DIFFERENTIAL PRESSURE CONTROLLER MODEL AND ITS APPLICATION IN DIFFERENT HEATING SYSTEMS

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Abstract

Variable flow distribution systems are promoted in complex heating and cooling systems as energy efficient solutions for new systems and as an upgrade of the existing constant distribution flow systems. In the case of variable flow, the pumping energy costs decreases and the preferable return temperature can be achieved which is beneficial for operation of the condensing boilers or district heating connection. As a result of different demand in heating/cooling zones, the pressure variations in the system can lead to hydraulic problems of single zones/consumers, which at the end result in control stability problems and decrease of thermal comfort for end users. High differential pressures at zones connections can in case of mixing circuits lead to back flows in the bypass ports of the three-way valves or backflow at the injection circuits with two port valves are used, increasing the return temperature and preventing the work of control valves in the full stroke. In these cases, the differential pressure controllers are necessary in order to stabilize the strong variation of differential pressure by preserving the high value of the regulating valve authority which guaranties the stable control over the entire stroke of the control valves, very close to the nominal conditions for which they are designed. This paper presents the simple algebraic model of the differential pressure controller as part of the hydraulic system and demonstrates the use of the proposed model at four characteristic examples applied at three different systems: Fixed flow radiators system, variable flow for injection circuit consumer connection and variable flow connection of indirect heating substation. The results of calculations demonstrate the applicability of the developed model for calculation of more complex cases.

Keywords: variable flow; differential pressure controllers; bivalent temperature; seasonal coefficient of performance

1. Introduction

In many practical situations high differential pressures are developed in the distribution systems which in turn affects the optimal work of control valves, expensive modulating PI controllers starts to work in introduces the ON/OFF mode and the benefits of modulating control are lost. This usually results with an increased energy consumption and negatively influence the thermal comfort of consumers.

Fig. 1. Relative proportional band (left) and minimum controllable flow (right), both as a function of valve authority [1]

Figure 1 (left) shows the relation between the control valve authority and relative proportional band of the controller, while Figure 1(right) provides the dependency of minimum controllable output of terminal as a function of control valve authority. It can be seen that the minimum controllable relative output is 32% for the valve authority of 0,1. It means that for the loads below 32% the controller works in ON/OFF mode and is not modulating. The final result is the temperature swings at consumer taking into consideration that proportional band has to be multiplied by 3 for this authority (Figure 1 left).

It means that if the proportional band is 2°C for control valve authority of 1, it should be 6°C for an authority of 0,1! One can suppose that the I function of the controller will compensate this deviation by slowly resetting the proportional controller but an integral function can only reduce such a deviation by half in 30 minutes [1]. This is usually the time during which the consumer will change the set point at the room or home controller and this may result in control loop instability and hunting the set point and increased energy consumption by overheating of the room/home.

Even if the control valve has been chosen properly with relatively high authority for design conditions, due to high variations of differential pressures the authority decreases dramatically and the described situation can happen. These variations of differential pressures are present in variable flow distribution networks, such as district heating or at larger buildings with multi-zone consumers. According to current literature review and recommendations for energy efficient design of HVAC systems, the variable flow distribution system is preferred.

In district heating systems and indirect connections of consumers, the distribution network is hydraulically decoupled from the consumer network via heat exchanger and differential pressure controllers are must-to-have elements of the substation connections. In that light, the simple algebraic model of the differential pressure controller should support the design engineers to analyse and optimize control strategies in order to decrease energy losses. These interventions and optimization can be the variation of the primary supply temperature, installation and control of variable speed drive pumps and installation of differential pressure controllers or any combination of these. One of the outcomes of these analysis should be the work of control valves in their full stroke, the possibility to control low loads with modulation and not ON/OFF, while providing the thermal comfort at consumers. Numerous studies have been carried out on improving the efficiency of heating systems that are focused on various heating elements, but control strategy was rarely analysed. Some of the suggestions for improved control strategy in case of decentralized heat supply are given in [2]. Also, operation and testing of the Danfoss pressure controller with integrated anti-oscillation system was done on district heating substation and results are presented in [3]. Results shows that developed system successfully removes oscillations in hydraulic system and automatically adjusts the pressure according to the system load.

