CHALMERS

M.Sc. Thesis E2009: 02
(Examensarbete)

Valve influence on the power requirement of centralized pumps in hydronic heating systems

- Theoretical study and evaluation of MathModelica as a hydronic systems simulation tool

Mattias Gruber

Gothenburg
June 2009

CHALMERS UNIVERSITY OF TECHNOLOGY
DEPARTMENT OF ENERGY AND ENVIRONMENT
BUILDING SERVICES ENGINEERING, S-412 96 GOTHENBURG, SWEDEN

Abstract

This master thesis was performed in collaboration with the company Tour & Andersson AB (TA). The purpose was to evaluate if the systems modeling and simulation software MathModelica, can be used in the area of hydronic systems. The basis for the evaluation was a requirement specification, containing the following four requirements:

1. the software should find solutions to hydronic systems of relevant sizes,
2. the software should be able to be used to model both balancing and control valves as well as other common hydronic components
3. the software should be able to produce results which agrees to reality regarding systems behavior, and finally,
4. the software should be able to produce results which agrees to reality regarding numerical accuracy

The evaluation was performed by executing two individual studies, which together were of such comprehensiveness that all four requirements were represented. One of these studies was called the verification study, and was used to test the fourth requirement presented above. The verification study was performed by model a test rig located at TA and to compare measured and simulated data. The result showed that the difference between the simulated and measured values normally was about ±7 %, which was considered as acceptable. However, much larger errors were also encountered in one case. For that reason, this requirement was not considered as fulfilled.

The other study was called the reference study and was an evaluation of how balancing and control valves in hydronic heating systems affect the power consumption of an including centralized pump. This study was used to test requirement one to three, and it was executed by modeling a general hydronic distribution system, as well as balancing and control valves. By simulating this system without any valves, only containing balancing valves and finally also with control valves, the effect that the gradual introduction of valves had on the drive power consumption of the pump was shown. Furthermore was the study broaden by including three different types of pumps as well as including both steady and unsteady conditions for which the power consumption was observed.

The result showed that the power consumption of the pumps in all cases was decreased when valves were introduced. The primarily reason for this was that the flow level in the system was decreased enough to overcome any other negative effects. Hence, it was theoretically shown that the flowrate has the largest effect on the drive power consumption of a centralized pump. In conclusion, requirement three was considered as fulfilled, while requirements one and two were considered as fulfilled in most, cases but not in all, cases.

Keywords: MathModelica, systems modeling and simulation, hydronic heating systems, balancing, controlling, pumps

Foreword

This essay is the master thesis of Mattias Gruber at Chalmers University of technology (CTH). It is the final phase of the master program called Sustainable energy systems and was performed at the faculty of energy and environment, department of building services engineering technology at CTH. The project, which is described in the thesis, was formulated by the development division of the company Tour & Andersson AB in Ljung, Sweden.

This project was executed in two steps, the first one during 10 weeks in the summer of 2008, and the second one over the spring period of 2009, starting the 13th of January and ending the 10th of June.

At Tour & Andersson, I especially like to thank development manager Nils Hjelte, who gave me the opportunity to work with this project. I would also like to thank the concerned development engineers at TA, and especially my company tutor, Peter Persson, for all the priceless help and input they have given me.

At CTH, I would like to thank my examiner Professor Per Fahlén and the rest of the involved faculty staff, for taking their time to guide me through the thesis.

Finally, I also like to thank Andreas Idebrant and Karin Gustafsson at MathCore for their valuable consultancy regarding the coding procedure.

Göteborg
2009-05-08

Mattias Gruber

Contents

Page

Symbols, subscripts, superscripts, definitions and abbreviations

Symbols

Latin letters

A	*Area;* m^2
B	Damping constant; $N/(m/s)$
c	Speed of sound; m/s
c_p	*Specific heat capacity at constant pressure; J/kg/K or kJ/kg/K*
c_p	Specific heat capacity; water
D, d	*Diameter; m or mm*
D	Outside diameter of pipes
d	Inside diameter of pipes
E	*Percentage difference*
e	*error*
e	control error
F	*Force; N*
F	Hydraulic force
H	*Distance; mm*
H	Valve opening
J	*Moments of inertia;* $(Nm\cdot s^2)/rad$ or $(Nm\cdot s^2)/m$
J	Polar moments of inertia
K	*Static gain*
K_c	Static gain of controller
k	Flow resistance factor; $Pa/(m^3/s^2)^2$
k_s	Spring constant; N/m
Kv	Valve capacity
h	*Specific valve opening*
h	Specific valve opening, normalized by the maximum opening
L	*Length; m or mm*
L	Length of the pipe in the direction of flow
M	*Mass; kg*
M	Mass of valve cone

$\dot{M}$	*Mass flowrate; kg/s or kg/h*
$\dot{M}$	Mass flowrate of water
n	*Number*
n	Total number of measurements
$\dot{Q}$	*Thermal capacity; W or kW*
$\dot{Q}$	Heating capacity
r	set value of regulator
s_x	Standard uncertainty of variable x (type A component; standard deviation)
T	Thermodynamic (absolute) temperature; K,
T	Torque; Nm
U	*Uncertainty*
u	*Velocity; m/s*
u	Velocity of water in pipe
V	*Volume; m^3*
V	Internal volume of component
$\dot{V}$	*Volume flowrate; m^3/s or m^3/h*
$\dot{V}$	Volume flowrate; water
$\dot{W}$	*Power (mechanical or electric); W or kW*
$\dot{W}$	Electric power input to pump

Greek letters

β	Bulk modulus; N/m^2
Δp	*Pressure difference; Pa or kPa*
Δp	Pressure drop or rise over a component
η	*Efficiency; -*
η	Energy efficiency of pump
ν	*Viscosity (kinematic); m^2/s or mm^2/s*
ν	Viscosity of water
ρ	*Density; kg/m^3*
ρ	Density of water

τ	*Time; s, min or h*
τ_c	Time constant
ω	Angular velocity; rad/s

Subscripts and superscripts

c	Controller
design	Design condition, i.e. at full load
f	friction
i	Inlet
L	Load
m	Measured
o	Outlet
p	Pump
s	Simulated
tot	Total
v	Valve
0	Reference value

Definitions - Dimensionless numbers

Re_d Reynolds number for flow in circular tubes; $Re = \frac{u \cdot d}{\nu}$

Abbreviations

ABC	Automatic Balancing and Control
CTH	Chalmers University of Technology
DN	Nominal Diameter
DPC	Differential Pressure Control
TA	Tour & Andersson AB

1 Introduction

This essay is the master thesis of Mattias Gruber at Chalmers University of technology (CTH) and corresponds to 30 academic points. The project described in this thesis, was performed in collaboration with the company Tour & Andersson AB (TA), located in Ljung, Sweden.

1.1 Background

TA is a company which is specialized in the area of hydronic balancing, and they are both developing and manufacturing products related to this. Traditionally, the development work of TA has mostly focused on product knowledge. However, this means that the overall behavior of hydronic systems has been lost, and such perspective has now been identified as important by the management. The reason is that an overall picture is most often needed in order to reach deeper technological insights, which in turn can lead to the introduction of new thinking patterns in the work of development. This is desirable, especially since the energy sector will be facing many changes in the nearby future, which probably will result in many new challenges for the development department. For that reason, TA has a goal of reaching a higher degree of innovation in their work, and especially the development department is going through large changes. One of their short term goals is to be able to foresight how their products interact and behave in different systems, which partly is thought to be achieved by expanding the laboratorial activity and by introducing a systems simulation tool. One of their options, MathModelica, is an object-oriented simulation software developed by MathCore in Linköping, Sweden. MathModelica hadn’t at the initiation of this project been used for hydronic applications before. Hence, there was no reference material to show whether or not MathModelica would satisfy TA´s needs. However, an evaluation was considered as useful since MathModelica had been proven valuable in other areas of applications, such as electronics, mechatronics and controlling, which made TA interested to evaluate its potential.

1.2 Purpose

The purpose of this study was to perform an evaluation of MathModelica, to test if the features needed to serve as a useful tool in the development department of TA exists. The base for the evaluation was a requirement specification formulated by TA. It contained the requirements that a system-simulation software should fulfill in order to be considered as a useful development tool. The containing requirements were phrased as;

1. the software should find solutions to hydronic systems of relevant sizes,
2. the software should be able to be used to model both balancing and control valves as well as other common hydronic components
3. the software should be able to produce results which agrees to reality regarding systems behavior, and finally,

4. the software should be able to produce results which agrees to reality regarding numerical accuracy

1.3 Method

The method was to use MathModelica in the execution of two individual studies, called the reference and verification study, which together were of such comprehensiveness that all four requirements were represented. By doing so, it was shown whether or not the requirements were fulfilled, and hence if the features of MathModelica are as desired.

The reference study was primarily included to test requirement one to three as listed above. However, it was also included since TA, according to Persson[21], has plans to perform this study in the future, partly using their new system-simulation-software. Hence by including this study, also a path on which TA later can continue on has been prepared. The reference study was an evaluation of how balancing and control valves in hydronic heating systems affects the power consumption of an included centralized pump. It was executed by modeling a reference hydronic distribution system with a centralized pump, as well as balancing and control valves. By simulating this system without balancing and control valves, only containing balancing valves and finally containing both types and comparing the results, the effect that the gradual introduction of valves had on the pump work was shown. Furthermore, three different types of pumps (one with constant rotational speed and two controlled ones) as well as both steady and unsteady conditions were included in order to broaden the study.

It should be mentioned that the reference study can not be considered as finished. The reason is that not all computer models included in the simulations were verified, which means that the results of the simulations can not be taken as fully reliable. However, the purpose of the reference study was not to test the accuracy of the models, but to test if desired component behaviors could be described and solutions to systems of relevant sizes could be found. Hence, further verification and tuning of the models was left for future studies.

Finally, the verification study was included to test requirement number four, and was performed by model a test rig located at TA and to compare measured and simulated data.

2 Work in this field

This project was characterized by computer simulations of hydronic heating systems, with the power consumption of a centralized pump as the primarily unknown variable. Since this project was an evaluation of the software MathModelica, the emphasis when comparing to other projects was in their choice of simulation software and how they perform. This comparison was of importance since if MathModelica was considered to fail the evaluation, other alternatives needs to be investigated by TA. For that reason, only projects using commercial software were included in this literature study.

There seems to exist a large amount of studies in which entire buildings are simulated, and most are aiming to find optimal values for design factors in order to reduce losses in a heat balance. Due to the commonness of such studies, some have been reflected on, even if the procedures do not match perfectly the procedure of this thesis. For example, Djuric[6] has optimized certain parameters to minimize the life-cycle cost by reducing the energy consumption in a building with hydronic heating system. The software used is called EnergyPlus which, according to their website [32], is a new-generation building energy simulation software based on DOE-2 and BLAST. As used by Djuric, EnergyPlus makes it possible to account for a numerous different energy flows in building, as well as the flow of water in a hydronic heating system.

The simulation software DOE-2, mentioned above, was used by Carriere[4] in a similar study as performed by Djuric. Carriere created a computer model of a 28 000 m^2 commercial building by using measured data from a real building which the model was based on. The model was later used to evaluate some conservation measures intended to reduce energy consumption. According to Carriere, DOE-2 has the ability to handle thermal mass effects, perimeter daylight and coupled interaction and part-load performance of primary and secondary HVAC equipment. One conclusion among others was that DOE-2 was found to agree with the monitored thermal performance with high enough accuracy required for this kind for simulations.

One study which is relevant as comparison to this project is described in an article by Rhy[25]. His purpose was to study the effect that balancing of water flowrate has on the indoor climate and the thermal comfort in a radiant floor heating system. The simulation software used is called Flowmaster which according to Rhy is a network fluid-flow system analysis tool, developed by Flowmaster International Ltd. Rhy simulated a verified model of a heating system in a larger single-family house which consisted of a boiler and radiant floor heating. The conclusions were that balancing became more crucial when the floor area increased and that cavitation in a balancing valve could occur if just a few rooms required heating.

Another relevant study is the PhD thesis by Trüschel[27], in which he both used Excel spreadsheet, to simulate a hydronic radiator system, and Flowmaster, to simulate an air-heater with an included valve-group. The purpose was to evaluate how the system design affects the system sensitivity to disturbances. Based on

Trüschel´s experiences, Excel spreadsheet can not be seen as a good alternative to MathModelica. The reason is that the Excel spreadsheet is based on steady-state conditions and cannot handle any dynamic processes. This may be of interest in some cases. Flowmaster, on the other hand, was used by Trüschel to describe the more complicated dynamic behavior of an air-heater. This software seems to have about the same outline as MathModelica (e.g. graphical system interface, equation-based) and it is primarily a software for simulation of hydronic systems, and might therefore be of special interest to TA. According to Trüschel, Flowmaster also has the possibility of handling heat transfer and the level of detail is high, at the same time as the pre-defined models easily can be altered to better correspond to a specific component. However, it should be pointed out that the primarily reason why Trüschel used Flowmaster was that it was easily available and he points out that there might be other as good or even better software´s to handle hydronic system simulations.

3 MathModelica, modeling and controlling

Since the largest part of this project was devoted to modeling, this chapter will emphasize that. The purpose of this chapter was to provide an overall picture, both of MathModelica and of general modeling. Furthermore was the importance of reduced energy consumption and controlling of hydronic systems included, since they both have significant value for the understanding of this project.

3.1 MathModelica

MathModelica is an object-oriented system simulation and modeling software, which is based on the Modelica language, and developed by MathCore in Linköping.

3.1.1 Brief history

The first initiative to develop the Modelica language was taken by a group, later called the Technical Committee 1, in September 1996. The group consisted of tool designers, applications experts and computer scientists and the goal was to create a new, unified object-oriented modeling language based on the experience from existing designs, but by starting from scratch. The first Modelica language description was released by the group in September 1997 and in the year 2000, the Modelica association was formed as an independent, non-profit international organization, which interest is to support and promote the development of the Modelica language. In 2003 the next description was released and, at that time, two complete commercial programs for modeling, which supported the Modelica language, existed[10].

3.1.2 The Modelica language

One common feature of object-oriented languages, like Modelica, is that the user defines their own objects, each with different or similar properties. The objects can be used more than once when a complete system is modeled, which means that the code, which describes the behavior of each object, is reusable. The computer language then has to deal with the interaction of the objects when simulated together in a system.

In the Modelica language, the objects are primarily defined by mathematical equations and the solver can handle both linear and non-linear equations as well as ordinary differential equations containing time derivatives. Furthermore, the object-orientation concept is viewed upon as a structuring concept, used to handle interactions between objects, i.e. the equation systems. Because the Modelica language primarily is based on equations, the same rules which applies in math also applies here. For example, the number of variables in the models has to be equal to the number of equations; all other cases results in that the solver can’t proceed since the equation systems are either under or over defined. Another effect is that division by zero and square rooting negative numbers have to be avoided, in order to make the models solvable[10].

Another specific feature of Modelica is that the laws of nature are implemented to decide the resulting direction of a flow variable. In the case of hydronic systems, the result is that the direction of the flow is pointed from a higher to a lower static pressure level, which means that an intended inlet very well can become an outlet. This feature adds a higher degree of realism since the desired direction has to be achieved by initial values instead of indicated by the computer code.

3.1.3 Features of MathModelica

All of the components and systems in MathModelica have two interfaces; one graphical and one which consists of computer code. The code interface is used when most of the modeling work is performed, i.e. when describing the behavior of specific component models. These component models are then used when systems are modeled, which normally is done in the graphical interface. Then the already defined components (which also have a defined graphical representation in the shape of an icon) are dragged and connected according to the desired system configuration. What happens is that the component equations are then arranged in an equations system, which describes the behavior of the whole system. Furthermore are trivial equations created each time a connection is made. These states that the corresponding variables that exist at both ends of a connection are the same regarding their solved numerical values.

The results from a simulation are mainly communicated to the user by the use of plots. The user can choose between two types of graphs in which any of the variables, parameters or constants included in the simulation can be plotted. Either, the graphs can be structured to show the desired parameter, variable or constant as a function of the simulated time, or, one of them could be plotted as a function of some other.

3.2 Computer models and simulations

Primarily, there are two different types of computer models. These are called normative and descriptive, and both types were used in this project. The specific feature of normative models is that they are able to describe changes. These are primarily created to contribute in some type of problem solving process, and are most often used when the corresponding reality can not be re-created in laboratories. During the reference study, such models were used as so called system tools, i.e. their purpose was to show how a system was affected by changes, which initially only referred to some part of the system. The descriptive models, on the other hand, merely describe a system as it is today, and hence, no changes are involved. This type of models was used in the verification study, to compare how well the models described a real corresponding system.

A computer model is based on the modeler's picture of the reality. This means that the simulated results are directly dependent on both the modeler's ability to interpret and describe the reality, as well as the degree of details incorporated in the models. Therefore, it is very important to be aware of any limitations when the results from a simulation are used. According to Rydén[24], some limitations can be identified by reflecting over the modeling ethics presented in his paper "Energy models for the technical energy system". In this paper, Rydén stresses that it should be kept in mind that the quality of the output can not be higher than the quality of the input. But, one should also keep in mind that the inputs never can be perfect. Simplifications are always made, and the picture given by the models corresponds at its most in a tolerably way to reality. For that reason, simulations should never be the only basis for decisions, but should merely add to further insights. Finally, in order to be aware of the limitations, models should always be verified to reality before the results are being used. In all other cases, simulated results should be considered as unreliable[24].

3.3 Energy efficiency for hydronic heating systems

The largest part of this project consisted of the reference study, in which the influence that valves have on the power consumption of a centralized pump was studied. As mentioned, this study were already planned to be performed by TA, primarily because a low operational energy consumptions of the systems, is an important factor for the customers when products are being chosen. Hence, by learning more, further improvements of their products might be possible which in turn could create market advantages[21].

For the customers, the incentives to reduce their electrical consumption could both be due to environmental and economical considerations. The environmental incentives are represented among customers who want to contribute to a global sustainability, and are due to the fact that the global energy system is facing a double challenge: on one hand it has to fit within the restrictions set by nature, regarding limited resources and limited possibility to assimilate waste. On the other hand it has to fulfill the needs for a growing population at the same time as the consumption pattern head towards an increased consumption per capita. Looking at electricity, the limited resources can for example be identified as the fuel used in power production, and the waste, the emissions of carbon dioxide related to combustion [14].

The most obvious economic incentive is that the operation becomes more expensive with high electrical consumption, but there is also an interest to reduce the electrical consumption on a national level in Sweden. This is due to the fact that the Swedish energy system mostly is dependent on electricity produced by nuclear and hydro power. However, since the nuclear in Sweden can not be expanded due to political decisions and the hydro due to environmental considerations, any larger increase of consumption has to be met by another technology. Since hydro and nuclear are the existing large scale technologies with the lowest running cost, the other alternatives will probably be more expensive per energy unit produced. Furthermore, since the Swedish electricity grid is connected to the rest of Europe, in which many countries relies on fossil fuels, any energy savings in Sweden will probably result in a reduction of fossil based power production elsewhere. The reason is that fossil fuel based production is a more expensive alternative, and is not used as long as cheaper alternatives are available.[13] Hence, from this point of view, the economic and environmental incentives overlaps.

3.4 Controlling of hydronic heating systems

The purpose of a heating system is to achieve a desired indoor climate, at the same time as the operational costs and disturbances should be minimized. The most common task is to achieve a constant temperature in the space, or a temperature that varies in a specific way; for example reduced temperature during night-time. However, since the room temperature constantly is submitted to disturbances, such as variations of outside temperature, ventilation rate and internal heat generation, the emitted thermal power of the thermal units has to be adjusted to meet the desired indoor climate. If the room temperature should be kept constant, the supplied heat has to be exactly equal to the losses, otherwise the room temperature will decrease or increase, resulting in a deviation from the desired level[20]. Hence, controlling is necessary and the purpose is to handle disturbances so that the system operates properly.

Since it is the room temperature which should be controlled, its actual value has to be known to the system and necessary corrections must be possible. Figure 3.1 shows an example of a control system, in which sensors measures the room temperature and sends a signal to a controller which compares the actual value to the desired value. If these are not consistent, the controller sends a correctional signal to an actuator which performs some kind of action in order to makes the necessary adjustments. In this study, the actuator adjusted the opening of a valve placed at the inlet of a corresponding thermal unit. This arrangements allowed the flows through the thermal units to be controlled, so that the corresponding emitted thermal power became as desired[20].

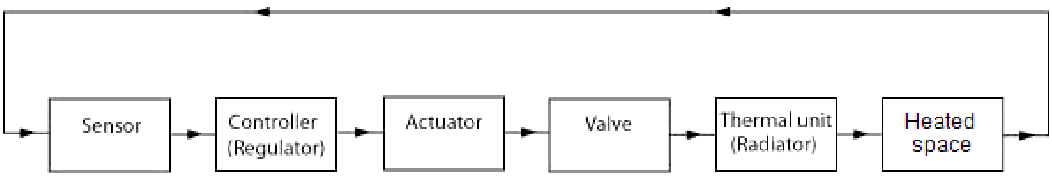

Figure 3.1 Block diagram of a principal control system.[20]

3.4.1 Controlling using valves

The valves, whose openings are controlled by an actuator, are called control valves and their purpose is to maintain the desired performance of the system by adjusting the flow and pressure level. Together with control valves, balancing valves are most often used in hydronic heating systems. Their purpose is to balance the flow through each thermal unit by adding flow resistance, so that the flow at full load (also called design condition) exactly matches what is prescribed.

This means that if the system is balanced, the control valves should be fully opened during the design condition.

Balancing valves are needed since irregularities initially are built into the system which causes overflows in some parts and underflows in others. If this is the case, some of the control valves might have to work during design conditions, i.e. at full load when they should be fully opened. This is problematic since each control valve only has a certain range to work in, and if some of that range already is covered during design conditions, less can be done about the disturbances. The purpose of balancing is to minimize this; i.e. to achieve the best pre-requisite for the control system as possible.

Another example of why controlling often is not enough is that areas with very low flows can occur in unbalanced systems. The problem is that even if the control valves in such areas are fully open, the flowrates might still be too low. In turn, since the emitted thermal power of a thermal unit is dependent on the flowrate, the prescribed heat during design conditions might not be delivered in these areas. The reason why such areas come into existence is that a hydronic heating system is composed of several hydronic circuits, which are differentiated by pipelengths and amount of hydronic component. This result in circuits with different flow resistance, and since the flowrate and flow resistance are dependent, the flow in different areas of an unbalanced system will differ: even if the prescribed flows are the same. When a system is balanced, the valves are used to even out the flow resistances in the circuits, which results in that the possibility that the prescribed flows can be achieved becomes higher.

A poorly balanced system will only not achieve the desired indoor climate, but also the energy efficiency will be low. The reason is that the pump in an unbalanced system has to be over-dimensioned in order to deliver high enough flowrates in all circuits. Hence, the emitted thermal power will be too high in some areas and too low in others, resulting in a poor matching of the loads. This implies that the size of the pump could be decreased in a balanced system, which will have a positive effect on the pump work. But, on the other hand, larger pressure drops are introduced by balancing valves which might have a negative effect [25][22].

4 Theory

The purpose of a model is primarily to describe, as well as possible, some relevant features of a corresponding real component. In many cases, the behavior which is considered as most relevant is the distinguished one, i.e. the most characterizing feature. When such is described mathematically and illustrated graphically, the plot is called the component characteristics, and depending on the component, the characteristics can be more or less general.

4.1 Pressure drop

The driving force for a fluid flow is the static pressure difference between two points but as the fluid flows, it is subjected to flow resistance causing pressure drops, i.e. reduction in driving force. For a component in a hydronic system, this relation can be described by equation 4.1[27] as long as the flow through the component is turbulent. In this equation, ξ is the flow resistance factor and Δp is the pressure difference between the inlet and the outlet, i.e. the pressure drop over the component, in Pa. If the flow through the component instead is laminar, the relation between the flowrate and pressure drop becomes linear. Furthermore, Idelchik[12] states that the total pressure drop in a system can be determined by the principle of superposition, which implies that the total pressure drop in the system can be determined by adding all the individual pressure drops over the components.

$$\dot{V} = \sqrt{\frac{\Delta p}{\xi}} \qquad \left[m^3/s\right] \qquad \text{(eq. 4.1)}$$

In the book "Fluid mechanics", White[31] states that the pressure drop in a system mainly derives from two sources; either from frictional losses or due to changes in geometries. The latter disturbs the flow, which result in that eddies are forming, which in turn increases the internal friction of the fluid. The increased internal friction further results in an increased internal energy, which is equal to a loss.

For most components in hydronic systems, the pressure drop is caused both by frictional losses and by geometry changes. In Idelchik, these pressure drops are referred to as "local" since they only occur at locations where a specific component, such as pipe-bend, valve etc, exists. The only pressure drops not designated as local in this project are the pure frictional losses caused by the flow through straight pipes.

When it comes to describing the relation between the flowrate and the pressure drop over a certain component in this project (equation 4.1), the components was classified into two groups; the ones that were considered as describable by standard models, and the ones which were considered as more unique, and therefore had to be described by specific models in order to maintain the accuracy. The difference between the components in these two groups is the number of parameters which are considered to have influence on the flow resistance at a certain flowrate. For a standard component, the relevant parameters are relatively few which allow standard models to be produced. This is possible since the flow resistance can be presented as a function of the flowrate for each combination of the parameters. Such relations are well documented in the literature. For specific components, on the other hand, there are a much larger number of flow resistance dependent parameters which makes it much harder to produce reliable standard models. For those, the best way is to use measured data, specific for the component, when modeling.

