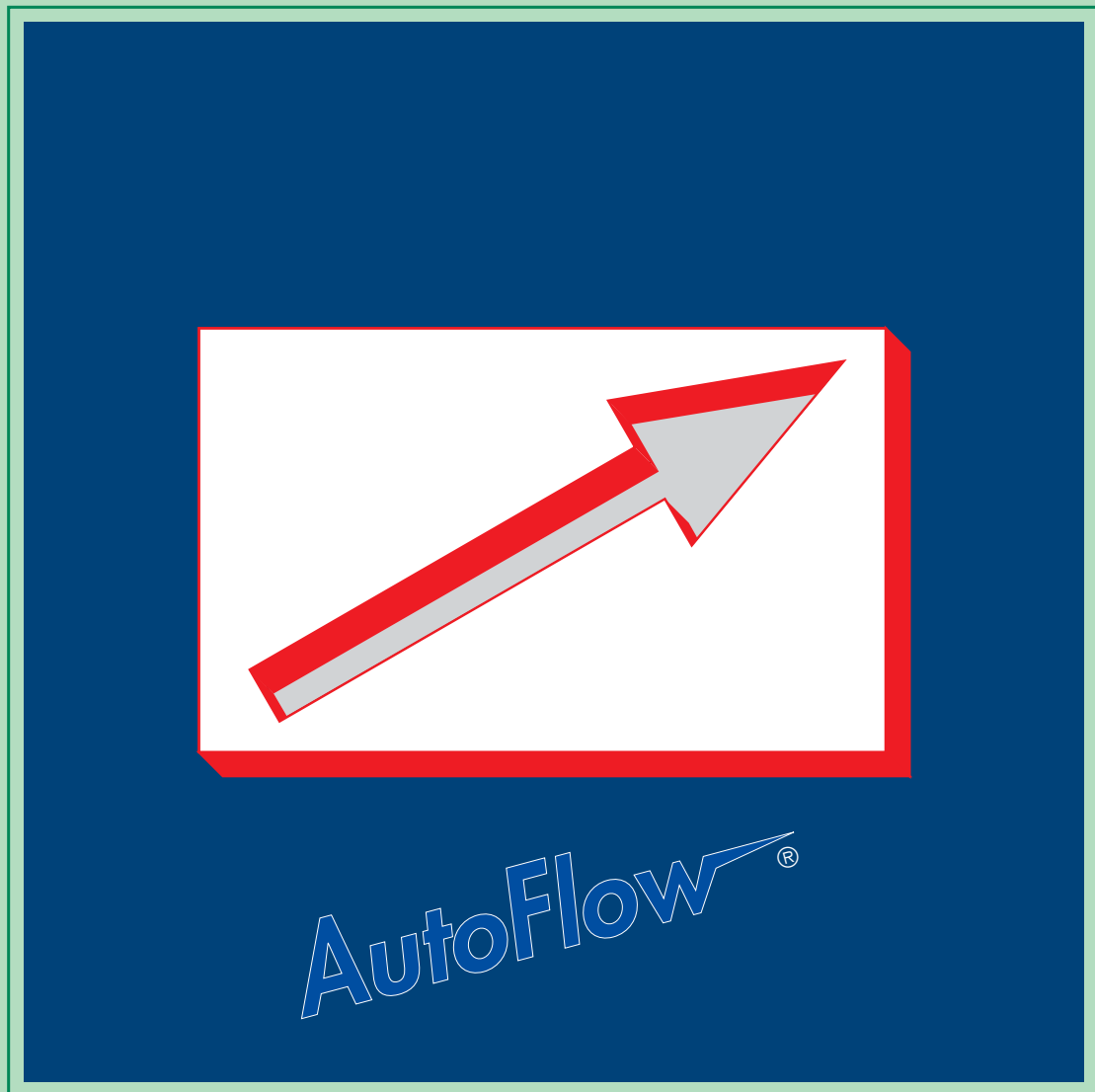


DYNAMIC BALANCING OF HYDRONIC SYSTEMS



THE AUTOMATIC
FLOW RATE REGULATOR

TECHNICAL REPORT

DYNAMIC BALANCING OF HYDRONIC SYSTEMS
THE AUTOMATIC FLOW RATE REGULATOR
AUTOFLOW

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Introduction

This is technical report showing the full advantage of using automatic flow rate regulators Autoflow - in hydraulic systems.

It allows you to carry out a so-called dynamic and continuous balance of the circuits in which they are inserted. The proposed analysis is made to focus on technical aspects which are easy to understand and refer to the specific appendix in the second part of this book for a more in depth study.

In particular, closed circuits of air-conditioning systems are considered; later on however, the results of the analysis are represented using other applications.

References to Bibliography are marked with: [n°].

Balanced circuits

Modern air-conditioning systems must comply with two fundamental requirements: **to guarantee a higher thermal comfort and to limit the consumption of energy.**

In order to do this it is necessary to supply the terminals units of the system with the right quantity of thermal fluid to make sure they can heat, cool and dehumidify according to the design specification.

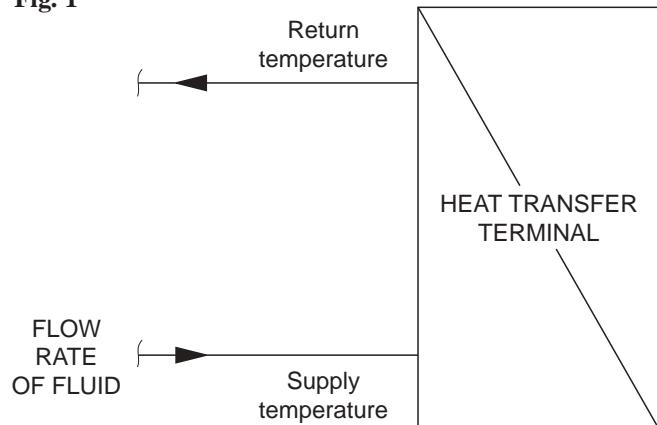
The physical laws on heat transfer are the basis of every energetic consideration; infact, determine that **the heat emission or removal** by a terminal unit **depends on the flow rate** passing through (Fig. 1).

For this reason, it is now possible to confirm that the building up of a system cannot be considered finished until the flow rate supplying each terminal is adjusted.

Only the terminals supplied with a **correct flow rate of fluid** can work under the nominal conditions for which they are dimensioned.

These are known as **balanced hydraulic circuits.**

Fig. 1



When a system is well-balanced:

- it guarantees a correct functioning of the terminals;
- it avoids too high velocities of the fluid, which can cause noise and wear;
- it prevents pumps from working in conditions of low efficiency or overheating;
- it limits the value of differential pressures which act on control valves, by preventing blow-by and working irregularities.

Manual balancing - Constant speed pump

On small to medium systems with a constant flow rate, satisfactory sizing of the pipework will normally be adequate to ensure that the circuits are balanced.

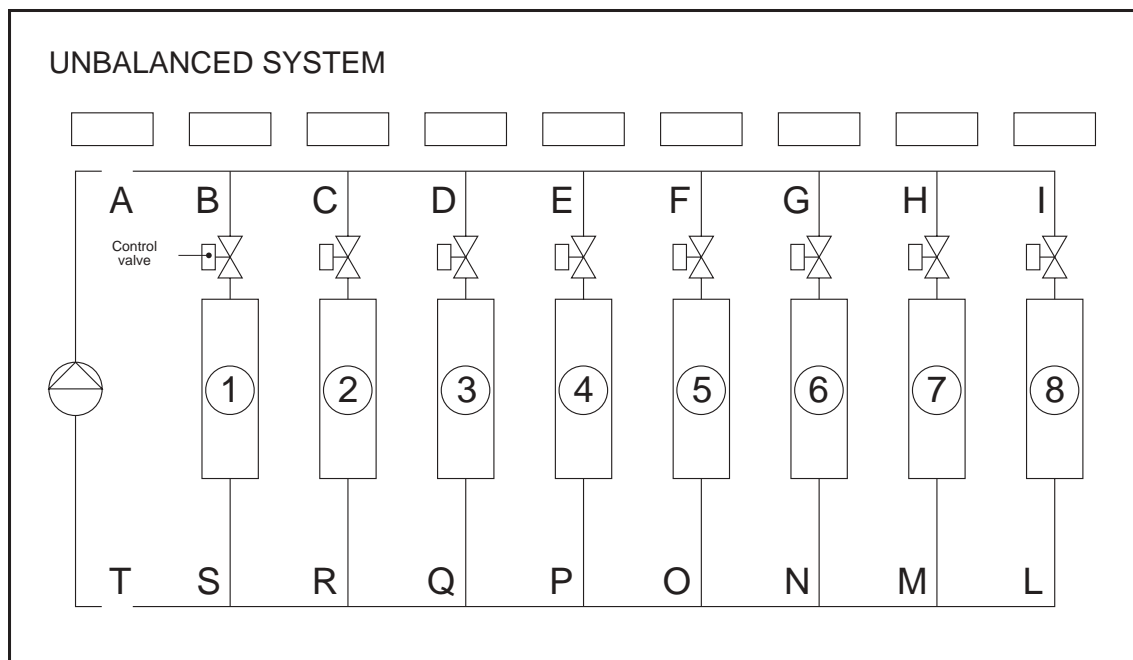
On systems with an extensive network, or with variable flow rates, in order to achieve balanced circuits, it is necessary to include equipment capable of regulating the flow of water.

The behaviour of the manual balancing valves when inserted in an unbalanced circuit will be analysed first. This is the classical method to balance a circuit.

Example

To clarify the problem, a numerical example is considered and used as a concrete support during the various following considerations.

Scheme 1



The example shows the **design results** of a circuit that provides 8 branch circuits from one main line (Scheme 1).

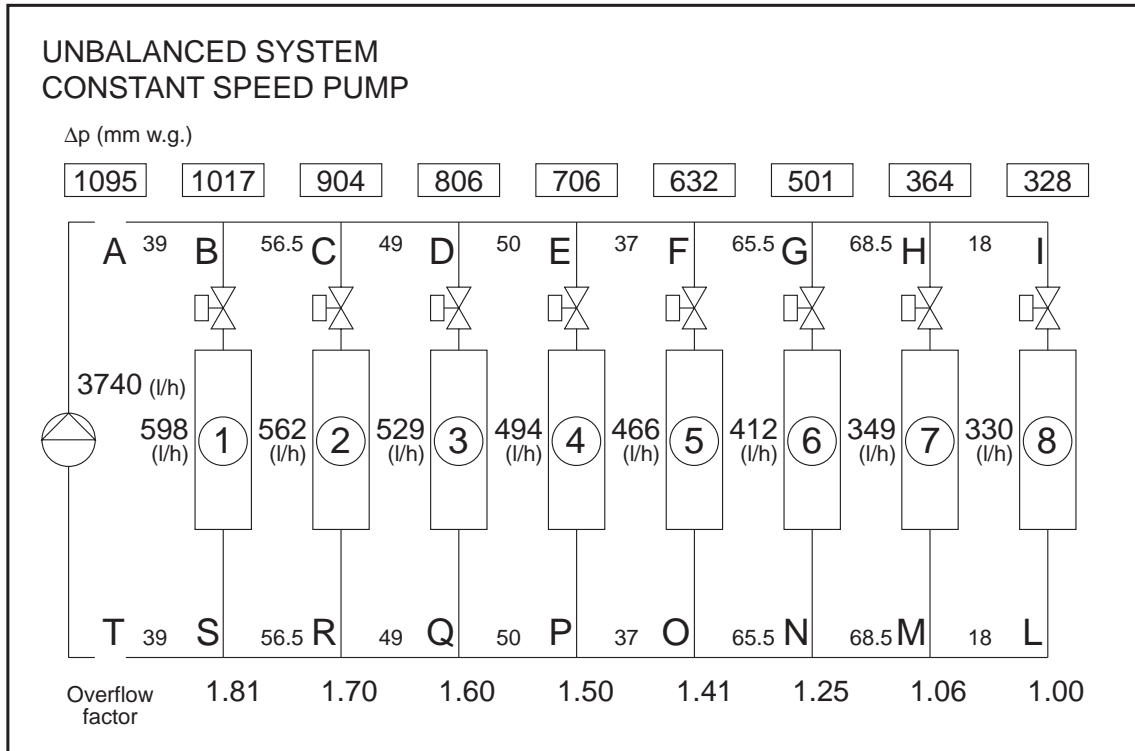
In each circuit a fan coil is inserted which must be supplied with a **nominal flow rate of 330 l/h**.

This causes a pressure drop of 150 mm w.g.

The calculations determining the final operating conditions (Scheme 2) are described in the second volume of Caleffi Handbooks on page 11.

For a logical and ideal continuity it is better to use accurate figures in terms of quantity, which can be found easily.

Scheme 2



The flow rate is often determined from heat transfer capacity and the temperature difference of water supplying the terminals.

If we take the most remote terminal of a circuit and try to supply it the way it should be, the other terminals will be **oversupplied** because of a higher Δp due to the common line pressure drop.

The aim of this example is to show that in order to guarantee the nominal flow rate to the most remote terminal (8), the closest terminals will be oversupplied.

The more remote this terminal, is the higher the overflow to the closer terminals.

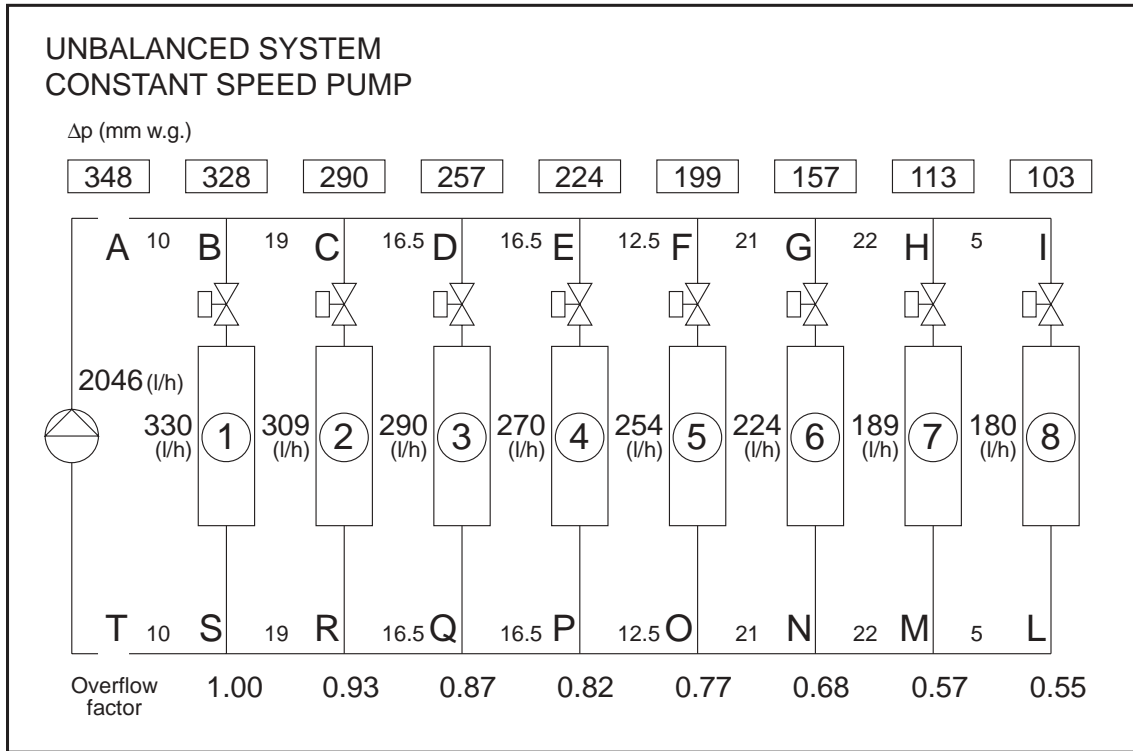
To **guarantee at least the nominal flow rate to every terminal** without using any balancing, an overflow will be created which will reach 81% in the first terminal.

In this example the total flow rate rises from 2640 l/h to 3740 l/h, which is an increase of 42%.

As we have already noticed, this produces a higher energy consumption absorbed by the pump, noise problems caused by the higher flow rates, causes problems in the functioning of the control valves and problems of erosion due to high speed of the fluid.

On the other hand, if the closest terminal (1) is supplied with its designed flow rate by reducing for example, the head and the flow rate of the pump, the other terminals will be **undersupplied** (Scheme 3). This situation should not be created as the necessary heat transfer in the terminals will not be reached.

Scheme 3

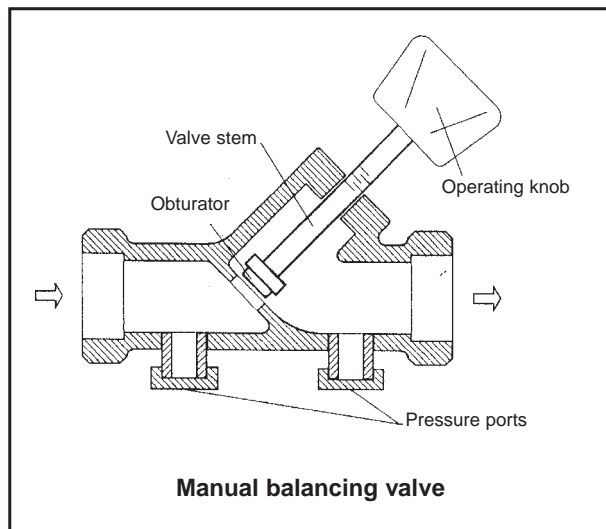


The total flow rate drops from a theoretical value of 2640 l/h to 2046 l/h.

To resolve such problems, balanced circuits must be made introducing balancing valves that are able to guarantee the designed flow rate (Fig. 2).

By turning the knob, the flow rate through the valve can be set reading the Δp measured at the piezometric ports.

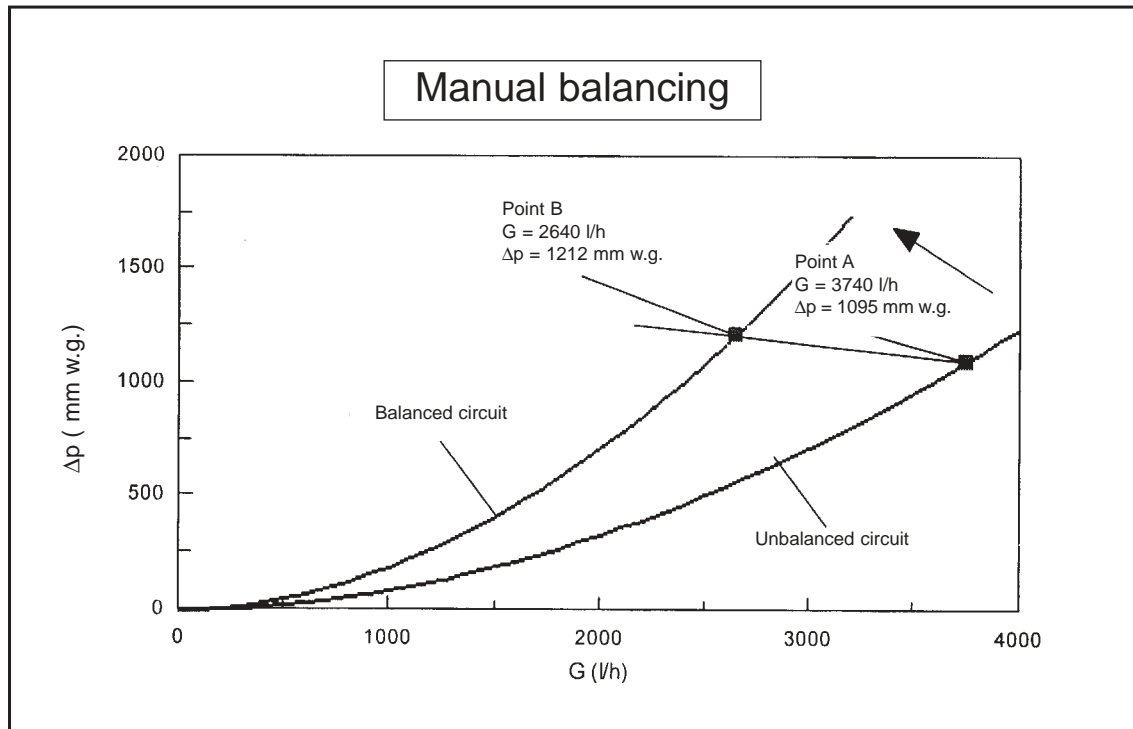
Fig. 2



A graph with the characteristic curves of the two different situations can be plotted (Fig. 3).

Operating points: balanced circuit, unbalanced circuit.

Fig. 3



The value of the characteristic of the balanced circuit is higher than the one of the unbalanced circuit (see appendix). This explains why the curve values of the balanced circuit are higher than the curve values of the unbalanced circuit.

Physically, the insertion of the balancing valves in the circuit and their adjustment made it possible to increase the pressure drops in the single branch circuits and therefore the total pressure drop of all the circuit.

Note: It should be taken into account that the pump characteristic curve can match the designed head and flow rate.

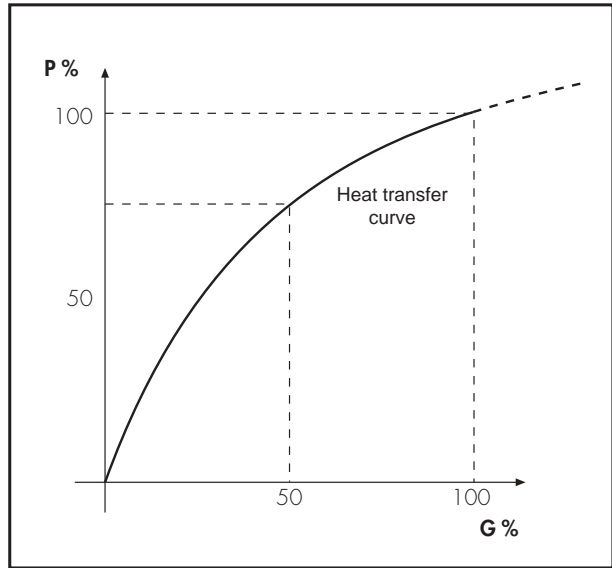
This however is a **static situation**, which in reality rarely occurs. In fact, a **control valve is able to automatically change the flow rate with which each terminal is supplied according to the thermal loads in heating or cooling.**

The example refers to the trend of a typical curve which relates the flow rate and the thermal output of a terminal (Fig.4). [4]

Operating on the basis of a temperature signal, the regulating valve reduces the flow rate automatically and consequently the heat emission of the same terminal; in this way, the flow rate varies, as the thermal load in the system varies.

A new working situation in the circuit with **a new equilibrium point** is now created.

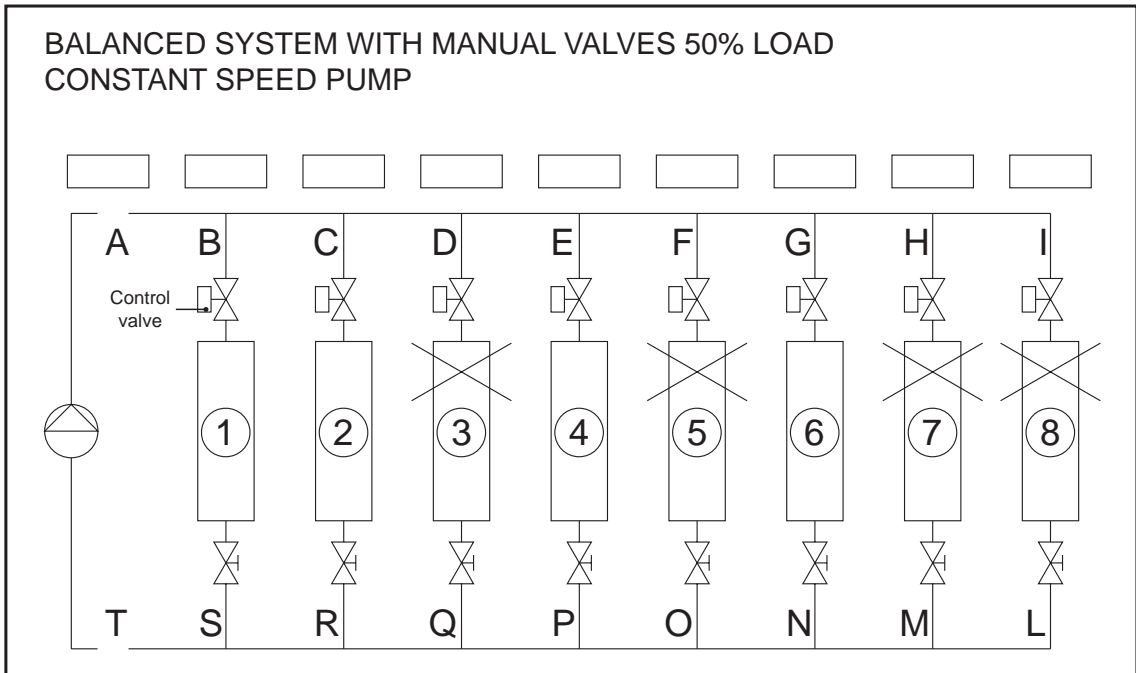
Fig. 4



Partial load

To get a better idea of what happens when **the circuit is partially closed**, some of the terminals of the previous example are assumed closed (Scheme 5).