In heating systems with variable flows in the distribution network, the connection to consumer can be made by installing two port valves with the bypass which provides the constant flow in consumer circuit or the three way – usually mixing valves. In both these cases it is supposed that the consumer requires the constant flow in the secondary network. In the first case, an increased differential pressure at the connecting point can result in reverse flow in the bypass, while in case of connection with mixing valve the mixing function of the three-way valve can be cancelled and the valve should start working as diverting valve, providing the counter flow at the bypass port of the three-way valve. In both cases the return water temperature will increase and in the second case the control of the consumer thermal comfort can become instable. Increased water return temperature will decrease the efficiency of condensing boiler and will result in increased energy consumption. In the second case, if the consumer with constant water flow is controlled with weather compensation control the supply temperature would be hard to achieve, since the mixing function is excluded.

Optimising the hydronic distribution of an existing system and thus reducing energy consumption can be achieved by combining a series of measures that bring small improvements one by one. Some of these measures were considered in [4]. Investigation conducted in [5] was used to analyze possibility for achieving hydraulic balance in building distribution systems, and consequently save energy and reduce the emissions by using pre-set radiator valves combined with differential pressure (DP) controllers. For this purpose, authors developed an integrated model for the simulation of the heat performance and hydraulics and applied it for pressure and flow conditions analysis of a two-pipe multistorey building heating system in China. Also, two solutions which involve the use of automatic differential pressure controller or proportional relief valves in order to avoid strong variation of differential pressure with load change are presented in [6].

This paper is designed to support the design engineers to understand and use this simple model as part of their calculation tool of the whole network. Equations which will be outlined in the next section are valid irrespective of the nominal pressure values and the model can be applied for one zone differential pressure controller or can be used as the main differential pressure controller positioned at the primary side of the large capacity heating substation in the district heating.

2. Mathematical model

In this subsection a mathematical model for the problem is outlined as well as the producer's data on available differential controller properties.

Fig. 2. Position of differential pressure controller in radiator heating system (left) and cross-section view at the valve body (right) [7]

Figure 1 (left) shows the differential pressure controller in the return pipe which upper part is connected to the supply pipe with capillary tube. Its main task is to keep constant differential pressure ($P1/P2$). At the right-hand side of this Figure the cross section of the valve body is presented. Valve seat position is determined by the resulting force coming from the spring and pressure difference at the membrane which separates the upper and lower parts of the valve at which corresponding pressure acts. In case that pressure $P1$ in the supply pipe increases, the resulting force, due to higher pressure at the upper side of the membrane, moves the valve seat downwards. As the valve closes, the pressure $P2$ increases and the valve seat find the new equilibrium position at which again the forces are in balance.

If pressure $P1$ drops, the force from the lower side of the membrane prevails and moves valve seat upwards, decreasing the pressure $P2$ and again establish new equilibrium position determined as new balance of the resulting force in the valve. The process is continuous, there is no additional energy provided to move the valve seat, everything is done by the resulting force coming from the spring and pressure difference between supply and return. Desired differential pressure value which needs to be maintained constant is determined by the choice of appropriate valve and the neutral position of the spring.

Fig. 3. System without the circulation pump (left) and typical connection with two/port valve and circulation pump (right) with positions of the differential pressure controllers for both cases

Figure 3 shows two characteristic connections of the consumers. Figure 3 (left) describes the connection without additional circulation pump in which the primary differential pressure is responsible for the water circulation towards consumer. Differential pressure controller (DPC) is positioned in the return pipe and is via measuring balancing valve (BVM) connected to primary side via capillary pipe. It is the DPC task to keep the constant differential pressure between points C and D or to keep the Δp_{sys} at the constant value. In Figure 3 (right), typical connection with two-port valve of consumer with constant flow is presented. In this case, DPC maintains the constant differential pressure and the control valve (CV) is working with practically constant authority.

