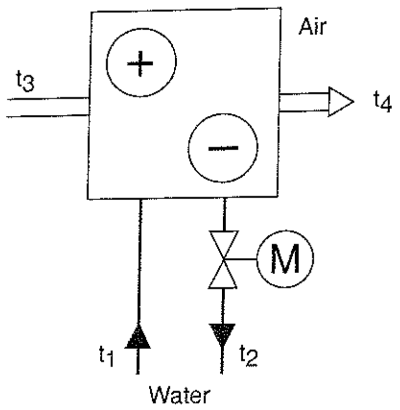


Fig 1.6 Heat exchanger

Dimensioning of heating installations

In water heaters and heat exchangers, the input heat equals the emitted heat.

In general:

$$P = q \cdot \rho \cdot c_p \cdot \Delta T$$

P = heat [kW]
q = flow [m^3/s]
ρ = density [kg/m^3]
c_p = specific heat [kJ/kg · K]
ΔT = temperature differential across heat exchanger [K]

After conversion to more easily handled units, we obtain:

For water:

$$P = 4.18 \cdot q_v \cdot \Delta T$$
$$P = 1.16 \cdot q \cdot \Delta T$$

P = emitted heat [kW]
q_V = volumetric flow [l/s]
q = volumetric flow [m^3/h]

For air:

$$P = 1.3 \cdot q_A \cdot \Delta T$$

P = emitted heat [kW]
q_A = volumetric flow [Nm^3/s] 0°C, 0.1 MPa (1 bar)

Emitted heat from saturated steam:

$$G = P \cdot 1.59$$

P = emitted heat [kW]
G = steam flow [kg/h]

Consideration has been given only to the heat which is released when steam is condensed to water.

Efficiency in heat exchangers

The following general formula applies to heat exchangers:

$$q_W \cdot \rho_W \cdot c_p \cdot (t_1 - t_2) = q_A \cdot \rho_A \cdot c_{PA} \cdot (t_4 - t_3)$$

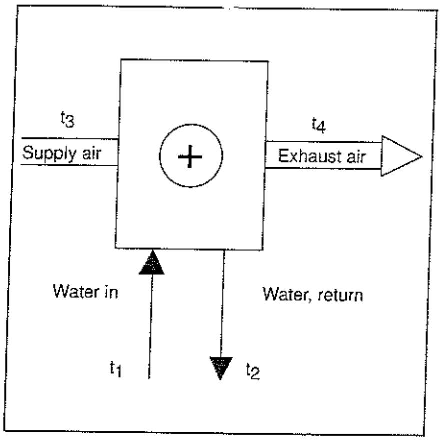


Fig 1.7 Heat exchanger for heating

For a given heat exchanger, with constant air and water flows, the following will apply:

Relative heating

$$\varepsilon_A = \frac{t_4 - t_3}{t_1 - t_2} = \text{constant air side}$$

$$\varepsilon_W = \frac{t_1 - t_2}{t_4 - t_3} = \text{constant water side}$$

which gives:

$$\varepsilon_W = \frac{q_A \cdot \rho_A \cdot C_{PA}}{q_W \cdot \rho_W \cdot C_{PW}} \cdot \varepsilon_A$$

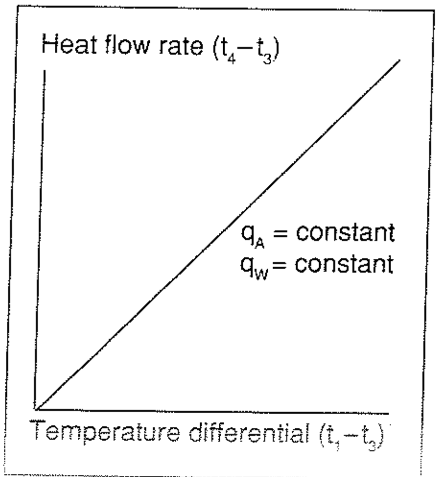


Fig 1.8 The heat output of the heat exchanger as a function of the temperature differential $(t_1 - t_3)$

The quantities ε_A and ε_W can be obtained from the calculations for the installation, or merely by performing simple temperature measurements at the installation. Sufficiently accurate water temperatures can be obtained from measurements made at the surfaces of pipes. As the relative heating, ε_A, enables us to determine whether or not the coil satisfies the requirements, ε_A is of considerable interest when approving an installation.

Transfer characteristic of heat exchangers

A heating or cooling coil's transfer characteristic is not linear but is a function of:

- the flow through the coil
- the supply water temperature
- the dimensioning temperature difference across the coil on the water side (ΔT, at full flow)

Fig. 1.9 shows the transfer characteristic for coils (convectors, radiators), for various dimensioning temperature differentials of the supply water.

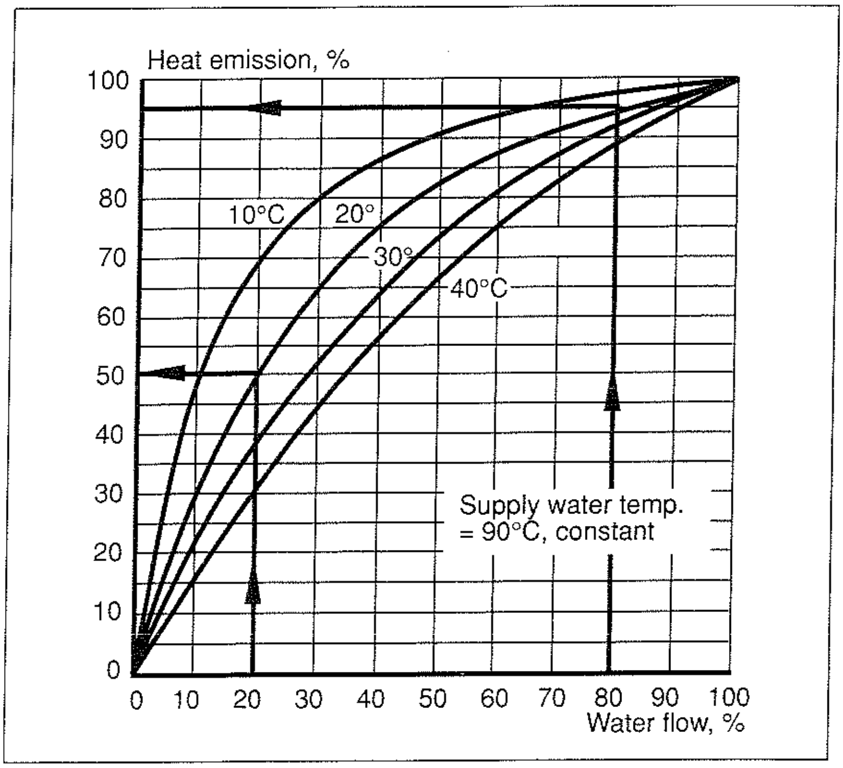


Fig 1.9 Heat emission of water to air heat exchangers, (radiators and convectors) at various temperature drops on the primary side. The diagram shows that if the flow is changed from 0 to 20%, the heat emitted by the heat exchanger will change from 0 to 50% (at ΔT = 20°C). If the flow is changed from 100 to 80% the heat emitted will change by a few percent.

The transfer characteristic of a water heat exchanger is largely a linear function between the heat emission and water flow, i.e. increasing flow results in increasing heat flow rate, as shown in Fig. 1.10.

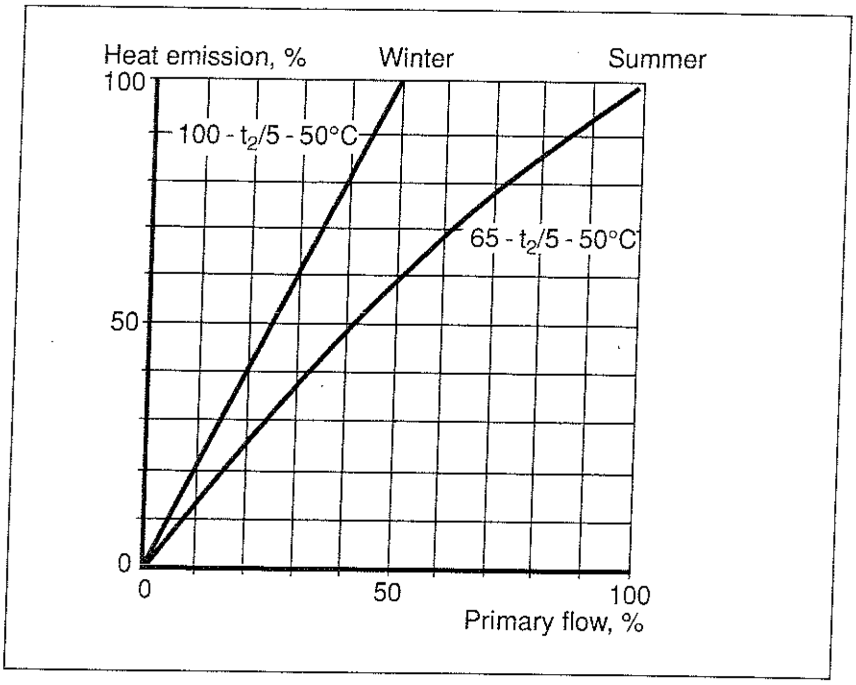


Fig 1.10 Heat emission of water/water heat exchanger at various temperature conditions.

Control of flow and temperature

The heat transfer of a heat exchanger can be controlled in two ways: by varying the flow through the heat exchanger while the temperature is held constant, or varying the temperature while the flow is held constant. The latter method is used to eliminate the risk of freezing for heating coils equipped with a separate circulating pump in the secondary circuit.

The different methods of control effect the transfer characteristic of the heat exchanger. Fig. 1.15 shows the transfer characteristic in the case of flow control and Fig. 1.16 shows the characteristic for temperature control.

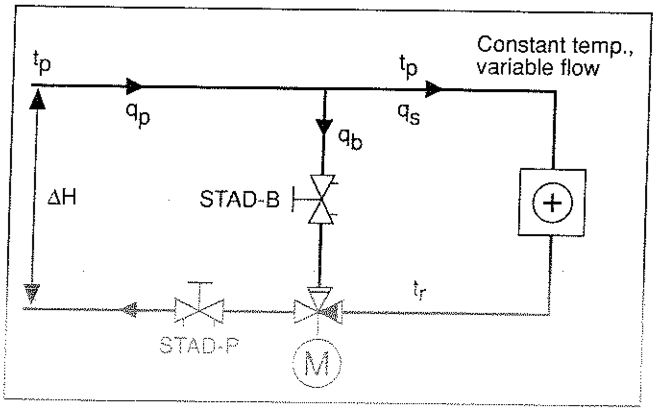


Fig 1.11 Example of flow control

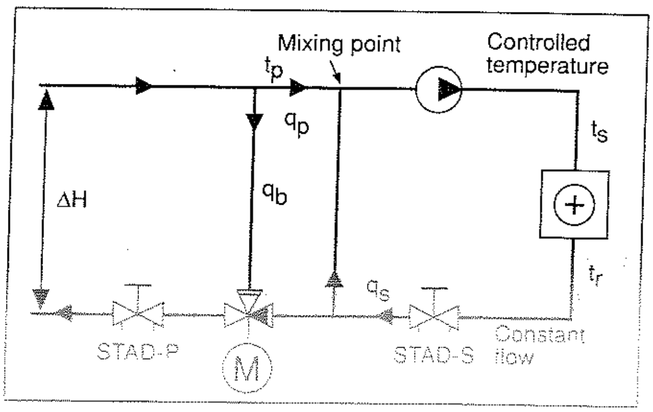


Fig 1.12 Example of temperature control

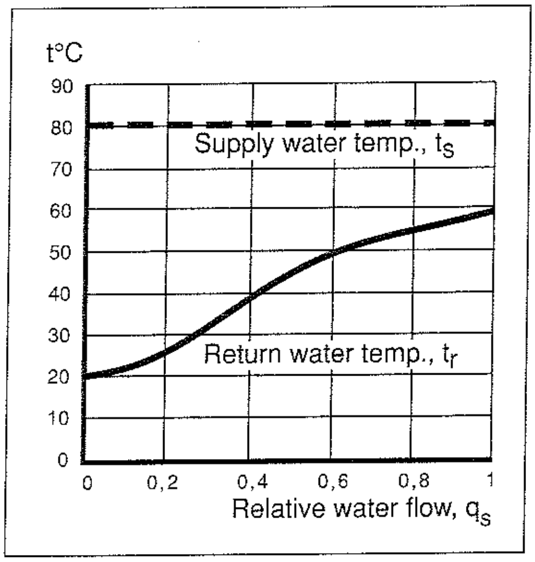


Fig 1.13 Supply and return temperatures at heat exchanger for variable water flow, i.e. water flow control.

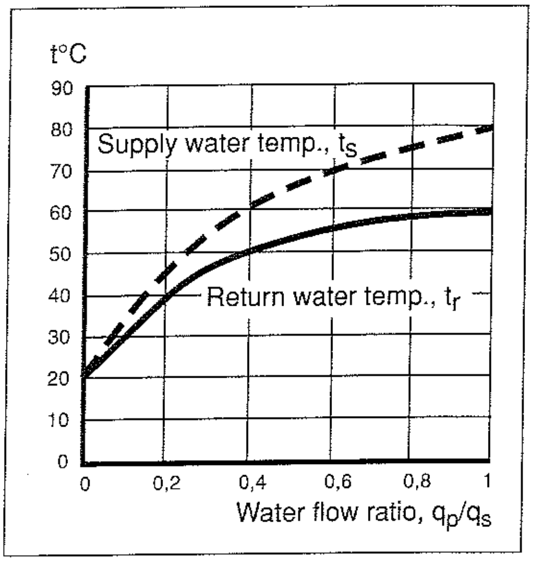


Fig 1.14 Supply and return temperatures at heat exchanger for constant water flow, i.e. temperature control.

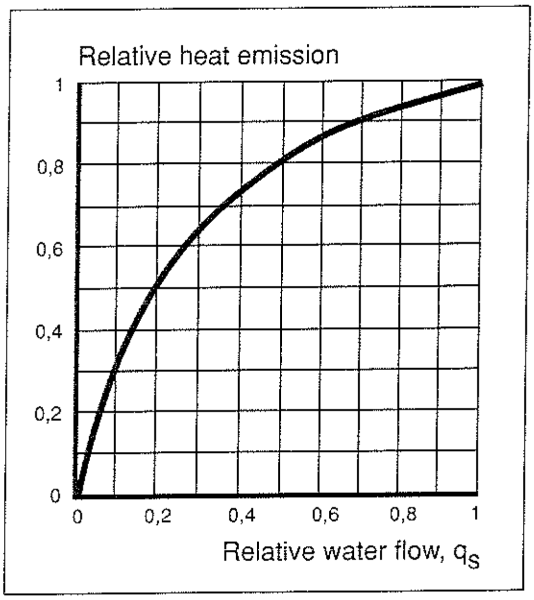


Fig 1.15 Heat emission curve versus flow control

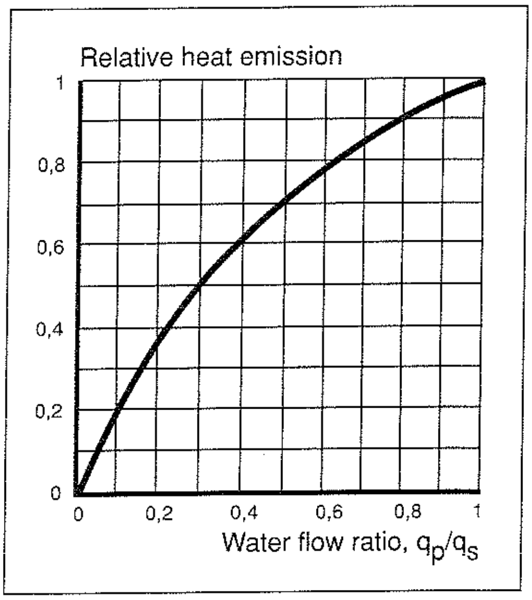


Fig 1.16 Heat emission curve versus flow

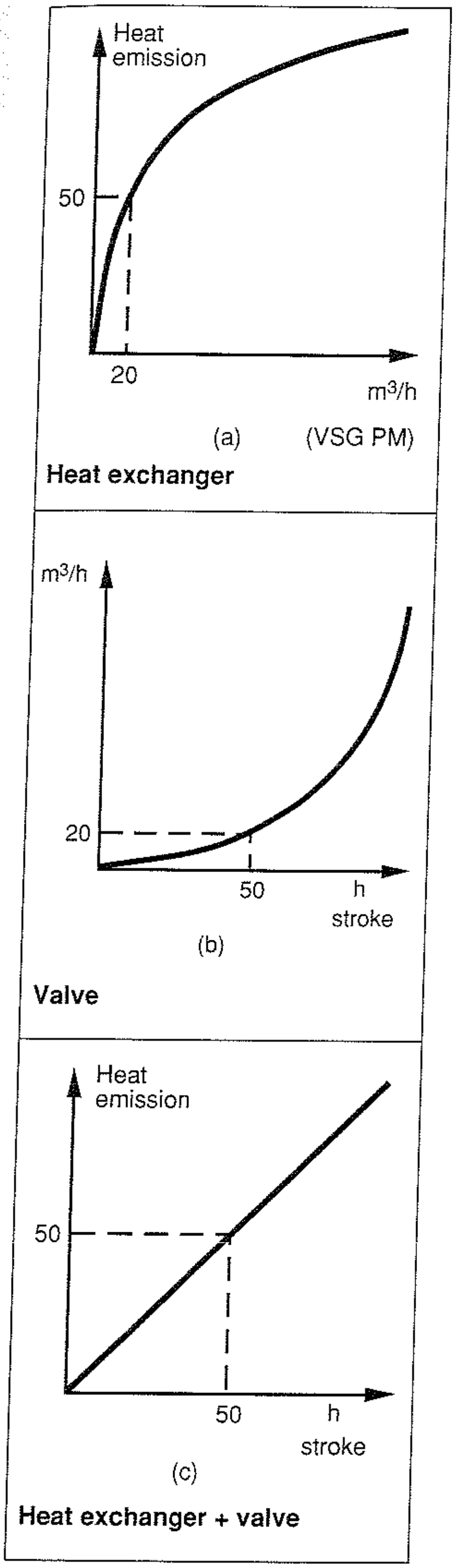


Fig 1.17 Diagrams, showing relationships between heat exchanger and control valve characteristics

Valves for controlling heat exchanger emission

The quantity of heat transfered by a heat exchanger is normally controlled with 2-way or 3-way valves.

2-way valves are used in systems with steam or district heating, to ensure the lowest possible return temperature. 3-way valves are used to ensure the minimum disturbance of the pressure and flow balance of the system.

Fig. 1.9 shows that, if the flow is changed from 0 to 20%, the heat flow rate of the heat exchanger will change from 0 to 50% (at $\Delta T = 20°C$). If the flow is changed from 100 to 80%, the heat flow rate of the heat exchanger will change by just a few percent. To attain stable control, it is therefore desirable to have constant gain in the control loop, at all loads.

For this reason control valves are designed so that the heat output (heat flow rate) and, therefore, the temperature at the sensor, are linear functions of the lift of the valve stem.

Fig. 1.17 shows how this is accomplished. A flow through the heat exchanger of 20% gives a heat output of 50%. The valve is therefore provided with a flow characteristic so that, when the valve is half open, only 20% of maximum water flow is obtained. Thus, when the valve is half open (i.e., 50% opening) the flow through the heat exchanger will be 20% and the heat output 50%. The flow characteristic of the valve therefore is nonlinear.

The most common standardized nonlinear flow characteristic is the *equal percentage (Eq%)*, or *logarithmic* characteristic.

In order to obtain a better control of the heat exchanger at low loads, the Eq% characteristic is often modified in the range just above the closed position. This flow characteristic is called *equal percentage modified, EQM*, or *modified logarithmic* characteristic.

Valves with linear flow characteristics are also standardized in a similar manner.

2 Valve types

General

In the fields of process and HVAC control, where the controlled medium is a liquid, gas or steam, the control valve is vital for controlling flows.

The control valve can be operated with various types of actuators, e.g.

- manual
- electrohydraulic
- electromechanical
- electrical, with solenoid
- pneumatic
- hydraulic
- electropneumatic
- self-acting (self-contained)

Controlled device

Standards for heating and ventilating installations typically use the following definition:

CONTROLLED DEVICE =
ACTUATOR + VALVE + LINKAGE

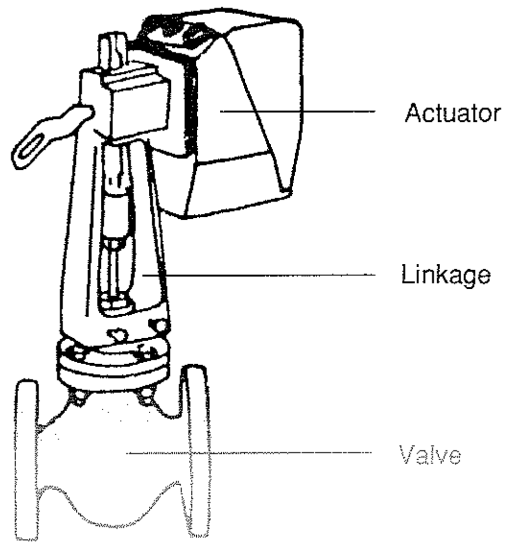
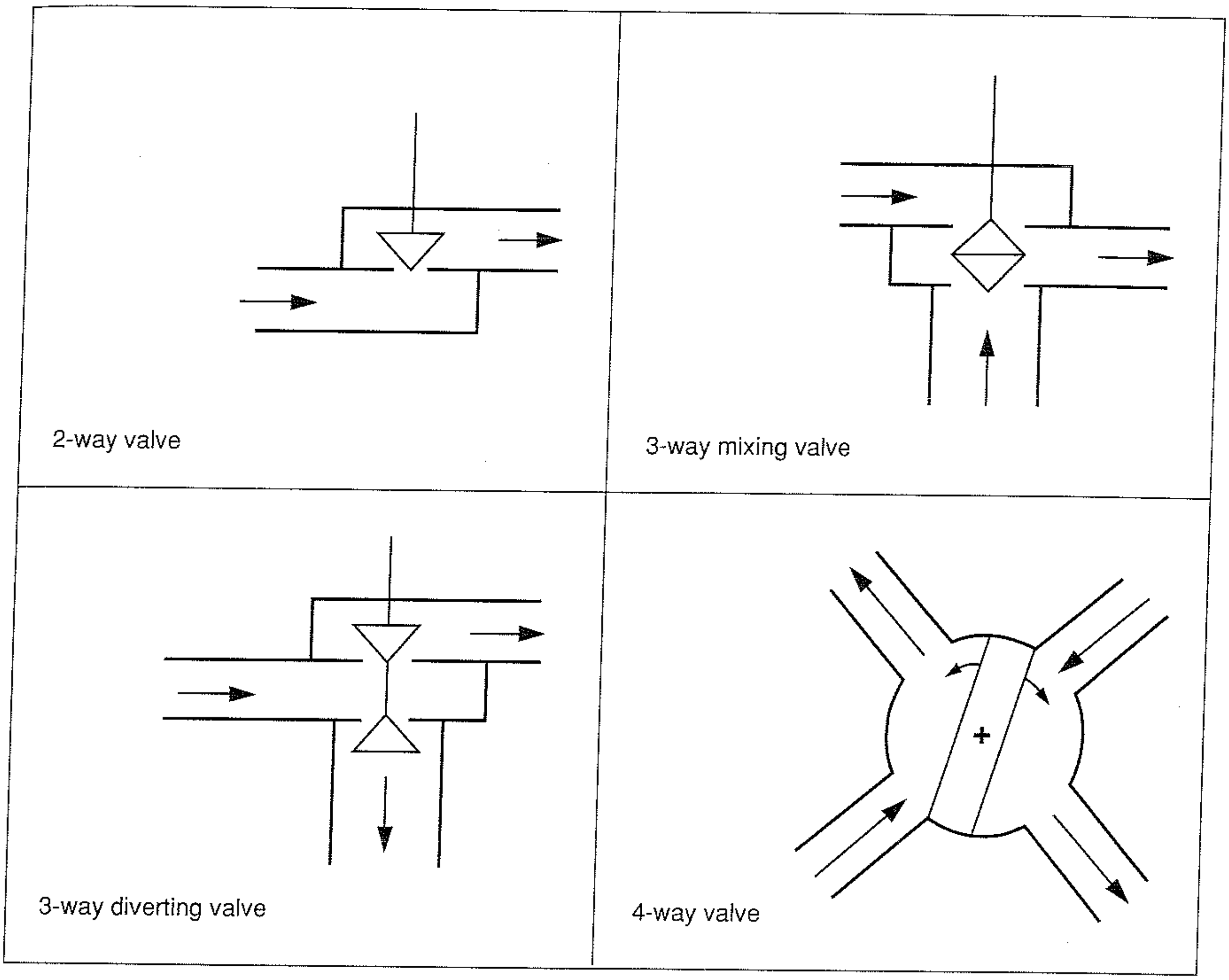


Fig 2.1

In HVAC applications valves with the following types of water paths are commonly used:



These water paths are generated using:

- plug valves
- rotating disk and sleeve valves
- piston valves
- gate valves
- ball valves
- diaphragm valves
- butterfly valves

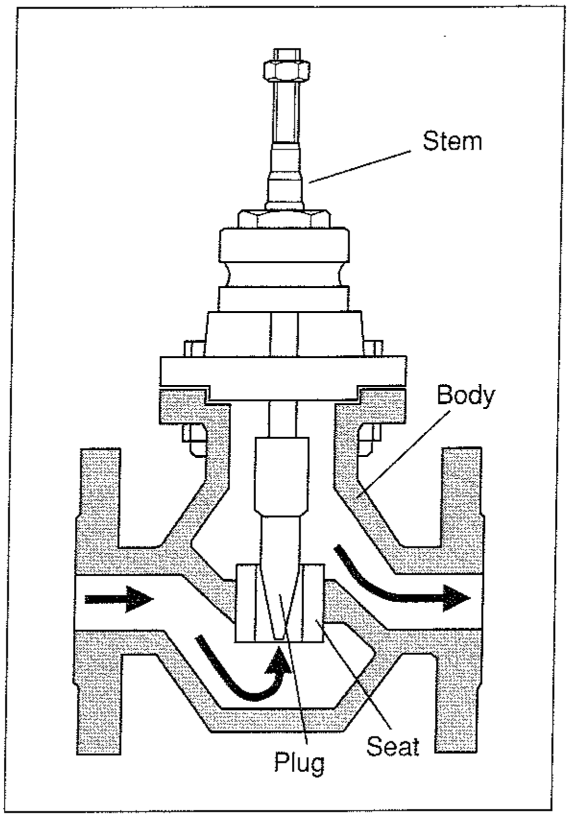


Fig 2.2 2-way valve

Plug valves

The 2-way plug valve is the most common type of control valve. It is designed to control and to shut off the flow. This type of valve is available in two designs, *single seated* and *double seated.*

Single-seated valves are used in cases where a tight seal is required. To ensure correct operation, the flow through the valve must be in the direction indicated on the valve body. As a rule, the direction of flow is such that the medium flows towards the plug. If the direction of flow is reversed vibration and noise can result.

For a new valve, the leakage flow through a closed valve will normally be less than 0.05% of the maximum flow, at the prevailing pressure drop.

When the pressure drop across the valve is high, large 2-way valves require a considerable actuating force. To reduce the necessary actuating force, such valves are more or less pressure-compensated, i.e. they are designed so that the force on the plug, caused by the pressure drop across the valve, is almost entirely eliminated. Figs. 2.3 and 2.4 show typical compensated valves.

The double-seated valve has two plugs, which are influenced by the water pressure. Opposing forces act on the plugs, balancing each other, whereby the required actuating force is minimized.

The leakage of a double-seated valve does normally not exceed 0.5% of the maximum flow. Even if both plugs are ground in, together with their respective seats, double-seated valves always have a higher leakage than single-seated valves. This is due to the differing coefficients of expansion of the valve stem and valve body. This means that when one of the plugs mates against its seat then there will still be a small gap between the other plug and seat.

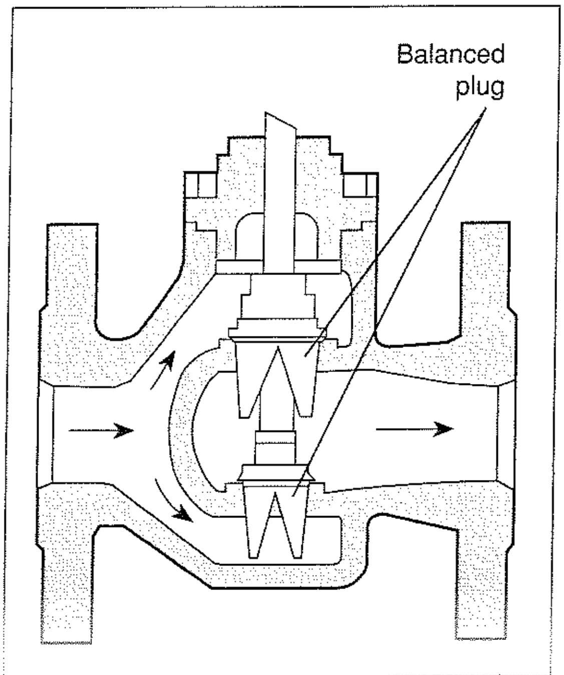


Fig 2.3 Double-seated valve

The effect of the pressure drop can be cancelled by applying the water pressure to the upper side of the plug as well. See Fig. 2.4. A seal is used to seal the upper part of the plug towards the valve body. The valve can be completely compensated and the actuating force will depend only on the friction caused by the seals.

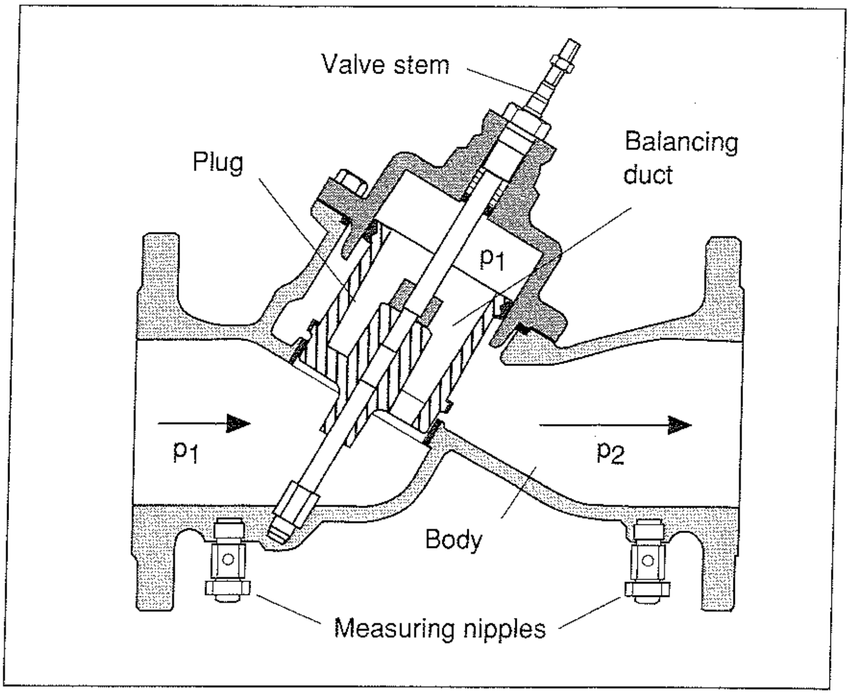


Fig 2.4 Balanced valve

Piston valves

Piston valves have two concentric cylinders, with the outer cylinder (sleeve) attached to the valve body and the inner cylinder operated by the valve stem.

As the inner cylinder moves upwards, holes in the outer cylinder will gradually be exposed, so that a water path is formed. Different characteristics can be obtained by varying the design and position of these holes.

Piston valves are always pressure compensated and require relatively little actuating force (but the friction of the seals must always be overcome).

This type of valve is normally used in district heating installations, where very high pressure drops occur.

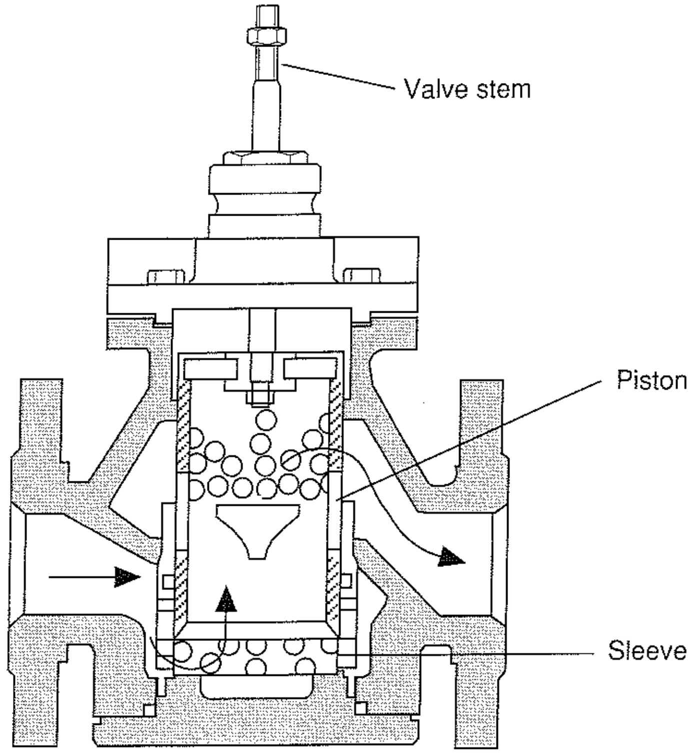


Fig 2.5 Piston valve (V260, V295)

Butterfly valves

This type of valve has a disk which governs the flow. The body is designed so that the valve can be clamped between standard pipe flanges. Butterfly valves are of simple design and require little space. The simplest type of butterfly valve has a flat disk of a diameter slightly less than the internal diameter of the valve body.

This type of valve does not seal completely. Typical leakage values vary, from 0.5% to 2% of the maximum flow. The seal can be improved by fitting a resilient packing around the disk, or by lining the inside of the valve body with rubber or both. Butterfly valves with such packings require greater actuating force than similar valves without packings. Such valves can seal as efficiently as plug valves.

When closed, the effect of the water pressure on the disk is in balance. However, the actuator must be capable of overcoming the friction forces in the valve stem bearings and the close-off friction.

When a butterfly valve opens, a certain degree of imbalance is introduced as the medium flows around the disk

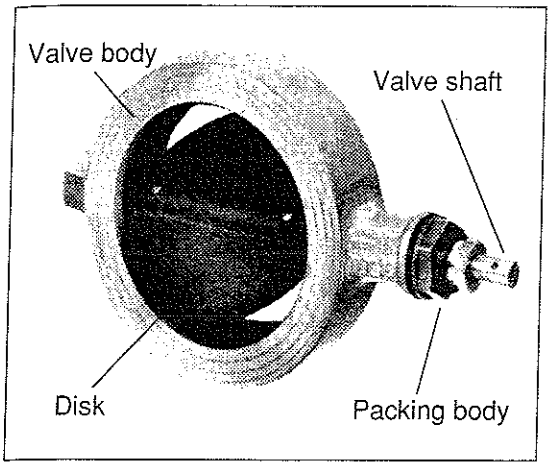


Fig 2.6 Butterfly valve

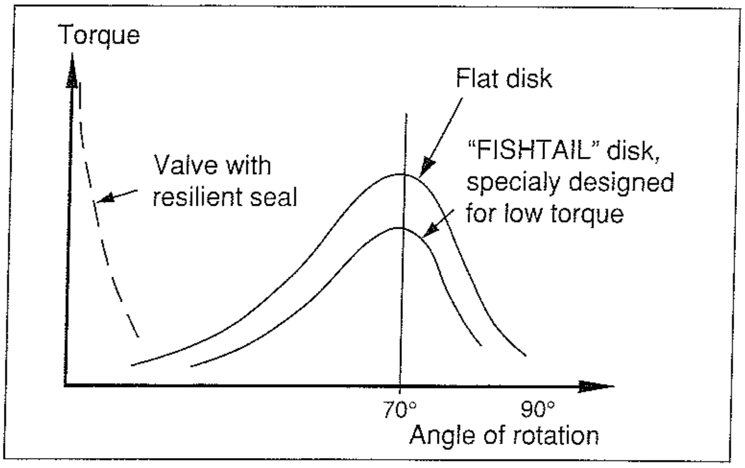


Fig 2.7 Torque curve of butterfly valve

(turbulent flow). The maximum imbalance occurs when the disk has rotated through about 70°. However, this imbalance can be almost completely eliminated, by means of a specially designed disk.

Fig. 2.9 shows the flow characteristic of a normal butterfly valve.

If two butterfly valves are connected together by linkages, they can be used as a 3-way valve (Fig. 2.8). Such configurations are typically used to control large flows, for example from boilers. The two valves can also be controlled by actuators, operating in opposition.

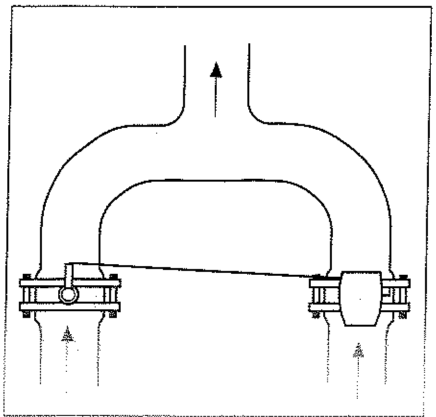


Fig 2.8 Linkage rod arrangement

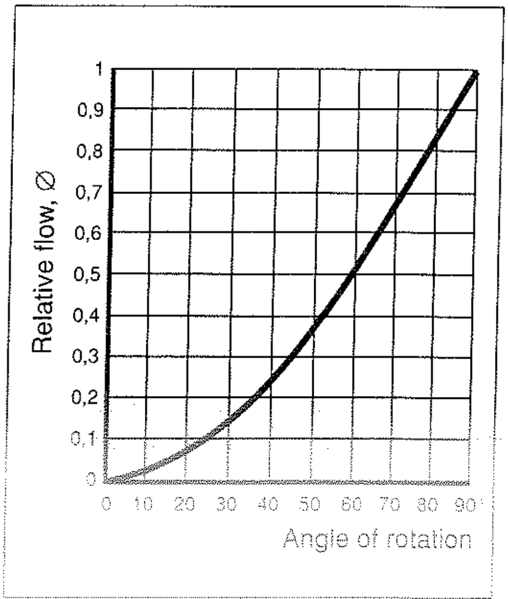


Fig 2.9 Flow characteristic

Split-range valves

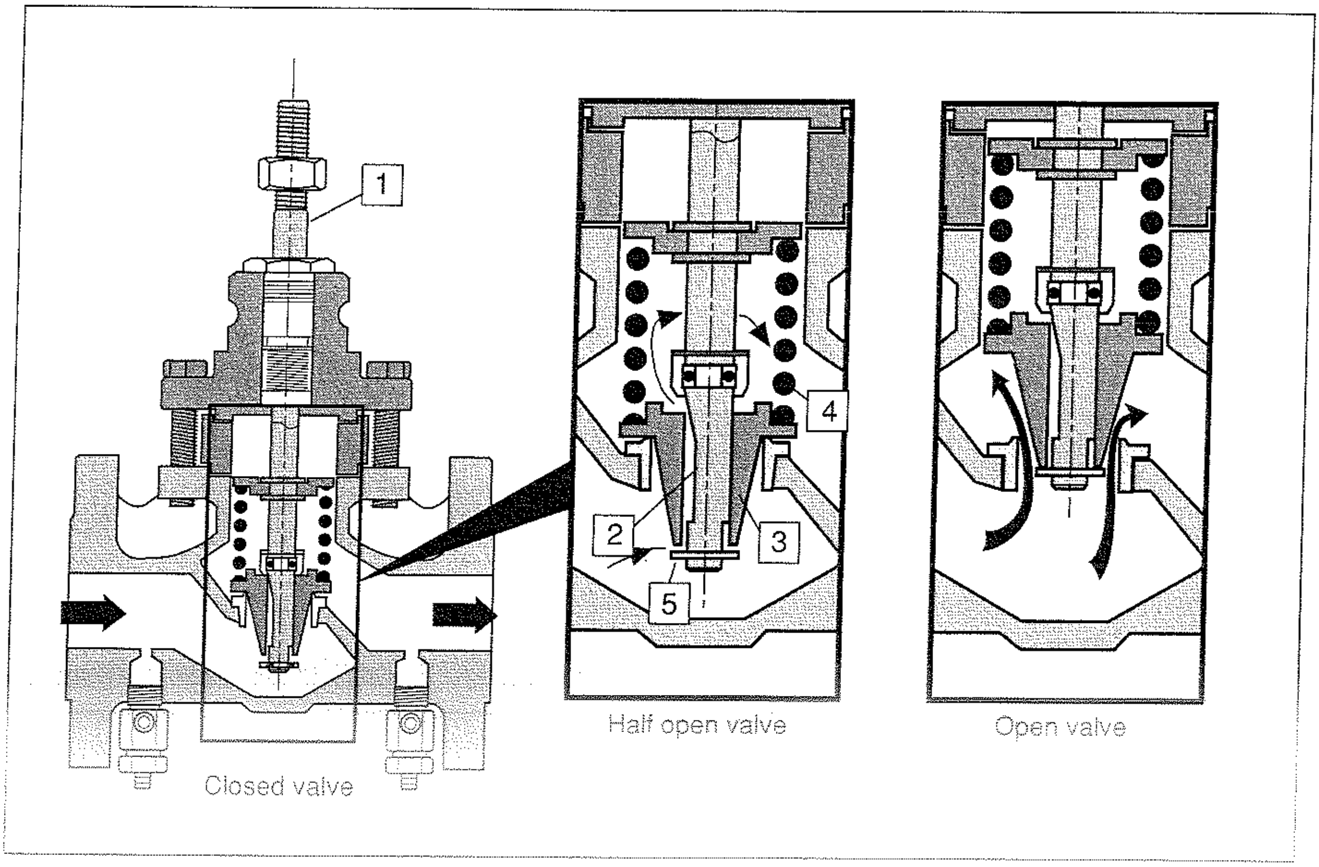
The split-range valve is a development of the plug valve. The split-range valve has two plugs, mounted on the same stem, which handle different flow ranges and which act sequentially, as the valve stem moves from one end of its travel to the other.