Scheme 5

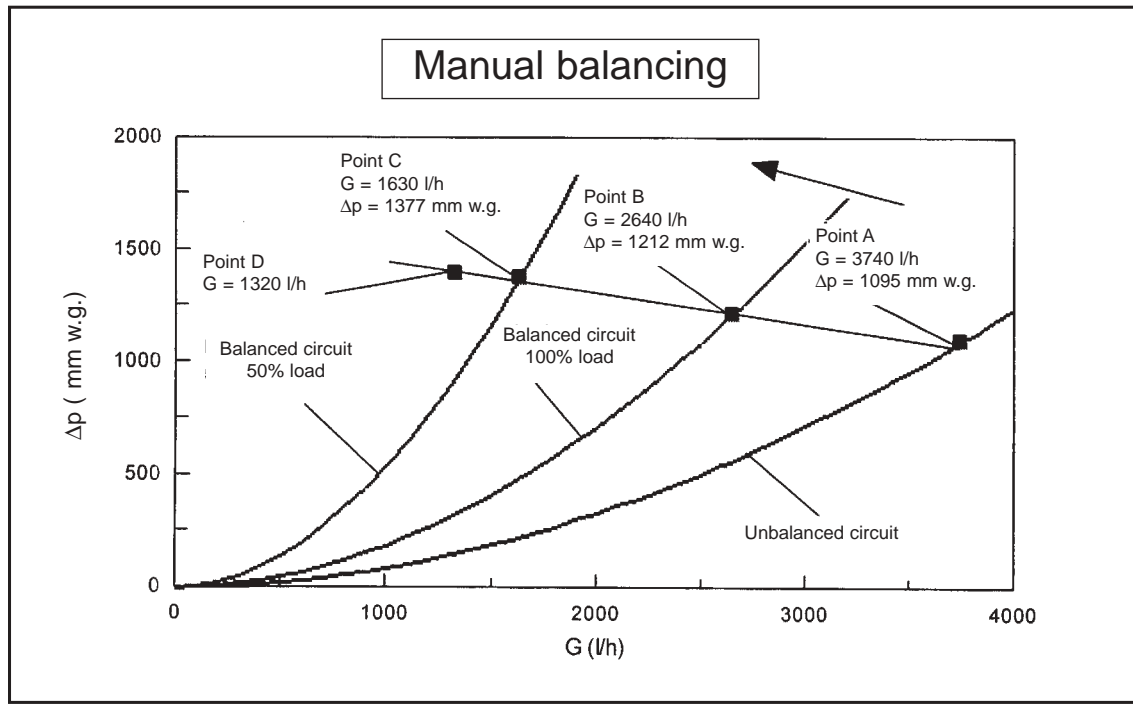


The new operating condition is the common point at the intersection of the pump curve and the circuit curve.

The representative point of the new functioning condition at partial load can be graphically represented (Fig. 5).

Operating points: total load, partial load.

Fig. 5



The intersection point with the curve of the pump is the new equilibrium point C.

$G = 1630$ l/h

$\Delta p = 1377$ mm w.g.

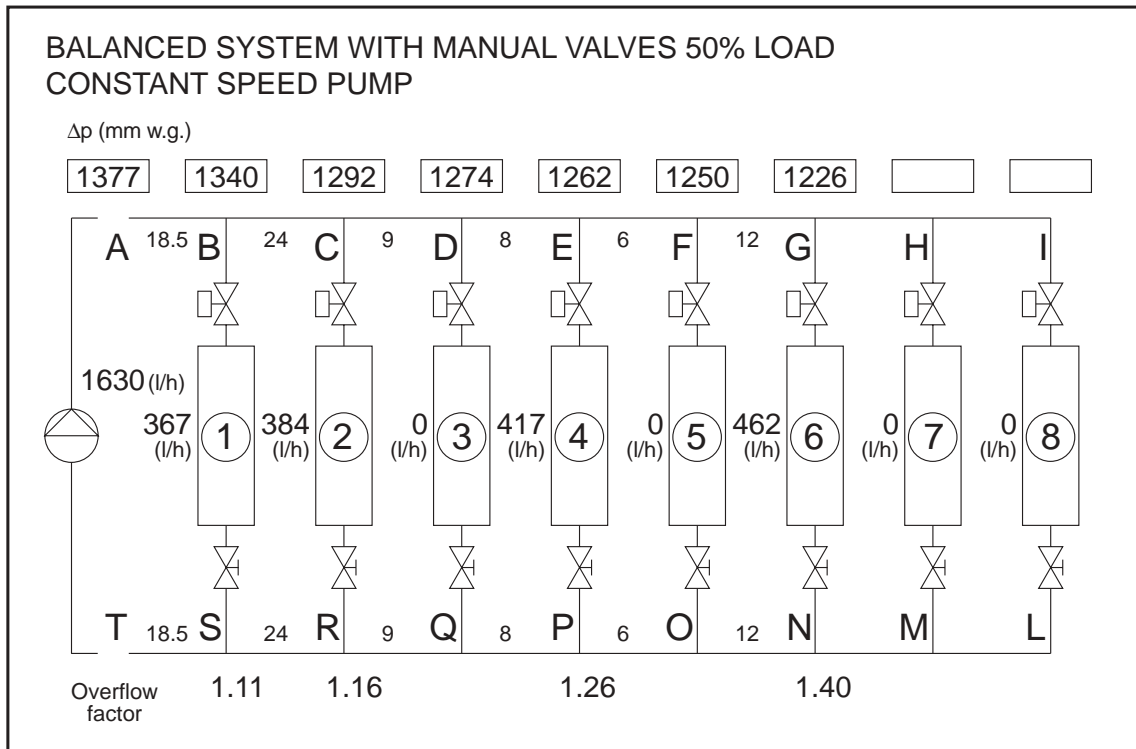
The total flow rate under these new conditions no longer corresponds to the nominal desired flow rate, that is 1630 l/h instead of $4 \cdot 330 = 1320$ l/h (point D).

The variation of the flow rate equals to + **23%**

The closing action of the regulating valves has created overpressures in the circuit and a **new distribution of the flow rates** at every branch circuit (Scheme 6).

The numerical analysis of the overpressures determined by the closing action of the valves is concluded in the appendix of this report.

Scheme 6



Compared to the nominal value, there has been an increase in the flow rate to all the terminals remained open.

The overflow factor is higher for the most remote terminals from the pump. The reason for this behaviour is due to the fact that for the last branch circuits, the balancing valve shows a lower pressure loss than those of the other branch circuits.

In order to have every terminal back to its nominal functioning conditions, it would be necessary to reset the valves, which is an impossible operation because of random closing of the circuits. It must be noticed that **in practice it will be difficult to set the valves**: in fact, as shown before the interdependence law of the hydraulic circuits (see appendix) states that any modification made in one of the circuits creates variation to the flow rates in the other circuits.

Energy consumption

It is well known that the input power of the pump increases as the flow rate increases (see appendix). If we calculate this power in the various cases of the previous example, we find (Tab. 1):

Pump power

Tab. 1

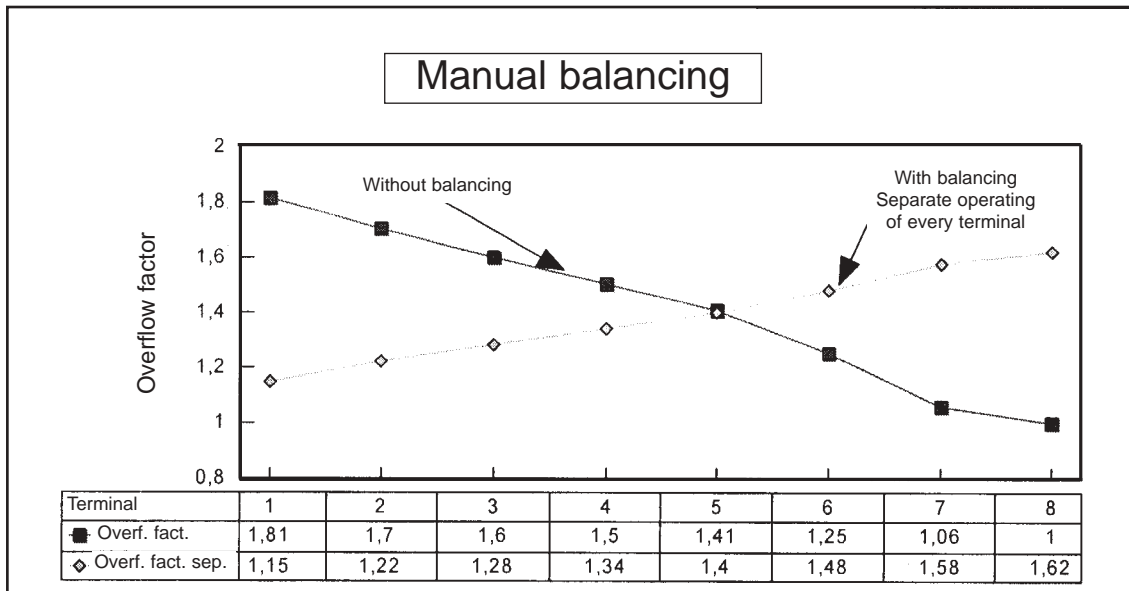
CASE	Load %	G (l/h)	Δp (mm w.g.)	Efficiency	Power %
1	100%	2640	1212	0,8	100%
2	50% (real)	1630	1377	0,7	81%
3	50% (theoretical)	1320	1400	0,6	75%

In case 2, the input power is 6% higher than the power that would have been obtained if the system at partial load had been recalibrated at the theoretical flow rate (case 3). The absolute power values in this case are small because they refer to the example in which the flow rates as well as the heads are small. What can be noted is a **percentage increase of energy consumption** compared to the theoretical condition which gives the limits of this type of balancing.

The figures illustrating the **extreme partial closing conditions** are also reported (Fig. 6). Only having one terminal working at a certain moment, corresponding conditions of oversupplying are found. It can be noticed that when getting nearer to the terminal which was considered the most undersupplied in the beginning, oversupplying conditions are increasing to a value of 60%.

Extreme partial load conditions

Fig. 6



In correspondence, the percentage increase of the input power rises up compared to the theoretical values to a maximum of **10%**.

Therefore, one can conclude that when referring to nominal design conditions, a perfectly balanced system **shows operating limits at partial load**.

Using manual balancing valves **the overflow that appears cannot be controlled**, therefore energy consumption increases compared to the desired conditions.

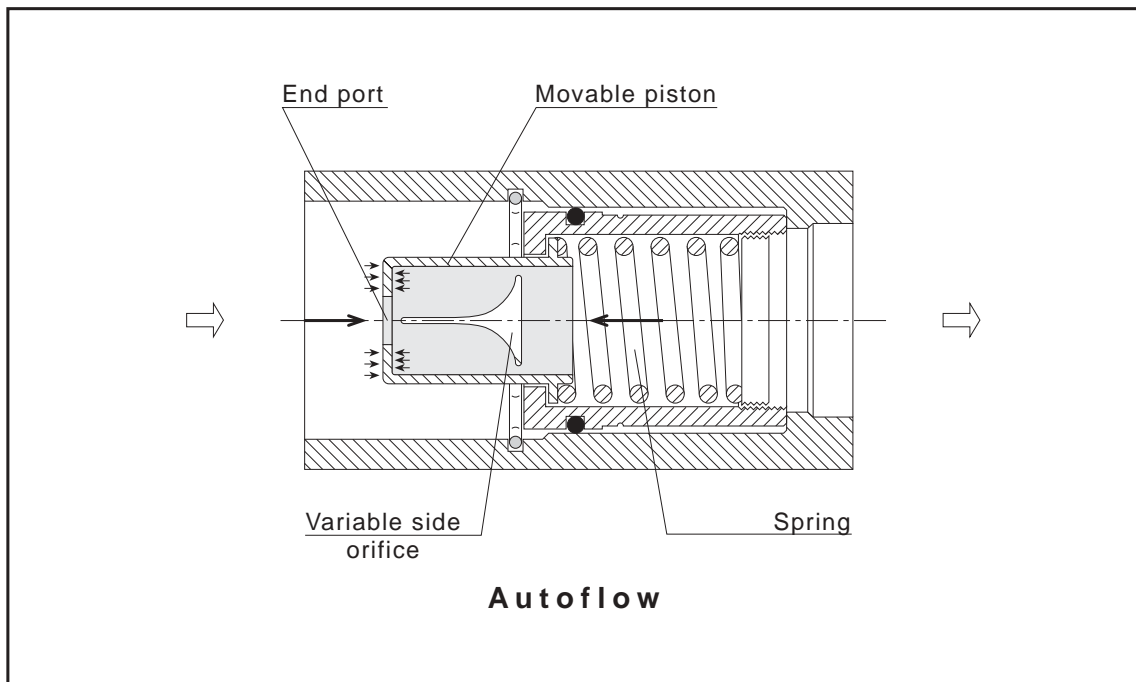
The Autoflow device

The Autoflow device is an automatic flow rate regulator, which function is to maintain a constant flow rate regardless of variations of the differential pressure between inlet and outlet.

The Autoflow regulating element is made of two essential parts (Fig. 7):

1. a piston that has two different types of section for the fluid passage: an end port and a variable side orifice;
2. an helical spring that opposes the thrust of the fluid on the piston.

Fig. 7



The equilibrium is set up between the spring force and the force created by the differential pressure on the surface of the piston head.

Autoflow equilibrium laws

	Relation	Description	Where:
1	$G=K \cdot A \cdot \Delta p^{0.5}$	Flow rate - Δp	G =Flow rate of fluid K =Equilibrium coefficient
2	$F=\Delta p \cdot A_0$	Force - Δp	A =Section for fluid passage F =Force
3	$F=K_s \cdot X$	Spring force - piston position	Δp =Differential pressure A_0 =Piston's head surface K_s =Characteristic of the spring X =Piston position

Autoflow geometry

Working out the previous relations, the geometry of the section necessary for the fluid passage in order to keep the flow rate constant when the differential pressure varies can be defined.

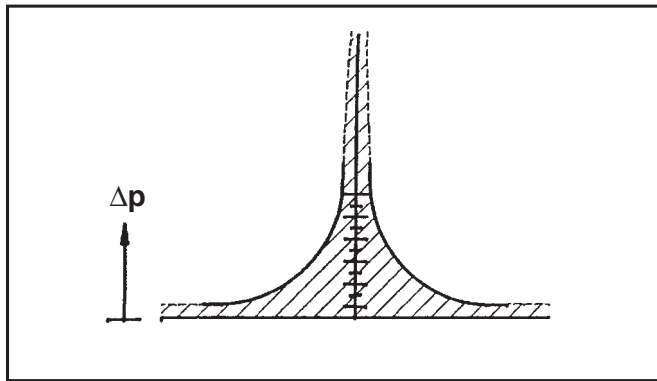
The curve can be plotted graphically, which identifies the shape of the section for the fluid passage versus Δp in order to maintain a constant flow rate (Fig. 8).

$A = f(\Delta p)$ with $G = \text{constant}$

$A = \text{variable cross section for flow passage}$

Shape of the cross section as function Δp

Fig. 8

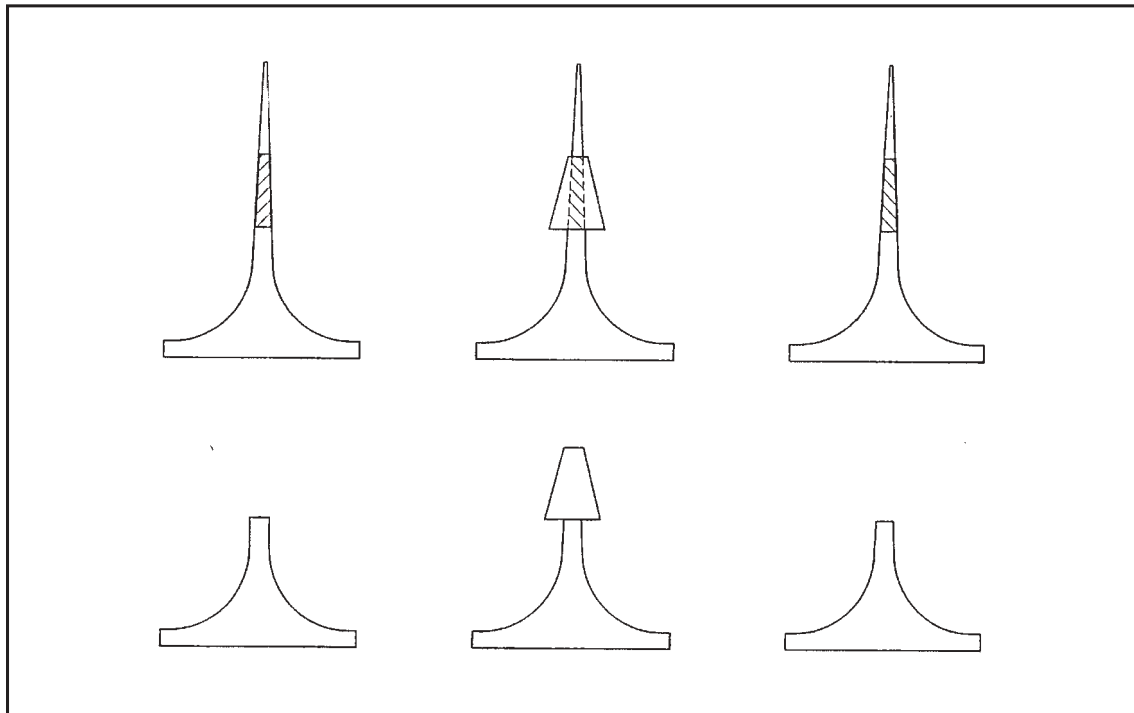


The shaded area represents the fluid passage section versus Δp and therefore the position of the piston inside the body. The more the piston moves inside, the smaller the area gets.

Here following are some examples of typical geometries (Fig. 9) obtained when bidimensionally developing the side surface of the piston. The two small shaded areas are composed to make a bigger central area.

Geometric composition

Fig. 9

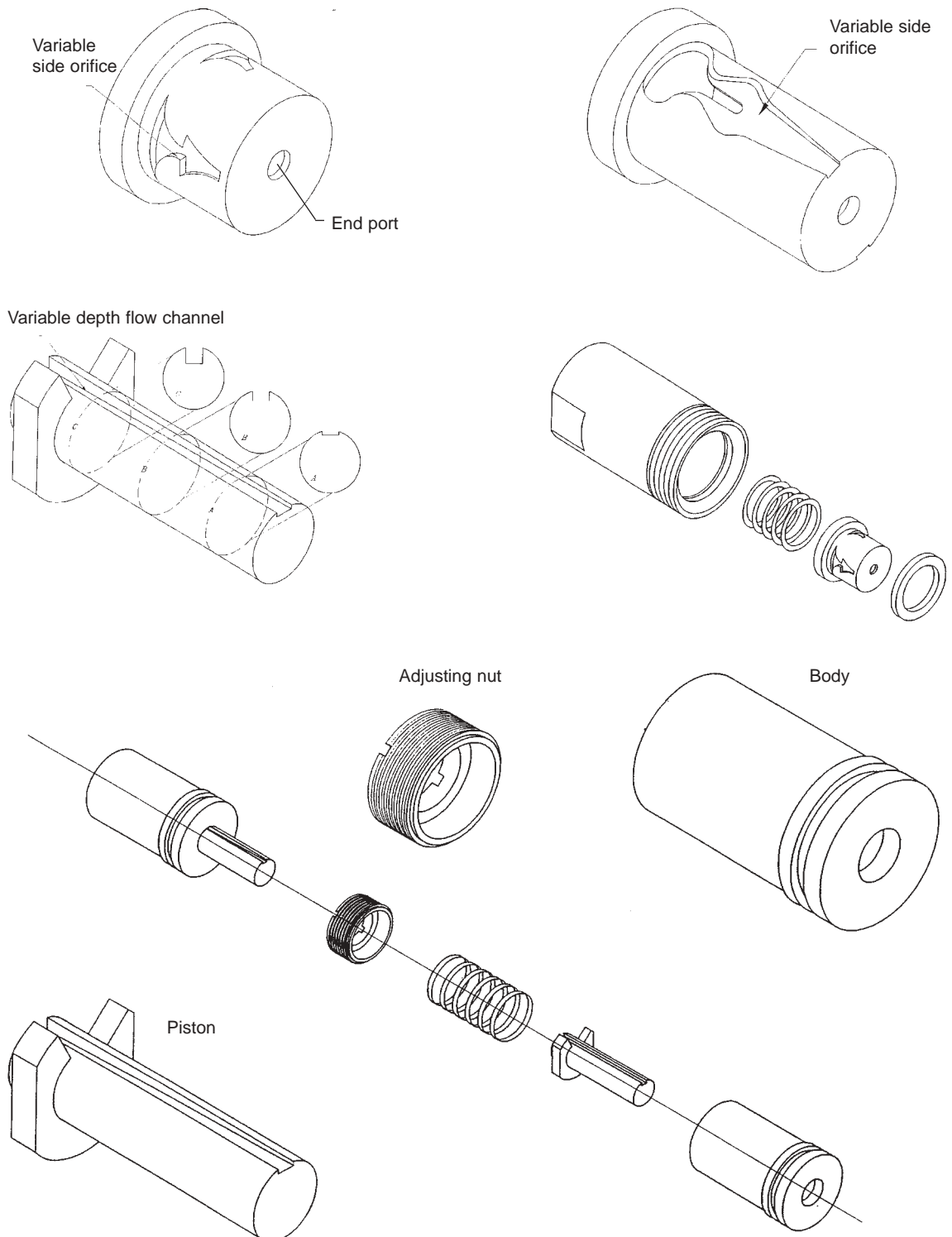


Autoflow sizes

Shapes and sizes of the fluid passage change according to the nominal flow rate and the differential pressure control range. Obviously these depend on the dimensions of the regulator so-called cartridge. As a consequence, Autoflow regulators are manufactured in different shapes and sizes (Fig. 10).

Geometries and shapes of Autoflow regulators

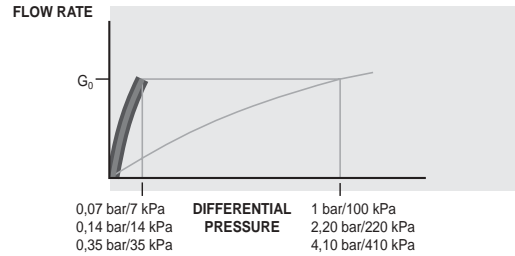
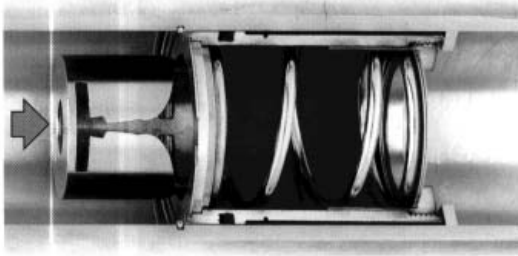
Fig. 10



Autoflow functioning

The functioning of the Autoflow device is easier to understand by referring to the curve Δp -G and a basic scheme, showing the relationship between these quantities.

BELOW THE CONTROL RANGE



In this case, the regulating piston remains in equilibrium without compressing the spring and giving the fluid the maximum free passage.

In practice the piston acts as a fixed orifice and therefore the flow rate passing through the Autoflow only depends on the differential pressure.

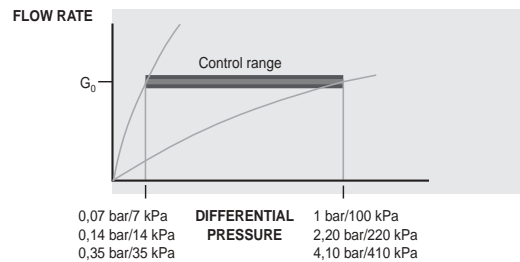
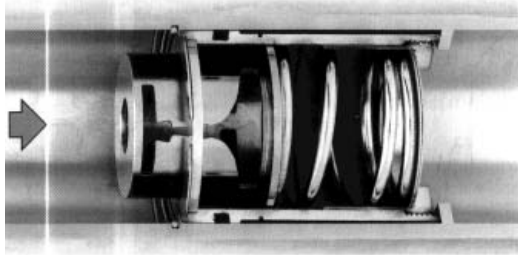
Range Δp 7÷100 kPa: $Kv_{0,01} = 0.378 * G_0$

Range Δp 14÷220 kPa: $Kv_{0,01} = 0.267 * G_0$

Range Δp 35÷410 kPa : $Kv_{0,01} = 0.169 * G_0$

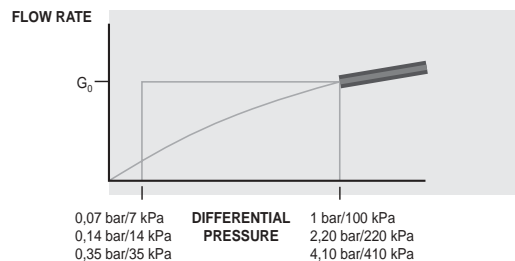
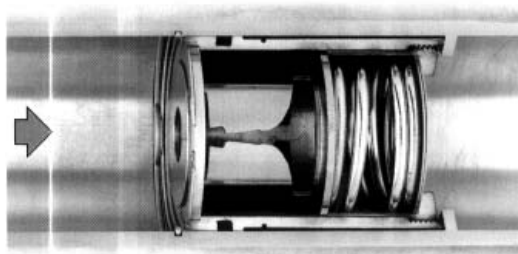
where G_0 = nominal flow rate

WITHIN THE CONTROL RANGE



When the differential pressure is within the control range, the piston compresses the spring and allows the fluid the correct passage, enabling the nominal flow rate to be constant.