Figure 4 presents the typical connection of consumer in district heating systems. This is indirect connection since the primary from secondary circuit is separated via heat exchanger. Control valve (CV) controls the primary flow and match the desired temperature in the secondary supply. It is in most cases weather compensated control [8]. Because of the strong pressure variations in the district heating network the differential pressure controller is positioned before control valve as shown in Figure 4.

Fig. 4. Indirect connection of consumer in district heating where heat exchanger separates primary from secondary circuit

We will first focus on the system shown in Figure 3 (left) but before we proceed, we will adopt the following assumptions, valid for all cases:

- It is assumed that the local hydraulic resistances prevail between points C and D in Figure 3. It means that the assumption is to keep the constant value of the friction factor in the pipeline. With the change of flow the friction factor will be assumed constant,
- Differential pressure controller will be approximated as the proportional controller with linear relationship between the pressure differential and the relative position of the valve (Y),
- Differential pressure valve is treated as the linear valve.

All three given assumptions are reasonable from the engineering side and introduce simplifications in the mathematical model which will be presented. First the corresponding equations and the problem formulation for the system given in figure 3 (left) will be given. The available differential pressure Δp_{AB} should be equal to sum of hydraulic resistances put in series.

$$\Delta p_{AB} = \Delta p_{BVM} + \Delta p_{sys} + \Delta p_{DPC} = \left(\frac{\dot{V}}{K_{v}^{BVM}} \right)^2 + \left(\frac{\dot{V}}{K_{v}^{SYS}} \right)^2 + \left(\frac{\dot{V}}{Y K_{v}^{DPC}} \right)^2 \quad (1)$$

Where Δp_{AB} is the differential pressure between point A and B, Δp_{BVM} is the pressure drop of the measuring balancing valve, Δp_{DPC} is the pressure drop of the differential pressure valve. Note that on the right-hand side of equation (1) the pressure drops are expressed in the form dependent on the volume flow $\dot{V}$ and K_v values of the measuring balancing valve with index BVM, system SYS and differential pressure valve DPC, where Y denotes valve relative position, $Y = 1$ (valve fully open) and $Y = 0$ (valve fully closed). Volume flow in equation (1) is given in m^3/h , while the pressure drop is given in bar. It is simple to convert the equation in other units, for example if the pressure drop is to be expressed in kPa than every expression in the brackets on the right-hand side of equation should be multiplied by 100.

Fig. 5. Graphical dependency of the position of differential pressure valve on the system differential pressure Δp_{sys}

With the presented assumptions listed above, it is possible to introduce the K_v^{SYS} value of the system, although this is not the usual way how to present the pressure drop at the consumer side. However, the fact that we assumed that our consumer or the system behaves as the fixed hydraulic resistance, our approach is justified. Equation (1) cannot be solved since we have two unknown variables, the volume flow $\dot{V}$ and the position of the DPC valve Y. In order to close the system, we will introduce the relationship which links the position of the DPC valve with the system differential pressure Δp_{sys} . The assumed relationship is presented graphically as shown in Figure 5.

In Figure 5 a graphical representation of the position of the DPC valve on Δp_{sys} is given. One can note three characteristic zones. In the Zone 1 (Z1) the differential pressure Δp_{sys} is lower than the nominal value Δp_{sys}^N . As it can be seen in this zone the DPC valve is fully open ($Y = 1$) and it is expected that the nominal flow in the system cannot be achieved. In the Zone 3 (Z3) the DPC valve is in its minimal position defined as $Y = Y_{min}$ and it does not depend on increase of the system differential pressure. Zone 2 is the most interesting for us, since it is expected that the differential pressure controller will operate in this zone for the most of the time during the system operation.