The valve stem (1) is shaped as a control plug. The larger plug (3) is pressed against its seat by means of the spring (4).

As the valve stem (1) moves upwards, the flow is throttled by the small plug (2). When the valve stem has been lifted so much that the washer (5) contacts the plug (3), the latter starts to move upwards and the flow is now controlled by the lift of the larger plug (3).

This principle gives the valve a rangeability which is about 10 times greater than that of a conventional valve. This means that a split-range valve is much better at controlling major load changes and over-sized systems.

Fig 2.10 Split-Range valves



Mixing and diverting valves

3-way valves are used for mixing or diverting two flows. Mixing valves are much more commonly used than diverting valves, about 90% of all 3-way valves are mixing valves in some cases. At low pressure drops (less than 30 kPa), mixing valves can be used as diverting valves.

NOTE: Using a *balanced valve* as a mixing valve can cause disturbances in the system. The valve tends to close quickly just before it has reached the fully closed position. When the water mass in the piping system comes to a rapid stop, annoying noise can be generated which is propagated throughout the building. This problem predominantly exists for valve sizes larger than DN50.

When planning new installations, mixing valves rather than diverting valves should be specified.

The plugs of mixing and diverting valves are designed differently. See Fig. 2.11. As in the case of 2-way valves, the direction of flow should always be ***towards*** the plug.

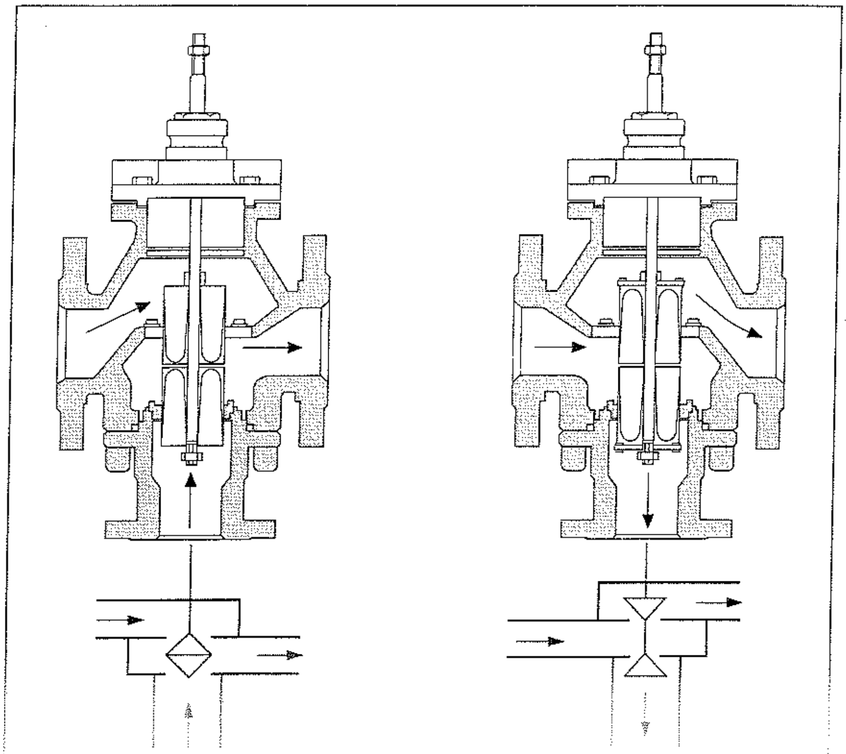


Fig 2.11
3-way valves

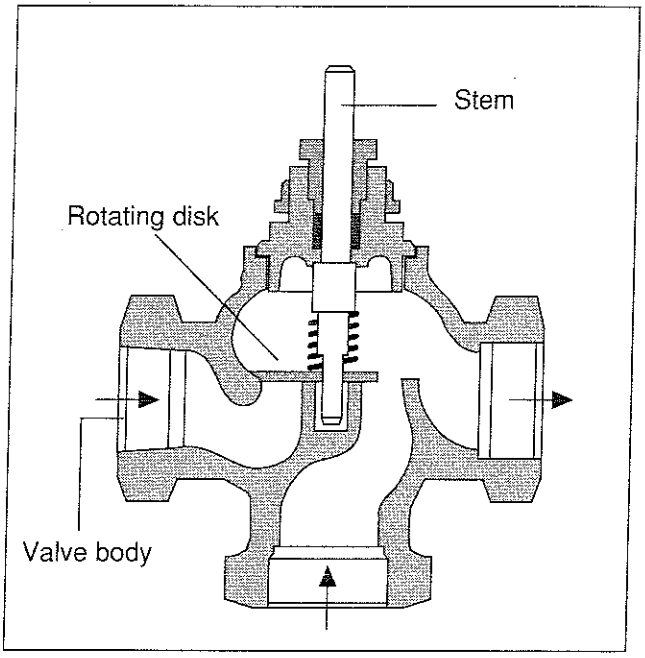


Fig 2.12 Disk valve (VTRA)

Disk and sleeve valves

Disk valves have a rotatable disk, which is held against a flat seat by a spring, or solid stem.

As the disk rotates, it cuts off a greater or lesser part of the flow. The angle of rotation between the closed and fully open positions normally is either 90° or 180°. When used as a 2-way valve, the direction of flow should be such that the water presses the disk against the seat, to ensure the best seal.

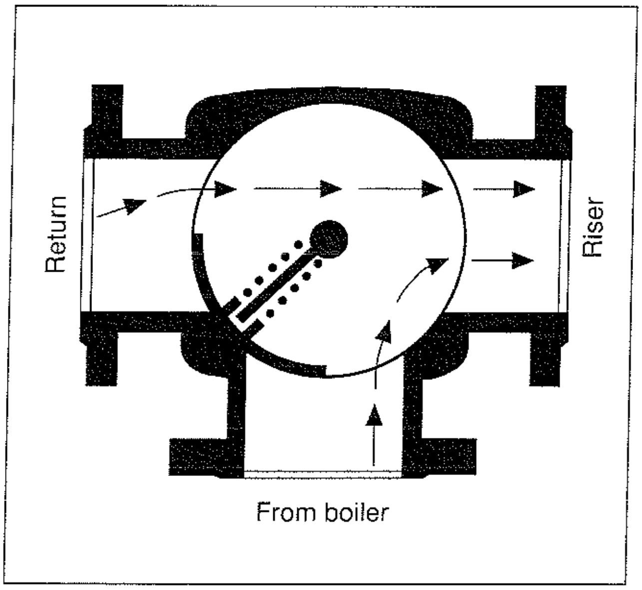


Fig 2.13 Rotating sleeve valve

A rotating sleeve valve consists of a rotatable sleeve which presses against the circular-shaped inside surface of the body of the valve. As the sleeve is rotated, flow paths are opened and closed, respectively.

3-way rotating disk and sleeve valves can usually be used as either mixing or diverting valves. These valves are relatively inexpensive but the leakage in the fully closed position is greater than that of single-seated valves. For this reason, they have largely been replaced by single-seated plug valves in applications demanding 2-way valves. However, 3-way rotating disk and sleeve valves are still used in 3-way applications in heating systems where the pressure drop across the valve is small.

Balancing valves

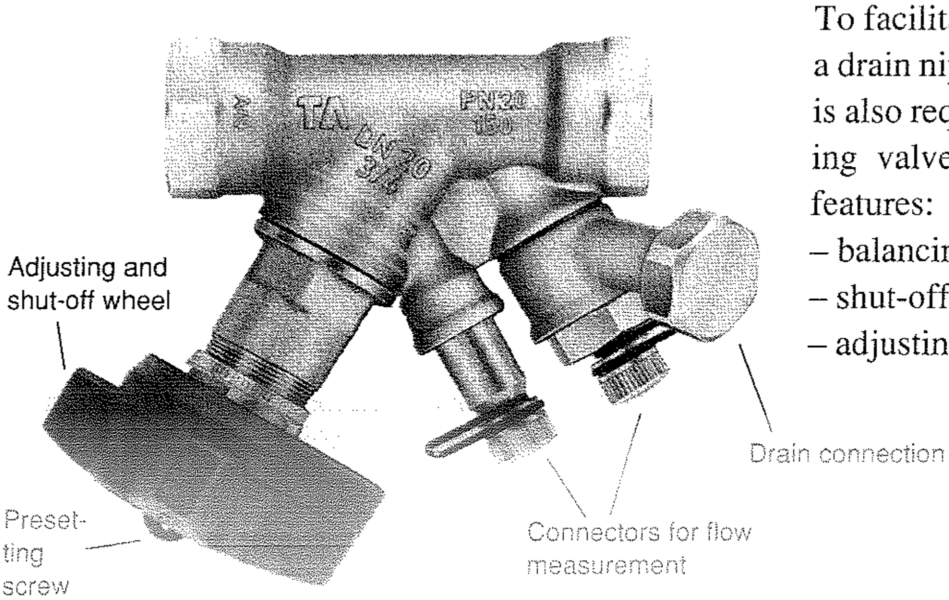
The most common control task is to control the energy so that the desired indoor climate is obtained in the various spaces throughout a building. To ensure that the correct amount of energy is supplied to each space, the flows in the various control circuits of the heating system must be *balanced*. Balancing of the heating system is a simple, effective operation which quickly produces tangible results in the form of energy savings, which often can be considerable.

Balancing is achieved by installing *balancing valves* in each group line of a heating plant. Balancing valves are designed to provide a fixed flow (set manually during installation) in branch lines and main lines. A balancing valve should also be installed in each riser of a heating installation. This makes it possible to achieve the desired distribution (balance) between the different risers.

A **balancing valve** is a plug valve, with a bronze or cast iron body. A hand wheel and scale facilitate setting the specified flow resistance, e.g. as specified in the installation drawings.

This type of valve has measuring nipples for measuring the pressure drop across the valve. The correct water flow can easily be determined from the pressure drop across the valve, using a pressure-drop chart, or an instrument which directly displays the flow.

Fig 2.14 Balancing valve



To facilitate the draining of risers, a drain nipple with hose connector is also required. A modern balancing valve has the following six features:

- balancing
- shut-off
- adjusting
- draining
- flow measurement
- presetting.

Gate valves, plug valves, ball valves

Gate valves, plug valves and ball valves are manually-operated shut-off valves. Gate valves and ball valves are designed so that a free flow is obtained when the valve is fully open. They seal completely in the fully closed position.

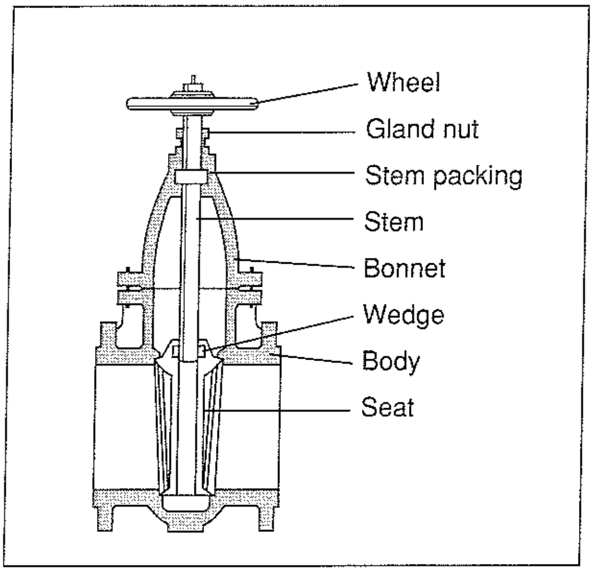


Fig 2.15 Wedge gate valve

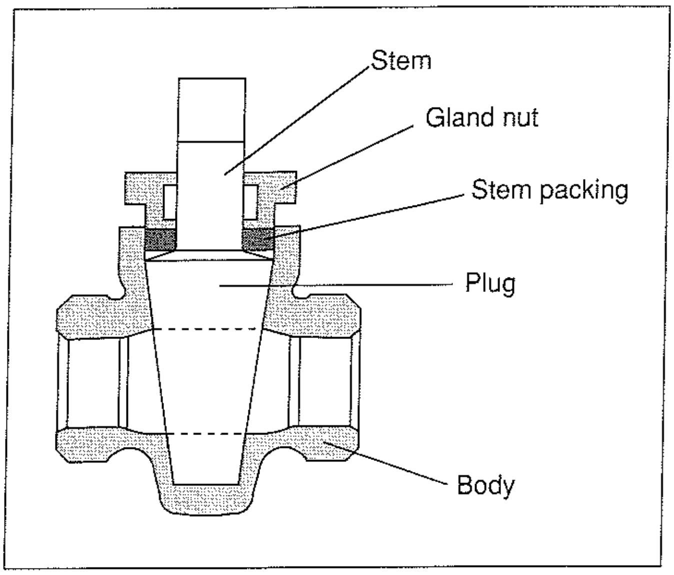


Fig 2.16 Plug valve

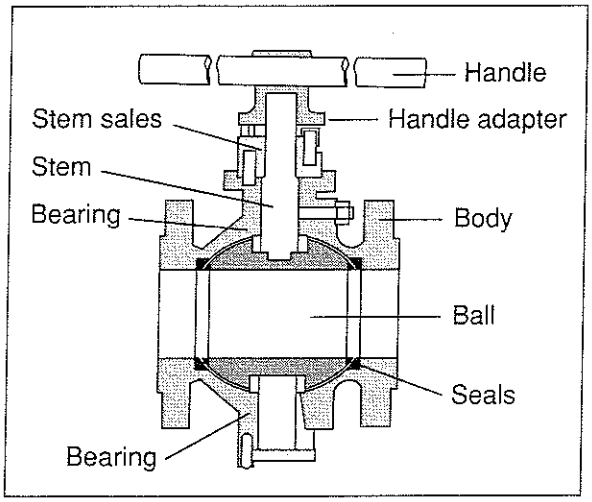


Fig 2.17 Ball valve

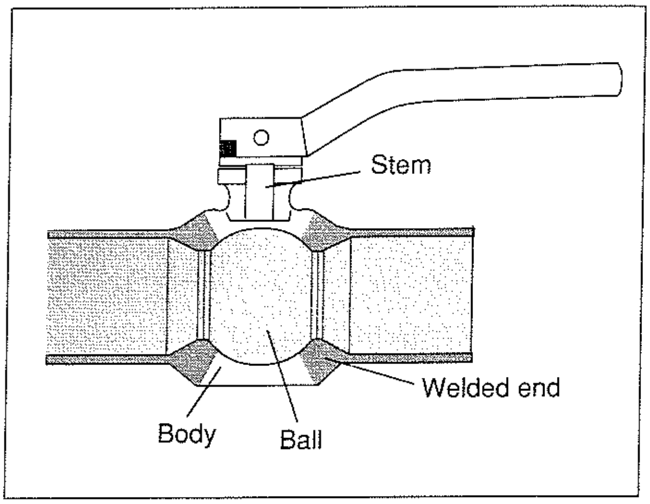


Fig 2.18 Ball valve with welded ends

Diaphragm valves

Diaphragm valves (also called *Saunders valves*) are used for viscous, corrosive and liquids containing suspended solids.

Instead of a plug and seat, diaphragm valves have a flexible diaphragm, which moves towards a ridge in the valve body. These valves seal well and have a small pressure drop when open. The diaphragm can be moved by means of the valve stem or by changes in pressure (e.g. of air or water) on the outside of the diaphragm. As this type of valve has a poor control characteristic, it is seldom used for flow control.

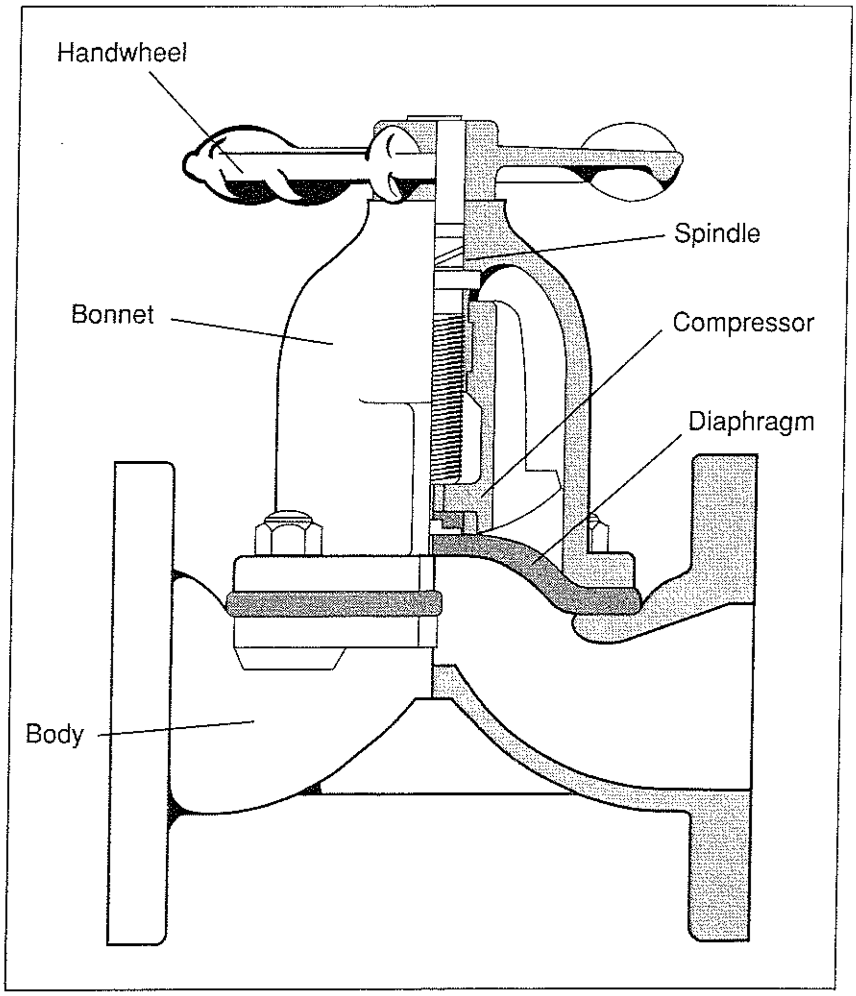


Fig 2.19 Diaphragm valve

3 Control valves

Valve body

The valve body normally comprises a sand casting of cast iron, ductile (nodular) iron, steel, brass or bronze.

The choice of material for the pressurized parts of a valve is determined primarily by the qualities of the medium:

– pressure
– temperature
– corrosiveness
– erosive characteristics

However, manufacturing cost is often the decisive factor. It is sometimes necessary to compromise on the choice of materials. Cast iron is used for valves in heating systems.

As cast iron is more brittle than ductile iron or steel, the prevailing temperatures and pressures impose strict limits on the use of this material. Cast iron valve bodies are less expensive than bodies of ductile iron or steel.

The corrosion resistance characteristics of cast iron are satisfactory for use in heating systems. Moreover, anti-corrosion additives normally are added to the water in heating systems.

Valves with bodies of bronze, brass or special alloys, are used in domestic hot water systems.

Brass contains about 58% copper and 42% zinc.

Bronze contains about 85% copper, 5% zinc, 5% tin and 5% lead. From the standpoint of resistance to corrosion, brass alloys with a high zinc content are inferior to bronze. Brass exhibits satisfactory resistance to neutral solutions but, in acidic liquids, dezincification occurs (the zinc is leached out) leaving a brittle copper skeleton. Brass alloys are prone to dezincification especially when installed in fresh water systems.

Normal bronze alloys, used for valves and couplings, cannot be hot pressed or die cast. AMETAL® is TA's own, patented copper alloy, developed especially for valves and couplings. AMETAL® can be hot pressed and is used for valve parts, which are in contact with heating and cooling mediums.

AMETAL® valve bodies are economically attractive and offer long life, even in aggressive media.

Typical standardized limiting temperatures for various valve body materials. The temperatures can vary from country to country.

Material	Designation	Max. temp. °C	Max. temp. °F	Min. temp. °C	Min. temp. °F
Grey cast iron	SS 14-0120 SS 14-0125	120 150	248 302	+3	+37
Ductile iron	SS 07 17-02 GGG 40	300	572	–20	–4
Bronze		225	437	–20	4
Brass		185	365		

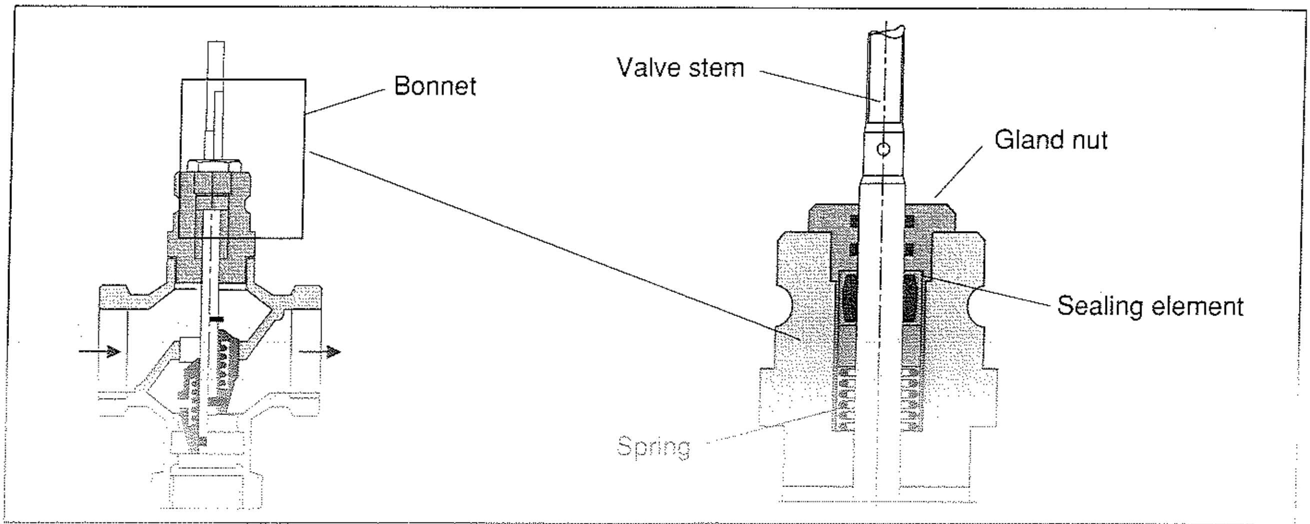
Valve stems

The valve stem transmits the motion of the actuator or hand wheel to the plug.

In plug valves, the stem motion is axial, whereas it is rotational in disk valves, ball valves and butterfly valves. Stem motion may be both axial and rotational in plug valves and gate valves. A variety of designs exist.

Valve stems are made of highly polished, stainless steel or brass.

Fig 3.1



Stem packing

The valve body has a packing box, which prevents the medium from leaking past the stem of the valve. The stem packing comprises a packing and a gland, which presses the packing against the packing box and the valve stem. The gland is tightened with a gland nut. The gland and gland nut are often the same part.

Lubrication of the valve stem with the correct grade of grease will prolong the life of the packing.

The choice of packing type and the design of the packing box are dictated by the motion of the valve stem (rotational or axial), its rate of travel, type of operation (continuous or intermittent) and the pressure and temperature range and type of medium.

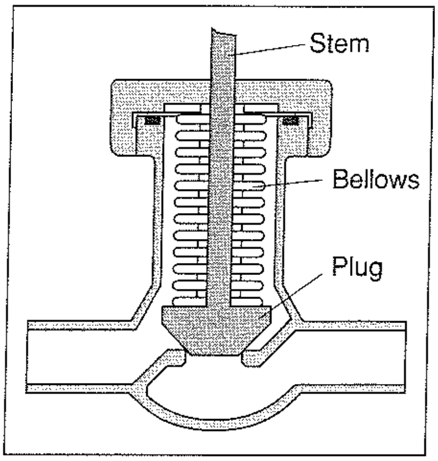


Fig 3.2 Valve with bellows seal. In cases where a total seal is required, a metal bellows is used. Such bellows are usually made of stainless steel or tombak (a copper alloy). To ensure a satisfactory bellows life, the length of the bellows should be about three times as long as the stroke of the valve. This makes for a costly design.

The design of stem packings differs for control valves and shutoff valves. In comparison with control valves, shutoff valves are operated relatively seldom during the life of the valve. If, when operating a shutoff valve, leakage occurs at the stem, it can frequently be cured by tightening the gland nut. Control valves often operate for long periods, without any attention.

Instead of requiring adjustment of the gland nut, some designs have a spring, which automatically tightens the packing, as it wears or becomes compressed.

In principle, the life of the packing is exceeded when the leakage rate reaches some specified maximum value. In some cases, the packing can be adjusted but in others it must be replaced.

In the HVAC field, the medium to be sealed against normally comprises hot water or steam. In the event of leakage through the packing box, the water will normally evaporate outside the packing box. Water and steam will leave crystalline deposits on the valve stem. These crystals are hard and are firmly bonded to the valve stem. When a stem coated with crystals passes the packing, axial grooves will be formed in the packing, which will cause further leakage.

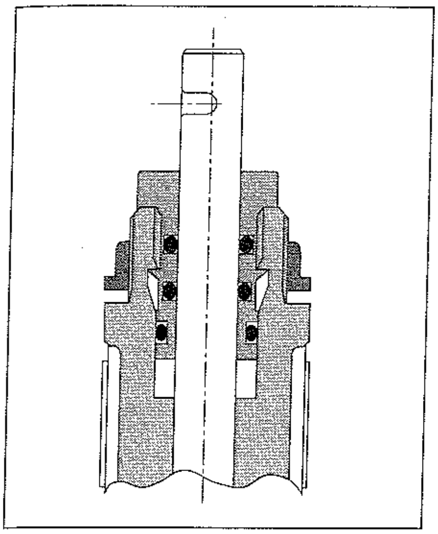


Fig 3.3 Valve with O-ring seal

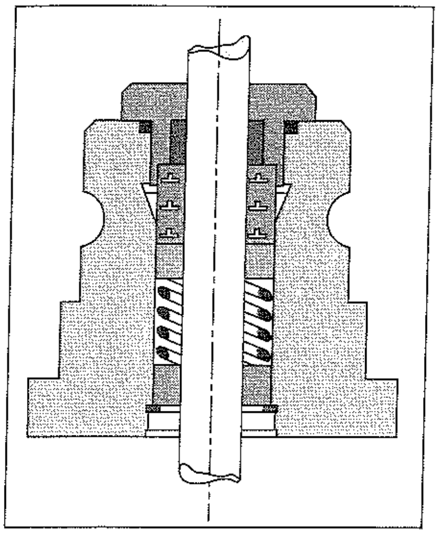


Fig 3.4 Valve with V-ring seal

In heat recovery systems and certain types of refrigeration systems, the medium is normally a water-glycol mixture or brine (water containing about 30% sodium chloride, NaCl, or 30% calcium chloride, $CaCl_2$), to prevent the water from freezing. The surface tension of glycol is lower than that of water, which means that such a mixture will leak past a standard packing. As the water is cold, there is little evaporation, which causes the leak to appear to be more serious than is actually the case. Heavy deposits of glycol, sodium sulphate or calcium sulphate may form around the packing box. Special packing boxes are used for these media.

On ships, for instance, oil is used as a heating medium. Oil is also difficult to seal against completely.

Packing types

O-rings of rubber make effective, inexpensive packings. The rubber quality selected will depend on the composition of the medium, the temperature range and the planned life. Good qualities in one respect are often combined with poor qualities in another. If the rubber, for example, can withstand high temperatures, it may have a poor resiliency. In HVAC valves, O-rings of EPDM rubber are commonly used. EPDM rubber is NOT suitable for use with oils.

Teflon can withstand temperatures in the range –200 to +200°C (–330 to +400°F) and it remains unaffected by chemicals. Its resistance to wear varies considerably, from extremely poor to good. In itself, it is less elastic than rubber. To ensure continuous pressure on the valve stem, springs are normally fitted to tighten the packing as it becomes compressed or worn. The packing is designed in the form of V-rings, which compress each other, through a spring. It is vital to the function of the packing box that the bearings of the valve stem are correctly designed, so that axial play is eliminated. The V-ring packing box works well and is the most common type of packing box in industrial and HVAC applications. Axial grooves on stems with reciprocating motion cause leakage.

Packing boxes for various media

Summary of packing boxes, used in TA valves

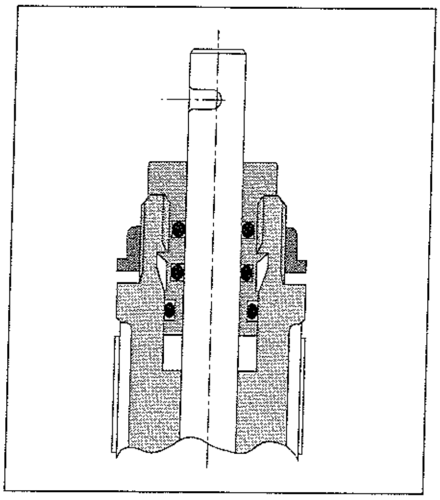


Fig 3.5

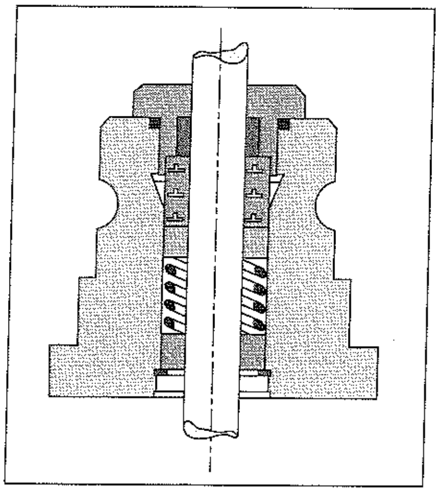


Fig 3.6

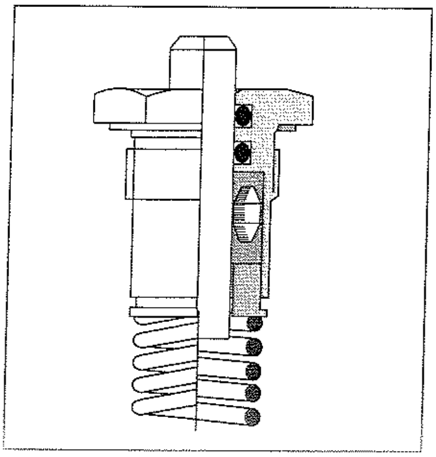


Fig 3.7

Packing box for valves with rotating stems, e.g. butterfly and disk valves

Figur 3.5
Medium: Water-glycol, max. 50% glycol (not oil)
Temperature range: –10° to +160°C (14°F to 320°F)
Packing material: EPDM, O-rings

Packing boxes for valves with rising stems, e.g. plug valves

Figur 3.6
Medium: Water and steam
Max temperature: 180° C (375°F) water, 180° C (375°F) steam
Packing material: Teflon rings, spring loaded

Figur 3.7
Medium: Water + glycol
Temperature range: –10°C to +30°C (14°F to 85°F)
Packing material: SI-WAX cartridge with two O-rings

Packing boxes for low temperatures

In refrigerating systems (in which the control valve operates below freezing point), moisture in the air condenses and freezes on the bodies and packing boxes of valves.

Normally, the pipes and valve body are insulated, to prevent the formation of ice. To prevent the valve stem from freezing, the valve is fitted with a heating element. The heating element heats the stem, so that no ice can form on it.

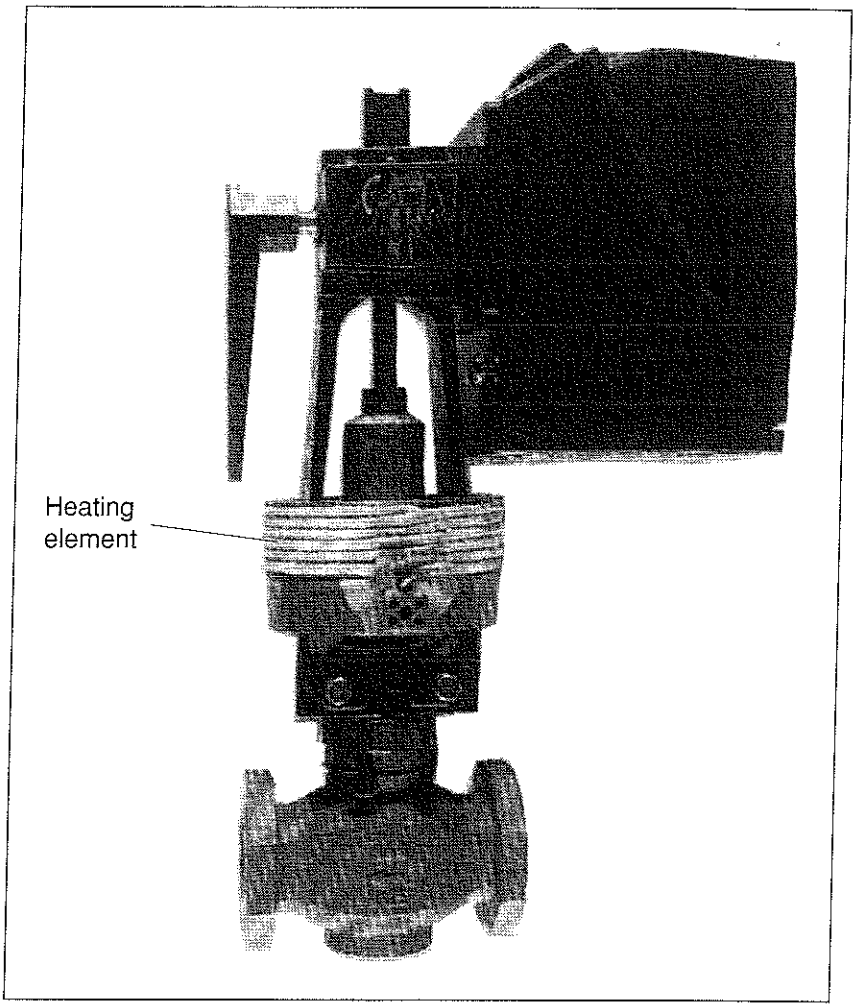


Fig 3.8 Valve with heating element

Valve trim

The trim of a valve is made up by the plug and the seat.

In conjunction with the valve seat, the plug varies the flow through the valve. The valve plug is connected to the actuator or hand wheel by the valve stem. In plug-style globe valves, the plug moves axially towards the valve seat.

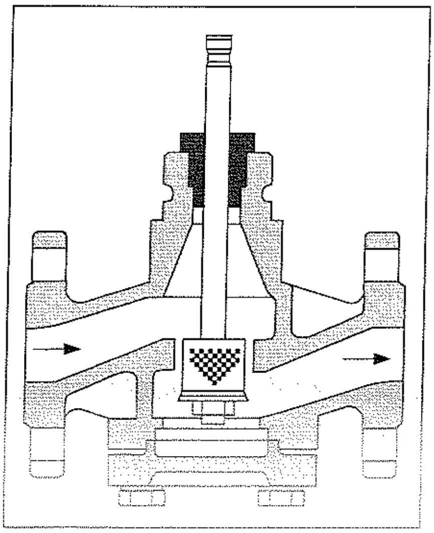


Fig 3.9

In 2-way valves designed for large pressure drops (from 100 kPa to about 800 kPa – about 14 psi to 110 psi), the plug and seat are often made of stainless steel and exhibit symmetry of rotation. If the valve seat is long, the water will accelerate smoothly, to reach the highest velocity where the flow area is smallest. To prevent large plugs from vibrating, they are often provided with some form of mechanical support. In cases where the pressure drop across the valve is low, the plugs are bronze castings. The plugs are designed, to be guided by the valve seat. One common type of plug is cylindrical (see *piston valve* in

Part 2 and also Fig. 3.9) with a large number of holes. As the plug moves in relation to the seat, the holes are gradually exposed and the water flow through the valve will gradually increase.

Rotating disk valves have a spring loaded disk, which presses against the seat. The disk may also be shaped as a rotationally symmetrical body, which fits closely into the valve body. Leakage can be reduced, by making the valve body and plug slightly tapered and spring loaded.

To ensure a good seal, the valve seat and plug are ground together. Valves for pressure drops of up to about 350 kPa (about 50 psi) are often equipped with a soft packing, to minimize leakage. When new, the maximum leakage of a single-seated valve is about 0.05% of the full flow. The maximum leakage is about 0.5%, in the case of double-seated valves.

Large pressure drops across a valve cause high flow rates. To prevent erosion of the plug and seat, the maximum permissible pressure drop across the valve is determined by the materials of the plug and seat.

To avoid vibration and cavitation in valves, the pressure drop can be distributed over several stages. See the section on "Cavitation" in Part 4.

Maximum pressure drop across valve plug and seat

Cast iron:	Pressure drop	about 150 kPa
Bronze, plastic, rubber	"	about 300 kPa
Stainless steel	"	800 to 1000 kPa
Stellite-coated plug and seat	"	greater than 1000 kPa

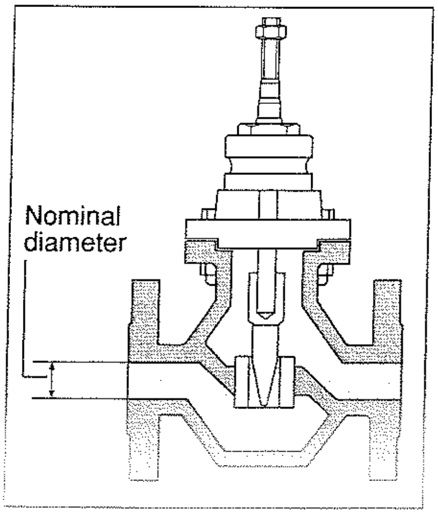


Fig 3.10

Valve size, DN

Valve size is specified as the nominal diameter (DN). The nominal internal diameter is the diameter, in millimeters, of the size of the pipe and pipe components (valves).

The numerical DN value usually corresponds to the size, in mm, of the internal diameter of the end connections.

Example: DN 32 corresponds to a 1-1/4 inch valve, in the Imperial or US system.

Pipe size

DN end connection numbers, with corresponding sizes for threaded steel pipes and copper pipes.

Table of dimensions for threaded steel pipes

Nominal size	External dia.	Average wall thickness SS 326 Wall thickness	Large wall thickness SS 327 Wall thickness
DN	mm	mm	mm
6	10.2	2.0	2.65
8	13.5	2.35	2.9
10	17.2	2.35	2.9
15	21.3	2.65	3.25
20	26.9	2.65	3.25
25	33.7	3.25	4.05
32	42.4	3.25	4.05
40	48.3	3.25	4.05
50	60.3	3.65	4.5
65	76.1	3.65	4.5
80	88.9	4.05	4.85
100	114.3	4.5	5.4
125	139.7	4.85	5.4
150	165.1	4.85	5.4

Table of dimensions, straight copper pipes, hardened. Length 5.5 m

External dia. mm	Wall thickness mm	External dia. mm	Wall thickness mm
6	0,8	15	1,2
8	0,8	18	1,0
8	1	22	1,0
10	0,8	22	1,5
12	1,0	28	1,2
15	1,0		

Excerpt from Standard SMS 1889-1890

Nominal pressure

(PRESSURE RATING), PN
PN = Pressure Nominal

The nominal pressure is the internal pressure, expressed in bar (1 bar = 10^5 Pa = 100 kPa), which forms the basis for the stress calculation for the valve, at 20°C (68°F). The nominal pressure is designated PN, followed by a numerical value.

Example: PN 16; maximum internal pressure = 1.6 MPa (16 bar).

The earlier designation was NT 16.

PN values follow the series: 2.5, 6, 10, 16, 20, 25, 40, 50, 64, 150, 400.

Delivery tests

Unless otherwise required, all valves are pressure and leakage tested, in accordance with SS IEC 534, prior to delivery.

Pressure test

The valve body is tested at a water pressure of 1.5 x PN.

Leakage, closed valve

The leakage between the plug and seat is tested with water, at a temperature of 20°C and at a pressure drop of between 100 and 500 kPa.

During delivery testing, it is ascertained that the leakage does not exceed the following values:

- plug valves, single-seated: 0.05% of K_V
- plug valves, double-seated: 0.5% of K_V
- pressure-compensated plug and piston valves: 0.1% of K_V
- disk valves: 0.5% of K_V

End styles

Valves are connected to pipes by means of the following types of end connections:

- threaded
- flanged
- clamping between flanges
- welded

The valve, which is a part of the pipe line, will be exposed to forces and moments, which occur in the pipe line as a result of over- and underpressure, thermal expansion and dead load. This loading – especially at high temperatures and pressures – is considerable and it degrades valve operation, through deformation.