4.2 Standard components

For local pressure drops which can be described by standard models the previous presented equation 4.1 can be used. However, in most cases a rewritten version (equation 4.2) is used, in which the resistance factor is denoted as *k*. The k-factor is specific to the component and dependent on different parameters such as the geometry, size and surface roughness of the component. The other terms on the right hand side of equation 4.2 describes together the dynamic pressure of the flow.

$$\Delta p_{local\,loss} = k \cdot \frac{\rho \cdot u^2}{2} \quad [Pa] \qquad \text{(eq. 4.2)}$$

Finding an analytical value to the k-factor, is not easy according to White, since the flow pattern often quite complex. This is much do to the geometry of the components, which creates complicated turbulent behavior of the fluid. White also states that the theory, for that reason, is very weak and most of the solutions are based on experimental data. In literature, experimental k-factors for standard components are often well documented.

4.3 Unique components

For estimated non-standard components, the characteristics are more or less dependent on the particular manufactures design. Hence, each individual type of component has to be treated as unique, even if standard models are available. This is necessary if the accuracy of the models is to be maintained. As an example it can be mentioned that, according to White, standard models of valves can differ a lot, between ±50 %, depending on the manufacturer. This would naturally result in very unreliable simulated results.

4.3.1 Valves

When it comes to valves, the most common approach is to express the pressure drop as a function of the valve capacity instead of the k-factor, and the advantage is that the valve capacity is easier to determine experimentally. The valve capacity (*Kv*, defined in eq. 4.3) is dependent on the valve opening, and corresponds to the magnitude of the flow which passes through the valve at a differential pressure of 1 bar[5]. If the valve if closed, the Kv-value will be zero, and the corresponding maximum value is instead achieved when the valve is fully opened. A common way is to represent the Kv-value as a function of the valve opening.

$$k_v = \frac{\dot{V}}{\sqrt{\frac{\Delta p}{\Delta p_0} \cdot \frac{\rho_0}{\rho}}} \qquad \left[\frac{m^3}{h} \Big/ \sqrt{bar} \right] \qquad \text{(eq. 4.3)}$$

4.3.2 Pumps

The behavior of a pump depends on a variety of parameters; e.g. the size of the motor, the type and working principle of the pump, the shape and size of the canals etc. This means that the behavior of a pump is dependent on the actual manufacturer, and pumps should therefore be treated as unique. However, the power consumption can generally be expressed as in equation 4.4[5]. In this, the product between the pressure rise and the volume flowrate represents the useful work delivered by the pump, and by dividing with the pump efficiency, the total supplied work is calculated.

$$\dot{W} = \frac{\Delta p \cdot \dot{V}}{\eta} \qquad [W] \qquad \text{(eq. 4.4)}$$

In a heating system, the purpose of the pump is to transport the water through the distribution system. If the system is closed, the pump only has to compensate for the pressure losses in the distribution system.[22] In this study, pumps of the centrifugal type were used which is the most common type used in heating systems. These are characterized by that the useful energy is delivered to the fluid by means of velocity changes which occurs as the fluid flows through a rotating impeller.[2]

4.3.2.1 Characteristics

The differential pressure over the pump depends on the flow through it at a given rotational speed of the impeller. An increase of the flow will decrease the pressure difference. Most often is this relation quadratic and can be illustrated graphically, with Δp on the y-axis and $\dot{V}$ on the x-axis. Such plot is called a pump curve, which is the characteristic for a pump and valid for a specific rotational speed[5].

Together with the pump curve, a pump characteristic which shows the relation of the power consumption and the volume flow is often used. This characteristic is called power curve and its behavior is of importance to determine the required power consumption for different operational cases.

4.3.2.2 Constant and variable rotational speed

Compared to pumps with constant rotational speed, there is a possibility to reduce the energy consumption by instead using controlled pumps. To describe this, the characteristics of a complete distribution system can be used. Just like the pump, the rest of the system can be illustrated in a Δp-$\dot{V}$ diagram. The relationship between the pressure drop in the system and the volume flowrate through it is called the system characteristic, which is determined by the actual flow resistance in the system at a certain time. As seen in figure 4.1, this curve is of opposite growth compared to the pump curve since the total pressure drop in the distribution system will increase as the flowrate increases. This means that the two curves obviously will intersect if illustrated in the same diagram. This intersection becomes the operation point of the pump (denoted A in figure 4.1), and is found in the point where the total flow through the system is such that the total pressure drop over the system is equal to the pressure rise of the pump. This point is specific for a certain total flowrate, which only can be changed either by altering the pump or the system characteristics. The most common way to change the pump characteristic is to alter the rotational speed, and for the system characteristics, the most common way is to change the openings of one or more valves in the system. The latter is illustrated in figure 4.2, where the total valve opening has been reduced from an opened state (A) to a more closed one (B)[5].

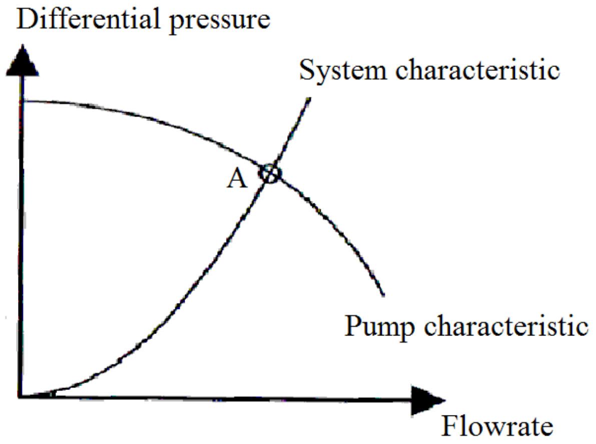

Figure 4.1 Pump and system characteristics, constant rotational speed pump.[5]

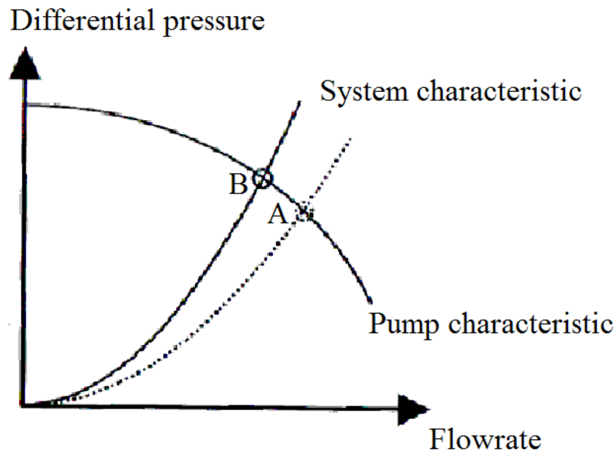

Figure 4.2 Pump and altered system characteristics, constant rotational speed pump.[5]

As seen in figure 4.2, the changed system characteristics result in a new operational point for the pump. This figure illustrates the response of a constant rotational speed pump, which is to move left along the pump curve (denoted A to B) reducing the flowrate and increasing the pressure rise. However, the response of a variable speed pump is instead to reduce the rotational speed which results in a new pump curve as shown in figure 4.3. Then, the new system characteristics can be intersected in a new operational point without increasing the pressure (denoted A to B in figure 4.3), which also results in lower total flowrate. According to equation 4.4, the operational point of the variable pump result in a lower power consumption compared to the constant speed pump, as long as the gain achieved by the lower pressure rise and flowrate is not overcome by a decreased efficiency.[22]

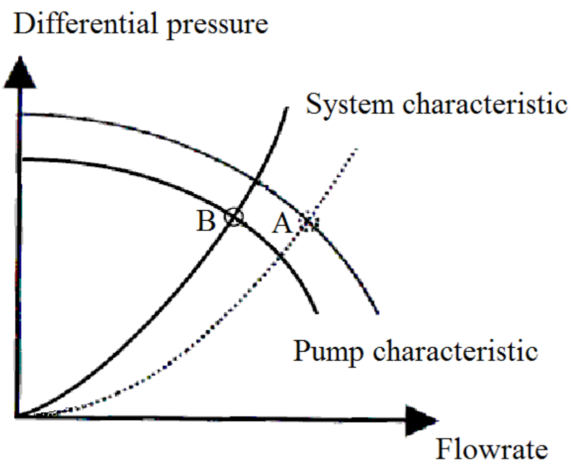

Figure 4.3 Pump and altered system characteristics, variable rotational speed pump.[5]

As can be seen, the variable rotational speed pump illustrated in figure 4.3 is controlled to maintain the pressure rise over itself or some point in the system (the pressure level of A and B is equal). However, there are other possibilities, which for example can be to control the pressure drop at some part of the system. Since the pressure drop is dependent on the volume flowrate, the pump instead adjusts its rotational speed to maintain this flowrate.

4.3.2.3 Efficiency

According to equation 4.4, the energy efficiency of the pump is variable, since it always has to be zero when the flowrate is zero. Furthermore, according to an arbitrary pump curve, there is a certain value of the flowrate for which the product between the pressure rise and volume flowrate reaches a maximum. And since the drive power in most cases is more or less constant for a certain rotational speed, this value also corresponds to the operational point for which also the energy efficiency reaches its maximum. This is shown in figure 4.4 for an arbitrary pump, where the energy efficiency is plotted as a function of the flowrate for a certain rotational speed.

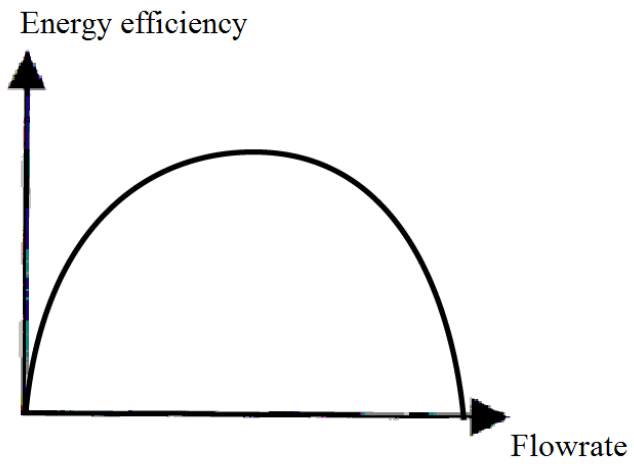

Figure 4.4 The energy efficiency of an arbitrary pump as a function of the flow rate for a certain rotational speed. [2]

When the rotational speed instead is adjusted, the resulting energy efficiency is depending on the relation between the old and new operational points. According to Alvarez[2] the energy efficiency can be regarded as constant on a so called load line, as long as the change of rotational speed is moderate. A load line is a parable which crosses the origo of a pump curve, as shown in figure 4.5. Mathematically, this can be explained by equation 4.5 and 4.6. Hence, an old and a new operational point results in about the same energy efficiency of the pump if their relation satisfies these two equations.

$$\Delta p_{new} = \Delta p_{old} \cdot \left(\frac{\omega_{new}}{\omega_{old}} \right)^2 \qquad [Pa] \qquad \text{(eq. 4.5)}$$

$$\dot{V}_{new} = \dot{V}_{old} \cdot \left(\frac{\omega_{new}}{\omega_{old}} \right) \qquad [m^3/s] \qquad \text{(eq. 4.6)}$$

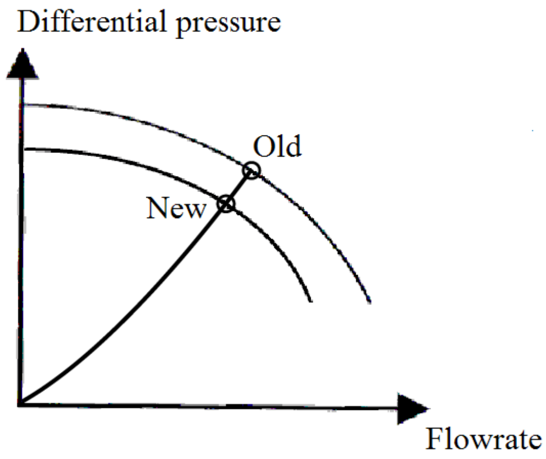

Figure 4.5 A load line is a parable which crosses the origo in a pump curve.[2]

4.4 Controlling

In figure 3.1, a block diagram of a control system was shown. In this, sensors measure the actual value of the variable which should be controlled, and sends a corresponding signal to the controller. The controller compares a set (desired) value to the measured one, and sends a correctional signal to the actuator, based on the difference between them which also is called the controlled error. In turn, the actuator makes the necessary adjustments which are needed in order for the set value to be reached. In figure 3.1 the actuator adjusted the opening of a valve, but the same principle can be applied on the controlled pumps discussed above. Then, the actuator most probably is an electrical motor connected to the pump and by controlling the speed of the motor, the rotational speed of the pump can be adjusted to meet the desired pressure rise or flowrate. Either way, this structure of the control system is referred to as feedback control, which means that the correctional signal is based on observations of actual conditions.

4.4.1 Controller types

There are different types of controllers, but according to Lennartson[16], the most common types are the P- and PI-regulators. The P-regulator, in which the *P* stands for proportional, implies that the correctional signal simply is proportional via a gain factor to the control error. This relation is shown as equation 4.5, where u is the correctional signal, e is the error signal, K_c is the gain factor, r is the set and y is the actual value of the control variable. This means that a large error results in a large correctional signal, i.e. large adjustments of the actuator. Furthermore, it can be seen that both negative and positive correctional signals can be achieved. This means that the controlled feature of the actuator both can be reduced and increased, which of course is necessary if compensations are to be possible for all types of errors. The PI-regulator includes besides the proportional part, also an integrating one. The purpose of the integrating part is to remove remaining control errors, which can't be fully compensated by the proportional part. By including an integrating part, yet another relation is introduced, which results in that the correctional signal also becomes dependent of the time integral of the error signal. This means that if the error signal not is equal to zero when the time increases, the correctional signal also will increase. In time, the integrating part will result in that the error signal becomes equal to zero[16].

$$u = K_c \cdot e = K_c \cdot (r - y) \qquad \text{(eq. 4.5)}$$

5 Method

Like mentioned in the introduction, a requirement specification was given by TA. This specification contained the requirements that a system-simulation software should fulfill in order to be considered as a useful development tool by TA. These requirements were the basis for this project and were phrased as:

1. the software should find solutions to hydronic systems of relevant sizes,
2. the software should be able to be used to model both balancing and control valves as well as other common hydronic components
3. the software should be able to produce results which agrees to reality regarding systems behavior, and finally,
4. the software should be able to produce results which also agrees to reality regarding numerical accuracy

The numerical accuracy requirement was tested for MathModelica by performing a verification study in which a test rig, located at TA, was modeled. By comparing simulated and measured data during different operational modes, the accuracy of the simulated results was tested. All other requirements were tested by performing the so called reference study. This study was actually based on a study which TA is intending to perform in the future, using a system-simulation tool. This means that by including this study, also a path was prepared, which TA has the possibility to continue on. The purpose of the reference study was to decide how the power consumption of a centralized pump (defined in equation 4.4) in a hydronic heating system is affected when control and balancing valves are introduced. It was executed by model a reference distribution system (shown in figure 5.1) which was thought to represent the structural foundation on which most hydronic distribution systems found in reality are based upon. Three different configuration of this system were simulated: without balancing and control valves, only containing balancing valves and finally containing both types. By comparing the results, the effect that the gradual introduction of valves had on the power consumption was shown.

5.1 Choice of simulation variables

When dealing with systems, it is crucial to describe the interactions between the components since this is determining the corresponding system behavior. For a hydronic system, these interactions are simply called hydronic. They occur when a change in flow through one component affects the pressure level in the system enough to change the magnitude of the flow through some other component. In this study, the pressure levels of the systems were determined by using the principle of superposition, presented in part 4.1. This means that both the pressure difference over, and the flow through, each component in the systems had to be known in order to determine the interaction. This was achieved by using the relations presented in part 4 of this thesis.[27]

For a hydronic heating system, the thermal interaction is also needed in order to fully define the system. However, it was chosen not to include these. This was motivated by that the power consumption of the pump was of primarily interest, and since the only dependent variables are the pressure rise and volume flowrate, the thermal interactions were of less importance. Hence, only including the hydronic variables in the models was a way to limit the study.

5.1.1 Overall initial condition

Initially, the pressure level for all components except the pumps was set to about atmospheric (100,000 Pa) but slightly higher (about 1 Pa) at each inlet compared to the outlet. This was done in order to create an initial driving force for the flow in the correct direction, and by doing this; backflows were avoided during the entire simulations. The initial low differential pressure also implied low initial flowrates through most of the components. The exception was the pump since, according to a pump curve; the pressure rise is at its maximum when the flow is at its minimum. This was modeled by adding the largest possible pressure rise to the atmospheric pressure at the outlet, and at the same time maintaining the atmospheric pressure at the inlet. Furthermore, by setting a prescribed flow direction of the pump, it became the origin of both flow and pressure in the system.

5.2 The reference study

All valve models used in the reference study were derived from corresponding products developed by TA. The balancing valve model derived from a product called STAD, which is a manual balancing valve with measuring ports. In turn, the control valve model derived from an automatic balancing and control (ABC) valve, which currently is under development. These products were chosen since the STAD is a very common type of valve, found in many applications, and the ABC was thought to represent the most up-to-date product among the control valves manufactured by TA. Hence, these choices reflected the reality in two different ways; both by using common and up-to-date components. Also the system modeling was performed in a way which was thought to reflect reality: all components in the systems were connected to each other as prescribed in user hand books, and all systems including balancing valves were balanced to achieve realistic valve settings.

In order to broaden the reference study, two different perspectives were further investigated. First, three different test set ups including different types of centralized pumps were modeled and simulated for each of the three distribution system. The intention was to show if the results also were dependent on the type of pump included in the system. One of the pumps had constant rotational speed, and the other two were controlled. The feature of both of the controlled ones was to keep the pressure difference at some part of the system at a certain value, which in one case was over the thermal unit located most far away from the pump, and in the other, over the pump itself. Second, disturbances were introduced to the systems during the simulations, and the corresponding response of the system and pump was observed. By doing this, the power consumption during unsteady conditions was investigated while the other parts of the study only handled steady state conditions. Even if the unsteady parts normally only constitute a fraction of the total operation of a system, it might be of importance and should therefore be investigated in order to get the whole picture.

5.2.1 Simulations

As indicated, the simulations included in the reference study consisted of nine different scenarios. These are shown in table 5.1, and as can be seen, the characterizing feature of a scenario was the combination of distribution system and type of centralized pump. The approach of the reference study was to compare the scenarios located in the same column. Hence, what really was observed was how the power consumption of a specific type of pump was affected by the stepwise introduction of valves.

The inter-grouping of the scenarios in the columns of table 5.1 indicates the size of the systems (i.e. the pump and distribution system combined) regarding the number of modeling equations. Hence, the systems located in the first row consisted of the fewest equations (about 1500) compared to the other two systems located in the same column. And naturally, the systems located in the last row consisted of the largest number of equations (about 2500). Furthermore, placing of the columns from left to right represents the complexity of the systems regarding how they were controlled.

Table 5.1 The simulation scenarios of the reference system.

Type of centralized pump / System configuration (distribution system)	Constant rotational speed	Controlled pressure rise over itself	Controlled pressure drop over the last thermal unit
Reference system as shown in figure 5.1	1	4	7
Also including balancing valves	2	5	8
Both including balancing and control valves	3	6	9

5.2.2 The reference system

The reference system (figure 5.1) was modeled as an arbitrary closed hydronic heating system, with a centralized pump and a 2-pipe distribution system. Totally, the system contained 50 meters of straight pipes, divided between one main circuit and two risers. On each riser, three thermal units were connected in parallel by the use of T-junctions.

The size of the reference system, regarding the number of branches and circuits, was primarily limited by the corresponding size of the result file when the system was simulated. Hence, the size constraining factors of the system were primarily what size a result file should not breach in order to be considered as acceptable and the available hard drive capacity of the computer. These factors became limiting since the option to set the output interval of a simulation was not available in MathModelica. For that reason, the result files tended to be very large and hard-to-handle when the number of branches and circuits increased.

There were at least two reasons why this system structure was chosen as reference. First, the configuration was thought to represent the foundation on which most hydronic heating systems are built according to. This was a desirable feature with the purpose to make the study as general as possible. The reason why the chosen configuration was seen as general was that it both contained risers, originated from a main circuit, and branches on these. This means that the most common branching possibilities were represented, even though in their simplest form. An arbitrary real system would most probably be larger, but the foundation the same, since the expansion would probably be made by adding branches, risers and thermal units. Second, this type of system was presented in technical documentation concerning how the ABC in combination with STAD should be configured. This means that the chosen connection scheme both is documented and approved by TA.

As can be seen, both the heat source and sinks were not included in the system boundary. The reason was that these were considered as more or less thermal, and their primarily functions could not be described without adding thermal interactions (thermal interaction are left out, see section 5.1). The primarily focus was instead on pure hydraulic components, with the exception of the thermal units, which were considered as the interface between thermal and hydraulic interactions. However, these were only used in the planning of the simulation performed during the balancing procedure (described in part 5.2.5), and then their thermal parts were treated as constant. Hence, the thermal units were not included in the system simulations. Instead were they approximated by their corresponding radiator valves, which mean that the pressure drops over the thermal units themselves were neglected.

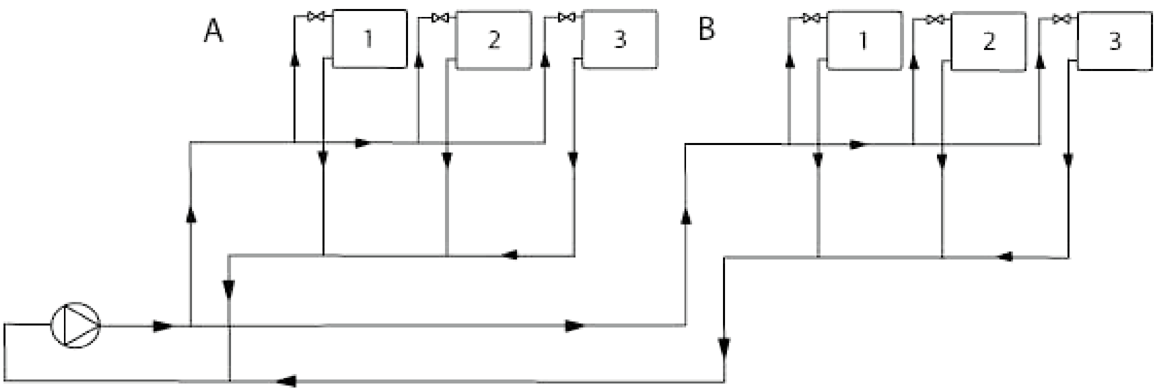

Figure 5.1 The reference system.

5.2.3 Introduction of balancing and control valves

Figure 5.2 shows the resulting system configuration when balancing valves were included. For each thermal unit, a corresponding balancing valve was added to the outlet, which enabled the possibility to balance each flow individually. Theoretically, the balancing valves could instead have been added to the inlets, since it is the combined pressure drop of the circuit (balancing valve in combination with the thermal unit) that determines the flowrate through it. However, conventionally the balancing valves are placed at the outlets to reduce the risk of air infiltration into the thermal units and, hence also in this study, to reflect the reality as far as possible. Furthermore was a main valve (denoted M) also introduced. However, if the configuration in figure 5.2 should correspond to reality even further, yet another two balancing valves, called partner valves, should be added to the return pipes of the risers. In a real system, these are used for compensation and are essential if a balancing procedure should be executable. However, in this study, the balancing was performed numerically (see section 5.2.5) which makes them obsolete. But, the main reason of why they were excluded is that they, according to TA, aren't needed in real systems which contain ABC valves. This means that if all systems should correspond to reality in this aspect, the partner valves had to be included in the system only containing the STAD valve, but excluded in the systems also containing the ABC. Naturally, this would have opposed the scope of the study, which was to gradually introduce valves to observe the effect on the pump work.

In the second step, an additional ABC valve was added to the inlet of each thermal unit, as shown in figure 5.3. This combination of valves corresponds to a specific type of valve group called direct connection. The most distinguished feature of a direct connection is that all water which passes through it, also passes through the thermal unit, and its purpose is to control the flow through a single thermal unit in order to achieve the desired thermal output[5]. The feature of the ABC valve in this group is to handle variations that occurs outside the circuit (thermal unit, control and balancing valve), by maintaining the pressure drop over, and hence also the flowrate through the circuit. The ABC can perform as long as the controlling region is not exceeded which occurs when the flowrate attains extreme values. If very high, the ABC valve almost has to close in order to maintain the desired flowrate. A small adjustment of the opening then results in a large change of flowrate which means that fluctuation will occur, i.e. unstable operation. If the flowrate on the other hand is too low, the ABC valve can't compensate what so ever. The reason is that even if the valve opening is at its maximum, the flowrate is still lower than prescribed. This scenario occurs if the thermal emittance of the unit is controlled by the means of a radiator valve, which has closed since the room temperature is larger than desired. Hence, if ABC valves are used, the flowrate can be controlled from inside the circuit, but should be unaffected by changes on the outside.