It is obvious that the linear dependency of the system differential pressure is defined. In Figure 5 the width of the zone 2 is the proportional band of the controller, while the value $x\Delta p_{sys}^N$ is the maximum value of the system differential pressure for which the DPC valve comes into the minimum position (Y_{min}). Note that the value of this differential pressure is given in relative numbers while it is obvious that $x > 1$. Since the DPC valve is proportional controller, it cannot provide the fix value of the system differential pressure value. Usually, relative variation of controlled differential pressure varies within $\pm 25\%$ which is acceptable. Equation of $Y = f(\Delta p_{sys})$ within zone 2 is given as follows:

$$Y - 1 = \frac{(Y_{min} - 1)}{(x - 1)\Delta p_{sys}^N} (\Delta p_{sys} - \Delta p_{sys}^N) \quad (2)$$

Where we assumed that the fully open DPC corresponds to the nominal differential pressure Δp_{sys}^N .

We will for the simplicity of presentation express the following:

$$\alpha_{DPC} = \frac{(1 - Y_{min})}{(x - 1)} \quad (3)$$

Where α_{DPC} constant is introduced, which is having in mind our assumption always positive value. If this simplification is introduced into equation (2) together with expressing Δp_{sys} via volume flow $\dot{V}$ it becomes:

$$Y = (\alpha_{DPC} + 1) - \frac{\alpha_{DPC}}{\Delta p_{sys}^N} \left(\frac{\dot{V}}{K_v^{SYS}} \right)^2 \quad (4)$$

Which now defines relative position of the DPC valve only in terms of the volume flow.

Once the nominal conditions are known, balance valve is set to provide the nominal flow in the system with fully open and set differential pressure control valve, we can define final form of the equation which will be solved in the form of Δe_{err} , so that:

$$\Delta e_{err} = \Delta p_{AB} - \left[\left(\frac{\dot{V}}{K_v^{BVM}} \right)^2 + \left(\frac{\dot{V}}{K_v^{SYS}} \right)^2 + \left(\frac{\dot{V}}{\boxed{Y} K_v^{DPC}} \right)^2 \right] \rightarrow 0 \quad (5)$$

Where $\boxed{Y}$ denotes the value of relative position of the DPC valve which is defined by equation (3). It can be noted that the equation (5) cannot be solved explicitly and iterative solution is necessary.

Calculation methodology applicable for the system presented in Figure 3 (left) is as follows:

- Based on known nominal conditions calculate values K_v^{BVM} , K_v^{SYS} and take K_v^{DPC} value from the supplier catalogue,
- Expose the system to differential pressure Δp_{AB} and solve equation (5) iteratively until the prescribed criterion is met – for example $\Delta e_{err} < 1 \cdot e^{-03}$,
- Calculate the relative position of DPC valve (Y) from equation (4) and calculate pressure drops of each component Δp_{BVM} , Δp_{sys} and Δp_{DPC} .

Please note that presented model does not take into account the dynamics of the DPC valve, it is rather model which can cover quasi-steady state calculations. It means that it is assumed that the under new imposed conditions (change of Δp_{AB}) system has enough time to reach new steady state condition. Proposed calculation methodology can be applied to practical situations where the designer knows all parameters in nominal conditions and is interested what happens to the system when it is exposed to change in differential pressure at its connection.

Let us now analyse second system which is more applicable for heating systems and connections of indirect heating substations in district heating. In this case it is assumed that the consumer water flow is constant and that weather compensated control is applied in which the control variable is supply temperature in the secondary circuit (see Figure 3 right). Desired supply temperature is dependent on the outside temperature and settings at the controller. Required supply temperature in the secondary circuit is achieved by action of the two-port valve, which determines the mixing ratio between the supply hot water and return cold water. The mixing point is in point I in Figure 3 (right) and the temperature sensor is placed after point I towards the consumer. As the outside temperature increases, less flow from the primary side is necessary to achieve desired supply temperature and control valve closes. With closing of the control valve, pressure at point C increases and resulting differential pressure measured between points C and D increases as well. At this stage, differential pressure valve, due to forces disbalance, acts and closes DPC valve trying to keep the differential pressure at the set value.