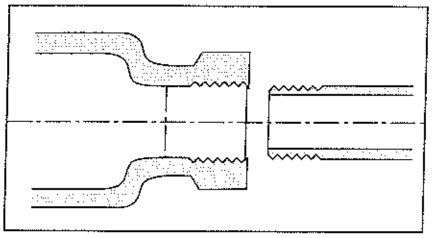


Fig. 3:11

Screwed end connections

Screwed connections are used for valve dimensions of up to DN 50. The valve body can have an internal or external thread.

In Europe, internally threaded valves have threads according to ISO/R7. Straight threads correspond to SMS 36 or BSP. NPT (tapered) threads are used in the USA and Canada.

In the case of valves with internal threads, pipes are screwed directly into the body. In the case of externally threaded bodies, a swivel coupling is used to connect the valve and pipe. The swivel coupling greatly facilitates the replacement of valves.

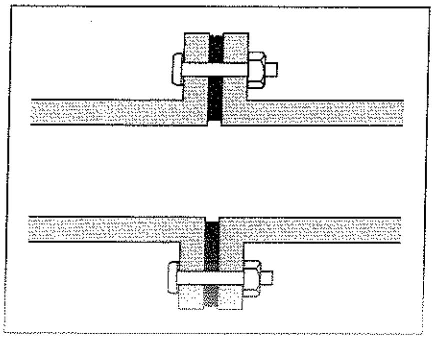


Fig. 3:12

Flanged connections

Flanged connections can be used for valves of size DN 15 and up. The dimensions of flanges depend on the pressure rating and the material of the valve body.

Bolt holes are drilled according to ISO/R2084.

Recommended pipe flanges

Nominal pressure		6	10	25
PN			16	40
Weld flange	DIN	2631	2633	2635
with collar	SMS	2031	2033	2035
Blind flange	DIN	2527	2527	2527

Clamping between flanges

Butterfly valves are designed to be clamped between standard valve flanges. This type of fitting is only suitable for relatively short valves, such as small 2-way valves and check valves. Where long pipe sections are installed, this type of fitting becomes less capable of absorbing the torque and shear forces encountered in major pipework.

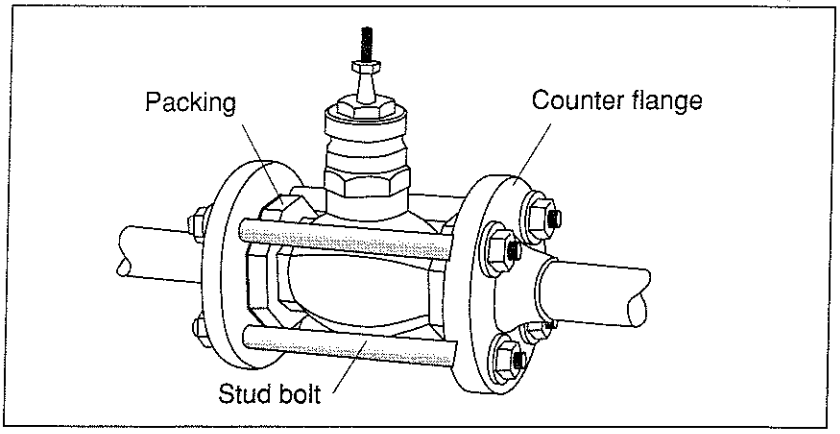


Fig. 3:13

Welded valve fittings

Shutoff valves, such as ball valves, often are available with weld ends. These are welded directly to the pipes.

NOTE: To avoid risking to damage the valve when welding, it should be cooled, for example by wrapping it with wet rags.

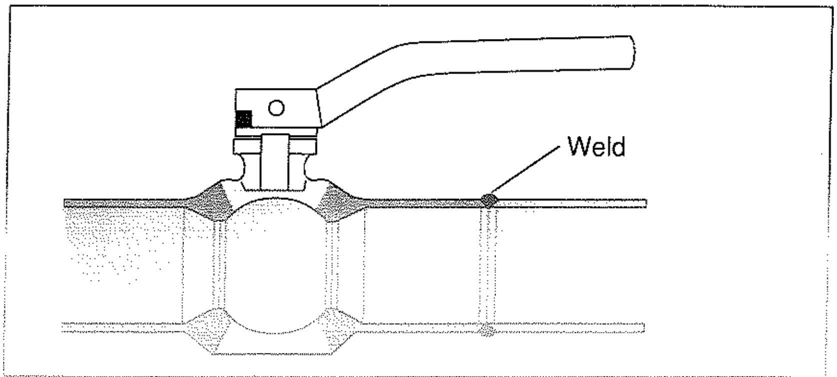


Fig. 3:14

4 Flow characteristics of control valves

Flow coefficients, A_v, K_v, C_v

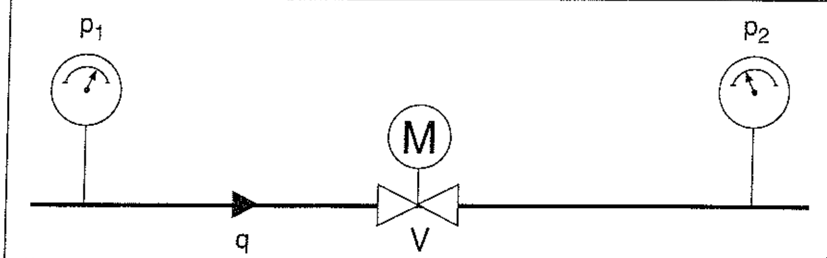


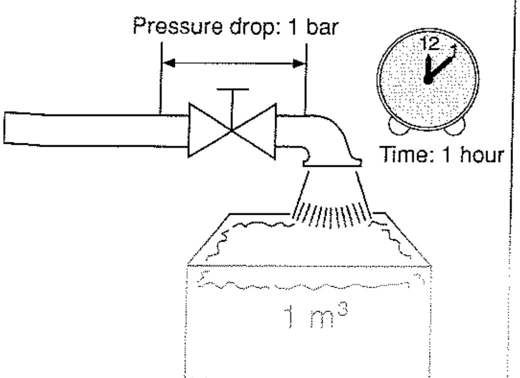
Fig 4.1

When a valve is installed in a pipe, its resistance to the flow will cause a pressure drop which is directly proportional to the square of the flow. Doubling flow rate increases pressure drop four times, tripling it increases pressure drop nine times, etc.

$$\Delta p = \xi \cdot \rho \cdot \frac{v^2}{2g}$$

$\Delta p = p_1 - p_2$ = pressure drop across valve
ξ = coefficient of resistance (dimensionless)
ρ = density of liquid
v = velocity of liquid

Valve flow coefficient



As a measure of the capacity of a valve the *flow coefficient*, K_v, is used, rather than the flow itself. The flow coefficient expresses the flow through the valve in m^3/h at a pressure drop of 1 bar.

K_v is calculated using the following formula:

$$K_v = \frac{q \cdot \sqrt{\rho}}{\sqrt{\Delta p}}$$

K_V = flow coefficient
q = flow, m^3/hr
$\Delta p = p_1 - p_2$, pressure drop across the valve, bar (= kp/cm^2)
ρ = density of the medium in kg/dm^3 (= 1 for water at 15°C)

Another flow coefficient, which is based on the SI system, is the A_V coefficient. One disadvantage of the A_V coefficient is that awkwardly small numbers are obtained.

$$A_V = \frac{q \cdot \sqrt{\rho}}{\sqrt{\Delta p}}$$

A_V = flow coefficient, m^2
q = flow, m^3/s
Δp = pressure drop across valve in Pascal, Pa
ρ = liquid density, kg/m^3

Conversion to the K_V coefficient is done by means of the following formula:

$$K_V = A_V \cdot 3.6 \cdot 10^4$$

In the process industries, the American valve coefficient, C_V, is frequently used. It is based on the U.S. system of units.

$$C_V = \frac{q}{\sqrt{\Delta p}}$$

C_V = flow coefficient
q = flow, US gallons per min
Δp = pressure drop, lbs. per sq.in. (psi)

The K_V coefficient is converted to C_V with the following formula:

$$C_V = 1.17 \cdot K_V$$

Example

Calculate required K_V for a valve with a water flow of 3 m^3/hr at a pressure drop across the valve of 1.5 bar.

$$K_V = \frac{3}{\sqrt{1.5}} = 2.5$$

Select $K_V = 2.5$.

Most European countries use K_V to denote the calculated requisite flow coefficient for valves.

All valves have a capacity value (the K_V value) shown for a fully open valve (stem lift, h = 100%). The K_V value is shown on the tag on the valve.

The K_V coefficients of a valve series are selected according to the Reynard series, i.e.

1.0   1.6   2.5   4.0   6.3   10   16 ... etc.

Example

Flow = 3 m^3/h, at pressure drop = 2 bar

$$K_V = \frac{3}{\sqrt{2}} = 2.12$$

$K_V = 2.12 - 20\% = 1.7$
$K_V = 2.12 + 40\% = 2.97$

Select $K_V = 2.5$

According to the Swedish standard for HVAC installations, VVS-AMA 83, the following applies for valves.

"The K_V coefficients specified by the manufacturer for control valves, must not deviate from the nominal K_V coefficients by more than –20% to +40%."

Pressure drop diagrams

To simplify the calculation procedure *pressure drop diagrams* are used. These diagrams show the relationship between flow and pressure drop. The axis are graduated logarithmically, which means that each K_V coefficient is represented by a straight line.

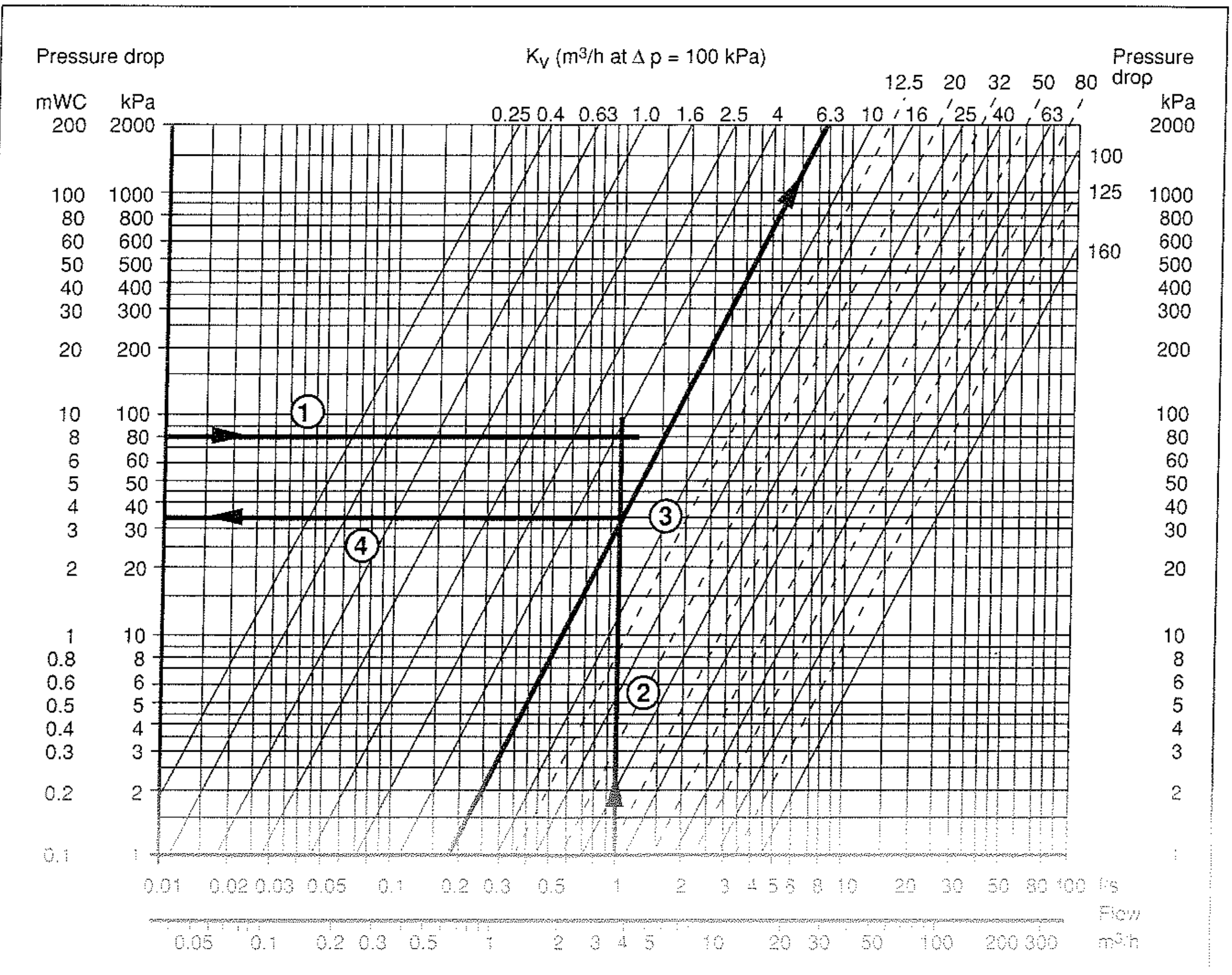
K_V coefficients can be calculated from the curve as follows:

1. Plot a horizontal line through the design value for pressure drop across the valve.

2. Plot a vertical line through the design flow through the valve.

3. Plot a line downwards, from the point at which the lines of 1 and 2 cut, to the nearest K_V line, and read off the K_V coefficient.

Fig 4.2 Pressure drop diagram for water



4. Plot a new horizontal line through this point. The pressure drop at the K_V coefficient can now be read off.

Using this method, the next larger K_V coefficient should be chosen, which can sometimes result in over-sizing. When performing accurate calculations, therefore, always check with the K_V formula that the selected K_V coefficient lies in the interval −20% to +40% of the K_V coefficient.

Pressure drop curves are easy to use for the calculation of changes in flow and pressure drop, for a given valve.

K_V of several, combined valves

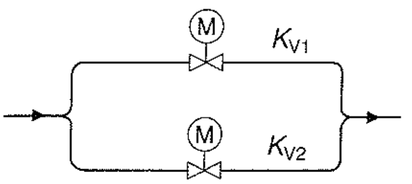


Fig 4.3

When two or more valves are connected in parallel, the resulting valve coefficient is the sum of the coefficients of the individual valves.

$$K_V = K_{V1} + K_{V2} + K_V ..$$

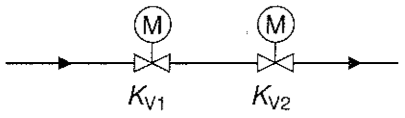


Fig 4.4

When two valves are connected in series, calculation of the resulting coefficient becomes slightly more complicated:

$$\frac{1}{(K_V)^2} = \frac{1}{(K_{V1})^2} + \frac{1}{(K_{V2})^2}$$

or

$$K_V = \frac{K_{V1} \cdot K_{V2}}{\sqrt{(K_{V1})^2 + (K_{V2})^2}}$$

Example

Two valves with $K_V = 2$ and 3, respectively, are connected in series. Calculate the resulting K_V coefficient.

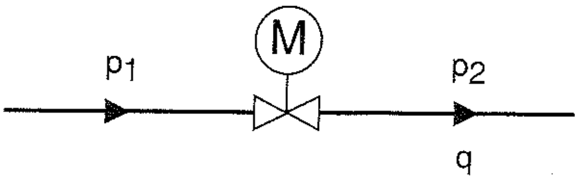
$$K_V = \frac{2 \cdot 3}{\sqrt{2^2 \cdot 3^2}} = 1.7$$

Flow characteristic

The "flow characteristic" of a valve denotes the ratio between the flow, q, recalculated for a constant pressure drop across the valve, and the position of the valve stem, h.

In the case of 3-way valves, this applies to each port.

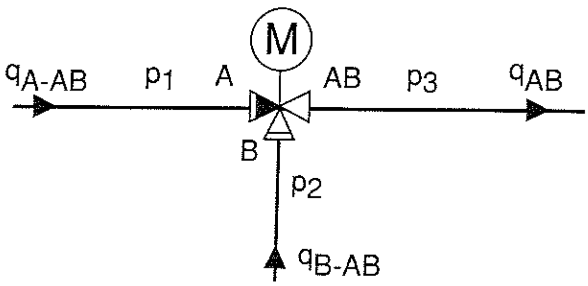
Two flow characteristics are standardized, *linear* and *equal percentage* characteristics, according to IEC534-1.



$$q = f(h)$$

$$p_1 - p_2 = \text{constant}$$

Fig 4.5 2-way valve



$$q_{A\text{-}AB} = f(h)$$

$$p_1 - p_3 = \text{constant}$$

$$q_{B\text{-}AB} = f(1\text{-}h)$$

$$p_2 - p_3 = \text{constant}$$

Fig 4.6 3-way valve

Linear flow characteristic

A *linear* flow characteristic is characterized by a theoretically equal increase in the stroke, h, corresponding to an equal change in the flow, q.

Expressed mathematically:
$q = q_0 + m \cdot h$, where:

q_0 = relative flow, at $h = 0$

m = the angle of the slope of the linear characteristic, i.e. the ratio between the increase in the flow coefficient and the increase in the relative travel.

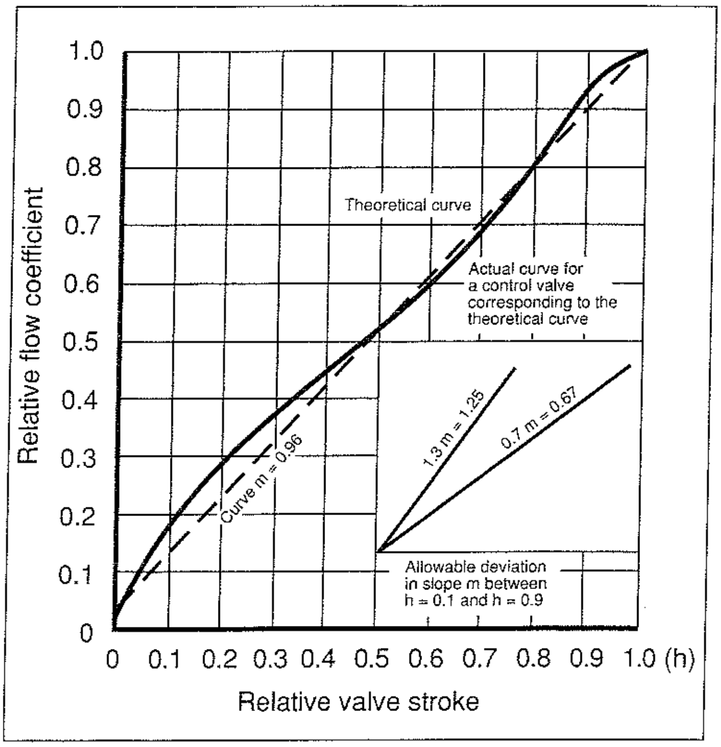


Fig 4.7 Linear flow characteristic

Equal percentage (logarithmic) flow characteristic

An equal percentage (logarithmic) characteristic denotes that an equal change in the stroke gives an equal percentual change in the flow coefficient.

Expressed mathematically :
$q = q_0 \cdot e^{nh}$, where:

q_0 = the relative flow coefficient, at $h = 0$

n = slope of the equal percentage (logarithmic) flow characteristic, when plotted with a logarithmic axis for q and a linear axis for h.

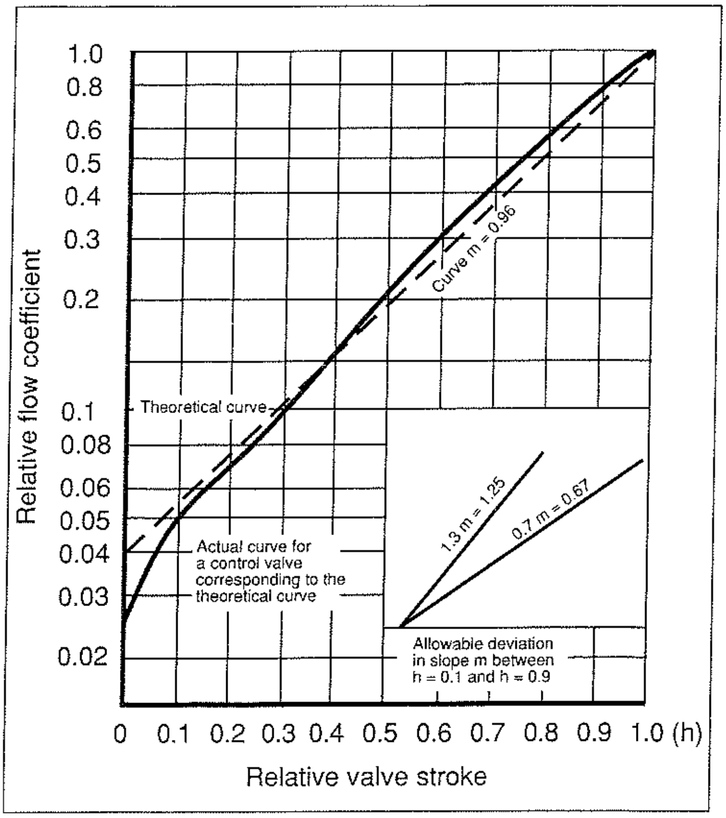


Fig 4.8 Equal percentage (Eq%, logarithmic) characteristic

The deviation of the characteristics of valves designated linear and equal percentage, respectively, from the theoretical curve must not exceed the tolerances specified in SS IEC 534–1, shown in Fig. 4.9.

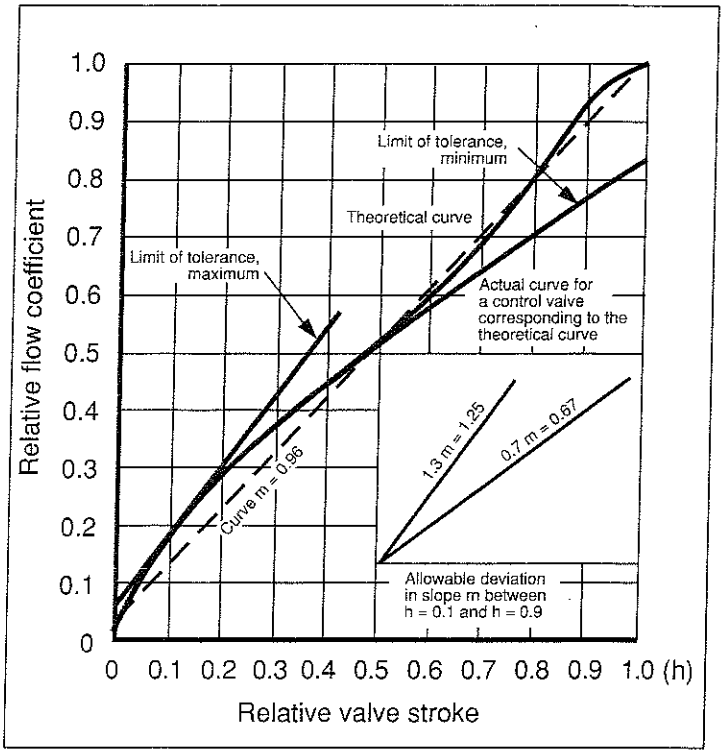


Fig 4.9 Linear flow characteristic according to IEC 534-1

Other valve characteristics

The flow characteristics of valves can also follow laws other than the above.

Quadratic characteristic

A *quadratic* flow characteristic is sometimes used. This means that the flow coefficient follows a square law, relative to the stroke. This characteristic constitutes a compromise between linear and equal percentage characteristics. It is not internationally standardized.

Quadratic characteristics are common in HVAC valves.

Quick-opening characteristic

This characteristic is characterized by a rapid change in the flow coefficient, as the valve is first opened, which diminishes as the valve approaches the fully open position.

This characteristic is *not* used for control valves, but may be encountered in shut-off valves.

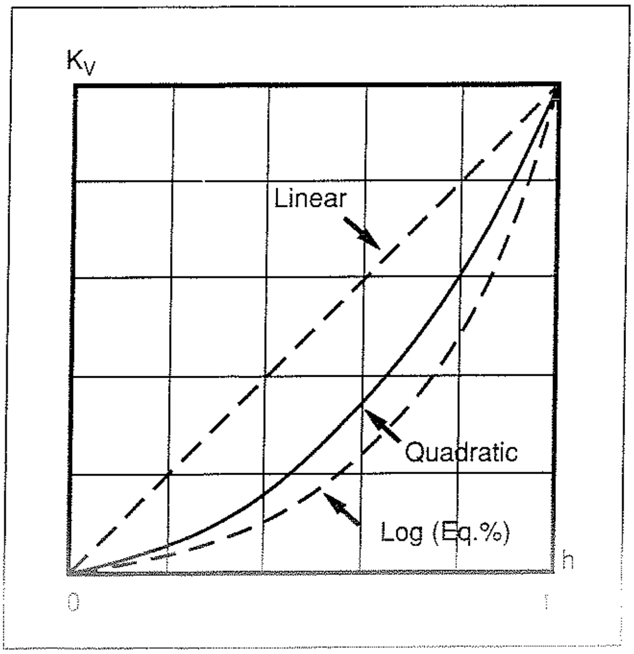


Fig 4.10 Linear, quadratic and equal percentage flow characteristics

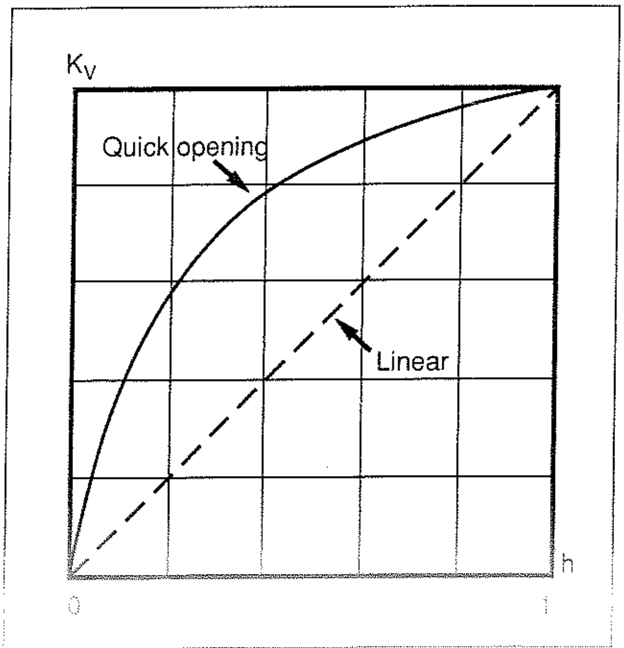


Fig 4.11 QUICK-OPENING characteristic

Flow characteristics of 3-way valves

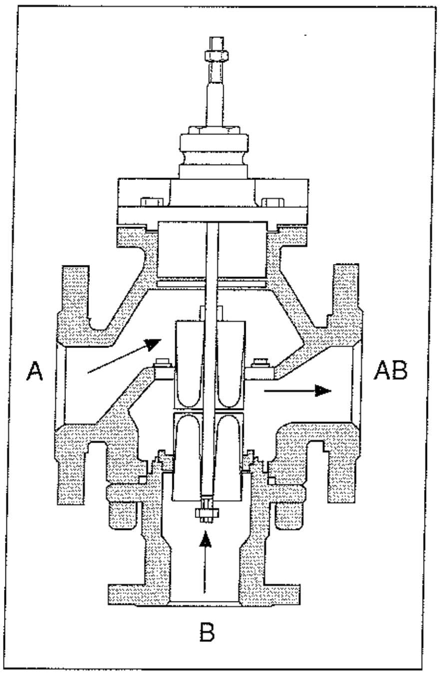


Fig 4.12
3-way mixing valve

As there are two controlled flows through a 3-way valve, such valves have two characteristics; one for the control path and one for the bypass path. If these two characteristics are identical, the valve is said to be symmetrical and if they are different, it is asymmetrical.

Symmetrical valves have the advantage that it is not necessary to keep track of which port is the control port and which is the bypass port, since their characteristics are identical.

In cases where the valve has a non-linear characteristic, there will be quite large variations in the common flow, throughout the operating range of the valve.

Fig's 4.13 and 4.14 show the inherent characteristics of a symmetrical, linear and a symmetrical equal percentage, Eq%, (logarithmic) 3-way valve. As can be seen, the common flow (A + B) of the Eq% valve varies considerably.

This disadvantage can be eliminated, or at least greatly reduced, by making the valve asymmetrical. See Fig's 4.15 and 4.16.

Figs. 4.15 and 4.16 show the characteristic of equal percentage-linear and modified-linear valves. The latter gives a constant flow from the common port, AB, provided the valve is installed according to certain prerequisites.

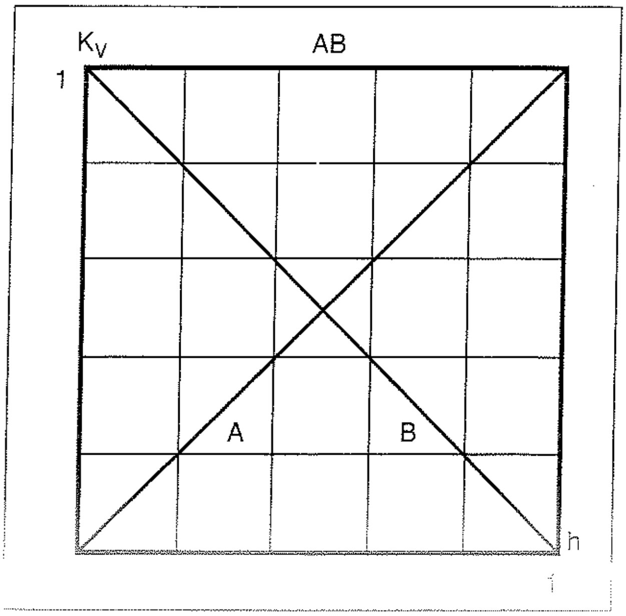


Fig 4.13 Symmetrical linear flow characteristic

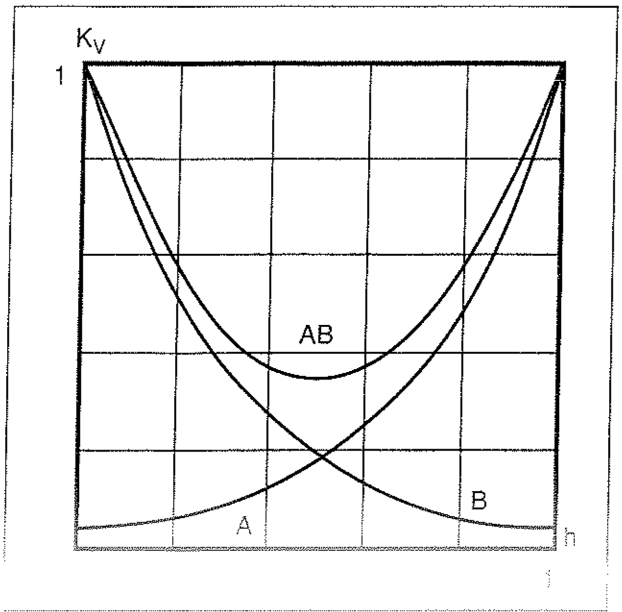


Fig 4.14 Symmetrical equal percentage flow characteristic, valid when

$$\frac{\Delta p \text{ control valve}}{\text{Pump head}} = 1$$

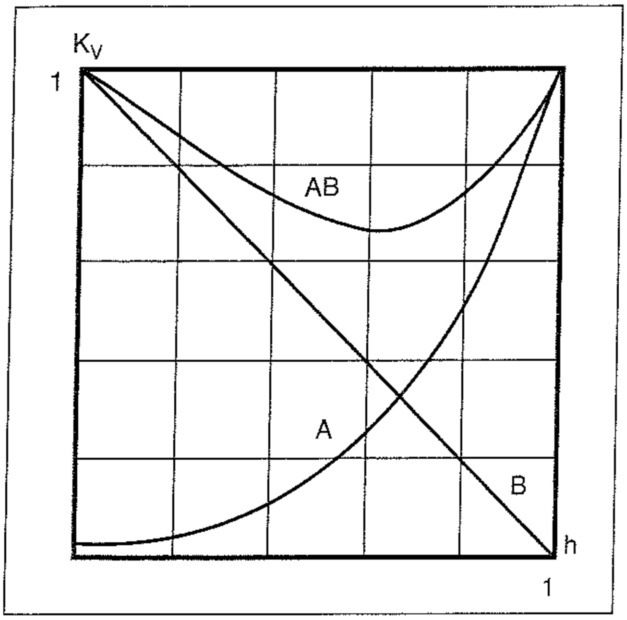


Fig 4.15 Equal percentage-linear flow characteristic

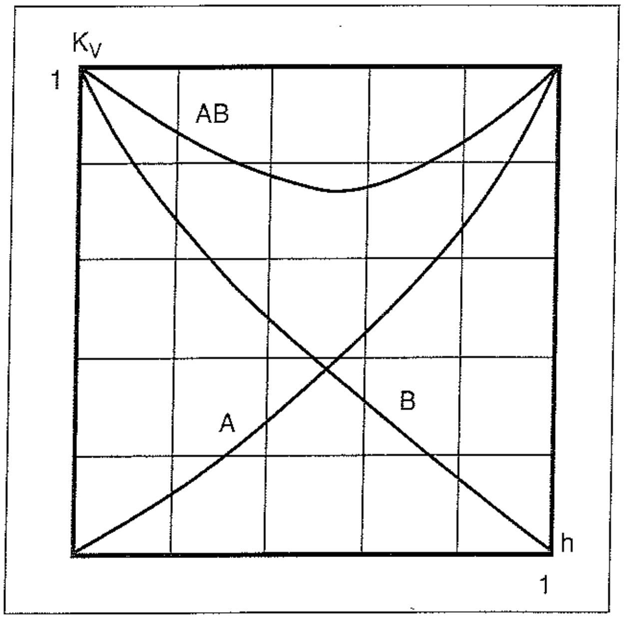


Fig 4.16 Equal percentage-complementary flow characteristic

However, it is not possible to determine the most suitable characteristic merely by studying the characteristic of a particular valve. The pressure drops occurring elsewhere in the system must also be considered. Total flow q_{AB} depends mainly on the ratio $\alpha = \frac{\Delta p \text{ control valve}}{\text{Pump head}}$.

For $\alpha = 1$ $\quad q_{AB\ min} = 40\,\%$

$\alpha = 0{,}25$ $\quad q_{AB\ min} = 69\%$

$\alpha = 0{,}05$ $\quad q_{AB\ min} = 91\%$

This is described in the sections on "Valve authority".

The control and bypass ports of asymmetrical valves must not be confused, as this will result in disastrously poor control characteristics.

Valve authority

The flow characteristic (i.e., *inherent* characteristic) of a control valve is defined for a constant pressure drop across the valve. When a valve is installed in a pipe system, in which the flow varies, the pressure drop across each component will also vary, as the pressure is proportional to the square of the flow. When the pressure drop across the valve varies, the flow characteristic will not be the same as the (theoretical) characteristic of the valve.

Fig. 4.17 shows a circuit with a heat exchanger, a control valve, and a balancing valve. This installation is assumed to be part of a larger network, in which the pressure drop remains unchanged, whether the valve is open or closed. The diagram shows how the pressure drop in the circuit varies.

The following extreme states are shown:

– the control valve is closed. The flow is, therefore, zero. The entire pressure drop takes place across the valve.

– the control valve is fully open. The pressure drop across the valve is now at a minimum. There is now a specific pressure drop across each component in the circuit.

Figure 4.18 shows how the valve's share of the pressure drop changes as the stem lift (*h*) is varied. This valve was selected, so that its share of the total pressure drop is 50%, when fully open.

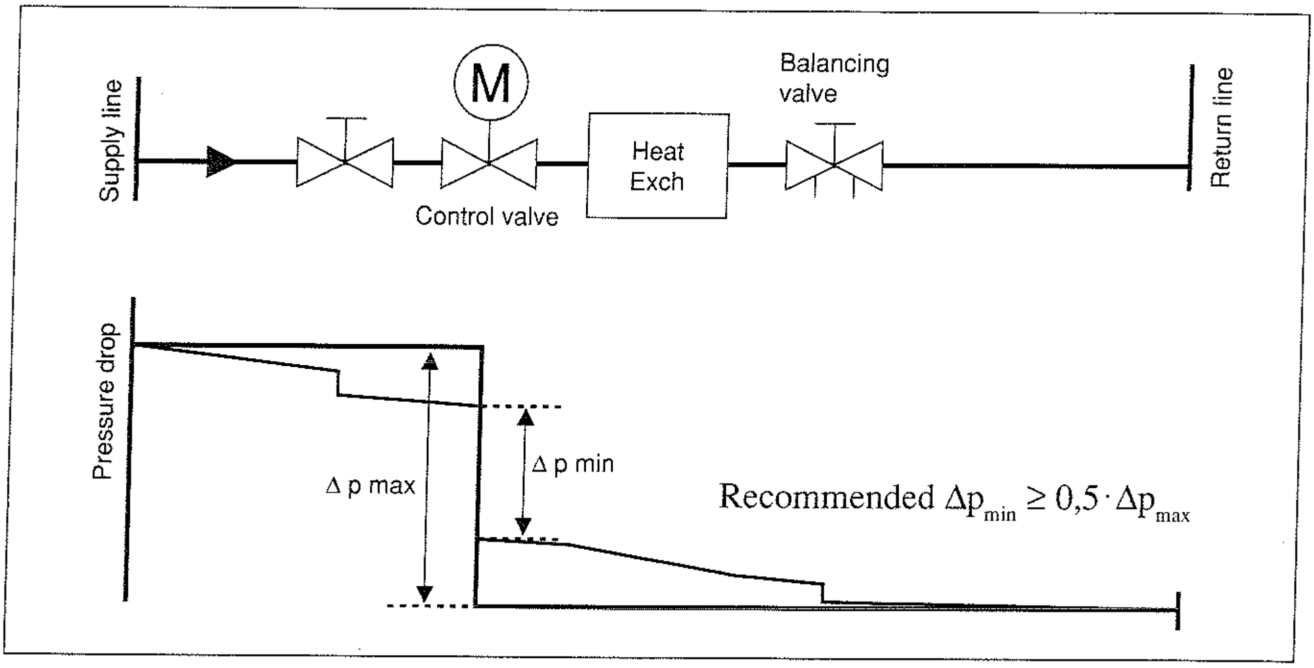


Fig 4.17

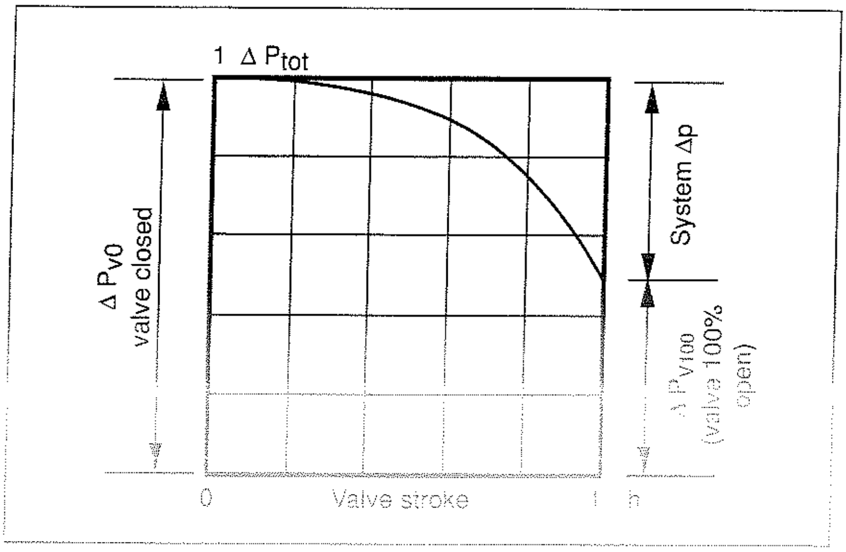


Fig 4.18

The "VALVE AUTHORITY" (β) concept is used to express in a simple way a control valve's share of the pressure drop in relation to the total pressure drop.

The valve authority, β, is defined as the quotient of the pressure drop across the valve, (ΔP_{V100}), and the pressure drop across the entire circuit affected by the valve, ΔP_{tot}, when the valve is fully open.

β can vary from 0 to 100%. A small β value denotes a small pressure drop across the valve, in relation to the total pressure drop. When β is small, the flow will be largely determined by the pressure drops across the other components of the system. This has the consequence that the true flow through the control valve will be greater than suggested by the flow characteristic at a constant pressure drop.