ABOVE THE CONTROL RANGE



In this range the piston compresses the spring completely and only leaves the end port for the fluid to pass. Like in the first case, the piston works as a fixed orifice.

The flow rate passing through the Autoflow depends on the differential pressure.

Range Δp 7÷100 kPa: $Kv_{0,01} = 0.1 * G_0$

Range Δp 14÷220 kPa: $Kv_{0,01} = 0.067 * G_0$

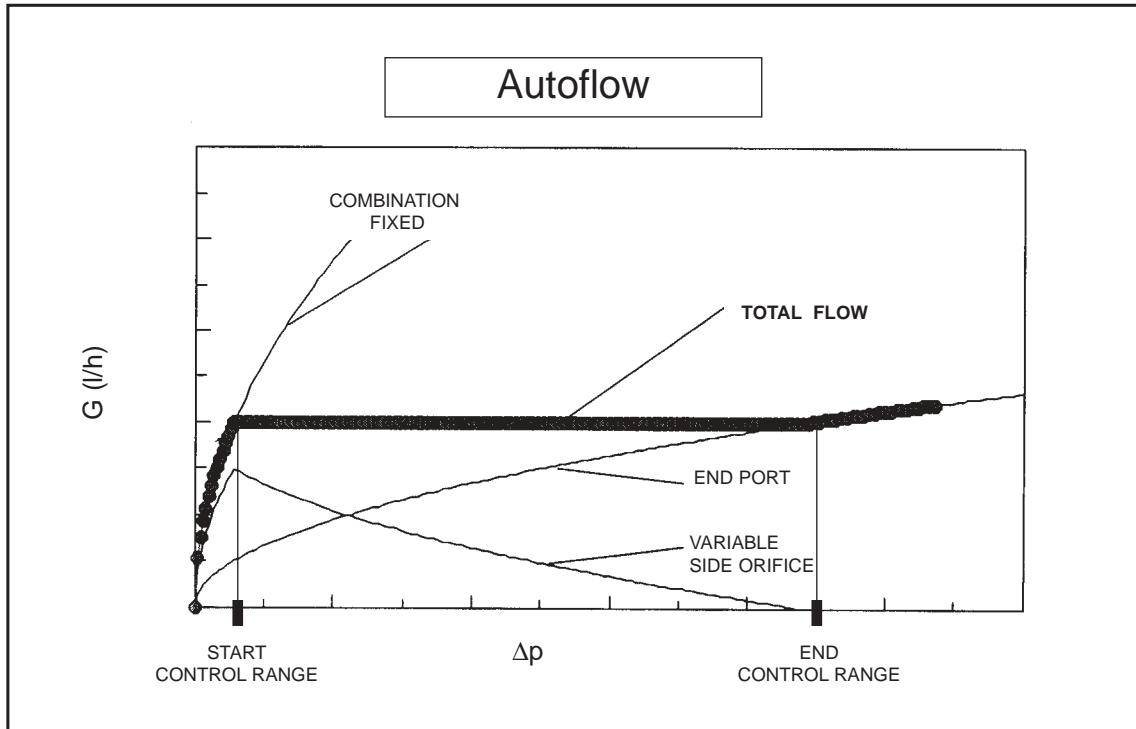
Range Δp 35÷410 kPa: $Kv_{0,01} = 0.049 * G_0$

where G_0 = nominal flow rate

It can be useful to represent the curves of the flow rates that compete at single sections and at their composition in function of the Δp in the same graph (Fig. 11).

Composition of the flow rates with Autoflow

Fig. 11



The total flow rate is the sum of the flow rates passing through the END PORT and VARIABLE side ports.

Remarks

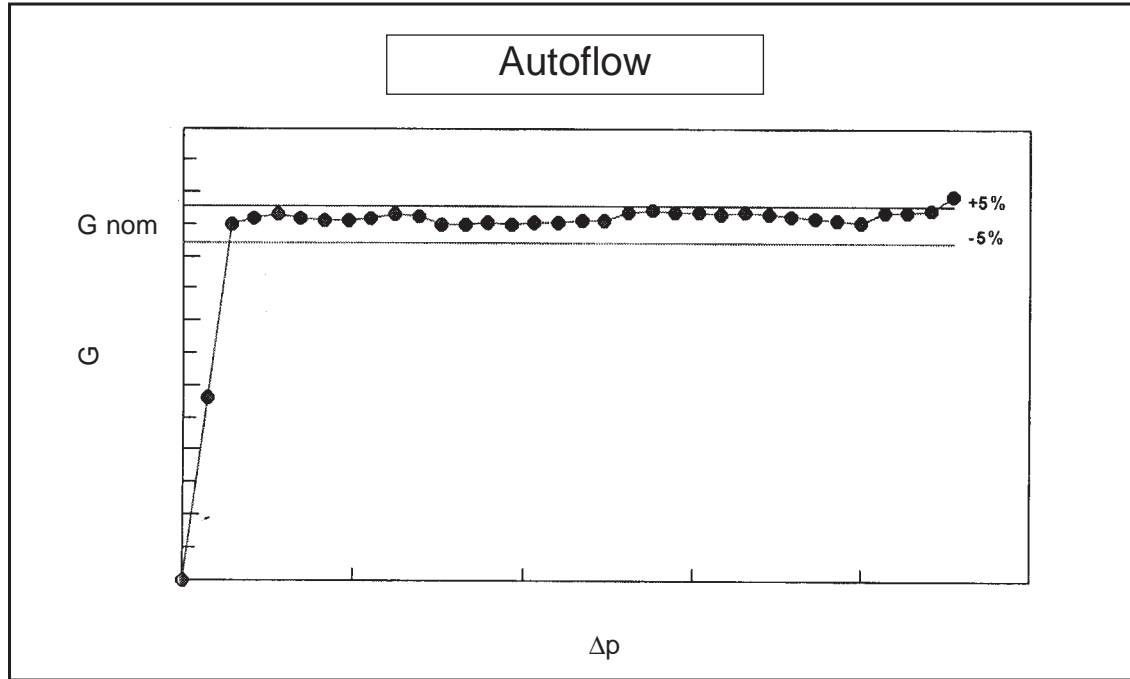
The control range of these devices is particularly large and able to satisfy the requirements of most systems.

Furthermore, different ranges of differential pressure are available (i.e. 14÷220 kPa and 35÷410 kPa), which increases the possibility of control even more.

Flow rate tolerances

The nominal flow rate is kept with a tolerance of $\pm 5\%$.

Fig. 12



This enables the flow rate to be maintained within the suggested limits, to obtain optimum functioning of the terminals and the regulating devices.

These limits depend on the function of the system, that is heating or cooling and on the temperatures for the thermal fluid.

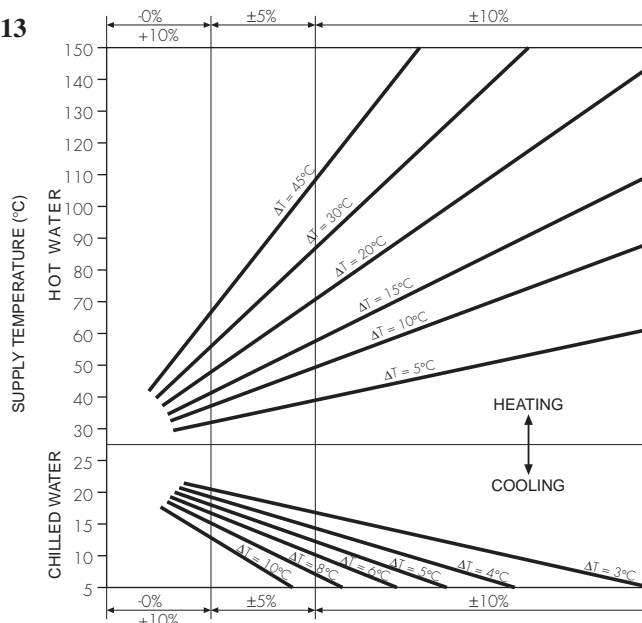
Generally it can be said, that the suggested tolerance for the nominal flow rate in case of heating is $\pm 10\%$, up to $\pm 5\%$ in case of cooling (Fig. 13).

Flow rate tolerances in heating-cooling

The laws regarding the heat transfer and the flow rate, lead to a representative graph.

This graph can be a useful guide for the choice of the flow rate tolerances. [6] [4]

Fig. 13



Autoflow balancing - constant speed pump

As seen before, the traditional balancing method with manual balancing valves only guarantees a constant flow rate under working conditions identical to the conditions they were adjusted, namely for **design conditions**.

If the system is partially loaded, the circuit characteristic changes and a new balancing is required to obtain the design flow rates through every terminal of the circuit.

In fact, in case of partial load there is always an overflow in the terminals.

It is obvious that in practice, this type of operation (new adjustment of the valves) is not possible and that balancing devices shall be used which are different from traditional manual balancing valves.

These devices must be dynamic and able to guarantee the design flow rate in a normal control range, under any working condition.

1. Continuous balancing

In an unbalanced situation the nominal flow rates through various terminals are only guaranteed when the excess of differential pressure on the various elements has been absorbed.

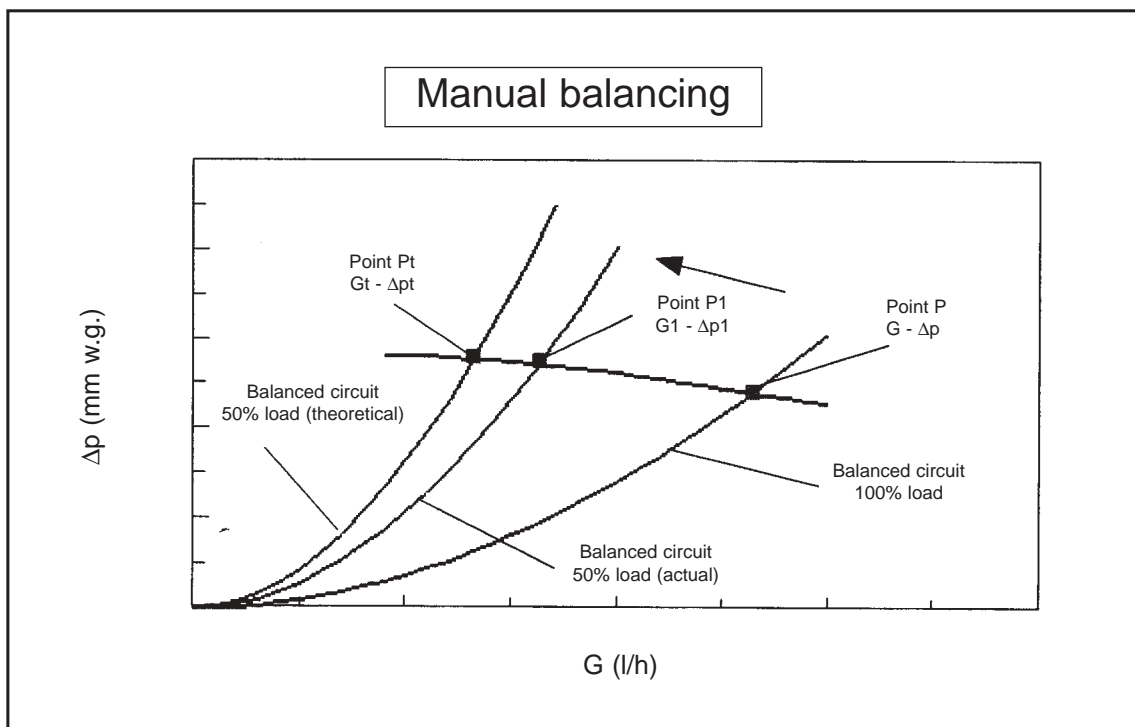
Once these elements are balanced the partial closing of the valves causes overpressures due to an increase of the pressure drops.

The new value determines the new working point of the system P1.

At this working point, there is a correspondent flow rate G1 which is higher than the desired theoretical one at partial load, theoretical Gt (Fig. 14).

Working points of the system: total load and partial load.

Fig. 14



To guarantee the desired flow rates, the circuit should be readjusted in order to absorb overpressures and reset the total flow rate at a theoretical G_t .

Thanks to its construction, Autoflow can absorb these overpressures while maintaining a constant flow rate. Therefore it can really be called a **flow rate regulator of a dynamic and continuous type**.

2. Planning with Autoflow

During the engineering of the project it should be taken into account, that the most remote branch circuit, provided with Autoflow is always supplied with the designed flow rate.

If it is not provided with Autoflow, it would be oversupplied in case of partial closing of the others circuits.

This is the circuit in which on the Autoflow there is a differential pressure lower than the others (the rules to calculate the pressure drops are always the same).

The pump has to be chosen in order to guarantee a minimum differential pressure on the Autoflow in this circuit, to enable the device to start its regulating action, maintaining a constant flow rate.

The following closings at partial load cause the well-known overpressures which in this case are absorbed by the device itself.

In effect, to select the pump, the pump head has to be calculated by summing up the pressure drops of the most remote circuit and the minimum working differential pressure of the Autoflow device.

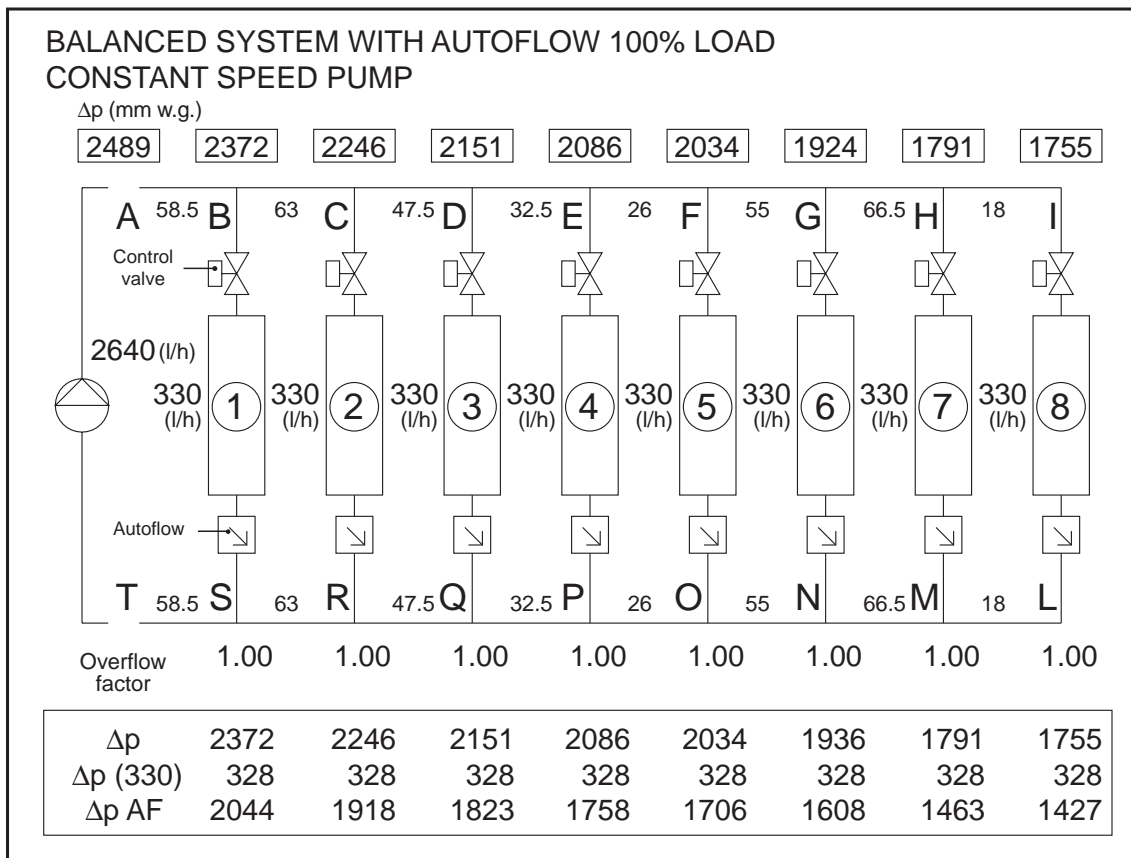
Example

Considering the previous example, Autoflow devices are inserted instead of manual balancing valves.

For analogy, only the relative data of the final working conditions are reported (Scheme 7).

The calculations executed in this case are also reported in the second volume of the Caleffi Handbooks on page 48, easier calculations.

Scheme 7



By means of the Autoflow, a perfectly balanced circuit is created.

Remarks

The Autoflow Δp of the last branch circuit is = 1427 mm w.g., so the Autoflow is at the minimum of its control range.

The closer to the pump the branch circuits are, the Δp on the Autoflow of the various branch circuits increases.

Consequently the spring of the Autoflow is more compressed.

$G = 2640 \text{ l/h}$

$\Delta p = 2489 \text{ mm w.g.}$

In this case, the total pressure difference of the circuit increases applying of the Autoflow.

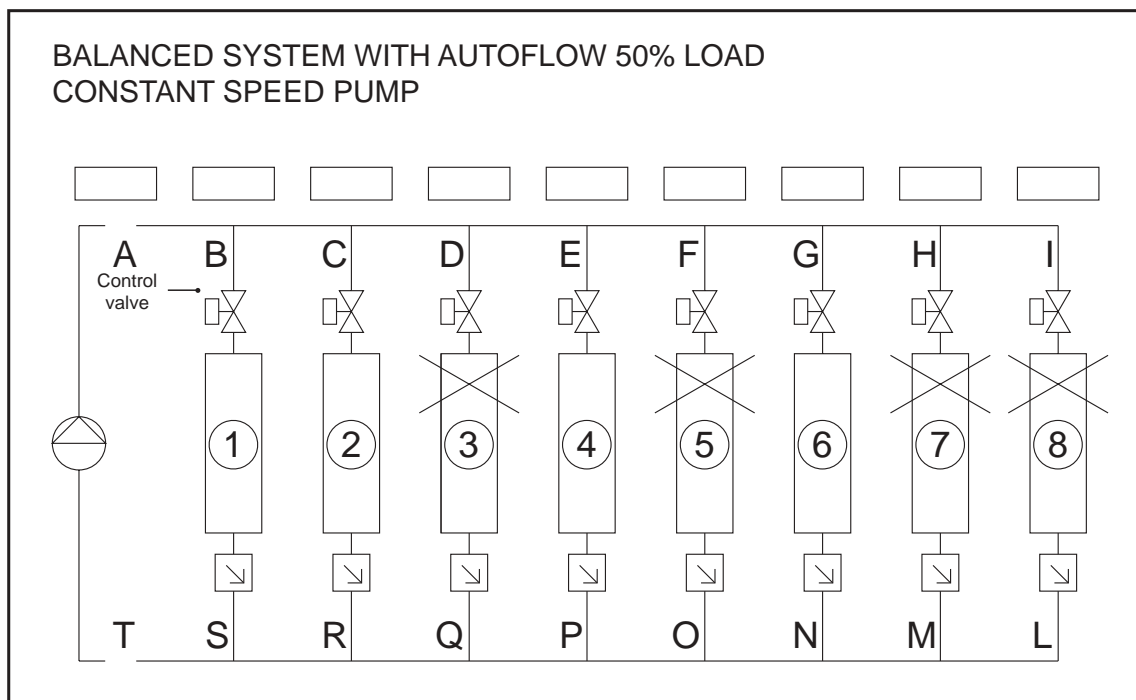
This increase however has to be evaluated on the global installation.

Partial load

Scheme 8 explains what will happen when the system works at partial load.

The control valves are handled simulating the closing of the same terminals of the previous example.

Scheme 8

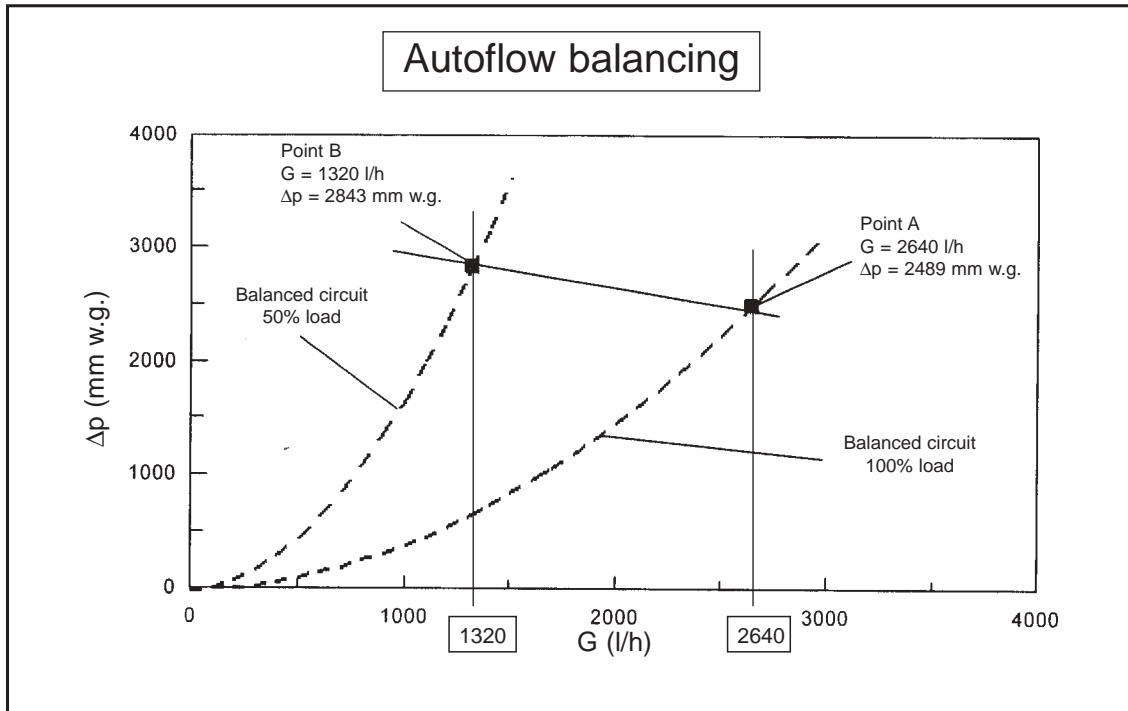


In these new conditions, the graph showing the various operating points is reported (Fig. 15).

Note: Because of different nominal working points Δp -G, the characteristic curve of the pump in this example is different if compared to the pump curve with manual balancing valves.

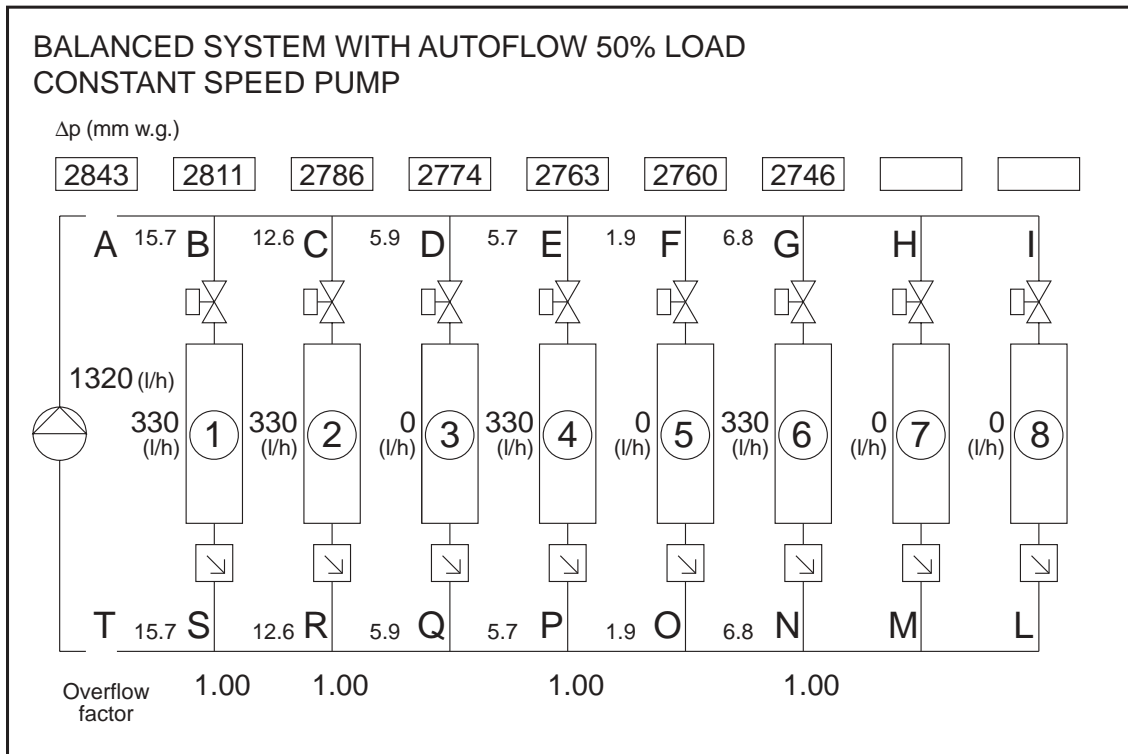
Working points of the circuit with Autoflow: total load & partial load

Fig. 15



Here following are the new functioning conditions of the dynamic balanced circuit with Autoflow (Scheme 9).