In case when the outside temperature decreases, control valve opens and pressure at point C decreases, the resulting force within the DPC valve changes the valve seat position and opens the valve until new equilibrium position is found. In terms of the definition of the problem, we here face a bit reverse problem.

So, even if the connection differential pressure is kept constant at the nominal value, as a result of change in the valve relative position, the change in differential pressure in the system is expected. The task of the differential pressure controller is to react on the control valve action and to provide the nearly constant differential pressure keeping the authority of the control valve close to its nominal value. Also, in terms of the definition of the engineering problem here, we have a different formulation. Namely, since the weather compensated control is in most cases applied as a control strategy in the secondary circuit, we know the volume flow both from primary supply line via two-port valve and from the circuit return. In order to keep the problem strictly on hydraulics here, we may confirm that the volume flow from the primary side is known.

Although, the system shown in Figure 3 (right) seems to be more complex because of the existence of the control valve which is modulating based on demand, it can be shown that the calculation procedure is even simpler in this case. Namely, since the pressure drop in the bypass between points I and II is negligible ($p_I \cong p_{II}$, or $\Delta p_{I-II} \cong 0$) we can confirm that the pressure drop at the control valve is equal to the Δp_{sys} . When this connection is designed, it is required that the nominal pressure drop of the control valve (fully open) compensates the available differential pressure at the connection subtracted by the pressure drop at the differential pressure controller and the pressure drop at the measuring balancing valve. If we put this in the form of equation, it is given as follows:

$$\Delta p_{AB} = \left(\frac{\dot{V}}{K_V^{BVM}} \right)^2 + \left[\left(\frac{\dot{V}}{Y_{CV} K_{VS}^{CV}} \right)^2 \right] + \left(\frac{\dot{V}}{Y K_{VS}^{DPC}} \right)^2 \quad (6)$$

Where the second term at the right-hand side of equation is practically the Δp_{sys} and contains flow coefficient of the control valve and its relative position as Y_{CV} . Note that in the nominal condition volume flow becomes nominal flow and the relative position of the control valve is $Y_{CV} = 1$.

Again, we have two unknown variables, relative position of the differential pressure control valve Y and relative position of the control valve Y_{CV} . However, we propose decoupling of the differential pressure controller and control valve in a way that the term presented in box in equation (6) is defined as Δp_{sys} which, according to our assumptions and discussion above, can be expressed as:

$$\Delta p_{sys} = \Delta p_{AB} - \Delta p_{BVM} - \Delta p_{DPC} = \Delta p_{AB} - \left(\frac{\dot{V}}{K_V^{BVM}} \right)^2 - \left(\frac{\dot{V}}{Y K_{VS}^{DPC}} \right)^2 \quad (7)$$

Note that we again have two unknown values but we can use already defined relation for differential pressure controller in equations (2) and (4). By combining equations (2, 4 and 6) we can reformulate our expression for relative position of the differential pressure controller and write it in the form of error function such as:

$$\Delta e_{gr} = Y - \left\{ (\alpha_{DPC} + 1) - \frac{\alpha_{DPC}}{\Delta p_{sys}^N} \left[\Delta p_{AB} - 100 \left(\frac{\dot{V}}{K_V^{BVM}} \right)^2 - 100 \left(\frac{\dot{V}}{Y K_{VS}^{DPC}} \right)^2 \right] \right\} \rightarrow 0 \quad (8)$$

Where we eliminated one unknown variable, wrote the equation in terms of the relative position of the differential pressure control valve Y (second term at the right-hand side of equation (6) and, since we again have equation in the form which cannot be explicitly expressed, we put it in the form of the error function. The solution is again to be sought iteratively until the prescribed tolerance is met. Equation (6) presents the key point for this system. At the end, we can calculate also the relative position of the control valve as:

$$Y_{CV} = \frac{1}{\sqrt{\Delta p_{CV}}} \left(\frac{\dot{V}}{K_{VS}^{CV}} \right) \quad (9)$$

Where volume flow is given in m³/h and pressure drop in bar and Δp_{CV} is the pressure drop of the control valve. Please note that it has been already said that the $\Delta p_{CV} = \Delta p_{sys}$ and this is valid for the system in Figure 3 (right).