Valve authority, β =

$$\frac{\Delta p_{V100}}{\Delta p_{V100} + \Delta p_{SYSTEM}} = \frac{\Delta p_{Vmin}}{\Delta p_{max}}$$

Flow characteristic (installed characteristic)

The curves in Figs. 4.19 and 4.20 show how the characteristic of a 2-way valve varies, according to the authority of the valve, β. The resulting characteristic is often referred to as *installed characteristic.*

The actual flow, q_a (%), is a function of the theoretical flow, q (%), with $\beta = 1$, and is obtained from the following formula:

$$q_a \% = \frac{1}{\sqrt{\frac{1-\beta}{10000} + \frac{\beta}{q^2}}}$$

To ensure satisfactory operation, β generally should not be less than 0.5, i.e. at least half of the available pressure

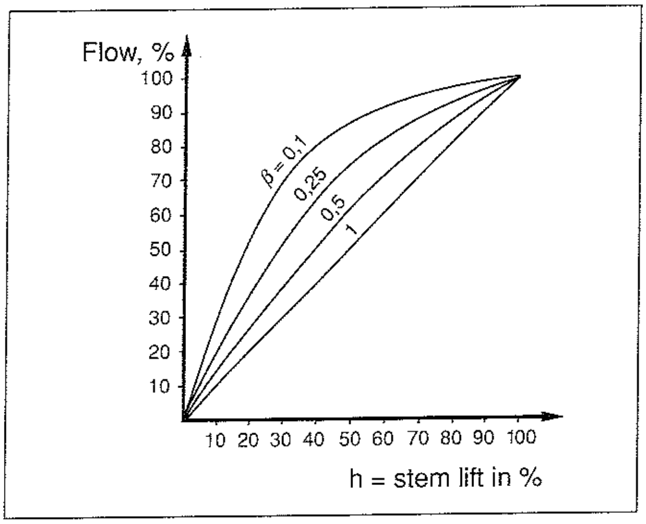


Fig 4.19 Linear flow characteristic for varying β values

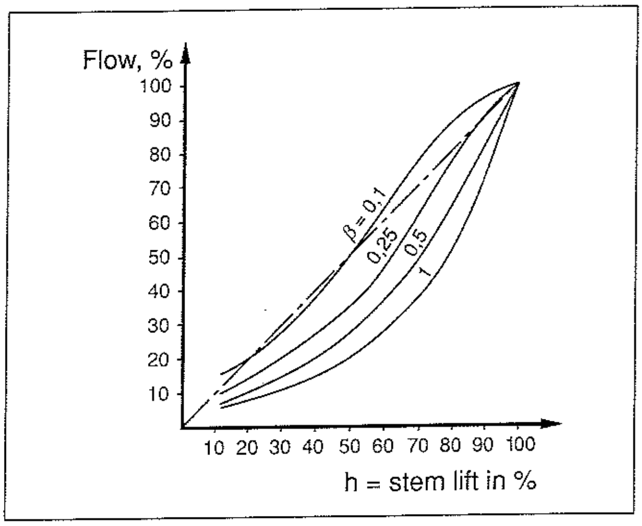


Fig 4.20 Equal percentage flow characteristic for varying β values

should be dropped across the mixing valve, when fully open.

Poor valve authority means that the K_V coefficient of the valve is unnecessarily large (i.e., the valve is oversized). This can be cured by selecting a valve with a smaller K_V coefficient. On the other hand, there is no reason to select a valve with too great an authority, as this would require an unnecessarily large pump.

A more practicable definition of the valve authority concept

A more logical and practicable definition of the valve authority concept is obtained by computing the ratio between Δp_{min} across the control valve at the design flow *and* Δp_{max} across the closed valve:

$$\beta' = \frac{\Delta p \text{ across fully open control valve at design flow}}{\Delta p \text{ across closed valve}}$$

As shown in Fig. 4.21 the authority β' provides a measure of the ***true*** distortion of the characteristic of a valve. The standard definition of the value authority, β, does not do this.

The relationship between the two types of authority is the following:

$$\beta = (Sq)^2 \cdot \beta'$$

where Sq is the relationship between the true flow and the design flow.

$Sq \geq 1$ for an open valve. When the maximum flow equals design flow, $\beta = \beta'$.

Could a balancing valve be installed in series with a control valve?

A control valve that has exactly the design K_V value is rarely available on the market. The control valves that are installed therefore typically are more or less oversized.

On start-up, for example after a night or holiday setback of system temperatures, most control valves will be fully open. Excess flows will then be found in favoured circuits, and too low flows in less favoured circuits. It is, therefore, important that flows through control valves are limited by means of balancing valves. Fig. 4.22 shows how a balancing valve influences the flow characteristic.

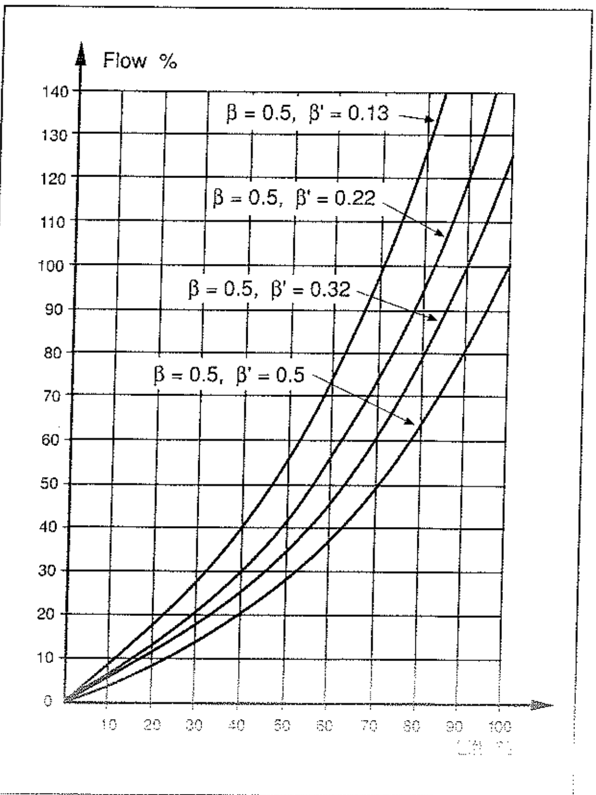


Fig 4.21 Flow as a function of the lift of a valve when the available pressure drop varies and valve authority. β is constant

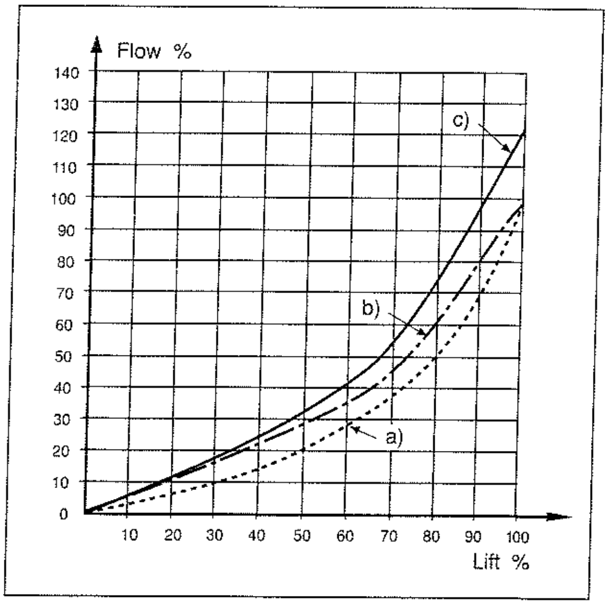


Fig 4.22 The figure shows how the characteristic of a control valve is influenced when limiting the maximum flow by means of a balancing valve

The operating characteristic of 3-way valves

In a 3-way valve the control path characteristic (A to B) as well as the by-pass path characteristic will be affected, as the pressure drop across the valve varies. The control path authority, β_A, should be 0.5 or greater. The pressure drop in the last part of the line is usually very small (unless a restriction valve is installed in the line). Therefore, β_B will be very close to 1.0.

The valve authority of a 3-way valve normally means the authority of the control path, i.e. β_A. The diagrams below show the inherent characteristic ($\beta = 1$) and the installed characteristic ($\beta = 0.5$) of a 3-way valve.

$\beta = 1$ — — — —
$\beta = 0.5$ ————

It can be seen from the curves that the sum (common) flow will not be constant throughout the entire range of stem motion.

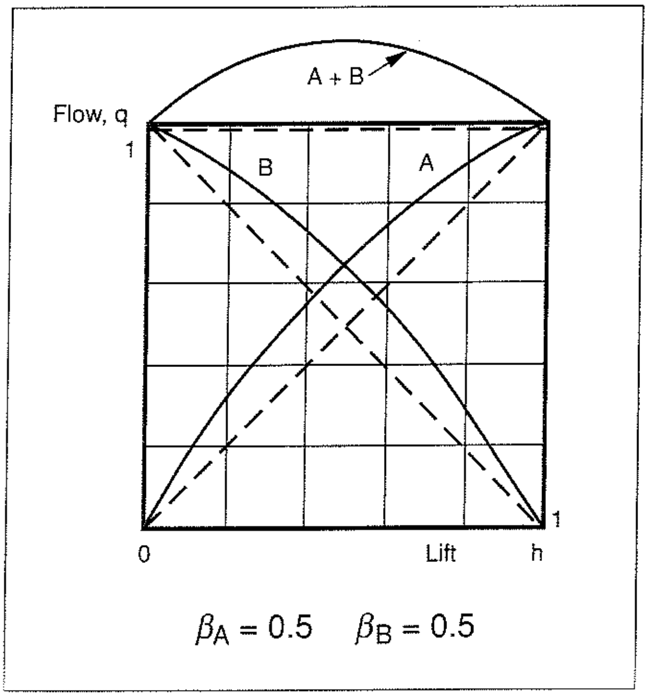


Fig 4.23 Valve with linear flow characteristic

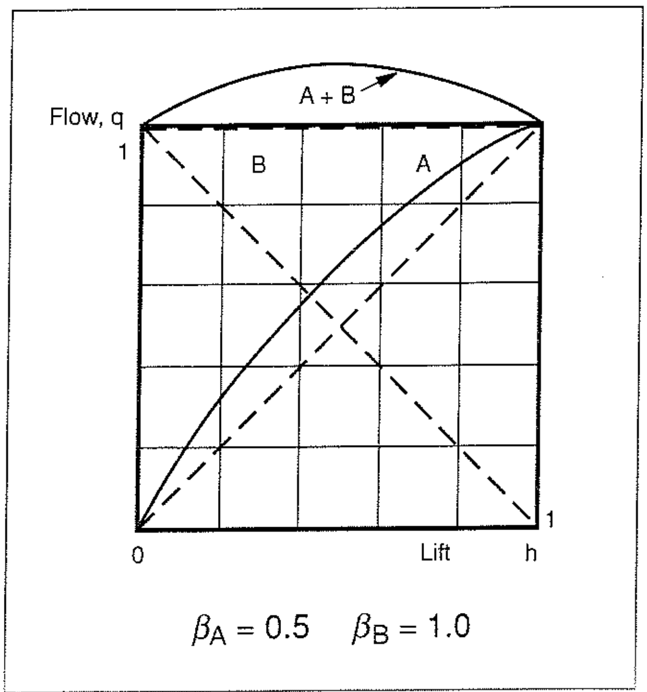
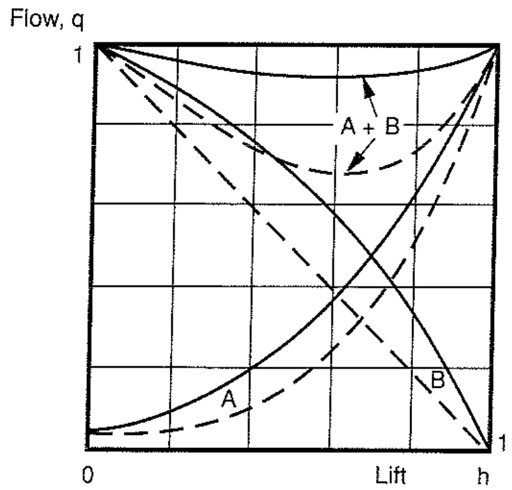


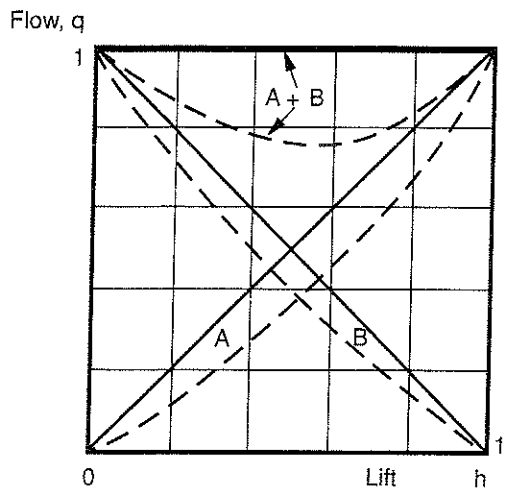
Fig 4.24 Valve with linear flow characteristic

The sum flow is not normally required to be constant throughout the entire motion of the valve stem. It is not sufficient to base your conclusions on only the characteristic of the valve. The influence of the remaining system components must also be considered. When the stem is in the mid position, a sum flow through the valve of greater than 100% (i.e. the flow when fully open) denotes that the coefficient of resistance of the valve is low in this point. This means that the flow in the circuit ought to increase. However, if the flow increases, the pressure drops across the remaining components will also increase, which will counteract the increase in flow. The characteristic of the pump will also affect the final result. The characteristic selected will depend on the type of control system in which the valve will be installed.



$\beta_A = 0.5 \quad \beta_B = 0.5$

Fig 4.25 Valve with equal percentage-linear flow characteristic



$\beta_A = 0.5 \quad \beta_B = 0.5$

Fig 4.26 Valve with modified-linear flow characteristic. When the valve is installed in a system having $\beta = 0.5$ for the A as well as the B port, the flow becomes a linear function of the lift. The sum (or common) flow of ports A and B will thus be constant over the entire lift range.

Calculation of valve authority

$$\Delta p = \Delta p_1 + \Delta p_2$$

$$\beta = \frac{\Delta p_V}{\Delta p + \Delta p_V}$$

Valve authority is calculated only for that part of the system, *in which the flow is variable.*

Δp_V = pressure drop across the fully open valve
Δp_L = pressure drop in section with variable flow (marked ————)
G = Generator (e.g. boiler)
C = Consumer (e.g. heating coil)

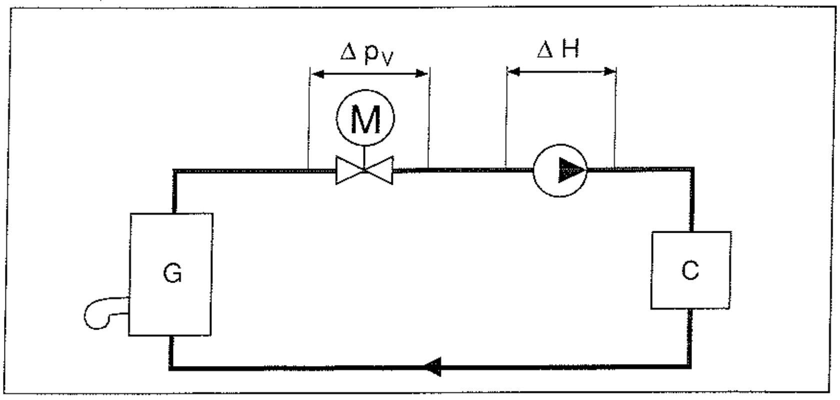


Fig 4.27 2-way valves

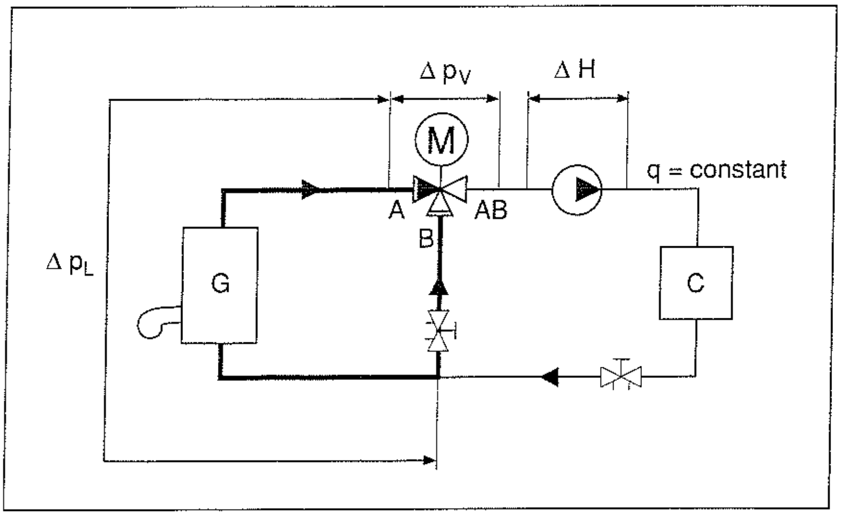


Fig 4.28 3-way valves. Mixing configuration

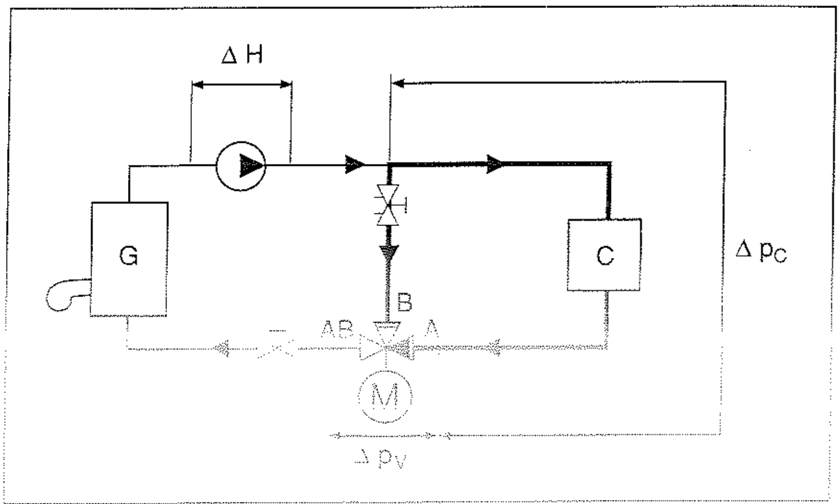


Fig 4.29 Diversion configuration, with mixing valve at mixing point

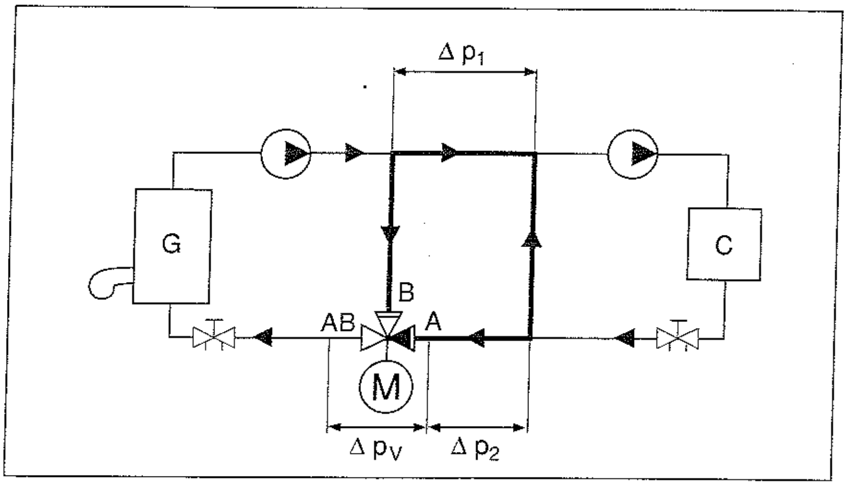


Fig 4.30 Diversion configuration, with internal circulation pump

Rangeability, *RF* (*R*)

The characteristic of a valve must conform to the theoretical curve, within the specified tolerances. Fig. 4.7 shows a theoretical and a true linear flow characteristic, with tolerances for the deviations. Fig. 4.31 shows the principle for the linear characteristic. Fig. 4.32 shows an enlarged part of a linear characteristic at low flows, just above the closed position.

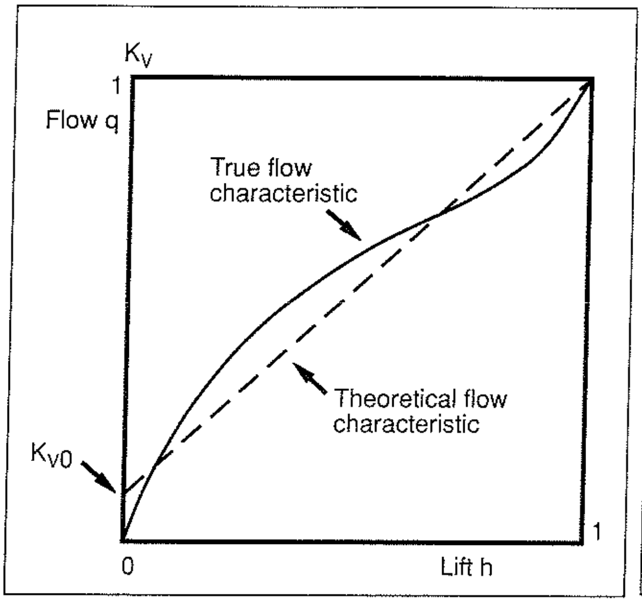


Fig 4.31 Linear flow characteristic

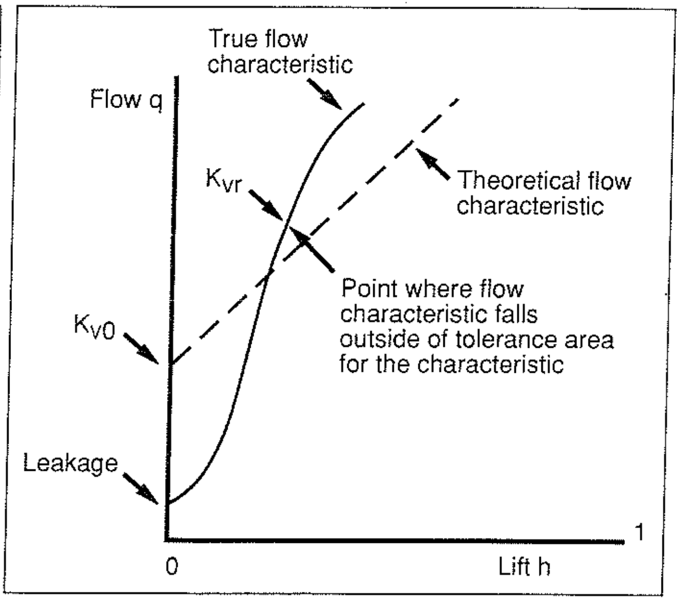


Fig 4.32 Linear flow characteristic, enlarged part at low flows

K_V = flow coefficient of valve, specifies valve capacity (in m^3/h), at the nominal travel and pressure drop (1 bar).

K_{V_r} = minimum flow through the valve, at a point on the characteristic where it starts to fall outside the specified tolerances, at a pressure drop of 1 bar.

K_{V_0} = theoretical minimum flow through valve (m^3/h), at a pressure drop of 1 bar.

h = relative travel (lift).

$$RF = \frac{K_V}{K_{Vr}}$$

The *rangeability*, *RF*, is the ratio of the flow coefficient to the minimum controllable flow within the tolerances for the characteristic. For HVAC control valves, *RF* is normally in the range 30 to 100. As a rule, the greater the rangeability the greater the valve cost, but also the greater the controllability.

System rangeability, R_β and R_S

If the mixing valve is installed in a system, with valve authority β', the pressure drop across the valve will increase, close to the closed position. This means that the flow will also increase, when the valve is nearly closed. In such a case, the true rangeability will be:

$$R_\beta = RF\sqrt{\beta'}$$

Exempel $RF = 30$ and $\beta' = 0.5$

R_β hence $30 \cdot \sqrt{0.5} = 21.2$

If the valve is oversized (from the flow standpoint) by a factor of S_P, the resulting rangeability (R_S) will be degraded by the factor S_P.

The resulting system rangeability, R_S, will be:

$$R_S = \frac{RF\sqrt{\beta'}}{S_P}$$

Exempel $RF = 30 : 1$

$\beta' = 0.5$

$S_P = 2$

hence $R_S = \frac{30 \cdot \sqrt{0.5}}{2} = 10.6$

Split-range valves

The "SPLIT-RANGE" concept originates in the division of the control range between two valves, operating in sequence. First the smaller valve opens. When this valve has reached 30% lift the larger valve starts to open. The resulting rangeablity will be about 100:1.

Normally, the K_v coefficient of the small valve will be selected to be between 1/4 and 1/3 of the combined K_v coefficient, so that the small valve, together with the large valve, gives the desired K_v coefficient.

TA's 2-way valves V299 and STL-SR are split-range valves. Their plugs comprise two plugs, where the small plug controls small flows and the large plug controls large flows.

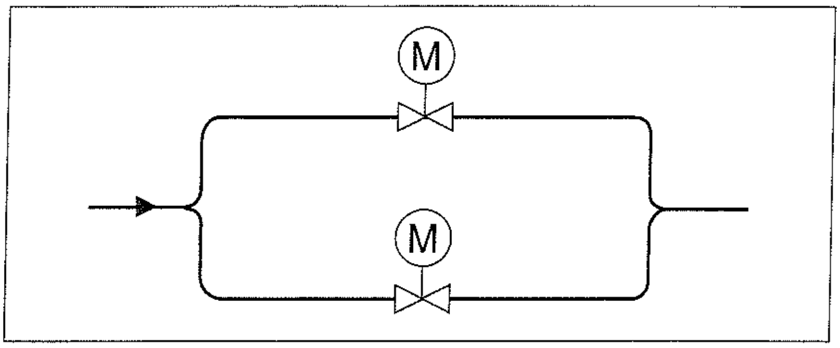


Fig 4.33 Split-Range valve

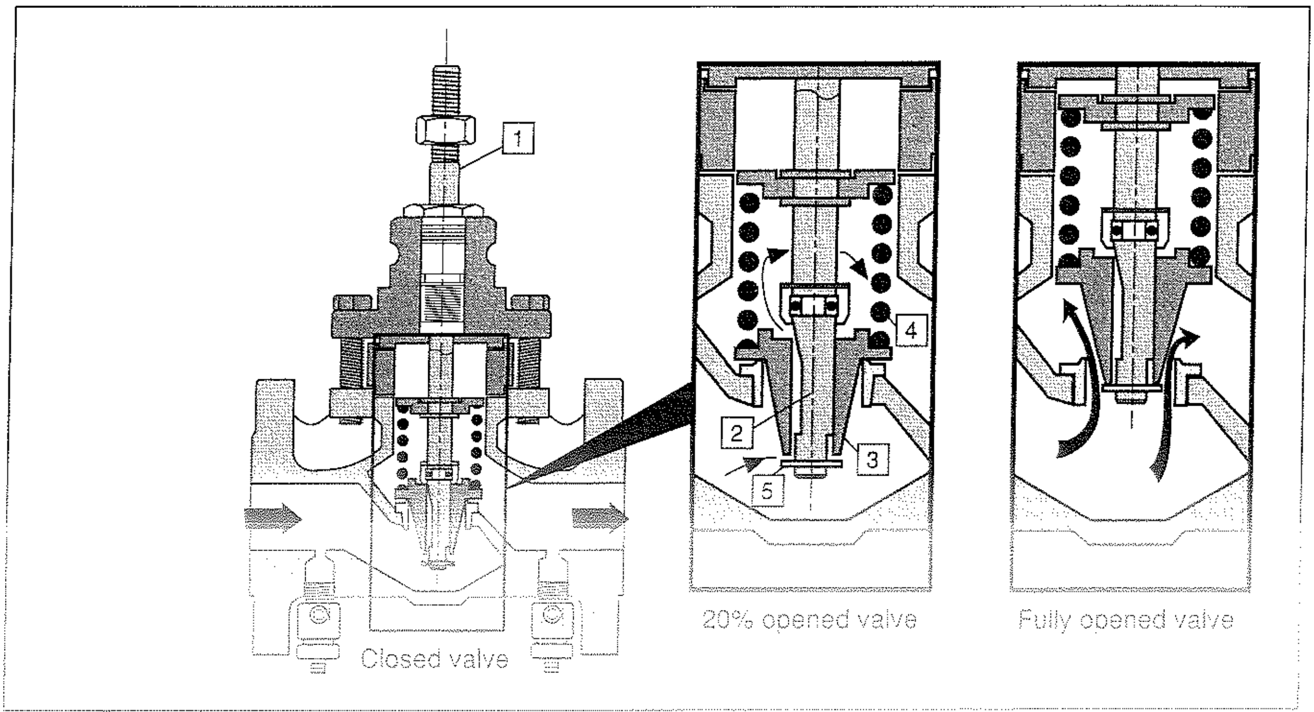


Fig 4.34
Split-Range valves V299 and STL-SR and an enlarged part of the plug

Valve noise

Valves that control large flows, at large pressure drops, often emit noise as the pressure drop exceeds the critical value.

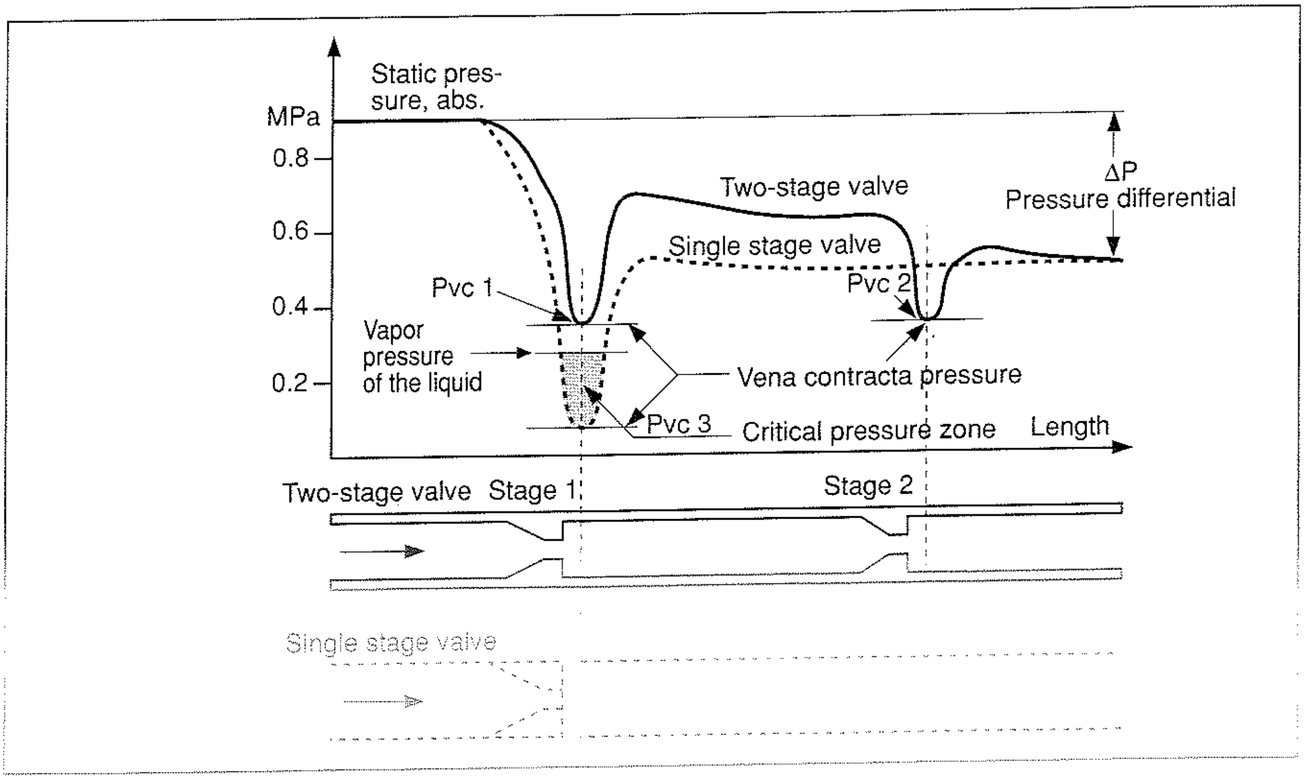
The pressure drop across 2-way valves used in district heating systems can be considerable. If possible, control should be exercised silently, since such valves are frequently installed in substations, close to dwellings.

In installations with high flow rates and pressure drops across valves, V-port valves and standard plug valves have largely been replaced with valves designed especially for these conditions.

Valve noise can be classified as follows:
– mechanical noise
– flow noise
– cavitation noise

Mechanical noise occurs, when parts of a valve start to vibrate, due to the flow. The tendency of a particular valve to vibrate will depend on how well the plug and stem are guided in the valve body. Such vibration will rapidly destroy the plug and seat and will cause metal fatigue, in the stem.

Fig 4.35 Pressure diagram for single and two stage valves



Flow noise is a hissing sound that occurs in the event of turbulent flow through the valve body. The sound level is normally low and not objectionable.

Cavitation noise occurs in liquids when the pressure drop across the valve exceeds the critical value.

As the liquid passes the valve plug and seat, the flow rate increases, whereupon the dynamic pressure will increase. Because the total pressure is constant, the static pressure will be correspondingly reduced. If the flow rate is so great that the static pressure drops below the vapor pressure of the liquid, steam bubbles will be formed, as the liquid starts to boil.

Once the liquid has passed the smallest area of the valve plug and seat, the flow rate will drop and the static pressure will increase. When the static pressure exceeds the vapor pressure of the liquid, the steam bubbles will implode.

When steam bubbles implode, powerful shock waves are generated, which can quickly erode and damage the valve. Cavitation is often the cause of vibration.

Apart from reducing the life of the valve, cavitation causes a considerable amount of objectionable noise, which sounds as if sand were swirling around inside the valve. It may also cause a chirping sound.

Fig. 4.36 shows how the static pressure varies in a valve.

Cavitation and how to avoid it

A few of the most common factors that increase the risk of cavitation are:

- low static pressure
- large pressure drop across the valve
- high fluid temperature
- unsuitable valve design

One method of avoiding the damaging effects of cavitation is to protect the parts of the valve that are exposed to erosion, by coating them with hard metal. Although this prevents cavitation damage, it does not reduce the noise level.

When the supply and return pressures that a valve is exposed are known, methods of eliminating cavitation include:

– installation of pressure reducing valves, before the control valve
– control valves are installed in the system where the pressure and temperature conditions preclude cavitation
– installation of two or more valves, in series, which are controlled so that each valve absorbs a part of the total pressure and the static pressure does not drop below the point at which cavitation can occur.
– installation of special valves, which have been designed to operate at large pressure drops, without cavitation.

A cavitation diagram is often shown in the data sheets of 2-way valves. A typical cavitation diagram is shown below. By means of this diagram it is possible to determine whether or not there is a risk of cavitation in the particular application.

The form of the cavitation curve will depend on the design of the valve.

Example

What is the largest pressure drop a valve can absorb, without incurring any risk of cavitation, in a system operating at a static pressure of 1000 kPa (~145 psi) (gauge pressure) and a water temperature of 120 °C?

Plot a horizontal line, from p_1 = 1000 kPa, to the line representing 120 °C. Plot a vertical line from this point, to the horizontal axis, and read off the maximum pressure drop:

Δp = 430 kPa

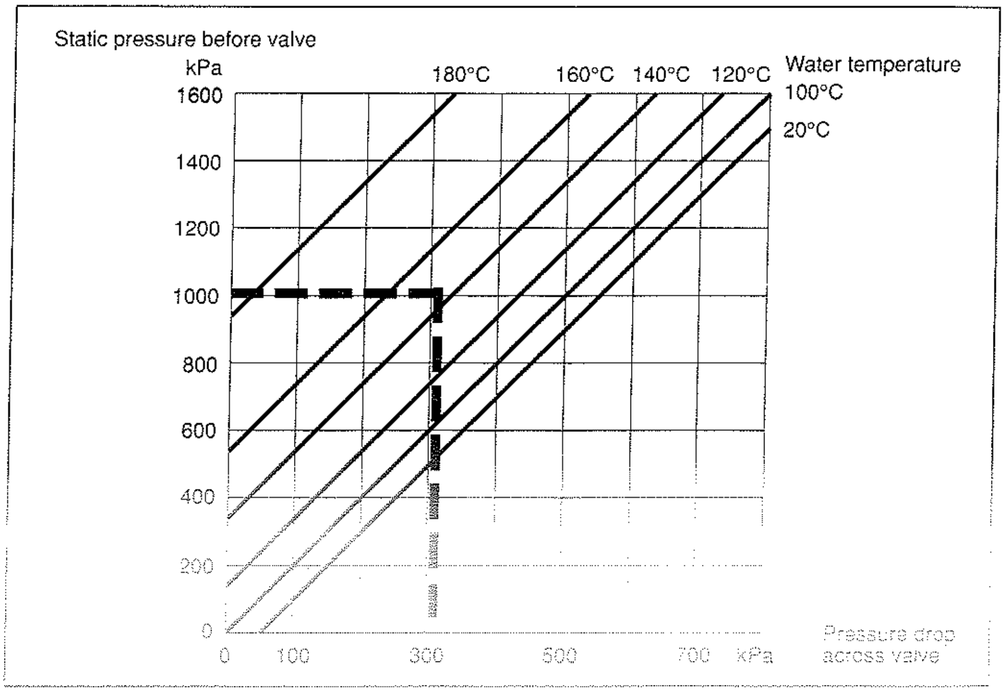


Fig 4.36
Cavitation diagram

5 Correction for viscosity

Correction for viscosity

The viscosity of a liquid is specified by its index of viscosity. The higher the index, the more slowly the liquid will flow. This means that, for liquids of high viscosity, the pressure drop across a valve or pipe must be higher than for low viscosity liquids, if a particular flow is to be maintained. To obtain a given flow, at a given pressure drop across the valve, the flow coefficient of the valve (K_V co-effi-cient) must be multiplied by a factor of K_R (see diagram 5.2).

Index of viscosity

Dynamic viscosity

Dynamic viscosity (η) is the force, required by an area of 1 m^2 to make it move an equally large area with a speed of 1 m/s. This can be expressed as:

$\eta = [N\ s/m^2]$

or, expressed in SI units:

$\eta = [Pa \cdot s]$

Another commonly used term is *poise* and *centipoise*:

1 poise $P = 0.1\ [Pa \cdot s]$

1 centipoise $cP = 0.01P = 1\ [mPa \cdot s]$

Kinematic viscosity

The *kinematic viscosity* (ν) is computed from the dynamic viscosity as the quotient of the dynamic viscosity (η) and density of the liquid (ρ), i. e.:

$$\nu = \frac{\eta}{\rho} \quad \left[\frac{\text{Pa} \cdot \text{s}}{\text{kg/m}^3}\right]$$

Kinematic viscosity plays an important role in the science of flow motion. It is independent of force and mass, i. e. it has the dimension of length unit squared, divided by time, in the SI system of units thus m^2/s.

The term most frequently used for kinematic viscosity is *stoke* (St):

$$1\ \text{St} = \frac{1\ \text{cm}^2}{\text{s}} = 100\ \text{cSt}$$

$$1\ \text{cSt} = \frac{1\ \text{mm}^2}{\text{s}}$$

Viscosity indices for some common liquids:

Water 20 °C	$\nu = 1$ cSt
Water 20 °C + 50 percent by weight glycol	$\nu = 3.17$ cSt
Engine oil, SAE 20	$\nu = 40$ cSt

When computing flow resistance, it is simplest to use kinematic viscosity, ν, as there is a direct proportionality between the resistance coefficient (measured as water column for laminar flow) and ν. To be able to use various viscosity figures, available commercially, in technical computations, these must normally be converted. Refer to table 5.1.

Table 5.1 Kinematic viscosity, conversion table

Centistoke (cSt)	1	4	7	10	15	20	25	30	40	60	120	250	330
°Engler	1	1.3	1.6	1.8	2.3	2.9	3.4	4.1	5.4	7.9	16	33	44
Redwood sec	28	35	43	52	68	86	105	125	164	245	486	1012	1336
Saybold sec	-	39	49	59	77	97	119	141	186	277	555	1157	1528
SAE No.	-	-	-	-	-	-	-	10	20	30	40	60	110

Control valves

Control valves that operate in the range of turbulent flow do not normally require correction for viscosity indices of not more than 10 cSt. However, correction for vicosity must be made in the case of valves controlling very viscous liquids (indices > 100 cSt) at very low flow rates. If the flow is within the laminar region (pressure drop across the valve < 0.5 kPa), valve calculations should give consideration to the viscosity of the medium.

The pressure drop in the pipes will also be effected by increased viscosity. If the pressure measurement nipple (for measuring the pressure drop across the valve) is located far from the valve, consideration must be given to the pressure drop across the pipe section.

Correction of K_V coefficient

Liquid mixtures can have widely varying viscosities. Exact calculation of the necessary K_V coefficient, for liquids of different viscosities, is difficult. However, the following method, which is based on empirical constants, gives acceptable results.