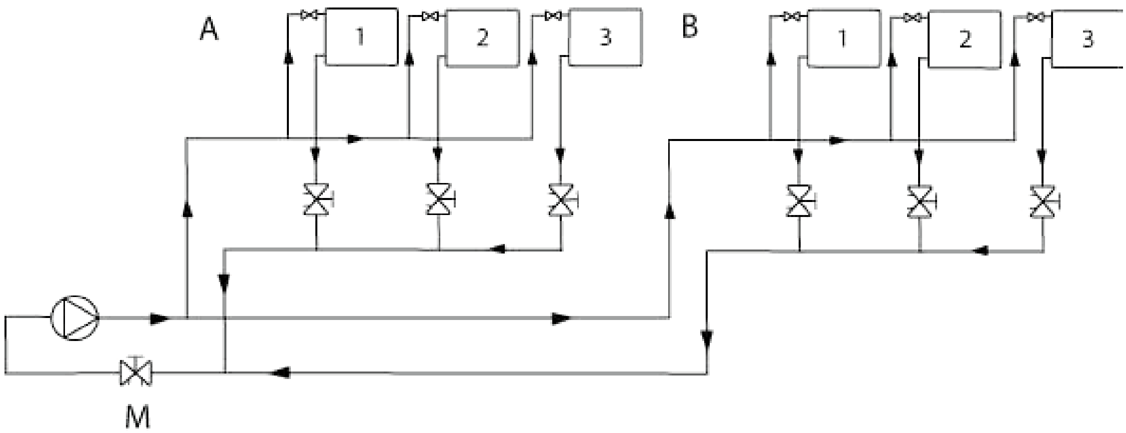

Figure 5.2 The reference system including balancing valves.

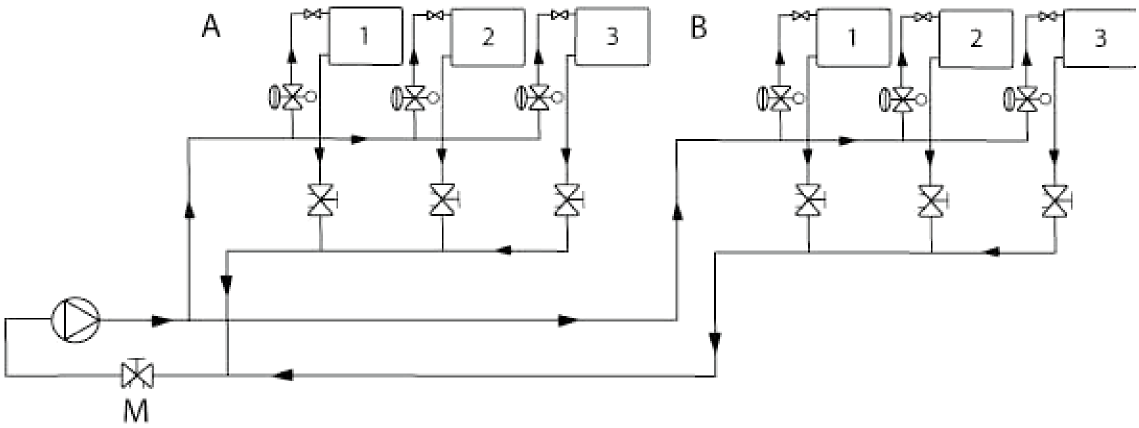

Figure 5.3 The reference system including balancing and automatic balancing and control valves.

In this study, the three system configurations also corresponded to three different flow levels; the flow in the system without valves corresponded to an overflow which was larger than the design flow at full load, the system with balancing valves corresponded approximately to design flow and the one also containing ABC valves corresponded to part load flow conditions. This approach seemed inevitable since the most relevant alternative approach would have been to maintain the design flow in all systems and to observe the difference in pressure rise of the pump. Compared to the chosen one this alternative approach had some large drawbacks. First, the difference between the system only containing the STAD and the system also containing ABC valves, would have become small and this comparison would hardly add to any new insights; this is further discussed in part 5.2.5. Second, this approach could not be applied to the reference system since the possibility to adjust the flow was not available in that case.

5.2.4 Types of pumps

The reference study included three different types of pumps. All of these were based upon a Grundfos Magna 32-120 which is located in the test rig used in the verification study (section 5.3). One the pumps were modeled to statically followed the pump curve, which corresponds to the maximum rotational speed.

The other two pumps were modeled as controlled in respect to the rotational speed. The first controlling approach was to control the differential pressure over the thermal unit most far away from the pump (denoted B3 in figure 5.1). This approach was chosen since, in a 2-pipe distribution system, the differential pressure over the last thermal unit, is the lowest among all thermal units in the system. This is due to that the fluid has to travel the longest way through the pipes to reach this point, and since the pressure drop in a pipe is proportional to the length, the driving force reaches its minimum. This means that by controlling the differential pressure over the last thermal unit, the pressure level in the entire system can both be maintained and minimized.[22] In this study, a certain pressure drop over the last thermal unit was achieved by maintaining a corresponding volume rate through it, which had to be met by a corresponding volume flowrate of the pump. This means that the volume flowrate of the pump was more or less fixed, but the corresponding pressure rise of the pump could attain any value within the valid working area of the pump by adjusting the rotational speed.

The other controlling approach, to keep the pressure rise over the pump itself at a certain level, was illustrated in figure 4.3. This means that the pressure rise of the pump was maintained throughout the simulations but, in this case, it was the volume flowrate that could attain any value within the valid working area of the pump.

5.2.5 Balancing procedure

According to Petitjean[22], the goal of a balancing procedure is to achieve a so called λ-value of one, through all thermal units at full load. As can be seen in equation 5.1, this means that the flow is as prescribed in the entire system at full load.

$$\lambda = \frac{\dot{V}}{\dot{V}_{design}} \qquad [\%] \qquad \text{(eq. 5.1)}$$

In the first step of the balancing procedure used in this study, the size of the units, regarding the required thermal emittance at full load, were determined. They were chosen to correspond to existing hydronic radiators manufactured by Purmo, and the available sizes were given in product specifications.[23] The different sizes were chosen arbitrarily but with a certain spread to include both large and small units. The purpose was to cause an unsymmetrical flow distribution, i.e. to model a system in which thermal units with different required maximum heat loads existed. This was considered as most realistic since different room volumes, number of outside walls, number of windows etc. would result in different required radiator sizes.

In the next step, the required flowrate at full load was determined by equation 5.2[15]. Since only hydronic interactions were described in this project, the temperature drop had to be treated as constant. A suitable value was found in Purmo, since together with the maximum thermal emittance, the required inlet temperature and the corresponding outlet temperature were presented. This resulting temperature drop, which was 10 K for all units, was used during the whole study. Furthermore, since many of the fluid properties used are temperature dependent, it was decided to assign them values at a level which corresponded to the mean value of the required inlet and outlet temperatures, which was 70 °C.

$$\dot{Q} = \dot{M} \cdot c_p \cdot (T_o - T_i) = \dot{M} \cdot c_p \cdot \Delta T \qquad [W] \qquad \text{(eq. 5.2)}$$

In the final step, the valve openings which corresponded to the flowrate at full load were determined numerically. This was done by modeling STAD valves, in which the required flowrates were set as constants, at the same time as the valve openings were set as the variables which should be solved for. The balancing procedure was performed by simulating a configuration containing these balancing valves and ABC valves whose first cone (further discussed in section 6.4.2) were constantly set to fully open. This was considered as the most realistic approach, since balancing is the last procedure performed before a hydronic heating system is taken into operation, which means that control valves are present[22]. The settings of the balancing valves which were gotten were used throughout the whole reference study. This procedure was one of the reasons why the approach to let the different systems corresponds to different flow levels, as discussed in part 5.2.3, was chosen. If instead the design flows were to be maintained in the system containing the ABC valves, the ABC valves would constantly have been fully open. This means that the result would have been the same as the simulation which was done during the balancing procedure, i.e. about the same as the system only containing the STAD valves. In table 5.2 the data used during the balancing procedure, including the resulting valve openings, are summarized for each thermal unit as denoted in figure 5.1.

Table 5.2 Data used for balancing and the resulting valve openings.

Valve	**Thermal power at full load, $\dot{Q}$, [W]**	**Temperature drop, ΔT, [K]**	**c_p [J/(kg K)]**	$\dot{M}$ $[kg/s]$	$\dot{V}$ $[m^3/h]$	**H [mm]**
A1	3759	10	4191	0.0896	0.324	1.968
A2	3258	10	4191	0.0777	0.280	2.599
A3	3759	10	4191	0.0896	0.324	1.968
B1	2005	10	4191	0.0478	0.173	1.460
B2	3759	10	4191	0.0896	0.324	1.971
B3	3258	10	4191	0.0774	0.280	2.612
Main				0.4717	1.70	3.805

Beyond the valve setting, also the sizes of the valves were determined during the balancing procedure. The aim was to find valve sizes which corresponded to the largest valve setting as possible, when the design flow was achieved, since this is recommended by TA for reasons discussed in part 5.3. The main difference between different valve sizes is that the Kv-value of a certain setting is larger for a larger valve than for a smaller. This means that the setting of a large valve has to be smaller to achieve a certain flow, compare to the corresponding setting of a smaller valve. Hence, the purpose was to find the smallest valves for which the desired flow conditions still were included in their valid working region. In conclusion, it was determined that all valve sizes should be of DN10 except for the main one which was better suited as DN20. This is explained by that the flow through the main valve was much larger than through the rest, which also can be seen in figure 5.1.

5.2.6 Disturbances

The largest parts of the simulations showed the systems during steady state. This was considered as the most important parts, but in order to do a complete investigation, also the unsteady parts of the pump work were included. This was the purpose of the disturbances; to illustrate the resulting system and pump response with regard to the power consumption, and how this was differentiated between the scenarios.

Disturbances were introduced via the radiator valves which were modeled as manual, and with normalized intervals of the valve openings. Hence, a valve setting of one corresponded to fully open, and zero to fully closed. Equal disturbances were introduced two times per simulation, but at different time steps and at different locations in the system. The state of the system just before the introduction of a disturbance was steady and all radiator valves were fully opened. The disturbances consisted of that one of the radiator valves, during 1.5 simulated seconds, almost closed (setting 0.143) and went back to fully opened again, as illustrated in figure 5.4. Such character of the disturbances was chosen, since the systems had to react both relatively fast and in different directions, in order to compensate. The purpose was to induce oscillations, which was thought to clarify the character of the response, and hence to make the observations more easily performed.

During the simulations, it was indicated that flowrates which approaches zero seems to be problematic for MathModelica to describe. Divergence problems were for example encountered when a variable valve opening approached zero. This was the reason why a disturbance was modeled as a radiator valve which went from fully open to *almost closed* and then back to fully open. That is, why the radiator valve in question did not completely close, which naturally would have resulted in even larger disturbances. Instead the value which represents the *almost closed* valve opening was tuned, and the chosen value corresponded to the lowest valve opening which could be solved by MathModelica.

Totally, each system was simulated during 30 seconds and disturbances were introduced at time 10 and 20 seconds. The first disturbance occurred at the thermal unit denoted as A3 in figure 5.1 and the second one at B1. These were chosen to include disturbances that both occurred close and far away from the pump.

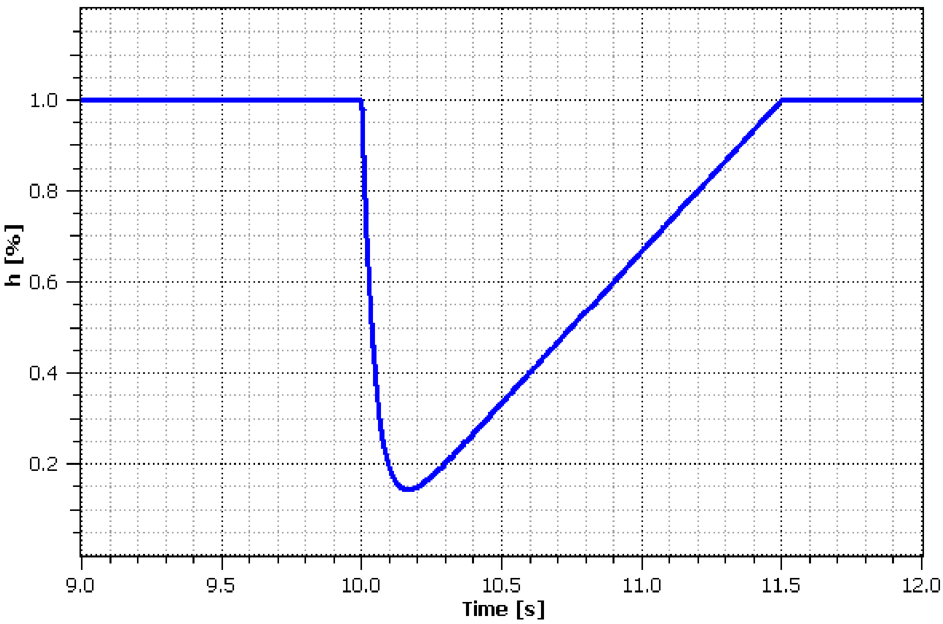

Figure 5.4 The behavior of the modeled disturbances.

5.3 The verification study

The verification study was performed to test the requirement which was phrased; the result from a simulation should agree with reality regarding numerical accuracy. This means that the variable values produced during a simulation, should be close to corresponding measurements of a real system.

This study was performed by modeling a test rig (figure 5.5) which is located at TA and used to test measuring equipment. By comparing the measured and simulated flowrates through different parts of the system, the validity of the solution produced by MathModelica was tested. The reason why this test rig was chosen was both a matter of availability and the fact that measurements easily could be performed; it namely includes plenty of measuring points.

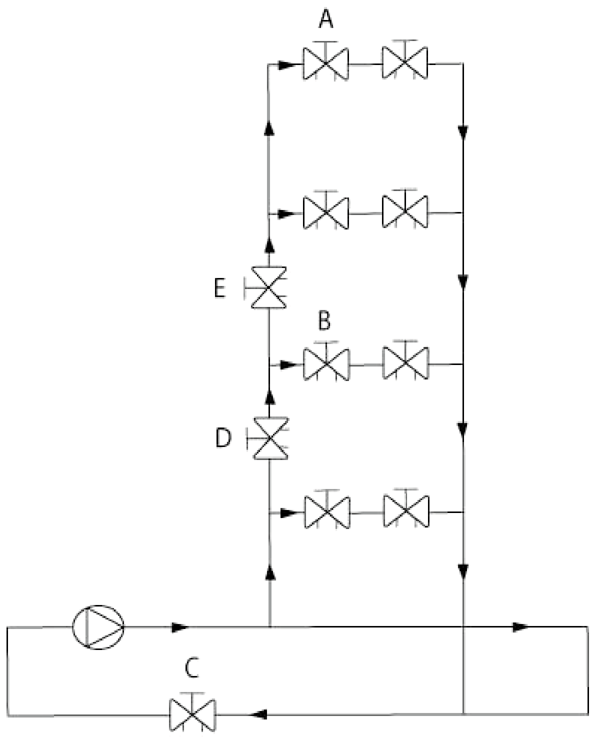

Figure 5.5 The test rig.

The test rig is a closed system, consisting of one riser divided into four branches. Each branch consists of one pair of STAD DN40 balancing valves, and before the inlet to the second and third branch, additional STAD DN40's are located. The main STAD valve, located at the inlet of the pump, is solely of a size DN50.

The frictional losses in the pipes were neglected in this study. This was motivated by the configuration of the test rig; there are a large amount of components in relation to the total length of the pipes which means that the pressure drop caused by the components will be totally dominating.

The pump, whose behavior all pumps in this project were based upon, is a Grundfos Magna 32-120. This pump is semi-controlled in respect to the rotational speed; i.e. the rotational speed can be set manually during operation. In the verification study, this pump was modeled and operated like a constant rotational speed pump by maintaining the rotational speed at its maximum. This means that the modeled pump merely operated in the upper periphery of the valid region. This approach was chosen, since the relation between a certain rotational speed and the resulting pressure rise and volume flowrate could not be determined. The reasons were that neither were such data available from Grundfos nor could corresponding measurements be performed. The latter depended on that the actual rotational speed was not displayed, or could be measured due to lack of instruments. In conclusion, this approach was thought as the only available one, since if different rotational speeds had been included in this study, their real absolute values had to be known in order to re-created for the simulations.

In order to make the study more comprehensive, and naturally also more certain, four different operational cases were included. The first two consisted of one with all valves fully opened and the other with all valves almost closed. These were chosen to represent both high and low flow conditions. In the other two operational cases, half of the valve-pairs were fully opened and the other halves were almost closed. The difference between these two cases was that the valves, denoted as D and E in figure 5.5, were in one case fully opened, and in the other almost closed. The purpose of these cases was to create conditions in which the interactions between the components played a large role, i.e. the pressure drop was larger over some valves and smaller over others. Furthermore, the ability of MathModelica to describe a system which contains both high and low flow regions simultaneously, was tested by decrease the openings of valves D and E.

In order to increase the certainty of the study, measurements for each operational case, were done at three different points in the system. These points are represented by the valves denoted as A, B and C in the figure 5.5. Hence, one point was located in the upper part of the test rig, one in the middle and one in the lower part. They were chosen like this to re-create an as good as possible picture of the system, at the same time as the number of measuring points was minimized. Furthermore, at each point, ten individual measurements were done. And the value used in the comparison between the measured and simulated data was the arithmetic mean value of them. The mean values were computed using equation 5.4[18], in which, x denotes each individual measured value, and n the number of measurements performed. The difference between the measured and simulated values was calculated by using the familiar equation presented as 5.5 below.

$$\bar{x} = \frac{1}{n} \cdot \sum_{j=1}^{n} x_j \qquad \text{(eq. 5.4)}$$

$$E = \frac{x_m - x_s}{x_m} \quad [\%] \qquad \text{(eq. 5.5)}$$

The maximum allowed value of equation 5.5 was determined based on the expected errors deriving both from the measurements and the simulations. These errors were both identified as random (type A errors) and systematic (type B errors) [7]. The type A errors were considered as primarily deriving from the measuring uncertainty, and were taken into account by calculating the standard deviation of the measurements at each point by using equation 5.6[18]. In turn was the standard deviation used in equation 5.7, to calculate the maximum normalized scatter of the measured values around the mean. The resulting values of equation 5.7 described the region around each measuring point in which measured values were expected to be found. These values were chosen as the representation of the type A errors. The type B errors, on the other hand, were identified as primarily dependent on the errors deriving from the computer models and from the manufacturing tolerances of the real components. The errors of the component models were never calculated in this study. Instead was its contribution estimated by looking at the magnitude of the most relevant influencing factors. These factors were the accuracy of the characteristics, how well the characteristics were approximated in the models and the tolerance implemented by the different flow assumptions. All of these are presented in chapter 6. The other factor which was considered as a contribution to the type B errors , namely the manufacturing tolerance of the real components, was represented by the corresponding values of the STAD valves, denoted as U_v in figure 5.6[28]. This approach was motivated by that the system mostly consists of these valves. Figure 5.6 returns the maximum possible deviation in Kv-value for STAD valves at a certain opening. As can be seen, the deviation is larger for a smaller valve opening: this is the reason why TA isn't recommending small valve openings, as discussed in section 5.2.5. Anyway, since the flowrate was the compared variable in this study, and the Kv-value is proportional to the flow; the same maximum deviation due to manufacturing tolerances can be expected for the measured data[21].

$$s_x = \sqrt{\frac{1}{(n-1)} \cdot \sum_{j=1}^{n} \left(x_j - \bar{x}\right)^2} \qquad \text{(eq. 5.6)}$$

$$U_m = \frac{s_x}{\bar{x}} \qquad [\%] \qquad \text{(eq. 5.7)}$$

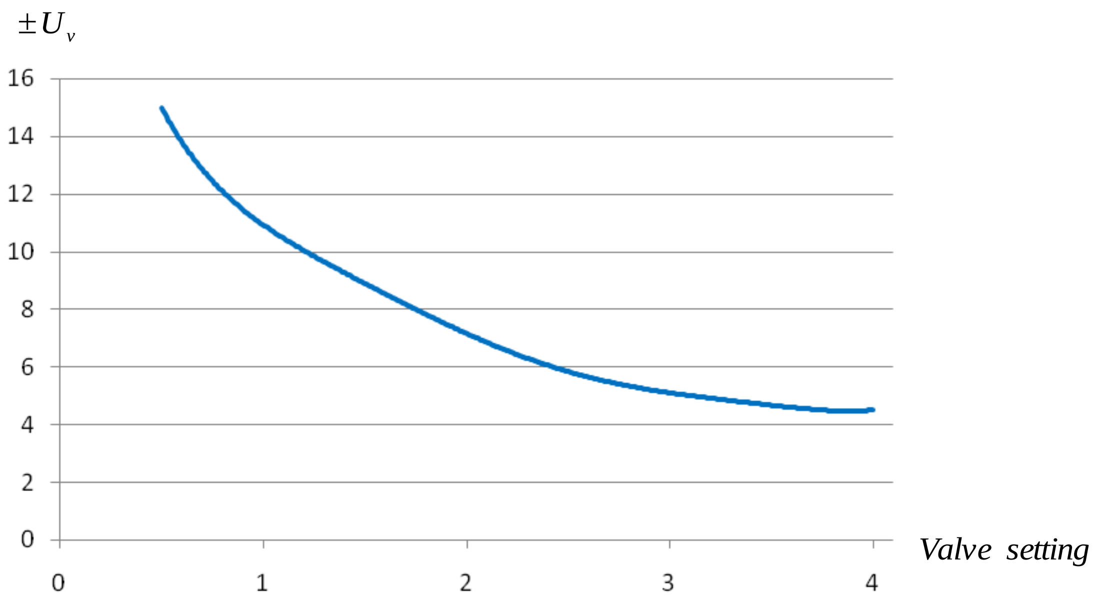

Figure 5.6 Manufacturing tolerances versus the setting of a STAD valve.[28]

6 Modeling approach and resulting component behavior

The primarily modeled behavior of the components was the pressure drop as a function of the volume flowrate. This was of interest since in a system; this relation describes the hydronic interaction between the components. The modeling was done by using mathematical descriptions of component characteristics together with other physical relations.

In this study, the component characteristics were based on measurements performed on the corresponding real component, but, dependent on the type of component, the sources differed. In order to ensure the accuracy for unique components, whose characteristics are specific, they were taken directly from the manufacturer. In those cases, the sources were product specifications, in which the characteristics were presented either in a graphical way or as discrete points. For standard components, the corresponding data are more general and could therefore be taken directly from literature. In table, 6.1 the modeled components along with the origin of the characteristics are presented.

Beyond these, some subcomponents were taken directly from the standard model library of MathModelica.

Table 6.1 Components along with the origin of the model.

Component/subcomponent	Origin
Manual balancing valve, STAD	TA
Automatic balancing and control valve	TA
Radiator valve, RV0-1 DN10	TA
Radiator, Ventil compact CV44	Purmo
Variable speed pump, Magna 32-120F	Grundfos
Constant speed circulation pump, Magna 32-120F	Grundfos
Combining T-junction	Standard
Dividing T-junction	Standard
90° pipe-bend	Standard
Straight pipe	Standard
P-regulator	MathModelica
PI-regulator	MathModelica
Pressure sensor	MathModelica

In this part of the thesis, the focus is on reviewing the component models and their resulting behavior when simulated in a system. Hence, the MathModelica code which was produced during the modeling stage is not reviewed at all. If a deeper understanding of the Modelica language is needed, this, together with some of the model codes, are presented in the company report by Gruber[11].

The component behavior plots shown in this chapter were all produced during the reference study. In order to be consistent, all of them derives from the simulations which were performed of the distribution system including both ABC and balancing valves. The reason why this system was chosen to be representative is that it was the most extensive one. Hence, by including the three scenarios which constitutes the last row of table 5.1, all components were represented.

6.1 Standard component models

The models describing the standard components were collected from two sources; the T-junctions from Idelchik[12] and the pipe-bend and straight pipe from White[31]. Both of these sources had presented their models in a graphical way, which was describing the flow resistance factor as a function of some dimensionless number.

Idelchik was chosen since the graphs illustrated the resistance factor as a function of the volume flowrate. Hence, they matched the working method used in this project perfectly. Also the conditions for which the models are valid were also well documented. Other sources considered were Miller[17], VVS 2000[3] and Alvarez[2] but then the function described was not relevant, the accuracy was considered as too low and the conditions were not specified, respectively.

White was chosen since the resistance factors for the pipe-bend and the straight pipe were illustrated as functions of the Re_d-number. This made the models more general, and hence, the possibility that the models would perform well was increased. Furthermore, curve fits were already performed which simplified the procedure. Other sources considered were Idelchik, VVS 2000 and Alvarez. Idelchik also showed the same kind of function but in this case an accurate polynomial curve fit could not be found. The reason why neither Alvarez nor VVS 2000 were chosen was that the accuracy in both cases was considered as too low.

6.2 Curve fits and accuracy

As mentioned before, the characteristics were based on data, either presented as graphical plots or as discrete values in tables. Either way, the data were initially interpreted as discrete points which had to be approximated into continuous mathematical expressions which could be solved by MathModelica.

The performed approximations were of polynomial curve fit types. The results from the approximations were mathematical equations, expressing the y-axis as a function of the x-axis. And the aim was to including as many discrete points which were presented in the original data as possible. Equation 6.1[26] is a generalization of a polynomial expression, where *n* denotes the order which determines the shape of the curve. Furthermore, the order is also equal to the number of solutions which can be found, and the solutions can be positive, negative, real or imaginary.

$$P(x) = a_n x^n + a_{n-1} x^{n-1} + ... + a_1 x + a_0 \qquad \text{(eq. 6.1)}$$

Generally, it was indicated that MathModelica had problems of finding reasonable solutions when high order polynomials were used in the models. In some cases, simulations of such models simply resulted in equation systems which were unsolvable by MathModelica. However, in most cases, high order polynomials resulted in solutions which were diverging. This might be explained by that the solver did find non-physical solutions which either were negative, unreal or both. If this was the case, this problem might have been avoided if the possibility to decide which part of the solution that should be kept after each iteration were available. However, this could not be done in MathModelica. Because of this, only absolute values of second order polynomials were used. Then the equation systems became relatively simple, at the same time as it was ensured that a real positive solution was kept. Since the characteristics expresses some feature as a function of the flow, the flow could still adopt negative values while the feature only could adopt positive ones. This was considered as a valid approach since all features included were both real and positive in their valid region. The advantage of this approach was that the possibility to end up with a solution which could be kept was increased. On the other hand, the accuracy was omitted since second order polynomials might not be the best approximation in all cases.