If we take a look at the indirect heating substation and the problem formulation in this case, it is obvious that the model presented for system in Figure 3 (right) is applicable in this case as well. The only difference between these two systems is that system differential pressure Δp_{sys} in case of system given on Figure 4 can be calculated as:

$$\Delta p_{sys} = \Delta p_{HEX} + \Delta p_{CV} = \left(\frac{\dot{V}}{K_V^{HEX}} \right)^2 + \left(\frac{\dot{V}}{Y_{CV} K_{VS}^{CV}} \right)^2 \quad (10)$$

Where Δp_{HEX} is the pressure drop of the heat exchanger and since it is treated as the local resistance in which pressure drop depends on the square of the volume flow and can be written via flow coefficient K_V^{HEX} .

Calculation methodology applicable for the system presented in Figure 3 (right) and system in Figure 4 is as follows:

- Based on known nominal conditions calculate value K_v^{BVM} , calculate Δp_{sys}^N (based on K_{vs}^{CV} and nominal flow) take K_{vs}^{DPC} value from the supplier catalogue, and in case of system in Figure 4 calculate K_v^{HEX}
- Expose the system to differential pressure Δp_{AB} and solve equation (8) iteratively by varying the Y until the prescribed criterion is met – for example $\Delta e_{err} < 1 \cdot e^{-06}$,
- Calculate the pressure drops of each component Δp_{BVM} (already known by the definition of the problem), Δp_{DPC} and Δp_{sys} as well as the Δp_{HEX} (for system in Figure 4),
- Calculate relative position of the control valve from equation (9) and for system in Figure 4 use equation (10) to get the Δp_{CV} .

Note that equation (8) poses no restriction in case of changing differential pressure Δp_{AB} and offers the possibility to take into account the simultaneous change in volume flow, as well as the change in differential pressure Δp_{AB} .

3. Results and discussion

For the validation of proposed approach four calculated cases are presented to demonstrate the applicability of the proposed methodology. First and second cases are related to system shown in Figure 3 (left). In the first case, the system is exposed to linear variation of the connection differential pressure and the volume flow and system differential pressure is calculated. Second case demonstrates the behaviour of the differential pressure controller when the system is exposed to the arbitrary variation of connection differential pressure. Third case is related to the system described in Figure 3 (right) where the periodical variation of the connection differential pressure and gradual change in flow are simultaneously applied.

The basis of the fourth case are measurements and results published in HERZ material [9] in which the indirect heating substation was taken (Figure 4). The connection differential pressure was kept constant while the three cycles of opening and closing the control valve are applied and the system differential pressure is measured. In this case, since we did not have data on exact measurements setup, an improvisation is used to check the applicability of our methodology. Even so, the hydraulic parameters can be clearly seen from the presented graph and our results correspond with the published data on measurements.

3.1. Linear increase of connection differential pressure

In this case the system in Figure 3 (left) is taken. The system is exposed to the linear increase of the connection differential pressure starting from 50 kPa up to 150 kPa, while the nominal connection differential pressure is 50 kPa. The following numbers were input for calculation:

$\Delta p_{AB}^N = 50 \text{ kPa}$, $\Delta p_{BVM}^N = 5 \text{ kPa}$, $\Delta p_{sys}^N = 20 \text{ kPa}$, $\Delta p_{DPC}^N = 25 \text{ kPa}$, nominal volume flow rate is $\dot{V} = 1,786 \frac{\text{m}^3}{\text{h}}$.

For the illustration purposes this system would be equivalent to the consumer with 40 kW in nominal conditions. In this case we will assume that the parameter $\alpha_{DPC} = 3$. Now, it is very important to demonstrate the ratios of pressure drops for individual components of the system.

Fig. 6. Contribution of pressure drop for each component of the system in Figure 3 (left) when the system is exposed to linear increase of the connection differential pressure Δp_{AB} from 50 kPa to 150 kPa.