Scheme 9



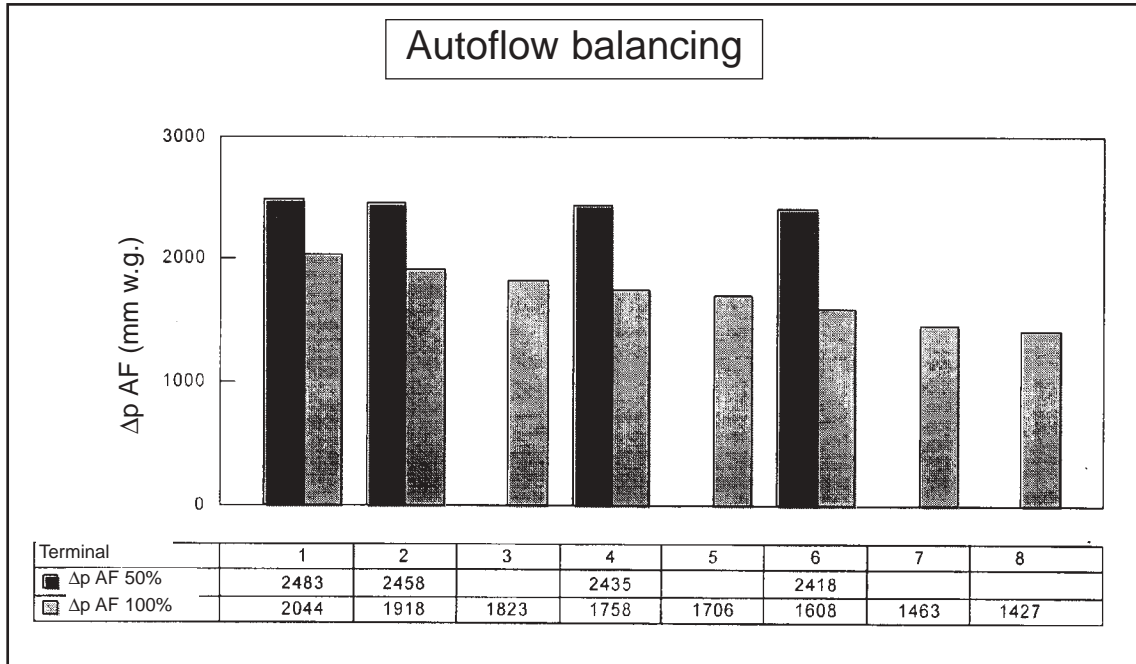
The flow rate remains constant at its nominal value.

The Autoflow absorbs the overpressure due to the control valves closing action (Fig. 16).

In the following graph the trend of these overpressures is reported in order to make a summary of the physical phenomenon.

Δp absorbed by the Autoflow

Fig. 16



Remarks:

At partial load, the increase of absorbed Δp by Autoflow (AF) is proportionally higher for circuit 6 than for circuit 1. Therefore:

- the less circuits connected, the higher the differential pressure that has to be absorbed by Autoflow;
- the more circuits connected, the lower the maximum differential pressure that has to be absorbed by Autoflow;

$$D = \Delta p_{\max} - \Delta p_{\min}$$

$$\text{with 50\% load: } D = 2483 - 2418 = 65 \text{ mm w.g.}$$

$$\text{with 100\% load: } D = 2044 - 1427 = 617 \text{ mm w.g.}$$

- the more circuits connected, the higher the D difference of the differential pressure among Autoflow (617 > 65).

Manual balancing - variable speed pump

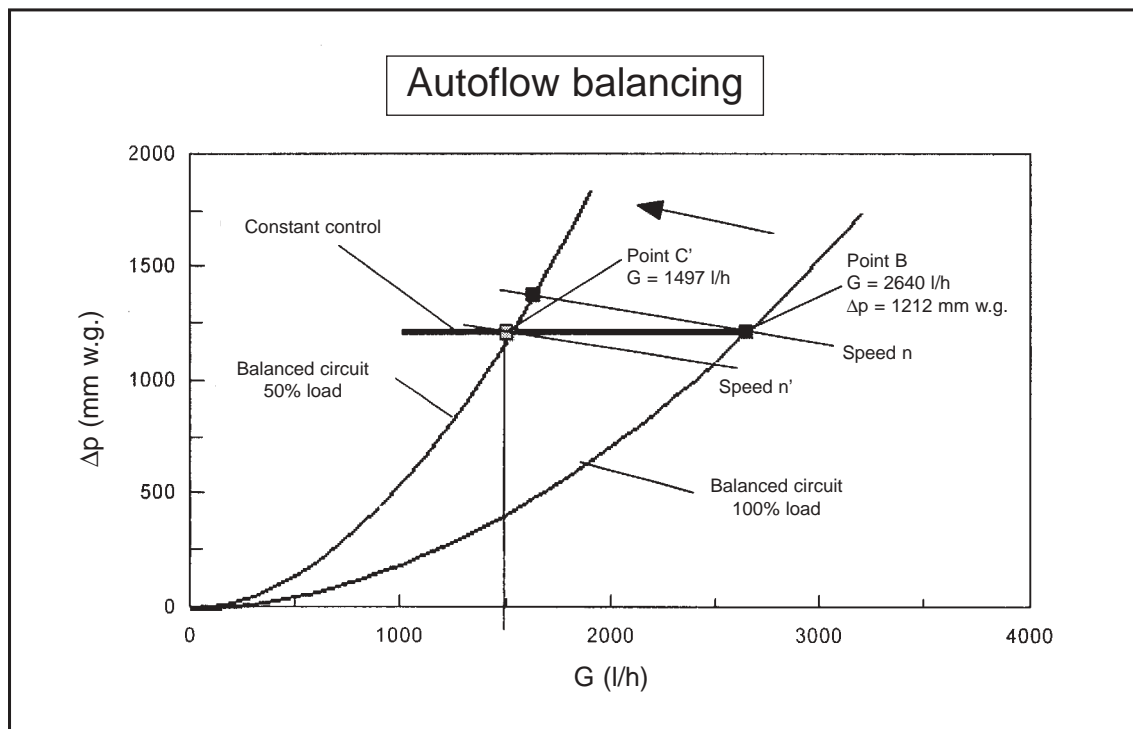
If in a manually balanced circuit, a variable speed pump which is able to modify its rotation speed is applied to satisfy the new requirements of partial load of the system, the following situations will occur.

Case 1 - Constant control

The pump reduces its speed and therefore the flow rate, maintaining a **constant Δp** . This method is called **constant control** (Fig. 17).

Variable speed pump with constant control

Fig. 17



The new speed of the pump is n' .

In this case the new working point is the point C' with $\Delta p = 1212$ mm w.g. This is the same value of point B that corresponds to the balanced circuit working at 100% load.

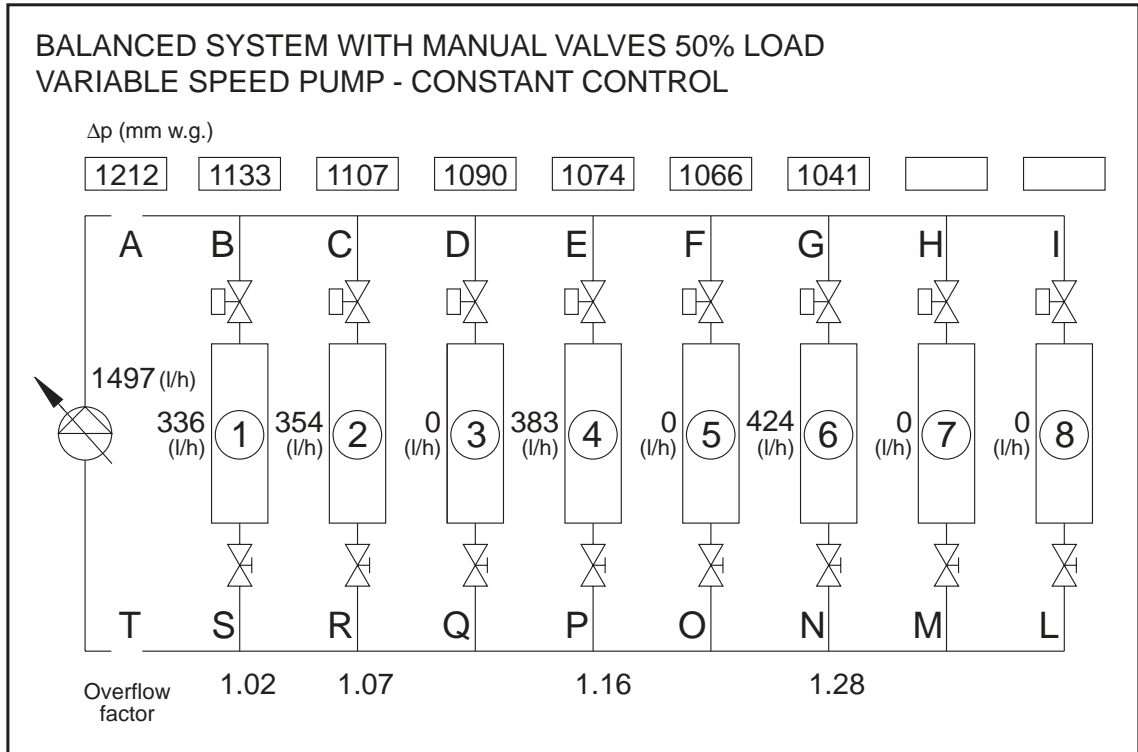
The flow rate corresponding to point C' is determined by the intersection of the curve of the circuit at partial load and the line $\Delta p = \text{constant}$.

In this case the flow rate is $G = 1497$ l/h.

Flow rates

In the scheme the new flow rate values of the branch circuits in these new conditions are reported (Scheme 10).

Scheme 10



The scheme shows that the overflow factors decrease, which means that there is an advantage compared to the constant speed pump. They are still > 1 though.

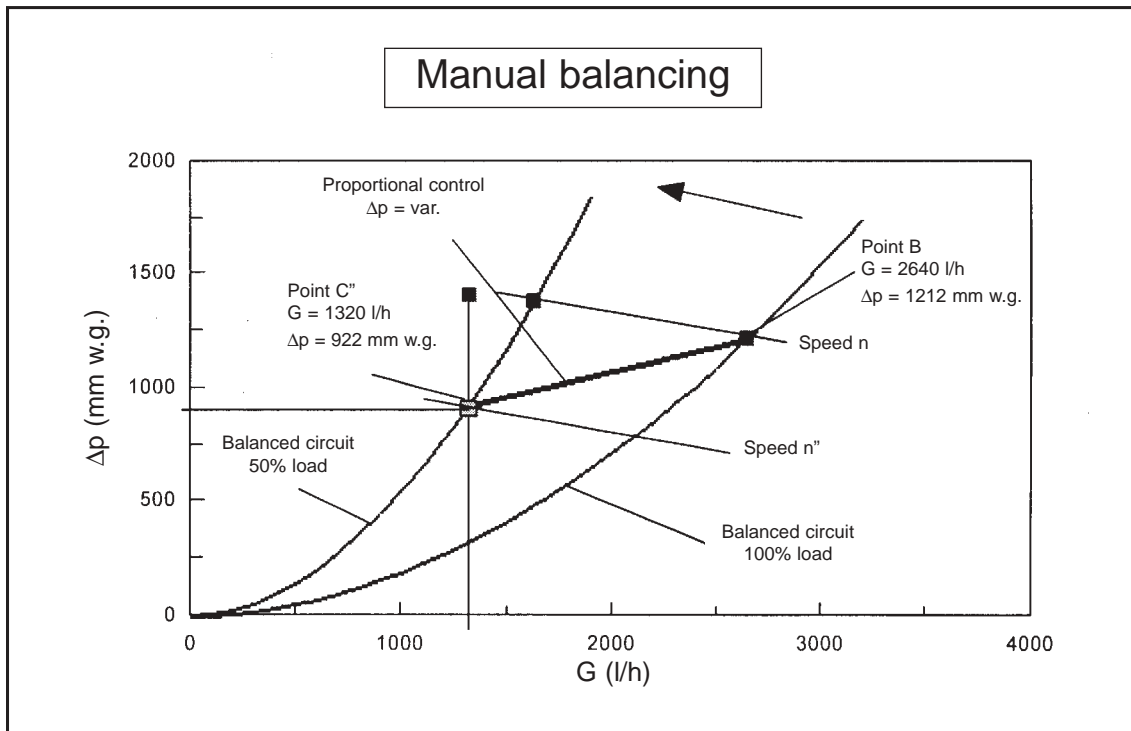
Even in this situation, manually balancing the circuit means that the design flow rate in the various circuits cannot be obtained.

Case 2 - Proportional control

The pump modifies its rotation speed in such a way that the **flow rate** as well as the **available head are reduced**. This method is called **proportional control** (Fig. 18).

Variable speed pump with proportional control.

Fig. 18



In these conditions, the branch circuits are supplied with the theoretical nominal flow rate that corresponds to a 50% partial load.

This means $G = 1320 = 330 * 4$ l/h.

A new point C'' can be found at the intersection point of the curve of the circuit at partial load and the line at constant flow rate $G = 1320$ l/h.

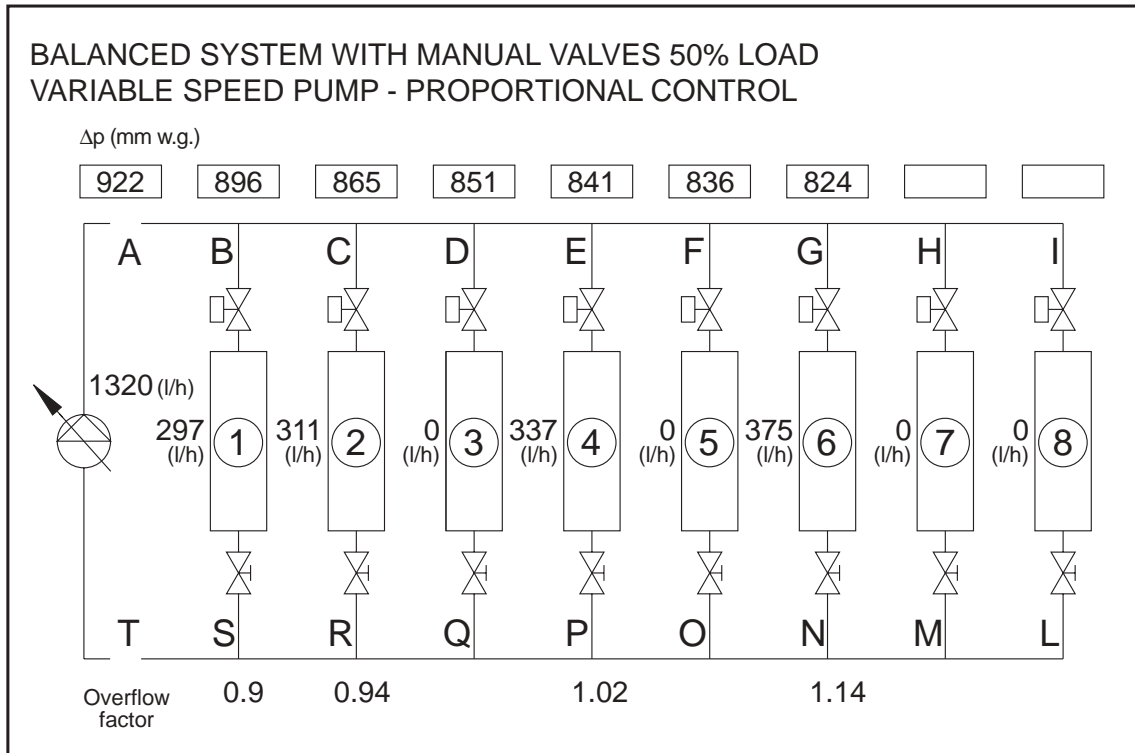
The new Δp value is: $\Delta p = 922$ mm w.g.

The new speed of the pump is n''.

Flow rates

In this case the new flow rates of the branch circuits can also be deduced (Scheme 11).

Scheme 11



Even in this case, the design flow rate values in the branch circuits are not obtained. In fact, there are not only oversupplying problems, but also **undersupplying** problems in the first terminals. These problems are caused by the manual balancing of these circuits according to the nominal values of the flow rate and Δp .

For example in the first branch circuit at total load, the following conditions were found:

$\Delta p = 1095$ mm w.g.; now $\Delta p = 896$ mm w.g.

As a consequence, the flow rate reduces by 10%.

When the level of partial load of the system is reduced, obviously these flow rate variations increase compared to the nominal values.

In this example a pump with an internal Δp sensor between inlet and outlet was used.

The choice of the position of the Δp sensor modifies the type of control chosen; it is noted in particular that installing the Δp sensor across the most remote circuit obtains a minimum energy consumption but creates problems of undersupplying in the first terminals. [8]

Thus, even when a variable speed pump is used in a manually balanced system, the problem of maintaining the designed flow rate when the working conditions vary remains.

Autoflow balancing - variable speed pump

On the contrary, if **Autoflow** had been used as an alternative to manual balancing valves, there would not have been these kind of problems.

The Autoflow **would automatically adapt itself** to the new load conditions of the system **by compressing itself if the Δp increases or releasing itself if the Δp decreases**, but still maintaining the same flow rate in the branch circuit.

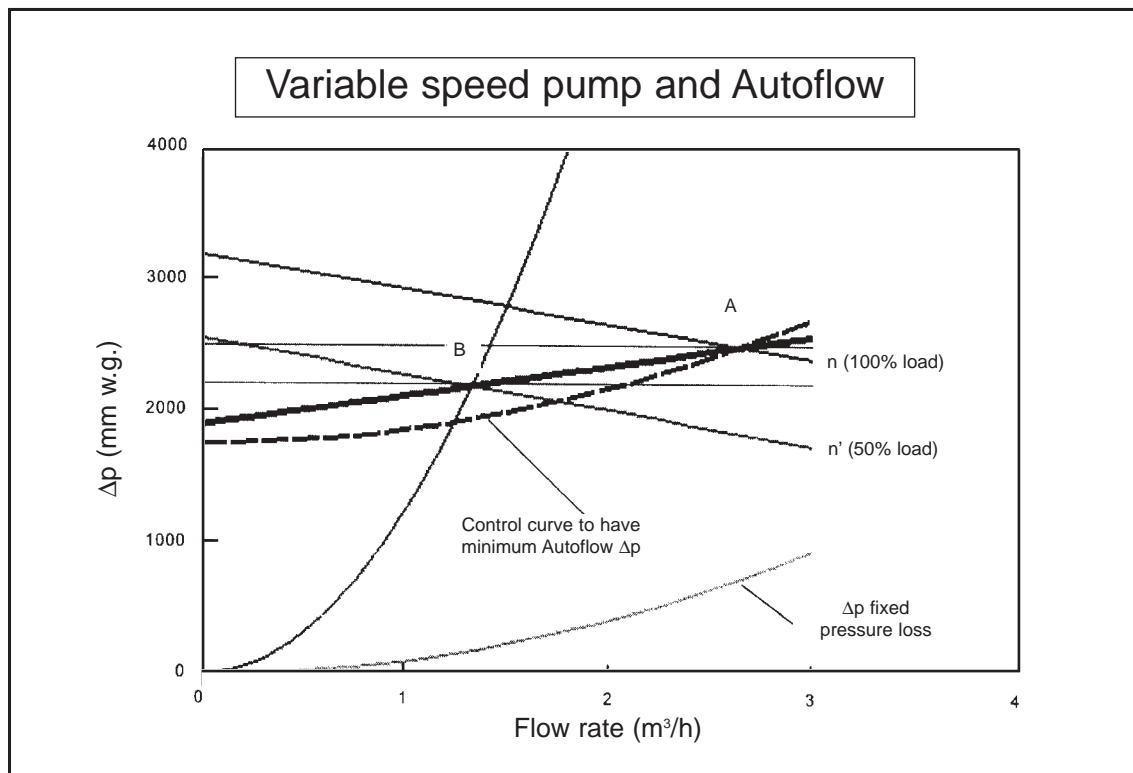
The only condition is that the Autoflow must have a minimum start up Δp , in accordance with the decrease of the head of the pump at partial load.

When the pump reduces the speed, it is necessary to make sure that the Autoflow will always be submitted to a Δp within its control range.

To simplify, it would be sufficient to guarantee a minimum operating Δp under every load condition for the Autoflow the most remote from the pump (Fig.19).

Minimum operating Δp

Fig. 19



Therefore, the point representing the new load condition (point **B**) in the graph, **must always be above the dotted curve.**

In fact, it is referring to the most remote circuit and operating in such a way, that the same Δp will be maintained on this circuit (fixed $\Delta p = 328$ mm. w.g. + minimum Autoflow $\Delta p = 1400$ mm w.g.).

If the Δp is constant in this circuit, the flow rate will also be constant.

If there is a minimum operating Δp on the Autoflow, the flow rate will be nominal.

Thus, if the Autoflow of the most remote circuit is submitted to a minimum operating Δp , all the others are submitted to a Δp within the control range.

Even in this case, the flow rate through every Autoflow will remain constant at its nominal value.

Energy saving

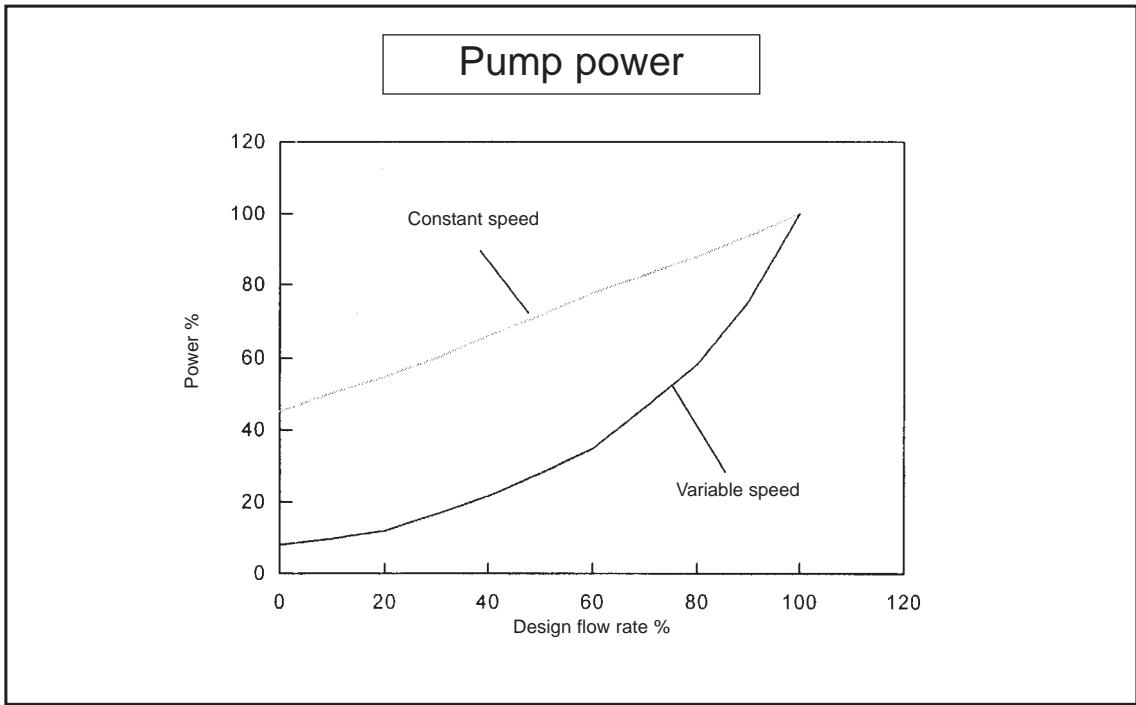
The application of a variable speed pump allows the input power to decrease when the load on the system decreases.

This decrease is shown in a graph comparing the input powers in case of reducing flow rates under working conditions at a constant or at a variable speed (Fig. 20).

For example, with a constant speed pump and a flow rate that equals 50% of the nominal value, the power will equal 75% of the power under full load conditions; with a variable speed pump on the other hand, the input power will equal 30% of the power under full load conditions.

Comparison constant speed pump - variable speed pump

Fig. 20



Conclusions

The following table summarizes the results of circuit balancing, obtained analysing the practical example.