1. Compute the K_V value of the valve.

$$K_{Vcalc} = \frac{q}{\sqrt{\Delta p}}$$

2. Compute viscosity correction factor, R_V:

$$R_V = \frac{3.85 \cdot 10^3 \cdot q}{\sqrt{K_{Vcalc} \cdot v}}$$

3. Read the correction factor, K_R, from the curve below.

4. Compute K_{Vcorr}:

$$K_{Vcorr} = K_{Vcalc} \cdot K_R$$

K_V = flow coefficient, m^3/h
q = flow, m^3/h
Δp = pressure drop, in bar (100 kPa)
v = kinematic viscosity, in cSt

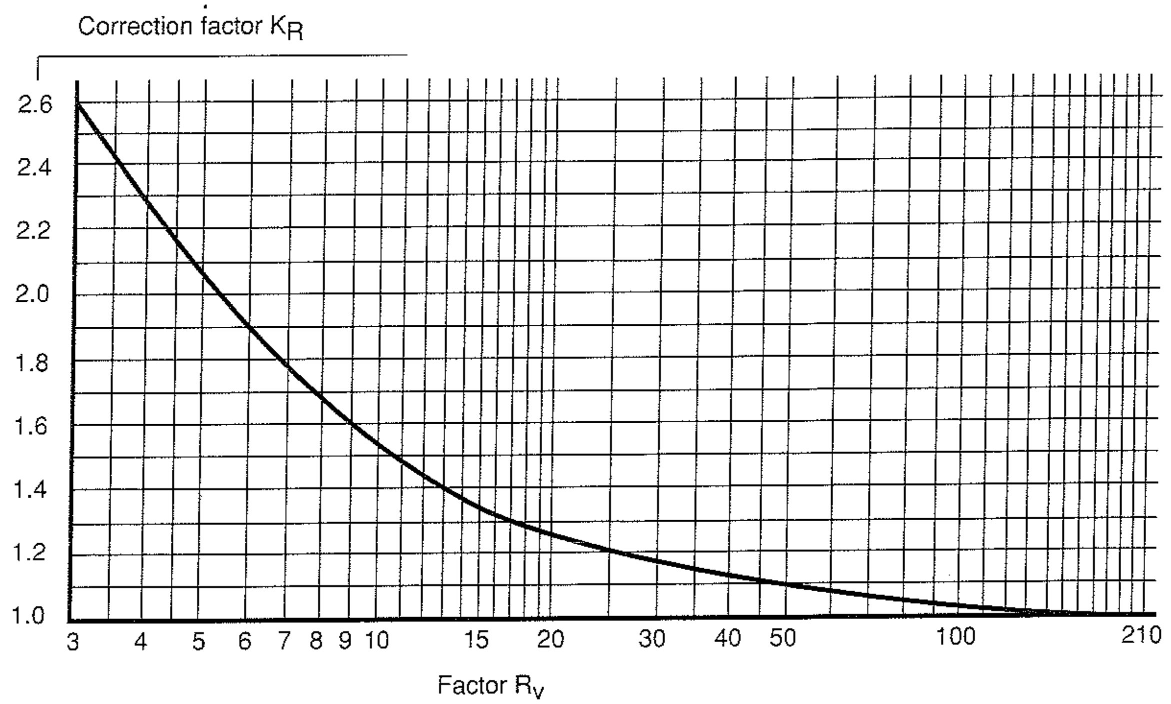


Fig 5.2 Viscosity correction factor, K_R

If the factor R_V is less than 4, the certainty of the results is doubtful, so tests should be performed.

Calculation examples

Example 1

Dimension a 2-way valve to control a water-glycol mixture with 50% glycol at –10 °C.

Prerequisites:

q = 10 m³/h
Δp = 100 kPa (1 bar)

Kinematic viscosity of glycol mixture at –10°C,
ν = 10 cSt

1.

$$K_{Vcalc} = \frac{q}{\sqrt{\Delta p}} = \frac{10}{\sqrt{1}} = 10\ m^3/h$$

2.

$$R_V = \frac{3.85 \cdot 10^3 \cdot q}{\sqrt{K_{Vcalc}} \cdot \upsilon}$$

$$R_V = \frac{3.85 \cdot 10^3 \cdot 10}{\sqrt{10} \cdot 10} = 1217$$

3.

$K_R \approx 1.01$, i.e. no correction required

4.

Select a valve with a $K_v = 10$

Example 2

Dimension a 2-way valve to control molasses (i.e., a sugar solution)

Prerequisites:

$q = 10\ m^3/h$
$\Delta p = 100\ kPa = 1\ bar$

Heavy oil has:
density $\rho = 1400\ kg/m^3$
dynamic viscosity $\eta = 1000\ cP$

1.

$$K_{Vcalc} = \frac{q}{\sqrt{\Delta p}} = \frac{10}{1} = 10\ m^3/h$$

2.
Conversion of dynamic viscosity, η, to kinematic viscosity, ν:

1000 cP = 10 Poise = 10 g/cms = 1 kg/ms

$$\nu = \frac{\eta[kg/ms]}{\rho[kg/m^3]} = \frac{1}{1400} = 7.14 \cdot 10^{-4}\ m^2/s$$

$$= 7.14\ cm^2/s = 7.14\ St = 714\ cSt$$

$$R_V = \frac{3.85 \cdot 1000 \cdot 10}{\sqrt{10} \cdot 714} = 17.05$$

3.

$$K_R = 1.3$$

4.

$$K_{Vcorr} = K_{Vcalc} \cdot K_R = 10 \cdot 1.3 = 13 \text{ m}^3\text{/h}$$

Select a valve with $K_V = 16$

Glycol

Monoethyleneglycol

is used as an anti-freeze agent in heating systems, where there is a risk of freezing. This type of glycol is poisonous and must not be used in systems for the refrigeration of food and drink.

Monopropyleneglycol

is not poisonous and used in heat exchangers that are in contact with food, tap water, etc.

Glycol mixtures

When dimensioning a heating system that will contain glycol, the necessary flow (as calculated for water, without additives) must be corrected, as the specific heat and heat transfer coefficient of glycol are not the same as for water.

Glycol, specific heat

The specific heat of glycol is temperature-dependent, refer to the diagram below.

Glycol, heat transfer index

The heat transfer index of glycol is about 25% lower than that of water, so it must be taken into account when dimensioning heat exchangers.

Recommendations for dimensioning of systems

1. Glycol-water systems should be dimensioned as water systems, with a 10% increase in the calculated pressure.
2. The flow of the glycol mixture should be increased by 30%, relative to water, which gives about the same heat transfer index.
3. Increased temperature drop in the heating coil.
4. Increased air speed in the air heaters, which increases the heat transfer number, *K*.

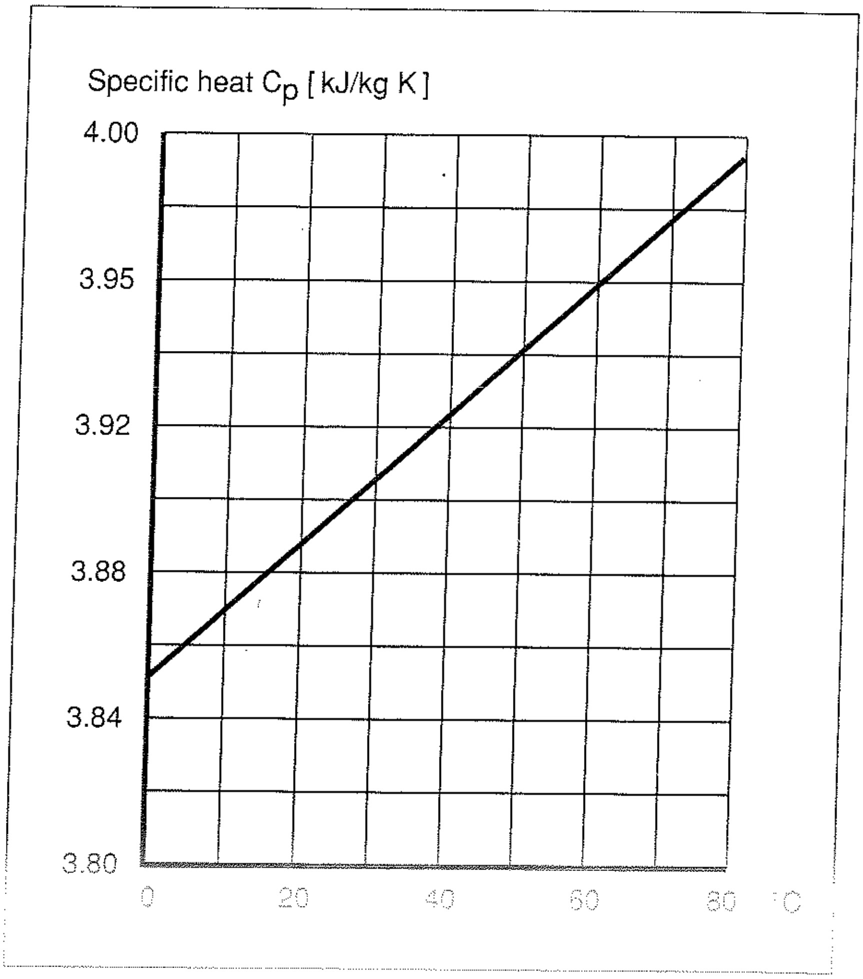


Fig 5.3 Specific heat of glycol, temperature dependence

6 Steam

General

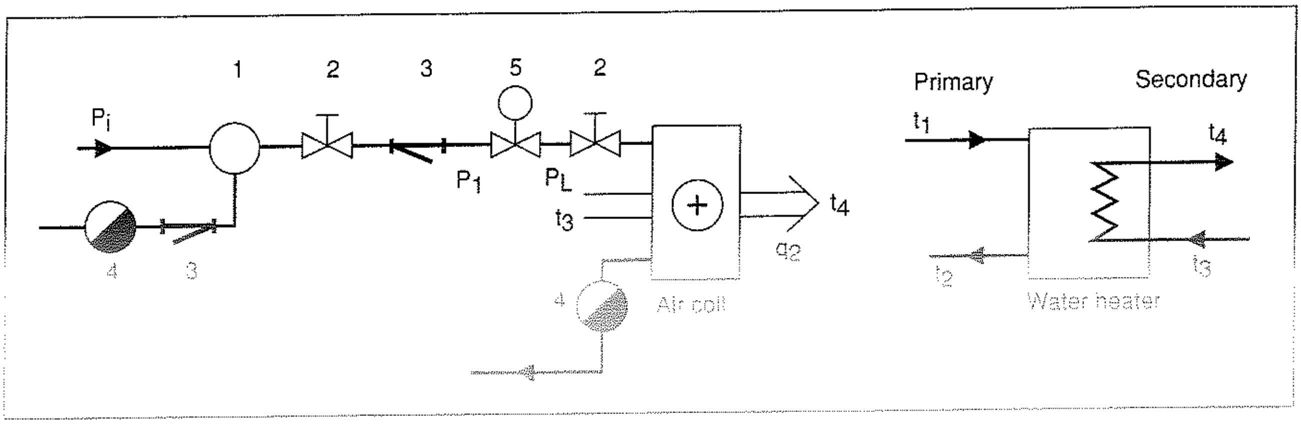
To attain satisfactory control of steam, the system must include a control valve, filter, steam trap, bypass line and shut-off valve, which must all be sized for the greatest anticipated load.

For proper function of the system, the installation must be planned so that no condensate reaches the valve. If condensate does reach the valve, it changes into steam, which results in the release of a large quantity of energy and "steam-hammering". The valve plug will be exposed to considerable forces, which can damage the linkage between the valve and actuator.

To avoid steam-hammering, the following recommendations apply:

- insulated piping, to prevent the formation of condensate
- lay the piping, so that there is slope in the direction of the steam flow
- branches should be connected on the upper side of the piping
- steam traps should be installed at the lowest point in risers
- steam traps should be installed at all low points
- a steam trap should always be installed on the supply side of steam valves.

Fig 6.1
1. Steam dryer
2. Shut-off valve
3. Strainer
4. Steam trap
5. Control valve



Outflow of gases

Assume that a gas is contained in a vessel, which has a small hole, through which the gas can escape. If the pressure drop across the hole (i. e., a valve) is increased, the speed of the escaping gas will also increase, up to a certain limit. It can be shown that this limiting speed is the speed of sound in the steam (or gas, steam is a gas), at the critical pressure, specific volume and temperature.

Once the above limit has been reached, the flow of escaping gas will depend on the type of gas, area of the hole and the conditions prevailing within the vessel. Additionally the limit will be independent of the external pressure.

The pressure ratio when the exhaust speed attains its maximum value is designated the *critical pressure ratio*, and is (p_K/p_1)*.

* p_1 = the absolute pressure in the vessel or the pressure before the valve. Absolute pressure = gauge pressure + 1 bar (100 kPa).

Critical pressure ratios for:
Superheated steam: $p_K/p_1 = 0.546$
Saturated steam: $p_K/p_1 = 0.577$

Hence the critical pressure drop limit:
For saturated steam: $p_K = 0.577 \cdot p_1$
For superheated steam: $p_K = 0.546 \cdot p_1$
Rounding off, for practical use: $p_K = 0.5 \cdot p_1$

Example
Prerequisites: Pressure of saturated steam before the valve is 2 bar = 300 kPa_{abs}.

Hence the critical pressure drop limit

$p_K = 0.577 \cdot 300 = 173.1$ kPa

Selection of pressure drop across valves

Modulating valves

The pressure drop across the valve must be greater than that across all other controlled components in the system. This means that it will primarily be the pressure drop across the valve that determines the pressure in the entire system.

The following rules apply to modulating valves.

a) In a modulating system, the pressure drop across the valve should be chosen to be at least 80% of the pressure differential between the supply and return.

b) If the 80%-pressure drop exceeds 50% of the supply pressure, calculated as absolute pressure, the latter should be used as the pressure drop across the valve.

A large pressure drop across the valve has little effect on the quantity of heat emitted, but it does contribute towards satisfactory control.

Two-position valves

(ON/OFF control)

A 2-position valve is often chosen so that its end connections matches the pipe connections. The pressure drop across the valve will be insignificant, which means that the required quantity of steam will be rapidly released into the heater. It is normal to permit a pressure drop of between 10 and 20% of the supply pressure.

Calculation of K_V's

Medium	Critical pressure drop $\Delta p \geq 0.5 \cdot p_1$	Sub-critical pressure drop $\Delta p < 0.5 \cdot p_1$
Saturated steam	$K_V = \frac{G}{11.35 \cdot p_1}$	$K_V = \frac{G}{22.7 \cdot \sqrt{\Delta p \cdot p_2}}$
Superheated steam	$K_V = \frac{G \cdot k}{11.35 \cdot p_1}$	$K_V = \frac{G \cdot k}{22.7 \cdot \sqrt{\Delta p \cdot p_2}}$

$$k = 1 + 0.0012 \cdot t_S$$

K_V = valve flow coefficient, at $\Delta p = 1$ bar

p_1 = absolute pressure before the valve in bar

p_2 = absolute pressure after the valve in bar

Δp = pressure differential across valve in bar

G = steam flow in kg/h

t_S = superheating temperature of the steam

k = correction factor for superheated steam

Step-wise valve calculations

1.
Calculate steam flow, G, in kg/h:

$G = 1.59 \cdot P$

$G = \dfrac{Q}{540}$

$P = \text{kW}$
$G = \text{kg/h}$
$Q = \text{kcal/h}$

2.
Calculate pressure drop across valve, in bar

$\Delta p = 80\% \cdot (p_i - p_r)$
or, alternatively:
$\Delta p = 0.5 \cdot p_1$

p_i = supply pressure of the system
p_r = return pressure of the system
p_1 = absolute pressure before the valve

3.
Calculate value of K_V (critical pressure drop)

$K_V = \dfrac{G \cdot k}{11.35 \cdot p_1}$

$G = \text{kg/h}$
$K_V = \text{m}^3/\text{h}$

4.
Select a valve that satisfies given conditions:

- temperature and pressure limitations
- K_V coefficient
- permissible pressure drop across valve
- close off pressure. Ensure that the actuator force is sufficient to provide the necessary close off pressure.
- flow characteristic
- correct type of connection (flanged or screwed)
- satisfactory seal between plug and seat, i.e. the valve must be designed for controlling steam.

Calculation example 1

Dry saturated steam

Calculate a 2-way valve for critical pressure drop

Prerequisites:

$P = 200$ kW (171 Mcal/h)

$p_{1abs} = 1\ 000$ kPa (10 bar) pressure before the valve

$p_{2abs} = 400$ kPa (4 bar) pressure after the valve

1.
Compute the critical pressure drop limit, p_K

$p_K = 0.577 \cdot p_1$

$p_K = 0.577 \cdot 1000 = 577$ kPa

2.
Compute the steam flow, *G,* in kg/h

$G = P \cdot 1.59$

$G = 200 \cdot 1.59 = 318$ kg/h

3.
Compute the K_V coefficient of the valve (critical pressure drop for saturated steam)

$$K_V = \frac{G}{11.35 \cdot p_1}$$

$$K_V = \frac{318}{11.35 \cdot 10} = 2.81$$

4.
Select valve

Select STL25 $K_V = 3.7$

If a type M5 or M15 actuator is used, the valve can close against 1000 kPa.

See also the section on calculation with steam diagram.

Calculation example 2

Superheated steam

Dimension the valve for a subcritical pressure drop

Prerequisites:

Steam flow, G = 60 kg/h

p_{1abs} = 500 kPa (5 bar)
p_{2abs} = 350 kPa (3.5 bar)

1.
Compute the critical pressure drop limit, p_K

$$p_K = 0.577 \cdot p_1$$

$$p_K = 0.577 \cdot 500 = 288 \text{ kPa}$$

2.
Compute pressure ratio, Z

$$Z = \frac{p_1 - p_2}{p_1} \cdot 100$$

$$Z = \frac{500 - 350}{500} \cdot 100 = 30\%$$

3.
Compute the superheating temperature of the steam, t_s

Steam temperature = 251 °C
Steam saturation temperature at 500 kPa_{abs} = 151°C
(obtained from steam table)
Superheating temperature, t_s: 251 − 151 = 100°C

4.
Compute the correction coefficient for superheated steam, K

$$K = 1 + 0.0012 \cdot t_s = 1 + 0.0012 \cdot 100 = 1.12$$

5.

Compute the K_V coefficient of the valve (subcritical pressure drop for superheated steam)

$$K_V = \frac{G \cdot k}{22.7 \cdot \sqrt{\Delta p \cdot p_2}}$$

$$K_V = \frac{60 \cdot 1.12}{22.7 \cdot \sqrt{(5-3.5) \cdot 3.5}} = 1.29$$

6.

Select valve

Select an STL 20, $K_V = 1.6$

Calculation example 3

Control valve for air heater

Calculate a control valve for an air heater with a steam consumption of 130 kg/h

Prerequisites:

Inlet relative pressure, $p_1 = 300$ kPa gauge

1.
Compute pressure differential (Δp) across the valve

$\Delta p = 80\% \cdot 300 = 240$ kPa

This gives outlet pressure, $p_2 = 300 - 240 = 60$ kPa

(Note: Gauge pressure)

2.
Compute the critical pressure drop limit for saturated steam (p_K)

* p_{1abs} = absolute pressure = $p_1 + 100$ kPa

$$p_K = 0.577 \cdot p_{1abs}{}^*$$

$$p_K = 0.577 \cdot (300 + 100) = 230 \text{ kPa}$$

3.
Compute the K_V coefficient of the valve

$$K_V = \frac{130}{11.35 \cdot 4} = 2.86$$

4.
Valve selection

Valve and actuator must close against 300 kPa and 145°C.

Valve with flanged or threaded end connections.

Select an STL 20 valve, $K_V = 2.5$ alt 4.0.

7 Resistance to chemicals and to mechanical stress

Resistance to chemicals

The table on page 78 specifies the resistance to chemicals normally encountered in HVAC media, for TA's range of valves. This table represents a summary of the expertise available in the TA company.

As new facts are constantly being discovered, such a table can never be complete. Thus, we cannot be held responsible for, or guarantee, the ability of control valves to withstand the media encountered in industrial processes. Instead, the table should be considered as a guide when selecting valves, primarily for HVAC applications.

Brines

Brines are solutions of salt (up to 30%) in water, comprising either sodium chloride or calcium chloride, and are intended for use in circulatory systems, in which a low freezing point is desired. Such systems may or may not contain oxygen. Even inexpensive materials, such as cast iron, offer excellent durability in systems that contain no oxygen. However, if there is any oxygen in the system, the life of even very resistant materials can be drastically reduced.

Recommendations for plants using brine-type refrigeration media

To avoid corrosion damages to fittings and pipes, that could cause operational disturbances in plants using brine-type refrigeration medias (typically composed by calcium chloride, $CaCl_2$, sodium chloride, NaCl plus additives such as sodium bichromate and caustic soda) the following should be observed:

Resistance table

Explanations:

a) See sedtion "BRINES". STM, VTRA, TRV = A
b) V294, V294F – contact TA
c) < 30 mg chlorine/litre
d) V386, V387 = A

A = excellent
B = fair; check before use wity TA representative
C = not usable

A number after the above letters means that, before use, the standard packing box must be replaced:
1 = packing box Q
2 = packing box X

Exempel:
Glycol, V221, A1 means that V221 is usable if the standard packing box is replaced by packing box Q.

	Valve series						
	DN 15	DN ≥20	DN 15	DN ≥20	DN 20	DN ≥20	
	V294 V394	V265 V294 V295 V298 V353 V386 V394 VTRE	V282 V384 V386	V282 V283 V384 V386	V354 V355	V221 V223 STM V395-1 V395-2 TRV V396-1 V396-2 VTRA	
Medium							
Brinol	C	C	A	A	A	C	
Kerosene	B2	B1	A2	A1	C	C	
Freon 12< 200°C	C	C	C	C	C	C	
Glycol, ethylene	A2	A1	A2	A1	C(B)	A1	
Glycol, propylene	A2	A1	A2	A1	C(B)	A1	
Hydrazine	A	A	A	A	A	A	
Calcium chloride	B2	B1	B2	B1	C(B)	A1	*a)*
Sodium chloride	B2	B1	B2	B1	C(B)	A1	*a)*
Oil, animal	A2	A1	C	C	C	C	*b)*
Oil, mineral	A2	A1	C	C	C	C	*b)*
Oil, linseed	B2	B1	C	C	C	C	*b)*
Oil, synthetic	A	A	C	C	A	A	*b)*
Oil, hot	A2	A1	C	C	C	C	*b)*
Water, desalinated	B	B	B	B	B	B	
Water, distilled	B	B	B	B	B	B	
Water, domestic	A	A	A	A	A	A	
Water, sea-	C	C	C	C	C	C	
Water, industrial	B	B	B	B	B	B	*c)*
Water, hot	A	A	A	A	A	A	
Water, pool-	B	B	A	A	A	B	*d)*

Metals and metal alloys corrode to a greater or lesser extent when in contact with electrolytes (this also includes water). If the electrolyte contains oxygen, ($H_2O + O_2$) various types of corrosion attacks can be initiated. The corrosion attacks go on until the oxygen is consumed. Strong electrolytes, such as water solutions with high electrical conductivity (normal brines, sea water) rapidly cause corrosion damage in many metals and metal alloys. These can be avoided by removal of the oxygen.

As brines cannot be thermally degassed, other degassing methods must be used. A suitable method is blowing nitrogen gas, N_2 through the solution. Then the major part of the oxygen reacts with the nitrogen to form nitrogen oxides (NO) which are removed from the solution with the nitrogen gas. Degassing must also be done when new (make-up) refrigeration solution is added.

It is recommended to install stationary equipment for the measurement of oxygen content, or to check the same with a portable oxygen meter. Thereby degassing can be done more effectively, and be kept under close control. If the given directions are followed, the plant will function well with most fittings and piping materials. If not, the plant must be constructed using special alloys, such as bronze.

Summary

1. Effective degassing, the nitrogen gas method being best.
2. Continuous monitoring of oxygen content in the refrigeration solution.
3. Effective degassing also of make-up solution.
4. Provided points 1, 2 and 3 above are met, the construction materials in all TA valves can be used in the types of plants mentioned above .

Glycol mixtures

Glycol mixtures are used, besides brines, as anti-freeze agents. Glycol concentrations up to 50% are used. Above this value, the freeze protection capability of glycol deteriorates.

Considerations when using glycol mixtures

- Viscosity of glycol mixtures is a function of glycol concentration. In accurate calculations of K_V values and pipe flow resistance, viscosity must be corrected for. See Part 5.
- The specific heat of glycol is temperature-dependent and differs rather much from that of water. Correction for specific heat must therefore be done in heat transfer computations. See Part 5.
- The heat transfer coefficient is lower than that of water. This must be taken into consideration when designing heat exchangers. See Part 5.
- Monoethyleneglycol is poisonous. For additional information on precautions, see Part 5.
- Monopropylene glycol, by contrast, is not poisonous. See Part 5.
- Surface tension of glycol is low which increases the tendency for leakage in packing boxes and pipe couplings. Therefore special types of packing boxes are used for glycol mixtures. See Part 3.

Corrosion

The concept of corrosion covers various chemical or electrochemical attacks on metals. In the case of ordinary steel, the word "rusting" is normally used.

Corrosion is costly and can result in plant shutdowns due to corrosion damages in piping and valves.

Forms of corrosion

- General corrosion
- Pitting
- Stress corrosion
- Crevice corrosion
- Intercrystalline corrosion
- Couple action
- Corrosion by hot gases
- Erosion corrosion
- Cavitation korrosion
- Abrasion corrosion
- Direct current corrosion

The different forms of corrosion are described below.

General corrosion

When general corrosion occurs, nearly the whole of the surface will be similarly affected (e.g. rusting). Rusting requires the presence of oxygen as well as water. One condition is that the relative humidity exceeds 60%. It is widely known that water always contains a certain amount of oxygen, dissolved (up to 15 ppm), which means that objects that are immersed in water can rust.

Hydrazine or ammonia additives are used to remove the oxygen from water. In the case of stainless materials, rusting often occurs as a result of incorrect surface finishing.

Pitting

When pitting occurs, there may be little surface damage, although the depth of the damage may be considerable. For this reason pitting presents a greater hazard than general corrosion.

Courses of action:
- Selection of suitable materials
- Deactivation of the medium
- Addition of inhibitors
- Good condition of material (ground or polished surface)
- Passivation of surface.

Stress corrosion

Stress corrosion occurs as a result of static tension in materials, in combination with a corrosive medium. This results in the formation of cracks. Tensile stress can be caused by welding or by local heating, when the material is not free to expand. It should be noted that this type of corrosion does not occur in pure metals.

- Brass cracks easily, in the presence of ammonia.
- Iron cracks, in the presence of nitrates.
- Stainless steel and certain aluminium alloys are attacked by water containing chloride ions, e.g. household salt.

Crevice corrosion

Crevice corrosion denotes an extremely localized, ridged type of corrosion occurring in narrow gaps and spaces, where liquids and dirt collect easily (e.g. in the gaps of flanged connections).

Intercrystalline corrosion

Intercrystalline corrosion occurs at the crystal boundaries of materials. It can occur in stainless materials. Such corrosion is only caused by acidic solutions. Nickel reacts with sulphur, to form nickel sulphide, which precipitates and undermines the structure of the material, until catastrophic failure occurs. This applies especially to 18–8 type stainless steels (with a carbon content of more than 0.08%), which are exposed to this type of corrosion at high temperatures (600–700°C).

NOTE: From the standpoint of corrosion, stainless steels are sometimes inferior to mild steel.

Couple action

Couple action occurs when a current flows (due to a difference in galvanic potential) through two metals, which are in electrical contact, in an electrolyte. Which of the metals will corrode is determined by their relative locations in the electrochemical series.

This type of corrosion can be troublesome in valves and it should be given consideration by the designer.

Corrosion in hot gases

Corrosion occurs in hot gases, where it is known as "scaling", as it is characterized by the flaking of oxide scales. Valve bodies contain such oxide scales, as a result of casting.

Oxide scales are also formed, e.g. when welding water pipes. These scales can break loose and fasten in the narrow gaps of a control valve, resulting in damage to the valve.

Erosion corrosion

Erosion corrosion (also known as "impingement attack") results from the mechanical effects of liquids on metals. The primary cause is the "blasting" to which the metal is subjected. This type of corrosion increases with the flow rate and when the liquid contains air bubbles or particles.

Cavitation corrosion

Cavitation corrosion occurs through the interaction of corrosion and mechanical stress. When the steam bubbles that occur during cavitation implode against a metallic surface, there is a very powerful mechanical shock, which destroys the passivating surface layer and so exposes the material to corrosion.

Abrasion corrosion

Abrasion corrosion occurs in boundaries, where surfaces meet under considerable pressure, without lubrication. The motion can be extremely small and is often invisible to the naked eye. The unevenness of the one surface scrapes away the oxide from the other, so that corrosion occurs. The presence of oxygen is necessary.

This type of corrosion is uncommon in valves.

Direct current corrosion

This type of corrosion has been included for the sake of completeness. It is rarely encountered in valves. DC corrosion occurs as a result of stray direct currents. Such currents occur only irregularly and are difficult to detect. The direct currents caused by arc-welding generators can be responsible for this type of damage.

Valve faults and fault identification

Examination of the valve plug and seat can reveal current or past valve defects.

Cavitation

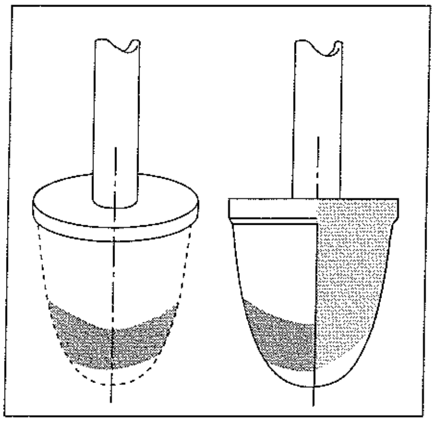


Fig 7.1
Valve plug, showing cavitation damager

Identification

The noise generated sounds like rough gravel passing through the valve, which always seems to occur towards the outlet side.

The damage gives the impression that small chips have been torn from the material. Beneath the damage the material is homogenous. See Fig. 7.1.

System action

The pressure on the supply side of the valve should be increased. The valve should be replaced, by a valve with another cavitation coefficient, e.g. a 2-stage valve (V260).

Mechanical vibration

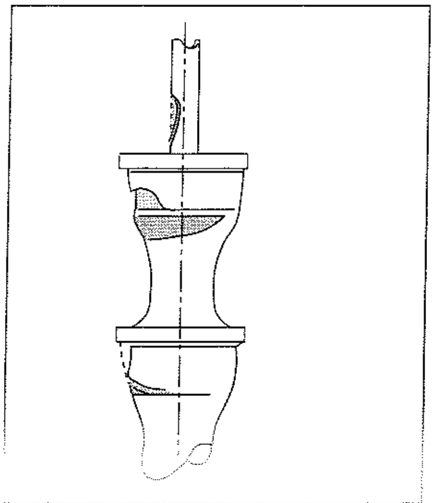


Fig 7.2
Valve plug with damages caused by vibrations

Mechanical vibration is due to radial (horizontal) movement of the plug, between the seats, bearing surfaces and bushes. Vibration can be caused by the effect of shock waves in the medium, on the plug, or by cavitation. It often causes asymmetrical damage.

Identification

A rattling sound, or sounds with a pitch of between 500 and 1500 Hz. See Fig. 7.2.

System action

- Use components with hard surfaces (Stellite).
- Better support for valve plug.
- Avoid cavitation.

Erosion

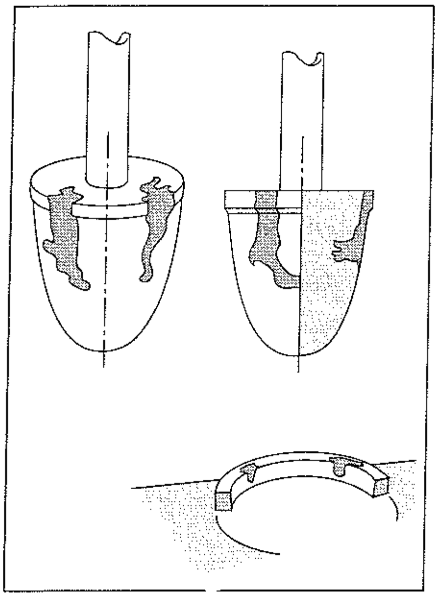


Fig. 7.3
Valve plug and seat with erosion damages

Erosion damages appear as pits and winding channels around sealing surfaces (flow velocity is at its maximum immediately adjacent to a closed valve). Erosion can also cause channels and borehole-like breakthroughs also through the threads, in threaded seats. This is caused by the grinding action on microcomponents in the material.

Identification

No noise generation. Damages are cavities with irregular shape.

Actions

Tight welding of threaded seats *or* sealing with sealing compound.
Valves with high pressure drops should not be allowed to leak through valve plug and seat.

Corrosion

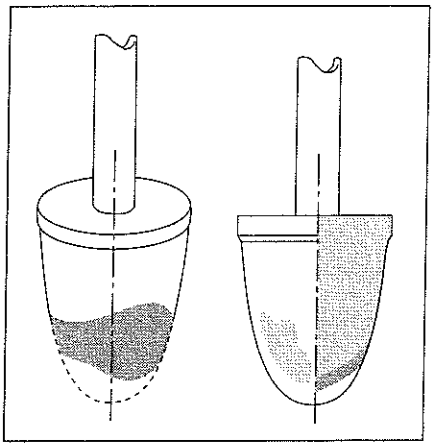


Fig 7.4
Valve plug, showing corrosion damage

Corrosion is caused by attack of chemicals on valve body, plug and seat, and can occur in all media.

Identification

No noise generation. The damage resembles cavitation damage, but is of uneven depth, with an indistinct border towards any undamaged areas. The damage can penetrate deeply and leave the material mesh-like and spongy in appearance. See Fig. 7.4.

System action

Discussions with a chemist and the valve supplier.

Vibration of internal valve components

The internal components of the valve can resonate at their natural resonant frequencies. This can also occur at low pressure and flow rates.

Identification

A pure tone, at a frequency of between 3000 and 7000 Hz, which is very annoying.

Damage

Fatigue fracture.

System action

Replace the valve with one that has better plug guidance.

Plug instability

Occurs when oscillatory, axial movements of the plug generate pressure variations in the medium which, in turn, cause shock waves against the plug – "bath tub plug" effect. Plug instability is usually caused by incorrect flow direction through valve (i.e. valve installed backward.

Identification

A low frequency sound, of 30 to 60 Hz – almost a humming sound. The pitch of the sound depends on the ratio of the mass/resilience characteristics of the actuator and valve components.

Damage

Sealing surfaces can crack and become deformed. The linkage between the stem and actuator may become damaged.

Positioners and other components can become damaged by the vibration.

System action

Ensure that the direction of flows through the valve is correct.

8 Systems

General

When planning installations that will be supplied by local boilers, consideration should always be given to the possibility of later connection to a district heating network. Additionally the probability of such connection should be discussed with the local energy utility.

If the system is to be prepared for connection to a district heating network, space heaters (radiators, convectors and heating coils) should be connected with 2-way valves, according to system 1 or system 2, and dimensioned for water temperatures of 80°C/40°C, at the prevailing outdoor temperature.

In case air heaters are dimensioned for a return temperature of 50°C, the air heater group must always be connected to the boiler in a bypass configuration, with an automatically controlled 3-way valve. This will ensure a sufficiently high return temperature. If air heaters are to be connected to a district heating network, their temperature and pressure specifications must be suitable for such operation.

If there is no likelihood of later connection to a district heating network, the system should be dimensioned for 80°C/60°C operation, at the prevailing outdoor temperature. The heaters should be connected with a 2-way valve, in a bypass configuration (system 2), which ensures circulation through the boiler, or with a 3-way valve (system 4). Select the configuration that gives the lowest system cost.

Air heaters for outdoor air or a mixture of outdoor air and return air should always be fitted with circulation pumps, to prevent freezing.

When such air heaters are installed, a freeze protection thermostat should be installed in the lowest water pipe, which, in case of freezing risk, automatically stops the supply air fan and closes outdoor air dampers.

Design requirements

Substations must be designed to satisfy the instantaneous and continuous demands of the subscriber. The engineering and economic conditions for distribution and production must also be satisfied. This means that:

- The temperature drops on the primary and secondary sides must be as great as possible, i.e. the quantity of water supplied must be controlled, downstream of the load. This facilitates the economic dimensioning of culvert networks and reduces pump energy consumption.
- Heat exchanger volume should be small, i.e. pure heat exchange
- The control of the domestic hot water circuit should be stable and precise.
- Control valves, with low sound levels should be used, dimensioned for the differential pressure, static pressure and temperature of the substation

Swedish district heating plants use primarily two types of heat exchanger configurations, 2-stage and 3-stage configurations. The 3-stage configuration is used in areas with soft water and must not be used for hard water. The 2-stage configuration is used in areas with hard water, but can also be used for soft water.

Concerning types of heat exchanger and their dimensioning, refer to the appropriate manufacturer.

Dimensioning regulations in Sweden

Swedish district heating plants control the supply water temperature in the primary circuit as a function of the outdoor temperature, according to the diagram below.

The Swedish National Board of Physical Planning and Building has issued regulations governing low temperature heating, which require a temperature of not more than 55°C, at the dimensioning value of the heating demand, for secondary heating systems. In certain cases, dispensation can be obtained for other temperatures, within the scope of the municipal energy plan. However, there is not always justification for designing for a supply water temperature of 55°C on the secondary side.

Thus, when modifying or expanding old heating systems and when designing new heating systems in buildings, the temperatures of the diagram in Fig. 8.1 are a useful guide. The dimensioning supply water temperature has been set to a maximum of 60°C. In district heating systems supplying only new estates, however, a temperature of 55°C should be chosen.

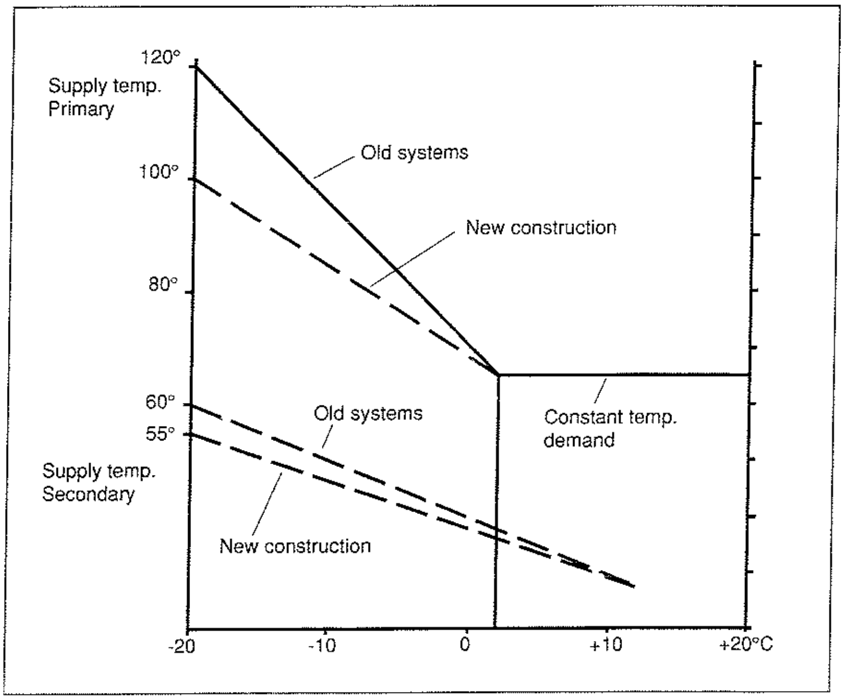


Fig 8.1

Differential pressure in substations

District heating water is distributed by culvert pipelines, from the plant. The main circulating pumps are designed, so that the most remote substations receive sufficient differential pressure.

In practice, this means that the substations close to the district heating plant receive the total differential pressure, ΔP, which, in certain cases, can be around 1.2 MPa, whereas 2-stage and 3-stage heat exchangers require only 100 kPa.

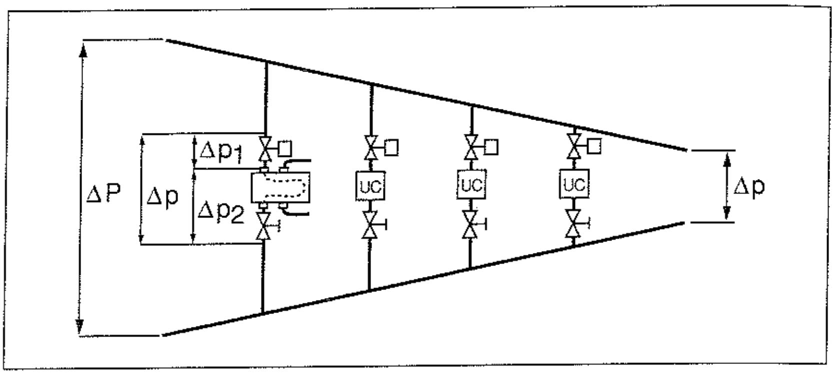


Fig 8.2 The excess pressure must be reduced by the control valves of the substations as silently as possible, without generation of cavitation noise.

3-stage configuration

Sequential control of the mixing valve and primary valve (summer valve) are common, in 3-stage district heating system configurations. (See page 125).

During the winter, the DHW temperature is mainly controlled by the mixing valve. As the heat demand of the radiator group and the need for preheating of the DHW drops, control is successively transferred to the primary valve. During the summer, only the primary valve exercises control. From a control standpoint, the sequential control of valves, with differing characteristics and time constants, is always a compromise.

The mixing valve has its own, individual characteristic and time constant and the primary valve has another characteristic and time constant, which are determined by the size and type of the heat exchanger. Depending on the characteristics of the valves, the system should be controlled with separate controllers for the mixing valve and primary valve.

A typical 3-stage configuration is shown in detail on page 125.

Mixing or diversion valves

As pointed out earlier, 3-way valves are used in order to minimize upsets in pressure and flow balances in the system. The difference between mixing and diversion valves from a design standpoint is shown in Part 2 (pages 15 and 21).

