Balancing Manual

Honeywell

Guideline for the hydronic balancing in heating and cooling systems

Contents

Index of Diagrams

Index of Tables

1. INTRODUCTION

The demands on modern heating and cooling systems are numerous. The end user expects the system to be easy to use and economical. The manufacturer is expected to supply state-of-the-art goods and keep the production costs of the system to a minimum.

Overdimensioned systems, together with poorly hydronically balanced systems, or systems that are not balanced at all, should be a thing of the past.

It is these systems that repeatedly cause problems in practice. Radiators do not heat up and thermostats cause irritating noises, resulting in higher costs than originally planned. This fact is actually quite remarkable, as the necessity for hydronic balancing of systems is common knowledge in the industry. Often, however, this knowledge is simply ignored, which means that many systems do not correspond to applicable regulations or the current status of the technology.

This manual is designed to provide an overview of current requirements and act as an instruction manual for hydronic balancing. The manual first covers the theory of hydronic balancing, followed by a description of the procedure for regulation using measurements, based on a practical example. The possibility of hydronic balancing according to calculations is also explained. Some examples are used to describe the dimensions of thermostatic radiator valves and balancing valves.

1.1 Consequences of failing to implement hydronic balancing

A modern water pump system distributes heat or cold equally to all rooms according to demand. Depending on system performance, a precisely defined volume of water is distributed through the pipes. As in all systems with locally separated heat/cold generation and output, the system is confronted with the problem of distribution on demand

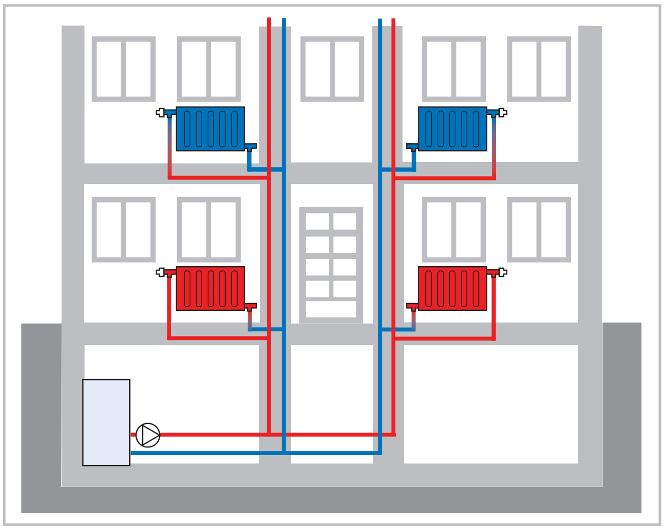


Fig. 1: Uneven distribution of water in a non-hydronically balanced system

Water always flows in the direction of least resistance. In a system that is not balanced, this means that the circuit closest to the pump always receives the largest volume of water. The resistance on the route to this circuit is less than for the other circuits, because in comparison the water has less distance to travel, and therefore has to overcome less resistance in the pipes. Within this circuit, the heat exchangers at the beginning receive the largest amount of water. Conversely, the circuit that is furthest from the pump has the least amount of water. The heat exchangers towards the end of this circuit receive the least heat or cold, because the distance to travel to these consumers is longer, and therefore more resistance in the pipes must be overcome. In the less hydronically efficient circuits, the heat exchangers that are further away from the pump thus receive unsatisfactory supply. The consequences of this are: Heat exchangers and circuits closer to the main supply are oversupplied, and the heat exchangers and circuits that are further away are undersupplied.

In practice, this problem is often incorrectly assessed. Frequently, the diagnosis is that the pump output or the supply temperature is too low, or that regulation is faulty. As a result of the incorrect diagnosis, pump outputs are increased or larger pumps installed, the supply temperature increased, or the regulation is adjusted.

A system “corrected” in this way may start to generate the required room temperature. However, the unnecessarily high mass flow rate or excessive supply and therefore also return temperatures result in further deficiencies, which can also be cause for complaint. For this reason, we have listed the most common deficiencies resulting from insufficient hydronic balancing below.

Uneven Heat Output

The heat output of a radiator or heat exchanger is largely determined by the quantity of water flowing through it. If the flow is too high, the room is overheated, while too little flow results in an insufficient room temperature. Radiator heating is tolerant within a certain range, which means that a slight reduction in water pressure will only result in a negligible drop in room temperature. On the other hand, doubling the flow of water only leads to a minor increase in room temperature. This relationship is often used as an excuse for not performing hydronic balancing. The diagram below shows this relationship on the basis of a radiator with the radiator exponent = 1.3 and a supply temperature/return temperature of 70/55°C at 20°C room temperature.

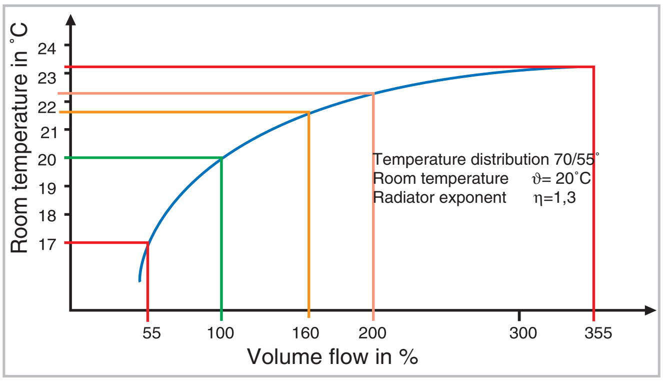


Fig. 2: Relationship between room temperature and volume flow

Note that a volume flow of 60-70 % causes the room temperature to sink to approx. 18°C, and therefore affects comfort.

Delayed heating following overnight temperature reduction

One typical consequence of insufficient hydronic balancing is detected following cold nights in spring and autumn. In the early morning, the system resumes heating after an overnight reduction in temperature, but only the consumers nearest the pump are supplied. Only when the demand is met and the thermostatic radiator valves reduce the mass flow rate, is sufficient water forwarded to the more distant consumers. It can therefore be midday or afternoon by the time all consumers are supplied and the rooms become warm. The phenomenon may have disappeared again by the following day. This only requires the building to have cooled down less due to a mild overnight temperature, and the heat demand for the heating phase therefore to be lower.

Noise problems

From the previous problem description (fig. 2), we can see that an excess volume flow is not necessarily seen as a fault, because the room temperature does not increase to the same extent as the temperature reduction. An unnecessarily high volume flow can often, however, cause a further defect in the form of streaming noises. Thermostatic radiator valves, which have the primary task of regulating room temperature through reducing volume flow, cannot be subjected to high levels of differential pressure, as the small diameters of the openings required for regulation otherwise lead to strong turbulence of the medium. It is this turbulence that causes the streaming noises. To avoid this, thermostatic radiator valves should therefore not be operated above a differential pressure of 200 mbar. If a valve is now subjected to a differential pressure, for example of 150 mbar, an increase in the volume flow quickly causes the threshold value to be exceeded.

Required temperature differences are not reached

The transported power of the circulating medium is defined as follows:

$$P = q \times c_p \times \Delta t$$

$$\Delta t = \vartheta_S - \vartheta_R$$

P	Performance of the heating medium in W
q	Mass flow rate in kg/s
c_p	specific heat capacity of the medium
c_p =	4186 J/kg K for water at 60°C
ϑ_S	Supply temperature in °C
ϑ_R	Return temperature in °C
Δt	Power distribution

From this formula, it is easy to recognise that doubling the mass flow q leads to a halving of power distribution Δt. This has consequences for systems and components that are dependent on a low return temperature.

For example:

Condensation boilers

These use the energy generated by condensation to increase thermal efficiency. This means that the high thermal efficiency specified by the manufacturer can only be achieved as long a certain minimum return temperature is not exceeded.

District heating connections

The operators of district heating networks aim to operate their networks as economically as possible. This is possible if many consumers can be supplied with little need for transporting water. Networks such as this are therefore operated with a high supply temperature and a low return temperature.

To encourage the customers to align their systems to the same high supply/return temperature differences, tariff bonuses, i.e. lower energy prices, are offered in return for customers who comply. This price advantage is lost if the return temperature has risen as a result of insufficient hydronic pressure.

Technical measurement and regulation problems

Control valves are designed so that the power output is regulated throughout the whole available valve stroke. An excessively high level of flow has the same effect as an over-sized regulator. The working range of the regulator is reduced, and the aperture is increased. This has an effect on the regulatory behaviour, which in some circumstances may be reflected in unstable regulation.

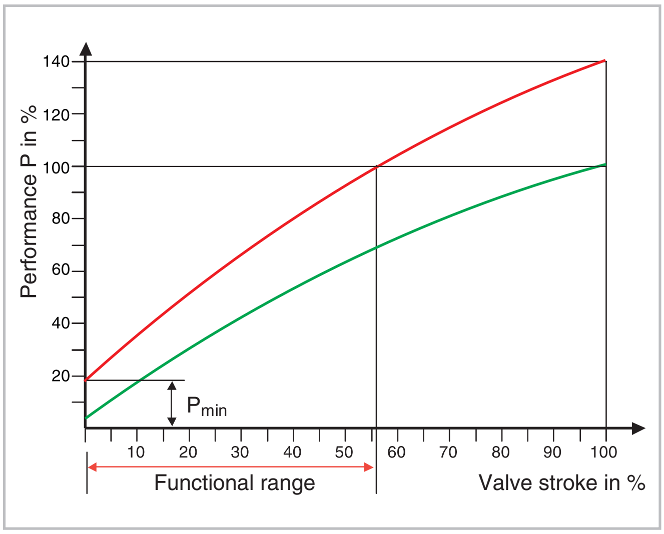


Fig. 3: Characteristic curve of a control valve in the case of excessive volume flow. The working range is reduced to approx. 55%

Unnecessarily high energy costs

The deficiencies above have an adverse effect on comfort and are usually reported as complaints. A further problem resulting from insufficient hydronic balancing can be unnecessarily high energy costs, caused by increasing the normal performance of the pump. Increasing the volume flow also increases the electrical power consumption. In well-insulated modern buildings, the proportion of pump energy in relation to heating performance has increased in importance, because the volume flow is not reduced with decreased demand for heating or cooling. The radiators still have the width of the window, and can therefore be operated with a lower supply temperature than in a poorly insulated building. A lower supply temperature also has a positive effect on the efficiency of the heat generator and on distribution loss through the pipe system. This general rule naturally also applies for systems that are used for climate control in buildings using cold water. The closer the average temperature of the supply and return water is to the room temperature, the lower the distribution loss.

1.2 A problem and its solution

The solution to all these problems is well known. Through hydronic balancing of the whole system, the resistance of the individual consumers is coordinated so that each heat exchanger receives exactly the required amount of water. Hydronic balancing results in:

- Equal supply to all heat exchangers and circuits
- Reduction of flow noise to a minimum
- Pumps only demand as much water as they require
- Energy savings

The installer is thus able to provide customers with a smoothly functioning economical and ecological system - and all with no additional energy requirements and no restrictions on comfort.

Advantages of hydronic balancing are:

- Energy savings
- Increased comfort, no noise
- Reduction in CO_2 emissions
- More efficient use of resources

2. BASIC PRINCIPLES OF HYDRONIC BALANCING

Hydronic balancing ensures that all circuits and heat exchangers receive water at the same resistance and are therefore equally supplied with heat or cold.

There are several possibilities for fulfilling this task. The procedure best suited to a particular system depends on several factors, including the size of the system, the layout of the distribution pipes and consumers and conditions in the surrounding environment.

2.1 Statically balanced Systems

In these systems, additional resistance is integrated in the form of adjustable throttle valves in the circuit, in the distribution, and in the heat exchanger. The circuit with the highest resistance is not regulated at all, and the circuit with the lowest resistance is regulated the most. The required resistance of the throttle valves can be calculated by comparing the pressure drop of the individual circuits. To calculate this, the required supply pressure for all circuits is determined and the difference between the supply pressures then needs to be applied to the corresponding flow control valve. This procedure is quick and simple to perform using modern calculation software. The result is a value for setting each individual valve, which can be used to generate the additional pressure. A system set in this way is statically balanced and optimized for a particular system state. This is normally the state of maximum demand. Chapter 5 contains some example calculations for this type of balancing.

2.1.1 Static Balancing with Thermostatic Radiator Valves

In smaller systems, such as heating systems in smaller family homes with a low number of radiators, balancing at the heat exchanger using the thermostat and/or lockshield valves is sufficient. Additional regulating valves are normally not necessary.

The valves and fittings industry offers various solutions intended to facilitate the planner and installer in making the presettings (hydronic balancing). There are two main options to consider: Some people prefer presettable thermostats, while others are in favour of fixed limited thermostatic radiator valves, coupled with a flow control valve in the return.

Limited Range Thermostatic Radiator Valves

When choosing valves with limiting K_v-discs, the installer takes advantage of the fact that the dimensioning of the thermostatic radiator valve can only be based on an assumed stationary state. When the system is put into operation, some thermostatic radiator valves immediately begin to close, while others may open further. This means that the individual thermostatic radiator valves have completely different hydraulic properties than those taken into account in the calculation. They may have smaller or larger differential pressures, which can result in a larger or smaller valve stroke. Honeywell therefore offers thermostatic radiator valves with a limited characteristic curve.

This means that when the thermostatic radiator valve opens further, no more water flows through the valve than with a valve stroke of 3 K - the characteristic curve of the valve levels off.

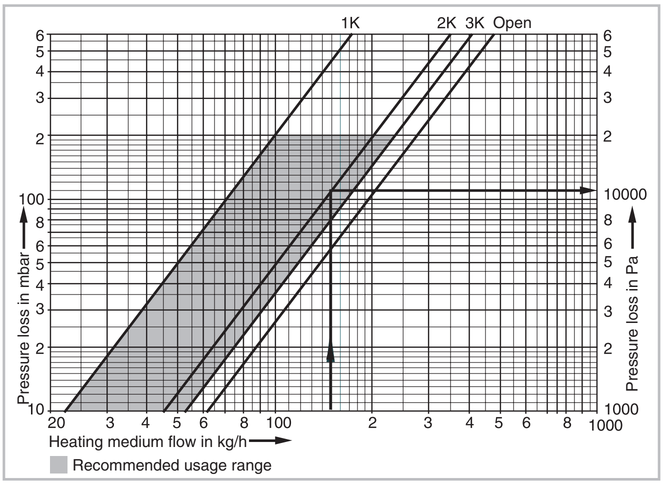


Fig. 4: Characteristic curve of a BB type thermostatic radiator valve, balancing of the individual radiators using the P-band, X_P = 1-3 K

The radiator now uses the whole regulated bandwidth of the valve. This means that for water volumes with the curves 1 to 3 K, the thermostatic radiator valve itself controls the quantity of water according to demand because of its higher authority. Regulation of the return valve is only required for extremely small quantities of water. This simplifies the task of the installer, who has to fit fewer radiator valves. In this respect, the lockshield valve is not considered an additional valve. It is currently a common requirement that radiators can be locked and emptied. The lockshield valve has thus been used as standard for some time.

Presettable Thermostatic Radiator Valves

The principle of presettable valves is based on the idea that the optimal thermostatic radiator valve should be selected on the basis of a characteristic curve of $X_P \leq 2$ K. If a lower quantity of water is now used in a calculation, the designer refers to the diagram of the corresponding valve and uses a characteristic curve with a lower presetting. This presetting can be equated with regulation at the lockshield valves mentioned above, because the regulating throttle is downstream of the actual valve disc. A presettable thermostatic radiator valve is therefore basically the same as a normal thermostatic radiator valve, but with a regulating lockshield valve integrated in one housing.