Type of circuit	Type of pump	Load condition	Overflow factor-total	Max. overflow factor-branch circuit	% Input power
Unbalanced	Constant speed	100%	1.42	1.81	168%
Balanced with manual valves	Constant speed	100%	1.00	1.00	100%
Balanced with manual valves	Constant speed	50%	1.23	1.40	81%
Balanced with Autoflow	Constant speed	100%	1.00	1.00	100%
Balanced with Autoflow	Constant speed	50%	1.00	1.00	75%
Balanced with manual valves	Variable speed	50%	1.00	0.9-1.14	30%
Balanced with Autoflow	Variable speed	50%	1.00	1.00	30%

Application of circuits with Autoflow in systems

Here following, typical schemes of air-conditioning systems where Autoflow regulators are represented.

In all the schemes there is the necessity of maintaining the flow rate constant in certain points of the hydraulic circuit, in order to satisfy the different but well-defined and well-known requirements.

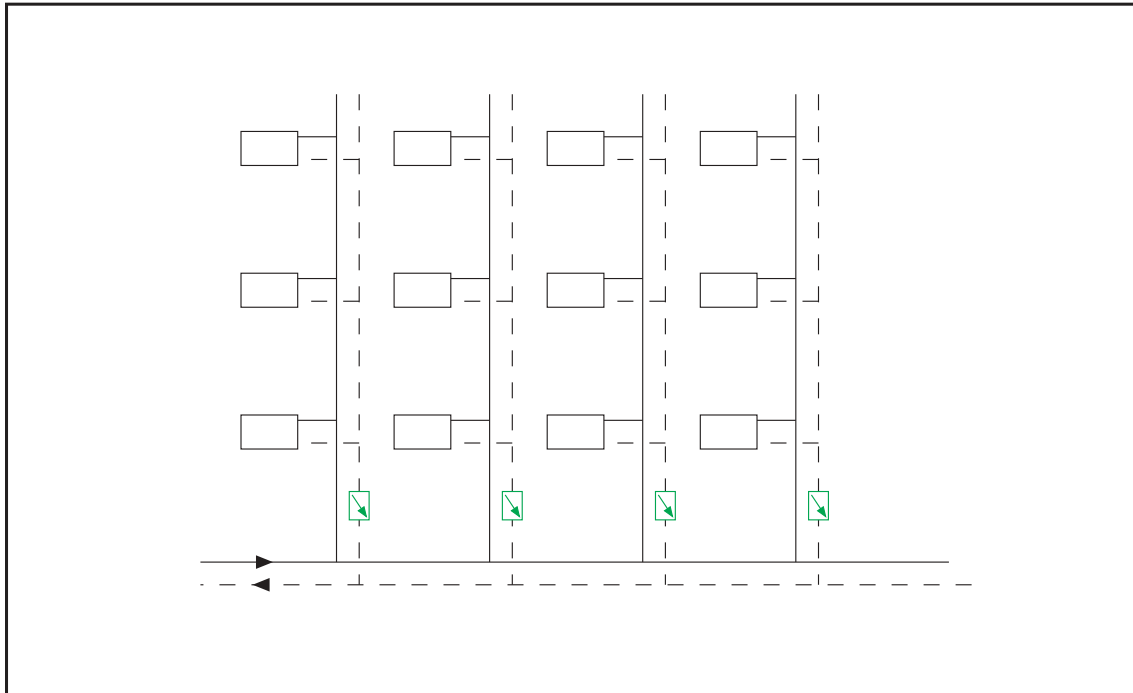
The same principle schemes with the necessary changes can also be used for other applications rather than the traditional air-conditioning systems, e.g. industrial process, refrigeration, chemical, hydraulic distribution systems, etc.

In air-conditioning systems, the circuits with Autoflow are especially used to assure the requested flow rates to the risers, the branches or the terminals.

They can be used with either 3-way or 2-way control valves in circuits with constant as well as with variable flow rate.

Balancing with Autoflow at the riser

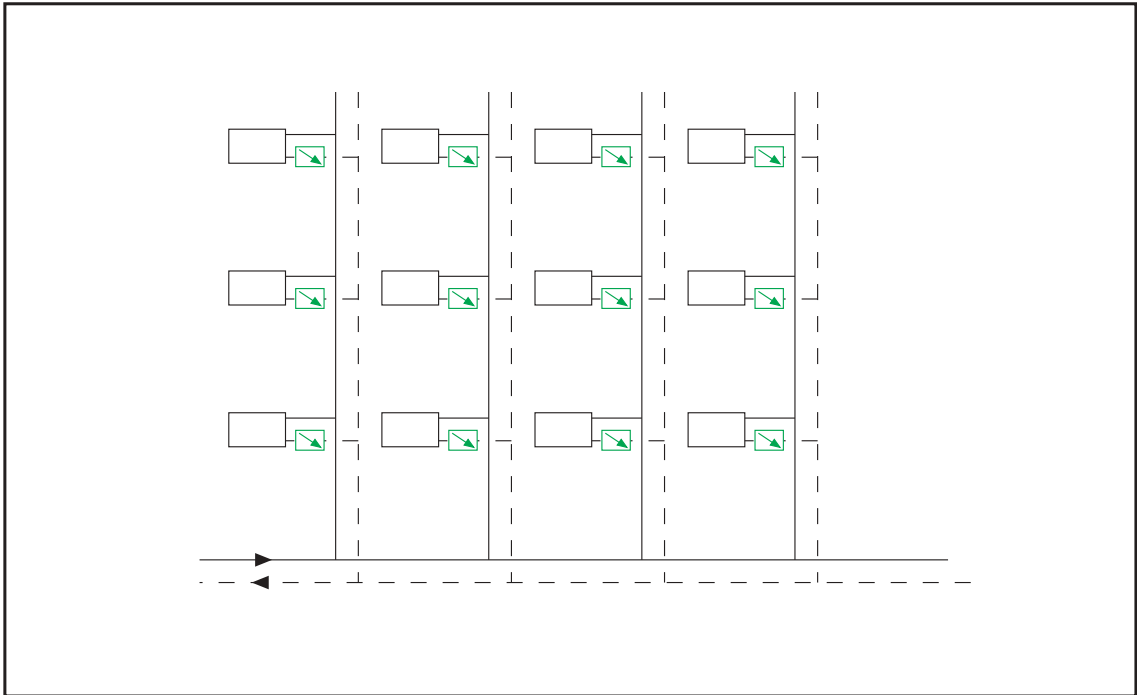
Fig. 21



The Autoflow devices are installed on the return pipe of the circuit.

Their function is to distribute the requested flow rate through the risers.

Unbalances in the distribution of the flow rates in the risers can occur for reasons evidenced in the examination of the simple circuits. For this reason, these circuits are normally used in buildings with no more than 5 or 6 floors.

Balancing with Autoflow at each terminal**Fig. 22**

The Autoflow devices are installed on the return pipe of the circuit.

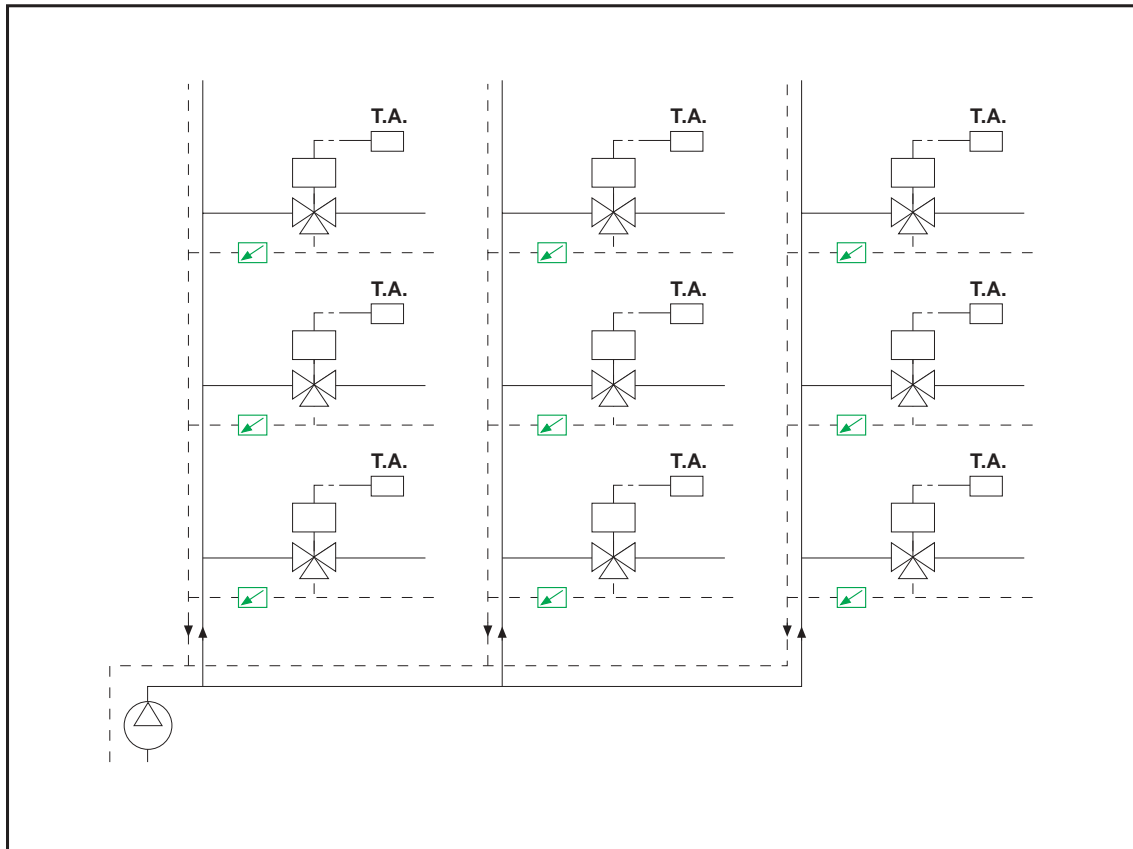
They are able to distribute the requested flow rates through each terminal.

Due to the wide control range of the Autoflow, it is not necessary to balance the risers on each other.

In case of manual valves on the other hand, it would be necessary to balance these risers too, by inserting an extra balancing valve at every riser itself.

Circuits with 3-way valves: balancing with Autoflow

Fig. 23

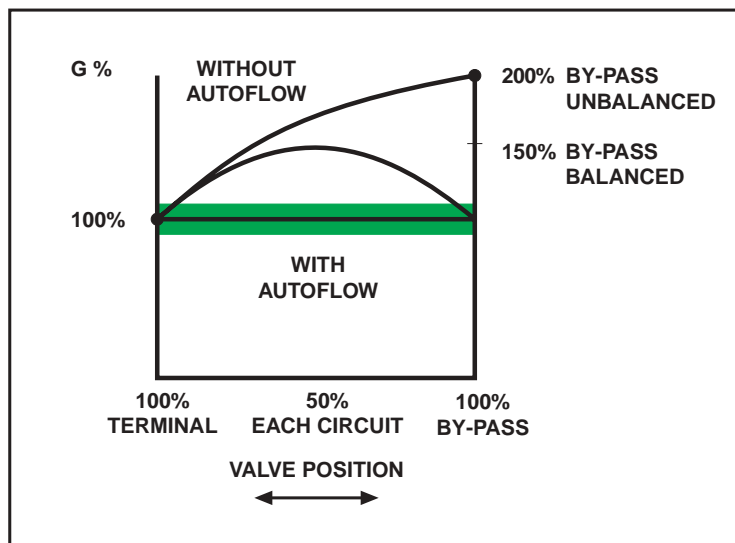


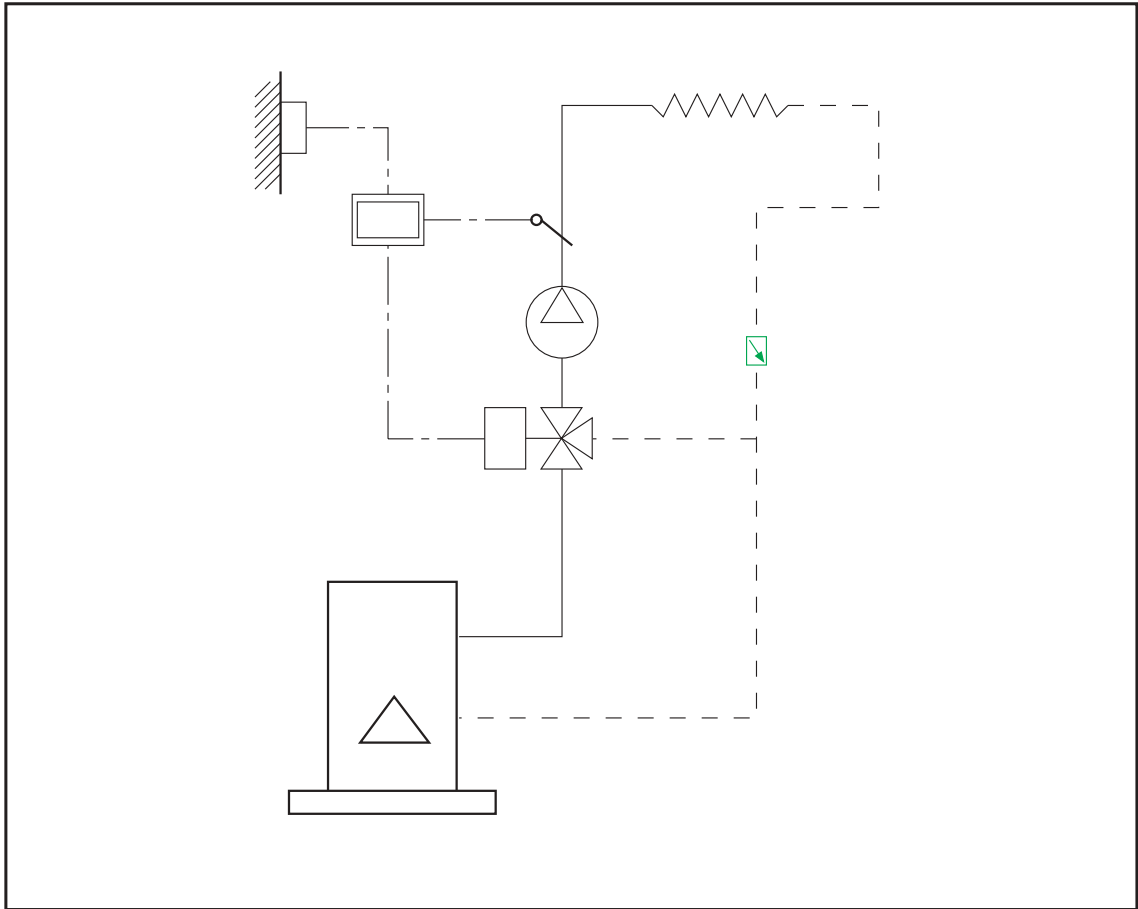
The Autoflow installed in the return of the zone circuits keeps the flow rate constant in every branch either with open valves or closed valves.

In this case, it is not necessary to balance the by-pass of the 3-way valves because Autoflow automatically absorbs eventual pressure drop variations on the way open to the zone or open to the by-pass.

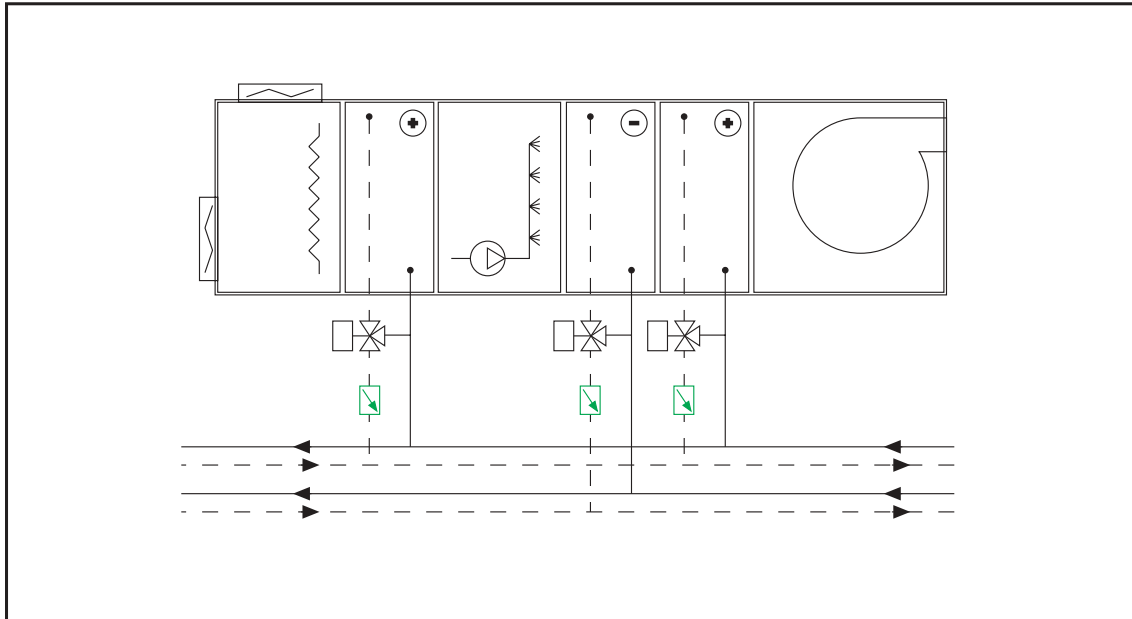
Furthermore, in case of modulating 3-way valves, the Autoflow keeps the total flow rate constant, even in the intermediate positions of the obturator of the valves. [5]

Fig. 24



Temperature control: balancing with Autoflow**Fig. 25**

The Autoflow keeps the flow rate of the circuit constant whether the valve is open or closed.

Control of coils with 3-way valves: balancing with Autoflow**Fig. 26**

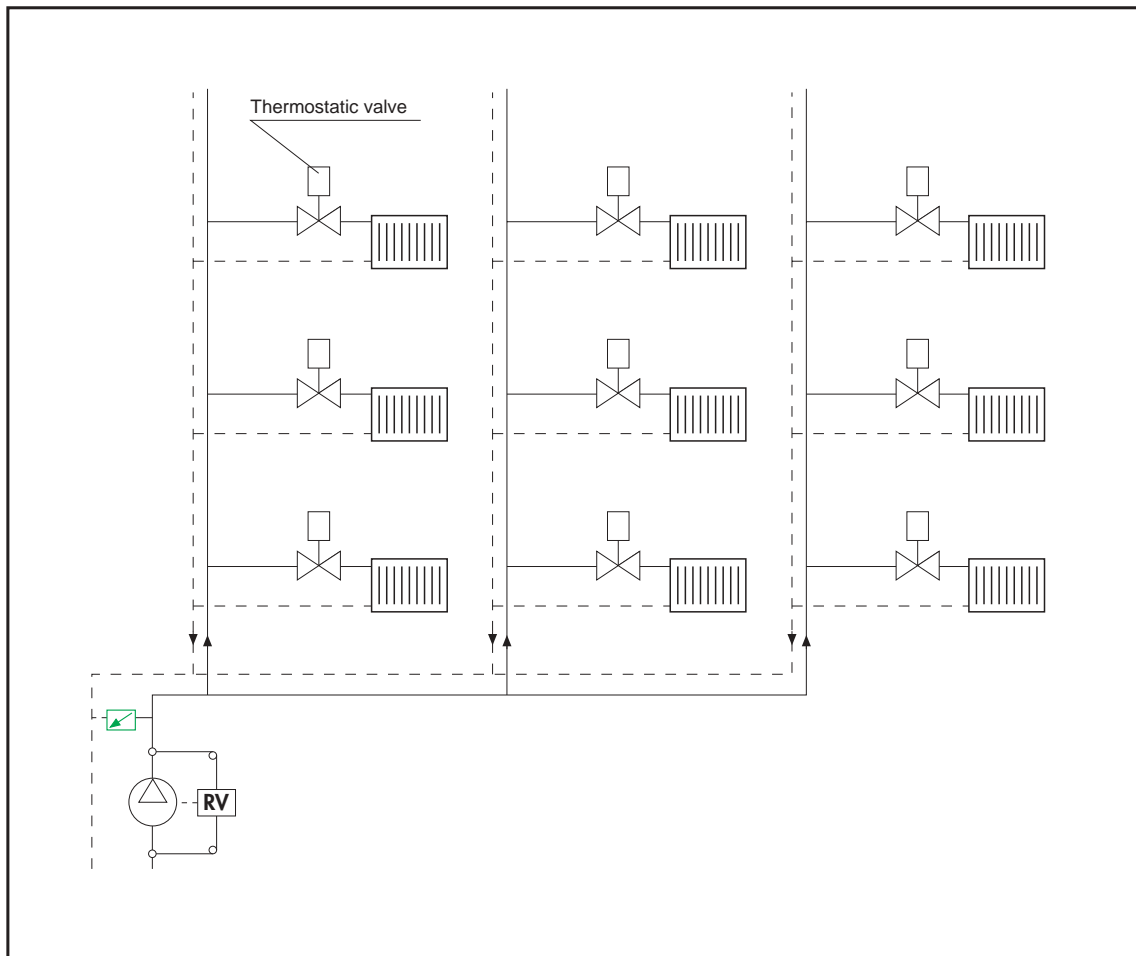
The Autoflow devices are installed on the return pipe of the circuits.

The Autoflow keeps the flow rates of the distribution circuits constant whether the valves are open or closed.

Furthermore, in case of modulating 3-way valves the Autoflow keeps the total flow rate constant, even in the intermediate positions of the obturator of the valves.

Differential pressure limiting device installed at the base of the circuit and by-pass with Autoflow

Fig. 27



This arrangement is mainly used on small systems using radiators fitted with thermostatic valves.

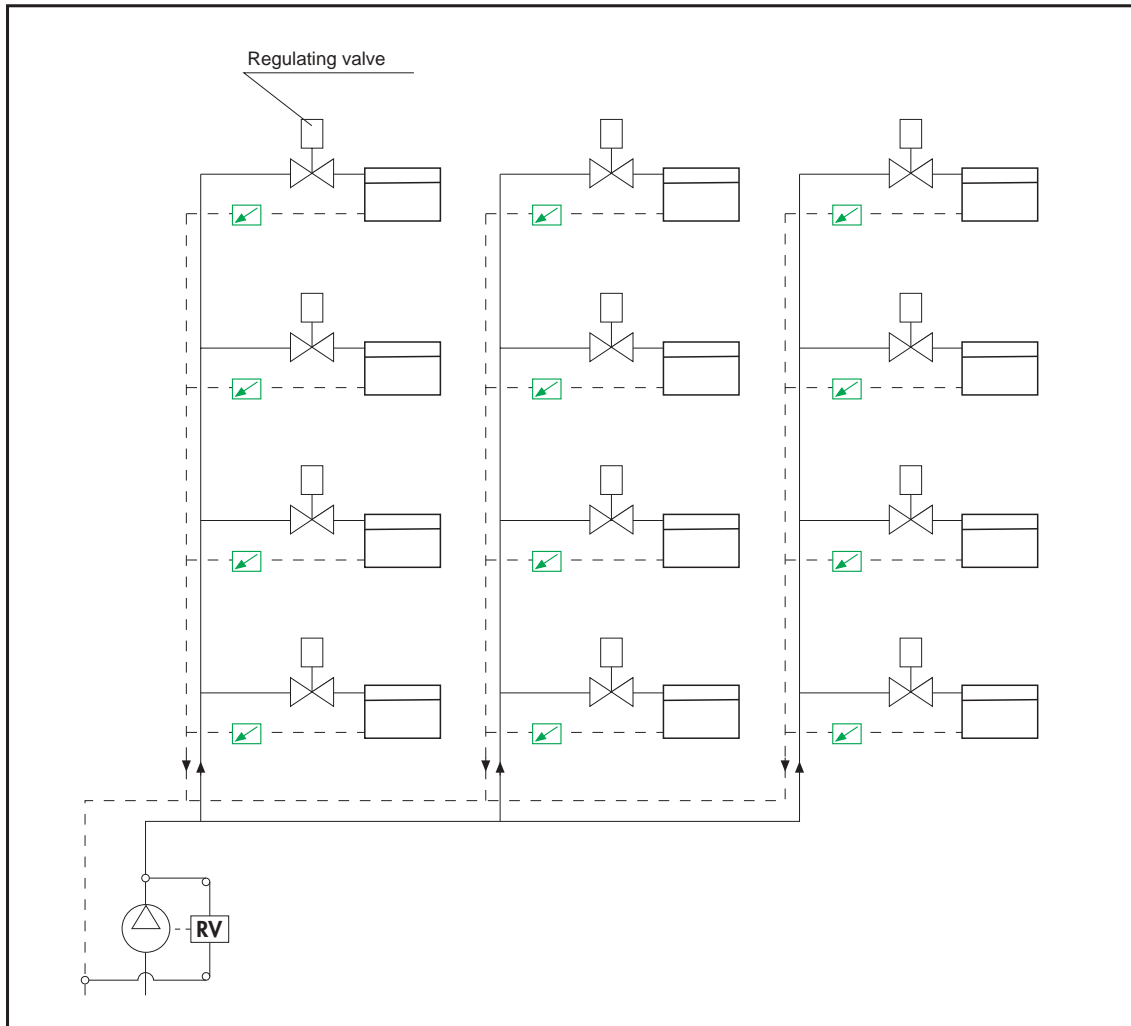
- **Setting pressure of the limiting device:**
It is recommended that this value is approximately 10% higher than the differential pressure (when the valves are open) between the points of the circuit in which the pressure ports of the limiting device are installed.