In every control circuit there is a mixing and a diversion point. In new construction, the valve is installed in the mixing point. When renovating old installations, the diversion valve should be retained, if the circuit was so designed originally.

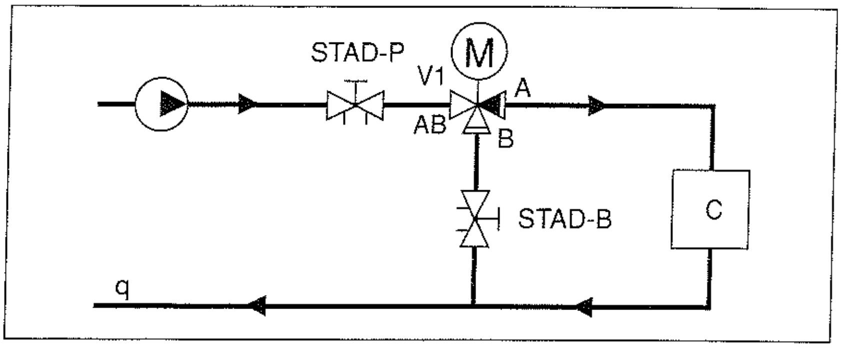


Fig 8.3 3-way valve as a diversion valve

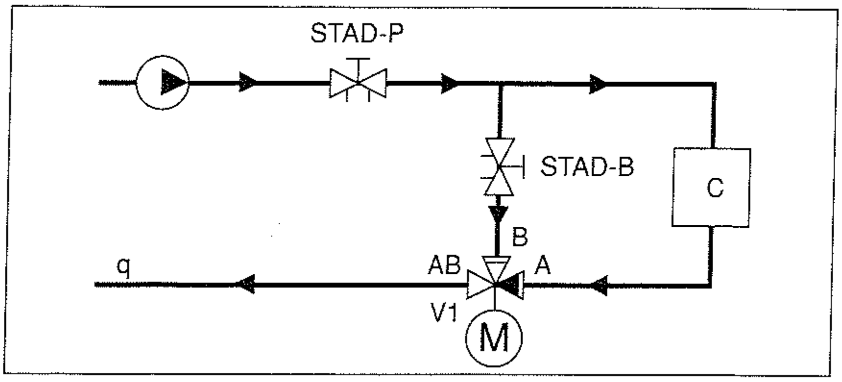


Fig 8.4. 3-way valve as a mixing valve

Pressure drop across 3-way valves

The pump pressure and the pressure drop across the 3-way valve are often confused. 3-way valves always have some water path open, which means that the total pressure from the pump does not affect the mixing valve.

Which pressure drop affects the valve plug?

Ignore pressure drops in pipes and pipe bends. Close path B of valve V1. The flow from the diversion point, C, through the balancing valve, V3, is zero. There can be no pressure drop in this line. This means that the same pressure prevails in diversion point C and at plug B. The flow from point C passes through the load, L, and valve port A.

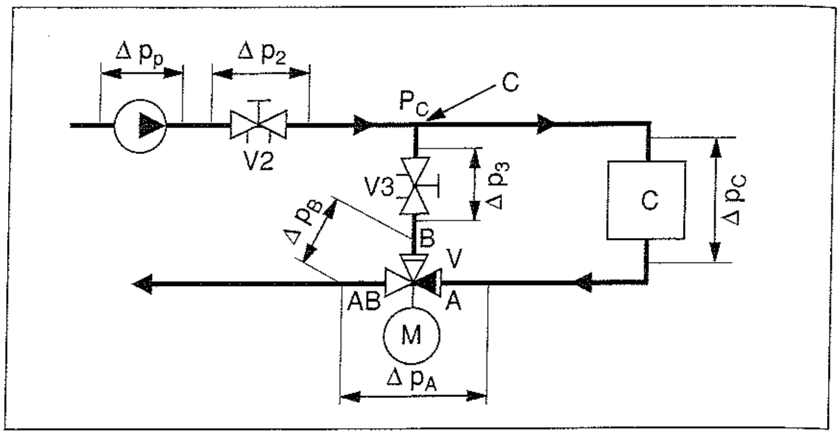


Fig 8.5 Pressure drops in typical mixing valve circuit

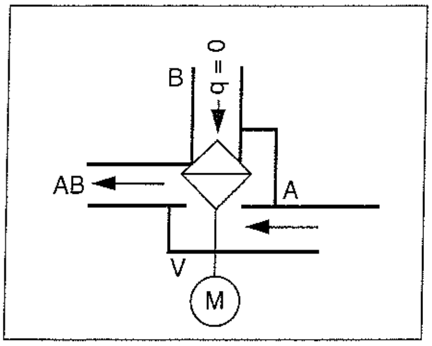


Fig 8.6 3-way valve with water path A open

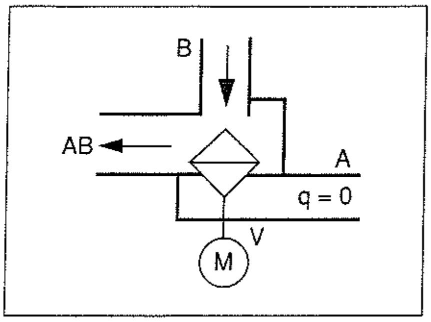


Fig 8.7 3-way valve with water path B open

The valve has been selected, so that its pressure drop will be Δp_A, for a given flow. For the same flow, the pressure drop across the load will be Δp_L.

Pressure at plug A: $p_A = p_C - \Delta p_C - \Delta p_A$

Pressure drop across the valve plug:

$$\Delta P_{plug} = p_B - p_A \rightarrow p_C - (p_C - \Delta p_C - \Delta p_A)$$

$$\Delta P_{plug} = \Delta p_C + \Delta p_A$$

The same reasoning applies, when plug A is closed. The above shows that the 3-way valve is only affected by the pressure drops in the circuit, where the flow is varied by the mixing valve.

The pressure drops that load a 3-way valve is equal to the total pressure drop in the open flow path, calculated from the point at which the flow is divided (C) to the common valve port (AB).

Valve authority

The valve authority should only be calculated for that part of the circuit, in which the flow is effected by the valve. Thus, the balancing valve, V2, in Fig. 8.5 does not effect the valve authority.

The 3-way valve effects the flow in the following parts of the pipe network:

(marked ——— in the figures)

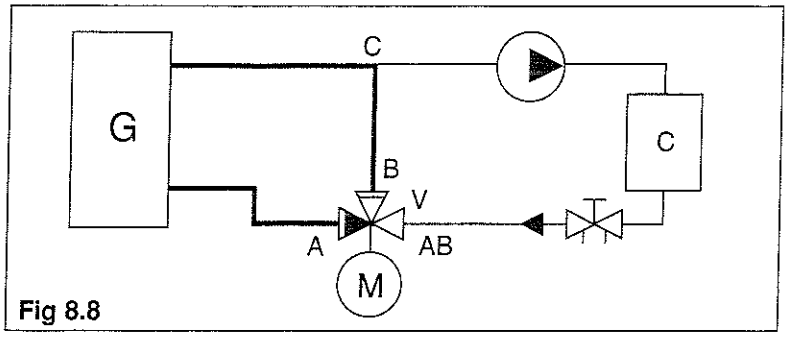


Fig 8.8

Fig 8.8 Diverting three way valve

Port A: Pipes AC + pressure drop across G.

Port B: CB.

$$\beta = \frac{\Delta pV}{\Delta pV + \Delta pG + \Delta pAC}$$

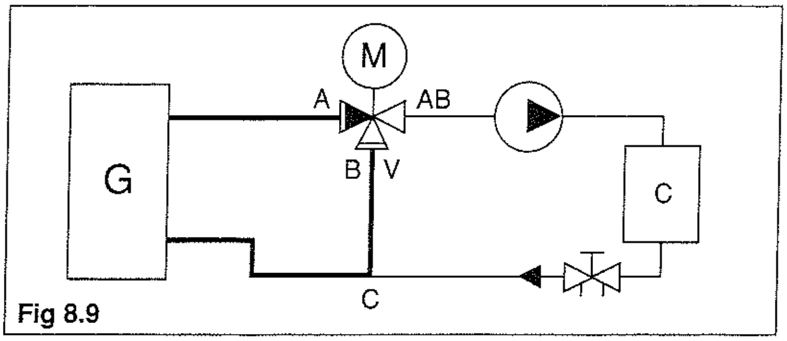


Fig 8.9

Fig 8.9 Mixing three way valve

Port A: Pipes CA + pressure drop across G.

Port B: CB.

$$\beta = \frac{\Delta pV}{\Delta pV + \Delta pG + \Delta pAC}$$

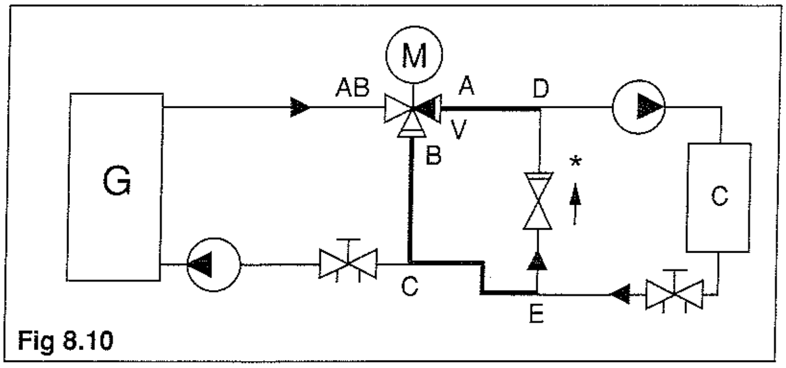


Fig 8.10

Fig 8.10 Diverting three way valve

Port A: Pipes AD + CE.

Port B: BC.

$$\beta = \frac{\Delta pV}{\Delta pV + \Delta pAD + \Delta pCE}$$

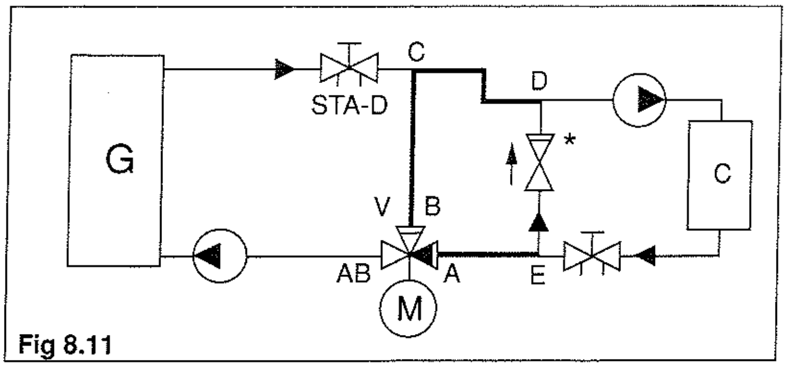


Fig 8.11

Fig 8.11 Mixing three way valve

Port A: Pipes AE + CD.

Port B: CB.

$$\beta = \frac{\Delta pV}{\Delta pV + \Delta pAE + \Delta pCD}$$

For the marked parts the pressure drops in Fig's 8.10 and 8.11 are relatively small. The authority of 3-way valves is therefore often close to 1. But to maintain correct characteristic in control valve V, don't select it for a Δp below 3 kPa.

* Check valve is only required when the load is a preheating coil to obtain a freezing protection if the secondary pump fails.

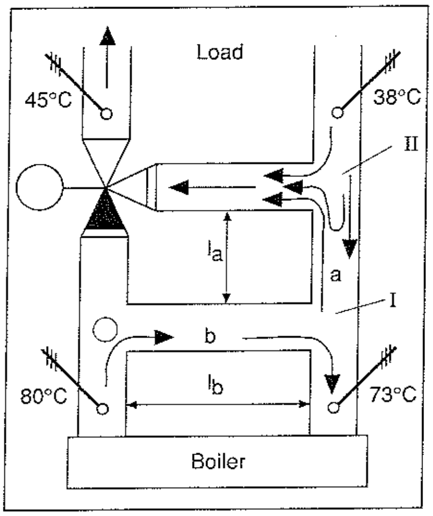


Fig 8.12

Bidirectional circulation

This phenomenon can occur in bypass circuits, if insufficient care was taken during dimensioning. Parasitic convection can cause the following difficulties, e.g. in circuits with thermostat valves:

- Hot spots
- Damaged thermostat
- Hot water in return water pipe

When the mixing port, e.g. of a 3-way valve, is closed, bidirectional circulation can occur, causing parasitic convection. Fig. 8.12 illustrates thermal currents (at I and II), when the valve is closed.

Bidirectional circulation and parasitic convection can be avoided if, during installation, it is ensured that the hot water connection is at a higher level than the cold water connection. Heat traps interrupt bidirectional circulation which, in turn, stops parasitic convection. The shaded pipes in Fig. 8.13 show examples of heat traps.

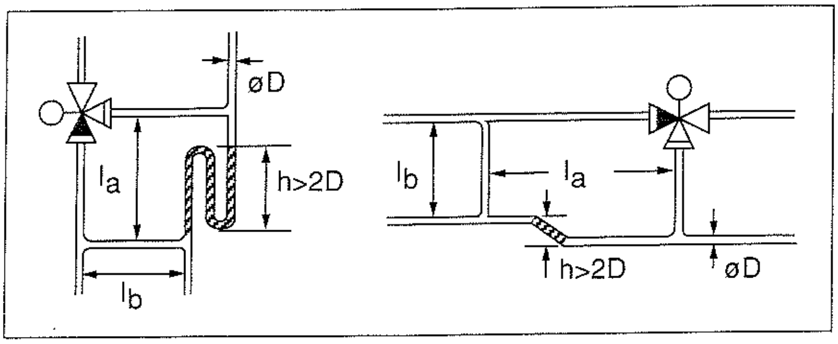
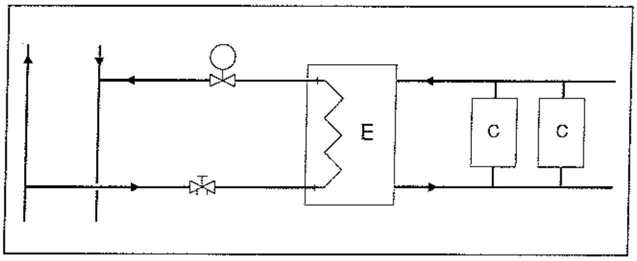


Fig 8.13 Prevention of bidirectional circulation

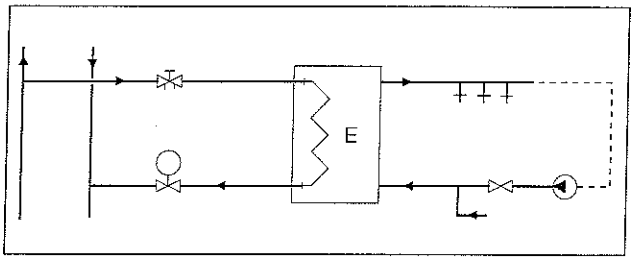
The primary parameter for bidirectional circulation, is the ratio between the distance between the pipes and the pipe diameter.

$l_a > 8 \cdot D$ to $10 \cdot D$

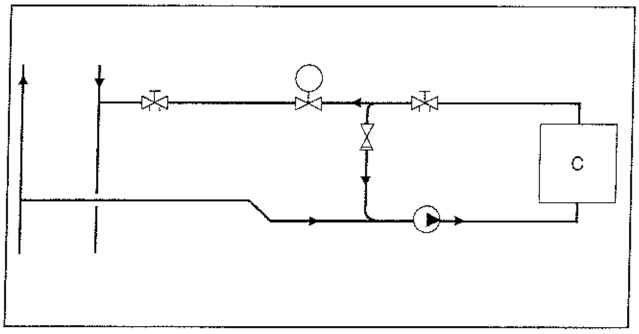
$l_b > 5 \cdot D$



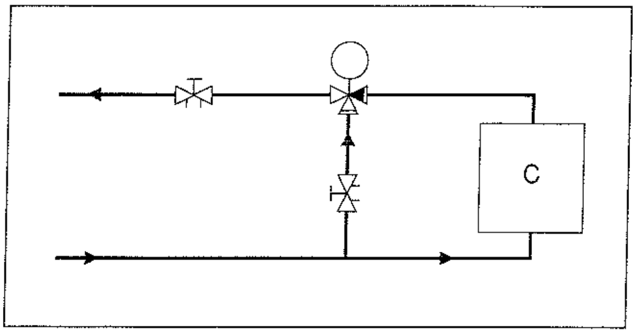
SYSTEM 1
Heating system connected to district heating network



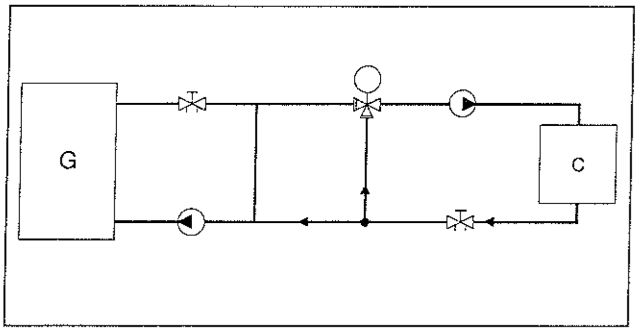
SYSTEM 2
Domestic hot water system connected to district heating network



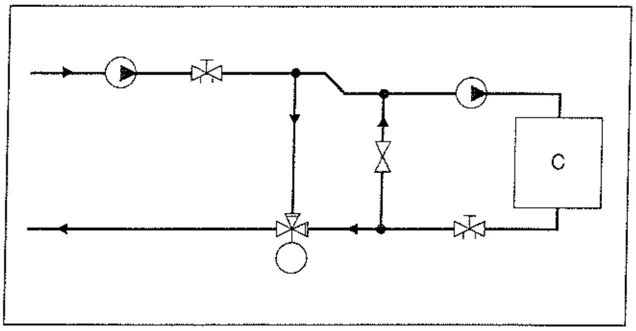
SYSTEM 3
Preheating coil which can be subject to freezing



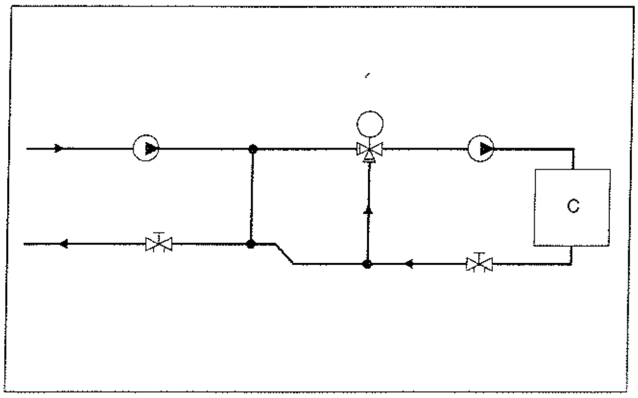
SYSTEM 4
Reheating system in cases not subject to freezing



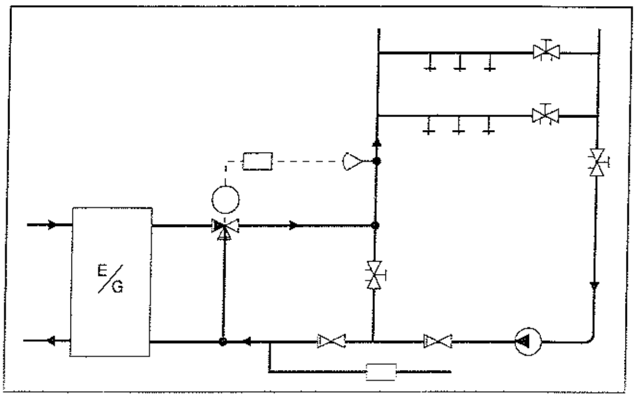
SYSTEM 5
Radiator circuit connected to boiler heating plant



SYSTEM 6
System with constant primary and secondary flows



SYSTEM 7
System with constant primary and secondary flows



SYSTEM 8
Domestic hot water circuit

System 1

2-way valve with primary pump water/water

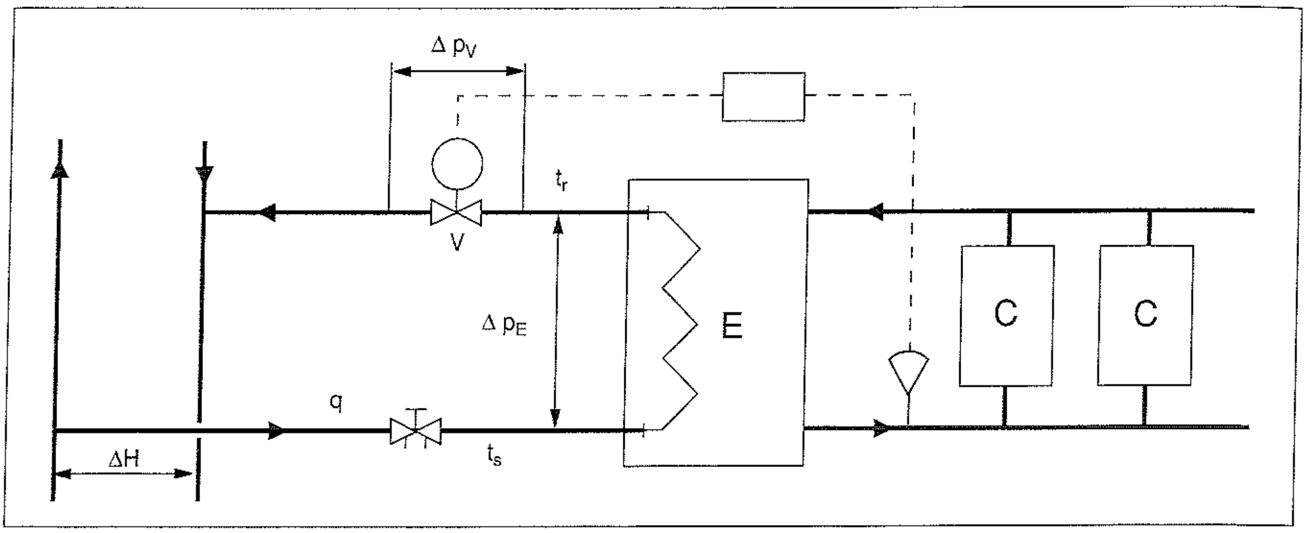


Fig 8.14 Heat exchanger, heating system

Functions and characteristics

- Flow control.
- Heating system (radiator groups and air heaters) connected to district heating network, with a requirement on low return temperature.
- Heat exchanger between primary and secondary circuits, is required if static pressure and temperature on primary side are incompatible with equipment in secondary circuit.
- Small air heaters, not subjected to freezing.

Valve sizing

$$\Delta p_V = \Delta H - \Delta p_E$$

$$K_V = \frac{36 \cdot q}{\sqrt{\Delta p_V}} \quad \text{(kPa, l/s)}$$

$$\beta = \frac{\Delta p_V}{\Delta H} \geq 0.5$$

Flow characteristic: Eq% (logarithmic)

System 2

2-way valve with primary pump, water/ (domestic) water

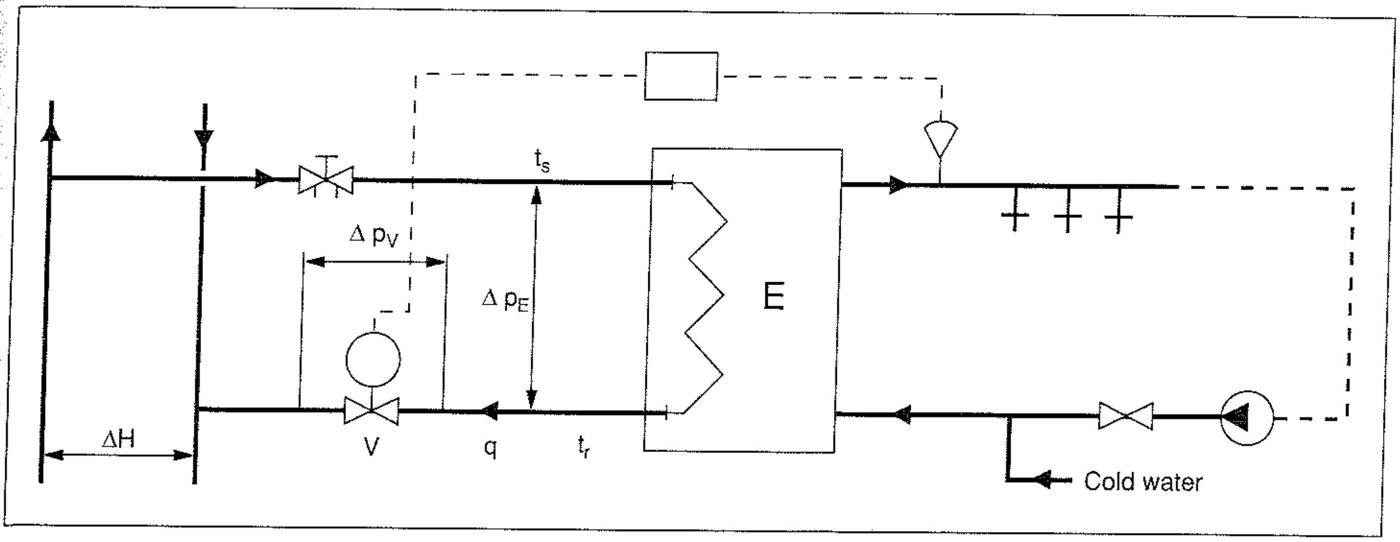


Fig 8.15 Heat exchanger, hot water

Functions and characteristics

- Flow control.
- Throttling away of excess pressure
- Domestic hot water system connected to district heating network
- System with requirements on low primary return temperature.

Valve sizing

$$\Delta p_V = \Delta H - \Delta p_E$$

$$K_V = \frac{36 \cdot q}{\sqrt{\Delta p_V}} \quad \text{(kPa, l/s)}$$

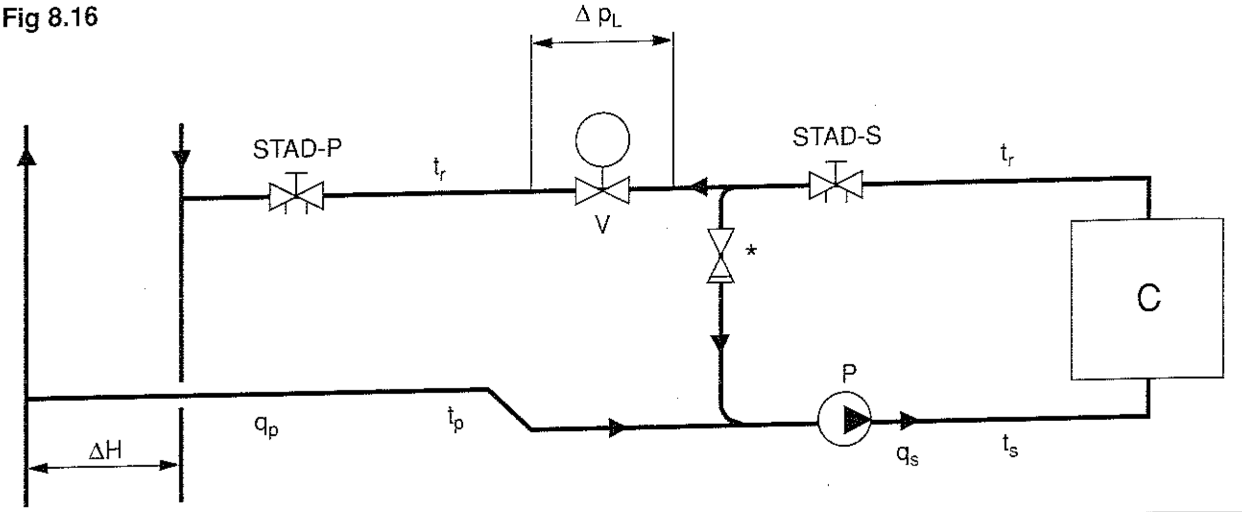
$$\beta = \frac{\Delta p_V}{\Delta H} \geq 0.5$$

Valve characteristic: Eq% (logarithmic)

System 3

2-way valve, with primary pump

Fig 8.16



Functions and characteristics

- Primary circuit: flow control, constant temperature
- Secondary circuit: temperature control, constant flow
- Connected to district heating network, with a requirement on low return temperature
- Heating installation, with long pipe runs
- Large air heaters, not subjected to freezing

* In the case of preheating coil, subjected to freezing, a check valve should be installed in the bypass line. If the pump stops, the water will be forced to circulate through the air heater, which will eliminate any freeze risk.

Valve sizing

Thermal equilibrium

$q_p \cdot (t_p - t_r) = q_s \cdot (t_s - t_r)$

Dimension the pump for the flow in the secondary circuit, q_s, and the total pressure drop in the circuit.

$\Delta p_V \approx \Delta H$

The pressure drop in the pipes of the primary side is negligible.

$$K_V = \frac{36 \cdot q_P}{\sqrt{\Delta H}} \quad \text{(kPa, l/s)}$$

Flow characteristic

ΔH = 3–5 kPa Eq% (Logarithmic)
ΔH = 5–10 kPa Modified linear (MOD. LIN)

Balancing*

1. Close V.
2. Adjust STAD-S for the design flow, q_s.
3. Open V fully.
4. Adjust STAD-P, so that the designed flow q_p is obtained.

* A more detailed and comprehensive description of balancing concepts and procedures is given in TA's handbook "Total Hydronic Balancing".

System 4

3-way mixing valve with primary pump

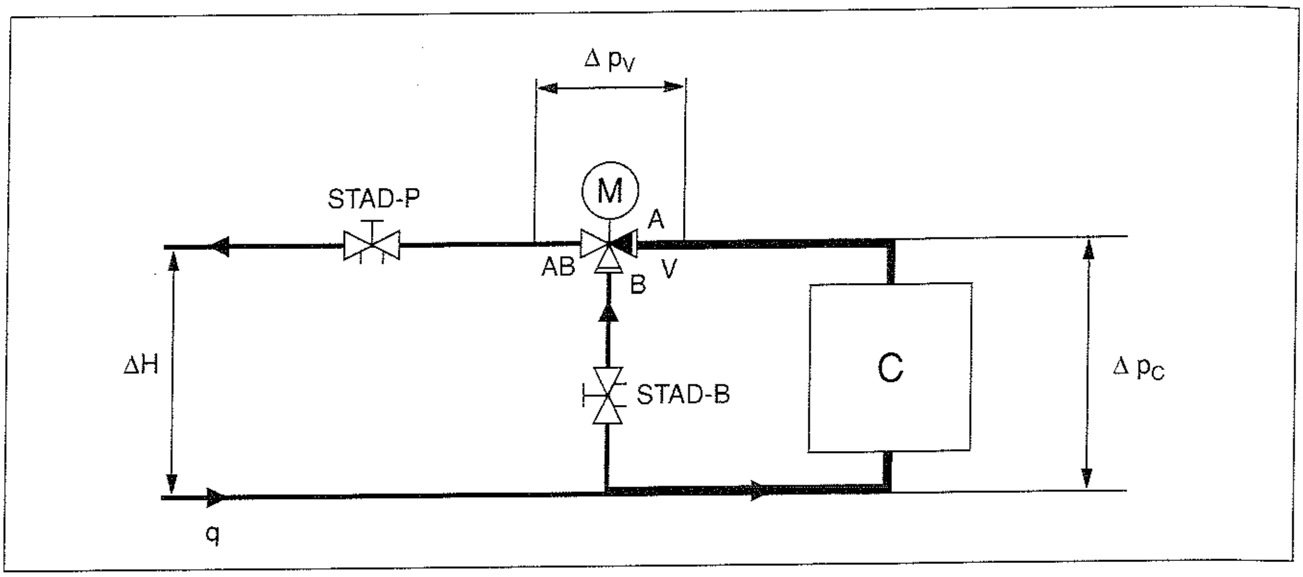


Fig 8.17

Functions and characteristics

- Primary circuit: constant flow
- Secondary circuit: variable flow, constant temperature
- The coil must not be subjected to freezing.
- With variable flow, this configuration does not provide an even temperature in the air coil.
- With small coils also risk for hunting at constant supply air temperature control.

Valve sizing

$\beta \geq 0.5 \quad \text{i.e.} \quad \Delta p_V \geq \Delta p_C$

$\Delta p_V = \Delta H - \Delta p_C$

$K_V = \dfrac{36 \cdot q}{\sqrt{\Delta p_V}}$ (kPa, l/s)

Flow characteristic:

A – AB = EQ% (logarithmic)

B – AB = linear (LIN)

Balancing*

1. Open port A–AB fully.
2. Reduce the flow with balancing valve STAD-P, to obtain designed flow.
3. Open port B-AB fully.
4. Adjust balancing valve STAD-B, to obtain the designed flow through STAD-P.

* A more detailed and comprehensive description of balancing concepts and procedures is given in TA's handbook "Total Hydronic Balancing".

System 5

Boiler, 3-way mixing valve

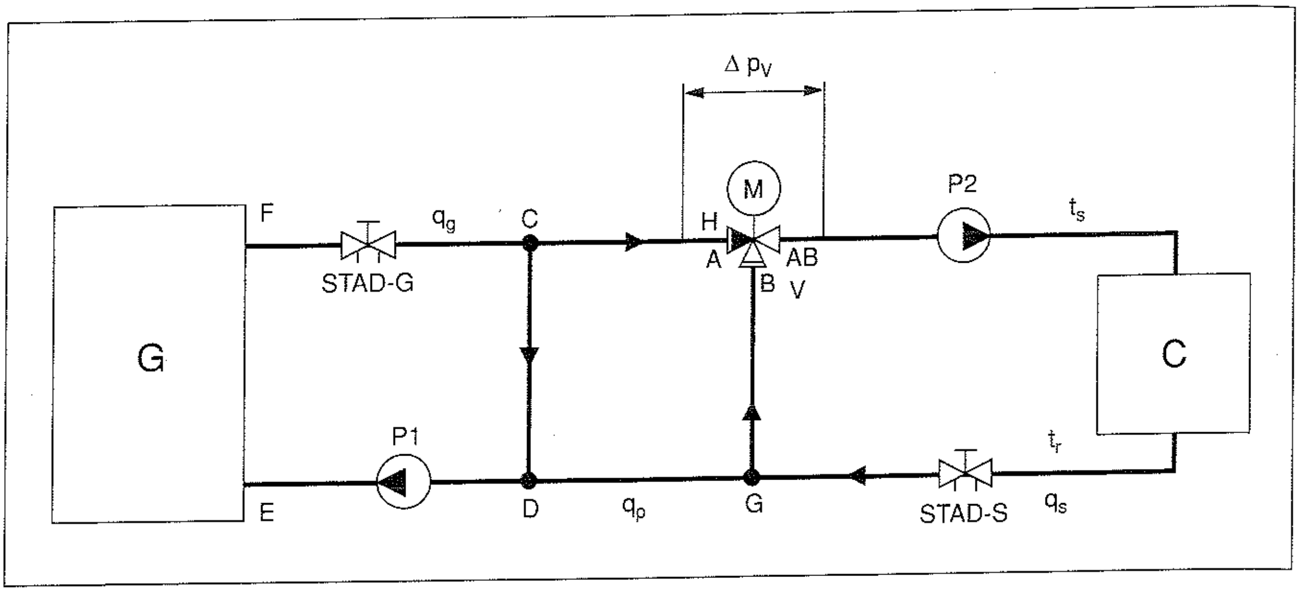


Fig 8.18 Constant flow in boiler

Functions and characteristics

- Primary circuit: variable flow, constant temperature
- Secondary circuit: constant flow, variable temperature
- System with local boiler

Valve sizing

$\beta \approx 1$

$\Delta p_V > \Delta_{GD} + \Delta_{CH}$

not less than 3 kPa

$$K_V = \frac{36 \cdot q_S}{\sqrt{\Delta p_V}} \quad \text{(kPa, l/s)}$$

Flow characteristic

Modified linear (MOD.LIN)

The resistance of pipe CD is considered to be negligible.

Balancing*

1. Open port A-AB fully.
2. Adjust balancing valve STAD-S, so that designed flow q_s is obtained.
3. Adjust balancing valve STAD-C, so that the flow q_g is slightly higher than the flow q_s.

* A more detailed and comprehensive description of balancing concepts and procedures is given in TA's handbook "Total Hydronic Balancing".

System 6

System with constant flows in primary and secondary circuits

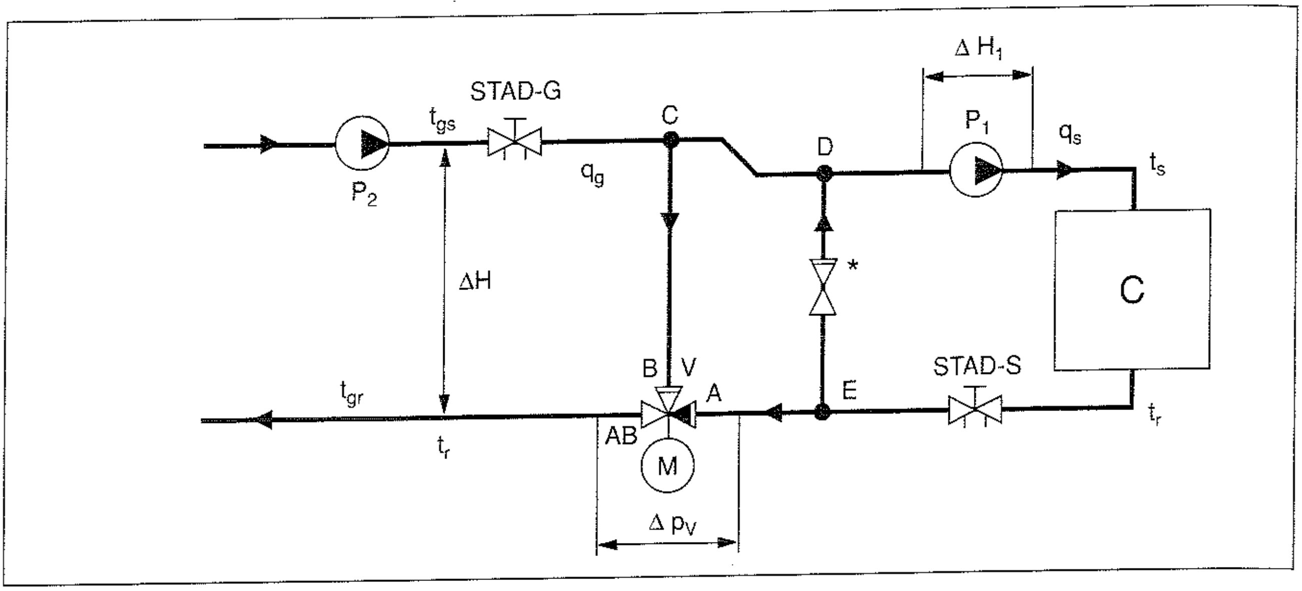


Fig 8.19 Coil in air handling unit

Functions and characteristics

- Primary circuit: constant flow, temperature control
- Secondary circuit: constant flow
- This configuration is used for large air cooling and heating coils
- It is suitable for connection to large boilers, where each object is individually controlled.

$$q_g \cdot (t_{gs} - t_{gr}) = q_s(t_s - t_r)$$

$$q_{gs} < q_s$$

$$t_{gs} > t_s$$

* The check valve in the secondary circuit forces the flow to pass through the coil, even if the pump in the secondary circuit has stopped (eliminates freezing hazard).

Valve sizing

Pipe sections C–D and E–A are the part of the pipe network, in which the flow is affected by the valve.

Valve authority, $\beta = 1.0$.

Flow characteristic for V:

$\Delta p_V > 3$ kPa (MOD.LIN – MOD.LIN) (modified linear)

$$K_V = \frac{36 \cdot q_S}{\sqrt{\Delta p_V}} \quad \text{(kPa, l/s)}$$

Balancing*

1. Close port A–AB of V and start pumps, $P1$ and $P2$.
2. Adjust STAD-S, so that the flow through the coil is correct.
3. Open port A–AB of V fully.
4. Adjust STAD-G, so that the designed primary flow is obtained.

* A more detailed and comprehensive description of balancing concepts and procedures is given in TA's handbook "Total Hydronic Balancing".