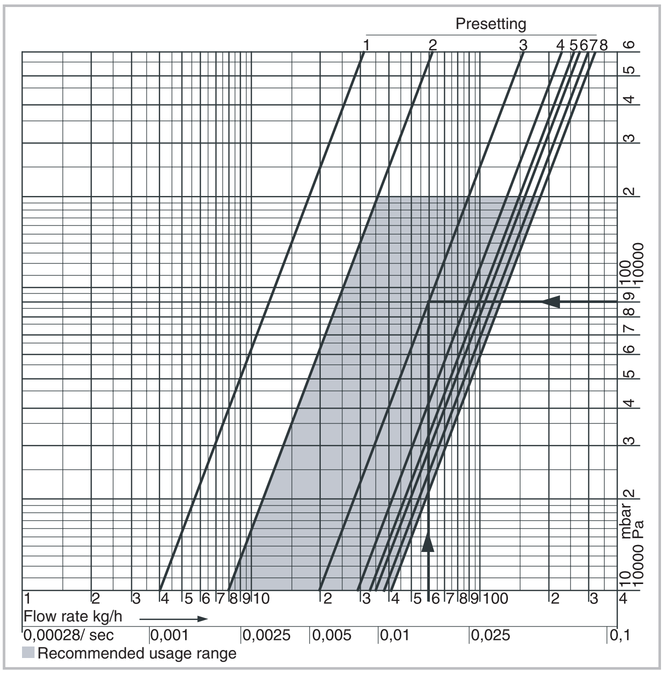


Fig. 5: Characteristic curve diagram of a V type thermostatic radiator valve, balancing using the presetting at $X_P = 2$ K

2.1.2 Static balancing with balancing valves

Hydronic balancing, however, does not end with the thermostatic radiator valves. Rather than one single circuit with only a few heat exchangers, systems usually consist of several circuits and multiple heat exchangers. This can have the consequence that the required pump performance may exceed a differential pressure of 200 mbar, which is generally accepted as the maximum for noise-free operation of thermostatic radiator valves. If this is the case, some of the differential pressure needs to be limited in the circuit. The purpose of balancing valves is thus to balance the individual circuits so that each circuit only receives the volume of water that is required according to demand. As described in section 2.1.1, balancing between the circuits also takes place at the consumer in this case.

Static balancing is particularly suitable for smaller systems or for systems with a constant water volume, such as single pipe heating systems or cooling systems with fan coil units. In systems with constant water volume, the flow is diverted by a bypass at the heat exchanger when demand is decreased. Fan coil units frequently have an integrated 2-way valve, which opens as soon as the fan is switched on. The room temperature is controlled by the fan speed instead of the flow rate.

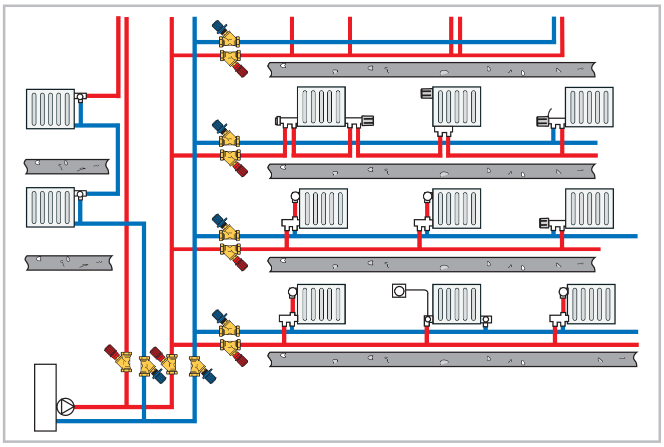


Fig. 6: Static balancing in a heating system

Whether determined as a result of calculations or measurement, the setting is based on operation at full load. This high setting is, however, only used on a few days during the heating period. Static balancing therefore means that the system is overdimensioned for most of the year and an unnecessary high amount of energy is consumed. In addition, static balancing is unable to reduce flow noises resulting from an increase in differential pressure in part-load operation. This is only possible with dynamic balancing.

2.2 Dynamically Balanced Systems

In systems with variable water volume and differential pressures, such as two-pipe heating systems with radiators, static balancing quickly exceeds the limits within which it can usefully be applied, particularly in larger systems.

In this type of system, the room temperature is controlled by the flow rate through the radiator. When outside temperatures are low, demand for heat is increased, and the system works at high load or full load, depending on the status assumed during planning. When outside temperatures are warmer, e.g. in spring or autumn, there is correspondingly less demand for heat. The thermostatic radiator valves are closed and a lower volume of water flows through the radiators. Within this partial load range, i.e. when the volume of water in the circuit decreases, balancing valves lose their effectiveness, because, in the same way as the piping, they behave like hydraulic fixed resistors and lose differential pressure. If the volume of water is halved, the differential pressure at the balancing valve is thus reduced to a quarter! At the same time, the demand on the circulating pump is increased, as the operating point in the pump diagram shifts to the left (see fig. 7). Within the partial load range, there is therefore an increase in differential pressure caused by the remaining open thermostatic radiator valves. This increase in differential pressure depends on the characteristic curve of the pump. For pumps with a flat characteristic curve, the increase in differential pressure is minimal and it increases as the curve becomes steeper.

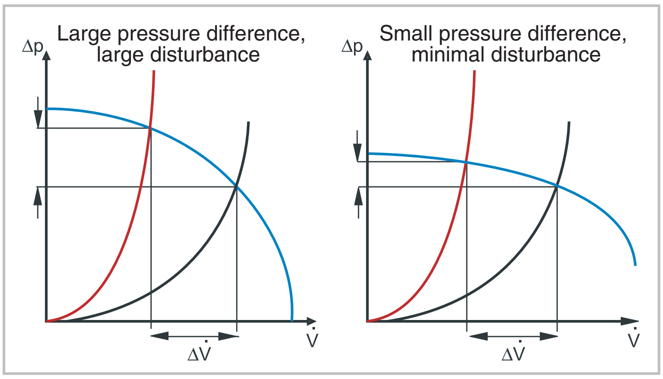


Fig. 7: Steep and flat characteristic curves for pumps

Depending on the dimensions of the pipe network and the size of the system, this differential pressure can exceed the maximum permitted value in terms of noise generation of 200 mbar. If a system generates noise in the partial load range, this is an indication that dynamic balancing with automatic balancing valves as differential pressure regulators are required.

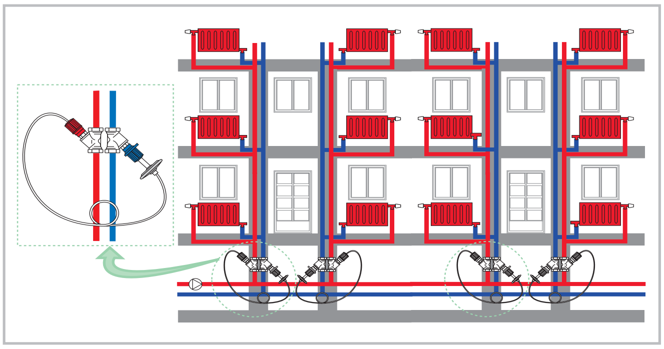


Fig. 8: Dynamic balancing in a two-pipe system

These valves basically function in the same way as manual balancing valves, except that automatic balancing valves continually control the pressure differential between supply and return. If, for example, the differential pressure in one circuit changes as a result of thermostatic radiator valves opening or closing in other circuits, the balancing valve automatically compensates. The system is therefore continually balanced depending on the current load, and the appropriate pressure conditions are maintained within the system.

In addition to maintaining an optimal hydronic balance, even under changing operating conditions, automatic balancing valves also limit the differential pressure increase in the partial load range. The thermostatic radiator valves therefore function under constant conditions. Flow noises, oversupply and undersupply of individual circuits or heat exchangers are avoided. The system functions economically, as only the required amount of water, and therefore heat or cold, is circulated. And above all: There is no need for time-consuming and costly setting and checking of calculated values.

Dynamic balancing is also well-suited for renovating heating systems in old buildings, as plan documents of the original pipe installations are normally not available. Of course, it is possible to measure the system after it is completed, but this is time-consuming and complicated to calculate, even for an expert. The end result is still ‘only’ a statically balanced system, i.e. it is set for the design flow.

In practice this is rarely achieved, however, as thermostatic radiator valves close and the system then operates under partial load conditions.

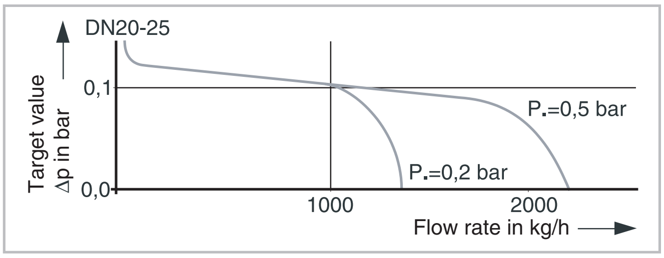


Fig. 9: Characteristic curve of an automatic balancing valve. The set differential pressure also remains fairly constant during partial load operation.

The use of automatic balancing valves saves the effort of measuring the system, and the system is also suitable for variable water volumes. This means that exactly the right volume of water flows, even if the system is working under partial load.

Dynamic balancing is therefore more suitable than manual balancing, in modern as well as in old systems.

A note on speed-regulated pumps

The simultaneous use of automatic balancing valves and pump regulator controlled by differential pressure is ideal, because the energy-saving effect of a decentralised differential pressure regulation supports the central regulation of the pump. The advantage of speed-regulated pumps is that the pump consumes less energy and therefore less costs arise, as the pump speed is automatically reduced under lower load conditions. Speed-regulated pumps are of no relevance for hydronic balancing, and cannot replace control valves. They generate the pressure differential required for transport at a central location and thus have no influence on distribution within the system.

2.3 Summary

- For small buildings, such as the average family home, it is sufficient to use static balancing using thermostatic radiator valves and, where necessary, lockshield valves.
- For medium-sized to large buildings, it is necessary to use static or automatic balancing valves for the distribution of water in the system.
- Static balancing with balancing valves is suitable for systems with a constant water volume, e.g. one-pipe heating systems or cooling systems with fan coil units or chilled ceilings. In two-pipe systems, balancing valves are only suitable under certain circumstances. They enable hydronic balancing according to the initial design. When operating under partial load, individual areas can be oversupplied due to the increase in differential pressure. This results in wasted energy or flow noises from the system.
- Dynamic balancing with automatic balancing valves is the optimal solution for systems with fluctuating water volumes, such as two-pipe heating systems with radiators. In contrast to static balancing only the amount of water that is required flows through the system, independent of load. This makes the system more economically efficient.
- A reliable system and the highest level of comfort, while keeping operating costs to a minimum are justifications for the installation of high-quality system components. These already pay off after a few years.
- The documentation on services supplied, together with an introduction to using the hydronically regulated system, all provide the customer with the necessary security and improve the market value of the property.

2.4 Balancing Strategies

There are two basic strategies for practical implementation of hydronic balancing, which are:

- Balancing at the consumers, and
- Balancing in the circuit.

2.4.1 Balancing at the consumers

In this balancing strategy, the pressure is first reduced at the consumers. Each consumer must therefore be equipped with the appropriate throttle valve and, if balancing is to be performed using flow rate measurement, also with an appropriate measuring valve.

It is important to take into account that each time a heat exchanger is adjusted, this also affects the flow rates of the heat exchangers that have already been set, due to the changing pressure relationships in the pipes. In this balancing strategy, it is therefore important that the pump and the network react as little as possible to changes in the flow rate. One ideal solution is the installation of automatic circuit control valves (section 2.2). These maintain a constant differential pressure within the circuit.

Balancing at consumers reaches its natural limitations as soon as the restriction of the valves result in a greater differential pressure than is practical for the fittings. For thermostatic radiator valves and lockshield valves, this is a maximum of 200 mbar.

2.4.2 Balancing in the Circuit

In order for balancing to take place in the circuit, all circuits must be equipped with throttle valves and, if balancing is to be performed using flow rate measurement, also with an appropriate measuring valve. Balancing circuits according to measurements is performed in hierarchical fashion: first the risers, then the circuits, and then the mains. This procedure may at first seem confusing, as at first sight it would appear more practical to balance larger water volumes first, followed by the smaller volumes. The following descriptions should explain why this hierarchical procedure is preferred.

Measurement based balancing is still the most frequently used method, as it is often not possible to set the system according to calculations. The fittings are then incorrectly set, or set based on what “feels right”. If first the circuit and then the risers within the circuit are balanced, it can happen that the throttle valve furthest away from the pump is also throttled. Each unnecessary restriction causes higher energy consumption as the pump has to compensate for the unnecessary differential pressure caused by the restriction.

It is therefore more practical to restrict the risers only as far as to allow at least one balancing valve to stay completely open, even if the required circuit water volume is still exceeded. When the individual circuits are balanced with each other in this way, balancing takes place in the circuits, so any excess water volume is restricted to the required amount in the circuits.

The following diagram shows the order of the throttle valves for balancing in the circuit. First the risers are balanced, i.e. pipes A1 and A2 and B1 and B2 with each other. Then the circuits A and B are balanced. Finally, the water volume in the main circuit is set.

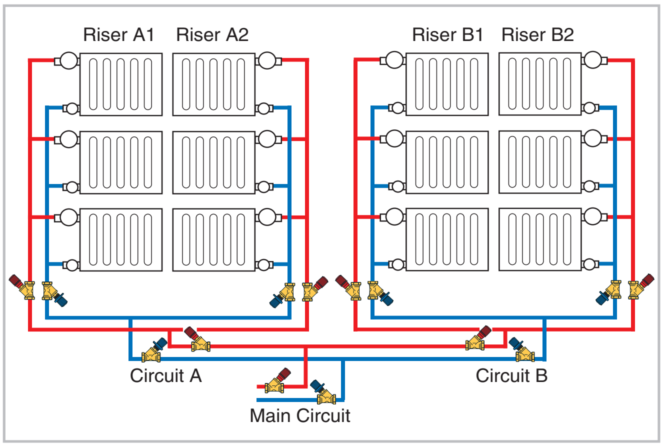


Fig. 10: Balancing in the circuit using Kombi-3-plus balancing valves

Note that the differential pressure set on the throttle valves of the risers can only be managed when the volume flow is 100%. If thermostatic radiator valves in the circuit restrict flow through the consumer, the circuit water volume is also reduced and therefore also the flow through the throttle valve. Halving the water volume reduces the differential pressure raised by the throttle valve to a quarter! The lack of differential pressure then needs to be compensated by the thermostatic radiator valves, which can lead to flow noises when working under partial load.

In practice, it is therefore highly recommended to balance the risers automatically. The manual throttle valves are replaced by automatic differential pressure balancing valves, e.g. Kombi-3-plus with Kombi-DU diaphragm unit.

2.5 Methods of Balancing

The underlying difficulty in hydronic balancing is that the necessity usually arises from insufficient supply of a heat exchanger or a circuit. In addition, the corresponding valves are fully open, which means the missing water volume must be supplied by throttling valves to divert water from other consumers or circuits to the under supplied component. Throttling the valve reduces the water volume flow, and the reduced volume influences all other consumers and circuits.

For this reason, specific balancing methods are required, which can be used for the previous balancing strategies. Some of these methods are described in the following:

2.5.1 The temperature method

This method is suitable for balancing by circuit as well as for balancing at the consumers, especially in the renovation of smaller systems.