Remark:

The by-pass with Autoflow device is used to guarantee a minimum flow rate, even when the thermostatic valves are closed. This flow rate is required to prevent the thermal inertia of the boiler causing localised overheating of the fluid and operation of the overheat thermostat or other safety devices, such as the fuel valve or temperature controlled safety valves.

Control with 2-way valves and variable speed pump: balancing with Autoflow

Fig. 28

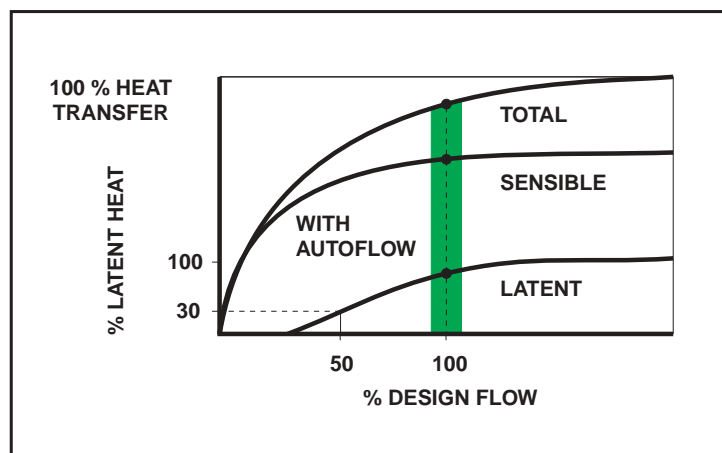


The Autoflow devices are installed on the return pipe of the circuit. This arrangement is mainly used in systems with fan-coil units and modulating control valves. The Autoflow stabilizes the flow rate passing through the terminal.

In a system with 2-way valves and a variable speed pump without automatic re-balancing of the system at reduced load (that is without Autoflow) terminals with an insufficient flow can occur.

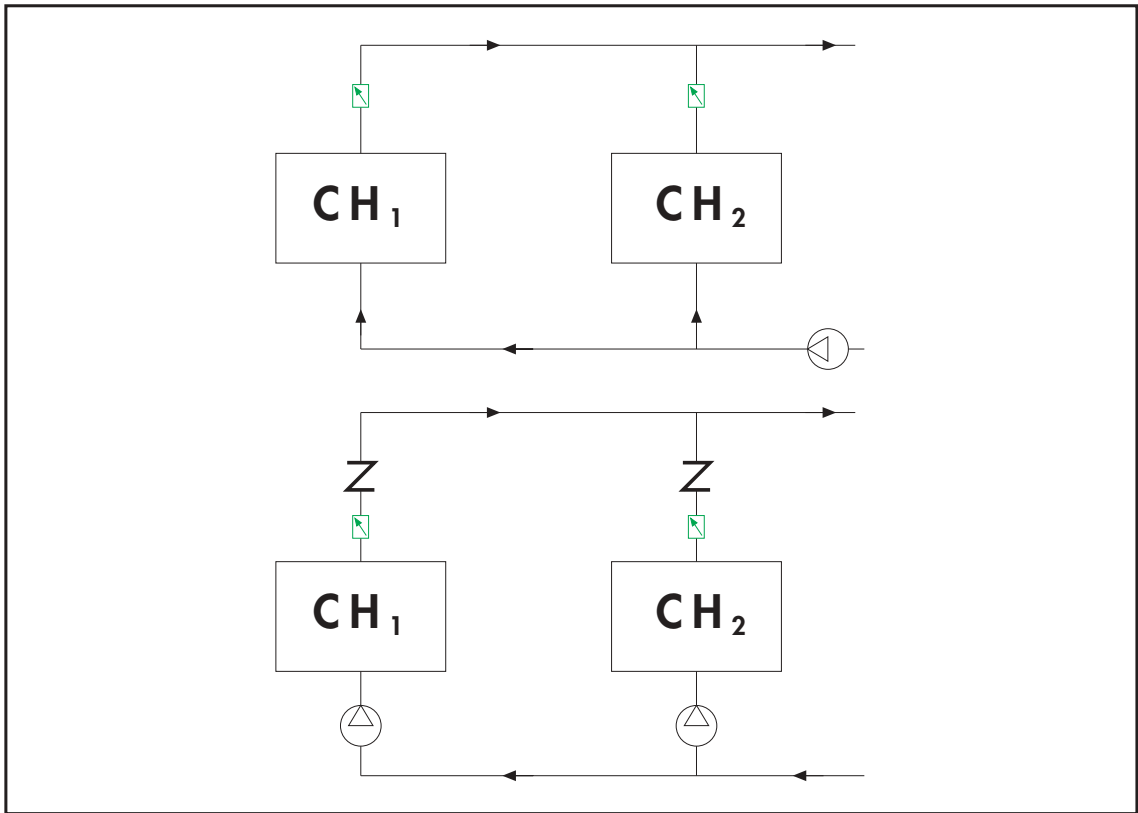
This situation reduces considerably, for example, the capacity of the dehumidification terminals. [6]

Fig. 29



Chilled water central plant: balancing with Autoflow

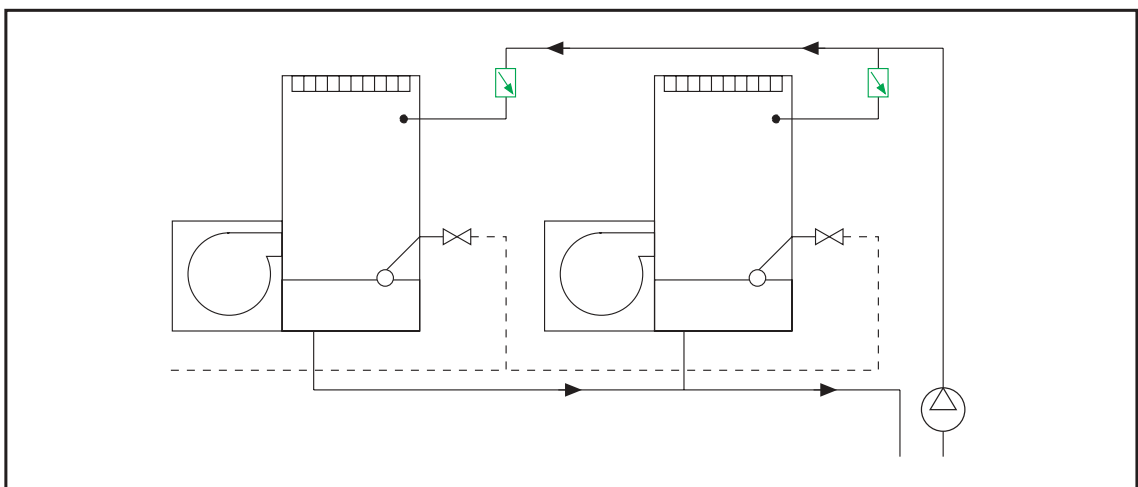
Fig. 30



The Autoflow devices are installed on the return pipe to balance the circuits supplying the evaporators or the condensers of the chillers. On the heat exchangers the flow rate is always constant even in case of shutting off the chillers, resulting into optimal performances.

Cooling towers: balancing with Autoflow

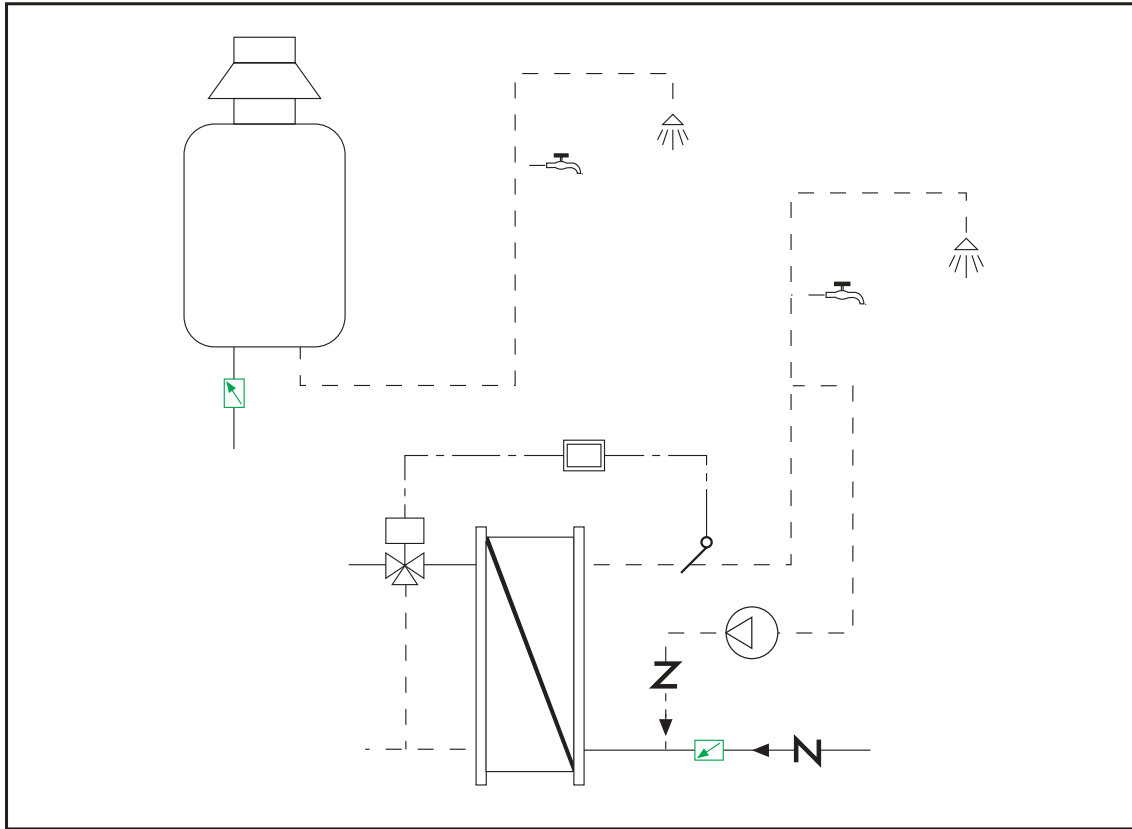
Fig. 31



They balance the circuits supplying the cooling towers.

Sanitary water production: limiting with Autoflow

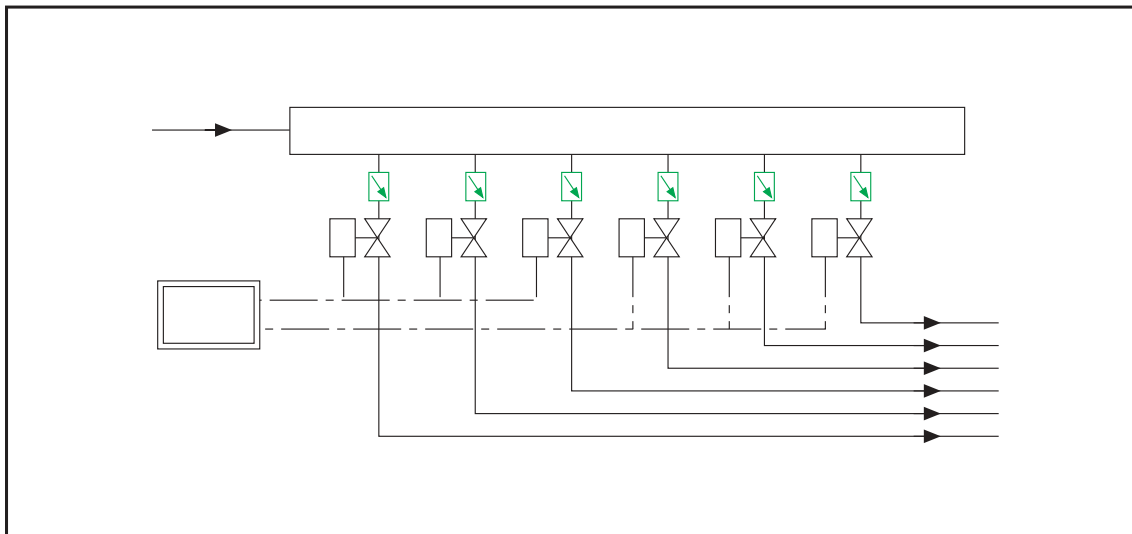
Fig. 32



The Autoflow limits the flow rate of the delivered hot water in the instantaneous production systems or in the production systems with limited capacity.

Water distribution systems: limiting with Autoflow

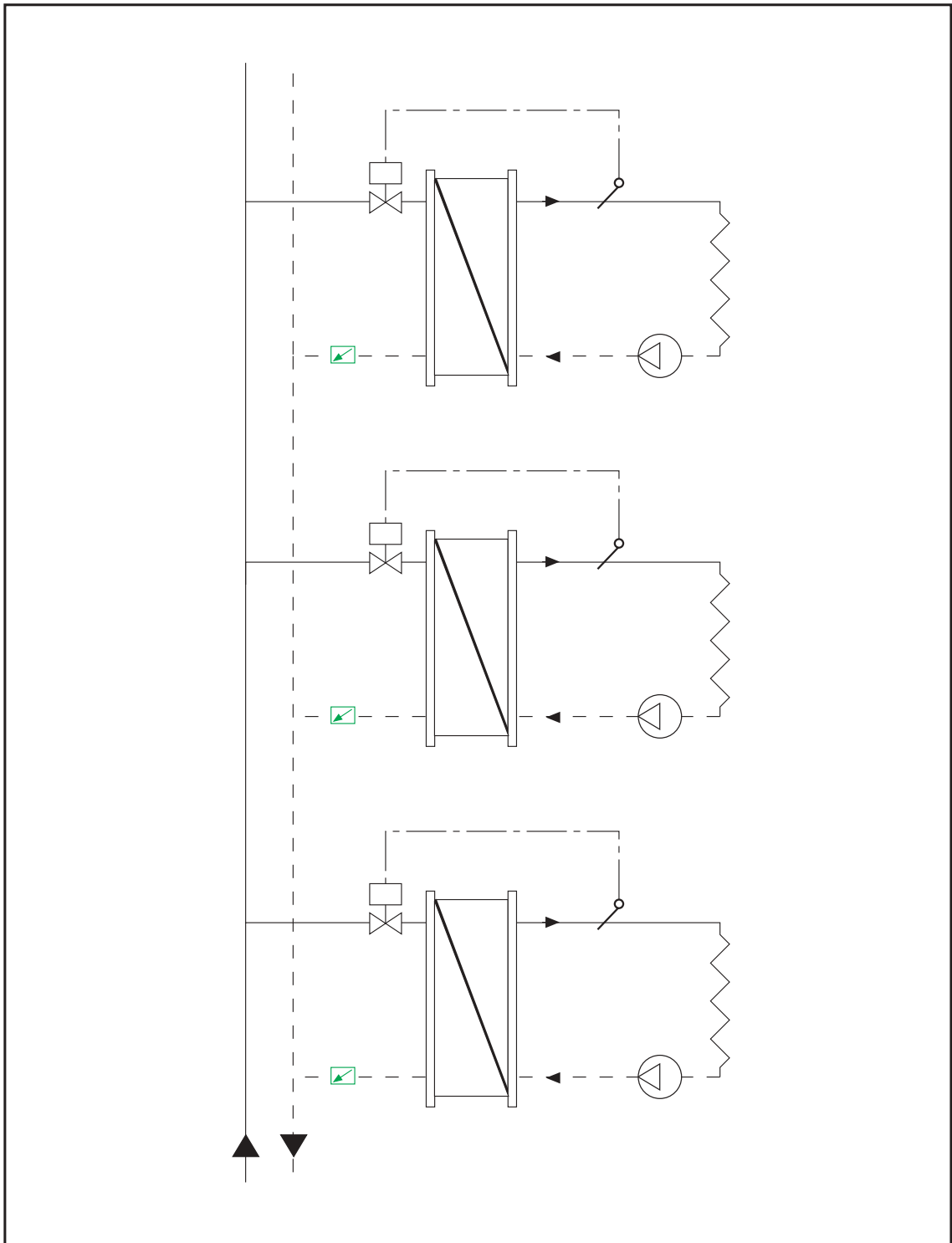
Fig. 33



To limit the delivered water flow rate and balance various circuits in irrigation systems.

District heating systems: balancing with Autoflow

Fig. 34



The Autoflow devices are installed on the return pipe of the circuit to balance the various secondary stations and limit the capacity at a nominal value in district heating systems. In this way, optimal performances of the heat exchangers are assured.

*TECHNICAL REPORT
APPENDIX*

DYNAMIC BALANCING OF HYDRONIC SYSTEMS

*THE AUTOMATIC FLOW RATE REGULATOR
AUTOFLOW*

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1st Section

Characteristics of circuits

Characteristic curves

The water supplying a circuit undergoes a **pressure loss** due to the passage through the piping and the other various components.

This pressure loss is the difference between the pressure at the initial point of the supply piping and the pressure at the final point of the return piping.

It is closely connected to the water flow rate that passes through the circuit according to the following relation:

$$\Delta p = Z * G^2 \quad \text{where: } \Delta p = \text{pressure loss}$$

$$G = \text{flow rate}$$

In this equation, **Z is the characteristic value of the circuit**. It only depends on the composition of the circuit itself. [3]

For example:

for a tube with a length of l : $Z = z * l$

in which **z** is a coefficient that depends on the tube diameter only;

for a localized resistance of x : $Z = x * m$

in which **m** is a coefficient that depends on the diameter only.

Knowing a working point $\Delta p - G$ of a circuit, a curve (parabolic type) showing the variation of Δp as a function of G can be made for one characteristic of the system Z .

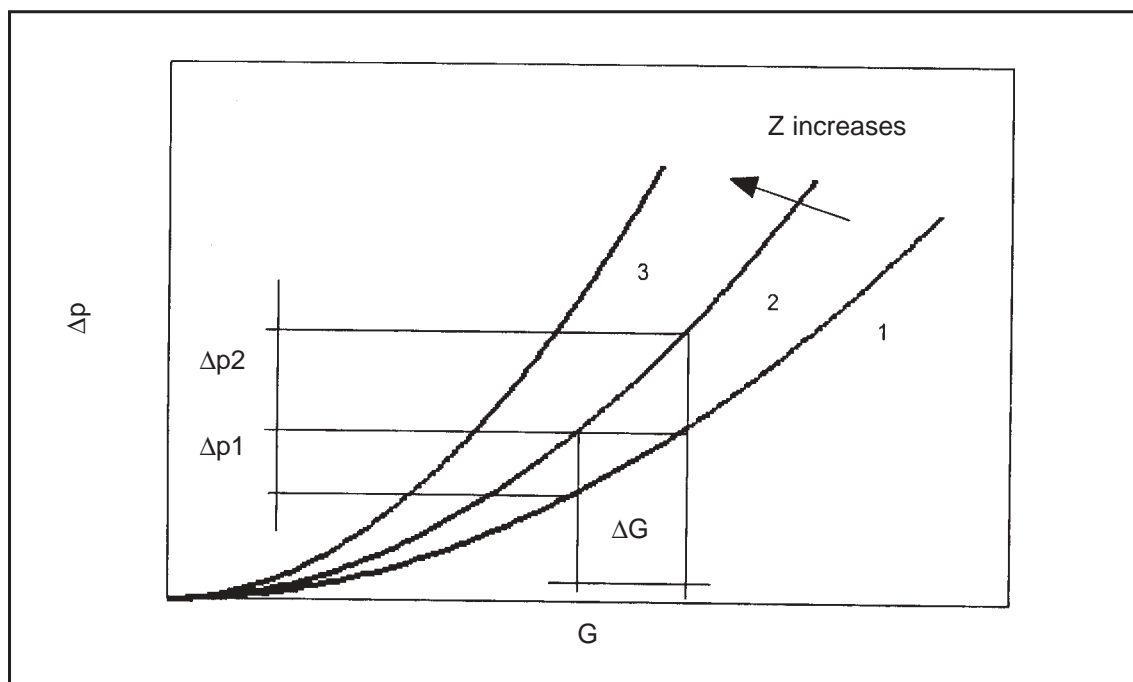
In a closed circuit the vertex of the parabola is at the origin of the axis.

This curve has the following **physical significance**:

by continuously maintaining the circuit under the same conditions and by only changing the flow rate the corresponding pressure drop can be determined.

In fact, in this case Z remains constant; it is the parameter that defines that particular curve.

Fig. 1



If, on the other hand the opening of the valves varies, a new condition of the circuit is created and therefore a new Z-value.

In particular a decrease of the passage of the fluid, causes an increase of the resistance of the circuit and therefore of Z.

The graph in Fig.1 shows that the lower the value of Z is, the less the flow rate variation influences the pressure drop of the circuit.

In fact, with the same flow rate variation ΔG , $\Delta p_1 < \Delta p_2$.

The main problem is to find a way to calculate each new characteristic of the circuit for every position of the closing and regulating devices.

Flow behaviour in a circuit

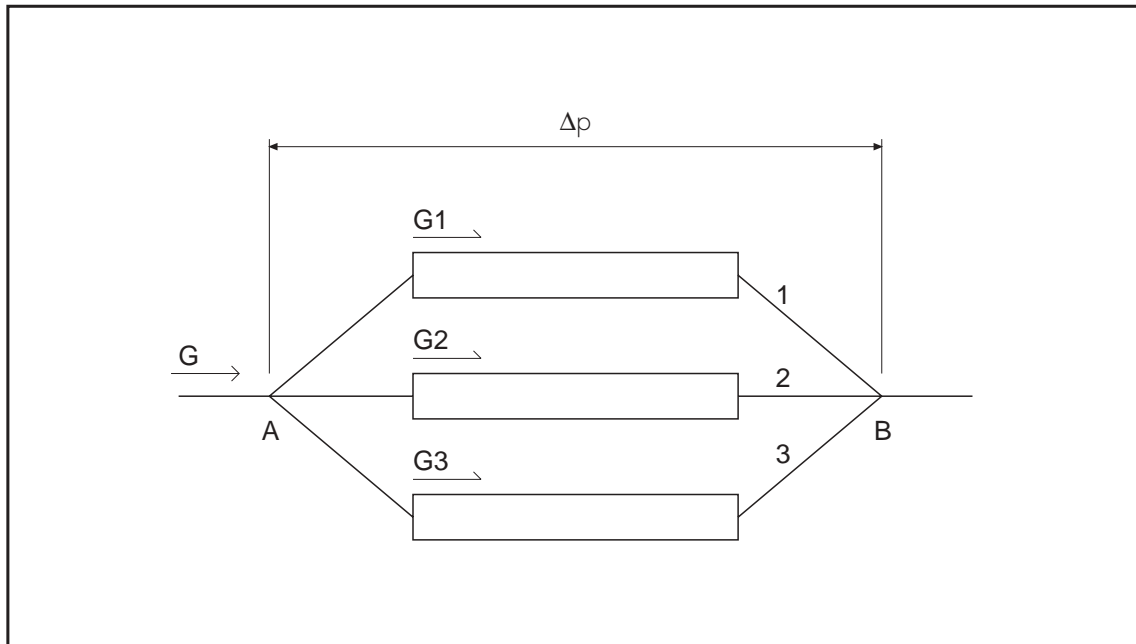
It may be helpful to point out the fundamental laws about the flowing of a fluid in a circuit (Fig. 2): [4]

- 1) the sum of flow rates that enter a node equals the sum of flow rates that leaves it;
- 2) the pressure drop between two nodes has the same value regardless of fluid path.

Furthermore the following laws are pointed out:

- 1) the flow between two communicating nodes (A-B) divides itself in single flow rates in order to have the same pressure drop in each branch of the connection;
- 2) a variation of the characteristic of one or more branches always causes a variation of the total characteristic and the distribution of the flow rates.

Fig. 2



$$\Delta p = \Delta p_1 = \Delta p_2 = \Delta p_3$$

$$G = G_1 + G_2 + G_3$$

Calculation of the characteristic of a circuit

The characteristic of a circuit can be calculated in different ways.

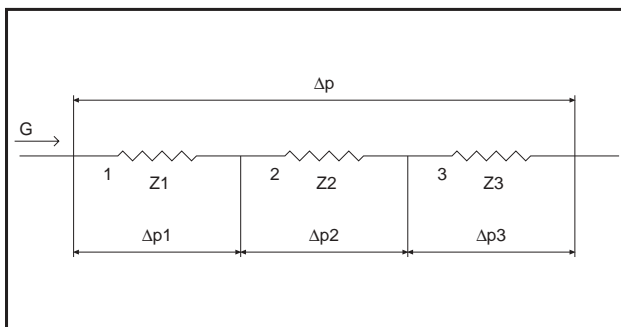
The calculation proposed hereafter is not a strict method but it has a series of operations that are easy to understand and to carry out.

This method especially allows a rather quick checking of the hydraulic unbalances that occur when the regulating devices are being closed or opened.

To calculate the characteristic Z of a circuit it must be divided in many basic branches. Each of them can be while remaining in analogy with the electrotechnics, a composition of elements in series or in parallel.