Figure 6 shows the contribution of the pressure drops of the DPC valve (area in orange colour), BMV valve (area in blue colour) and available Δp_{sys} (area in green colour). Note that the increase of the connection differential pressure is given in black line at the top of the diagram. It can be seen that DPC valve compensates the increase of the connection differential pressure, while preserving the system differential pressure almost at constant level.

At the left-hand side of the diagram given in Figure 6 the numerical nominal values of pressure drops are given while at the right-hand side the final values of all components are presented when the system is exposed to $\Delta p_{AB} = 150$ kPa.

Final remark is that the increase of the connection differential pressure of 100 kPa, over the nominal value resulted in increase of the system differential pressure in 3,4 kPa. In other words, due to the DPC valve action the disturbance has been reduced 30 times! This is exactly was is expected from the differential pressure controller.

3.2. Arbitrary variation of connection differential pressure

In this case, the same system with identical characteristic values is used and an arbitrary variation of the connection differential pressure is applied. The goal is to proof that the variations of the system differential pressures will remain within prescribed tolerances.

Fig. 7. Arbitrary variation of connection differential pressure Δp_{AB} (blue line) and corresponding system differential pressure Δp_{sys} (red line) in time

Figure 7 shows an arbitrary variation of connection differential pressure Δp_{AB} (blue line) and corresponding system differential pressure Δp_{sys} in time. Note that although the time is one of the axes, the dynamic behaviour of the differential pressure controller is not taken into account and the results are presented as the quasi-steady-state solutions. As it can be seen the variation of the connection differential pressure is 100 kPa while the connection differential pressure never falls below nominal connection differential pressure. In this case the maximum amplitude of the system differential pressure is 3,4 kPa as shown in diagram in Figure 7.

3.3. Variation of connection differential pressure and step change in flow

This case is applied to the system shown in Figure 3 (right). In order to test the proposed calculation methodology simultaneous change in connection differential pressure and change in volume flow is applied. Sinusoidal variation of the connection differential pressure is generated, while the step change in the relative volume flow is applied. The flow is changed from the starting 10% , with increment of 10% and ends with 100% value or the nominal flow. The following data are used for the calculations:

$\Delta p_{AB}^N = 50$ kPa, $\Delta p_{BVM}^N = 5$ kPa, $\Delta p_{sys}^N = 20$ kPa, $\Delta p_{DPC}^N = 25$ kPa, nominal volume flow rate is $\dot{V} = 1,786 \frac{m^3}{h}$. Here we also supposed that minimal relative position of the differential pressure control valve is $Y_{min} = 0,045$, which corresponds to the maximal differential pressure $x\Delta p_{sys}^N = 1,32 \Delta p_{sys}^N = 26,4$ kPa. So, the proportional band for DPC valve is in this case is 32% (see Figure 5). In this case, we said that pressure drop at the fully open control valve should be equal to the available nominal differential pressure $\Delta p_{sys}^N = \Delta p_{CV}^N = 20$ kPa.

If we now apply preliminary analysis, it is obvious that the authority of the control valve in nominal conditions is 1. With maximal differential pressure (26,4 kPa) the authority will change to 0,76, which is again very good. At contrary, if we assume that no differential pressure control is installed, the authority in nominal conditions would be 0,4 and when system is exposed to maximal connection differential pressure of 150 kPa, an authority would decrease to only 0,13 which brings the problems elaborated in the introductory part of this paper.

Figure 8 presents results of calculation for applied boundary conditions for the system given in Figure 3 (right). It is visible that despite the variation of the connection differential pressure and step change in the volume flow presented in red line and given at the right-hand side vertical axes, the system differential pressure remains almost constant and that the differential pressure control valve takes over almost all excessive pressure.