- **Advantage:** Balancing is based purely on temperature measurements, so no flow measurements are required.
- **Disadvantage:** Time consuming, as each time you need to wait for the system to settle back into a steady state. This can only be performed if the outside temperature is sufficiently low (below 0°C, with no sun). Modern systems also regulate demand, by adjusting (lowering) the supply temperature when the outside temperature increases. The difference between the supply temperature and the room temperature is therefore lower, and an error of just 1°C gets more weight.

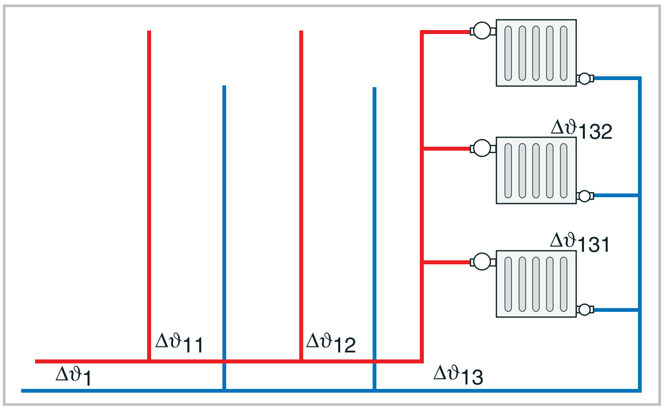


Fig. 11: The temperature method

Assuming that all consumers are correctly dimensioned, when the system is hydronically balanced, the same temperature difference should be set in all circuits and consumers.

$$\Delta\vartheta_1 = \Delta\vartheta_{11} = \Delta\vartheta_{12} = \Delta\vartheta_{13} = \Delta\vartheta_{131} = \Delta\vartheta_{132}$$

Only the temperature drop in the main circuit is not taken into account. In the temperature method, all circuits and consumers should be set to the same temperature difference.

In the simplest case, the temperature difference in the main circuit (i.e. in the pump) can be used as the target temperature difference.

The target K_v value can then be calculated as follows:

$$K_{v,\ target} = K_{v,\ actual} \frac{\Delta\vartheta_{actual}}{\Delta\vartheta_{target}}$$

For the actual temperature difference, the measured difference between the supply and return temperature needs to be set at the appropriate throttle valve. The target K_v value can also be determined from the current operating conditions (inside and outside temperature) and the design data (maximum supply and return temperature, minimum outside temperature, heat gain expressed in Kelvin):

$$K_{v,\,target} = K_{v,\,actual} \frac{(\vartheta_S - \vartheta_R)\,(20 - \vartheta_{o,\,min})}{(\vartheta_{S,max} - \vartheta_{R,\,max})\,(\vartheta_i - \Delta\vartheta_{HG} - \vartheta_o)}$$

ϑ_S	measured supply temperature
ϑ_R	measured return temperature
$\vartheta_{S,\,max}$	design supply temperature
$\vartheta_{R,\,max}$	design return temperature
ϑ_o	measured outside temperature
$\vartheta_{o,\,min}$	design outside temperature
ϑ_i	measured room temperature
$\Delta\vartheta_{HG}$	estimated heat gain (in Kelvin)

2.5.2 The proportional method

This is the simplest method of circuit balancing for systems with measuring valves.

- **Advantage**: Can be performed by one person using a single measuring device.
- **Disadvantage**: Several runs are required.

The proportional method is based on the fact that several parallel flows, which have a certain interrelationship, maintain the same relationship even if the overall flow is changed.

Firstly, all flows are aligned to the same flow quotient. This is defined as follows:

$$\frac{\text{ACTUAL - flow}}{\text{TARGET - flow}}$$

Then the overall flow is adjusted until the flow quotient is 1.0.
A detailed description of the proportional method is in chapter 4.

2.5.3 The reference circuit method

This method is suitable for balancing larger systems with main, circuit and risers, which all need to be balanced against each other.

- **Advantage:** Can be carried out quickly and efficiently. Only needs to be carried out once.
- **Disadvantage:** Requires more resources than the proportional method (3 people!)

This method generally requires 3 people (A, B and C) and 2 measuring devices (pressure difference and flow rate)

- A reads the pressure difference and flow on the reference circuit (= least efficient circuit), and reports this value via radio to B.
- B continually adjusts the overall flow rate so that the reference value at A remains constant.
- C balances the remaining circuits, and any repercussions on the whole network are automatically compensated by A and B.

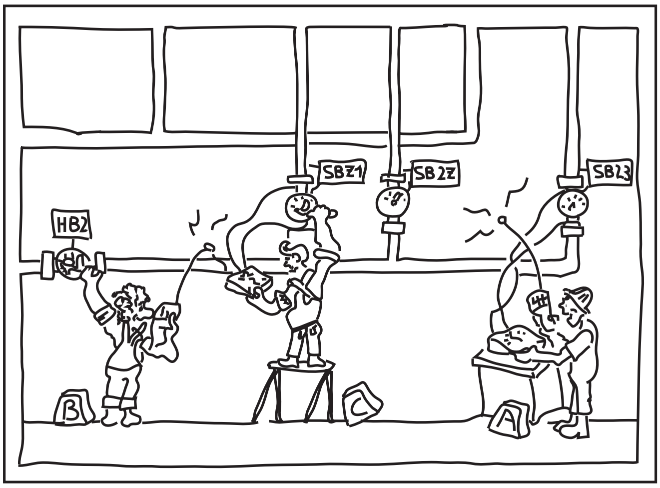


Fig. 12. The reference circuit method

2.5.4 Auxiliary method for balancing valves with pressure test cocks

When performing circuit balancing using balancing valves with pressure test cocks (Kombi-2-plus, Kombi-F-II), the pressure difference is measured and then the flow rate is calculated using the K_v value (which depends on the valve setting). Because the K_v value needs to be recalculated for each valve setting, the flow cannot be continually displayed - even when using a microcomputer - as the new valve setting needs to be entered following every run. Finding the right valve setting is thus a very gradual process and can be laborious.

This auxiliary method enables you to perform this task much faster: you can calculate the correct valve setting almost exactly using any two valve settings (balancing system with 2 unknowns). The solution is explained as follows:

$$K_{v,TARGET} = \sqrt{\frac{\dot{V}^2_{TARGET}}{H - \left(\frac{\dot{V}_{TARGET}}{K}\right)^2}}$$

with

$$K = \sqrt{\frac{\dot{V}^2_{2} - \dot{V}^2_{1}}{\left(\frac{\dot{V}_1}{K_{V1}}\right)^2 - \left(\frac{\dot{V}_2}{K_{V2}}\right)^2}}$$

$$H = (\dot{V}_1 / K_{V1})^2 + (\dot{V}_1 / K)^2$$

$\dot{V}$ = volume flow in m^3/h

Index 1 = 1st measurement
Index 2 = 2nd measurement

2.5.5 The Honeywell measuring method

The Honeywell measuring method makes use of the fact that for every balancing valve in the return, there is a shut-off valve in the supply. If pressure test cocks are mounted on a valve in the supply, each changed setting of the return valve results in a changed differential pressure via the supply valve. Using an electronic measuring computer, the flow rate can be continually displayed. It is calculated on the basis of the K_{vs} value of the valve on the supply side.

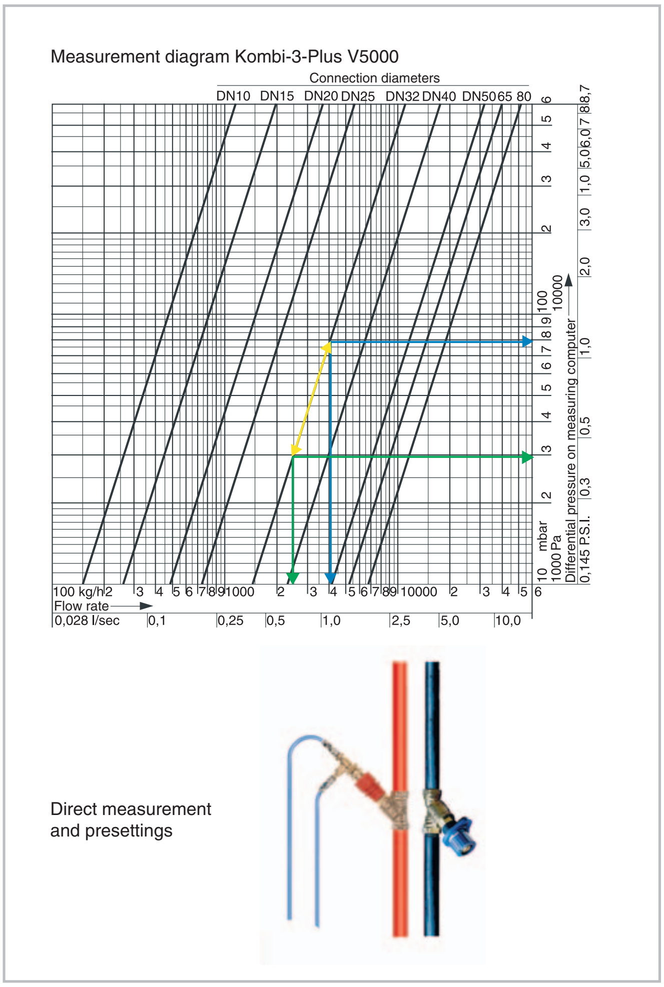


Fig. 13: The Honeywell measuring method

3. PROCEDURE

3.1 Planning and Preparation Work

The foundations for successful hydronic balancing begin at the planning stage. The system should be divided into separate logical sections (subareas): supply pipes, risers, floor distribution, radiators and thermostats. The advantages and disadvantages of the most common systems of distribution are described in chapter 3.1.2.

Next, the required valves are determined for the supply pipes, risers, floor distribution and radiators. You also need to plan where the appropriate shut off valves will be located in the system, which can be used to isolate different parts of the system in case of repairs or construction work.

The dimensioning of the pipework is then designed, along with determining the position of the balancing valves and selecting suitable speed-regulated pumps.

3.1.1 Determining the required mass flow rates

In new systems, the required mass flow rates of the individual consumers can be calculated using the heat or cool load and the final dimensions of the heat exchangers. This usually provides reliable planning values. In old systems, the heat or cool load cannot be calculated, or at least not easily. Alternatives are therefore required in order to determine the required mass flow rate as accurately as possible. It may help to consider that a deviation in the mass flow rate of approx. ± 10% of the demand can hardly be noticed (see fig. 2). A system balanced within this tolerance range therefore saves more energy than an unbalanced system. In an unbalanced system, the inefficient consumers can only be supplied by increasing pump energy.

Here, it is helpful to determine the overall requirements of the building. This value can be divided among the different rooms according to area in square metres. Some additional adjustments are still required to take different proportions of external surface area and window surface area into account.

Calculation of overall energy demand according to available cost calculations

When replacing or renovating a heating system, records of energy costs in recent years are often available. If the hours in which the heating system was in operation are also known, you can calculate the heat requirement of the building using the following formula:

$$P_B = \frac{B_a \times H_u \times \eta_{ges}}{b_{vH}}$$

P_B	Building heat requirement
B_a	Annual fuel consumption
H_u	Heating value
η_{tot}	Annual degree of utilization (from manufacturer's documentation or standards)
b_{vH}	Operating hours

Use of specific tables and diagrams

Energy providers use specific diagrams and tables in consulting discussions, which can be used to determine the heat requirement of the building based on the age of the building.

Estimated values based on permitted limits

In many countries, there are limit values for the maximum permitted specific heat demand. These values also distinguish between buildings of different ages. As this is a maximum value, using this value will always result in over-dimensioning. It is therefore advisable to check the result by calculating the heat requirements of some rooms (if possible).

Calculation based on the installed heating surface

This procedure is used the most frequently, as the performance of the heating surfaces can be taken from the manufacturer's documentation. This is also, however, the least accurate of the methods described here. The supply and return temperatures that were originally used to design the heating surfaces are not known. Heating surfaces are generally not set to the required performance, but are overdimensioned.

Examples:

- The required length was not available, and the next largest radiator was selected.
- For aesthetic reasons, the radiator size was adjusted to match the width of the window.
- The building has been insulated, but the existing radiators were not replaced.

After demand has been determined based on the possibilities described above, the heating surfaces have to be checked in order to calculate the required mass flow rate. If the manufacturer's documentation for the installed heating surfaces is still available, the required radiator power at the intended system temperatures can be determined according to the following general rules:

The power of a radiator is calculated according to the formula:

$$P = K \times A \times \Delta\vartheta$$

P	Radiator power
K	Heat transfer coefficient
A	Radiator surface, from manufacturer's documentation
$\Delta\vartheta$	average logarithmic temperature difference from supply temperature, return temperature and room temperature

$$\vartheta\Delta = \frac{(\vartheta_S - \vartheta_i) - (\vartheta_R - \vartheta_i)}{\ln\left(\frac{\vartheta_S - \vartheta_i}{\vartheta_R - \vartheta_i}\right)}$$

ϑ_S	Supply temperature
ϑ_R	Return temperature
ϑ_i	Room temperature

In the manufacturer's documentation, the "standard performance" P_N for the nominal conditions according to DIN EN 442, which is calculated based on ϑ_S=75°C, ϑ_R=65°C, $\vartheta_{i\,Room}$=20°C. The power output is often also specified according to the old standard at ϑ_S=90°C, ϑ_R=70°C, $\vartheta_{i\,Room}$=20°C. If the supply and return temperatures deviate from these conditions, the actual radiator output is also lower. The actual power output of a radiator is calculated according to the formula:

$$P = P_N \times \left(\frac{\vartheta\Delta_N}{\vartheta\Delta}\right)^n$$

P	Actual heat output of the radiator
P_N	Nominal heating performance at 90/70/20°C or 75/65/20°C
$\vartheta\Delta_N$	Average logarithmic temperature difference under nominal conditions 90/70/20°C or 75/65/20°C
$\vartheta\Delta_N$ =	59.44 K at 90/70/20°C
$\vartheta\Delta_N$ =	49.83 K at 75/65/20°C
$\vartheta\Delta$	Average logarithmic temperature difference under initial design conditions
n	Radiator exponent, approx. 1.3. More precise specifications can be found in the manufacturer's documentation.

The heat emitted is supplied by the heating medium and calculated according to the formula:

$$P = q \times c_p \times (\vartheta_S - \vartheta_R)$$

P	Performance of the heating medium in W
q	Mass flow rate in kg/s
c_p	Specific heat capacity of the medium
c_p =	4186 J/kg K for water at 60°C
ϑ_S	Supply temperature in °C
ϑ_R	Return temperature in °C

The general rules named above are illustrated in the radiator diagram (fig. 14 and on page 14). With the aid of this diagram, you can therefore work out the actual power output for a known nominal heat output for the selected supply and return temperatures.