System 7

System with constant primary and secondary flows

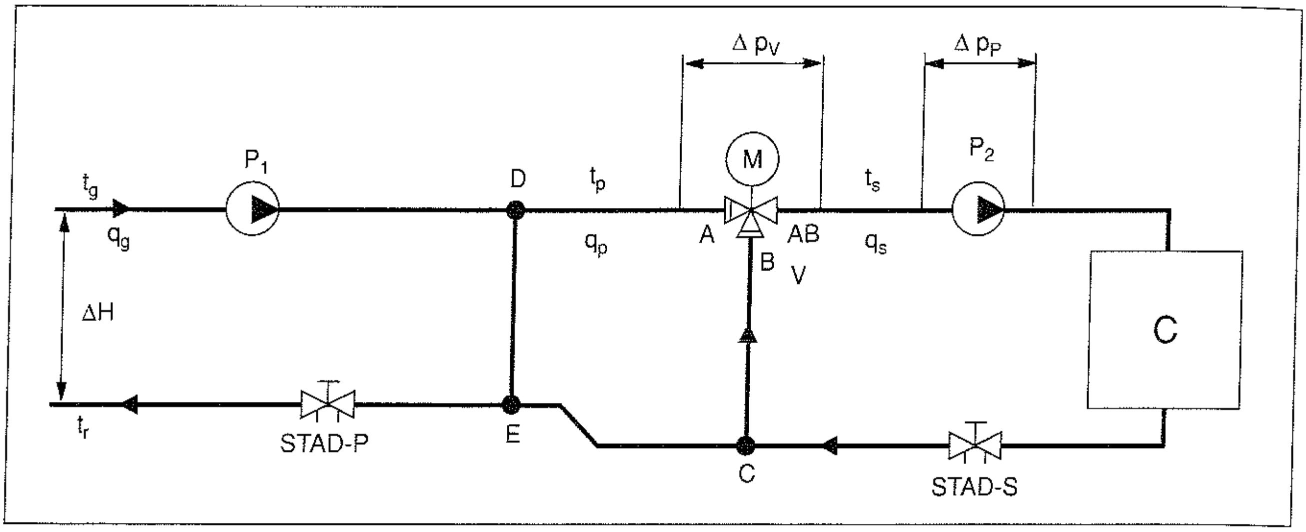


Fig 8.20

Functions and characteristics

- Primary circuit: constant flow, constant temperature
- Secondary circuit: constant flow, variable temperature
- Pressure changes in the primary circuit do not affect the secondary circuit, which also means that the secondary circuit cannot affect the primary circuit.
- This configuration is used for large systems, with multiple mixing valve – bypass groups.

Valve sizing

Pipe section D–E is the part of the pipe network, in which the flow is affected by the valve. The pressure drop in D-E is negligible, which means that the authority of the valve, $\beta = 1$, but the valve must be designed for a pressure drop of at least 3 kPa.

Flow characteristic for V

$\Delta p_V \geq 3$ kPa (MOD.LIN – MOD.LIN) (modified linear)

$$K_V = \frac{36 \cdot q_S}{\sqrt{\Delta p_V}} \quad \text{(kPa, l/s)}$$

Balancing*

1. Close port A–AB.
2. Adjust STAD-S, so that the flow through the coil is correct.
3. Open port A–AB fully.
4. Adjust STAD-P, so that the designed flow is obtained.

* A more detailed and comprehensive description of balancing concepts and procedures is given in TA's handbook "Total Hydronic Balancing".

System 8

Domestic hot water system for multi-family buildings. 3-way mixing valve DHWC pump

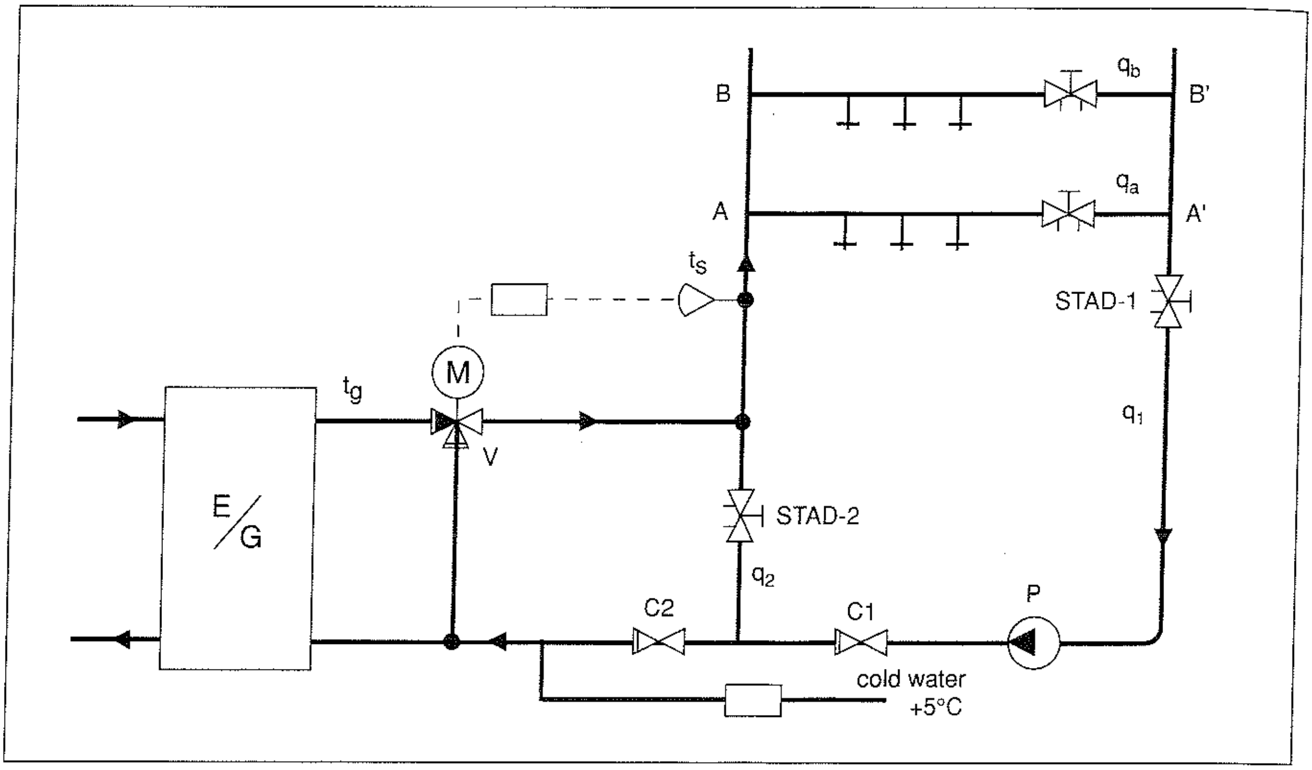


Fig 8.21

Functions and characteristics

- Hot water from the water heater is mixed with cold water in the 3-way valve, V.
- To compensate for heat losses in the pipes, a minimum flow q_1 is pumped back to the heat exchanger, through the loop.

Valve sizing

The following example applies for Swedish conditions.

Assume N standard apartments, each with an area of 65 m^2. Normal DHW demand per apartment, q_n, = 0.8 l/s.

Standard flow = N · q_n l/s.

The probable demand, q_e (l/s), can be obtained from the graph of Fig. 8.22.

Dimension the 3-way mixing valve for q_e and for a pressure drop, ΔpV = 10–80 kPa.

Convert the flow, q_e, to m^3/h:

$$q = q_S \cdot \frac{3600}{1000} \ \ m^3/h$$

$$K_V = \frac{q}{\sqrt{\Delta p_V}}$$

Suitable characteristics for the mixing valve are Eq% (logarithmic) for the boiler/heat exchanger port, and linear characteristic for the cold water port.

Balancing*

Recirculation flow in the different branches A-A', B-B', ... etc. is balanced so designed flows q_a, q_b, ... etc. are obtained. Circulation flows should be larger in the branches furthest away due to heat losses. STAD-1 is used as a partner valve in the balancing procedure.

When the total cirulation flow is obtained in STAD-1, the three way mixing valve, V, should be balanced with STAD-2, see DHWC pump, page 111.

* A more detailed and comprehensive description of balancing concepts and procedures is given in TA's handbook "Total Hydronic Balancing".

Sizing diagram for mixing valve in DHW systems, based on the number of standard apartments (Swedish standard).

Example
50 standard apartments. The probable water flow, q_s, = 1.8 L/s. Select a pressure drop across the mixing valve V = 40 kPa. In highrise buildings lower pressure drops across the mixing valve are selected, compared to buildings of normal height. Select a 3-way valve with K_v = 10. Flow characteristic: Symmetrical quadratic or Eq.%-Lin. (Eq.% towards the hot water port).

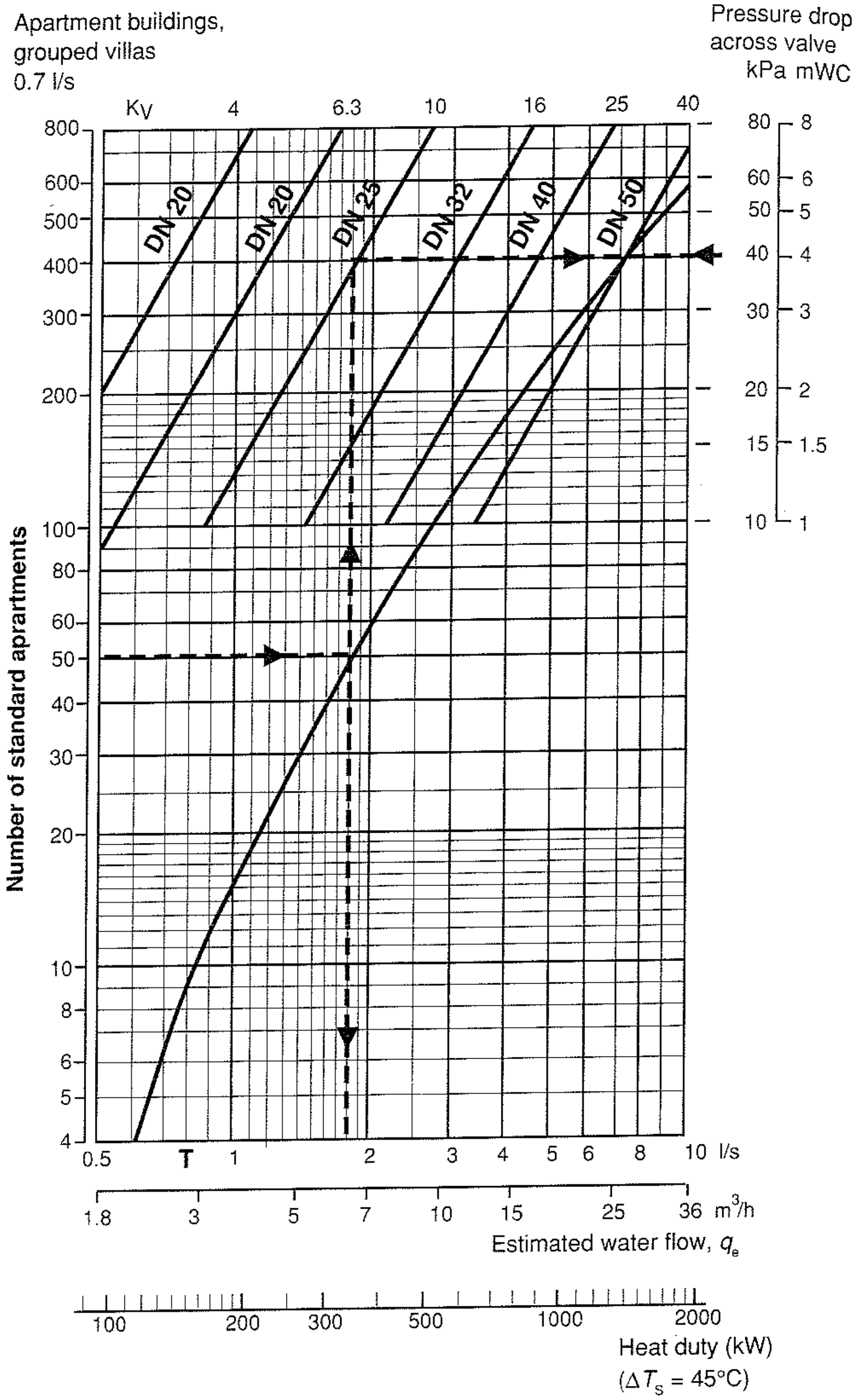


Fig 8.22 Estimated water flow in distribution pipe in dwellings and office buildings

Control of domestic hot water (DHW)

When controlling DHW, the fundamental difficulty lies in the considerable flow variations. Combined with this is the requirement for nearly constant temperature resulting in a requirement for exacting performance of the control equipment.

The properties of the control valve are also of great importance to obtain good DHW control. By selecting a correct flow characteristic (see page 116) good control performance is obtained for all loads. During periods when there is only DHWC flow in the system (typically during night-time) a valve with a great rangeability further enhances the DHW control performance (see page 115).

In district heating systems, improved control accuracy means that accumulated excess temperature in the heat exchanger is reduced and the return temperature to the district heating plant is lower.

DHWC pump

Cold water is heated from about 5°C, to the DHW temperature, $t_2 = 50 - 55$°C (maximum, according to the standard in force is 65°C). If there is no hot water usage for a long period of time, e.g. during the night, the hot water in the hot water pipe gets cold. The person who first uses hot water in the morning will have to let the water flow for a long time before it is hot. In apartment buildings with long pipe runs, a domestic hot water circulating (DHWC) pump is therefore installed which circulates a small flow in the pipe system.

The DHWC flow is between 5–10 % of the dimensioning domestic hot water flow. When the DHWC flow has returned to the boiler or heat exchanger, the temperature has dropped by about 3–10°C, depending on the DHWC flow rate, the pipe insulation and length of the pipes in the circuit. An acceptable typical DHWC heat loss is about 5–10 W/m of pipe. In a single family home pipes normally are so short that there is no need for a DHWC pump.

To compensate for the temperature drop, ΔT_{DHWC}, the flow, q, is split into q_1 and q_2,

$$q_1 \cdot t_1 + q_2 \cdot (t_2 - \Delta T_{DHWC}) = q \cdot t_2$$

Example

$t_2 = 50°$

$\Delta T_{DHWC} = 5°$

The flow, q, then will be split into $q_2 = 90\%$ and $q_1 = 10\%$.

To compensate for heat losses in the DHW pipe, adjust STAD-2 so that 10% of the DHW flow passes through the heat exchanger.

DHWC pump flow

DHWC heat loss rate, P_{DHWC}:

$$P_{DHWC} = q_{DHWC} \cdot \Delta T_{DHWC} \cdot 4.18 \text{ [kW]}$$

q_{DHWC} is in l/s

Required pumping head, p_{DHWC}:

$$p_{DHWC} = \frac{R \cdot L}{1000} \text{ [kPa]}$$

R = Pressure drop in the DHWC pipe [Pa/m]
L = Length of the DHWC pipe [m]

Select a suitable pump for:

- flow, q_{DHWC} [l/s]
- pumping head, p_{DHWC} [Pa].

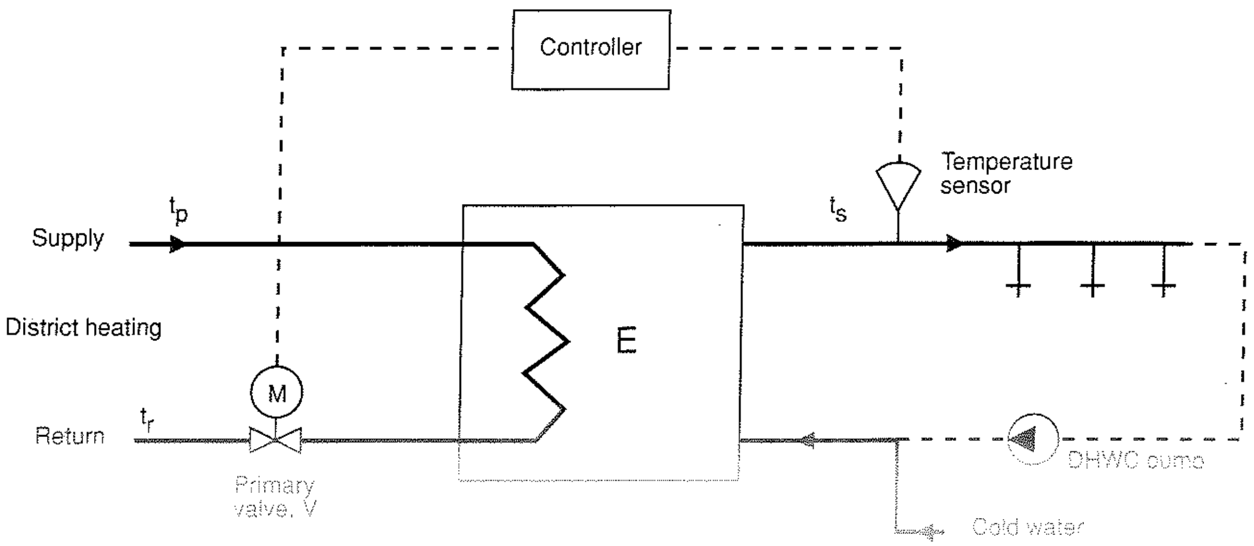
Domestic hot water with primary control valve

In domestic hot water systems, connected to district heating networks, typically the following components are found:

- A *heat exchanger* separates the primary from the secondary side.
- A *temperature sensor* senses the temperature of the outgoing domestic hot water.
- A *domestic hot water controller* compares the signal from the temperature sensor with a setpoint and actuates the primary control valve.
- A *primary control valve* throttles the primary flow to the heat exchanger.
- A *DHWC pump* guarantees that there is at all times a certain minimum flow in the circuit.

In district heating systems, with the control valve on the primary, great demands are made on the control valve's ability to control small flows, to compensate for the DHWC heat losses. The control valve therefore must have a large rangeability, R.

Fig 8.23



The temperature drop across the primary heat exchanger is, for small DHWC flows, about 45°C in the summer and about 60°C in the winter. The primary control valve is sized for the small pressure drop across the subscriber station, prevailing during the summer, about 150 kPa.

In the winter, when there is a heat demand also for heating of the building, the pressure drop across the primary is increased up to 600 kPa, sometimes even greater. This means that in the winter, the flow capacity of the control valve is oversized by a factor of about 3.

Required rangeability of the primary control valve

The primary control valve is sized for summer conditions:

$\Delta T_{prim} = 45\ °C$

$\Delta T_{sec} = 50 - 5 = 45\ °C.$

Dimensioning primary flow:

$q_P \cdot 45 = q_S \cdot (50 - 5)$

$q_P = q_S$

q_S = probable water flow (see page 110).

$$K_V = \frac{36 \cdot q_P}{\sqrt{\Delta p}} \quad \text{(kPa, l/s)}$$

NOTE: q_p is expressed in m^3/h

Primary flow to compensate for the DHWC losses:

The DHWC flow is 10 % of the secondary flow and the temperature drop in the DHWC pipe is about 5°C.

$q_{PDHWC} \cdot 45 = q_S \cdot 0.1 \cdot 5$

$q_{PDHWC} = 0.011 \cdot q_S$

$$\text{Rangeability, } R = \frac{q_P}{q_{PDHWC}} = \frac{1}{0.011} = 90.9$$

In the winter there will be a 4-fold increase of the pressure drop across the subscriber station. The temperature drop across the primary will then increase, while the DHWC losses remain unchanged. This means that the primary valve is oversized by a factor of 3, and should therefore be sized for a rangeability, R, ≈ 300.

Control valves have as standard a rangeability, R, ≈ 100. The need for a greater rangeability can be satisfied by installing two valves in a parallel split-range arrangement, one valve with a $K_V \sim 1/3$ and the other with a $K_V \sim 2/3$ of the required K_V value. See page 110. With a split-range valve can in this way a sufficiently great rangeability be obtained to handle the DHWC heat losses.

NOTE: If a primary valve does not have a sufficiently great rangeability, the system could enter into a state of hunting, when only the small DHWC flow circulates in the system. This could cause rapid wear of actuators, packing boxes, etc.

Flow characteristic of the primary control valve

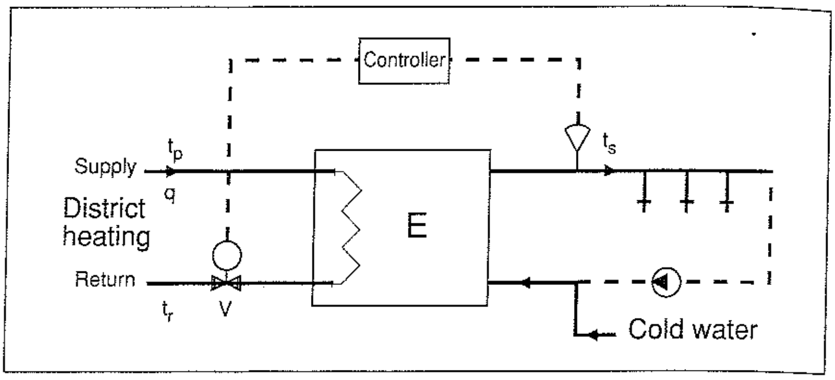


Fig 8.24 Direct acting DHW system

In heating and DHW systems the supply line temperature is controlled through the use of a temperature sensor. The temperature of the DHW must be kept constant within specified limits.

The transfer characteristic of a water-to-water heat exchanger is virtually constant within the pertinent flow range (refer to Fig's 1.9 and 1.10 in Part 1). The conclusion could be made, therefore, that the flow characteristic of the primary control valve should be linear. This is not the case, however. If a temperature offset (Δt_s) arises at the temperature sensor on the secondary, it is to be corrected by the primary control valve. Disregarding whether Δt_s occurs at full or at a small secondary flow, the lift of the valve stem required to correct the offset should be about equal. There should be a linear relationship between Δt_s and the stem lift.

Secondary

Heat rate, $P = q_S \cdot (t_{ww} - t_{cw})$; ($_{ww}$ and $_{cw}$ mean warm and cold water, respectively)

Assume q_S = constant

During an upset in the system, when for example t_{ww} or t_{cw} is changed, the upset is corrected by a change in P. A small change in t_S (i.e. small Δt_S) gives a small ΔP.

$\Delta P = q_S \cdot \Delta t_S$

Gain, G_S:

$$G_S = \frac{\Delta t_S}{\Delta P} = \frac{1}{q_S}$$

Boundary conditions:
For $P \approx 0$, q_S is small, i.e. G_S is large
For $P \approx 100\%$, q_S is large, i.e. G_S is small

Primary

$P = q_p \cdot (t_{in} - t_r)$
Assume $(t_{in} - t_r)$ = constant

Gain:

$$G_P = \frac{\Delta p}{\Delta q_P} = t_{in} - t_r$$

For $P \approx 0$ $\quad t_{in} \approx 90°C \quad t_r \approx 20°C$
$t_{in} - t_r \approx 70°C$

Conclusion: G_P is large

For $P = 100\%$

q_P = max.
$t_{in} = 90°C$
$t_r = 50°C$
$t_{in} - t_r = 40°C$

Conclusion: G_P is small

System gain, G_{syst}

$G_{syst} = G_S \cdot G_P$

$$G_{syst} = \frac{\Delta t_S}{\Delta E} \cdot \frac{\Delta P}{\Delta q_P} = \frac{t_{in} - t_r}{q_S}$$

Valve gain

$$G_V = \frac{\Delta q_V}{\Delta h}$$

Δq_v = flow change
Δh = stem lift change

Summary

The best control performance is obtained with a linear relationship between the temperature offset at the sensor (Δt_S) and the stem lift, Δh:

$$\frac{\Delta t_S}{\Delta h} = \text{constant}$$

The flow characteristic which best satisfies the above conditions is the EQ% (logarithmic) characteristic:

$$q_{prim} = e^{n(h-1)} = q_{max} \cdot R^{(h-1)}$$

The factor n should lie in the range 3.4 to 5.0. It is a function of:

- the variations in supply temperature, summer/winter
- the changes in the pressure differential, summer/winter
- type and characteristic of heat exchanger

9 Valve calculations

Formulas

Heating

Water	$P = 4.18 \cdot q_W \cdot \Delta T$
	$P = 1.16 \cdot q \cdot \Delta T$
Air	$P = 1.3 \cdot q_A \cdot \Delta T$
Steam	$G = 1.59 \cdot P$

Units and designations

$P = \text{kW}$
$q = \text{m}^3/\text{h}$
$q_W = \text{l/s}$
$q_A = \text{Nm}^3/\text{h}$
$G = \text{kg/h}$

Temperatures (standard values)

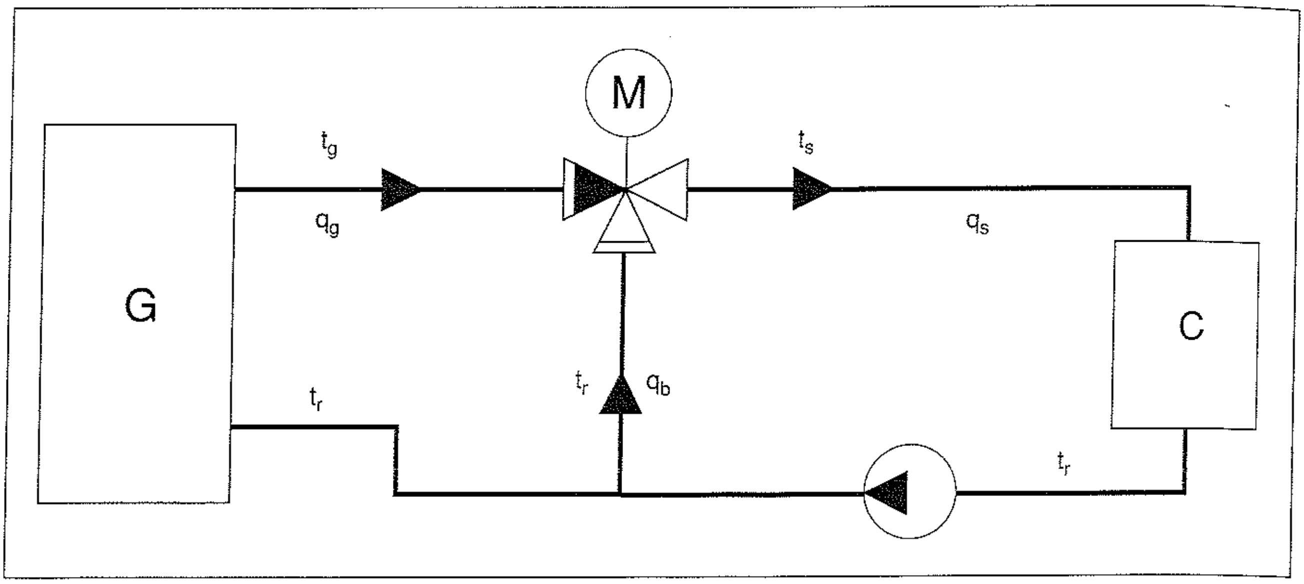
Heat exchanger, primary, district heating	$\Delta T = 40$ K
Heat exchanger, other	$\Delta T = 20$ K
Heat exchanger, radiators, low flow system	$\Delta T = 50$ K (80–30 K)
Heat exchanger, coolings coils	$\Delta T = 5$–10 K

Heating demands of dwellings

New buildings	40 W/m^2 living area
Well insulated buildings	50 W/m^2 living area
Not very well insulated buildings	60 W/m^2 living area
Poorly insulated buildings	100 W/m^2 living area
Cellars	15 W/m^2 living area

Fig 9.1 Calculation of leakage using the temperature method

Calculation of valve leakage, using the temperature method



$$\frac{q_g}{q_s} = \frac{t_s - t_r}{t_g - t_r} \qquad \frac{q_b}{q_s} = \frac{t_s - t_g}{t_r - t_g}$$

Valve flow coefficient

Liquid	$K_V = \frac{q \cdot \sqrt{\rho}}{\sqrt{\Delta p_V}}$	$C_V = 1.17 \cdot K_V$
Steam	CRITICAL PRESSURE DROP $\Delta p \geq 0.5 \cdot p_1$	SUB-CRITICAL PRESSURE DROP $\Delta p < 0.5 \cdot p_1$
Saturated steam	$K_V = \frac{G}{11.35 \cdot p_1}$	$K_V = \frac{G}{22.7 \cdot \sqrt{\Delta p \cdot p_2}}$
Superheated steam	$K_V = \frac{G \cdot k}{11.35 \cdot p_1}$ $k = 1 + 0.0012 \cdot t_S$	$K_V = \frac{G \cdot k}{22.7 \cdot \sqrt{\Delta p \cdot p_2}}$

Fig. 9.2

K_V = flow coefficient, m³/h, at Δp = 1 bar.
C_V = flow coefficient, US gallons/min, at Δp = 1 psi.
p_1 = pressure before valve, bars absolute.
p_2 = pressure after valve, bars absolute.
p_V = pressure drop across valve, $p_1 - p_2$, bar.

ρ = density, kg/dm^3 (note units).
q = liquid flow rate, m^3/h.
G = steam flow rate, kg/h
t_S = steam superheating temperature, °C.
k = correction factor for superheated steam.

Valves connected in parallel

$$K_V = K_{V1} + K_{V2} + K_V \ldots$$

Valves connected in series

$$\frac{1}{(K_V)^2} = \frac{1}{(K_{V1})^2} + \frac{1}{(K_{V2})^2}$$

General

When designing HVAC systems, often uncertainty exists regarding, the magnitude of the pressure drop across various components. The following information will suffice for rough estimates, although the manufacturers' specifications always should be consulted when making accurate calculations.

In the Stockholm area, for example, a design pressure drop of 150 kPa is specified for connection to district heating networks, although the available pressure drop will normally be many times greater. Inquire of your local authorities or energy utility concerning the specifications that apply for your area.

Guide for quick estimates

The following are commenly encountered pressure drops:

Δp_P = pressure drop on primary side of heat exchangers.
Δp_S = pressure drop on secondary side of heat exchangers.

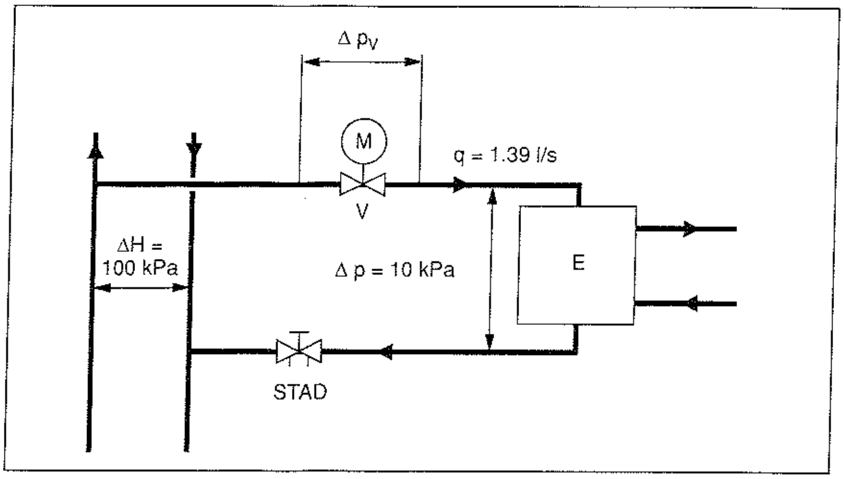
Water heater	(tap water)	Δp_P = 2–7 kPa, 20 kPa, max. Δp_S = 10–30 kPa, 50 kPa, max.
Heat exchanger Heat exchanger Heat exchanger	(radiator network) (air conditioning) (snow melting)	Δp_P = 20 kPa, max. Δp_S = 15 kPa, max.
Radiators without radiator valves		Δp = 0.5 kPa
Low flow systems with radiator valves		Δp = 10 kPa
Convectors		Δp = 5–20 kPa
Fan coils		Δp = 5–20 kPa
Heating/cooling coils		Δp = 5–20 kPa
Boilers	single family houses	Δp = 1–5 kPa
Boilers	apartment houses	Δp = 0.5–10 kPa
Water meter, district heating		$\Delta P_P \approx$ 15 kPa
Filters		$\Delta p \approx$ 15 kPa
Pipe resistance	copper pipe	$\Delta p \approx$ 0.2 kPa/m
Pipe resistance	steel pipe	$\Delta p \approx$ 0.4 kPa/m
Pipe resistance	total in a substation*	$\Delta p \approx$ 10 kPa

* District heating

Calculations

▬▬ = pipe network affected by the variable flow through the valve.

Valve sizing



To obtain a flow in the primary circuit of 1.39 l/s a pressure drop of 10 kPa is required. A pressure drop of 100 kPa is available. Calculate the flow coefficient, K_V, and the authority, β, of the valve.

Solution

$p_V = 100 - 10 = 90$ kPa

$$K_V = \frac{36 \cdot q}{\sqrt{\Delta p}} = \frac{36 \cdot 1.39}{\sqrt{90}} = 5.27 \text{ (kPa, l/s)}$$

$+40\% = 7.38$

$K_V = 5.27$

$-20\% = 4.2$

Select $K_V = 6.3$

Valve authority, β (see page 49):

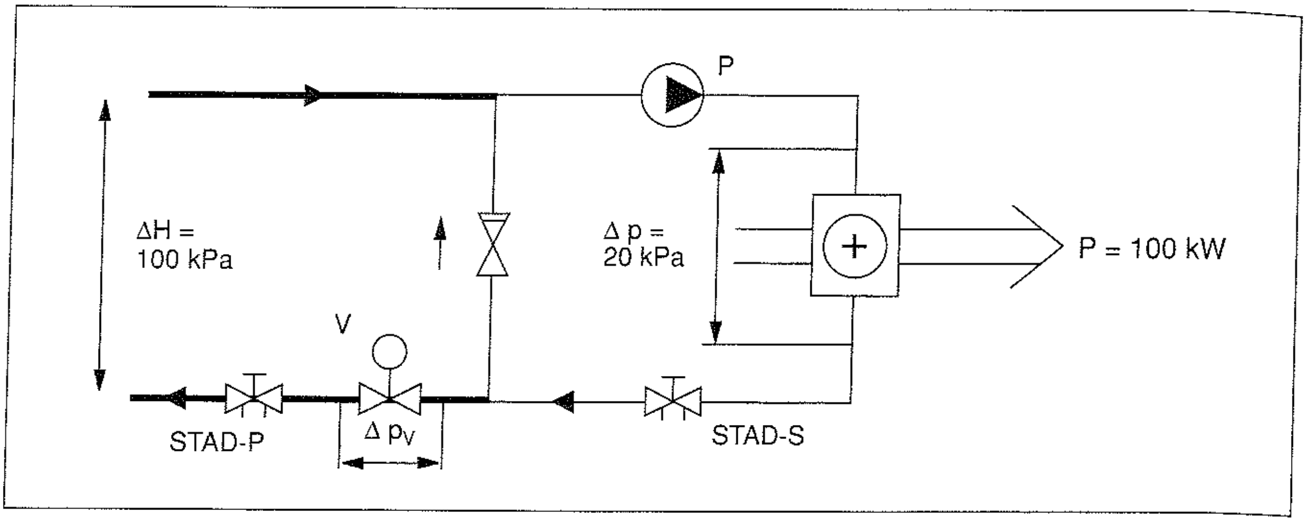
$K_V = 6.3 \qquad q = 1.39$ l/s

$$\Delta p_V = 90 \cdot \left(\frac{5.27}{6.3}\right)^2 = 63 \text{ kPa}$$

$\beta = \frac{63}{100} = 0.63$ (a good value as β should be > 0.5)

Pressure drop to create in the balancing valve

$\Delta H = \Delta p_V - \Delta p_E = 100 - 63 - 10 = 27$ kPa



An air preheater must deliver 100 kW.

- Sizing of V.
- Sizing of circulating pump, P.
- Calculate the authority of the valve.

Solution

Seek the flow, q.

$P = q \cdot \Delta T \cdot 1.16 \cdot \text{kW}$

$100 = q \cdot (100 - 35) \cdot 1.16$

$q = 1.3\ \text{m}^3/\text{h} = 0.37\ \text{l/s}$

The pump P should be dimensioned for the flow, $q = 1.3\ \text{m}^3/\text{h}$, and $\Delta p = 20$ kPa, plus the remaining pressure drops in the circuit. Select the nearest larger pump and compensate with STAD-S.

Control valve, V

The pump P provides a constant flow in the secondary circuit and overcomes the pressure drops in the secondary circuit. V should be dimensioned for the entire pressure drop, $\Delta p = 100$ kPa.

$$K_V = \frac{36 \cdot q}{\sqrt{\Delta H}} = \frac{36 \cdot 0.37}{\sqrt{100}} = 1.33 \text{ (kPa, l/s)}$$

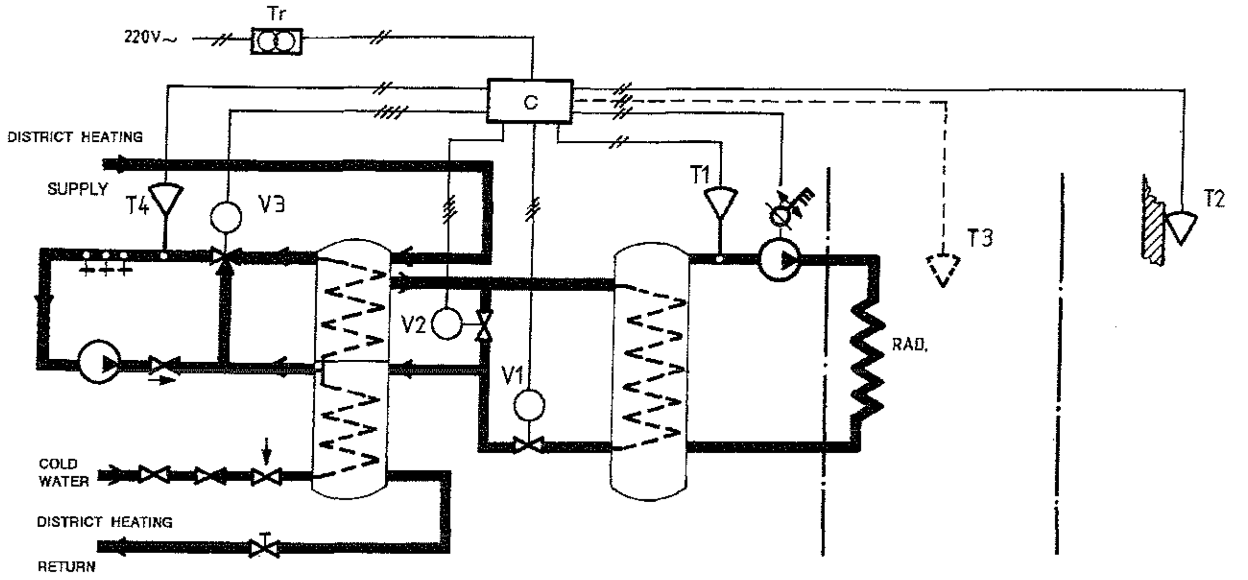
Valve authority, $\beta = 1.0$.

Choose right K_V

Select STA-P

Sizing of control valves in district heating substations

A substation supplying 50 standard-size apartments is connected to a district heating network, according to the diagram below. A 3-stage configuration is used, as the tap water is soft.



Function

Sensors T1 and T2 control supply line temperature via C and V1 according to the selected reset curve.

The circulating pump stops and V1 closes if outdoor temperature exceeds the value set in C.

Restart is computed by C.

High and low limiting of supply temperature are set in C.

Sensor T4 keeps DHW temperature constant via C, V2 and V3 (sequence control).

Night setback of radiator circuit and DHW are set in C.

Reference sensor T3 adjusts the selected reset curve and opimization time.

EQIPMENT SPECIFICATION

DESIGN.	DESCRIPTION	TYPE	REMARKS
C	CONTROL UNIT		
	Controller	TA 230U	
	Controller	TA 239 W	
	Auxilliary unit	TS	
	Auxilliary unit	RP	
V1	VALVE ACTUATOR	M15/24V	
V2, V3	VALVE ACTUATOR	EM5/24V	
T1	TEMPERATURESENSOR	EGW	
T2	TEMPERATURESENSOR	EGU	
T3	TEMPERATURESENSOR	EGRL/EGL	Reference sensor
T4	TEMPERATURESENSOR	EGK	
Tr	TRANSFORMER	YT 60	

RADIATOR CIRCUIT/DHW

85 – 115

Sizing data according to city of Stockholm power utility

Sizing data for heat exchangers and water heaters.

TABLE*	1 °C	2 °C	3 MPa
Radiator systems			
Primary side	120 – 65	100 – 50	1.6
Secondary side	80 – 60	60 – 45	0.6
Air heaters (*preheating or overall heating*)			
Primary side	120 – 45	100 – 35	1.6
Secondary side	80 – 40	60 – 30	0.6
Air heaters (*reheating, driers*)			
Primary side	65 – 35	65 – 35	1.6
Secondary side	55 – 30	55 – 30	0.6
Water heaters			
Primary side	65 – 25	65 – 25	1.6
Secondary side (*DHW*)	50 – 5	50 – 5	1.0

* **Table**
1 = Sizing data for old buildings.
2 = Sizing data for new construction, mixed old and new buildings.
3 = Maximum permissible pressure.

Use Table 2 for complete renovations, i.e. when new radiators are to be installed.

Pressure drop across installation

– Winter 150 – 600 kPa.
– Summer 150 – 400 kPa.

Static pressure

– Winter 1 300 – 1 600 kPa.
– Summer 1 000 – 1 600 kPa.

Primary valve

If the primary flow is greater than 5 l/s (18 m^3/h), two control valves should normally be installed, connected in parallel, and controlled in sequence. Stage 1 should pass one third of the flow and stage 2, two thirds of the flow.

NOTE: A type TA 239W controller + TS unit should be used for the sequential control (DHW) of primary valves. The mixing valve is controlled in sequence with the primary valve.

Domestic hot water

Dimension the primary and secondary DHW valves in a 3-stage configuration, connected to a district heating network. Dimension the DHW flow according to Swedish Standard BFS 1988.