Fig. 8. Sinusoidal variation of connection differential pressure (Δp_{AB}), step change in relative flow (red line) and contributions of pressure drops for Δp_{DPC} , Δp_{sys} (or Δp_{CV}) and Δp_{BVM}

It should be said that the nominal conditions have never been met in this example. Namely, when the connection differential pressure drops to the nominal value, the relative flow through the primary side is at the level of 80% and also when the flow is nominal which corresponds to the last point in diagram in Figure 8, the connection differential pressure is at 100 kPa.

One can also note that the blue area which presents the pressure drop coming from the balance measuring valve is almost invisible until the relative flow exceeds 50% which is reasonable, since the balance valve is local resistance and its pressure drop is proportional to the square of the flow. In cases where flow is small the pressure drop at this valve is almost negligible.

3.4. Indirect heating substation with differential pressure controller

This case was chosen to show the applicability of the developed calculation methodology to treat the stabilisation of the differential pressure in indirect connection in district heating application. Results of measurements for comparison purposes are taken from the published material [9]. Although exact experimental setup was not available, based on the presented results the case was reconstructed and calculated.

This case deals with the system in Figure 4, where differential pressure controller stabilises the differential pressure for the control valve which is in series placed in the return pipe after the heat exchanger. In this case the connection differential pressure was kept constant during the measurements and the control valve controlled the flow through the heat exchanger in three full cycles from the fully open to the almost closed position. As a result of changes in system differential pressure, the differential pressure control valve reacted and was trying to keep the differential pressure at the almost constant value. The following data are used for the calculations:

$\Delta p_{AB}^N = 250$ kPa, $\Delta p_{BVM}^N = 35$ kPa, $\Delta p_{DPC}^N = 25$ kPa, $\Delta p_{sys}^N = 19$ kPa, $\Delta p_{CV}^N = 12,25$ kPa, $\Delta p_{HEX}^N = 6,75$ kPa, nominal volume flow rate is $\dot{V} = 3500 \frac{m^3}{h} = 3,5 \frac{m^3}{h}$.

Fig. 9. Volume flow, connection differential pressure and system differential pressure / measured (top left) [9], calculated (bottom left) and comparison measured vs. calculated (right)

It is estimated that based on the assumed nominal temperature difference at the primary side of $\Delta t = 55^\circ\text{C}$, (110/55) the heat exchanger capacity was close to 230 kW, and the pressure drop for the heat exchanger together with connections was estimated by the available producer's data. Control valve and the DPC valves are also chosen according to the market available K_{vs} values. In our case, the volume flow was read from the diagram and this volume flow was used to solve the equations (8, 9 and 10) and to determine the system differential pressure for every given volume flow.

Also, based on our estimation maximum differential pressure for the almost closed DPC valve ($Y_{min} = 0,05$) is $x\Delta p_{sys}^N = 1,4 \Delta p_{sys}^N$.

Figure 9 (left) shows the results of measurements (up) and results of calculations (down) and diagrams one over the other (right). Variation of the flow (blue line) is given in litres per hour at the right-hand side vertical axes, while the differential pressure (red line) is given in kPa at vertical left-hand side axes. Please note that the system differential pressure is multiplied by 10 in order to make measured variable clearly visible on the graph. It is obvious that our estimation was good and the results in terms of the system differential pressure are very good. Discrepancies are visible only at the time intervals where the flow is changing from minimum to the maximum position or vice versa. This is expected since our proposed model do not treat the dynamics of the differential pressure controller than rather treat steady state conditions which means that DPC valve reacts instantly on any disturbance.

4. Conclusions and further work

Simplified algebraic model of the differential pressure controller is presented in this paper, which can be of use for design engineers in preliminary and design phases. This model can be incorporated into the modelling tools which then treat interaction of different components in the heating/cooling systems or indirect heating substations in district heating. It has been demonstrated on four examples that the proposed calculation methodology is useful for calculation of different properties such as individual pressure drops of different components in the heating system and relative positions of DPC and control valve.

Future work is focused on implementation of this model to Modelica Buildings library, which then allows analysis of different design concepts and include, but is not limited to, the variable speed drive pumps, advanced models of plate heat exchangers, different control strategies etc. Preliminary work has been already done and the results are promising.

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