Example:

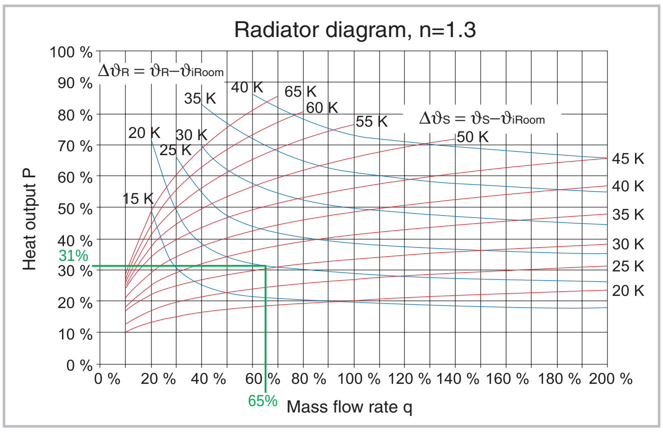


Fig. 14: Radiator diagram for nominal conditions ϑ_S=90°C, ϑ_R=70°C, $\vartheta_{i\,Room}$=20°C

The nominal heat output of an installed radiator is 1200 W at 90/70/20°C. We need to know the actual heat output and the required mass flow rate at the selected system temperatures of 50/40/20°C.

The power output is then

$$P = 0.31 \times 1200\ W = 372\ W$$

The nominal mass flow rate at heat output 1200 W is

$$q_N = \frac{1200\ W}{4186\ J/kg\ K \times 20\ K} = 0.014\ kg/sec = 51.6\ kg/h$$

The required mass flow rate can now be derived

$$q = 0.65 \times 51.6\ kg/h = 33.54\ kg/h$$

In some circumstances, it is necessary to further reduce the calculated heat output and therefore also the mass flow rate, as the nominal output has been calculated under ideal conditions. Other influencing factors are, for example, installation in radiator alcoves or behind cladding, and the type of pipe connection used:

Reduced output	due to installation type
Under shelves or window sills	-2% … -4%
In alcoves, open at the front	-4% … -8%
Behind radiator cladding, open at the top	-4% ... - 8%
Behind radiator cladding, closed at the top	-15% ... - -25%

Tab. 1: Reduced power due to particular installation situations

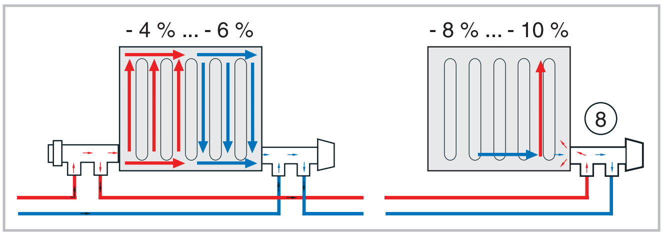


Fig. 15: Reduced power due to particular connection variants

3.1.2 Influence of the heat emission system

Radiator heating in a two-pipe system

Two pipes are laid to every radiator, one for supply, and one for return. In central Europe, two-pipe systems with cellar distribution and riser pipes are the most common type of system.

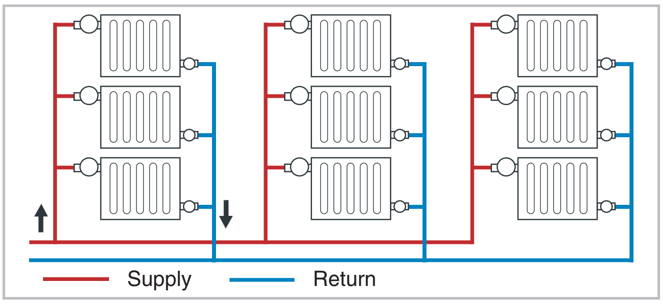


Fig. 16: Two-pipe installation with risers

- **Advantage**: Radiators have a relatively low inertia, and can therefore be easily controlled. All radiators have the same supply temperature. This means that the design is relatively simple and transparent, and thermostatic radiator valves have relatively little influence on the heating output of neighbouring radiators. These systems also have a long lifespan and subsequent alterations and enhancement are simple.
- **Disadvantage**: These systems require considerably more input of resources in terms of materials and work than single pipe systems. Heat metering of individual flats in systems with risers is not possible.

Hydronic balancing: Ideal conditions for simple balancing. All balancing strategies and methods described above can be used.

Radiator heating in a one-pipe system

The supply and return pipes are connected to the radiators by a circular loop made of copper, soft steel, or plastic pipes. These are usually laid in the floor. It is also possible to lay them in the skirting boards (e.g. in renovations).

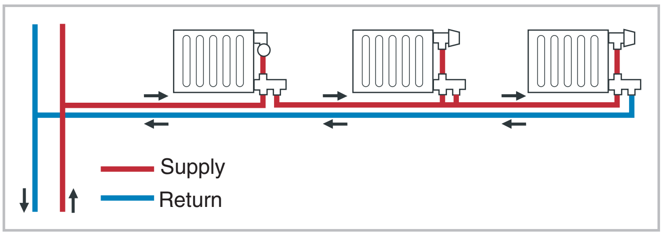


Fig. 17: One-pipe installation

- **Advantage**: Fewer riser pipes and saves material and labour. The loops can be installed by auxiliary personnel (no welding necessary). Heat metering per individual flat is possible.
- **Disadvantage**: Uneven water temperature in the radiators and therefore difficult to design. Thermostatic radiator valves influence the heating performance of downstream radiators. No large temperature drop can be achieved between supply and return.

Hydronic balancing: Balancing between the loops is not a major problem, because these are designed as two-pipe systems. In contrast, balancing with flow rate measurement in the individual radiators within the loop is not possible, as there are no opportunities to take measurements. In this case, the only option is to balance the (very small and therefore difficult to measure!) temperature differences using the individual radiators.

Radiator heating in a two-pipe system with star formation

In contrast to the one-pipe system, in this case the individual radiators are not connected by a loop, but rather each radiator is connected to the distributor in a star formation.

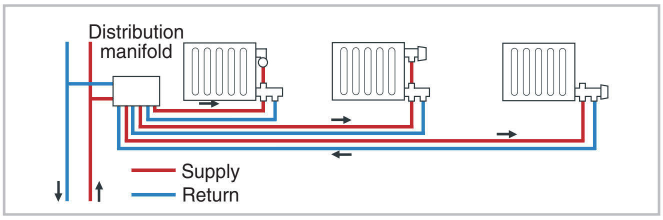


Fig. 18: Two-pipe installation with manifold

- **Advantages:** All radiators have the same supply temperature. This makes design relatively simple and transparent. Thermostatic radiator valves have little influence on the heat output of neighbouring radiators. Fewer risers, thus saves material and labour. Installation of radiators can be done by auxiliary personnel (no welding necessary). Meter reading for individual flats is possible.
- **Disadvantages**: Generally greater pipe friction loss than in conventional two-pipe heating systems, pipes are longer and of smaller size.

Hydronic balancing: Hydronic balancing is simple. The measuring points for flow rate measurement are best located in the manifold cabinet.

3.2 Preparations in the System

1. Study the position of the supply pipes, risers, floor distributors, radiators and thermostats on the construction plans and the specifications, check the flow rate, pressure drop, and control position of each valve.
2. All shutoff valves and balancing valves must be opened - set the control valves according to calculations or estimation. If the system is equipped with thermostatic radiator valves, all valves on the calculated nominal flow must be set to the calculated design flow. This setting can prove very difficult:
   - The valves have been designed for different proportional ranges X_p.
   - A proportional range X_p can not be set manually.
   - The radiator thermostat opens the valve more or less depending on room temperature.

   Due to their particular construction, Honeywell thermostatic radiator valves are limited to a maximum proportional range of $X_p = 3$ K. This enables you to achieve a value close to the required value when the thermostatic radiator valves are taken off during balancing. Presettable valves therefore also need to be set to the calculated value.
3. If the radiators are connected using 3-way bypass valves, the pressure drop across the bypass should be identical to the pressure drop across the heat exchanger. If this is the case, the hydronic relationships are the same in every operating state.
4. Set all the balancing valves to the presetting specified in the construction plans. If no data is available, set the valves to an estimated level. The valve furthest from the pump should be completely open. The closer the valve is to the pump, the smaller the required valve presetting.
5. Ensure that you have access to the flow diagrams and data sheets for the valves used.
6. To avoid later problems, thoroughly rinse and bleed the system before hydronic balancing is performed.
7. Then check the filling pressure and refill water if necessary.

System balancing should be recorded so that the system data is available if any building work takes place later.

4. THE PROPORTIONAL METHOD

The simplest and most common method for hydronic balancing of heating or cooling systems is the “proportional method”. This method is therefore explained in more detail using an example. The proportional method is called such because all values behave and change in relation to each other.

With the proportional method, heating and cooling systems are hydronically balanced in three steps:

1. Measurement of the actual flow rate and comparison with required flow rates
2. Balancing of the system
3. Restriction of the pump or balancing of the next level (e.g. circuit level, see chapter 2.4.2)

4.1 Measuring the Actual Flow Rate

The established actual flow rate and the desired target flow rate are used to determine the quotient of the relevant part of the system:

$$\text{Flow quotient } R = \frac{\text{Actual flow rate } q_{actual}}{\text{Target flow rate } q_{target}}$$

The values for each circuit are recorded in the measuring log:

		Measurement 1		Measurement 2	Measurement 3	Result
		Review		Run 1	Run 2	Run 3
Measuring point	Target-Flow rate q_{target} (kg/h)	Actual-Flow rate q_{actual}	Quotient R	Quotient R	Quotient R	Quotient R
Circuit 1	955	1165	1.22			
Circuit 2	225	353	1.57			
Circuit 3	515	695	1.35			
Circuit 4	215	232	1.08			

NOTE:

- If all quotients >1, the system or part of the system is overdimensioned.
- If all quotients <1, the system or part of the system is underdimensioned.

4.2 Balancing Circuits

The least efficient circuit can be identified by the lowest quotient, in our example this is circuit 4. No further throttling should be performed here, to avoid pump energy being wasted. The quotient of the least efficient circuit acts as the reference value for setting the rest of the circuits. Depending on the characteristic curve of the pump and network, the final value may deviate more or less from the reference value (greater flow rates thus have relatively more influence). For our example, the first approximate quotient is estimated at R=1.15. The circuits 1, 2 and 3 are then set to this value with the aid of the measuring device. This starts at the circuit with the highest quotient. In our example, the circuits are therefore set in the order 2, 3 and then 1. During the setting process, the individual circuits influence each other. Water that is restricted in one circuit remains available, but simply takes another route. For this reason, a step-by-step procedure is required. In the second run, for example, the settings are made to a quotient of 1.20.

The system is balanced when all circuits are balanced to the same quotients (within a certain tolerance range). In this example, the quotients could be as follows after the second measurement:

		Measurement 1		Measurement 2	Measurement 3	Result
		Review		Run 1	Run 2	Run 3
Measuring point	Target-Flow rate q_{target} (kg/h)	Actual-Flow rate q_{actual}	Quotient R	Quotient R	Quotient R	Quotient R
Circuit 1	955	1165	1.22	1.15	1.20	1.20
Circuit 2	225	353	1.57	1.15	1.20	1.20
Circuit 3	515	695	1.35	1.15	1.20	1.19
Circuit 4	215	232	1.08	?	?	1.21

4.3 Reducing the Quotients to R=1.0

All circuits are now in balance with each other, and the flow rate is too high by a factor of approximately R=1.2. The flow rate must therefore be reduced centrally, e.g. by balancing to the next highest level. If no higher levels are available, it is possible to set the pump to a lower level or exchange the pump with a smaller type. This also saves electrical energy.

5. CALCULATION EXAMPLES

Following the description of hydronic balancing by measurement and adjustments in the previous sections, this section explains the estimated determination of thermostatic radiator valve settings, balancing valves, and differential pressure regulating valves based on two small examples. This calculation is normally sufficient, as a stationary state is assumed when designing the thermostatic radiator valves (see section 2.1.1).

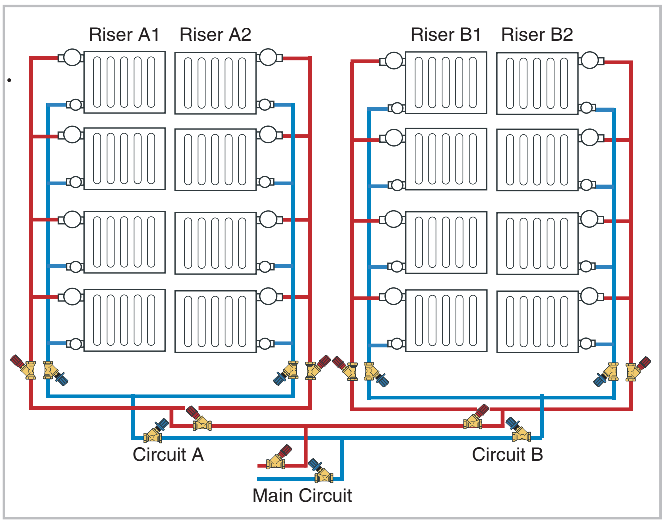


Fig. 19: Circuit diagram for the calculation example

5.1 Example of Thermostatic Radiator Valve Dimensioning

The model for the example is based on riser A1 and A2 in above circuit diagram. In the first step, the piping is dimensioned, and the dimensioning of the thermostatic radiator valves and lockshield valves is determined. To do this, the radiator length and the pipe lengths must be known.

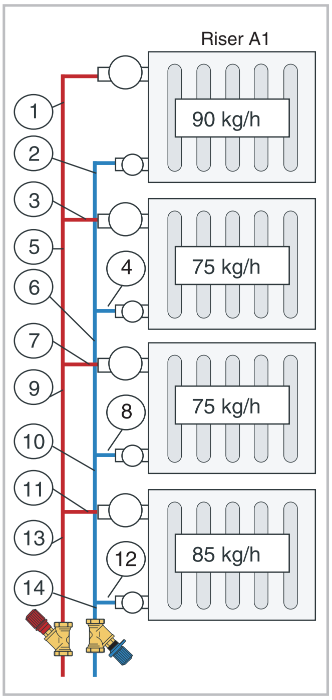


Fig. 20: Circuit diagram for the calculation example details

The dimensioning of the pipes requires at least a maximum permissible speed to prevent flow noises.

Radiator connectors are calculated with a maximum speed of 0.5 m/s, while 1.5 m/s is more normal for main distributor pipes. This speed, however, causes great pressure loss in the pipes, which requires high pump output. To limit pump output, the pressure losses in smaller systems are therefore set between 0.5 mbar/m and 1 mbar/m, and in larger systems between 1 mbar/m and 2 mbar/m. This example is based on a maximum pressure loss R ≤ 1.5 mbar/m.

The pressure losses depend on several factors, including the roughness of the internal surface of the pipes and the density of the medium. They can be calculated or derived from diagrams. Fig. 21 shows a diagram of this type for steel pipes and heating water at 40°C.