Elements in series

Fig. 3



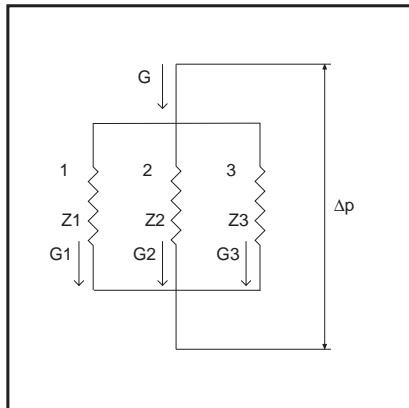
This means that to calculate the value of Z for a branch composed of elements in series, the various Z -values only have to be summed up (Tab. 1).

Tab. 1

Series
$\Delta p = Z \cdot G^2$
$G = G_1 = G_2 = G_3$
$\Delta p = \Delta p_1 + \Delta p_2 + \Delta p_3 =$
$Z_1 \cdot G^2 + Z_2 \cdot G^2 + Z_3 \cdot G^2 =$
$(Z_1 + Z_2 + Z_3) \cdot G^2 =$
$Z \cdot G^2$
$Z = Z_1 + Z_2 + Z_3$

Elements in parallel

Fig. 4



Tab. 2

Parallel	
$\Delta p = \Delta p_1 = \Delta p_2 = \Delta p_3$	
$G = G_1 + G_2 + G_3$	
$\Delta p = Z_1 \cdot G_1^2$	$G_1 = (\Delta p / Z_1)^{0.5}$
$\Delta p = Z_2 \cdot G_2^2$	$G_2 = (\Delta p / Z_2)^{0.5}$
$\Delta p = Z_3 \cdot G_3^2$	$G_3 = (\Delta p / Z_3)^{0.5}$
$\Delta p = Z \cdot G^2$	
$G = (\Delta p / Z_1)^{0.5} + (\Delta p / Z_2)^{0.5} + (\Delta p / Z_3)^{0.5} =$	
$\Delta p^{0.5} \cdot (1 / Z_1^{0.5} + 1 / Z_2^{0.5} + 1 / Z_3^{0.5})$	
$G = (\Delta p / Z)^{0.5} = \Delta p^{0.5} \cdot 1 / Z^{0.5}$	
$1 / Z^{0.5} = 1 / Z_1^{0.5} + 1 / Z_2^{0.5} + 1 / Z_3^{0.5}$	

In order to calculate the value of Z in this case, it is necessary to consider the reciprocal of the various terms (Tab. 2).

Knowing the Δp of the branch, the branch's flow rate and the characteristics of the branch and of one element, **the flow rate flowing through this element** can be calculated.

Tab. 3

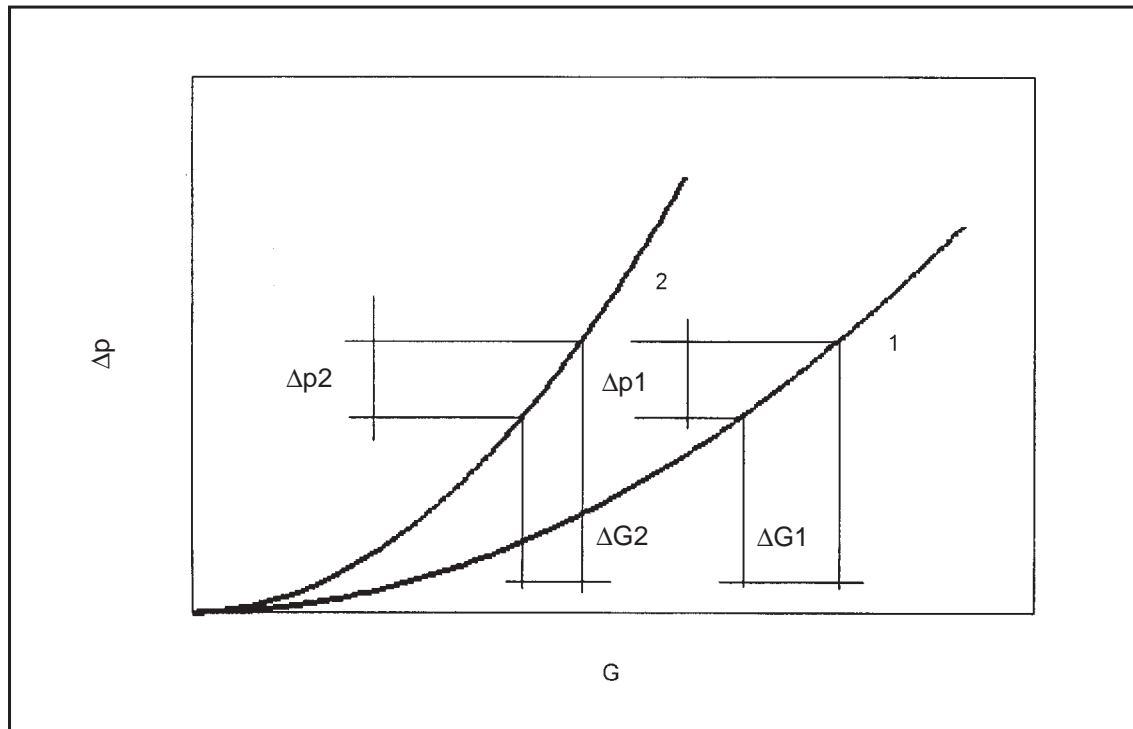
Flow rates
$G_i / G = (\Delta p / Z_i)^{0.5} / (\Delta p / Z)^{0.5} = (Z / Z_i)^{0.5}$
$G_i = G \cdot (Z / Z_i)^{0.5}$

Applying this formula, the new flow rate in an element can be calculated when the total flow rate in the branch varies without the Z-values being changed (neither in the branch nor in the element) (Tab. 3).

Stability of a circuit

A circuit is more stable than another circuit when a same variation in pressure drop Δp causes lower flow rate variation ΔG (Fig. 5).

Fig. 5



Considering the graph there is:

$$\Delta p_1 = \Delta p_2 \quad Z_2 > Z_1$$

$$\Delta G_1 > \Delta G_2$$

Circuit n° 2 is more stable

Relative unbalance

In case of a variation of the available differential pressure in a parallel circuit, the relative unbalance between these circuits is less perceptible when they have similar characteristics between the various circuits (Z is almost identical).

If the Z -values of the various circuits are identical the variation of the flow rate of each element will have the same relative value.

Interdependence

If the circuits are in parallel the risk to have an unbalanced element is higher when the common branch has a relatively high pressure loss (high Z -value).

In this case the circuits are very interdependent.

On the contrary, if the common branch has a negligible pressure drop compared to the one of the various circuits in parallel there is no risk of having a relative unbalance. In this case the working of each circuit is almost independent of the working of the others.

Authority

At this point the general hydraulic authority can be defined as the ratio between the pressure loss of the determined element and the total pressure drop of the circuit (this latter equals the head of the pump which will be shown later).

The higher the authority of an element, the more independent its working will be.

Balancing and calibration will be easier because they are not influenced by the balancing and the calibration of the other elements.

For example, a terminal fitted with a control valve with a low general hydraulic authority is considered. If this terminal is part of a system in which all the control valves are closing except for the one of this terminal, the working of this valve will be very disturbed because of an increase of the available differential pressure that comes into this circuit.

In case of high authority on the other hand, the functioning of the valve is not disturbed noticeably by the limited increase of differential pressure.

Characteristics of the pumps

In a heating, cooling or air conditioning system the function of the pump is to circulate the water from the producer, which can be a boiler or a chiller, to the terminals.

The pump has to supply a certain water flow rate in determined pressure conditions, in order to win the resistance along the distribution circuit.

Among all the different types of pumps, the behaviour of the centrifugal pump will be analysed here, which is mostly used for this type of application.

The figures relating to the characteristics of the pump, experimentally determined, are represented graphically by the manufacturer.

Characteristic curves

The characteristic curve of the pump shows the **relation between head and flow rate**, which allows to choose the pump that best satisfies the requirements of the circuit. The pump head is a quantity indicating the energy supplied to the fluid; it can also be indicated in terms of pressure. [4]

For every rotation speed n there is a characteristic curve.

Fig. 6

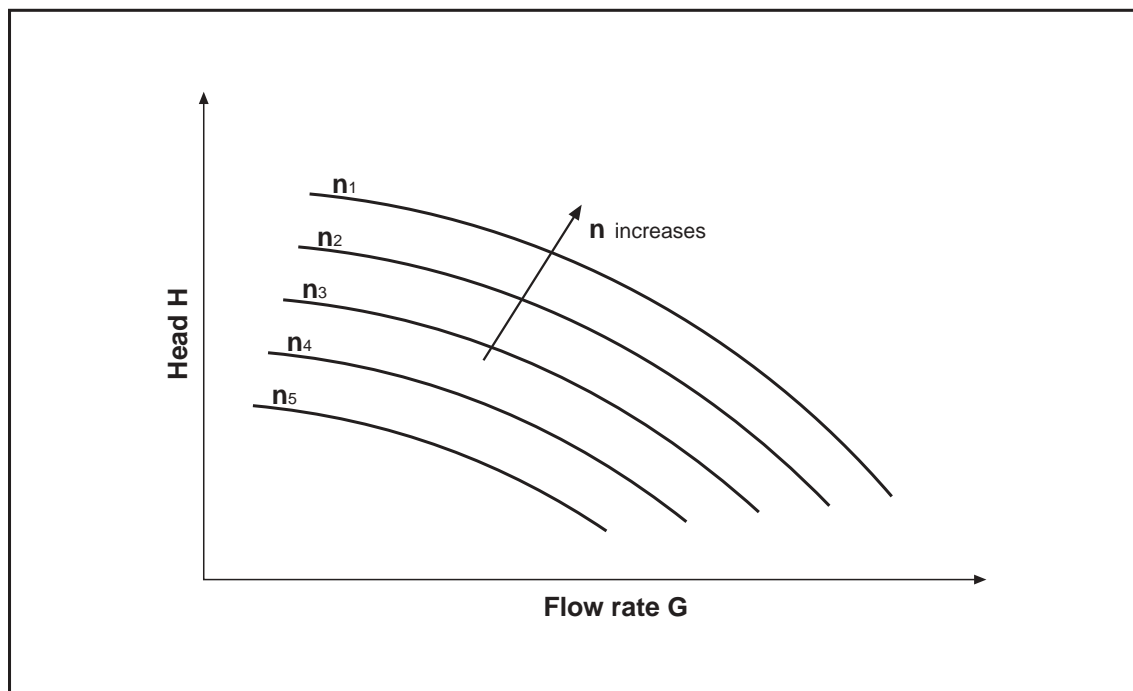


Figure 6, shows that when the speed decreases, the maximum discharge head and the maximum flow rate decrease (from n_1 to n_5).

The different curves can be obtained one from the other by a simple translation.

Additional curves

Furthermore, the manufacturer still in function of the flow rate, experimentally determines the curves related to other quantities. This allows the designer to get a wider view of the pumps behaviour under different working conditions (Fig. 7).

Input power - flow rate curve

The input power depends on the working characteristics, flow rate and head of the pump; it is the input power at the pump shaft.

$$P = \frac{\rho \cdot G \cdot H}{367,2 \cdot \eta}$$

Where: P = input power of the pump, kW
 ρ = density of the fluid, kg/dm³
 G = flow rate, m³/h
 H = head, m w.g.
 η = efficiency, dimensionless

Efficiency - flow rate curve

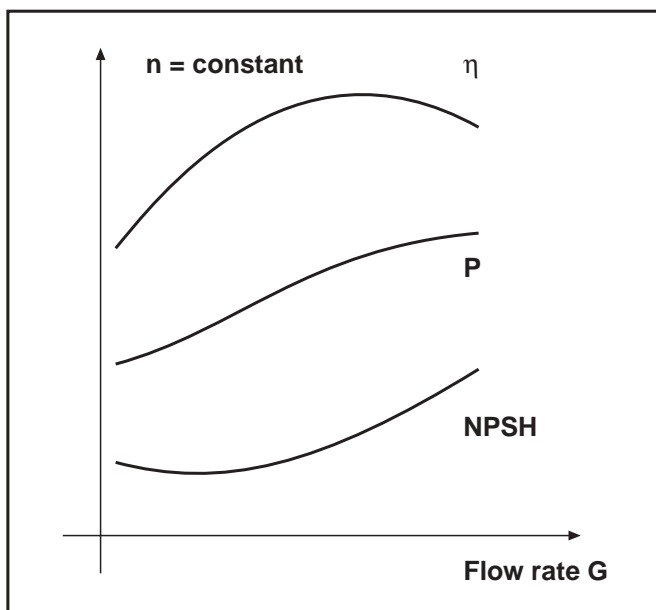
The efficiency of a pump is the ratio between the power delivered to the fluid in terms of flow rate, head, and the input power at the shaft.

$$\eta = \frac{P_{\text{OUTPUT}}}{P_{\text{INPUT}}}$$

NPSH - flow rate curve

The NPSH (Net Positive Suction Head) values represent the minimum pressure that has to be guaranteed at the inlet of the pump in order to avoid cavitation damages.

Fig. 7



In particular it is noticed that **the input power curve P** always rises with the **flow rate G**, maintaining a constant rotation speed n.

The efficiency η however, has a maximum value corresponding with a certain flow rate G: it always decreases in case of changes of the flow rate G from this value.

Isoefficiency curves

It can be interesting to report the characteristic curves in the same graph showing the pumps efficiencies while its rotation speed, flow rate and head vary. These curves called isoefficiency, show some areas in which the efficiencies are constant (Fig. 8).

Fig. 8

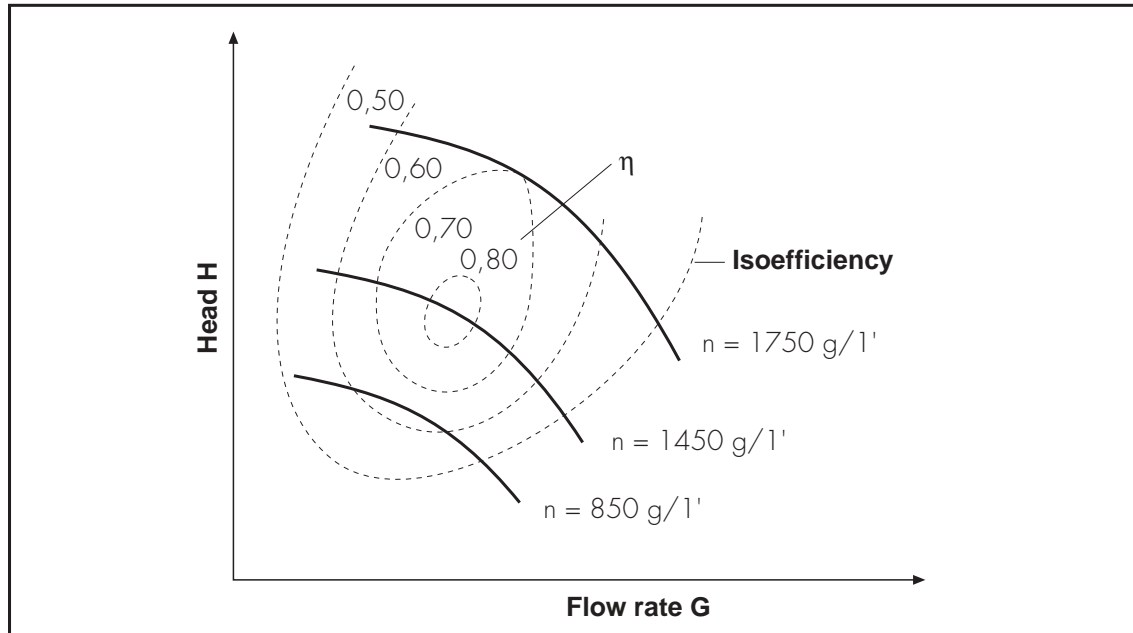
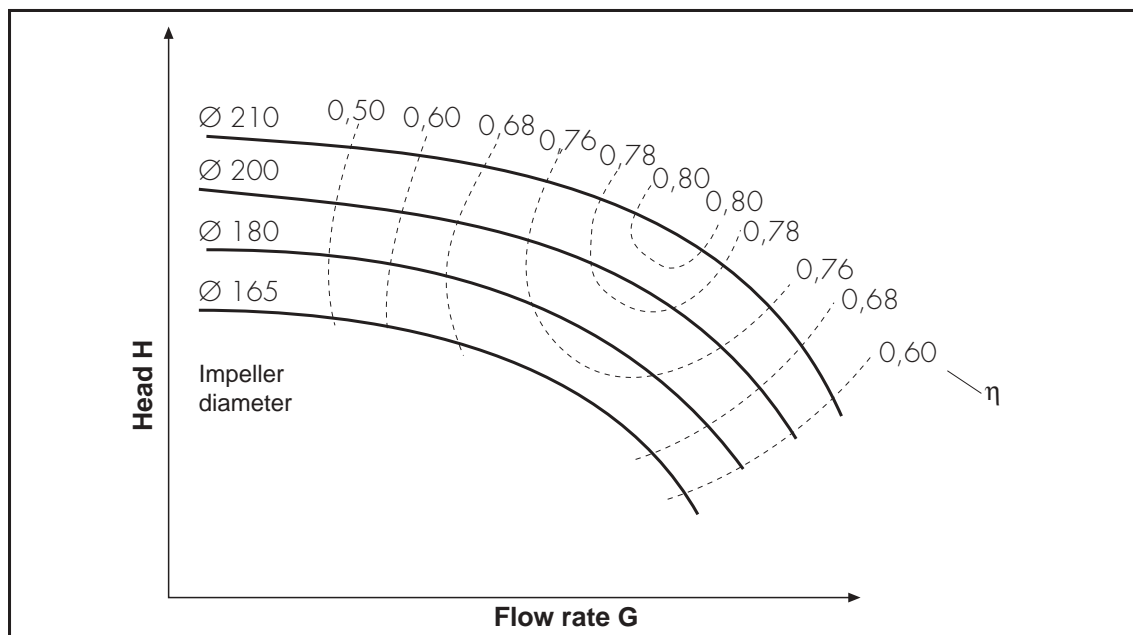


Fig.8 shows how much the area of maximum efficiency decreases and how this efficiency reduces as soon as it deviates a little from this highest efficiency, whether the flow rate increases or decreases.

In many cases the manufacturers also supply curves showing the operating of the pump when the diameter of the impeller varies maintaining a constant rotation speed (Fig. 9).

Fig. 9



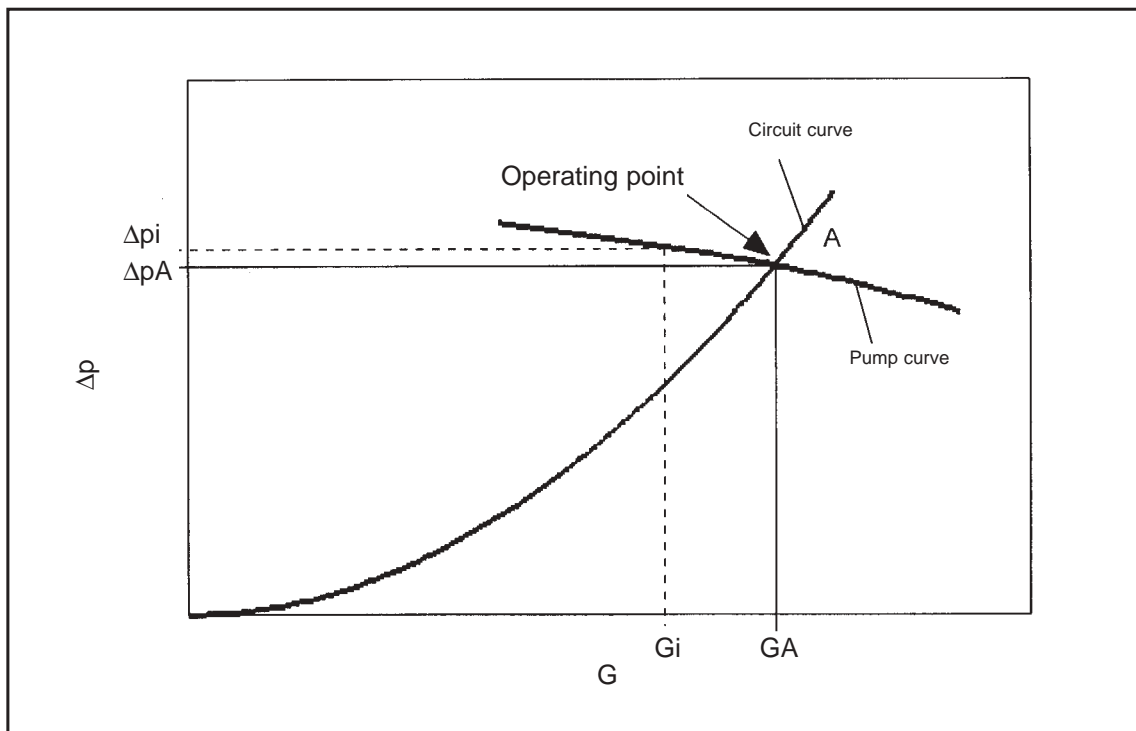
Pump - circuit functioning

Operating point

As shown before, the pump has a working characteristic that represents the relation between flow rate and head. This curve has been experimentally determined by the manufacturer who measured the correspondent head by varying the flow rate.

In practice these flow rate variations can be obtained by varying the resistance on the discharge of the pump. When a pump is installed into a circuit, this circuit can be considered as the resistance mentioned above.

Fig. 10



Operating point A is the equilibrium point between the pump and the circuit (Fig. 10). This point corresponds to:

$$\Delta p_{\text{pump}} = \Delta p_{\text{circuit}}$$

$$G_{\text{pump}} = G_{\text{circuit}}$$

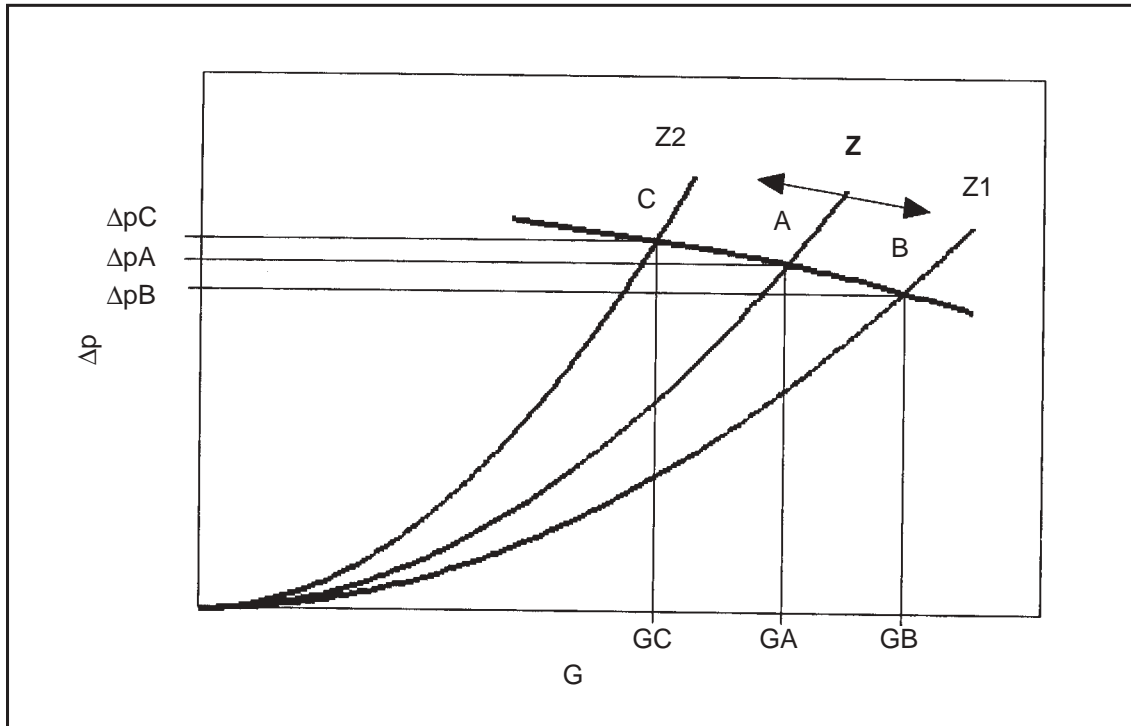
Knowing the characteristic of the pump and the characteristic of the circuit, it is easy to determine this common point at the intersection of the two curves in a graph.

Change of the operating point

In case of variation of any of the closing or regulating devices, the characteristic curve of the circuit varies according to the new Z-value.

This operation determines a new operating point (Fig. 11).

Fig. 11



The operating point shifts from A to B or C and determines the corresponding flow rates and heads:

from A to B opening of the valves ($Z_1 < Z$)

from G_A to G_B $G_B > G_A$ $\Delta p_B < \Delta p_A$

gives a smaller available differential pressure in order to guarantee the G_B flow rate;

from A to C closing of the valves ($Z_2 > Z$)

from G_A to G_C $G_C < G_A$ $\Delta p_C > \Delta p_A$

gives a higher available differential pressure in order to guarantee the G_C flow rate.