The accuracy of the approximations was assessed using the method called "coefficient of determination" which produces a result denoted as R^2. The R^2-value is a measure of how many percentage of the variability in the data that is accounted for in the approximation. This is a widely used measure for determining the adequacy of a regression model, i.e. an empirical model expressing relations between variables not directly developed from theoretical or first-principle understanding of the underlying mechanism[18].

The polynomials are in themselves unrestricted equations. This means that solutions to the polynomials can be found in a region which is stretching from zero to infinity on the y-axis, and from minus infinity to infinity of the x-axis. However, physical solutions can only be found close to the points which were given by the sources. Since it is only here that the real corresponding components have been observed operating. In order to leave out unphysical solutions, the regions in which solutions could be found were limited by the smallest and largest values of the data given by the sources.

6.3 Overall conditions

In this part, the assumptions which were shared by all models and/or systems are reviewed. Most of these assumptions resided during the modeling stage. And primarily, they were done in order to re-create the conditions for which all used equations and characteristics are valid. Besides these overall assumptions, also the equation used to describe the transfer of information between the components is commonly shared by all components. Hence, this equation is also presented in this part of the thesis.

6.3.1 System inertia

The transfer of information between components is not predefined in MathModelica. Hence, in default mode, any changes in one outlet will instantaneously also occur at the next components inlet, which has to be regarded as unphysical.

If a hydronic system is filled, the information is distributed by the means of pressure waves, which causes the flow distribution in the system to change. Since the speed of pressure waves is limited, this interaction is time dependent. This behavior was modeled by assigning equation 6.2 to each component. This equation describes the speed of a pressure wave in a liquid and was derived from equation 6.3[19]. The purpose of the derivation was to include variables which were used in this project.

$$\frac{dp}{d\tau} = \frac{\dot{V}}{V} \cdot \beta \qquad [Pa/s] \qquad \text{(eq. 6.2)}$$

$$c = \sqrt{\frac{\beta}{\rho}} \qquad [m/s] \qquad \text{(eq. 6.3)}$$

Beyond making the solution more realistic, equation 6.2 also unburdens the solver by instructing it to find time-wise solutions instead of a complete one from the beginning. Such approach will further result in more stable simulations.[29]

In equation 6.2, the *β-value* was set to *2.2 GPa* which corresponds to water at 70 °C according R. Fox[8]. In the same equation, *V* denotes the water-filled volume of the current component, and the corresponding estimated values which were used are presented in table 6.2.

Table 6.2 Estimated water volume in the components.

Component	Volume [dm^3]
Manual balancing valve, size DN40	0.4
Manual balancing valve, size DN50	0.5
Radiator valve and radiator	10
Automatic balancing and control valve	0.4
Pump, Magna 32-120F	2
Dividing T-junction	0.1
Combining T-junction	0.1
90° pipe-bend	0.075
Straight pipe	1 / m

6.3.2 Assumptions

The fluid in the systems was described as water. When it came to variations in the density which occurred due to the flow, the fluid was assumed to be incompressible. However, as indicated above, when it came to density changes due to the distribution of pressure waves, the fluid was assumed as compressible. Furthermore, the flow was assumed to be strictly one dimensional spatially, which means that the variables were only allowed to vary in the flow direction. In other words, only the mean value of the velocity and pressure profiles at the inlets and outlets were described by the equations. However, all variables were allowed to vary over time.

The systems in total were set to comply with the continuity equation which, based on the two assumptions above, can be described as in equation 6.4[31]. In this equation, the indexes *i* and *o* denotes the inlet and the outlet of the system, respectively. In this perspective were the inlet and outlet of course non-existing, since the systems in total were closed and hence, no water was either added or removed. Consequently, equation 6.4 became equal to zero if applied to a complete system. The reason why equation 6.4 looks like it does is that ingoing flows always were defined as positive and outgoing as negative. This feature is predefined in Math-Modelica and was hence used throughout the whole project.

$$\sum_j \left(\rho \cdot \dot{V}_j\right)_i + \sum_j \left(\rho \cdot \dot{V}_j\right)_o = 0 \quad [kg/s] \qquad \text{(eq. 6.4)}$$

Equation 6.4 implies that the mass flow through the outlets is equal to the mass flow through the inlets at all times. This was, on the other hand, only true for single components during steady state. The reason was the pressure inertia which was described by equation 6.2 and placed between the inlet and the outlet of all components. Since this pressure inertia also acted as inertia for the flow, the flows through the outlets were lagged compared to the flows through the inlets. Hence, small differences between the outgoing and the ingoing flow of the components could be found during the unsteady states.

Since the assumption that the fluid was incompressible regarding the flow case, the density was assumed to be constant. This was also assumed for the other properties of the fluid. As mentioned in part 5.2.5, all fluid property values were taken at a temperature of 70 °C. This corresponded to the design mean value of the implied inlet and outlet temperatures of the radiators which was given in the product specification by Purmo[44].

Finally, the models that were dependent on the Re_d-number were only valid in a region between 4000 and 10^5. Since the transition between laminar and turbulent flow was assumed to occur at approximately 2300[31], the flow was assumed to be fully turbulent at all times. This was also true for most of the solved time-steps, since the flow velocities in most cases were relatively high and the pipes generally were relatively small. However, this implies that the models initially were out of their valid region since the flow velocities in the systems began at approximately zero. For that reason, the initial part of the solutions was considered as unreliable, and the solutions were not taken into account until the flow had reached the lower valid Re_d-number.

6.4 Valves

The feature of a valve is that its opening can be adjusted, either manually or automatically, which causes a change of flow resistance. Thereby the fluid flow can be controlled in a way which results in a desired behavior of the corresponding system. The valve sizes are usually given as the nominal diameter which is denoted DN. The nominal diameter of a valve is the size of the internal diameter of the inlet and outlet given in mm.[30]

As mentioned before, the characteristic of a valve is specific to the manufacturer. This means that even if two valves are of the same type, large differences can occur between them if the brands are different. This means that general models are hard to produce and instead, it was decided to use measured data, produced at TA, to describe the characteristics of the valves. Consequently, the behaviors of all valve models in this study were based on actual valves produced by TA.

The characteristics of a valve is usually given as the Kv-value as a function of the valve opening. The definition of the Kv-value was previously presented as equation 4.3, which now can be rewritten as equation 6.5 due to the assumptions which since then have been made. This is now possible since the reference pressure drop, Δp_0, is equal to one and that the fluid was assumed to be incompressible water. Hence, since the reference density, ρ_0, refers to water and the density was assumed to be constant, the density ratio consequently becomes one[5].

$$k_v = \frac{\dot{V}}{\sqrt{\Delta p}} \qquad \left[\frac{m^3}{h} \Big/ \sqrt{bar}\right] \qquad \text{(eq. 6.5)}$$

6.4.1 Balancing valves

The modeled balancing valve was based on TA´s product called STAD which is a manual plug valve containing measuring nipples. It was modeled in four different sizes and their characteristics along with the accuracy of the approximations are shown in figure 6.1 and table 6.3, respectively[28] [20].

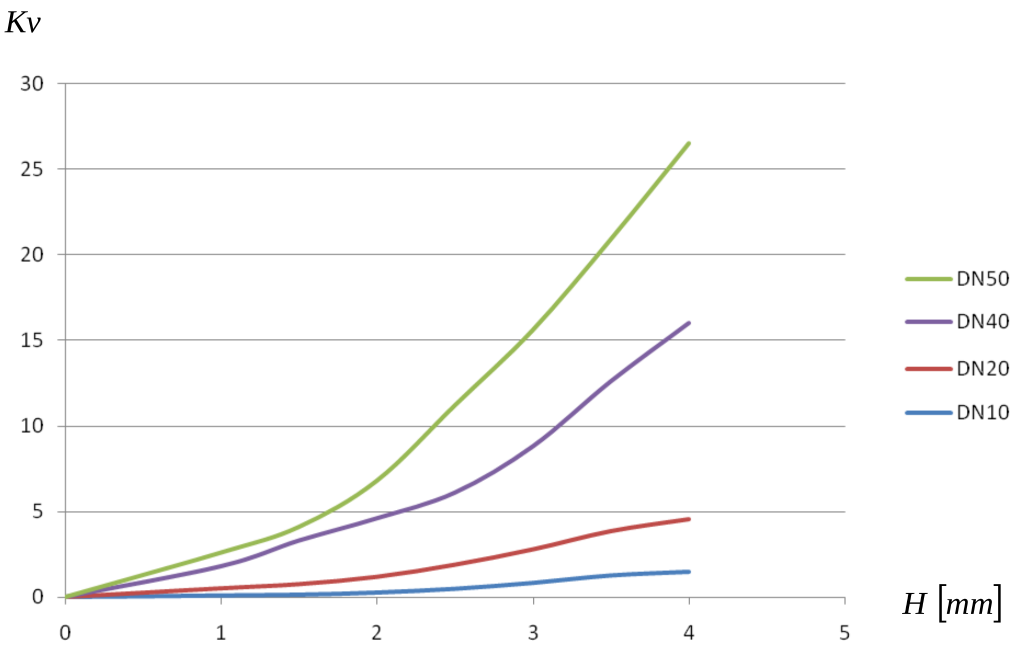

Figure 6.1 STAD characteristics.

Table 6.3 Accuracy of the STAD polynomial approximations.

Size	R^2 value
DN10	0.988
DN20	0.995
DN40	0.995
DN50	0.999

The primarily modeled feature of the balancing valve is shown figure 6.2. In this figure, the pressure drop is plotted against the volume flowrate for two different valve settings of a STAD DN40. What can be seen is that the pressure drop for a certain flow became larger as the valve opening became smaller. Hence, the flow resistance of the valve was larger for a smaller opening. In this case, the openings correspond to 1 mm (continous line) and 2 mm (dashed line).

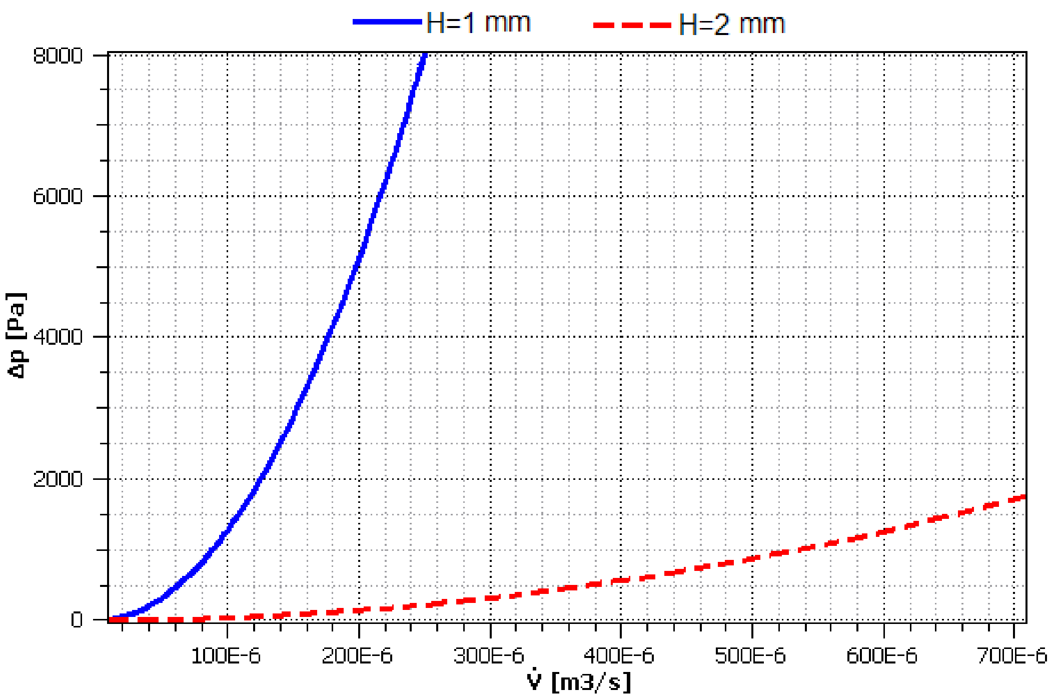

Figure 6.2 Primarily modeled feature of a balancing valve.

6.4.2 Automatic balancing and control valve

In contrast to balancing valves which have a fixed opening, control valves are able to change their flow resistance depending on the prevailing situation. The modeled control valve was based on an automatic balancing and control valve (ABC) which currently is under development by TA. In this context, “automatic” refers to that the valve is a flow limiter that adjusts its opening automatically depending on the actual differential pressure across it. In turn, “balancing” refers to that the valve is designed to maintain a constant flow in the circuit which it controls.

The ABC consists of two separate cone valves which are connected in series as shown in figure 6.3. The left one in figure 6.3 is a differential pressure controller (DPC), and the right one is a control valve. A feature of the ABC is that a desired magnitude of the flow can be obtained by adjusting the opening of the control valve, independent on the opening of the DPC. Furthermore, by controlling the opening of the control valve in respect to the actual room temperature, the ABC valve can operate during full as well as part load.

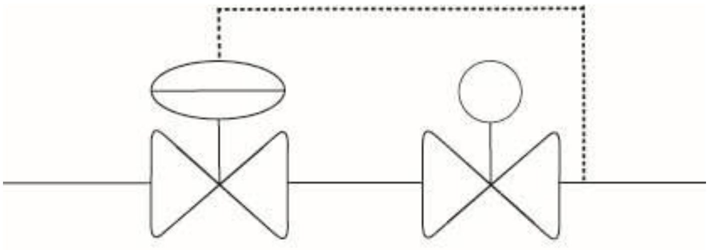

Figure 6.3 Schematics of the ABC valve.

As mentioned in parts 5.2.3, the system which contained the STAD’s should approximately represent a design flow case, and the system also containing the ABC valves should represent a part load case. As mentioned in part 5.2.5, the value of the design flows through the thermal units were constant, specific and determined by assigning design thermal outputs of the radiators. A part load flow, on the other hand, can be non-specific, variable and virtually take on any value; as long as the value is lower than the design flow of the corresponding thermal unit. Since it did not really matter what flowrates that were chosen to represent the part load case, the flowrates which corresponded to the simplest modeling approach were chosen. This approach was to set the openings of the control valve to an arbitrary, constant level of 2 mm. This approach resulted in that the mean value of the steady-state flowrates were the same through all thermal units during the part load case. The reason was that the DPC’s adjusted their openings so that the pressure drops over the control valves were the same for all ABC valves in the system. And since all control valves had the same settings, the volume flowrate which corresponded to this pressure drop also were the same. There were two other options considered which were ruled out for different reasons. First, the part load flows could have been determined individually for each unit, based on estimated room temperatures. But, this was considered as outside of the scope, since temperature variables were left out. Second, to set the part load flow of each thermal unit based on the corresponding design flow, for example 80 %. This approach would definitely have been more relevant. But, the models of the ABC valve would in that case have to include yet another control loop and would naturally become more complicated. Hence, the current approach was chosen partly due to lack of time.

The procedure of the DPC is to achieve a constant flow by maintaining a constant pressure drop of 10 kPa over the control valve. This value is predetermined in the manufacturing of real ABC valves, and was hence given as input data and used throughout this study. This modeled behavior is shown in figures 6.4, 6.5 and 6.6.

In the first figure, both the pressure drop over the DPC and the control valve is plotted against time. It can be seen that the desired pressure drop of 10 kPa over the control valve is reached after about 2.3 seconds. However, this was the case when the total inertia of the system was included. The individual response time of the ABC was modeled as about 0.5 seconds which was given as input data by TA. In the next figure, the volume flowrate through the ABC valve located before the thermal unit denoted as A2 in figure 5.3, is plotted against time, together with the volume flowrate through the thermal unit denoted as A3. What is illustrated is the effect of the disturbance introduced by the radiator valve belonging to the thermal unit A3. This disturbance occurred at the simulated tenth second, and was causing oscillations of the volume flow through the corresponding thermal unit A3. However, as can be seen, the volume flow through the ABC was unaffected during the same period. This was the case despite of the fact that this ABC was connected to the same T-junction as the thermal unit A3. This changed flow level in the T-junction was compensated by the ABC by altering the valve opening of the corresponding DPC from time to time. Because of this compensation, the momentary pressure drop which corresponded to the desired flow, set by the opening of the control valve, could be achieved although the disturbance. This procedure is illustrated in the final figure where the pressure drop over the DPC during the same period of time is shown.

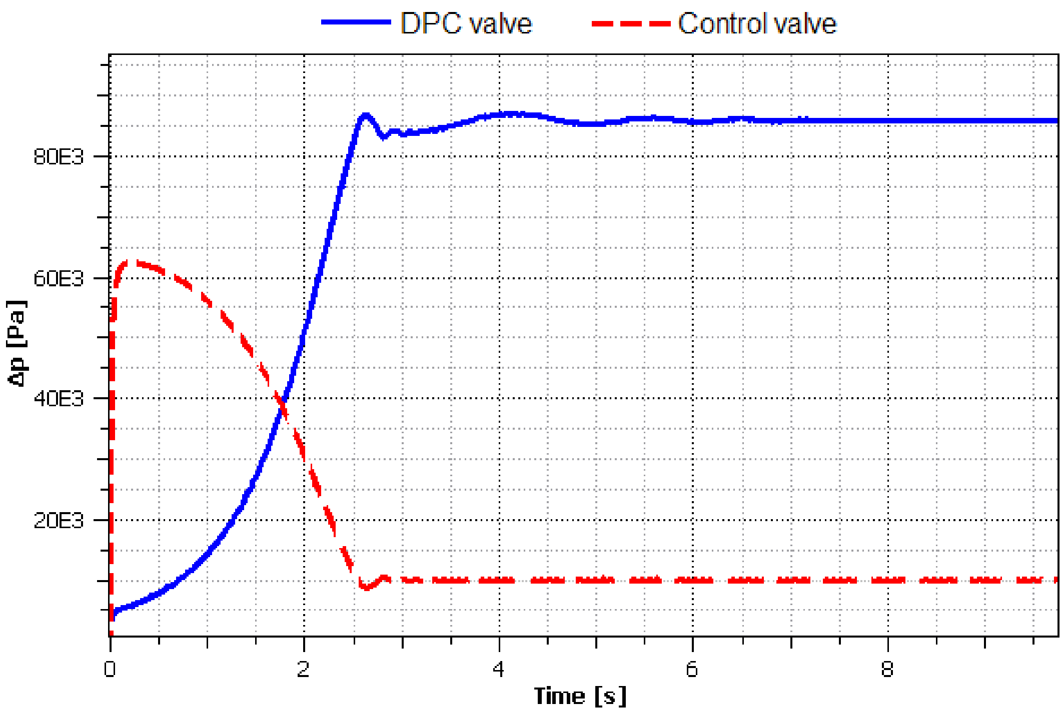

Figure 6.4 Working procedure of the modeled ABC valve.

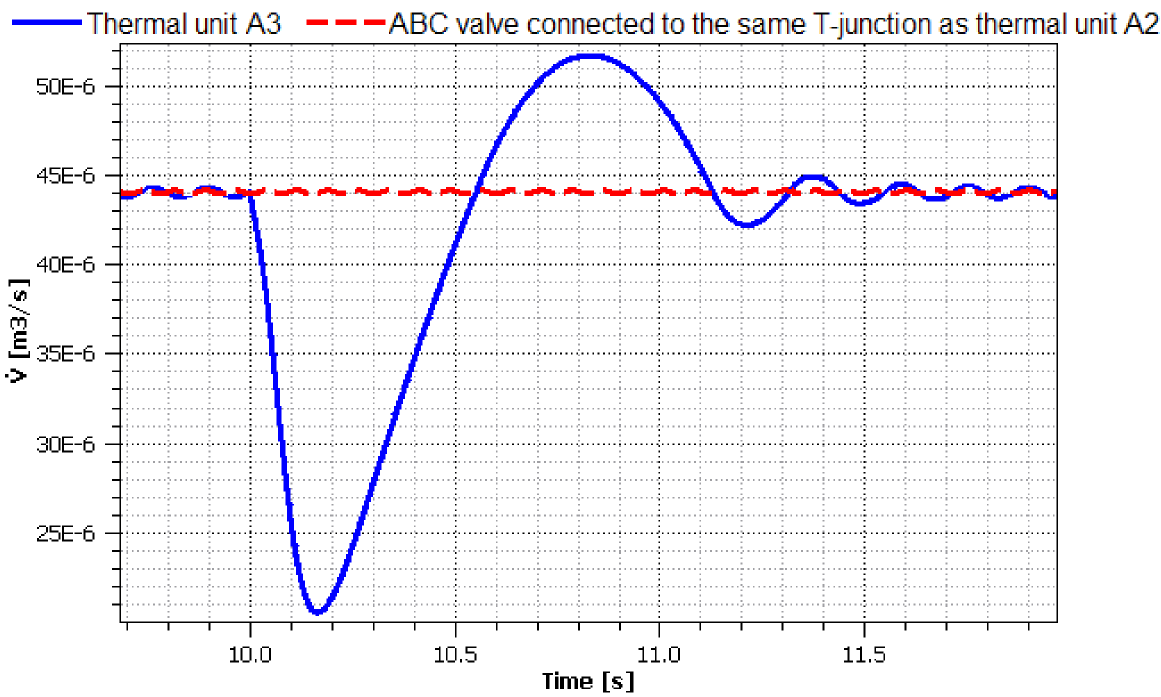

Figure 6.5 Fluctuations of volume flowrate through a modeled thermal unit and a modeled ABC valve during a disturbance.

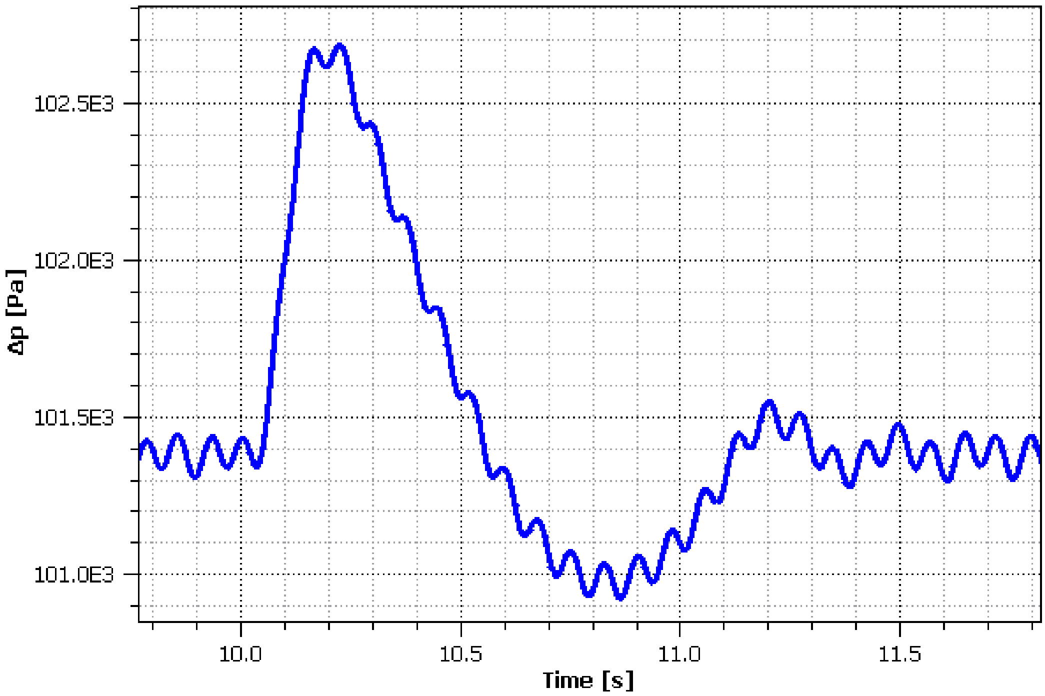

Figure 6.6 The corresponding fluctuation of pressure drop over the modeled ABC valve.

In principle, the control mechanism of the ABC consists of a membrane, whose upper chamber is connected to the outlet of the control valve by capillary tubes. The lower chamber is filled with a non-flowing liquid and in turn connected to the top of the DPC's cone, also by capillary tubes. Hence, the force that the membrane subjects on the DPC cone is dependent on the pressure difference between the outlet of the control valve and the liquid. At the bottom of the DPC's cone a spring acts, and when the pressure at the outlet of the control valve becomes higher than the force from the spring, the liquid presses down the cone of the DPC. The effect of the reduced opening is that the flow through the ABC is decreased, which in turn results in a reduced pressure drop over the control valve. When the value of the desired flow is changed by adjusting the opening of the control valve, the pressure level of the upper chamber is changed and hence also the P-band of the control mechanism. However, not even during steady state the opening of the DPC can achieve an exact pressure drop of 10 kPa over the control valve. The reason is that there is some dead-time between the DPC and the control valve which results in that the desired opening of the DPC is constantly overshot. This means that the actual flow will oscillate around the desired flow, which also has been described in the model and can be seen as the low frequency oscillations in figure 6.5 and 6.6.

The characteristics of the DPC and the control valve are shown in figure 6.7 and 6.8, and the corresponding R^2-values of the approximations are about 0.994 and 0.984, respectively. In the model, the control mechanism was approximated by a feedback loop containing a static P-regulator. By using this type of regulator, it was assumed that a certain change of the pressure drop should, at all times, result in the same rate of change of the valve opening. This was considered as reasonable since ideally the valve opening of the DPC is in direct relation to the pressure drop the control valve. However, this might not be completely true since the magnitude of the friction force, subjected on the DPC's cone, is dependent on its position. Hence, the required force to move the cone varies, and the controlling mechanism might not be linear. However, this effect is assumed to be negligible at this point[5].