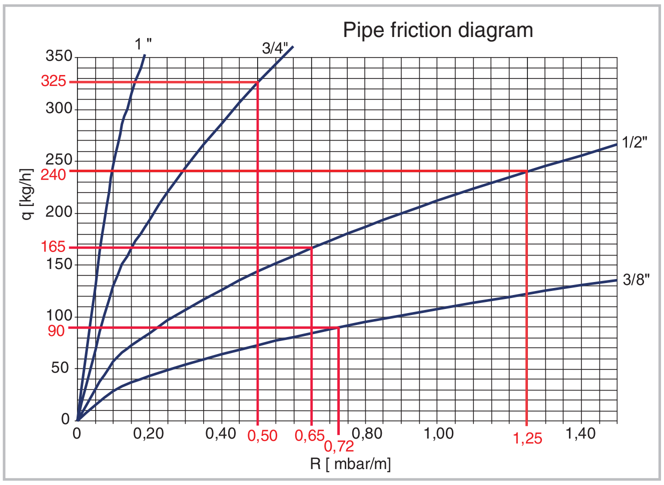


Fig. 21: Pipe friction diagram for determining pressure loss in parts of the system, in mbar/m

Section S	q (kg/h)	L (m)	Dim.	Δp_{pipe} mbar/m	Δp_{pipe} mbar
1	90	3.60	3/8"	0.72	2.59
2	90	3.10	3/8"	0.72	1.30
3	75	0.10	3/8"	0.53	0.05
4	75	0.10	3/8"	0.53	0.05
5	165	3.50	1/2"	0.65	2.28
6	165	3.50	1/2"	0.65	2.28
7	75	0.10	3/8"	0.64	0.06
8	75	0.10	3/8"	0.64	0.06
9	240	3.50	1/2"	1.25	4.38
10	240	3.50	1/2"	1.25	4.38
11	85	0.10	3/8"	0.64	0.06
12	85	0.10	3/8"	0.64	0.06
13	325	1.00	3/4"	0.50	0.50
14	325	0.50	3/4"	0.50	0.25

Tab. 2: Pressure loss of different sections of piping

Dimensioning with V type thermostatic radiator valve

For determining the least efficient flow route, the pressure losses of the open thermostatic radiator valves and lockshield valves are taken from the manufacturer's diagrams or calculated using the K_v-value in the first calculation run. Presettable valves are designed with preset values or K_v values of $X_p \leq 2$ K. For the V type thermostatic radiator valve, this is K_v=0.41.

$$K_v = \dot{V} / \sqrt{\Delta p} \qquad \Delta p = (\dot{V} / K_v)^2$$

K_v	Flow capacity in m³/h at 1 bar pressure loss
$\dot{V}$	Volume flow in m³/h
Δp	Pressure loss in bar

The mass flow rates in kg/h have been determined for the individual radiators from the dimensioning of the radiators, (see fig. 20). For the above formula, however, we need the volume flow in m³/h. The mass flow therefore needs to be converted to the volume flow using the density ρ. The density of water at 4°C ρ = 1 kg/dm³ and at 90°C, is still ρ = 0.96 kg/dm³. For this reason, the conversion is not used in practice and the factor 1 is used for all temperatures. This results in the following values:

V type thermostatic radiator valves

Highest-level radiator **HK1**:

$\Delta p = (\dot{V}/K_v)^2 = (0.09\ \text{m}^3/\text{h} / 0.41)^2 = 0.048$ bar = 48.19 mbar

Middle radiators **HK2** and **HK3**:

$\Delta p = (0.075\ \text{m}^3/\text{h} / 0.41)^2 = 0.03346$ bar = 33.46 mbar

Lower radiator **HK4**:

$\Delta p = (0.085\ \text{m}^3/\text{h} / 0.41)^2 = 0.04298$ bar = 42.98 mbar.

Lockshield valve, Verafix, angle pattern:

Highest radiator **HK1**:

$\Delta p = (V/K_v)^2 = (0.09\ \text{m}^3/\text{h} / 1.7)^2 = 0.0028$ bar = 2.80 mbar

Middle radiators **HK2** and **HK3**:

$\Delta p = (0.075\ \text{m}^3/\text{h} / 1.7)^2 = 0.00195$ bar = 1.95 mbar

Lower radiator **HK4**:

$\Delta p = (0.085\ \text{m}^3/\text{h} / 1.7)^2 = 0.00250$ bar = 2.50 mbar

		Dim.	L m	Δp mbar/m	HK1	HK2	HK3	HK4
q	kg/h				90	75	75	85
Δp_{valve}	mbar				48.19	33.46	33.46	42.98
$\Delta p_{Lockshield}$	mbar				2.80	1.95	1.95	2.50
$\Delta p_{pipe\ S1}$	mbar	3/8	3.6	0.72	2.59			
$\Delta p_{pipe\ S2}$	mbar	3/8"	3.1	0.72	1.30			
$\Delta p_{pipe\ S3}$	mbar	3/8"	0.1	0.53		0.05		
$\Delta p_{pipe\ S4}$	mbar	3/8"	0.1	0.53		0.05		
$\Delta p_{pipe\ S5}$	mbar	1/2"	3.5	0.65	2.28	2.28		
$\Delta p_{pipe\ S6}$	mbar	1/2"	3.5	2.28	2.28			
$\Delta p_{pipe\ S7}$	mbar	3/8"	0.1	0.53			0.05	
$\Delta p_{pipe\ S8}$	mbar	3/8"	0.1	0.53			0.05	
$\Delta p_{pipe\ S9}$	mbar	1/2"	1.0	1.25	4.38	4.38		
$\Delta p_{pipe\ S10}$	mbar	1/2"	0.5	1.25	4.38	4.38		
$\Delta p_{pipe\ S11}$	mbar	3/8"	0.1	0.66			0.07	
$\Delta p_{pipe\ S12}$	mbar	3/8"	0.1	0.66			0.07	
$\Delta p_{pipe\ S13}$	mbar	3/4"	1.0	0.50	0.50	0.50	0.50	0.50
$\Delta p_{pipe\ S14}$	mbar	3/4"	0.5	0.50	0.25	0.25	0.25	0.25

Tab. 3: Riser A1, pressure losses at thermostatic radiator valve connectors and pipes

Note that a large proportion of the pressure loss is generated by the thermostatic radiator valve and the loss from the pipe and lockshields are almost negligible. For this reason, it is not necessary to determine the pressure loss for T-joints, bends etc.. The pressure loss from the radiator itself is also negligible. In contrast, pressure loss from fan coil units or heat exchangers can be in the range of several 100 mbar and must be included in the overall calculation.

The system is balanced, when the sum of all pressure losses across the individual flow routes is equal. To achieve this, the individual pressure losses are compared in the table below:

		Dim.	L m	Δp mbar/m	HK1	HK2	HK3	HK4
q	kg/h				90	75	75	85
Δp_{total}	mbar				68.95	49.58	45.02	46.37
$\Delta p_{Differential}$	mbar					19.37	23.93	22.58

Tab. 4: Pressure loss balance in riser A1

This overview shows that the greatest differential pressure is required to supply the highest radiator, HK1. The supply to this radiator is therefore the least efficient flow route. To avoid oversupply of radiators HK2, HK3 and HK4, the same differential pressure is required. This means that the thermostatic radiator valve must also increase this differential pressure:

Required differential pressure for the thermostatic radiator valve at HK2:

$\Delta p_{THV,\ target} = \Delta p_{THV} + \Delta p_{Differential} = 33.46 + 19.37 = 52.83$ **mbar**

Required differential pressure for the thermostatic radiator valve at HK3:

$\Delta p_{THV,\ target} = \Delta p_{THV} + \Delta p_{Differential} = 33.46 + 23.93 = 57.39$ **mbar**

Required differential pressure for the thermostatic radiator valve at HK4:

$\Delta p_{THV,\ target} = \Delta p_{THV} + \Delta p_{Differential} = 42.98 + 22.58 = 65.56$ **mbar**

In the function diagram of the thermostatic radiator valve type V, we now need to find the intersection between the mass flow rate and required differential pressure.

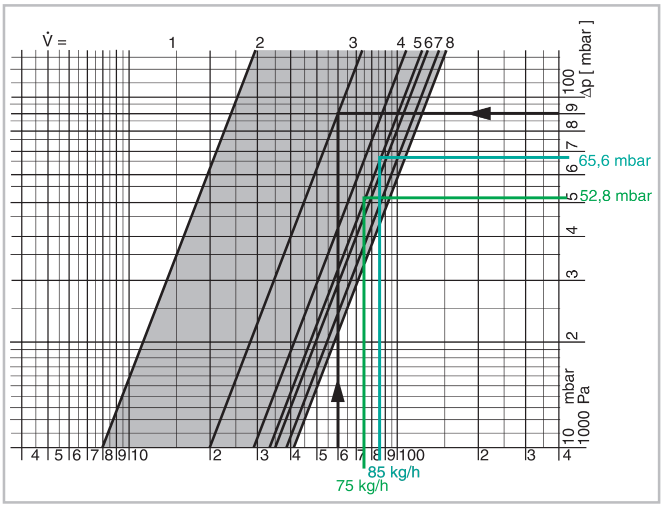


Fig. 22: Function diagram for V type thermostatic radiator valve

Or, the presetting is determined based on the K_v value and the presetting table:

	$\dot{V}$	1	2	3	4	5	6	7	8
X_p=1 K	K_v	0.04	0.06	0.13	0.16	0.19	0.19	0.19	0.19
X_p=2 K	K_v	0.04	0.08	0.20	0.29	0.33	0.35	0.38	0.41
X_p=3 K	K_v	0.04	0.09	0.21	0.34	0.42	0.44	0.51	0.55

Tab. 5: Presetting and K_v value for V type thermostatic radiator valve

Presetting **HK2**:

$K_v = \dot{V} / \sqrt{\Delta p} = 0.075\ \text{kg/h} / \sqrt{0.05283} = \mathbf{0.33}$ $\quad \dot{V} = 5$

Presetting **HK3**:

$K_v = \dot{V} / \sqrt{\Delta p} = 0.075\ \text{kg/h} / \sqrt{0.05739} = \mathbf{0.31}$ $\quad \dot{V} = 5$

Presetting **HK4**:

$K_v = \dot{V} / \sqrt{\Delta p} = 0.085\ \text{kg/h} / \sqrt{0.06556} = \mathbf{0.33}$ $\quad \dot{V} = 5$

The above table only contains a preset value that corresponds to the calculated K_v value for the radiators HK2 and HK4. The setting is therefore very simple, presetting 5. The calculated K_v value for the radiator HK3 is between the settings $\dot{V} = 4$ and $\dot{V} = 5$. The setting $\dot{V} = 5$ is also selected in this case. The valve is then set between X_p=2K and X_p=1K.

Dimensioning with BB type thermostatic radiator valve

In the previous example, the presettings of the lower radiators are the same, and only the presetting for the radiator with the longest flow route is slightly higher. This is because the mass flow rates only vary slightly. The necessity for balancing in this case is mainly a result of the varying flow routes. In installations of this type, it is sufficient if the radiators are connected via the more cost-effective BB type thermostatic radiator valve.

The following example shows a calculation of the riser A2 with the BB type thermostatic radiator valve. The pressure losses of the pipes are taken from the pipe friction diagram as before. To determine the least efficient flow route, in this example, in the first calculation the pressure loss of the open / non-preset thermostatic radiator valve is applied. For limited thermostatic radiator valves, it is still common practice to start the calculation with the K_v value at X_p = 3 K. In the example, the K_v value K_v = 0.52 is used for the calculation.

The water volumes for the individual radiators are:

Radiator	HK1	HK2	HK3	HK4
Water volume	80 kg/h	65 kg/h	65 kg/h	75 kg/h

This results in the following values:

BB type thermostatic radiator valves

Highest-level radiator **HK1**:

$\Delta p = (\dot{V}/K_v)^2 = (0.08\ \text{m}^3\text{/h} / 0.52)^2 = 0.0237\ \text{bar}$

$\Delta p = 23.67\ \text{mbar}$

Middle radiators **HK2** and **HK3**:

$\Delta p = (0.065\ \text{m}^3\text{/h} / 0.52)^2 = 0.01563\ \text{bar} = 15.63\ \text{mbar}$

Lower radiator **HK4**:

$\Delta p = (0.075\ \text{m}^3\text{/h} / 0.52)^2 = 0.02080\ \text{bar} = 20.80\ \text{mbar}$

Lockshield valve, Verafix, angle pattern:

Highest radiator **HK1**:

$\Delta p = (\dot{V}/K_v)^2 = (0.08\ \text{m}^3\text{/h} / 1.7)^2 = 0.00221\ \text{bar} = 2.21\ \text{mbar}$

Middle radiators **HK2** and **HK3**:

$\Delta p = (0.065\ \text{m}^3\text{/h} / 1.7)^2 = 0.00146\ \text{bar} = 1.46\ \text{mbar}$

Lower radiator **HK4**:

$\Delta p = (0.075\ \text{m}^3\text{/h} / 1.7)^2 = 0.00195\ \text{bar} = 1.95\ \text{mbar}$

The result is listed in the table below:

		Dim.	L m	Δp mbar/m	HK1	HK2	HK3	HK4
q	kg/h				80	65	65	75
Δp_{valve}	mbar				23.67	15.63	15.63	20.80
$\Delta p_{Lockshield}$	mbar				2.21	1.46	1.46	1.95
$\Delta p_{pipe\ S1}$	mbar	3/8"	3.6	0.59	2.12			
$\Delta p_{pipe\ S2}$	mbar	3/8"	3.1	0.59	1.83			
$\Delta p_{pipe\ S3}$	mbar	3/8"	0.1	0.43		0.04		
$\Delta p_{pipe\ S4}$	mbar	3/8"	0.1	0.43		0.04		
$\Delta p_{pipe\ S5}$	mbar	1/2"	3.5	0.51	1.79	1.79		
$\Delta p_{pipe\ S6}$	mbar	1/2"	3.5	0.51	1.79	1.79		
$\Delta p_{pipe\ S7}$	mbar	3/8"	0.1	0.43			0.04	
$\Delta p_{pipe\ S8}$	mbar	3/8"	0.1	0.43			0.04	
$\Delta p_{pipe\ S9}$	mbar	1/2"	3.5	0.98	3.43	3.43		
$\Delta p_{pipe\ S10}$	mbar	1/2"	3.5	0.98	3.43	3.43	3.43	
$\Delta p_{pipe\ S11}$	mbar	3/8"	0.1					0.05
$\Delta p_{pipe\ S12}$	mbar	3/8"	0.1					0.05
$\Delta p_{pipe\ S13}$	mbar	3/4"	1.0	0.37	0.37	0.37	0.37	0.37
$\Delta p_{pipe\ S14}$	mbar	3/4"	0.5	0.19	0.19	0.19	0.19	0.19
Δp_{total}	mbar				40.83	28.17	24.59	21.46
$\Delta p_{Differential}$	mbar					12.66	16.24	19.37

Tab. 6: Pressure loss balance in riser A2

Required differential pressure for the thermostatic radiator valve at HK2:

$\Delta p_{THV,\ target} = \Delta p_{THV} + \Delta p_{Differential} = 15.63 + 12.66 = \mathbf{28.29}\ \text{mbar}$

Required differential pressure for the thermostatic radiator valve at HK3:

$\Delta p_{THV,\ target} = \Delta p_{THV} + \Delta p_{Differential} = 15.63 + 16.24 = \mathbf{31.87}\ \text{mbar}$

Required differential pressure for the thermostatic radiator valve at HK4:

$\Delta p_{THV,\ target} = \Delta p_{THV} + \Delta p_{Differential} = 20.80 + 19.37 = \mathbf{40.17}\ \text{mbar}$

In the function diagram of the thermostatic radiator valve, we now need to find the intersection between the mass flow rate and required differential pressure. For all radiators, this is within the proportional range X_p=1K….3K.

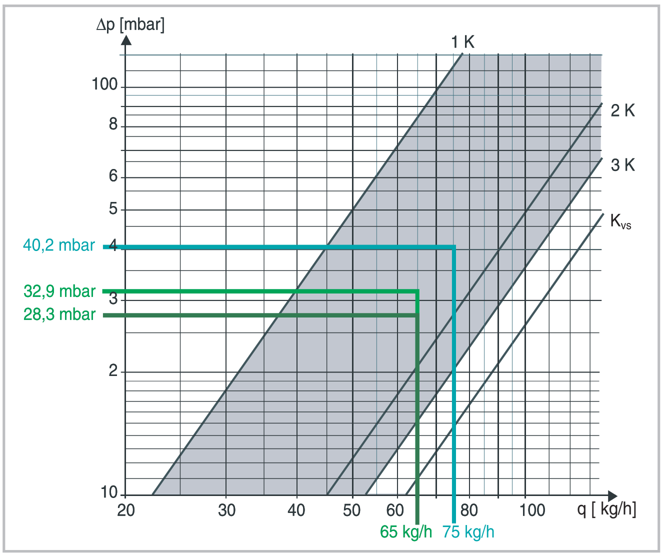


Fig. 23: Function diagram for BB type thermostatic radiator valve

Summary:
The system is thus balanced without any further presettings.