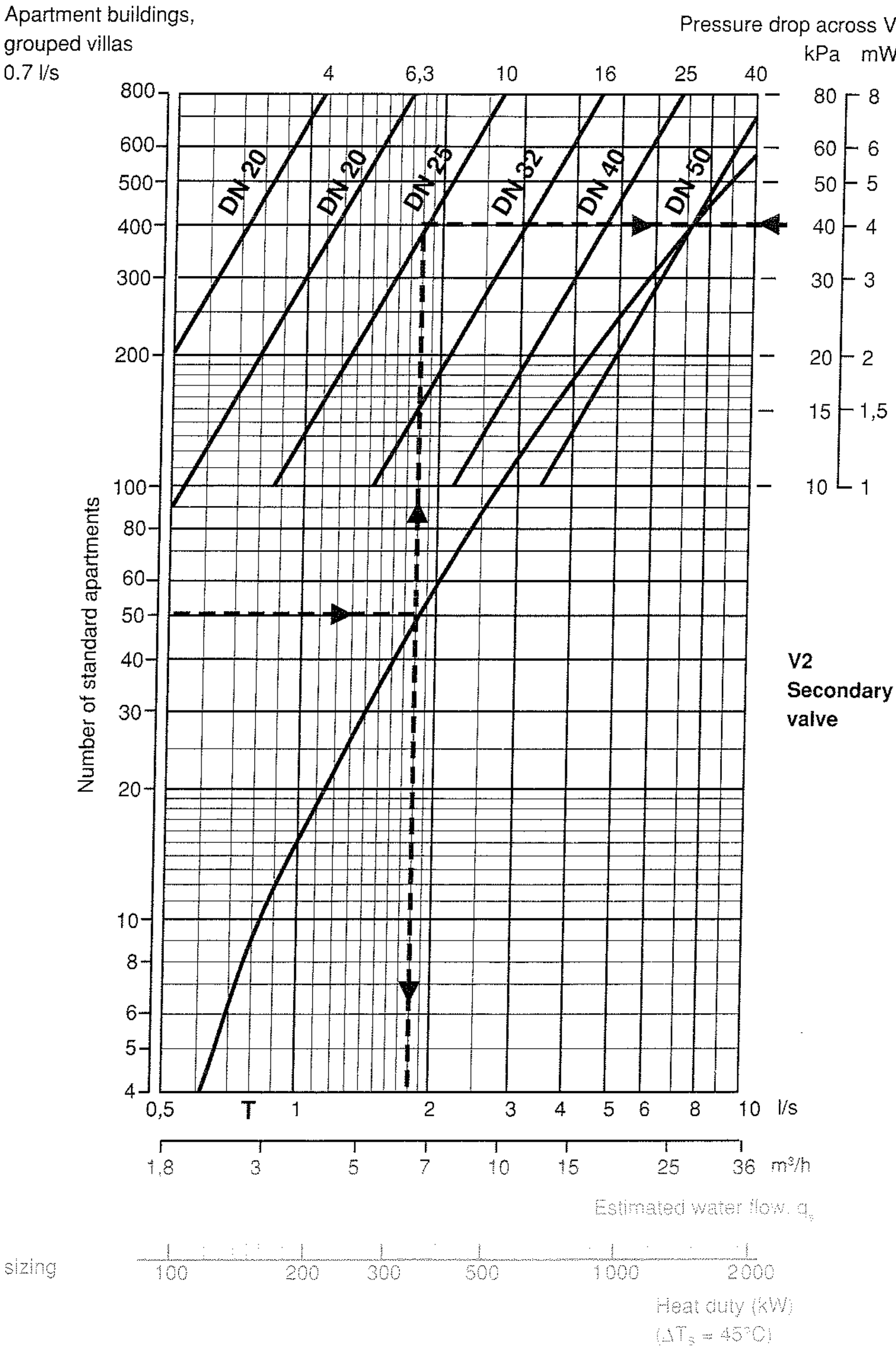
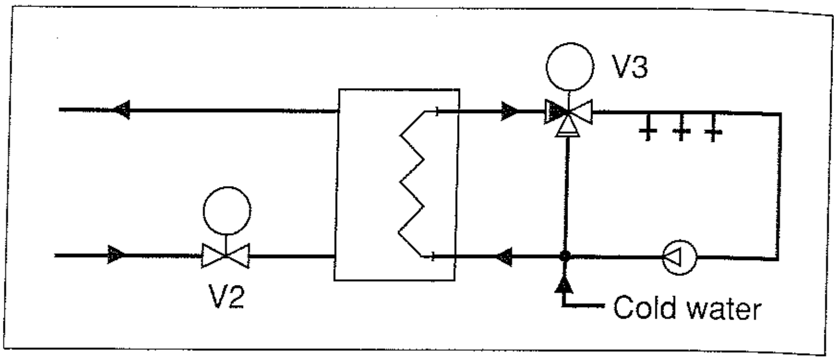


Fig 9.2 Valve sizing diagram



The estimated maximum water demand for 50 appartments, calculated for 50 appartments

Bath tub:	0.4 l/s
Kitchen sink:	0.2 l/s
Wash-basin:	0.1 l/s
Total	**0.7 l/s**

Standard flow = 50 · 0.7 l/s =35 l/s.

According to the diagram below, the estimated flow is:
$q_s = 1.8$ l/s.

Heat rate demand

From the diagram is obtained P = 319kW.

Mixing valve, V3

Select a pressure drop of 10 to 80 kPa across the valve, or $\beta = 50\%$. Assume $\Delta p = 40$ kPa.

In the case of high rise buildings, choose a lower pressure drop across the valve.

$$K_V = \frac{36 \cdot q}{\sqrt{\Delta p}} = \frac{36 \cdot 1.8}{\sqrt{40}} = 10.2 \quad \text{(kPa, l/s)}$$

Select a V386 valve, DN 32, $K_V = 10$.

Primary valve, V2

$P = q \cdot \Delta T \cdot 1.16$

$319 = q \cdot (65 - 25) \cdot 1.16$

$q = 6.9$ m³/h = 1.92 l/s

Pressure drop in the circuit

Water heater:	$\Delta p_p = 7$ kPa
Water meter + piping:	$\Delta p = 25$ kPa
Available differential pressure:	$= 150$ kPa

$$\Delta p_V = 150 - 7 - 25 = 118 \text{ kPa}$$

$$K_V = \frac{36 \cdot q}{\sqrt{\Delta p}} = \frac{36 \cdot 1.92}{\sqrt{118}} = 6.36 \text{ (kPa, l/s)}$$

Select a valve with K_v in the interval

$$K_V = 6.36 \quad \begin{matrix} +40\% \rightarrow 8.90 \\ -20\% \rightarrow 5.09 \end{matrix}$$

Select STL SR valve, DN 20, $K_V = 6.3$.

Valve authority, β

$\Delta p_{V100} = 118$ kPa

Available pressure drop: 150 kPa

$$\beta = \frac{p_{V100}}{\Delta p_{tot}} = \frac{118}{150} = 0.78$$

The valve can also be sized using a pressure drop diagram.

Caviation

When the valve is closed the pressure drop across the valve equals the total available differential pressure. The differential and static pressures will vary as follows:

Δp, winter: 150–600 kPa.
p_{stat} 1.3 – 1.6 MPa.

Δp, summer: 150–400 kPa.
p_{stat} 1.0 – 1.6 MPa.

Example

Winter: Δp_{max} = 600 kPa
p_{stat} = 1 300 kPa
temp. = 100 °C
No cavitation

Summer: Δp_{max} = 400 kPa
p_{stat} = 1 000 kPa
temp. = 65 °C
No cavitation

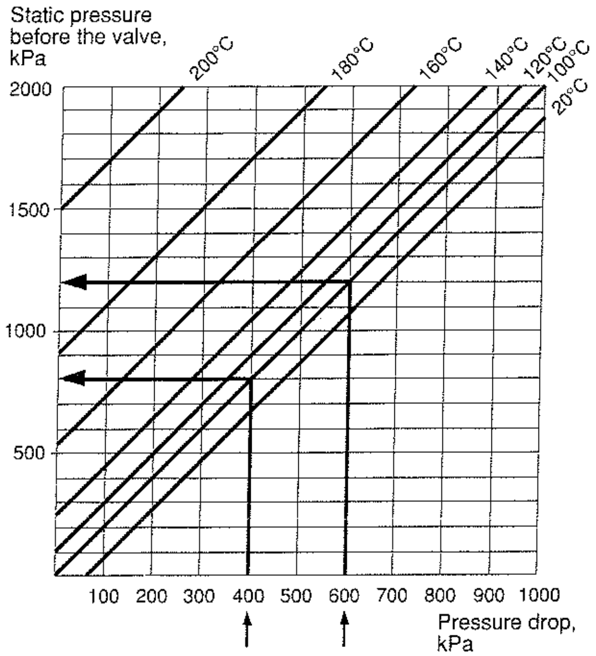


Fig 9.3 Cavitation diagram with pressure drops in the example in the text entered. For the pertinent temperatures, the static pressures fall well below the pressures where there is risk of cavitation.

Heating, radiator circuit

Heat demand

50 apartments, each with an average area of 65 m². A heating demand of 60 W/m² gives

$P = 50 \cdot 65 \cdot 60 = 195$ kW.

Similarly, a cellar area of 600 m², with a heating demand of 15 W/m², gives 9 kW.

$P_{tot} = 195 + 9 = 204$ kW

Radiator valve, V1

$P = q \cdot \Delta T \cdot 1.16$
$204 = q \cdot (100 - 50) \cdot 1.16$
$q = 3.5\ m^3/h = 0.97\ l/s$

Pressure drop in the circuit

Heat exchanger	$\Delta p_p = 35$ kPa
Water meter and piping	$\Delta p\ = 25$ kPa

$\Delta p_V = 150 - 35 - 25 = 90$ kPa

$$K_V = \frac{36 \cdot q}{\sqrt{\Delta p_V}} = \frac{36 \cdot 0.27}{\sqrt{90}} = 3.68\ \text{(kPa, l/s)}$$

$+ 40\% = 5.2$
Basic K_V value = 3.68
$- 20\% = 2.9$

Select: STL294 $K_V = 4.0$
EQ% (logarithmic) characteristic.

Valve authority, β

$$\beta = \frac{90}{150} = 0.6$$

Cavitation

See primary valve for domestic hot water.

Valve close off pressure

Can the primary valve close off the maximum differential pressure?

Is Δp_C (maximum permissible Δp across a closed valve) lower than the maximum value permitted by the combination of actuator, valve type and valve size? If not, the valve leakage will be excessive (> 0.05% of K_V).

From the valve specification sheet, the Δp_C value for some valve-actuator combinations are:

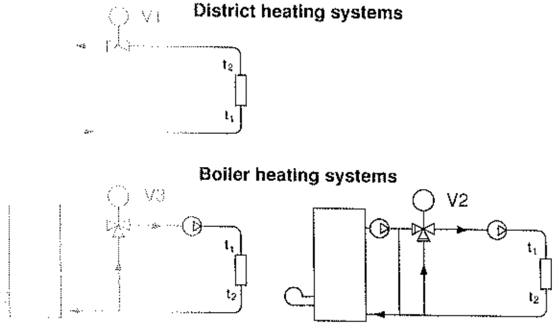
EM5C - STL 20	$\Delta p_C = 1\,600$ kPa
M5C - STL 25	$\Delta p_C = 1\,500$ kPa

These calculations can also be made using a sizing diagram, see next page.

Radiator systems

District heating systems

Boiler heating systems



Dimensioning pressure drop across control valve
V1 Data are obtained from heating utility company,
V2 Δp = 10 kPa, V3 Δp = 5 kPa.

Heating demand
- Poorly insulated buildings
- Less well-insulated buildings
- Well insulated buildings
- New construction
- Cellars

APPLICATION EXAMPLES

Radiator systems
Apartment buildings
- less well-insulated buildings; 60 W/m²
- 20 apartments, 65 m² each
- Δt = 20°C
- V2, Δp = 10 kPa

According to diagram:
- heat duty 78 kW
- water flow 0.93 l/s (3.3 m³/h)

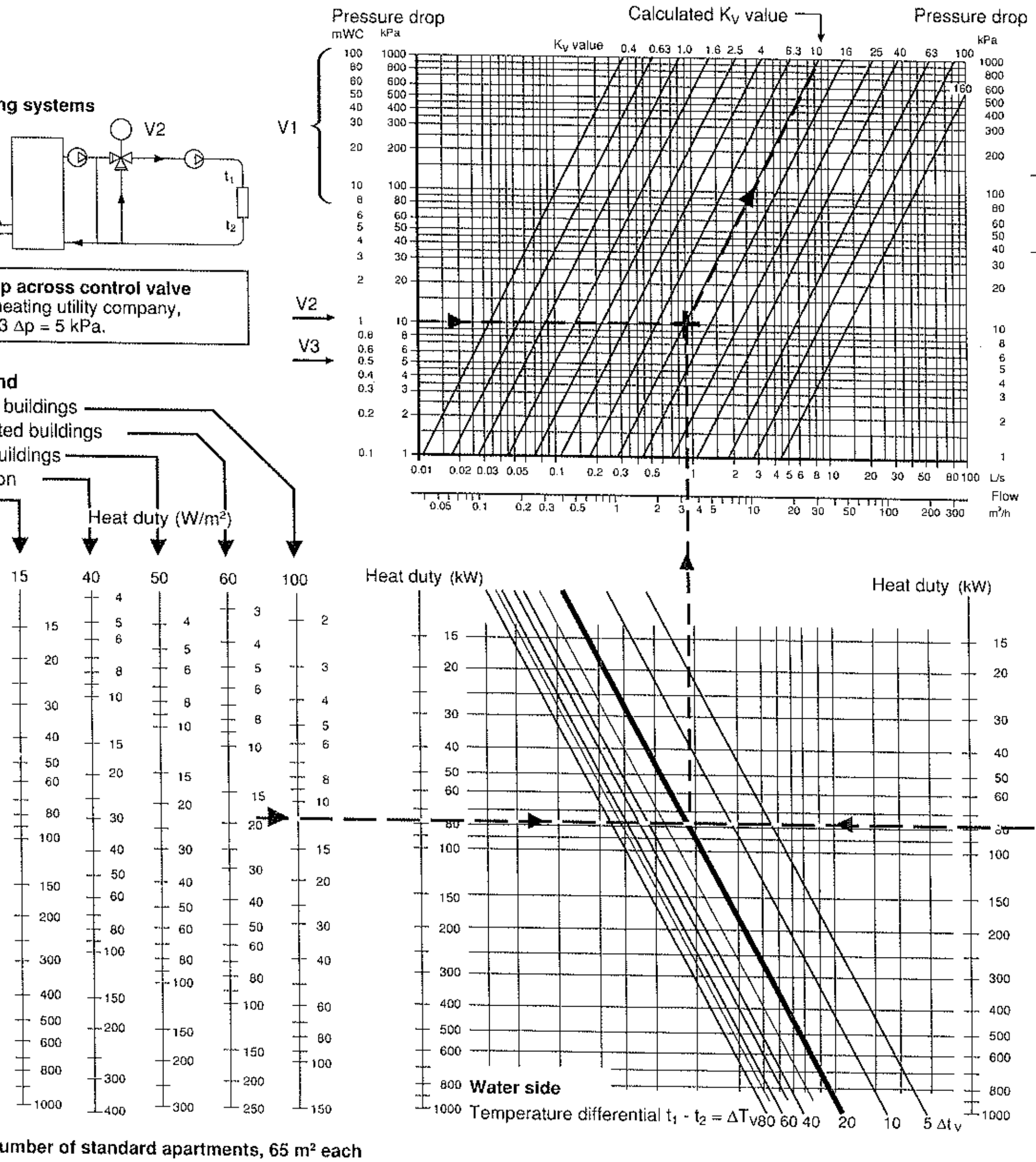
K_v = 10
Select V392 DN25 K_{vs} = 10

Air handling systems
- air side $t_4 - t_3$ = 40°C
- air flow 5500 m³/h
- water side $t_1 - t_2$ = 20°C
- V2, Δp = 10 kPa

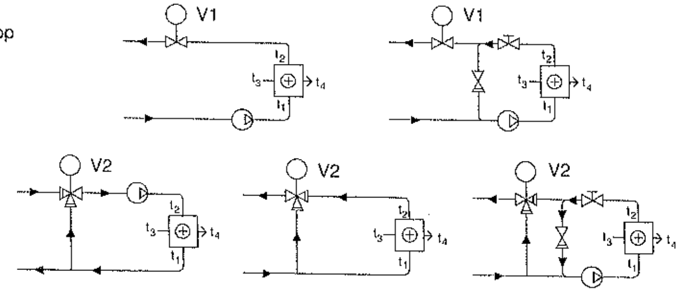
According to diagram:
- heat duty 78 kW
- water flow 0.93 L/s (3.3 m³/h)

K_v = 10
Select V392 DN25 K_{vs} = 10

Heating systems

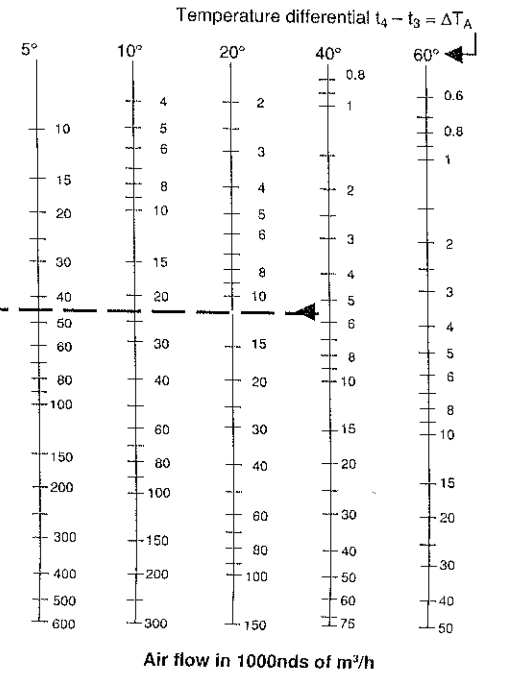


Air handling systems



Dimensioning pressure drop across contol valve
V1 Δp Depends on installation type and size, V2 Δp = 10 kPa.

Air heaters



Conversion factors

Energi

J joule	kWh kilowatt-hours	kpm kilopond-metres	kcal kilocalories	HPh horsepower-hours	ft - lbf (foot pound-force)	Btu (British thermal unit)
1	0.277 78·10^{-6}	0.101 97	0.238 85·10^{-3}	0.377 67·10^{-6}	0.737 56	0.947 82·10^{-3}
3.6·10^{6}	1	0.367 10·10^{6}	859.85	1.359 6	2.655 2·10^{6}	3.412 1·10^{3}
9.806 6	2.724 1·10^{-6}	1	2.342 3·10^{-3}	3.707 7·10^{-6}	7.233 0	9.294 9·10^{-3}
4.186 8·10^{3}	1.163·10^{-3}	426.94	1	1.581 2·10^{-3}	3.088 0·10^{3}	3.968 3
2.647 8·10^{6}	0.735 50	0.27·10^{6}	632.42	1	1.952 9·10^{6}	2.509 6·10^{3}
1.355 8	0.376 62·10^{-6}	0.138 26	0323 83·10^{-3}	0.512 06·10^{-6}	1	1.285 1·10^{-3}
1.055 1·10^{3}	0.293 07·10^{-3}	107.59	0.252 00	0.398 47·10^{-3}	778.17	1

1 erg = 0.1 · 10^{-6} J

Power

W	kpm/s	kcal/s	kcal/h	HP horse-power metric	HP horse-power	ft · lbf/s	B.t.u./h
1	0.101 97	0.238 85·10^{-3}	0.859 85	1.359 6·10^{-3}	1.3410·10^{-3}	0.737 56	3.412 1
9.806 6	1	2.342 3·10^{-3}	8.432 2	13.333·10^{-3}	13.151·10^{-3}	7.233 0	33.462
4.186 8·10^{3}	426.94	1	3.6·10^{3}	5.692 5	5.614 6	3.088 0·10^{3}	14.286·10^{3}
1.163	0.118 59	0.277 78·10^{-3}	1	1.581 2·10^{-3}	1.559 6·10^{-3}	0.857 79	3.968 3
735.5	75	0.175 67	632.42	1	0.986 32	542.48	2,509 6·10^{3}
745.7	76.04	0.187 11	641.19	1.013 9	1	550	2.544 4·10^{3}
1.355 8	0.138 26	0.323 83·10^{-3}	1.165 8	1.843 4·10^{-3}	1.818 2·10^{-3}	1	4.626 2
0.293 07	29.885·10^{-3}	69.999·10^{-6}	0.252 00	0.398 47·10^{-3}	0.393 02·10^{-3}	0.216 16	1

Pressure, mechanical tension

Pa	bar	kp/cm^2 at	kp/mm^2	torr (≈ mmHg)	atm	lbf/in^2 (psi)
1	10·10^{-6}	10.197·10^{-6}	0.101 97·10^{-6}	7.500 6·10^{-3}	9.869 2·10^{-6}	0.145 04·10^{-3}
100·10^{3}	1	1.019 7	10.197·10^{-3}	750.06	0.986 92	14.504
98.066·10^{3}	0.980 66	1	10·10^{-3}	735.56	0.967 84	14.223
9.806 6·10^{6}	98.066	100	1	73.556·10^{3}	96.784	1.422 3·10^{3}
133.32	1.333 2·10^{-3}	1.359 5·10^{-3}	13.595·10^{-6}	1	1.315 8·10^{-3}	19.337·10^{-3}
101.32·10^{3}	1.013 2	1.033 2	10.332·10^{-3}	760	1	14.696
6.894 8·10^{3}	68.948·10^{-3}	70.307·10^{-3}	0.703 07·10^{-3}	51.715	68.046·10^{-3}	1

1 mm water column ≈ 9.81 Pa 1 (eng) inch water column ≈ 249.09 Pa 1 inch Hg ≈ 3 386.4 Pa

Temperature

	Kelvin scale	Celsius scale	Rankine scale	Fahrenheit** scale[1]	Physical conditions
Interrelated temperatures	0 K	-273.15 °C	0°R	-459.67°F	Absolute zero
	255.372 K	-17.778 °C	459.67°R	0°F	
	273.15 K	0 °C	491.67°R	32°F	Melting point of ice*
	273.16 K	0,01 °C	491.688°R	32.018°F	Tripel point of water
	373.15 K	100 °C	671.67°R	212°F	Boiling point of water*
Interrelated temperature differentials	1 K	1 °C	1.8°R	1.8°F	
	0.555 56 K	0.555 56 °C	1°R	1°F	

*Assumes precise conditions. **Value in °C = $\frac{1}{1.8}$ · (value in °F – 32).

Water – Steam table

Pressure p kPa	p bar abs	Temperature t °C	Heat of evaporation kJ/kg
1	0.01	6.9808	2 485
2	0.02	17.513	2 460.2
3	0.03	24.1	2 444.6
4	0.04	28.983	2 433.1
5	0.05	32.898	2 423.8
6	0.06	36.183	2 416
7	0.07	39.025	2 409.2
8	0.08	41.534	2 403.2
9	0.09	43.787	2 397.9
10	0.1	45.833	2 392.9
20	0.2	60.086	2 358.4
30	0.3	69.124	2 336.1
40	0.4	75.886	2 319.2
50	0.5	81.345	2 305.4
60	0.6	85.954	2 293.6
70	0.7	89.959	2 283.3
80	0.8	93.512	2 274
90	0.9	96.713	2 265.6
100	1	99.632	2 257.9
150	1.5	111.37	2 226.2
200	2	120.23	2 201.6
250	2.5	127.43	2 181
300	3	133.54	2 163.2
350	3.5	138.87	2 147.4
400	4	143.62	2 133
450	4.5	147.92	2 119.7
500	5	151.84	2 107.4
600	6	158.84	2 085
700	7	164.96	2 064.9
800	8	170.41	2 046.5
900	9	175.36	2 029.5
1 000	10	179.88	2 013.6
1 100	11	184.07	1 998.5
1 200	12	187.96	1 984.3
1 300	13	191.61	1 970.7
1 400	14	195.04	1 957.7
1 500	15	198.29	1 945.2
1 600	16	201.37	1 933.2
1 700	17	204.31	1 921.5
1 800	18	207.11	1 910.3
1 900	19	209.8	1 889.3
2 000	20	212.37	1 888.6
2 500	25	223.94	1 839
3 000	30	233.84	1 793.9
4 000	40	250.33	1 712.9
5 000	50	263.91	1 639.7
6 000	60	275.55	1 571.3
7 000	70	285.79	1 506
8 000	80	294.97	1 442.8
9 000	90	303.31	1 380.9
10 000	100	310.96	1 319.7
11 000	110	318.05	1 258.7
12 000	120	324.65	1 197.4
13 000	130	330.83	1 135
14 000	140	336.64	1 070.7
15 000	150	342.13	1 004
20 000	200	365.7	591.9
22 000	220	373.69	184.5
22 120	221.2	374.15	0

Temperature conversion table

40 to 0			0 to 100						100 to 1 000					
°C	t°	F°	°C	t°	°F	°C	t°	°F	°C	t°	°F	°C	t°	°
-40	**-40**	-40	-17.8	**0**	32	10	**50**	122	38	**100**	212	260	**500**	932
-39.4	**-39**	-38.2	-17.2	**1**	33.8	10.6	**51**	123.8	43	**110**	230	266	**510**	950
-38.8	**-38**	-36.4	-16.7	**2**	35.6	11.1	**52**	125.6	49	**120**	248	271	**520**	968
-38.3	**-37**	-34.6	-16.1	**3**	37.4	11.7	**53**	127.4	54	**130**	266	277	**530**	986
-37.7	**-36**	-32.8	-15.6	**4**	39.2	12.2	**54**	129.2	60	**140**	284	282	**540**	1004
-37.2	**-35**	-31	-15	**5**	41	12.8	**55**	131	66	**150**	302	288	**550**	1022
-36.6	**-34**	-29.2	-14.4	**6**	42.8	13.3	**56**	132.8	71	**160**	320	293	**560**	1040
-36.1	**-33**	-27.4	-13.9	**7**	44.6	13.9	**57**	134.6	77	**170**	338	299	**570**	1058
-35.5	**-32**	-25.6	-13.3	**8**	46.4	14.4	**58**	136.4	82	**180**	356	304	**580**	1076
-34.9	**-31**	-23.8	-12.8	**9**	48.2	15	**59**	138.2	88	**190**	374	310	**590**	1094
-34.4	**-30**	-22	-12.2	**10**	50	15.6	**60**	140	93	**200**	392	316	**600**	1112
-33.9	**-29**	-20.2	-11.7	**11**	51.8	16.1	**61**	141.8	99	**210**	410	321	**610**	1130
-33.3	**-28**	-18.4	-11.1	**12**	53.6	16.7	**62**	143.6	100	**212**	413	327	**620**	1148
-32.8	**-27**	-16.6	-10.6	**13**	55.4	17.2	**63**	145.4	104	**220**	428	332	**630**	1166
-32.2	**-26**	-14.8	-10	**14**	57.2	17.8	**64**	147.2	110	**230**	446	338	**640**	1184
-31.7	**-25**	-13	-9.44	**15**	59	18.3	**65**	149	116	**240**	464	343	**650**	1202
-31.1	**-24**	-11.2	-8.89	**16**	60.8	18.9	**66**	150.8	121	**250**	482	349	**660**	1220
-30.6	**-23**	- 9.4	-8.33	**17**	62.6	19.4	**67**	152.6	127	**260**	500	354	**670**	1238
-30	**-22**	- 7.6	-7.78	**18**	64.4	20	**68**	154.4	132	**270**	518	360	**680**	1256
-29.4	**-21**	- 5.8	-7.22	**19**	66.2	20.6	**69**	156.2	138	**280**	536	366	**690**	1274
-28.9	**-20**	- 4	-6.67	**20**	68	21.1	**70**	158	143	**290**	554	371	**700**	1292
-28.3	**-19**	- 2.2	-6.11	**21**	69.8	21.7	**71**	159.8	149	**300**	572	377	**710**	1310
-27.7	**-18**	- 0.4	-5.56	**22**	71.6	22.2	**72**	161.6	154	**310**	590	382	**720**	1328
-27.2	**-17**	+ 1.4	-5	**23**	73.4	22.8	**73**	163.4	160	**320**	608	388	**730**	1346
-26.6	**-16**	+ 3.2	-4.44	**24**	75.2	23.3	**74**	165.2	166	**330**	626	393	**740**	1364
-26.1	**-15**	+ 5	-3.89	**25**	77	23.9	**75**	167	171	**340**	644	399	**750**	1382
-25.5	**-14**	+ 6.8	-3.33	**26**	78.8	24.4	**76**	168.8	177	**350**	662	404	**760**	1400
-24.9	**-13**	+ 8.6	-2.78	**27**	80.6	25	**77**	170.6	182	**360**	680	410	**770**	1418
-24.4	**-12**	+10.4	-2.22	**28**	82.4	25.6	**78**	172.4	188	**370**	698	416	**780**	1436
-23.9	**-11**	+12.2	-1.67	**29**	84.2	26.1	**79**	174.2	193	**380**	716	421	**790**	1454
-23.3	**-10**	+14	-1.11	**30**	86	26.7	**80**	176	199	**390**	734	427	**800**	1472
-22.8	**-9**	+15.8	-0.56	**31**	87.8	27.2	**81**	177.8	204	**400**	752	432	**810**	1490
-22.2	**-8**	+17.6	0	**32**	89.6	27.8	**82**	179.6	210	**410**	770	438	**820**	1508
-21.7	**-7**	+19.4	0.56	**33**	91.4	28.3	**83**	181.4	216	**420**	788	443	**830**	1526
-21.1	**-6**	+21.2	1.11	**34**	93.2	28.9	**84**	183.2	221	**430**	806	449	**840**	1544
-20.6	**-5**	+23	1.67	**35**	95	29.4	**85**	185	227	**440**	824	454	**850**	1562
-20	**-4**	+24.8	2.22	**36**	96.8	30	**86**	186.8	232	**450**	842	460	**860**	1580
-19.5	**-3**	+26.6	2.78	**37**	98.6	30.6	**87**	188.6	238	**460**	860	466	**870**	1598
-18.9	**-2**	+28.4	3.33	**38**	100.4	31.1	**88**	190.4	243	**470**	878	471	**880**	1616
-18.3	**-1**	+30.2	3.89	**39**	102.2	31.7	**89**	192.2	249	**480**	896	477	**890**	1634
-17.8	**±0**	+32	4.44	**40**	104	32.2	**90**	194	254	**490**	914	482	**900**	1652
			5	**41**	105.8	32.8	**91**	195.8				488	**910**	1670
			5.56	**42**	107.6	33.3	**92**	197.6				493	**920**	1688
			6.11	**43**	109.4	33.9	**93**	199.4				499	**930**	1706
			6.67	**44**	111.2	34.4	**94**	201.2				504	**940**	1724
			7.22	**45**	113	35	**95**	203				510	**950**	1742
			7.78	**46**	114.8	35.6	**96**	204.8				516	**960**	1760
			8.33	**47**	116.6	36.1	**97**	206.6				521	**970**	1778
			8.89	**48**	118.4	36.7	**98**	208.4				527	**980**	1796
			9.44	**49**	120.2	37.2	**99**	210.2				532	**990**	1814
						37.8	**100**	212				538	**1000**	1832

Interpolation values

°C		°F
0.56	**1**	1.8
1.11	**2**	3.6
1.67	**3**	5.4
2.22	**4**	7.2
2.78	**5**	9.0
3.33	6	10.8
3.89	7	12.6
4.44	8	14.4
5.00	9	16.2
5.56	10	18.0

Temperature conversion formulas:
°C = 5/9 (°F –32)
°F = 9/5 (°C +32)

Enter the "t°" column (bold figures) with the figure to be converted, then seek the corresponding Celsius degrees to the left, and Fahrenheit to the right.

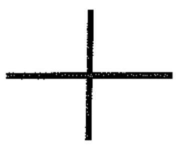
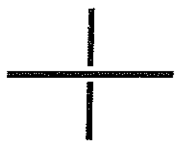
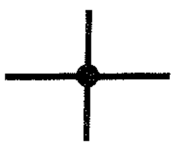
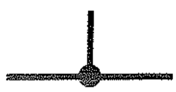
10 Miscellaneous

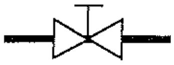
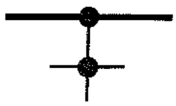
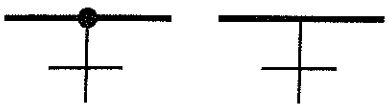
Symbols for heating installations

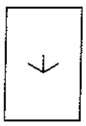
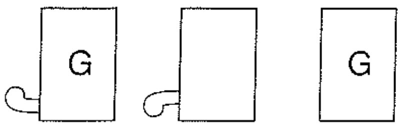
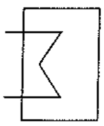
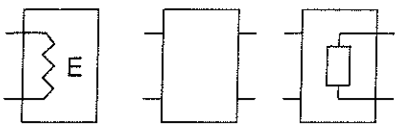
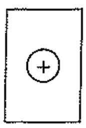
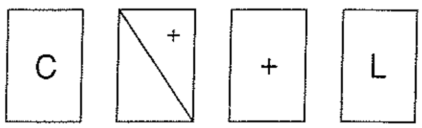
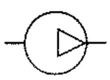
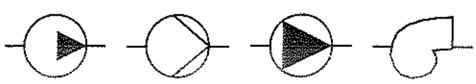
The symbols used in this handbook are mainly in coordination with the international standard ISO 4067/1 – 1984, Part 1 and the Swedish standard SS 03 22 60.

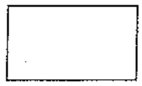
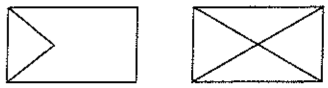
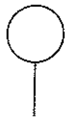
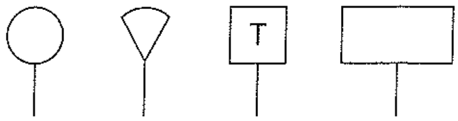
These symbols are shown in column "ISO" in the following diagrams.

Apart from the ISO symbols other standard will be shown (also occur in this handbook).

Piping	**ISO**	**Others**
Pipe, general symbol		
Direction of flow		
Crossing pipe, not connected		
Crossing pipe, connected		
Tee connection		

Valves	ISO	Others
Valve, general symbol, also control valve, two-way		
Non-return valve (check valve)		
Control valve, three-way (mixing)		
Shut-off valve		
Balancing valve		
Draw-off valve (tap)		

Equipment	ISO	Others
Boiler (oil)		
Heat exchanger		
Terminal unit (coil, radiator)		
Pump		

Control equipment	**ISO**	**Others**
Controller		
Actuator (motor)		
Temperature sensor		
Pressure meter		

Symbols and units used in this handbook

Symbol	Unit	Description
C_V	USGPM	Flow coefficient at $\Delta p = 1$ psi
K_R		Viscosity correction factor
K_V	m^3/h	Flow coefficient, nominal, at $\Delta p = 1$ bar
K_V	m^3/h	Flow at $\Delta p = 1$ bar
K_{V0}	m^3/h	Theoretical smallest valve flow at $\Delta p = 1$ bar
K_{Vr}	m^3/h	Smallest flow within tol. of charact. at $\Delta p = 1$ bar
p	kPa (bar)	Pressure
p_K	kPa (bar)	Critical pressure
Δp	kPa (bar)	Differential pressure, pressure drop
Δp_m	kPa (bar)	Max. permissible pressure drop across a valve to to avoid erosion damages
Δp_{V100}	kPa (bar)	Pressure drop across fully open valve
Δp_{V0}	kPa (bar)	" " " closed valve
Δp_C	kPa (bar)	Max. permissible pressure drop across closed valve
p_{VC}	kPa (bar)	Pressure at vena contracta
p_V	kPa_{abs}(bar_{abs})	Vaporization pressure of liquid
q	m^3/h	Volumetric flow (water)
q_V	l/s	Volumetric flow (water)
q_L	Nm^3/s	Volumetric flow, air, at 0°C, 1 atm
q_a	%	Momentary flow, in %
t_1	°C	Supply temperature, primary
t_2	°C	Return temperature, primary
t_3	°C	Supply temperature, secondary
t_4	°C	Return temperature, secondary
ΔT	K	Temperature differential, (Kelvin)
Δt	°C	" " , (°C)
P	kW	Power (heat rate)
Q	kcal/h	Heat quantity per hour (i. e., power)
c_p	kJ/kg·K	Specific heat
c_{pA}	kJ/kg·K	Specific heat – air at constant pressure
R		Rangeability of valve

Symbol	Unit	Description
R_b		Rangeability of valve at authority β
R_S		Resultant rangeability of valve
R_R	kPa/m	Flow resistance per unit length of pipe
R_V		Calculation factor, viscosity
K_R		Correction factor, viscosity
k		Correction factor for superheated steam
DN	mm	Nominal diameter
PN		Nominal pressure in bar
d_i	mm	Internal diameter
h	%	Relative lift of a valve
L		Load
v	m/s	Flow velocity
G	kg/h	Steam flow
Z		Pressure ratio
S_P		Flow oversizing factor
β (beta)		Valve authority
ξ (ksi)		Resistance factor
η_W (eta_W)	%	Efficiency of heat exchanger, water side
η_A (eta_A)	%	Efficiency of heat exchanger, air side
ε_A ($epsilon_A$)		Relative heating of an air heater, air side
ε_W ($epsilon_W$)		Relative heating of an air heater, water side
η (eta)	Pa·s	Dynamic viscosity
ν (ny)	m^2/s	Kinematic viscosity
ρ (ro)	kg/m^3	Density
ρ_W (ro_W)	kg/m^3	Density of water
ρ_A (ro_A)	kg/m^3	Density of air
σ (sigma)		Cavitation coefficient
Φ (fi)	mm	Diameter

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Subject index

Brief explanations (in alphabetical order) of certain concepts and keywords used in this handbook, with references to texts.

Page

69 **ABSOLUTE PRESSURE**
Pressure above absolute zero. Is obtained from gauge pressure by adding the atmospheric pressure (=100 kPa, or 1 bar).

6 **ACTUAL VALUE**
Value of the measured variable entering the controller

14 **ACTUATOR**
Device for positioning of a valve.

7 **COIL**: (Preheat, reheat, hot water, heating, cooling, steam)

7 **Preheat coil**
Air heat exchanger for air duct. The coil can be exposed to outdoor air (i.e., not preheated) which means that the coil is subject to freezing risk.

7 **Reheat coil**
Air heater for air duct, for heating preheated air.

7 **Hot water coil**
Air heater, connected directly to a district heating network.

7 **Heating and cooling coil**
Heat exchanger in air conditioning system for heating or cooling air. Medium on primary side is water.

7 **Steam coil**
Air heater directly connected to a steam system.

59 **CAVITATION**
The formation of steam (bubbles) in valves, typically caused by pressure drop.

60 **CAVITATION DIAGRAM**
Diagram showing the lowest static pressure, upstream of a valve, which is permitted to avoid cavitation..

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48	**CLOSING PRESSURE, Δp_C** The largest pressure drop across the valve where the plug – seat leakage is less than specified, at a given operating thrust.
14	**CONTROLLED DEVICE** The combination of actuator and control valve.
10	**CONVECTOR** Small air heater (or cooler). Primary side medium: water, warm or cold.
69	**CRITICAL PRESSURE, P_K** The steam pressure where the outlet velocity of the steam has reached a maximum.
42	**FLOW CHARACTERISTIC** Relationship between the lift of a valve and the flow coefficient (K_V, C_V) or the equivalent area (A_V).
37	**FLOW COEFFICIENT, K_V, C_V, A_V** The flow through a fully open valve, at a specified pressure drop across the valve.
6	**HEAT EXCHANGER** Device for transferring heat from one medium to another. The media are typically water, air or steam.
34	**LEAKAGE, CLOSED VALVE** The maximum flow that will leak past the plug and seat of a fully closed valve.
32	**MAXIUMUM PRESSURE DROP ACROSS THE VALVE PLUG AND SEAT, Δp_m** The largest permissible pressure differential without risk of erosion damages to valve plug and seat.
34	**NOMINAL PRESSURE, PN** Maximum nominal internal pressure in a valve, expressed in bar. PN = Pressure Nominal
40	**PRESSURE DROP DIAGRAM** Diagram showing the relationship between pressure drop and flow and the flow capacity value of a valve.

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34 **PRESSURE TEST**
Pressure testing of the valve body.

6 **RADIATOR**
Heat exchanger for room heating. Media: Water on the primary, air on the secondary.

55 **RANGEABILITY**
The ratio between the largest and the smallest flow that a control valve can handle with maintained flow characteristic.

6 **SETPOINT**
The desired value of the controlled variable.

57 **SPLIT RANGE CONFIGURATION**
Two valves, of different size, connected in parallel, that open sequentially. Split range configuration is used to obtain a large rangeability.

20 **SPLIT RANGE VALVE**
Valve with two plugs that open sequentially. Split range valves have a much greater rangeability than normal valves.

47 **VALVE AUTHORITY**
The ratio between the pressure drop across a valve at fully open valve and the total pressure drop.

42 **VALVE CHARACTERISTIC = *f* (VALVE AUTHORITY)**
Describes how the flow characteristic changes as the valve authority changes

32 **VALVE SIZE, DN**
Valve size in mm. DN 32 used to be $1\text{–}{}^{1}/_{2}$".
DN = Diameter nominal

7 **WATER HEATER**
Heat exchanger for heating of water. Primary medium is water, secondary medium tap water.

61 **VISCOSITY INDEX**
The viscosity of a liquid is denoted by the viscocity index of the liquid.