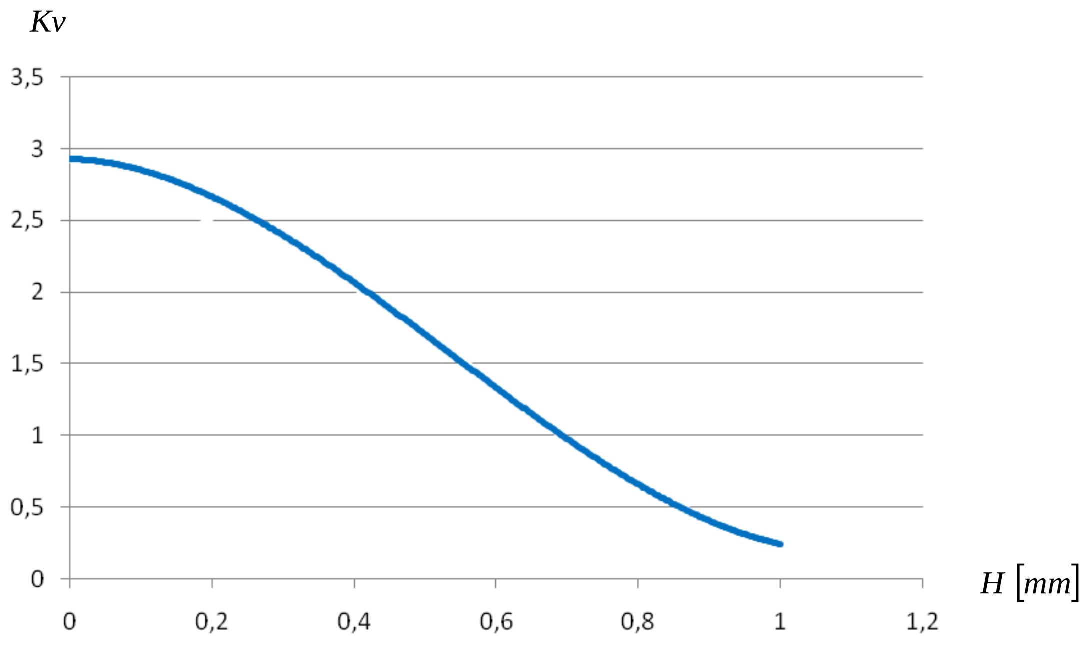

Figure 6.7 Characteristics of the DPC.

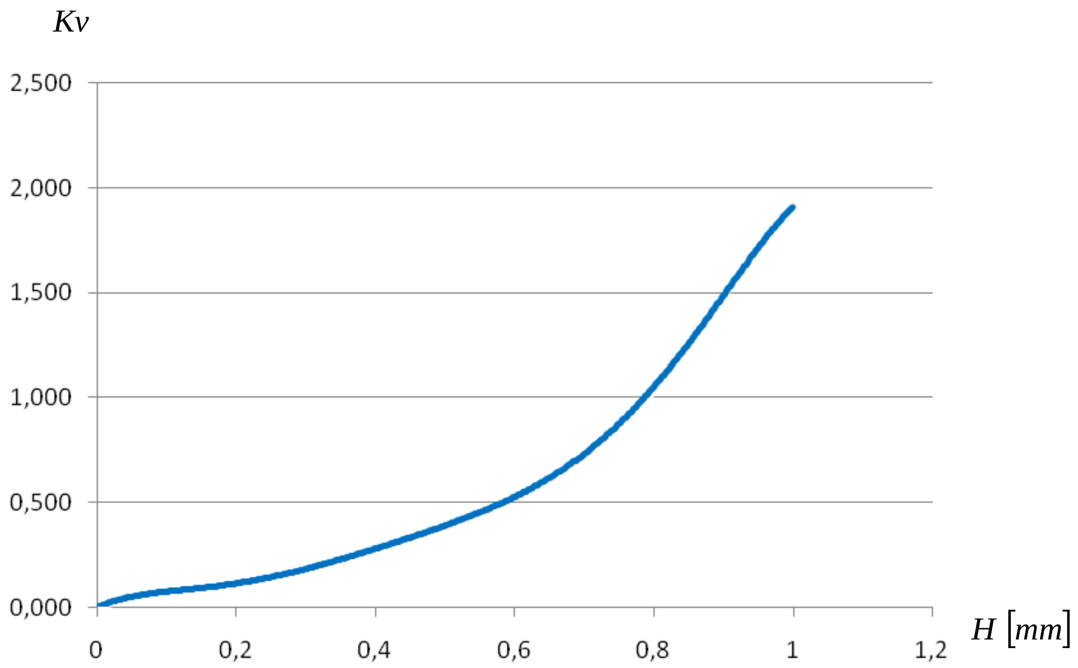

Figure 6.8 Characteristics of the control valve.

The gain of each P-regulator, *Kc*, was estimated by using equation 6.6[5]. This equation was used to determine gain values which matched the system in order to avoid unstable control. The numerator of equation 6.6 was interpreted as the available change of the DPC´s opening which was equal to 1.2 mm for all ABC valves. When it came to the denominator, it was interpreted as the possible change of the controlled variable, which in this case was the mass flowrate through the corresponding thermal unit according to equation 5.2. The lower limit of $\Delta\dot{M}$ appeared when the corresponding opening of the DPC's was zero, and was consequently also zero for all ABC valves. The upper limit, on the other hand, appeared when the opening instead was at its maximum, i.e. 1.2 mm. This scenario corresponded to the flow distribution which was achieved during the balancing procedure, i.e. all the DPC cones were fully opened and the openings of the control valves were set to 2 mm. Hence, the upper limit corresponded to the design flows, as presented in table 5.2 for each thermal unit. Consequently, the gains of the controllers became different from each other and are presented in table 6.4.

$$K_c = \frac{\Delta H}{\Delta \dot{M}} \qquad [mm/(kg/s)] \qquad \text{(eq. 6.6)}$$

Table 6.4 The gains of the controllers.

ABC valve belonging to the thermal unit as denoted in figure 5.3	$\Delta H\ [mm]$	$\Delta \dot{M}\ [kg/s]$	K_c
A1	1.2	0.0896	13.4
A2	1.2	0.0777	15.4
A3	1.2	0.0896	13.4
B1	1.2	0.0478	25.1
B2	1.2	0.0896	13.4
B3	1.2	0.0774	15.5

The modeled ABC contained two ideal (no pressure drop) pressure sensors that were connected to the inlet respectively the outlet of the control valve. The difference between the signals was send to the controller that controlled the time derivative of the DPC's opening. Between the controller and the DPC a signal inertia was modeled. This inertia resulted in a time delay between the controller and the cone, and was thought to represent the dynamic behavior of the ABC valve. This was done by using equation 6.7, in which u is the input signal, y the output signal, b is a damping constant, k_s is the spring constant and M is the mass of the cone.

$$u = k_s \cdot (y + y_0) + b \cdot \frac{dy}{d\tau} + M \cdot \frac{d^2 y}{d\tau^2} \qquad \text{(eq. 6.7)}$$

$$F = k_s \cdot y + b \cdot \frac{dy}{d\tau} + M \cdot \frac{d^2 y}{d\tau^2} \qquad \text{(eq. 6.8)}$$

Equation 6.7 is based on equation 6.8[16], which can be illustrated as an oscillation with dampened resonance. This equation describes the dynamic behavior of a spring-mass system when the spring is viscously dampened, and it was assumed that the DPC could be considered as such a system. When comparing theses two equations, it can be seen that the input signal should be interpret as the force from the membrane that acts on the cone. Furthermore should the output signal be interpreted as the position of the cone. The counter-forces on the right hand side of equation 6.7 were assumed to derive from the spring located at the bottom of the real DPC's cone, the viscous damping of the liquid that is located between the membrane and the cone, and finally the inertia of the cone´s mass. The constant y_0 was introduced since even when the cone was at its upper location, which corresponds to y=0, the spring still acted with a certain force. The constants in equation 6.7 are not reviewed in this study since the product is under development and they are still classified. However, it can be mentioned that the mass and spring constant, along with y_0 were known from the beginning, and the damping constant was tuned to correspond to the observed behavior of the ABC.

In figure 6.9 the resulting dynamic behavior of the DPC´s cone as described by equation 6.7 is shown. The figure illustrates the end of the unsteady state which begins when the system is initialized. What can be seen is that the opening of the DPC, which initially is set to 1.2 mm (fully opened), overshoots the desired value about seven times before the steady state, characterized by the constant periodic oscillations, is found.

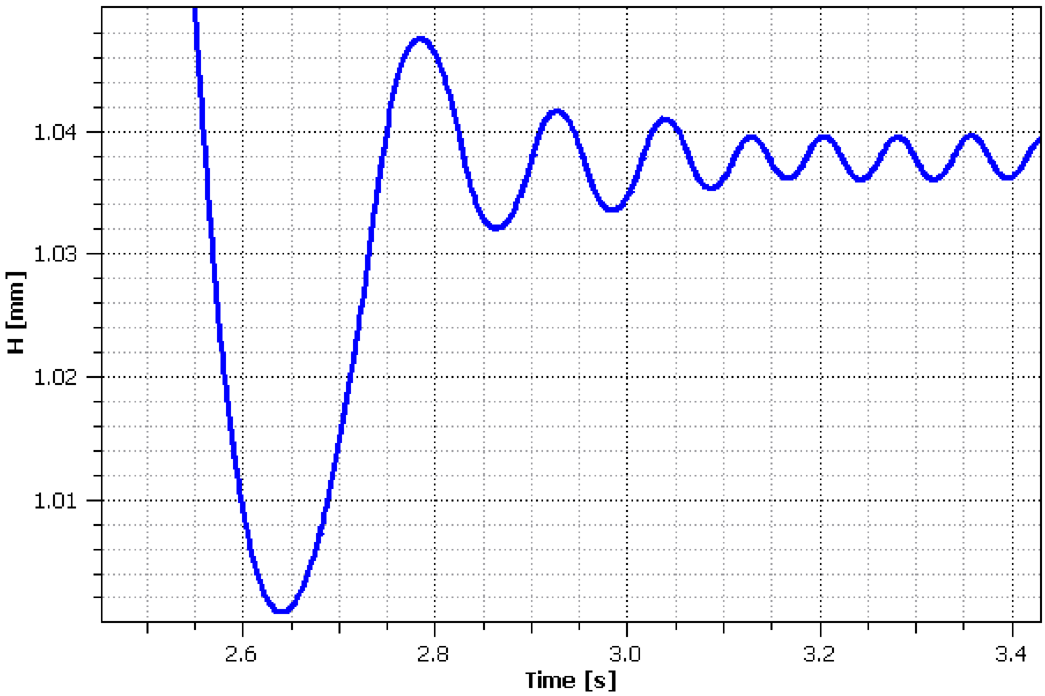

Figure 6.9 Dynamic behavior of the modeled DPC, valve opening versus time.

6.4.3 Radiator valves

The radiator valves were based on the product RV0-1 of size DN10, produced by TA. They were modeled without thermostat function and hence functioned as manual valves in this project. The reason was of course that thermal interactions were not included in the models.

The default setting of the radiator valve openings were set to one, which corresponded to fully open and a Kv-value of 1.7. For most of the radiator valves, this setting was treated as a constant throughout the simulations. However, this was not the case for the two valves which generated the disturbances in the system: a disturbance consisted of that one of these two valves first reduced its opening to a level which corresponded to a Kv-value of 0.07 and then back to fully open again (see section 5.2.6).

The modeled dynamic behavior of a radiator valve when the opening was changed during a disturbance corresponded to a system including a time constant. This means that the valve opening reached its final value first after a certain amount of time, but without overshooting like in the case of the ABC valve. This behavior is described by equation 6.9 and the result has already been shown in figure 5.4.

$$H(t) = H_0 \cdot \left(1 - e^{-\left(\frac{\tau}{\tau_c}\right)}\right) \qquad [mm] \qquad \text{(eq. 6.9)}$$

$$y(t) = K \cdot \left(1 - e^{-\left(\frac{\tau}{\tau_c}\right)}\right) \quad \text{(eq. 6.10)}$$

Equation 6.9 is based on equation 6.10[16] which is visualized in figure 6.10 as a step response of a first order system including a time constant. As can be seen in this figure, *y(t)* is the time dependent output signal and *K* is the final value, translated as *H(t)* and H_0 in equation 6.10, respectively. Furthermore, the time constant τ_c is defined as the elapsed time until *H* has reached a value corresponding to 63 % of H_0.

The purpose of introducing equation 6.10 into the radiator valve models was to make the shape of an original disturbance more realistic. The original disturbance was modeled by a when-loop which stated that at a certain time, *H* should be reduced from fully opened to a certain value and then back again, all during 1.5 seconds. This resulted in that the total changes of *H* occurred instantaneously which resulted in that its time derivative at times became infinite. Instead, equation 6.10 was used to smooth these changes, and the result is shown in figure 5.4. For that reason the time constant was chosen to relatively small value of 0.05 seconds, since the primarily time dependent change, i.e. the 1.5 seconds was achieved by the when-loop.

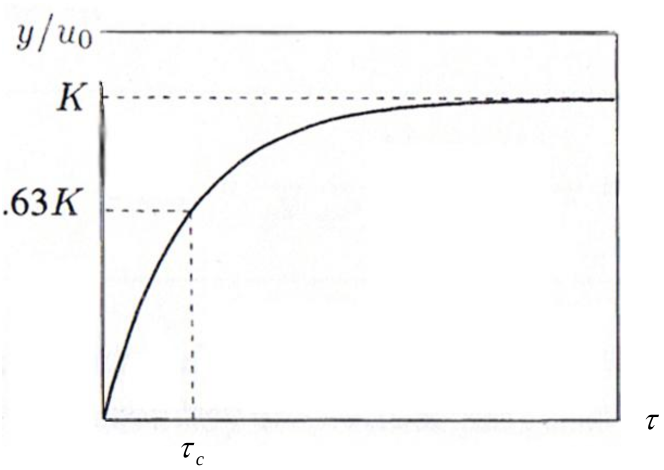

Figure 6.10 Step response of a first order system with a time constant.[16]

6.5 Pipe-bend

All bends in this study were of 90° and their diameter were set either to 0.01 or 0.04 m, dependent on the size of the majority of valves included in the system. In the verification study, they were placed where bends actually were found in the test rig, and in the reference study, at locations where it was estimated that a change of the flow direction was needed.

The losses related to the turning of the flow are both due to changes of the static pressure and of the velocity distributions that occur inside the bend. Furthermore, secondary flow, flow separation and eddies are generated[17]. The corresponding pressure drop can be described by combining equation 6.11 and 6.12, where 6.11

is a curve fit already performed by the White[31] based on measurements. The simulated result of these equations is shown in figure 6.11 where the pressure drop is plotted as a function of the volume flowrate.

$$k = 1.49 \cdot \mathrm{Re}_d^{-0.145} \qquad \text{(eq. 6.11)}$$

$$\Delta p = k \cdot \frac{\rho \cdot u^2}{2} \qquad [Pa] \qquad \text{(eq. 6.12)}$$

The dynamic viscosity used to calculate the Re_d -number was set to $4.1 \cdot 10^{-7}$ m^2/s, which corresponds to water of 70 °C [31]. For more information about the equations above, see part 4.2 in this thesis.

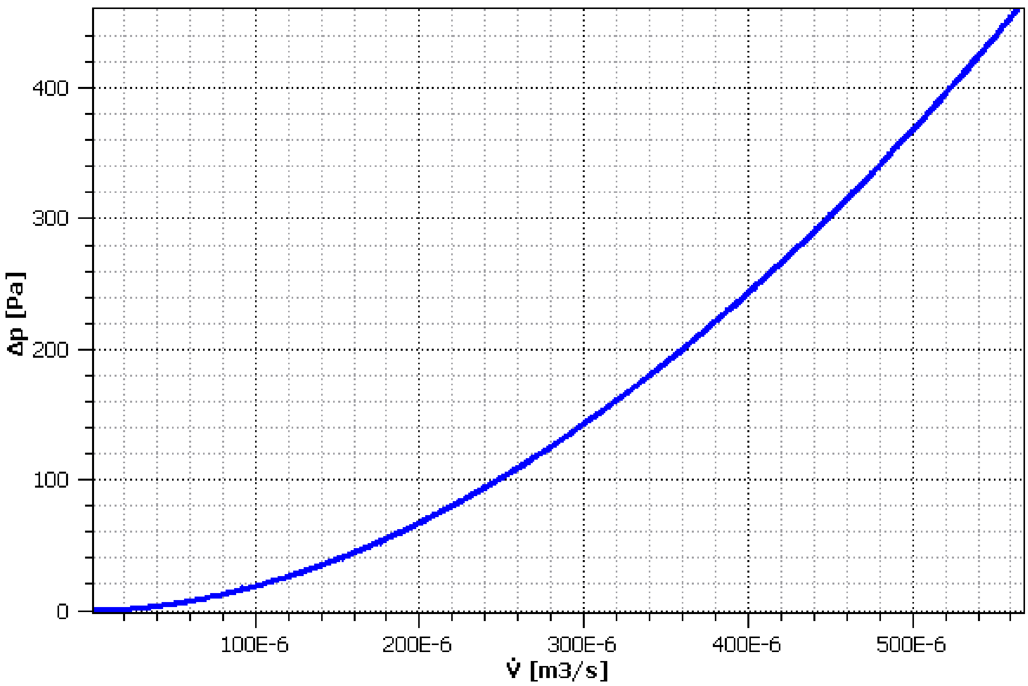

Figure 6.11 The pressure drop as a function of the volume flowrate for the modeled pipe-bend.

6.6 Straight pipes

The straight pipes were modeled in a similar way as the pipe-bends, but with the exception that also the length was introduced as a parameter that affected the pressure drop. In the reference system, pipes were not placed at all locations where such were expected to be found, but the aim was instead that the total length should correspond to an estimated total pipe length in a corresponding real system.

Losses in straight pipes occur since the fluid rubs against the pipe walls and, due to the friction, some of the velocity is transformed into internal energy. The equation which describes the corresponding pressure drop is presented as equation 6.13[31], in which the velocity should be the mean value over a cross-section of the pipe. This equation is valid for fully developed flow, both turbulent and laminar, but the mean value, when derived from the maximum value of the velocity profile, is calculated differently in those two cases. However, as it was mentioned in part 6.3.2, the flow was assumed to be one-dimensional. This resulted in that the calculated volume flowrate, and hence also the velocity, already was the mean of the velocity profile. Hence, this enabled equation 6.13 to be rewritten as equation 6.14[31]. In this equation, L is the pipe length and d is the inner diameter which was set to the same size as the majority of valves included in the system.

$$\Delta p = k \cdot \frac{L}{d} \cdot \frac{\rho \cdot \bar{u}^2}{2} \quad [Pa] \qquad \text{(eq. 6.13)}$$

$$\Delta p = k \cdot \frac{L}{d} \cdot \frac{\rho \cdot u^2}{2} \quad [Pa] \qquad \text{(eq. 6.14)}$$

The k-value was calculated using equation 6.15[31], which is a curve-fit of the Moody diagram. The Moody diagram describes the k-factor for a pipe as a function of the Re_d-number and the surface roughness. However, equation 6.15 is only valid for pipes with smooth surfaces, which hence all pipes in this project were assumed to have. This assumption might not be the most realistic one, since all pipes have a surface roughness, but it was chosen due to its simplicity. If the roughness were to be included, the expression would become more complicated, so for now, this assumption is considered as relevant.

$$k = 0.316 \cdot \mathrm{Re}_d^{-\frac{1}{4}} \qquad \text{(eq. 6.15)}$$

The result of equation 6.14 in combination with 6.15 is shown in figure 6.12 where the pressure drop is plotted as a function of the volume flowrate for a modeled straight pipe.

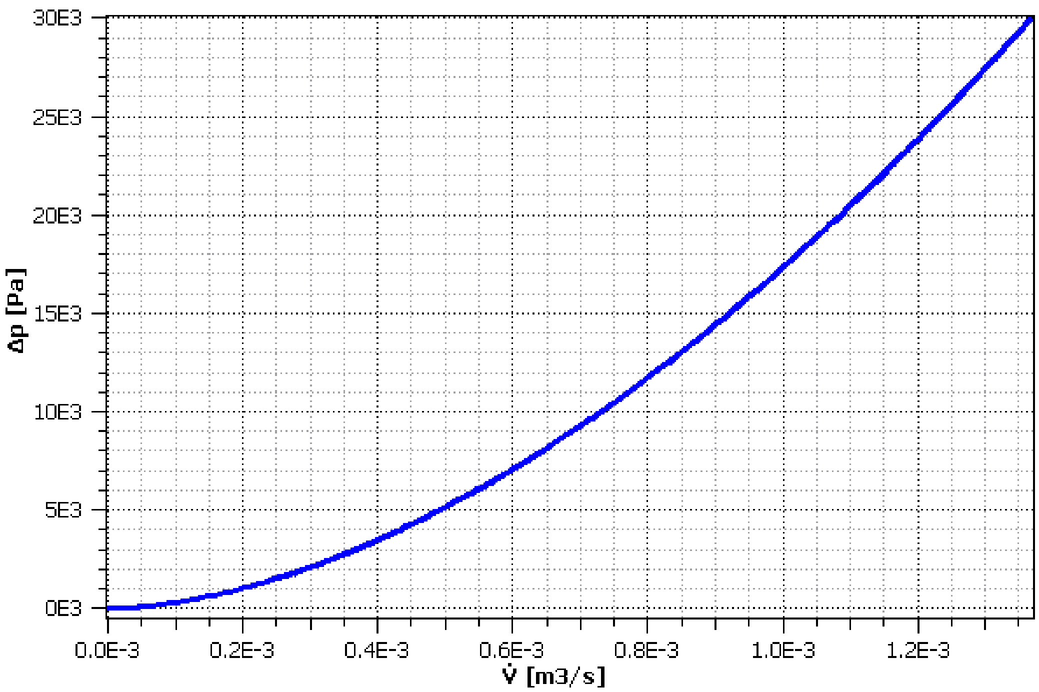

Figure 6.12 The pressure drop as a function of the volume flowrate for the modeled straight pipe.

6.7 T-junctions

The modeled T-junction had three different openings, as seen in figure 6.13. The T-junctions were either used to divide one stream into two, or merge two streams into one. Specific for this study, the difference between the merging and dividing T-junction was that the direction of the flow through the branch denoted as C changed direction, i.e. the flow took either on negative or positive values as discussed in part 6.3.2.

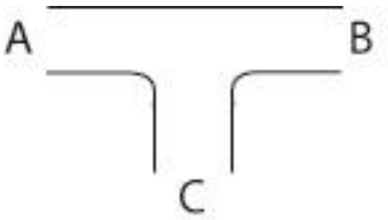

Figure 6.13 Schematics of a T-junction.

For a general T-junction, the resistance factor mainly depends on the angle between the branches C and A as denoted in figure 6.13, the area relation between all branches and the volume flow ratio through them. The losses mainly occur due to shocks, i.e. irreversible losses, which are caused by sudden expansions, frictional losses and turning of the flow. Furthermore, the turning of the flow also causes losses due to flow separation and production of eddies. Also for merging T-junctions, a turbulent mixing of two streams occurs. These might have different velocities and if the difference is large, a tangible transmittance of kinetic energy called an ejector effect occurs. This might results in shocks as well as an internal acceleration of the stream with the lowest velocity[12].

In “Handbook of hydraulic resistance”, Idelchik[12] has presented plenty of data for T-junctions. He has for example chosen to present characteristics in which the resistance factor is dependent on the radiuses of the pipes, the flow between different branches, the different angles between the branches etc. Generally, these characteristics were presenting the k-factor as a function of the ratio between the volume flows through the branches in question. And the returned k-factor should be used in equation 6.12 to calculate the corresponding pressure drop. An example of a T-junction characteristic presented by Idelchik is shown in figure 6.14, which illustrates the k-factor between branches C and B as denoted in figure 6.13. This characteristic is specific for a merging T-junction with 90 ° angle between the branches, and pipes of equal radius. Here, the ejector effect is visible as the negative pressure drop, which occurs when the flow through B is much larger than the flow through C. When this is the case, the flow through A is very large and acts as an ejector of the flow through branch B. However, this was practically the only case among the T-junction included in this study, where the ejector effect was visible. In the other cases, only positive pressure drops were represented.

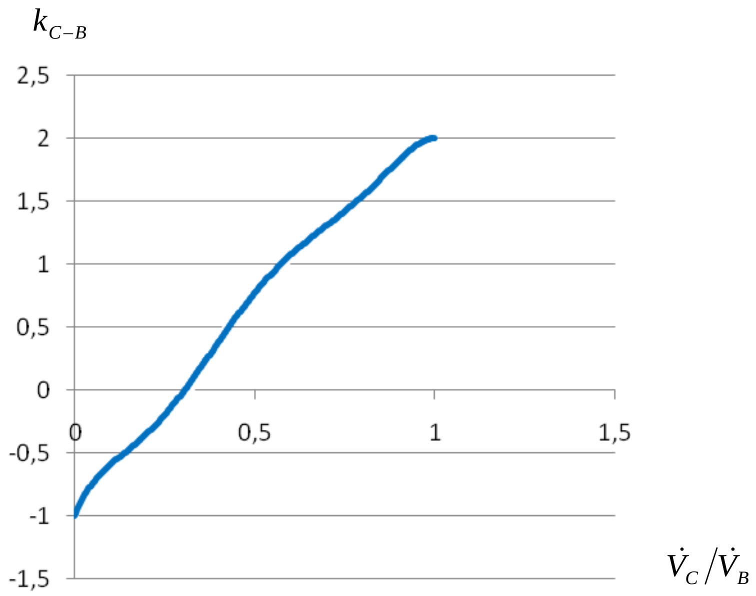

Figure 6.14 Characteristic of a T-junction. [12]

However, this was not the approach used to model the T-junctions included in this study. The reason was that MathModelica could not find solutions to the polynomials derived from the Idelchik characteristics. Furthermore was also a stepwise linear version of the polynomials tried. In such form could the solver on one hand handle the equations, but on the other, the produced solutions were always diverging. Instead were the T-junctions approximated by a straight pipe and a pipe-bend connected in parallel. This might be a source of error, even if the length of the straight pipe was tuned to, as well as possible, fit the actual pressure drops presented by Idelchik. For example were both the ejector effect and the specific shape of the characteristics failed to be described. Furthermore was it assumed that all branches had the same diameters, which were equal to the size of the majority of the valves included in the actual system.