Dimensioning for horizontal distribution, originating from one heating circuit distributor.

In some regions, radiators are not connected as described above using vertical ascending pipes, but through horizontal distributor pipes within the living space. With this installation method, there are two possible variants:

Connection of the radiators in a type of tree structure with one main pipe and several supply pipes branching off it :

In this case, the same hydronic general rules apply that were described in the previous examples, so there is no need to enter into detail at this point.

Individual connection of the radiators, starting from one heating circuit distributor:

This connection method simplifies the calculation considerably, as, apart from the pressure loss from the fittings, there is only the friction loss from the pipes to consider in each flow circuit (see fig. 18). This procedure is explained with the following example.

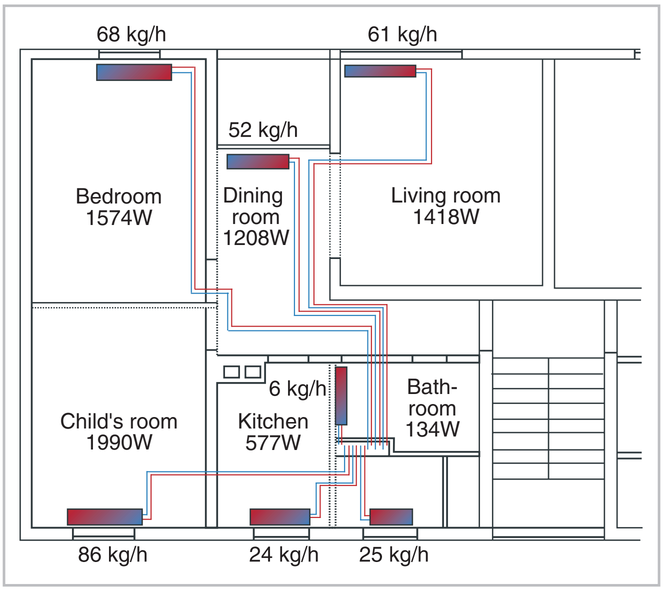


Fig. 24: Horizontal distribution, originating from one heating circuit distributor

In practice, horizontal systems of this type are available in copper, soft steel, plastic, or composite pipe. The same standard pipe diameter is used for all radiator connections, regardless of the required mass flow rate. In this example, the installation has 16 x 2 mm plastic pipes. Depending on the length of the pipe and the mass flow, the following pressure losses result:

Room		Bath	Bed	Dining	Living	Child	Kitchen1	Kitchen2
Flow rate	[kg/h]	6.00	68.00	52.00	61.00	86.00	24.00	25.00
Pipe Ø		16x2	16x2	16x2	16x2	16x2	16x2	16x2
Length	[m]	1.50	21.00	13.50	17.00	13.50	8.70	4.50
Δp_{pipe}	[mbar]	0.04	20.00	8.10	13.00	20.00	1.40	0.70
$\Delta p_{radiator}$	[mbar]	0.00	0.76	0.44	0.59	1.17	0.09	0.09
Δp_{TRv}	[mbar]	0.21	27.51	16.09	22.14	44.00	3.43	3.72
$\Delta p_{return\ flow}$	[mbar]	0.02	2.24	1.31	1.80	3.58	0.28	0.30
Δp_{total}	[mbar]	0.27	50.51	25.94	37.53	68.75	5.20	4.81

Tab. 7: Pressure losses when connecting via heating circuit distributors

The same applies here: The system is balanced when the pressure loss is equal across several flow paths. To determine the presetting, you therefore need to calculate the least efficient flow route and the required pressure via the thermostatic radiator valve. Together with the mass flow rate, this results in the K_v value, which is set using the presetting.

Room		Bath	Bed	Dining	Living	Child	Kitchen1	Kitchen2
Flow rate	[kg/h]	6.00	68.00	52.00	61.00	86.00	24.00	25.00
Δp_{total}	[mbar]	0.27	50.51	25.94	37.53	68.75	5.20	4.81
Difference	[mbar]	68.48	18.24	42.81	31.22	0.00	63.55	63.13
Required Δp_{TRV}	[mbar]	68.69	45.75	58.90	53.36	44.00	66.98	67.66
K_v-value		0.02	0.32	0.21	0.26	0.41	0.09	0.10
Pre-setting		1	5	4	4	8	3	3
X_P	K	1.30	1.95	1.38	1.77	2.00	0.68	0.74

Tab. 8: Determining the presetting

5.2 Dimensioning Examples for Balancing and Regulating Valves

To calculate the system balancing, the same principle is applied as when dimensioning the thermostatic radiator valves. The circuit water volume and the required differential pressure for supply can be compared with the water volume of a radiator or heat exchanger and its differential pressure. The balance then still has to be enhanced to include the balancing valves and the pressure losses of the connecting pipes.

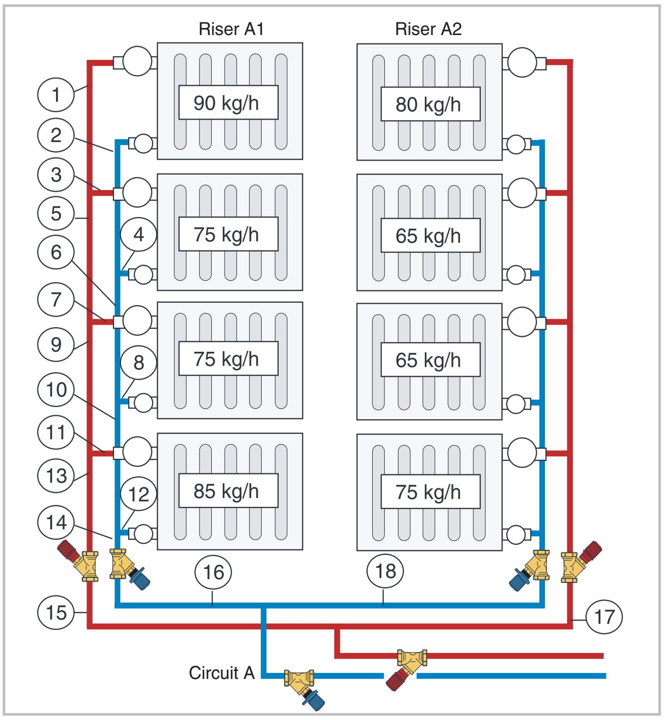


Fig. 25: Circuit balancing using Kombi-3-plus

Circuit A1:

Differential pressure of Kombi-3-plus, RED:

$\Delta p = (\dot{V}/K_v)^2 = (0.325\ m^3/h\ /4.5)^2 = 0.00522\ bar = 5.22\ mbar$

Differential pressure of Kombi-3-plus, BLUE, without presetting:

$\Delta p = (\dot{V}/K_v)^2 = (0.325\ m^3/h\ /6.4)^2 = 0.00258\ bar = 2.58\ mbar$

Circuit A2:

Differential pressure of Kombi-3-plus, RED:

$\Delta p = (\dot{V}/K_v)^2 = (0.285\ m^3/h\ /4.5)^2 = 0.00401\ bar = 4.01\ mbar$

Differential pressure of Kombi-3-plus, BLUE, without presetting:

$\Delta p = (\dot{V}/K_v)^2 = (0.285\ m^3/h\ /6.4)^2 = 0.00198\ bar = 1.98\ mbar$

		Dim.	L	Δp	Circuit A1	Circuit A2
q	kg/h				325.00	285.00
$\Delta p_{Kombi\text{-}3\text{-}RED}$	mbar				5.22	4.01
$\Delta p_{Kombi\text{-}3\text{-}BLUE}$	mbar				2.58	1.98
$\Delta p_{circuit}$	mbar				86.95	40.83
$\Delta p_{pipe\ S15}$	mbar	3/4"	5.0	0.50	2.50	
$\Delta p_{pipe\ S16}$	mbar	3/4"	4.5	0.50	2.25	
$\Delta p_{pipe\ S17}$	mbar	3/4"	4.5	0.37		1.67
$\Delta p_{pipe\ S18}$	mbar	3/4"	5.0	0.37		1.85
Δp_{total}	mbar				99.50	50.34
$\Delta p_{Differential}$	mbar					49.16

Tab. 9: Pressure loss balance when balancing circuits

At the balancing valve in circuit A2, an additional differential pressure of 49.16 mbar needs to be generated. The required differential pressure at this valve is then:

$$\Delta p = 40.83\ mbar + 49.16\ mbar = 89.99\ mbar$$

The corresponding presetting can be read from the diagram of the selected valve.

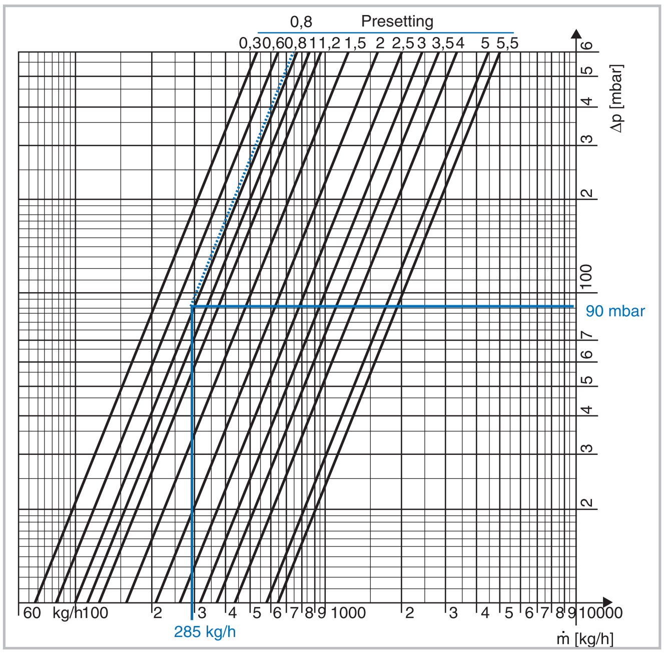


Fig. 26: Function diagram Kombi-3-plus, BLUE, DN 20

Or it is calculated using the K_v-value and the relevant table.

$$K_v = \dot{V} / \sqrt{\Delta p} = 0.285\ kg/h / \sqrt{0.08999} = 0.95$$

$\dot{V}$	0.3	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8
K_v	0.68	0.72	0.84	0.97	1.10	1.30	1.50	1.70	1.90
$\dot{V}$	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.4	3.6
K_v	2.10	2.30	2.50	2.70	2.91	3.12	3.36	3.60	3.86
$\dot{V}$	3.8	4.0	4.2	4.4	4.5	4.6	...	5.9 = open	
K_v	4.12	4.40	4.69	4.99	5.14	5.28	...	6.40	

Tab. 10: Presetting and K_v-value Kombi-3-plus, BLUE, DN 20

The required presetting is **0.8**.

5.3 Dimensioning Example for the Circuit Differential Pressure Regulator

After calculation, the next step for hydronic balancing is to balance the subsidiary circuits. This can be performed as described earlier, using manual throttle valves, or also with an automatic differential pressure control valve. As we have already described the calculation processes for the manual valves, we will now explain the dimensioning of the differential pressure control valve.

The differential pressure control valve provides a maximum differential pressure and a maximum water volume for the downstream circuit. Its dimensioning can therefore be compared to that of a pump.

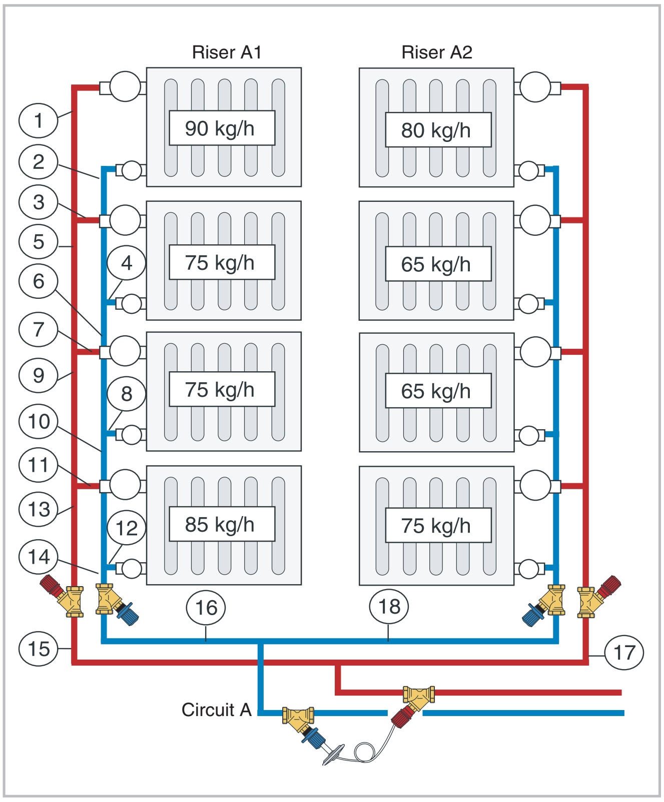


Fig. 27: Hydronic balancing using the differential pressure control valve

The water volume is formed from the sum of the water volume in riser A1 and A2:

$q = q_{riser\,1} + q_{riser\,2}$

$q_{riser\,1} =$ 90 kg/h + 75 kg/h + 75 kg/h + 85 kg/h

$q_{riser\,1} =$ **325 kg/h**

$q_{riser\,2} =$ 80 kg/h + 65 kg/h + 65 kg/h + 75 kg/h

$q_{riser\,2} =$ **285 kg/h**

q = 325 kg/h + 285 kg/h = **610 kg/h**

The required dimension of the differential pressure control valve can be simply described using the characteristic curve diagram of the valve.

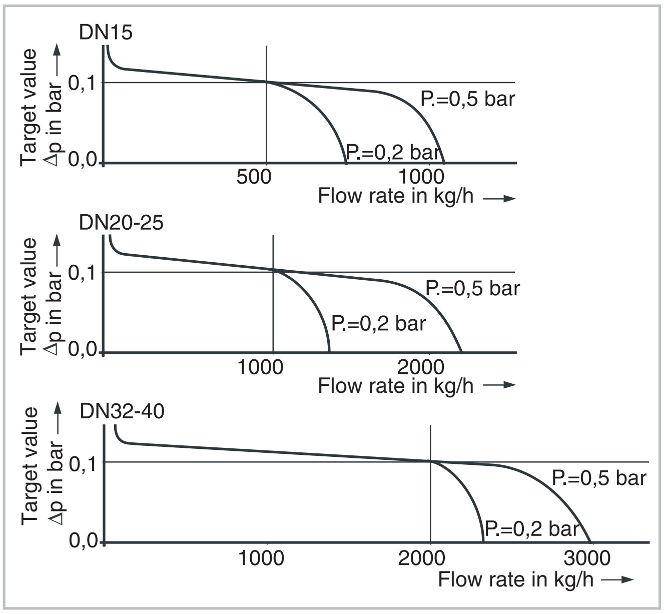


Fig. 28: Characteristic curves for the differential pressure control valve (automatic balancing valve)

The diagram for the standard size DN20 shows an intersection at 1000 kg/h with the set differential pressure 0.1 bar. The value is higher than the required water volume 610 kg/h, so the valve is big enough.