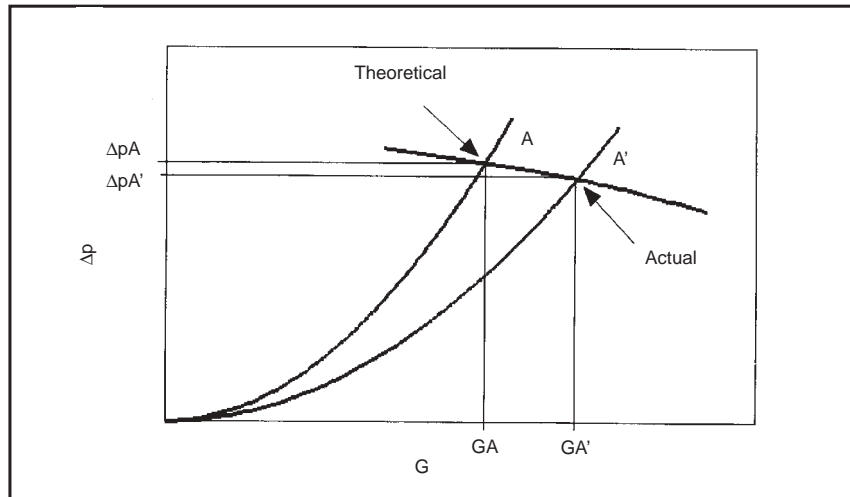
Theoretical point - Actual point

It often happens that during the development of the project, several safety factors are taken into consideration to calculate Δp .

Usually, for an installed system, the pressure loss in the designing phase is higher than actually experienced, the consequence is a shifting of the operating point towards the right.

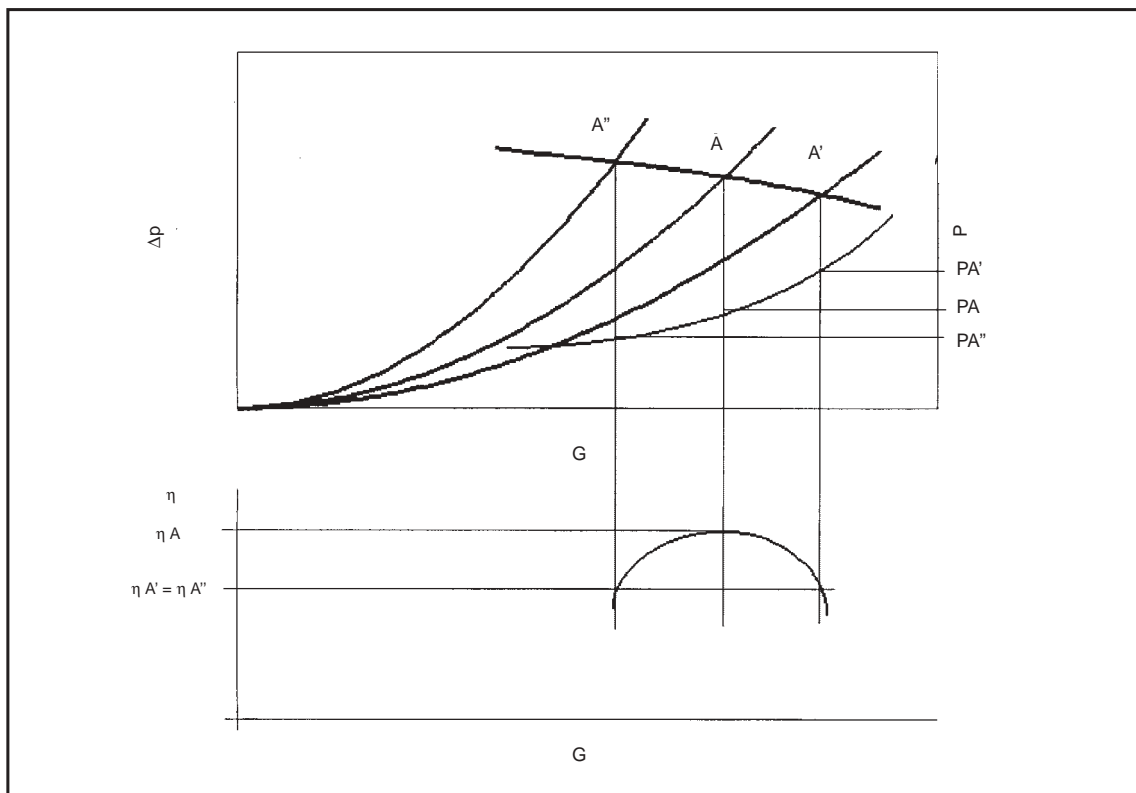
Therefore, the actual flow rate is higher than the theoretical one (Fig. 12).

Fig. 12



This is unacceptable because it causes the pump to work with a lower efficiency and a higher energy consumption (Fig. 13).

Fig. 13



It can be noted that $\eta A' = \eta A''$. However, at point A'' the input power, and therefore the energy consumption, is lower than at point A' .

$PA'' < PA'$.

Variable speed pump

Functioning

With the highest efficiency area being rather limited in some cases, it can be useful to let the pump work in such a way that the rotation speed reduces when a **contemporary reduction of the flow rate and the head**, following a reduction of the load of the system, is required.

The speed of the pump is reduced automatically, connected with a signal of a Δp sensor positioned along the circuit. In this case the pump is able indeed to provide a lower head than necessary when working at a constant speed (Fig. 14).

The input power is lower.

Fig. 14

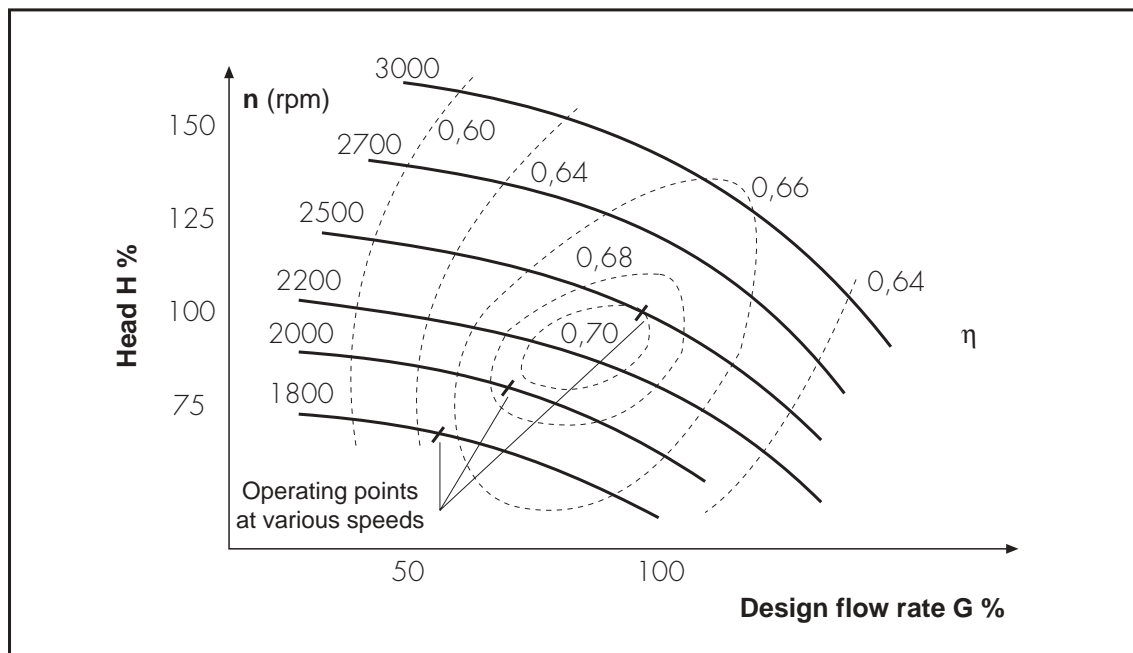
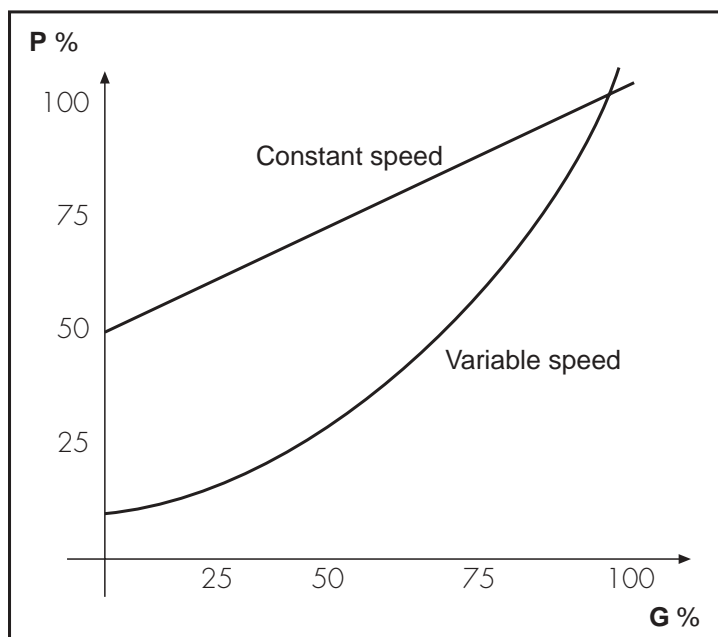


Fig. 15



Energy saving can be represented in a graph displaying the input power when working at a constant or variable speed (Fig. 15).

For example:

$G = 50\%$
 $P_{var} = 30\%$
 $P_{const} = 75\%$

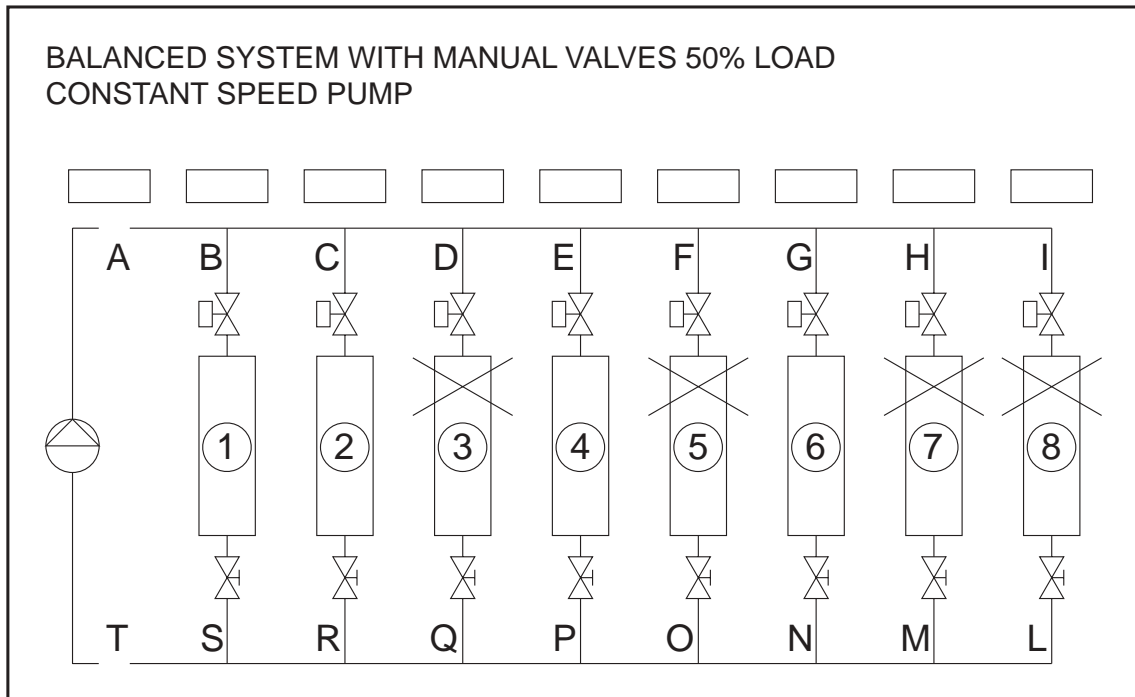
In this case the comparison is made with a Δp control sensor of the pump positioned across the most remote circuit; this position guarantees the lowest operating pumping costs. [8]

2nd Section

Balanced circuit with manual valves at partial load

In order to understand what happens in case of partial closure of the circuit, some terminals of the previous example are closed.

Scheme 1



In order to know the new working conditions, the common point between the characteristic of the pump and the circuit has to be calculated. It is necessary to know the Z-value which characterizes the new situation.

With the calculations made to dimension the circuit, the Z-values are obtained, knowing the Δp and G of every terminal and branch.

Using the relation $\Delta p = Z \cdot G^{1.9}$ the Z-values can be obtained (Tab. 1).

(In this case the exponent 1.9 is used, determined experimentally, because it is more representative for an actual system). [3]

Z-values of the circuit**Tab. 1**

TERMINAL	ΔP (mm w.g.)	G (l/h)	Z	BRANCH LINE	ΔP (mm w.g.)	G (l/h)	Z
1	1095	330	0,017957159	AB-ST	58,5	2640	1,8454E-05
2	969	330	0,01589055	BC-RS	63	2310	2,5614E-05
3	874	330	0,014332928	CD-QR	47,5	1980	2,5883E-05
4	809	330	0,013266978	DE-PQ	32,5	1650	2,5041E-05
5	757	330	0,012414218	EF-PO	26	1320	3,0611E-05
6	647	330	0,01061303	FG-NO	55	990	0,00011186
7	514	330	0,008429205	GH-MN	66,5	660	0,00029221
8	478	330	0,007838832	HI-LM	18	330	0,00029519

Calculation of Z, system at partial load

The Z-coefficient characteristic of the new working situation, can be calculated applying the laws of circuits in series and in parallel. The same rules enabled us to determine these values for the balanced as well as for the unbalanced circuit.

The following example shows the various calculating operations in case of partial load. In circuits where the flow rate is reduced to 0, $Z = \infty$ (infinite).

Tab. 2

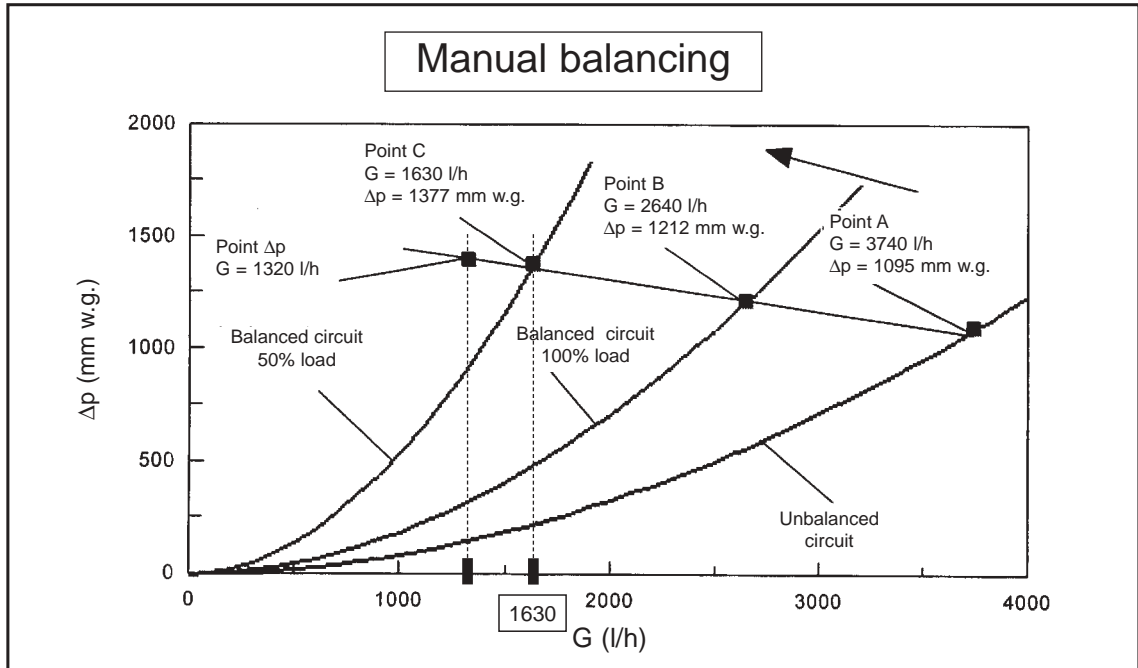
LINE	COMPOS.	CALCULATION	$1/Z^{0.525}$	Z·(Composed)
6				0.0106103
FG-6-ON	Series	$Z+ZFG+ZON=.0106103+2 \cdot .00011185$		0.010834
EF-OP	Series	$Z+ZEF+ZOP=.01083400+2 \cdot .00003061$		0.001089522
4	Parallel	$1/Z^{0.525}+1/Z4^{0.525}=1/0.01089522^{0.525}+1/0.0132675^{0.525}$	20.398952	0.003202645
DE-PQ	Series	$Z+ZDE+ZPQ=.003202645+2 \cdot .00002504$		0.003252725
CD-QR	Series	$Z+ZCD+ZQR=.003252725+2 \cdot .00002588$		0.003304485
2	Parallel	$1/Z^{0.525}+1/Z2^{0.525}=1/.003304485^{0.525}+1/.015891^{0.525}$	28.86468	0.00165329
BC-RS	Series	$Z+ZBC+ZRS=.00165329+2 \cdot .00002561$		0.00170451
1	Parallel	$1/Z^{0.525}+1/Z1^{0.525}=1/.00170451^{0.525}+1/.017957^{0.525}$	36.657465	0.00104868
AB-TS	Series	$Z+ZAB+ZTS=.00104868+2 \cdot .00001845$		0.00108558
			ZTOT	0.00108558

New operating point

Knowing the Z-coefficient characteristic of the new situation, the correspondent curve can be plotted.

Operating points: total load, partial load.

Fig. 1



The intersection point with the curve of the pump is the new balancing point C, at:

$$G = 1630 \text{ l/h}$$

$$\Delta p = 1377 \text{ mm w.g.}$$

Calculation of the flow rates

Knowing the total flow rate and the Z-values of each line, the new flow rates for each branch and terminal can be calculated.

For every terminal the formula in Tab. 3 applies.

Tab. 3

FORMULA
$G_i = G \cdot (Z/Z_i)^{0.525}$

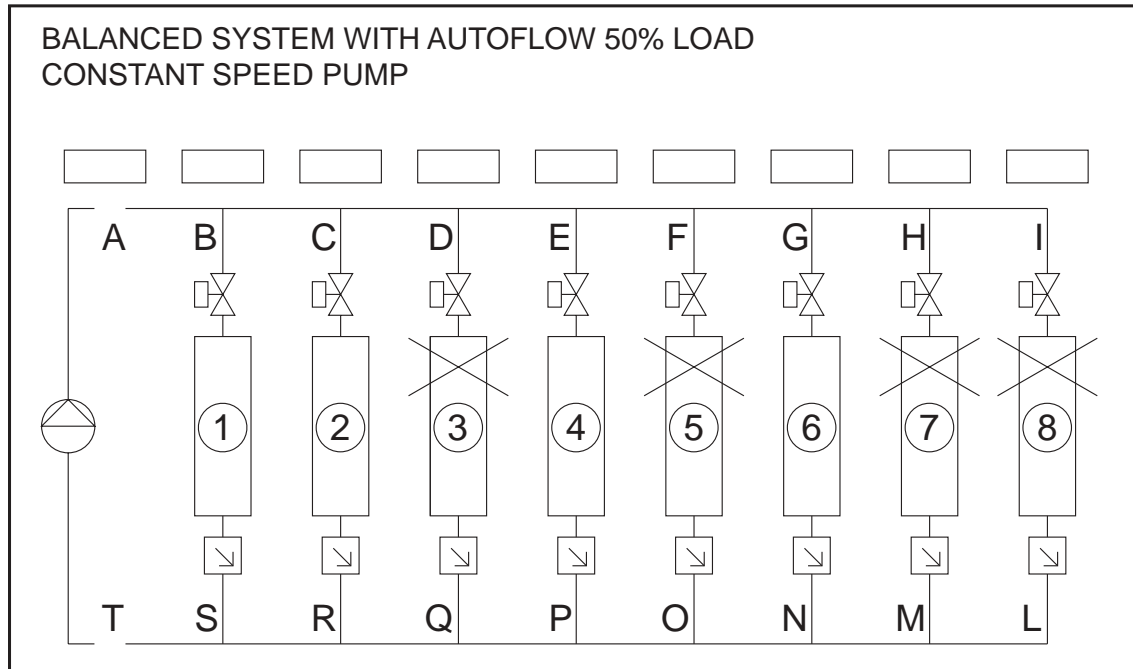
TERMINAL	CALCULATION	G (l/h)
1	$1630 \cdot (.00104868 / .017957)^{0.525}$	367
2	$1263 \cdot (.00165329 / .015891)^{0.525}$	384
4	$879 \cdot (.003202645 / .013267)^{0.525}$	417
6	879-417	462

The new values of the flow rates are represented in the scheme of the circuit.

Balanced circuit with Autoflow at partial load

The next scheme represents what happens when the system works at partial load. The control valves close the same terminals as in the previous example.

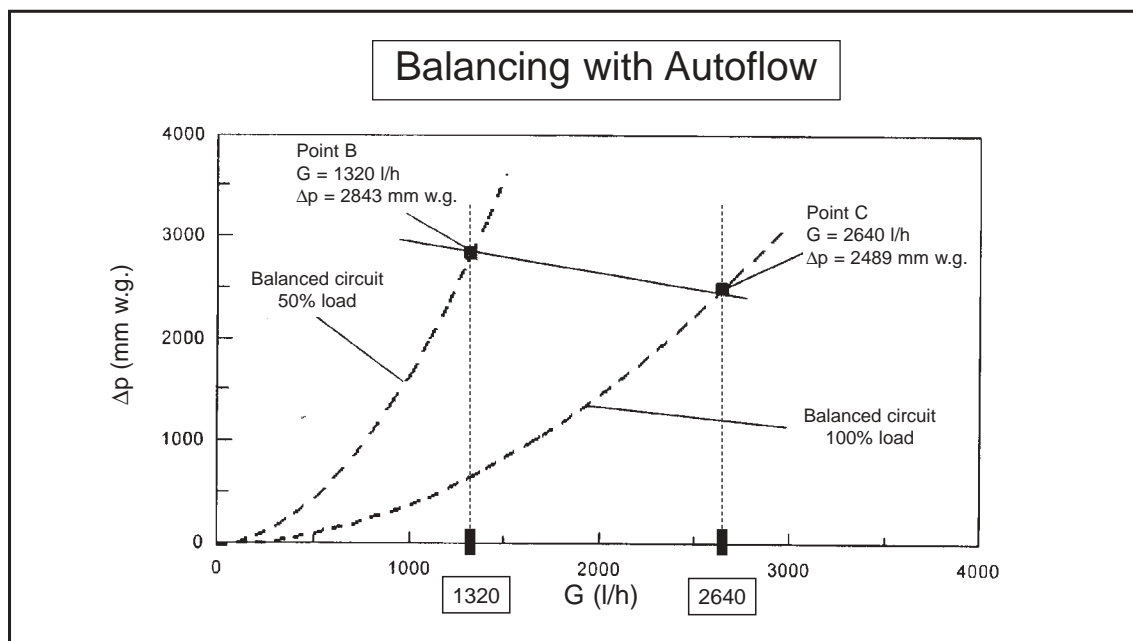
Scheme 2



Even in this case a graph represents the various operating points. Obviously the characteristic curve of the pump in this example is different from the last case, because the nominal operating points $\Delta p - G$ are different.

Operating points of the circuit with Autoflow: total load, partial load

Fig. 2



Carrying out the following calculations makes it possible to determine **the overpressure, which must be absorbed by Autoflow to bring the system back to its nominal design conditions.**

Differing from the case with a balanced circuit with manual valves at partial load the dynamic nature of the regulator does not allow simple considerations on the Z-values.

It allows however to obtain the exact value of the flow rate corresponding to the required partial load. Knowing this, the new operating point on the curve of the pump can be identified and the new differential pressure values of every circuit calculated.

Autoflow overpressures

Tab. 4

Calculation of Autoflow overpressures

$\Delta p_{AT}=2843$	
$\Delta p_1=\Delta p_{AT}-2\cdot\Delta p_{AB}=\Delta p_{AT}-2\cdot Z_{AB}\cdot G^{1.9}=2843-2\cdot 15.7=$	2811
$\Delta p_2=\Delta p_1-2\cdot\Delta p_{BC}=2811-2\cdot 12.60=$	2786
$\Delta p_4=\Delta p_2-2\cdot\Delta p_{CD}-2\cdot\Delta p_{DE}=2786-2\cdot 5.9-2\cdot 5.7=$	2763
$\Delta p_6=\Delta p_4-2\cdot\Delta p_{EF}-2\cdot\Delta p_{FG}=2763-2\cdot 1.9-2\cdot 6.8=$	2746

Autoflow Δp

Tab. 5

TERMINAL	Δp circuit (mm w.g.)	Δp branch (mm w.g.)	Δp AF (mm w.g.)	Δp AF (100%) (mm w.g.)	Δp AF extra (mm w.g.)
1	2811	328	2483	2044	439
2	2786	328	2458	1918	540
4	2763	328	2435	1758	677
6	2746	328	2418	1608	810

The results obtained from the calculation are reported in the scheme of the circuit.

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