One specific feature of a T-junction is shown in figure 6.15, where the steady-state volume flowrates through the branches of a merging T-junction is plotted. Here, it can be seen that outgoing flows (branch B) are defined as negative and ingoing flows (branch A and C) as positive. Furthermore can it be seen that the sum of the flowrates through A and C is equal to the absolute value of the flowrate through B. This shows that the continuity equation applies in this case.

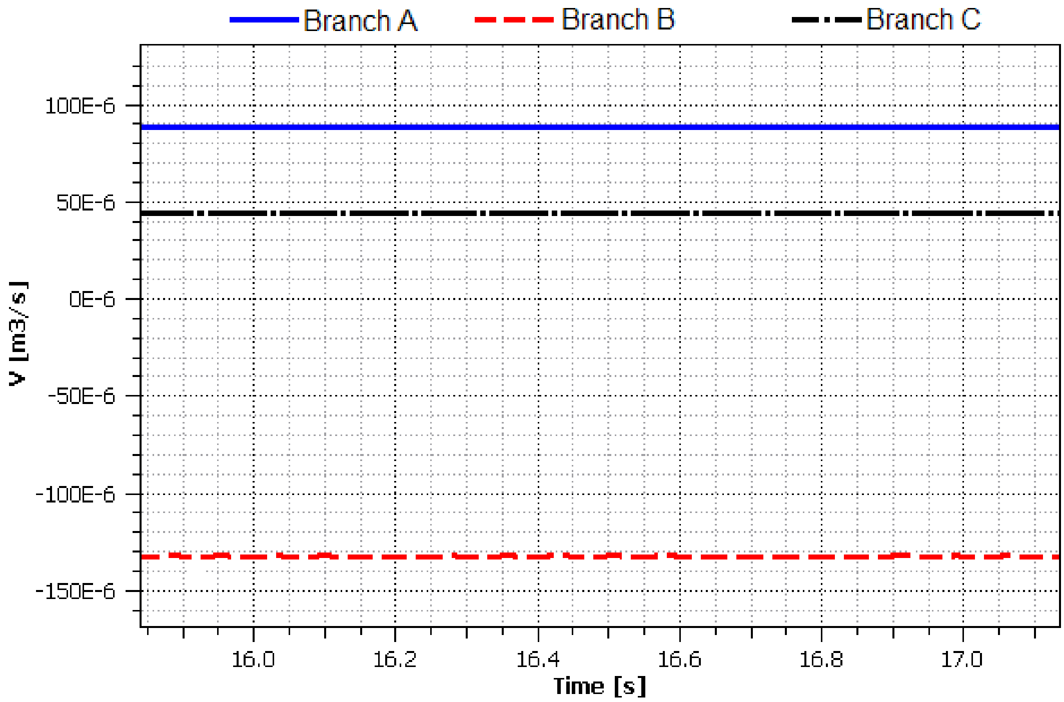

Figure 6.15 Characteristic feature of a modeled merging T-junction.

6.8 Pumps

All three pumps in this project were modeled by using the same characteristics. The characteristics were represented by a pump and a power curve, and both of them derived from the Grundfos Magna, located in the in the test rig used during the verification study (see section 5.3). These two curves are shown in figure 6.16 and 6.17, and the corresponding R^2-values of the approximations were calculated to 0.996 and 0.718, respectively. As indicated, the approximation of the power curve is relatively inaccurate which further on is taken up as a possible source of error.

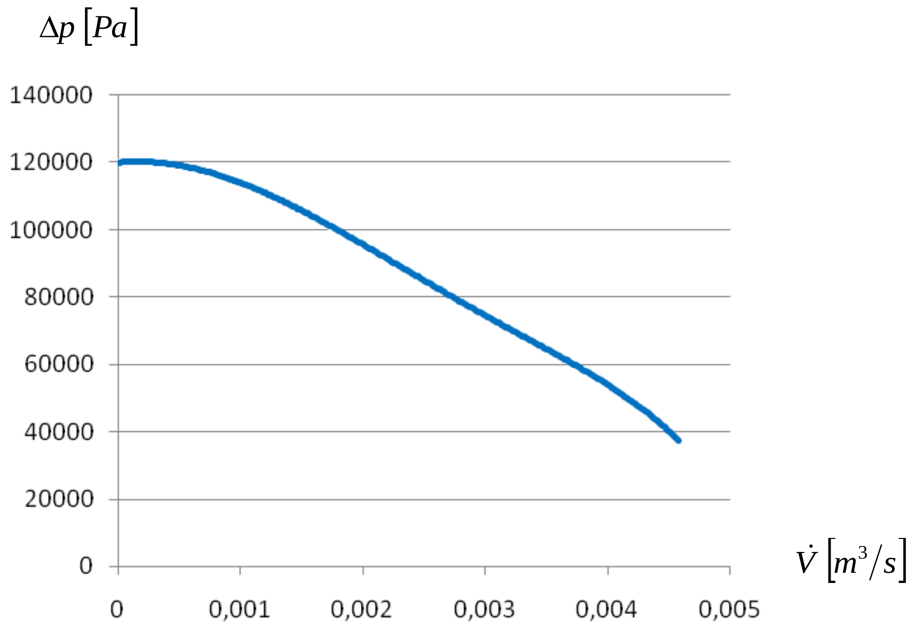

Figure 6.16 The pump curve of a Grundfos Magna pump.

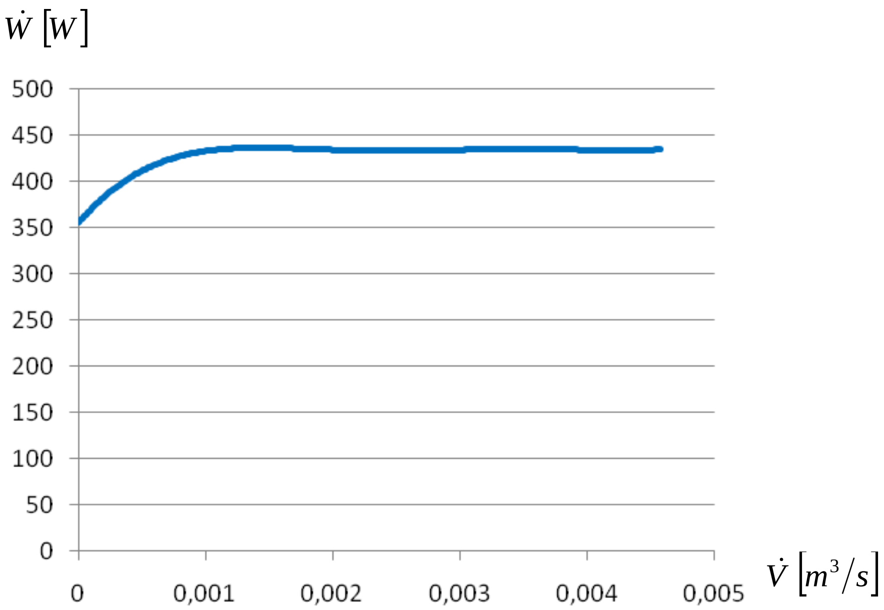

Figure 6.17 The power curve of a Grundfos Magna pump.

The Grundfos Magna in question is driven by an electrical motor whose rotational speed can be adjusted manually, and figure 6.16 and 6.17 shows the resulting behavior when the rotational speed is at its maximum. This was how the constant rotational speed pump was modeled, i.e. at all times follow these two curves. For the controlled pumps, on the other hand, the polynomials of these curves were manipulated so that solutions could be found within an area limited by the maximum curves, the positive parts of the y- and x-axis and the maximum allowed flow of the pump. This approach was considered as necessary since the actual relation between a certain absolute value of the rotational speed and the corresponding pressure rise and volume flowrate, could not be determined, as discussed in part 5.3. This manipulation was done by introducing a scaling factor (denoted as *n*) to the polynomials of the pump and power curves, as seen in equation 6.16 and 6.17, respectively. This factor was thought to represent the change of the rotational speed and naturally, it was chosen as the variable whose time derivative was controlled by the regulator. As long as $0 \le n \le 1$ in combination with the usual restrictions discussed in part 6.3, the obtained values of the pressure rise and the power were located within the valid working area of the pump. Initially for all pumps, the volume flow was set to about zero and the pressure rise at its maximum. This corresponded to an operational point located almost furthest to the left of the maximum curves. Consequentially, the rotational speed was initialized at its maximum value (i.e. *n* equals to one). Throughout the remaining parts of the simulations it was observed that the steady state values of *n* were between 0.9 and 0.3, i.e. 10 and 70 % lower than the maximum.

$$\Delta p = funct\left(\dot{V}\right)\cdot n \qquad [Pa] \qquad \text{(eq. 6.16)}$$

$$\dot{W} = funct\left(\dot{V}\right)\cdot n \qquad [W] \qquad \text{(eq. 6.17)}$$

This approach assumes that a certain decrease in the rotational speed resulted in the same rate of decrease in both the pressure rise and the power consumption. Furthermore was the volume flow rate at the same time kept at a constant level, or the other way around. The main advantage of this approach was that the relation between the pressure rise, the volume flowrate and the power consumption was kept. This resulted in that the power consumption could be described as a function of the pressure rise and the volume flowrate, which can be seen by combining the two equations above. This relation was also maintained for the uncontrolled pump, since it followed the maximum curves statically.

As mentioned in part 4.3.2.3, the energy efficiency of the pump is dependent on the corresponding flowrate. By using equation 6.18 (formerly presented as 4.4), in combination with the pump and power curves, it was shown that such relation already was included in the characteristics. This relation, derived from the maximum curves, is shown in figure 6.18 where the energy efficiency is presented as a function of the volume flow rate. Since the real energy efficiency was included in the characteristics, and since the relation between the curves was maintained, the real energy efficiency was accounted in each operational point of the pump. This resulted in that the calculated power consumption of the pump actually was the necessary drive power consumed by the pump in order to supply the required useful work.

$$\dot{W} = \frac{\Delta p\cdot\dot{V}}{\eta} \qquad [W] \qquad \text{(eq. 6.18)}$$

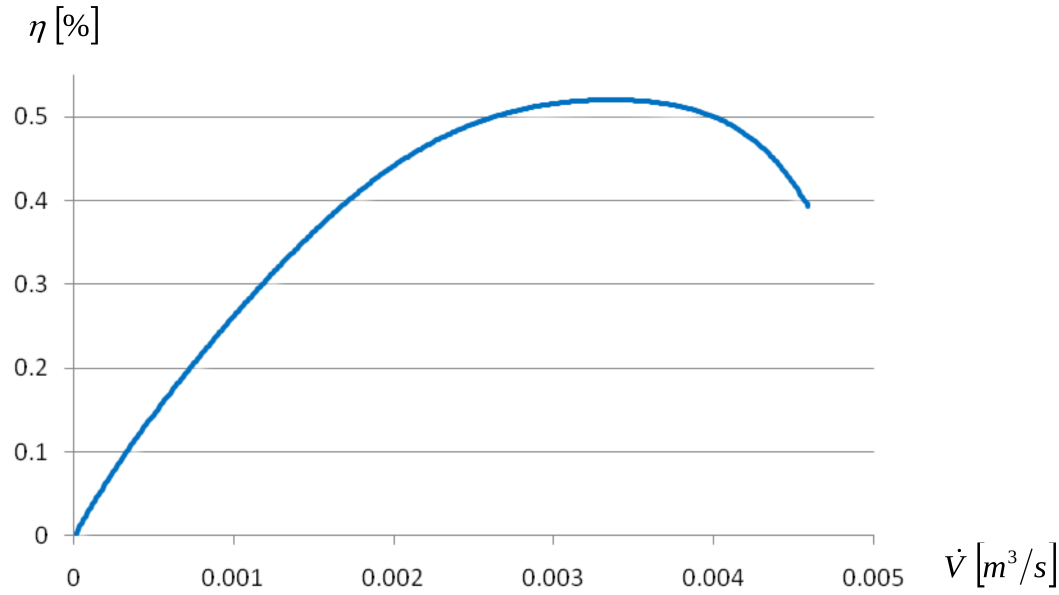

Figure 6.18 The energy efficiency as a function of the corresponding volume flow rate for maximum rotational speed of the pump.

The regulator included in both of the controlled pumps was of a PI type. This choice was motivated by that this type of regulator is the type most frequently occurring in industry[16]. As mentioned in section 5.2.4, one of the controlled pumps should maintain a certain pressure drop over the last thermal unit, and the other one a certain pressure rise over itself. The magnitudes of the corresponding set values were chosen arbitrary, and the used values were 10 kPa and 160 kPa for the pressure drop and pressure rise, respectively. Both of the control-loops were built just like in the case of the ABC; pressure sensors measured the differential pressures over the controlled objects and the regulators adjusted the control variables in order to reach the set values. The settings of the regulators, such as static gain and time constant of the integrator part, were unknown and their values were instead tuned to result in behaviors which were estimated as realistic.

The dynamic behaviors of the controlled pumps were described by equation 6.19, where *J* is the moments of inertia, *T* is the torque and *u* and *y* are the output and input signal, respectively. This equation introduced a time-delay between the controller and the pump and was thought to represent the inertia of the pump.

$$u = (J) \cdot \frac{dy}{d\tau} + T \qquad \text{(eq. 6.19)}$$

$$T = (J_p + J_L) \cdot \frac{d\omega}{d\tau} + T_f + T_L \qquad [Nm] \qquad \text{(eq. 6.20)}$$

Equation 6.19 is based on equation 6.20, which, according to Alciatore[1], describes the actual dynamic behavior of a pump driven by an electrical motor. If comparing these two equations, it can be seen that the output signal from the modeled regulator should be interpreted as the torque of the motor, and the input signal to the modeled pump as the rotational speed. This seemed reasonable since if a motor was included in the models, it would have been placed between the regulator and the pump, and hence the rotational speed would have been its output. Just like in the case of equation 6.7, which was presented in the same context regarding the ABC, equation 6.19 describes a system with dampened resonance. In the case of equation 6.19, the counter forces were considered as due to the friction in the pump, the counter torque caused by the load and the inertia of the pump and load. Of these three, the first two were assumed to be independent of rotational speed, while the third one was assumed to be dependent. The reason why equation 6.19 was simplified compared to 6.20, was that none of the constants were known. Instead, were the number of constant reduced as much as possible, and the values of the remaining were tuned to correspond to a behavior of the pump which was considered as reasonable[9].

In figure 6.19, the resulting behavior of a controlled pump is shown, where the pressure rise is plotted against time. Most interesting are the oscillations which occurred at the system initiation and during the disturbances. These are namely the resulting behavior described by equation 6.19, and it can be seen that the time it takes for the pump to find a steady-state after a disturbance is about 5 seconds.

The reason why the three oscillations in figure 6.19 have different amplitudes is unclear. When it comes to the two disturbances (introduced at simulated time 10 and 20 seconds), the reason might be that one of them occurred close to the pump and the other one far away from the pump (see section 5.2.6). Hence, they might have affected the pump differently. The reason why the initiation of the system affected the pump more than the disturbances might be an effect caused by the high initial pressure level of the pump. Hence, this high pressure level caused high activity of the regulator since the corresponding control error was large.

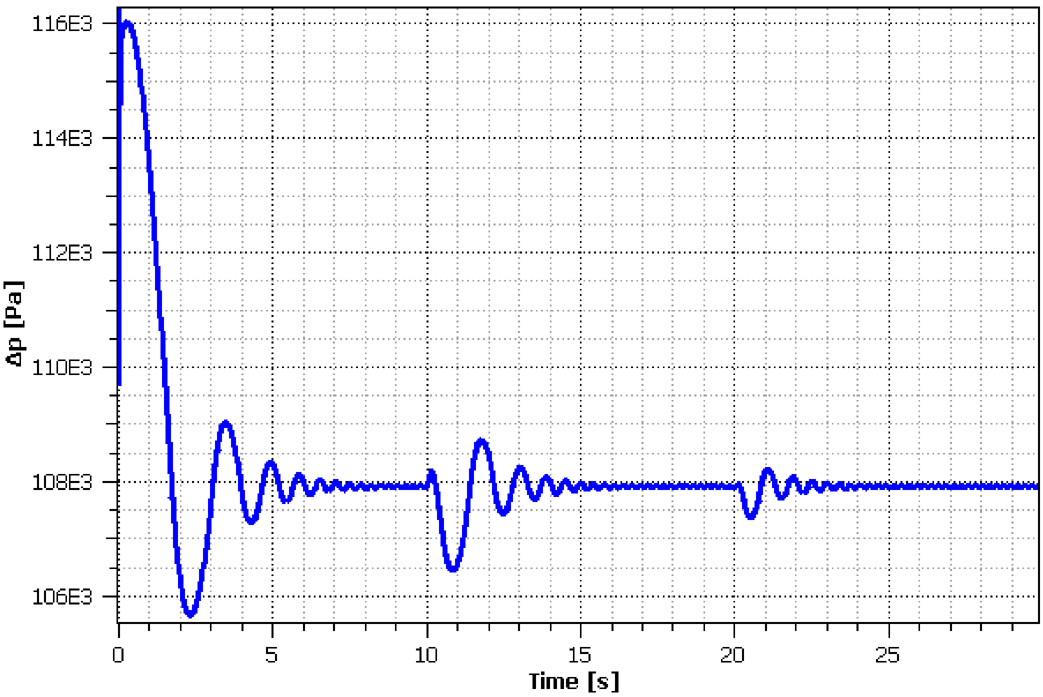

Figure 6.19 Modeled behavior of a controlled pump.

7 Results

In this part the results from the two studies are presented and the corresponding conclusions are presented in part 8. It could not be pointed out enough that the numerical results from the simulations performed during the reference study were not verified and should for that reason not be considered as reliable. Furthermore, it should be remembered that all of the pumps were based on the one located in the test rig used during the verification study. This means that the sizes of the pumps were not adjusted to the actual systems, which in turn means that the resulting power consumptions could be much higher than if the pumps instead were optimized to the systems. However, the purpose of the reference study was not to produce numerical values which could be considered as reliable. Instead was it performed to observe the effect of the graduate introduction of valves, which in turn could be used to draw conclusions regarding the abilities of MathModelica.

7.1 Results from the reference study

The result from the reference study is presented as plots acquired when the scenarios presented in part 5.2.1 were simulated. In each graph, the corresponding solutions to some variable related to the centralized pump are plotted for three different scenarios. These three scenarios are the ones which were placed in the same column of table 5.1. Hence, each graph shows how the operation of a specific type of pump was affected when valves were graduate introduced in the distribution system. Generally, the continous lines represent the scenarios when neither balancing nor ABC valves were included. All of those scenarios were characterized by an overflow in the system, i.e. a flow level which resulted in flows which were larger than the design flows. The dashed lines represent the scenarios in which balancing valves were introduced, and their openings corresponded to a flow level which was characterized as the design flow. And finally, the dotted lines represent the scenarios when both balancing and ABC valves were included, which in turn corresponded to a part load flow case. Hence, all scenarios included radiator valves. Furthermore were also disturbances introduced via the radiator valves during all of the simulations which were included in the reference study. These were consistently introduced at time-steps 10 and 20 seconds and consisted of that one of the radiator valves almost closed and then opened again (see section 5.2.6).

In figure 7.1 the resulting drive power consumptions are shown for the three scenarios when the centralized pump was of a constant rotational speed type. As can be seen, the unsteady regions of the two upper curves are primarily represented by the introduction of the disturbances via the radiator valves. But for the lower curve, including the ABC valves, yet another unsteady region is visual at the initiation of the system. This region is represented by that the ABC valves continuously are adjusting their openings to achieve the desired pressure drop. Hence, the steady-state of the system is reached when the openings of the ABC valves corresponded to the set value of the controller.

As can be seen, the drive power consumption became stepwise lower when the different valves were introduced. Another interesting observation is that the amplitude of the oscillations caused by the disturbances tends to be smaller as new valves are introduced.

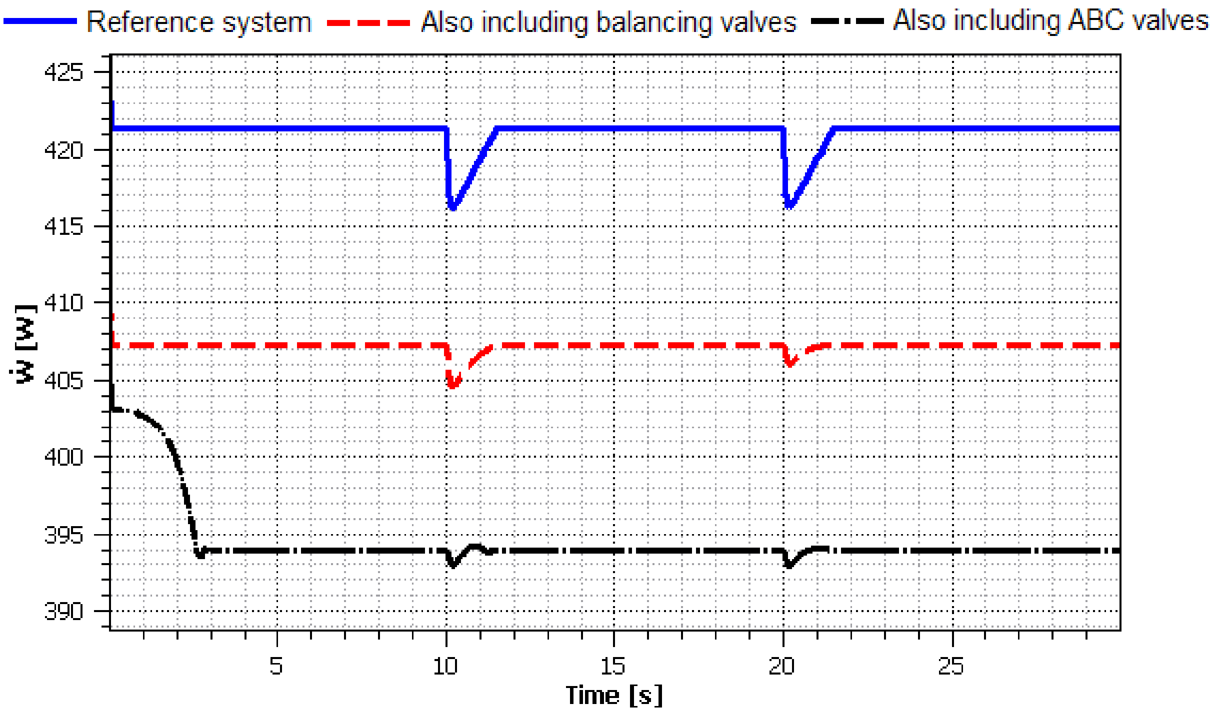

Figure 7.1 The effect of the graduate introduction of valves on the drive power consumption of the constant rotational speed pump.

In figure 7.2 and 7.3 are the resulting volume flowrate and pressure rise of the constant speed pump when simulated in the three different distributional systems shown. What can be seen is that the STAD and ABC valves throttle the flow at the same time as they introduce a higher pressure drop in the systems. Hence, higher pressure rises of the pumps was required to compensate for these losses.

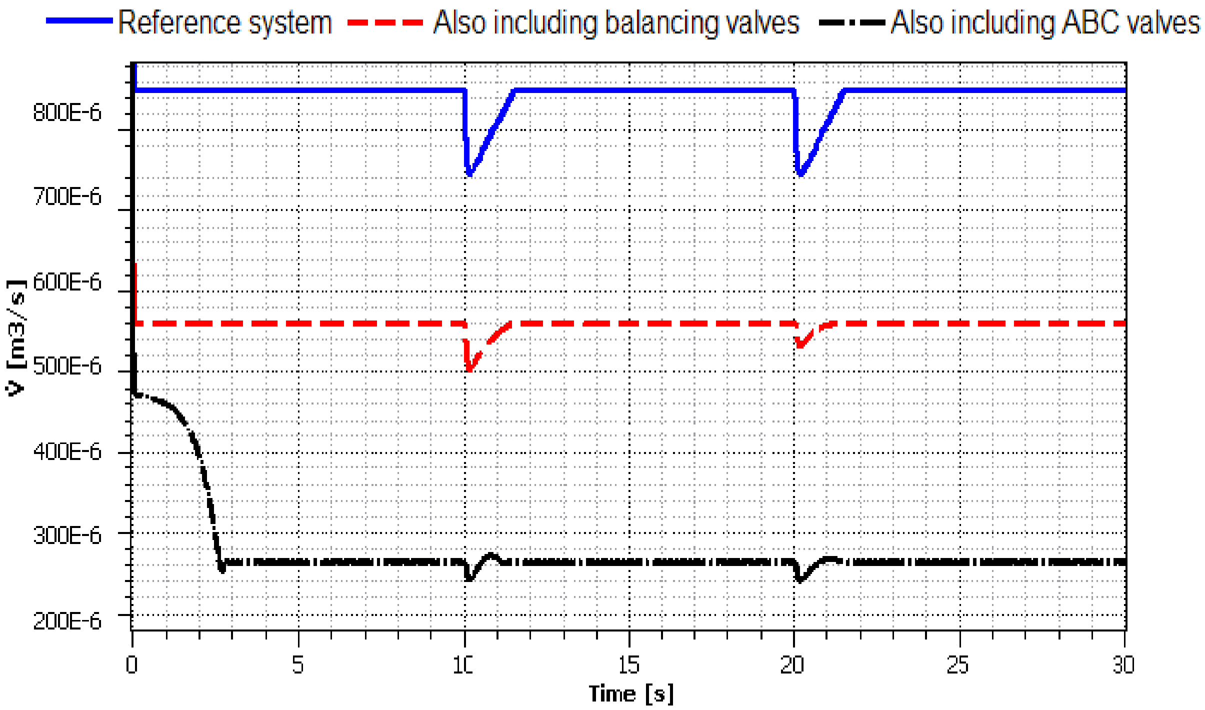

Figure 7.2 The effect of the graduate introduction of valves on the volume flow of the constant rotational speed pump.