The required differential pressure is calculated from the pressure loss of the least efficient circuit and the pressure loss of the circuit sections 15 and 16. The least efficient circuit in our example is the riser A1 (see Tab. 9).

The pressure loss is Δp = 99.5 mbar.

The pressure losses of the connecting sections 15 and 16 and the overal balance are taken from the table below:

		Dim.	L m	Δp mbar/m	Circuit A1
q	kg/h				610.00
$\Delta p_{circuit}$	mbar				99.50
$\Delta p_{pipe\,S15}$	mbar	3/4"	3	0.50	1.50
$\Delta p_{pipe\,S16}$	mbar	3/4"	3	0.50	1.50
Δp_{total}	mbar				102.50

Tab. 11: Pressure loss balance in circuit

The required differential pressure 102.5 mbar is slightly higher than the factory setting for the differential pressure of 100 mbar. The valve is therefore sufficiently dimensioned.

If the calculation results in a differential pressure of more than 100 mbar, the target value of the membrane control valve can also be increased. Each turn of the spindle on the membrane control valve raises the target value by 15 mbar. After it is turned once, the set differential pressure is therefore:

$$\Delta p = 100 \text{ mbar} + 15 \text{ mbar} = 115 \text{ mbar}$$

If the required differential pressure is lower than the factory setting, a larger amount of water flows than is actually required. For further optimization, the differential pressure of the valve can be reduced to the pressure that is actually required, using the counter spring no, VA2502A001. This accessory reduces the factory-set lowest differential pressure to 0.05 bar. By adjusting the target value setting of the membrane control valve, the required value is then set. One turn of the target value spindle increases the differential pressure by 15 mbar. To increase the pressure by 50 mbar, e.g. to 72.5 mbar, therefore requires 1 1/2 turns.

The controller is now set to exactly the required differential pressure. This means that only the required water volume of 610 kg/h is permitted to flow, even if the max. capacity of the controller is 1000 kg/h.

The characteristic curve levels off at a pump pressure of p_0=0.2 bar. The available pump pressure in the circuit must therefore be 0.2 bar.

With this setting, the circuit is automatically and dynamically balanced. This means that when individual thermostatic radiator valves close under partial load conditions, the supply pressure does not increase: the other thermostatic radiator valves only receive the water volume that they require, and energy is saved.

6. PRODUCT OVERVIEW

6.1 Kombi-3-plus

The Honeywell Kombi-3-plus valve range consists of two valves and three accessory parts:

- Kombi-3-plus, RED measuring and shut-off valve
- Kombi-3-plus, BLUE balancing and shut-off valve
- Kombi-DU diaphragm unit
- Measuring kit
- Draining adapter

The valves are maintenance-free due to an O-ring seal, and easy to insulate as there are no additional connections on the valve housing.

The special feature is that the accessories allow additional functions to be subsequently added as required without interrupting operation: measurement, regulation, draining, and filling. All functions are performed via the valve spindle in the top section and can be simply upgraded during normal operation of the system.

The basics: Kombi-3-plus, RED and BLUE

The basic solution is already equipped for measuring or differential pressure control:

- Security from the start
- Fast upgradeability without interrupting operation

Upgrades: Kombi-3-plus, RED and BLUE with diaphragm unit

If required, the Kombi-3-plus, BLUE can be subsequently fitted with the diaphragm unit. The manual throttle valve then becomes an automatic differential pressure control valve

- Upgrade without interrupting operation
- Automatic hydronic balancing, including under partial load conditions
- Time consuming measuring and calibration is no longer necessary
- No flow noises

Technical data sheets

Technical data sheets on Kombi-3-plus valves can be downloaded from our Internet site.

http://europe.hbc.honeywell.com/products/.htm

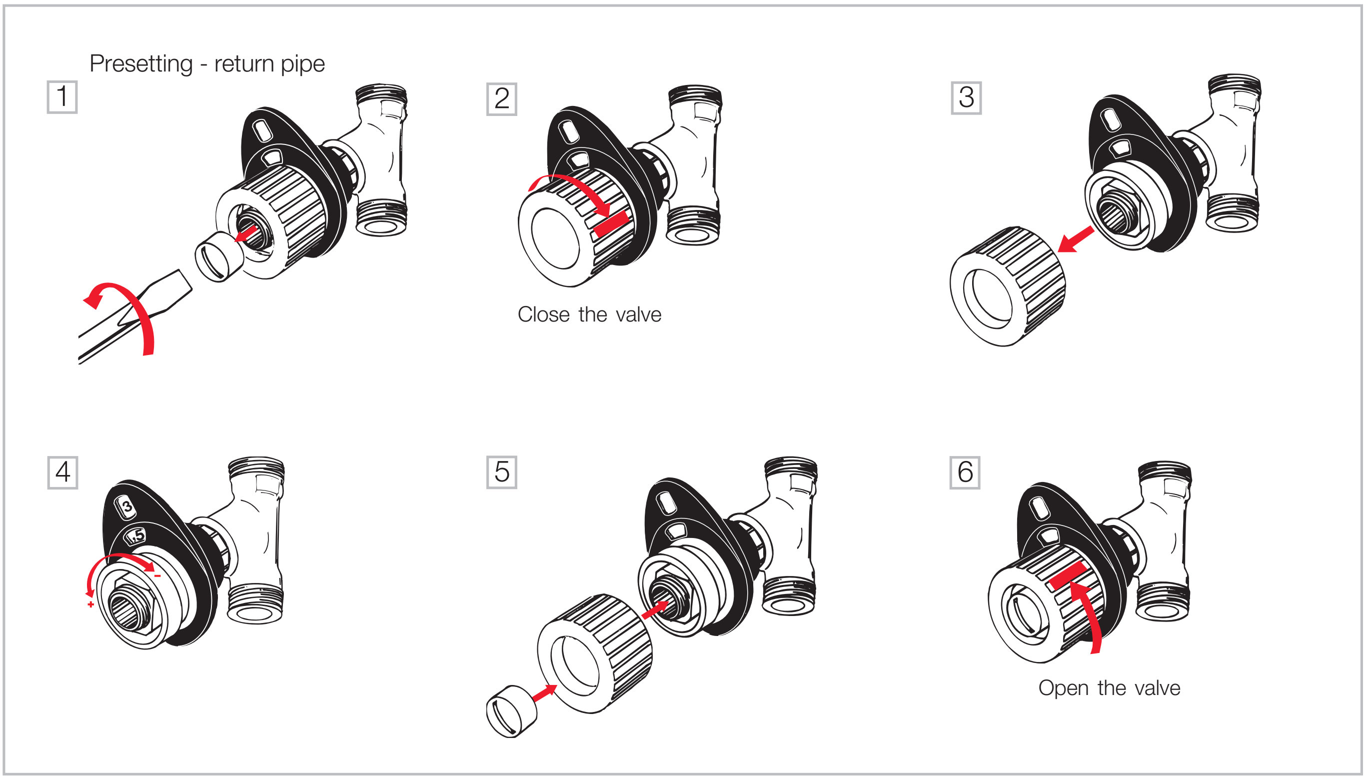
The website also includes software to help you quickly locate the correct valve pre-settings.

http://www.honeywell-valvesizing.com

6.1.1 Operating instructions

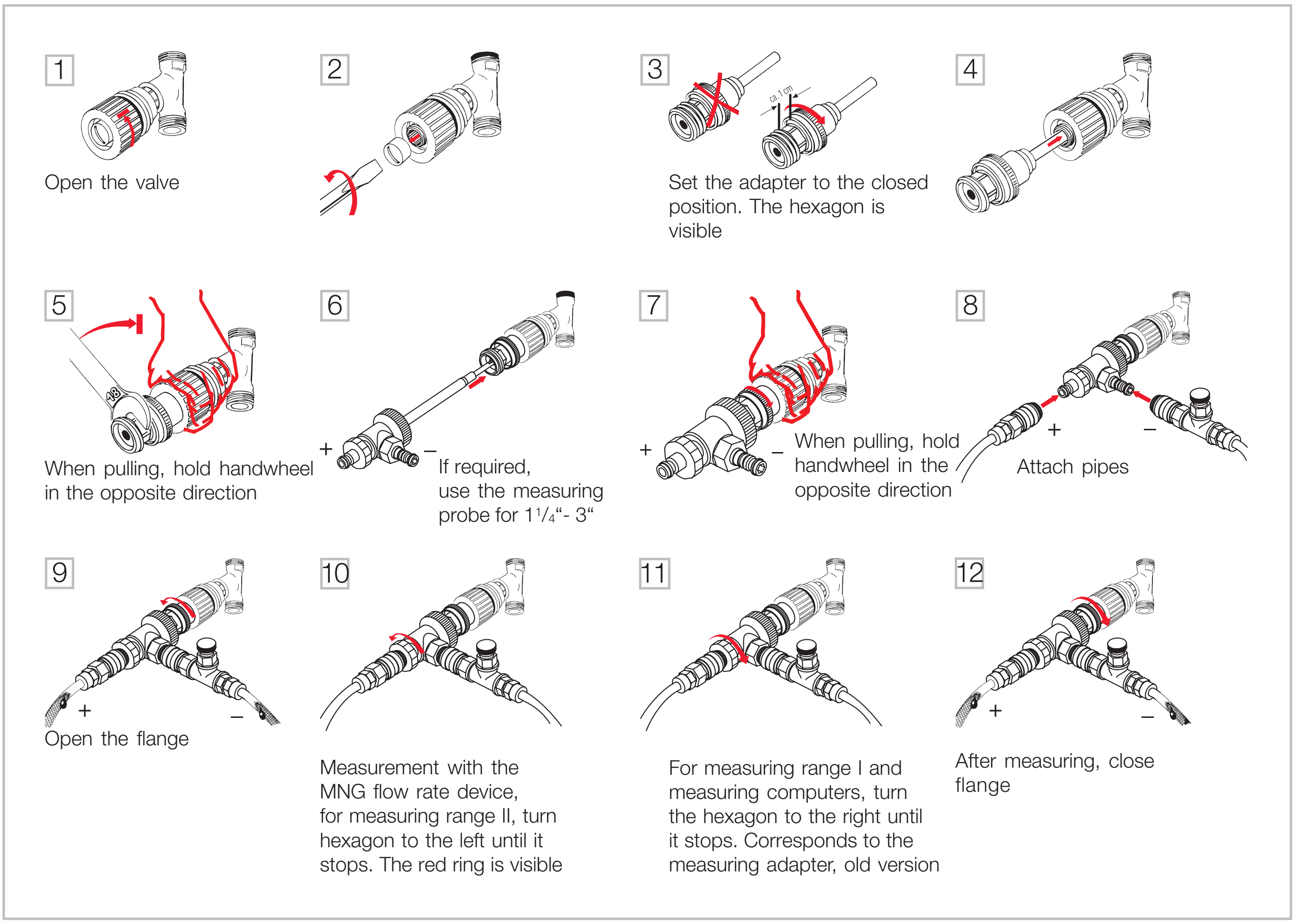
Regulation

Precise presetting through finely-tuned preset values on the Kombi-3-plus, BLUE in the return pipe.



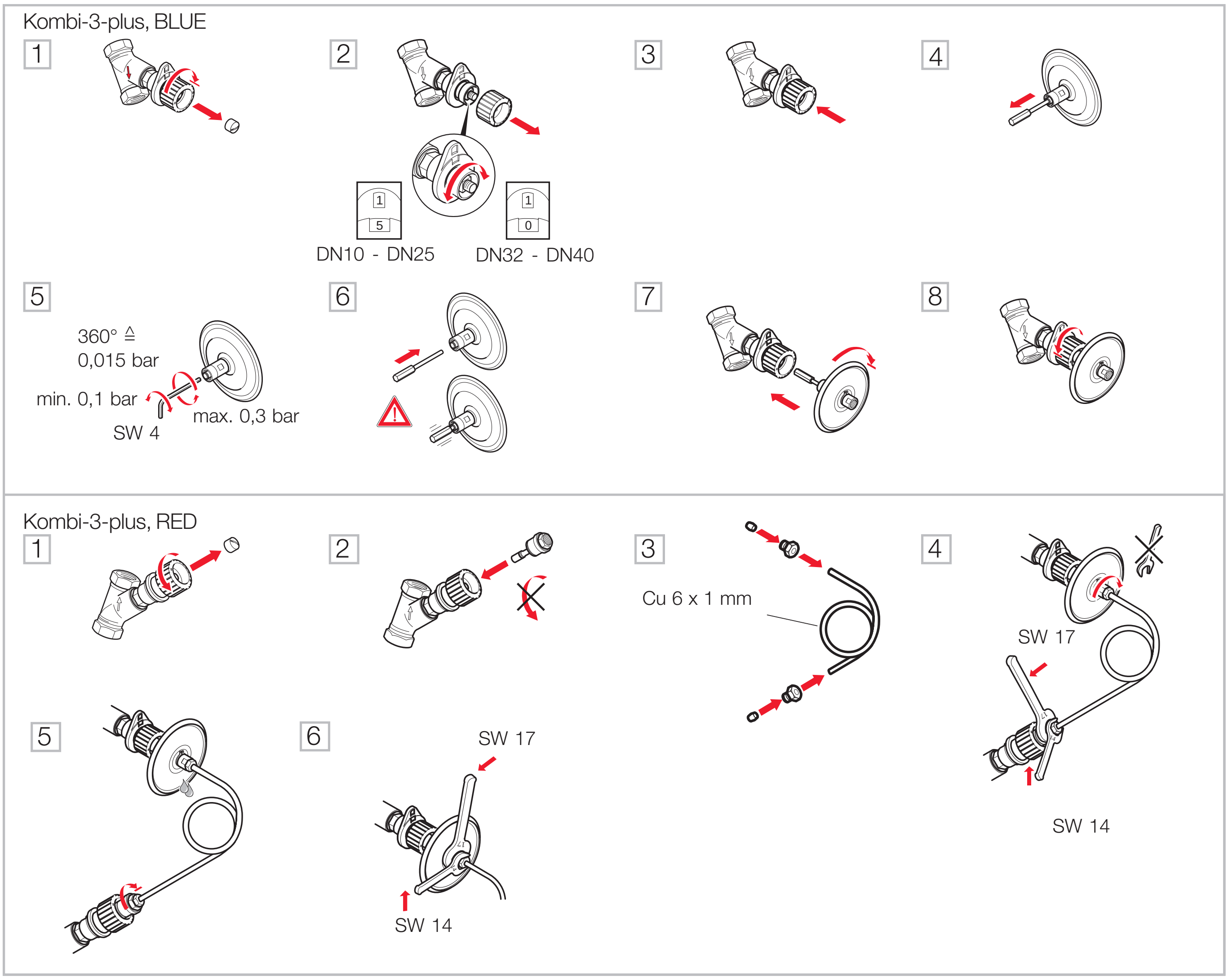
Measuring using the Honeywell measuring system

Measuring at the Kombi-3-plus, RED with defined K_v-value in the supply pipe and simultaneous flow rate setting (accurate to the litre) on the Kombi-3-plus, BLUE in the return pipe.