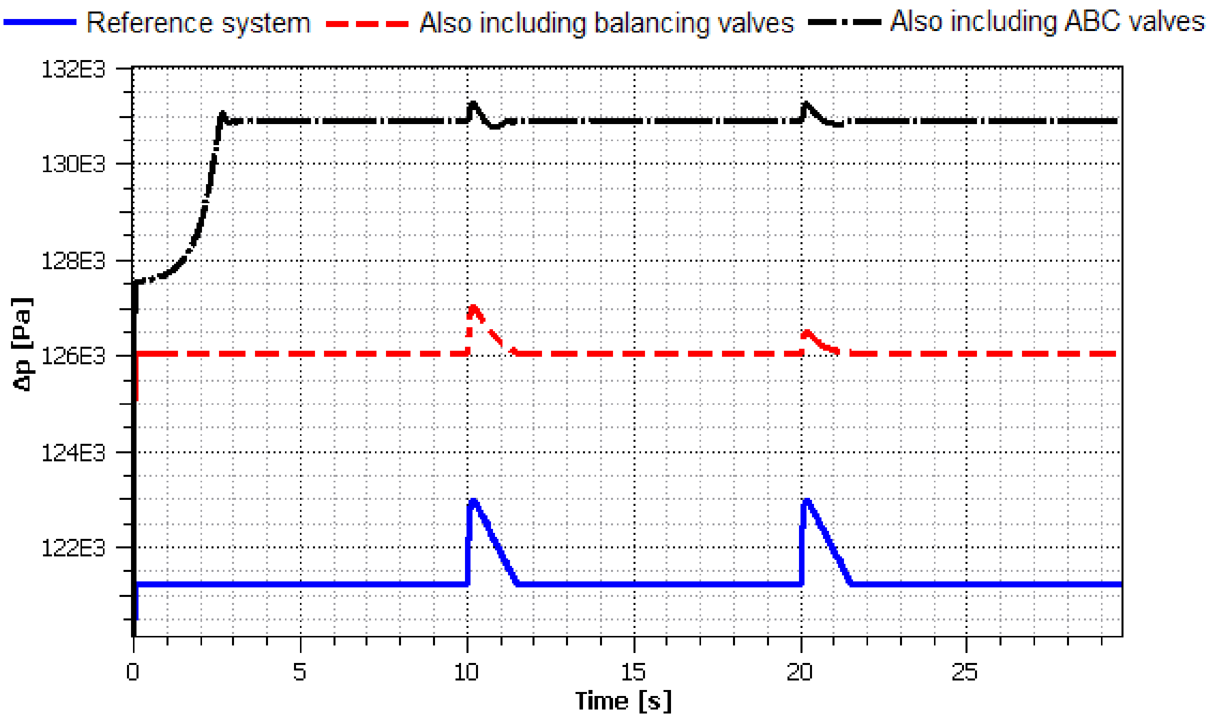

Figure 7.3 The effect of the graduate introduction of valves on the pressure rise of the constant rotational speed pump.

In the same way, the results from the simulations when the pumps instead controlled the pressure rise over itself are shown in figure 7.4, 7.5 and 7.6. The overall result, compared to the scenarios when the previously discussed pump was included, is about the same. However, one large difference is that the pressure rise of the pump in this case was the controlled variable. Hence, the steady-state values became the same in all three scenarios as seen in figure 7.6. Otherwise became the required drive power lower when valves were introduced which was due to the reduced total volume flow of the pump. Both of these effects can be seen in figure 7.4 and 7.5 respectively.

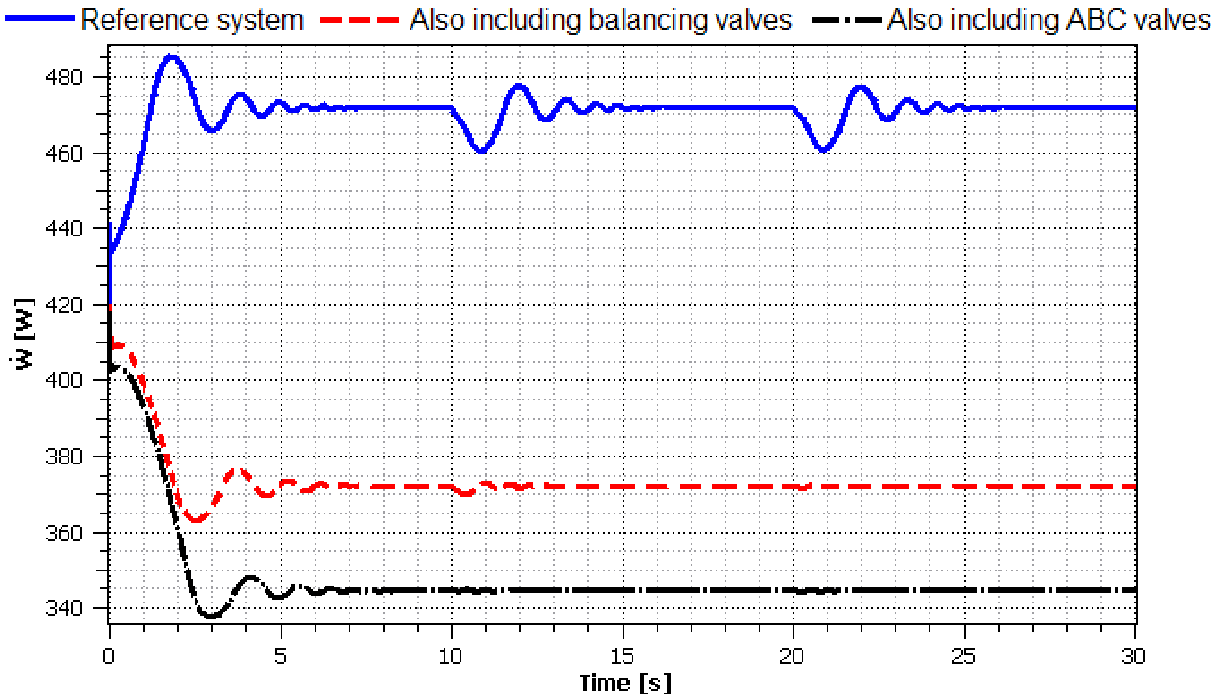

Figure 7.4 The effect of the graduate introduction of valves on the drive power consumption of the pump which controls the pressure rise over itself.

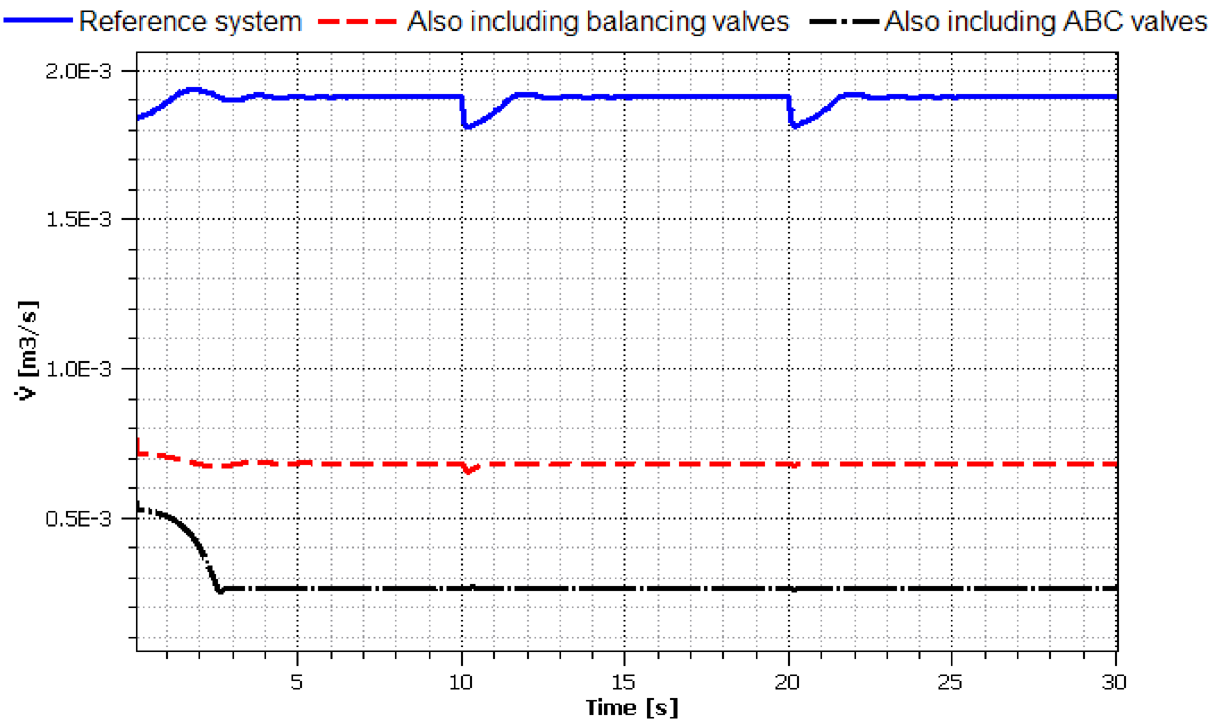

Figure 7.5 The effect of the graduate introduction of valves on the volume flow of the pump which controls the pressure rise over itself.

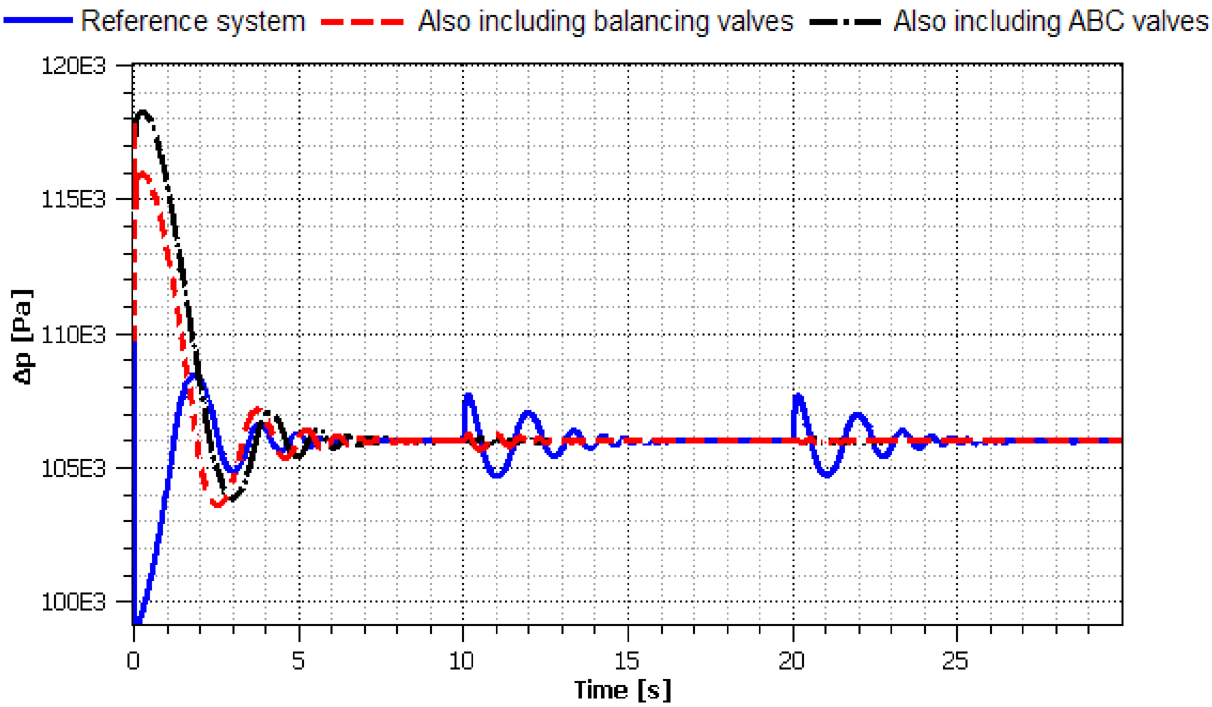

Figure 7.6 The effect of the graduate introduction of valves on the pressure rise of the pump which controls the pressure rise over itself.

In the final figures presented below, the result from the simulations of the three distribution systems including the pump which was controlling the pressure drop over the thermal located most far away from the pump is presented. Since this pressure drop was the controlled variable, it should reach a certain stead-state which should be the same in all three scenarios. This also implies that the stead-state flowrate through the last thermal unit should be the same in all three cases, which is shown in figure 7.7. However, the flowrates only have to be the same in this particular circuit which the last thermal unit belongs to. Hence, the total flow rate should also in this case be affected when valves were introduced. This effect is shown in figure 7.9 and the corresponding pressure rises in figure 7.10. Overall, it was identified that both the volume flowrate and the pressure drop were decreased when valves were introduced. When it came to the drive power of the pump, the result also in this case shows that it was decreased when valves were introduced, even if the difference between the part load and design flow case was not as distinct as when other types of pumps were used.

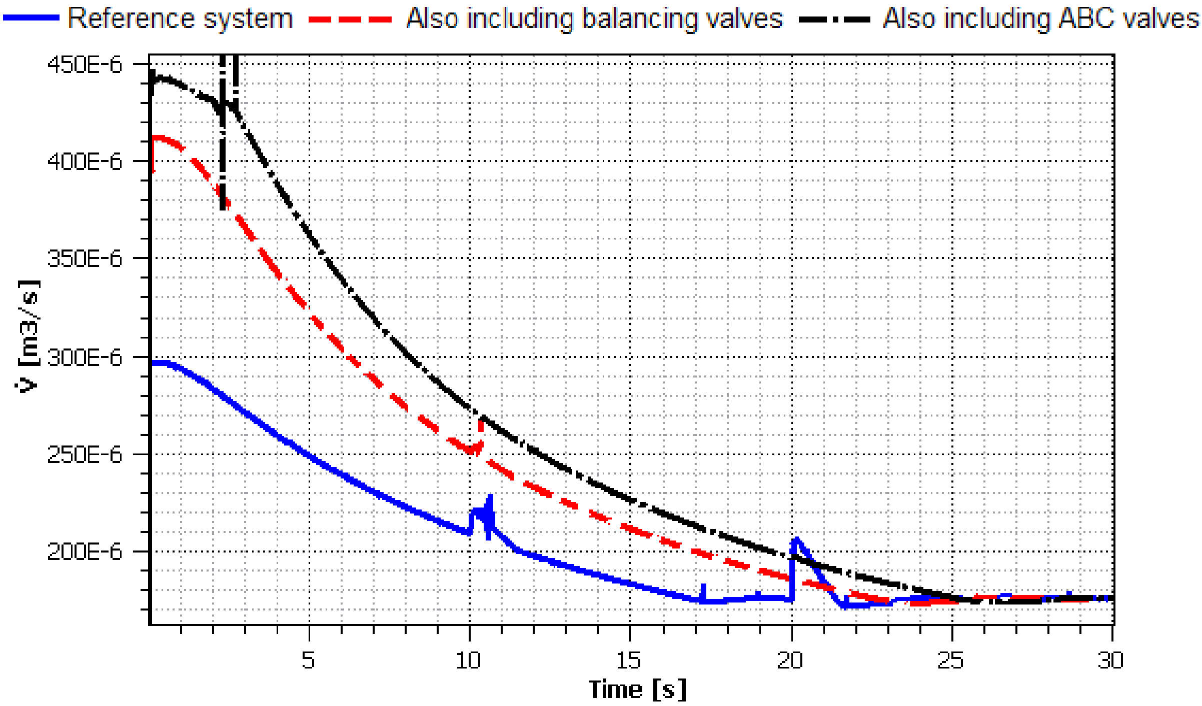

Figure 7.7 The controlled volume flowrate through the thermal unit located most far away from the pump.

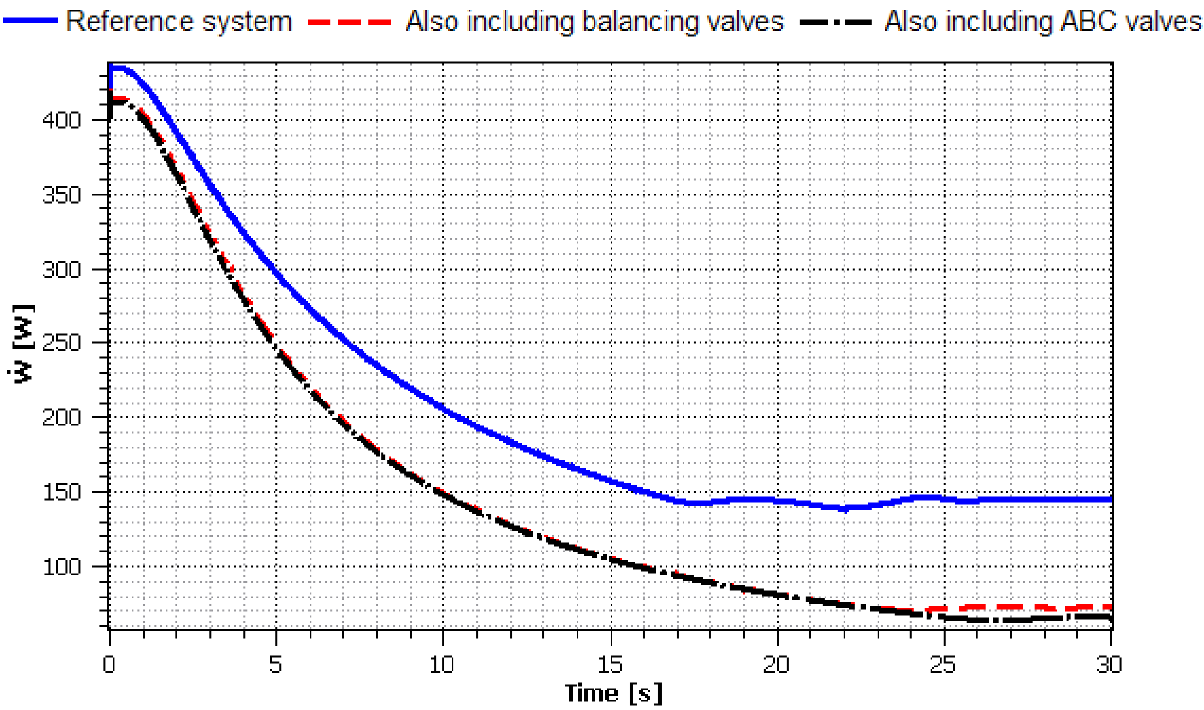

Figure 7.8 The effect of the graduate introduction of valves on the drive power consumption of the pump which controls the pressure drop over the last thermal unit.

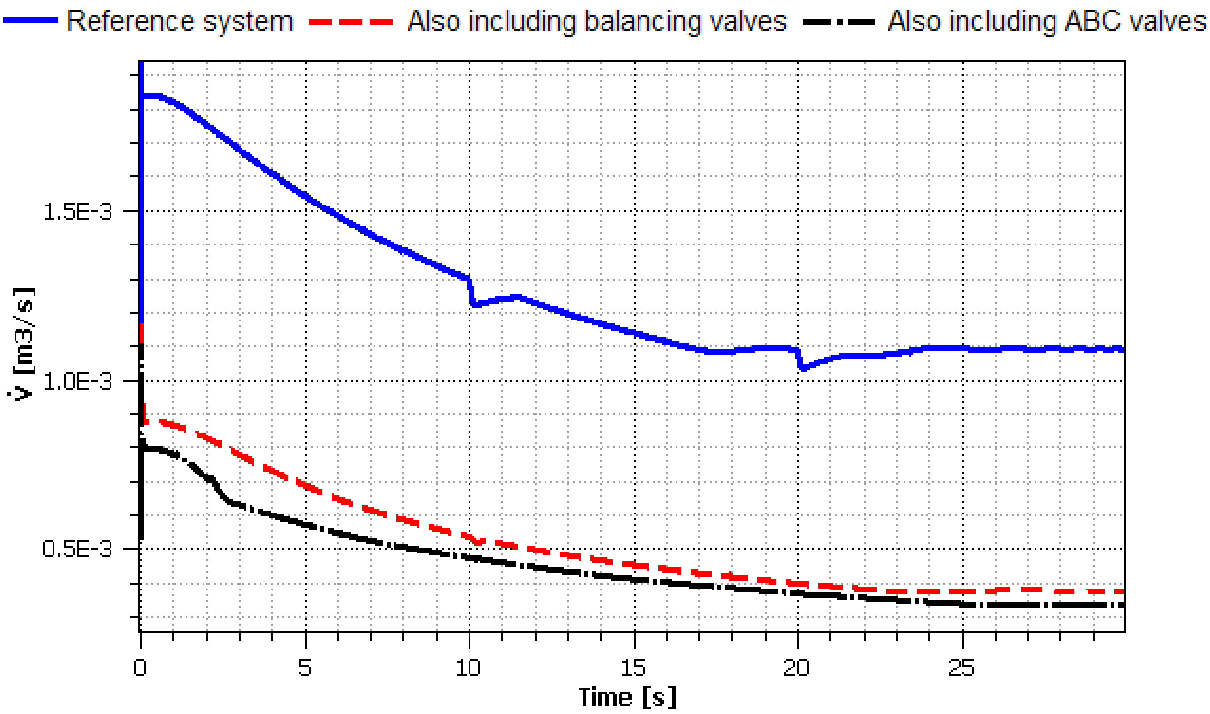

Figure 7.9 The effect of the graduate introduction of valves on the volume flowrate of the pump which controls the pressure drop over the last thermal unit.

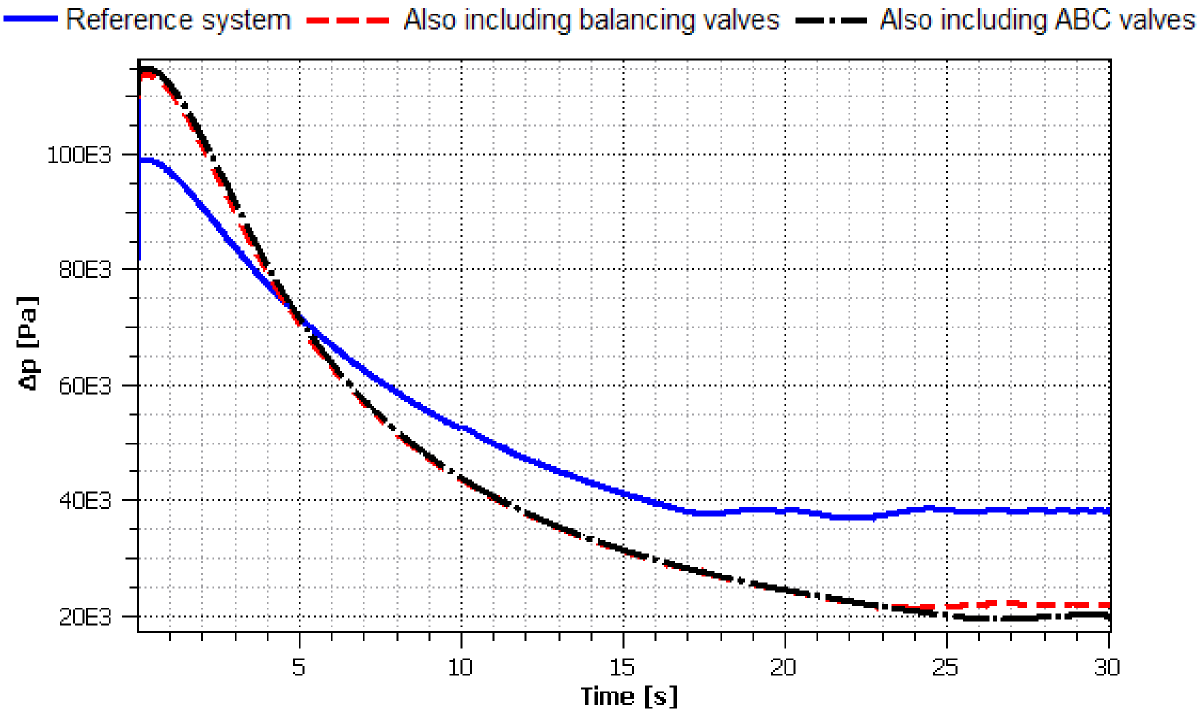

Figure 7.10 The effect of the graduate introduction of valves on the pressure rise of the pump which controls the pressure drop over the last thermal unit.

7.2 Results from the verification study

The measuring of the test rig was performed at three points, denoted as A, B and C in figure 5.5. Throughout the verification study, the measured volume flowrate at these points and the corresponding simulated values were used to validate the simulated data. At each point, ten different measurements were done and the study included four different operational scenarios. The resulting values are all presented in appendix A.