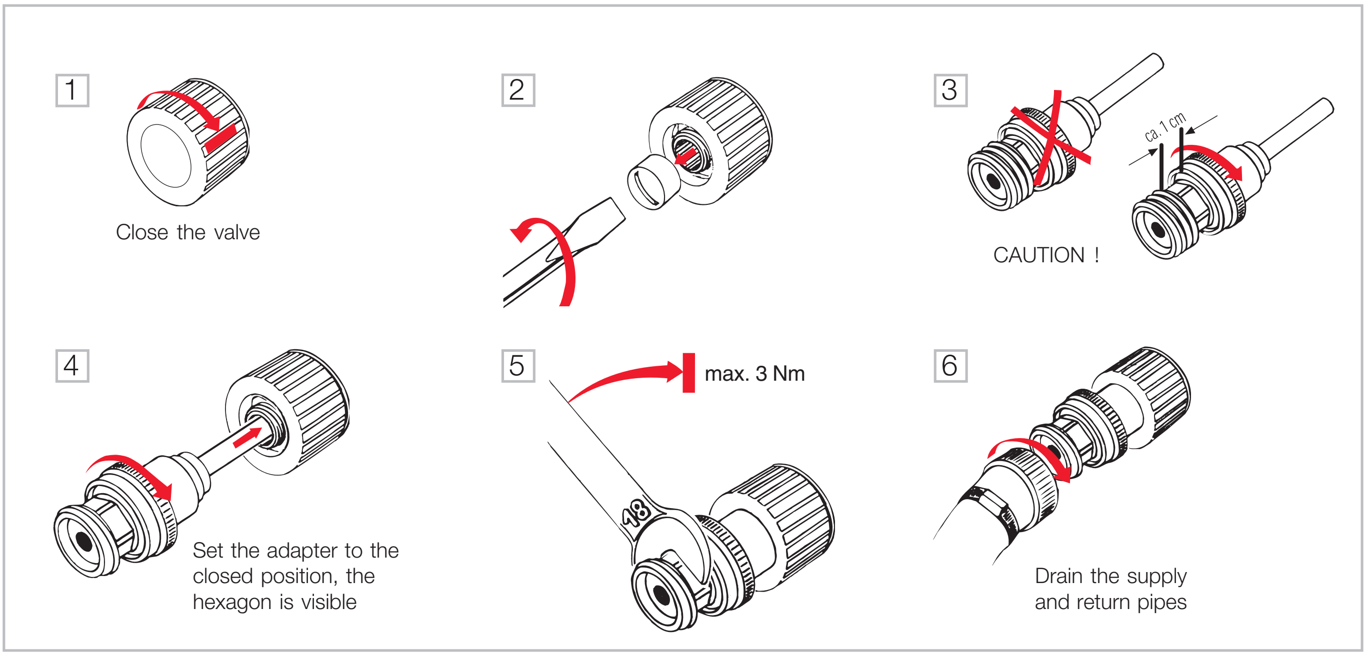
Controlling differential pressure

Simple upgrade of the valve combination Kombi-3-plus, RED and BLUE by adding the diaphragm control unit. Or install the cost-effective complete solution, from the start, consisting of the Stop Valve-3 and Kombi-3-plus, BLUE and Kombi-DU diaphragm unit.



Draining and filling

Simple filling and draining with the draining adapter.



6.2 V5032 Kombi-2-plus

Y-pattern shuttoff valve with red bronce housing, for heating and cooling applications.

- For pres-setting and shutoff of pipelines, risers, heat exchangers etc.
- Measuring function by using auxiliary method (see 2.5.4)
- DN15…DN40 retrofittable with Kombi-Diaphragm Unit for automatic hydronic balancing, including under partial load conditions
- With draining function via adapter
- Compact design

6.3 V5016 Kombi-PC

Differential pressure controller for higher flow rates, e.g. in cooling applications or local district heating stations:

- Automatic hydronic balancing, including under partial load conditions
- Shuttoff and draining / filling function

6.4 V5015 Kombi-FC

Flow controller for applications with constant flow rates, e.g. fan coil units, monotube radiator installations:

- Automatic hydronic balancing
- Shuttoff and draining / filling function
- Direct presetting of the needed flow rate

6.5 V6000 Kombi-F-II

Flanged balancing and shutoff valve for hydronic heating and cooling applications.

- Pre-setting by stroke limitation with digital position display
- With pressure test cocks for differential pressure measurement.

6.6 V9406 verafix-cool

Control valve with measuring function, e.g. for chilled ceilings

- Flow measuring and pre-setting at the same time
- Six-functions: control, pre-setting, shutoff, measuring, draining and filling
- Compact design
- Robust, low noise, flow efficient valve housing

6.7 V5100 Stop Valve-3

Y-pattern shuttoff valve with red bronce housing, for heating and cooling applications.

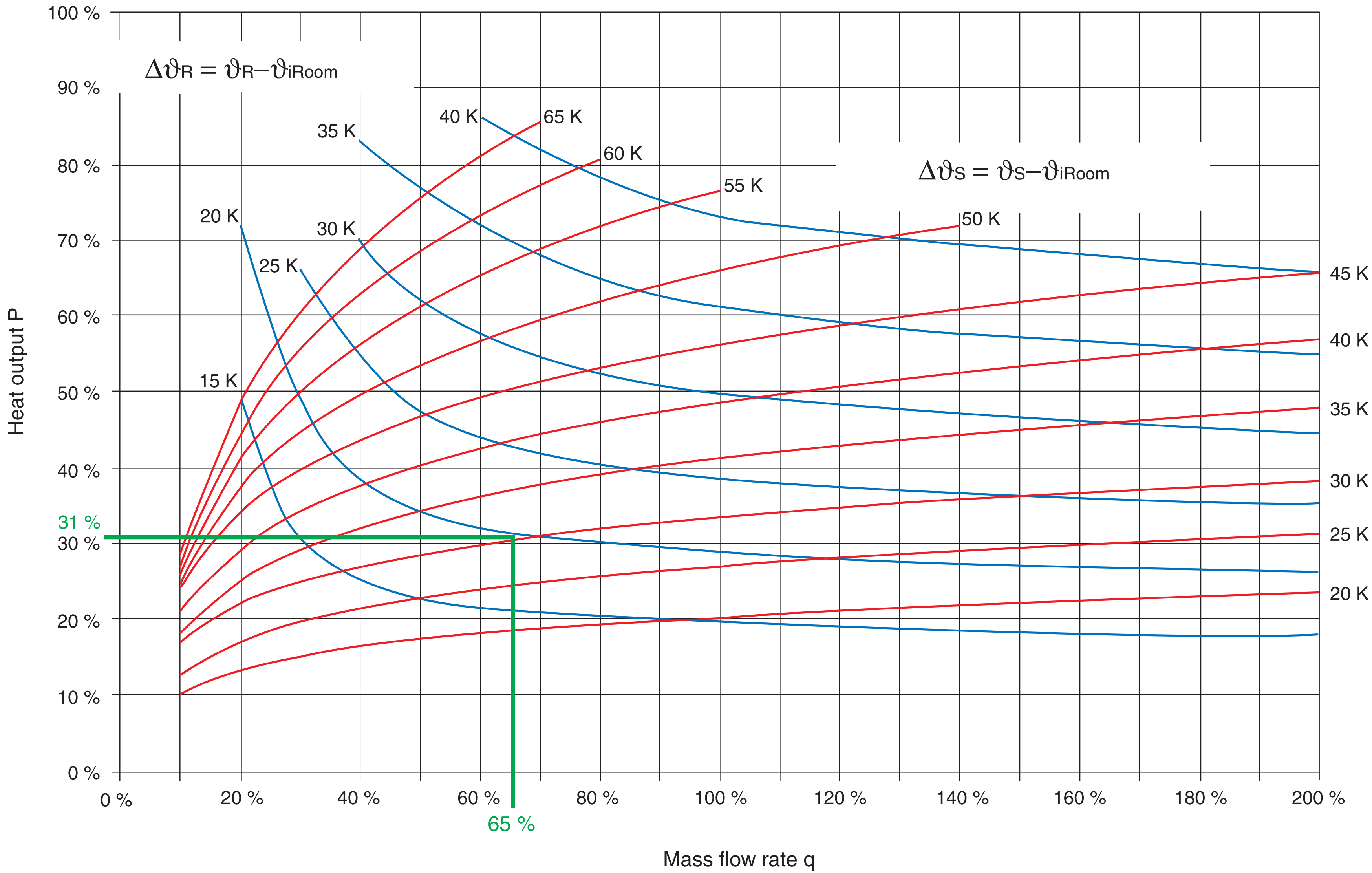
- For shutoff of pipelines, risers, heat exchangers etc.
- With draining function via adapter
- Connection of impulse tube for Kombi-DU diaphragm unit possible
- Compact design

6.8 VB550 Stop-Ball

Brass made shuttoff valve for heating and cooling applications.

- For shutoff of pipelines, heat exchangers etc.
- Quick and economical solution
- Compact design and easy to use

Radiator diagram, n = 1.3 für $\vartheta_S = 90^\circ C$, $\vartheta_R = 70^\circ C$, $\vartheta_{Room} = 20^\circ C$



Example: You want to know the heat output and the required mass flow rate for a radiator that operates at 50/40 at a room temperature of 20°C.

Solution: $\Delta\vartheta_S = \vartheta_R - \vartheta_{Room} = 50°C - 20°C = 30K$

$\Delta\vartheta_S = \vartheta_R - \vartheta_{Room} = 40°C - 20°C = 20K$

The intersection of the lines $\Delta\vartheta_S$ and $\Delta\vartheta_R$ results in a heat output of 31% of the nominal heat output and a mass flow rate of 65% of the nominal mass flow.

Austria
Honeywell Austria Ges.m.b.H.
Handelskai 388
1023 Wien
Phone: (01)72 78 00
Fax: (01)72 78 08
www.honeywell.at

Belgium
Honeywell S.A.
3, avenue du Bourget
1140 Bruxelles
Phone: (02) 7 28 23 86
Fax: (02) 7 28 26 97
www.honeywell.be

Bulgaria
Honeywell EOOD
14, Iskarsko Chaussee
POB 79
1592 Sofia
Phone: +359 (02) 9 79 00 23
Fax: +359 (02) 9 79 00 24

Czech Republic
Honeywell spol., s r.o.
Na Strzi 65/1702
140 00 Praha 4
Phone: (02) 42 44 21 11
Fax: (02) 42 44 22 82
www.honeywell.cz

Denmark
Honeywell A/S
Strandvejen 70
2900 Hellerup
Phone: 39 55 55 55
Fax: 39 55 55 51
www.honeywell.dk

Finland
Honeywell 5A
Ruukintie 8
02330 Espoo
Phone: 02 07 52 20 00
Fax: 02 07 52 26 94
www.honeywell.fi

France
Honeywell EVC
72, Chemin de la Noue
74380 Cranves-Sales
Phone: (04) 50 31 67 30
Fax: (04) 50 31 67 40
www.honeywell-confort.com

Germany
Honeywell GmbH
Vertriebsbüro Haustechnik
Hardhofweg
74821 Mosbach
Phone: (01801) 466300
Fax: (02932) 988346
www.honeywell.de/haustechnik

Greece
Honeywell Ldt.
25 Politechniou Str.
14122 N.Iraklio
Ahtens, Greece
Phone: +30 21 02 84 80 49
Fax: +30 21 02 84 80 55

Hungary
Honeywell Kft.
Petneházy u. 2-4
1139 Budapest
Phone: (01) 4 51 43 00
Fax: (01) 4 51 43 43

Italy
Honeywell S.p.A.
Via Gobetti, 2b
20063 Cernusco Sul Naviglio
Phone: +39 (02) 92 14 61
Fax: +39 (02) 92 14 64 02
www.honeywell.it

Norway
Honeywell A/S
Askerveien 61
Postboks 263
1372 Asker
Phone: 66 76 20 00
Fax: 66 76 20 90
www.honeywell.no

Poland
Honeywell Sp.z.o.o.
ul Domaniewska 41
02-672 Warszawa
Phone: (0 22) 60 60 900
Fax: (0 22) 60 60 901
www.honeywell.com.pl

Portugal
Honeywell S.A.
Edificio Suecia II
Av do Forte Nr. 3 - Piso 3
2795 Carnaxide
Phone: (02) 14 24 50 00
Fax: (02) 14 24 50 99

Romania
Honeywell Romania srl,
Calea Floreasca nr. 48 B,
Sector 1
014462 - Bucharest
Phone: (02 12) 31 64 37
Fax: (02 12) 31 64 39

Russia
ZAO Honeywell
4th floor, 24 Luzhniki,
Moscow 119048, Russia
Phone:+7(095)796 98 00/797 63 01
Fax: +7 (095)796 98 94/796 98 93

Slovak Republic
Honeywell Ltd
Mlynske Nivy 73
PO Box 75
82007 Bratislava 27
Phone: (02) 58 24 74 00
Fax: (02) 58 24 74 15

Spain
Honeywell S.L.
Josefa Valcárcel, 24
28027 Madrid
Phone: 913 136 272
Fax: 913 136 129
www.honeywell.es

Sweden
Honywell AB
Storsätragränd 5
127 86 Skärholmen
Phone: (08) 7 75 55 00
Fax: (08) 7 75 56 00
www.honeywell.se

Switzerland
Honeywell AG
Honeywell-Platz 1
8157 Dielsdorf
Phone: (01) 8 55 24 24
Fax: (01) 8 55 24 25
www.honeywell.ch

Turkey
Honeywell Otomasyon Ve
Kontrol
Sistemleri San. Ve Tic. A.S.
Cayiryolu Sok. (Halyolu)
Ucgen Plaza No: 7, Kat 5-6-7
Icerenkoy 81120 / Istanbul
Phone: (02 16) 5 75 66 20
Fax: (02 16) 5 75 66 37
www.honeywell.com.tr

Ukraine
Honeywell
4, I. Lepse av.
03067 Kiev
Phone: (0 38) 4 42 01 44 74
Fax: (0 38) 4 42 01 44 75

United Kingdom
Honeywell Control Systems Ltd.
Honeywell House
Arlington Business Park
Bracknell, Berkshire RG12 1EB
Phone: (0 13 44) 65 60 00
Fax: (0 13 44) 65 62 40
www.honeywell.com/uk

Middle East Headquarters
Honeywell Middle East
PO Box 6034
14th Floor, Golden Tower
Corniche Road
Sharjah
United Arab Emirates
Phone: +971 65 72 61 43
Fax: +971 65 72 61 65

Distributors & Africa Headquarters
Honeywell S.p.A.
International Operations
Via Gobetti, 2b
20063 Cernusco Sul Naviglio
Phone: +39 (02) 92 14 61
Fax: +39 (02) 92 14 68 88

South Africa
Honeywell Southern Area
(Pty) Ltd.
PO Box 66 74
Midrand 1685
Phone: (01) 18 05 12 01
Fax: (01) 18 05 15 54

Asia Pacific Headquarters
Honeywell (China)
Investment Co., Ltd.
35F., Tower A, City Center,
100 Zun Yi Road,
Shanghai 200051, China
Phone: +86 (21) 52 57 45 68
Fax: +86 (21) 62 37 28 96

America
Honeywell Water Controls
65 Access Road
Warwick
Rhode Island 02886
USA
Phone: (401) 7 38-42 90
Fax: (401) 7 38-41 56
Honeywell Worldwide

ACS Control Products
Honeywell GmbH
Möhnestraße 55
59755 Arnsberg
Phone (49) 2932 988 0
Fax (49) 2932 988 224
www.honeywell.com

EN5H-0105GE23 R0505
May 2005