The measured mean values, along with the corresponding standard deviation and the simulated values are shown in table 7.1, for the setup where all the valves in the test rig were fully opened. The corresponding results are shown in table 7.2, where the total expected measuring error is presented as the sum of the normalized random and the systematic errors (see section 5.3). Furthermore is also the difference between the simulated and mean measured data presented in the same table. As discussed in part 5.3, the systematic errors were both due to the errors in the computer models and the manufacturing tolerance of the valves. However, when it comes to the systematic errors presented in table 7.2, only the manufacturing tolerance was taken into account. Instead was the error due to the computer models kept in mind and used when the results were discussed (section 8.2). That's why the systematic errors are equal at all three measuring points: all three valves which were representing these points had namely the same setting.

The comparisons between the measured and simulated data are also presented in table 7.2. The result of the comparisons was calculated as the difference between E and U_{tot}, which corresponded to the simulated values breaching of the region in which measured values were expected to be found. That is, the scatter of the measured values. If the simulated values were located within this region, the breach was set to zero. The explanation to why the breach in some cases was larger than zero can only be found in the accuracy of the models or of the solver, since the variability of the measurements already were taken into account. If the simulated results are to be considered as accurate enough this breach cannot be too large.

In conclusion for this operational case, it can be seen that the simulated data at all three points breached the region in which measured values were expected to be found. However, the errors are not very large and can definitely be explained by the errors of the component models.

Table 7.1 Statistics from setup for which all valves are fully opened (4 mm).

Valve	**A**	**B**	**C**
Measured mean value, $\bar{x}$, [m^3/s] $\times 10^{-6}$	99.5	165	815
Standard deviation, s_x, [m^3/s] $\times 10^{-6}$	0.85	5.35	1.81
Simulated volume flowrate, $\dot{V}$, [m^3/s] $\times 10^{-6}$	110	189	753

Table 7.2 Expected measuring errors and results, all valves fully opened (4mm).

Valve	A	B	C
Random errors, U_m, [%]	±0. 9	±3.2	±0.2
Systematic errors (not incl. modeling errors), U_v, [%]	±4.5	±4.5	±4.5
Total expected measuring error, U_{tot}, [%]	±5.4	±7.7	±4.7
Difference between measurement and simulation, E, [%]	13.6	12.7	-8.2
Simulated values breaching of the real scatter region [%]	8.3	5.0	3.5

The corresponding data for the operational case when all valve openings were set to 1 mm are shown in table 7.3 and 7.4. The region in which measured values were expected to be found, i.e. the scatter of the measured values, was for this operational case larger compared to the previous discussed operational case. This was both due to the increased uncertainty of the valves, which followed by the decreased openings, and due to the larger expected measuring error, which probably was due to an increased fluctuation of the flow. Partly because this region was larger, the simulated value at two points (A and B) were located inside of it. However, the simulated value of the third point (C) breaches this region with about 12 %.

Table 7.3 Statistics from setup for which all valve openings are set to 1 mm.

Valve	A	B	C
Measured mean value, $\bar{x}$, [m^3/s] $\times 10^{-6}$	307	499	2340
Standard deviation, s_x, [m^3/s] $\times 10^{-6}$	30.5	15.3	76.8
Simulated volume flowrate, $\dot{V}$, [m^3/s] $\times 10^{-6}$	295	460	1840

Table 7.4 Expected measuring errors and results, all openings set to 1 mm.

Valve	A	B	C
Random errors, U_m, [%]	±9.92	±3.06	±3.27
Systematic errors (not incl. modeling errors), U_v, [%]	±12	±12	±12
Total expected measuring error, U_{tot}, [%]	±21.9	±15.06	±15.27
Difference between measurement and simulation, E, [%]	-4.07	-8.48	-27.17
Simulated values breaching of the real scatter region [%]	0	0	11.90

In the next case, the openings of the valve-pairs were set differently, and the valves denoted as D and E in figure 5.5 were set to 4 mm. As can be seen in table 7.5 and 7.6, this operational case was not much different from the previous, but the breaching occurred instead at point A.

Table 7.5 Statistics from setup for which some openings are set to 1 and some to 4 mm, openings of valves D and E is set to 4 mm.

Valve	**A**	**B**	**C**
Measured mean value, $\bar{x}$, [m^3/s] $\times 10^{-6}$	568	892	2020
Standard deviation, s_x, [m^3/s] $\times 10^{-6}$	28.4	19.7	54.3
Simulated volume flowrate, $\dot{V}$, [m^3/s] $\times 10^{-6}$	674	834	1890

Table 7.6 Expected measuring errors and results, some openings are set to 1 and some to 4 mm, openings of valves D and E is set to 4 mm

Valve	**A**	**B**	**C**
Random errors, U_m, [%]	±5.0	±2.2	±2.7
Systematic errors (not incl. modeling errors), U_v, [%]	±4.5	±4.5	±4.5
Total expected measuring error, U_{tot}, [%]	±9.5	±6.7	±7.2
Difference between measurement and simulation, E, [%]	15.7	-7.0	-6.9
Simulated values breaching of the real scatter region [%]	6.2	0.2	0

The valve settings which constituted the final operational case were the same as for the previous one, with the exception that the openings of valves denoted as D and E instead were set to 1 mm. This scenario was somewhat different from the other ones since the resulting flowrate at point A was measured to be zero. Furthermore, the difference between the simulated and measured values was very large, up to about 1000 %, which of course is totally unacceptable.

Table 7.7 Statistics from setup for which some openings are set to 1 and some to 4 mm, openings of valves D and E is set to 1 mm.

Valve	**A**	**B**	**C**
Measured mean value, $\bar{x}$, [m^3/s] $\times 10^{-6}$	0	533	12400
Standard deviation, s_x, [m^3/s] $\times 10^{-6}$	0	31.2	36.3
Simulated volume flowrate, $\dot{V}$, [m^3/s] $\times 10^{-6}$	118	526	1140

Table 7.8 Expected measuring errors and results, some openings are set to 1 and some to 4 mm, openings of valves D and E is set to 1 mm.

Valve	**A**	**B**	**C**
Random errors, U_m, [%]	±0	±0. 585	±0. 292
Systematic errors (not incl. modeling errors), U_v, [%]	±4.5	±4.5	±4.5
Total expected measuring error, U_{tot}, [%]	±4.5	±5.06	±4.79
Difference between measurement and simulation, E, [%]	100.00	-1.33	-987.72
Simulated values breaching of the real scatter region [%]	95.5	0	-982.43

8 Conclusion

In this part, the results presented in chapter 7 are analyzed. The purpose was to use these results in order to determine whether or not the requirements which were the basis for the evaluation of MathModelica were fulfilled or not.

8.1 Reference study

The goal of the reference study was to show if:

1. MathModelica can find solutions to hydronic systems of relevant sizes
2. MathModelica can be used to model both balancing and control valves as well as other common hydronic components
3. MathModelica produces results which agrees to reality regarding systems behavior

8.1.1 Requirement one

During this study, the first requirement was considered as fulfilled in most cases, but not in all. The reason was that convergence problems were encountered for the system in which the flow in some part approached zero, as well as for systems in which the T-junctions based on the Idelchik models were included. However, the latter is considered as a modeling problem and is instead discussed under requirement number two.

So, convergence problems were primarily encountered for systems in which the flow in some part approached zero. And this problem was primarily tangible when the disturbances were modeled which was discussed in part 5.2.6. It should be pointed out that there is a possibility that this problem derived from the model equations and could perhaps in that case be avoided by adjusting them. However, at this point, it seems like MathModelica generally has problems to describe zero flows. This is backed up by the fact that the zero flow in the last operational case in the verification study was failed to be described. This operational case resulted in large errors, while the errors in the other cases were relatively small. The possibility that both the modeling problem of the disturbances and the large errors in the verification study were related was considered. Hence, if this is the case, it might be concluded that MathModelica fails to describe zero flows, and if forced; a solution can not be found.

8.1.2 Requirement two

Also the second requirement was in most cases considered as fulfilled, but not entirely. The reason was that MathModelica was limiting the types of equations that could be used in the models in two different perspectives: both what types of equations that could be defined and what types that could be solved. Generally, problems were encountered when polynomials of higher order then two was used (discussed in part 6.2) and differential equations could only be ordinary and only include time derivatives. So, the reason why requirement two was considered as fulfilled in most cases were that the including components could be described by these relatively simple types of equations. But in some cases, the accuracy of the models had to be set aside in order to make this possible.

However, as mentioned above, the T-junctions could not be simulated at all when the Idelchik models were used. The most probable explanation to this problem was that MathModelica could not solve the polynomials which were derived from the Idelchik models. These polynomials included the ratio between two different volume flowrates which resulted in that three variables had to be iterated from each equation. Furthermore, each T-junction consists of two polynomials which, on top of it all, were nested, since one of the variables was included in both. The resulting equation system might become relatively complicated and that is thought to be the reason why solutions could not be found.

8.1.3 Requirement three

Requirement three was considered as fulfilled and the evaluation was performed by analyzing the simulations presented in part 7.1. The purpose of the analysis was to determine whether or not the resulting system behaviors could be regarded as realistic, and thereby if requirement three was fulfilled or not. The basis for the analysis was equations 4.4 which were used to reflect over the change in drive power consumption when valves were introduced. Furthermore were also the graphs of the corresponding pressure rises and volume flow rates evaluated to see if they corresponded to the observed change of drive power consumptions.

Generally for all distribution system, regardless of what pump that was included, it was shown that both the volume flow rate and the pressure rise of the pump were affected when valves were introduced. According to equation 4.4, a positive effect on the drive power consumption occurred when the flowrate and/or the pressure rise was decreased and a negative effect occurred when one of them were increased. Besides of those, also the energy efficiency had an effect on the power consumption according to equation 4.4. However, whether or not this effect was positive or negative was solely dependent on the corresponding flowrates of the new and old operational points (see figure 4.4). For that reason the volume flowrate and the energy efficiency were regarded as nested and it was primarily their combined effect that was of interest.

But to begin with, it was shown in the graphs presented in part 7.1 that regardless of what pump which was included in the systems; the effect of the introduced disturbance tended to become smaller as more valves were introduced. This is considered as consistent with the models since when new components were introduced, they might add further damping to the system. This means that the systems became more and more elastic and hence, disturbances had less influence.

8.1.3.1 Conclusions related to the systems including the pump with constant speed

The results of the simulations related to the pump with constant speed are shown in figure 7.1, 7.2 and 7.3. Since this pump statically followed the characteristic curves related to the maximum rotational speed, the magnitude of the drive power consumption corresponded to the operational point in which the maximum pump curve and system curve were intersecting. In this case was the flowrate decreased and the pressure rise was increased when valves were introduced. In conclusion; since the drive power consumption became lower when valves were introduced, the results imply a system, in which the ratio of the flowrate and energy efficiency was decreased more than the corresponding increase of pressure drop when valves were introduced. Hence, the volume flow was the variable which in this case had the largest influence on the drive power consumption. Such system was considered as theoretically possible and for or that reason, this result was considered as reasonable.

8.1.3.2 Conclusions related to the systems including the pump with controlled pressure rise over itself

The corresponding results of the systems including the pump which controls the pressure drop over itself are shown in figure 7.4, 7.5 and 7.6. In this case, the set value of the pressure rise was maintained, which means that the power consumption only was dependent on the ratio between the flowrate and the energy efficiency. Since the drive power consumption also in this case became lower when valves were introduced, the results imply a system in which the decrease of the volume flowrate was larger than the corresponding decrease of energy efficiency when valves were introduced. This is also considered as a system which is theoretically possible and for that reason were these results also considered as reasonable.

One interesting observation which can be done in figure 7.4 is the relatively high drive power which was required in the system represented by the continous line. This drive power was both substantially higher than for the other two scenarios in the same graph, as well as for the corresponding system when the pump with constant speed was included (shown in figure 7.1). However, it should be reminded that the pressure rise of the pump was fixed to a level which apparently was much higher than the corresponding pressure drop in the system. This means that both the pressure rise and the volume flow rate became very large in this scenario. This can also be seen by looking at the pressure rises for corresponding scenarios presented in figure 7.1 and 7.7.

8.1.3.3 Conclusions related to the systems including the pump which controlled the pressure drop over the last thermal unit

The results from the final case, which was characterized by the pump which controlled the pressure drop over the last thermal unit, are presented in figure 7.7, 7.8, 7.9 and 7.10.

The specific feature of this pump was the resulting flow was determined by the value which was needed in order to meet the desired pressure drop at the last unit. Overall, also this case showed that the drive power consumption became lower when valves were introduced, mostly since the flowrate was decreased. For that reason was also these results considered as theoretically possible. However, this case resulted in some behaviors which had not been observed before. These are presented and analyzed in the following text.

As can be seen in figure 7.8, the resulting drive power consumption of this type of pump was consistently lower compared to the corresponding scenarios including a different type of pump. The reason was the relatively low required pressure rise of this pump which can be seen in figure 7.10. This might be a result of that the required volume flowrate could be met by any pressure rise within the valid region by adjusting the rotational speed, according to equation 6.16. This means that the degree of freedom of the pressure rise was large, and hence it was not forced to any specific value. The hypothesis based on this reasoning, was that the distribution system was in this case determining the pressure rise of the pump. This was motivated by the fact that the inlet/outlets, modeled in MathModelica, states that the pressure and flowrate was equal in all ports which were directly connected to each other. Hence, since the pump was not determining the pressure rise, the pressure rise was instead determined by the distribution system. This would result in that the pump would supply the absolute minimum pressure needed in the system, i.e. the pressure drop of the distribution system. This might on the other hand not be true for the other two pumps, especially not the one with controlled pressure rise over itself.

In figure 7.8, 7.9 and 7.10 it can be seen that the valves had smaller influence on both the volume flowrate and the pressure drop compared to the corresponding scenarios when the two other pumps were included. This might partly depend on that the pressure levels consistently were lower in the cases when this pump was used, as discussed above. However, it might also depend on that also the flow levels in most cases were different compared to the cases when other types of pumps were used. This is especially visible if comparing the curves in figure 7.9 with the corresponding ones in figure 7.2 and 7.5. It can then be seen that most of the flow levels in this case were lower, and that the part load and design flow cases were more or less equal. This difference in the flow cases lies in that the flowrate through the last circuit in this case was determined by the set value of the pump-controller, and not by the balancing valve like in the other cases. This could result in different total volume flowrates of the pump in all three distribution system, which in turn further would affect the pressure level.

Another interesting observation can be done if the pressure rises in figure 7.10 are further studies and compared to the other corresponding pressure rises of the pump with constant speed (figure 7.3). It can then be seen that the pressure rise in this case actually became lower when valves were introduced. This effect can most probably be explained by the relatively low flow levels in the systems; the flow was throttled in a way that the effect of the added pressure drops from valves were overcome by the reduced pressure drop of the already existing components when the flowrate through them were decreased.

8.2 Verification study

The final requirement, stated in the introduction and in part 5, was tested in the verification study. It was phrased that the simulated results should agree to reality regarding numerical accuracy. The results which were the foundation of the conclusions below are presented in part 7.2.

In most cases, it was shown that the simulated values on average deviated from the region in which measured values were expected to be found by about ±7 %. This means that about ±7 % of the total error can be derived from the models or from the accuracy of the solver. However, this error is about of the same magnitude as was embedded in the characteristics used in the components models. The errors of the characteristics were unavoidable and are due to simplifications, generalizations etc. made by the authors. For that reason, the ±7 % error was considered as acceptable. Furthermore did also the approximations of these characteristics introduce an uncertainty of about ±5 %.

However, in one of the operational cases the deviation between the measured and simulated values was much larger, about 100-1000 %. In this case, the measurements indicated that the flowrate in the upper part of the test rig was zero. And the large deviation is for now on explained by that MathModelica has problem to describe zero flows. Hence, requirement four was not considered as fulfilled.

9 Discussion

This main part of this chapter consists of the sources or error which might have influenced the verification study. Furthermore are some of the experiences of MathModelica which were gained during this project reviewed. The purpose to include them was to support future users of MathModelica in the area of hydronic system.

9.1 Sources of error

Besides the errors embedded in the characteristics, the simplification of the T-junctions into a straight pipe and a pipe-bend is an error which could be found in the component models. This simplification might partly explain the commonly occurring deviation between the simulated and measured values of about ±7 % in the verification study. However, it should be remembered that this simplification had to be done since MathModelica could not solve the more relevant equations. This means that MathModelica had to be considered as the reason for the part of the deviation encountered in the verification study which followed by the T-junction simplifications. Furthermore, the straight pipes were neglected in the verification study, which, on the other hand, is thought to have a smaller effect.

Another modeling error was the relatively inaccurate approximation of the power curve which was presented in part 6.8. However, this has no affect on the verification study since the volume flowrate is the measured variable. But, it has, on the other hand, influenced the reference study regarding the numerical values. This also has to be considered as due to MathModelica itself since polynomials of higher order would have been more accurate but could not be solved.

9.2 Experience of MathModelica

This project has provided knowledge regarding how a model preferably should be described in order to perform well during a simulation in MathModelica. First, the importance of initial values; if they are unrealistic the solution process will be very time consuming and the result might be something other than desired. For example; it is very easy to end up with backflow in the components if the initial values not prescribe the desired direction. Furthermore, negative pressures and other unphysical tendencies are not that uncommon if the initial values do not avoid this. Second, the system initiation has to be included in the simulations. It is very hard to initially describe as system that is in operation. This means that all components have to include time dependent equations, which are able to describe the development from an unsteady to a steady solution. Third, if a component is described by, let's say, more than four equations, it is favorable to divide it into subcomponents containing not more than four equations each. This is worth mentioning, since the order in which the equations should be solved can not be set, and hence, the structure is very hard to grasp if many equations are combined. Finally, the structure of the equations are of large importance; avoid as far as possible both division and square roots since this can result in unsolvable equation systems, also avoid expressions which allows some other solution then desired; for example unreal or negative numbers.

Finally, it has to be mentioned, that even if some issues have been uncovered, there are a lot of positive aspects of MathModelica. Especially the language, which is based on equation, allows the user to be creative in a possible familiar language. This means that the mathematical skill of a new MathModelica user is of more importance than the ability to learn a new computer language. Further-more, the component-based structure opens up for the possibility to continuously expand the models and systems by taking-off from the previous ones. This means that step-wise improvements can be made without too much effort.

References

1. Alciatore D. G., Histand M. B., 2007. Introduction to mechatronics and measuring systems. (McGraw-Hill, ed.3)

2. Alvarez, H., 1990. Energiteknik del 1 (Energy technology part 1). (Studentlitteratur). Lund

3. Andersson Chan A., Hedström A., 2004. VVS 2000: tabeller och diagram. Vatten- och avloppsteknik (VVS 2000: tables and diagrams. Water and wastewater technology). (Förlags AB VVS). Stockholm

4. Carriere M., Schoenau G. J., Besant R. W., 1998. Investigation of some large building energy conservation using DOE-2 model. (Pergamon, Energy conversion & management 40, 1999, page 861-872)

5. The Commtech group, 2003. Achieving the desired indoor climate. (Studentlitteratur, chapter 9)

6. Djuric N., Novakovic V., Holst J., Mitrovic Z., 2006. Optimization of energy consumption in buildings with hydronic heating systems considering thermal comfort by use of computer-based tools. (ELSEVIER, energy and buildings 39, 2007, page 471-477)

7. Fahlén P., 2009. Analysis of Measurement – Concepts, Procedures, Uncertainties. (CTH, Building Services Engineering). Göteborg.

8. Fox R. W., McDonald A. T., Pritchard P. J., 2004. Introduction to fluid mechanics. (Wiley, ed. 6)

9. Fredriksson J., 2009. Electro mechanical actuators (Lecture held at CTH 2009-02-26 in the course Mekatronik (Mechatronics))

10. Fritzson P., 2004. Principles of object-oriented modeling and simulation with MathModelica 2.1. (Whiley) Linköping

11. Gruber M., 2008. Implementation of the modeling and simulation tool MathModelica into the predevelopment procedures of Tour & Andersson AB. (Company report owned by Tour & Andersson AB)

12. Idelchik I. E., 1986. Handbook of hydraulic resistance. (Springer-Verlag, ed.2). Berlin, Germany

13. Johnsson F., 2008. The global and European energy system. (Lecture held at CTH 2008-01-21 in the course Heat and power systems engineering)

14. Karlsson, Holmberg., 2004. The societal metabolism. (CTH)

15. Khartchenko N. V., 1998. Advanced energy systems. (Taylor & Francis)

16. Lennartson B., 2002. Reglerteknikens grunder (The foundations of automatic control engineering). (Studentlitteratur, ed. 4:6)

17. Miller D. S., 1990. Internal flow Systems - Design and performance prediction. (Unwin Brothers Ltd. Ed 2)

18. Montgomery D. C., Runger G. C., 2006. Applied statistics and probability for engineers. (Wiley, ed.4)

19. Nordling C., Österman J., 2006. Physics handbook for science and engineering. (Studentlitteratur, ed. 8)

20. Palmertz H., 1993. Valve handbook – theory and practice. (Tour & Andersson AB)

21. Persson P., pre-development engineer at TA in Ljung. Oral reference 2009-03-04

22. Petitjean R., 1995. Total balancing. (Tour & Andersson Hydronics AB)

23. Purmo, 2007. Technical brochure, Ventil Compact 200 mm

24. Rydén B., 1999. Energy models for the technical energy system – role, construction and use. (CTH, department of Energy Systems Technology)

25. Ryu S-R., Rhee K-N., Yeo M-S., Kim K-W., 2008. Strategies for flowrate balancing in radiant floor heating systems. (Taylor and Francis, Building research & information 2008, 36(6), page 625-637)

26. Råde L., Westergren B., 2004. Mathematics handbook – for science and engineering. (Studentlitteratur, ed.5)

27. Thrüschel A., 2002. Hydronic heating systems – The effect of design on system sensibility. (CTH, Building Services Engineering). Göteborg.

28. Tour & Andersson AB, 2005. Product catalogue 2005/2006'

29. Versteeg H. K., Malalasekera W., 2007. An introduction to computational fluid dynamics. The finite element method. (Pearson Prentice Hall, ed.2). Loughborough, USA

30. Walski P. E., Chase D V, Savic D A, 2001. Water distribution modeling. (Haestad Methods Inc.)

31. White F. M., 2008. Fluid mechanics. (McGraw Hill, ed.6). Rhode Island, USA

32. http://www1.eere.energy.gov/buildings/about.html (Cited 2009-02-03)

Appendix A

Table A.1 Measured flowrate when all valves are fully opened.

Valve	A	B	C
Measured volume flow 1 [m^3/s]	0.0000994	0.000150	0.000813
Measured volume flow 2 [m^3/s]	0.0000986	0.000166	0.000811
Measured volume flow 3 [m^3/s]	0.000100	0.000166	0.000815
Measured volume flow 4 [m^3/s]	0.000100	0.000165	0.000814
Measured volume flow 5 [m^3/s]	0.0000988	0.000168	0.000815
Measured volume flow 6 [m^3/s]	0.000100	0.000167	0.000816
Measured volume flow 7 [m^3/s]	0.000100	0.000166	0.000816
Measured volume flow 8 [m^3/s]	0.0000991	0.000168	0.000818
Measured volume flow 9 [m^3/s]	0.0000980	0.000166	0.000813
Measured volume flow 10 [m^3/s]	0.0001	0.000167	0.000816

Table A.2 Measured flowrate when all valves are fully closed.

Valve	A	B	C
Measured volume flow 1 [m^3/s]	0.000262	0.000497	0.00225
Measured volume flow 2 [m^3/s]	0.000326	0.000479	0.00232
Measured volume flow 3 [m^3/s]	0.000315	0.000504	0.00241
Measured volume flow 4 [m^3/s]	0.000312	0.000523	0.00233
Measured volume flow 5 [m^3/s]	0.000306	0.000491	0.00231
Measured volume flow 6 [m^3/s]	0.000299	0.000474	0.00237
Measured volume flow 7 [m^3/s]	0.000306	0.000504	0.00240
Measured volume flow 8 [m^3/s]	0.000255	0.000516	0.00246
Measured volume flow 9 [m^3/s]	0.000354	0.000504	0.00233
Measured volume flow 10 [m^3/s]	0.000338	0.000491	0.00221

Table A.3 Measured flowrate when some valve openings are set to 1 mm and other are fully opened, D and E are fully opened.

Valve	A	B	C
Measured volume flow 1 [m^3/s]	0.000576	0.000893	0.00209
Measured volume flow 2 [m^3/s]	0.000542	0.000867	0.00192
Measured volume flow 3 [m^3/s]	0.000571	0.000914	0.00205
Measured volume flow 4 [m^3/s]	0.000578	0.000901	0.00198
Measured volume flow 5 [m^3/s]	0.000623	0.000857	0.00204
Measured volume flow 6 [m^3/s]	0.000531	0.000878	0.00208
Measured volume flow 7 [m^3/s]	0.000576	0.000890	0.00203
Measured volume flow 8 [m^3/s]	0.000595	0.000899	0.00195
Measured volume flow 9 [m^3/s]	0.000556	0.000899	0.00204
Measured volume flow 10 [m^3/s]	0.000535	0.000920	0.00202

Table A.4 Measured flowrate when some valve openings are set to 1 mm and other are fully opened, the openings of D and E are set to 1 mm.

Valve	A	B	C
Measured volume flow 1 [m^3/s]	0	0.000468	0.00128
Measured volume flow 2 [m^3/s]	0	0.000508	0.00123
Measured volume flow 3 [m^3/s]	0	0.000537	0.00121
Measured volume flow 4 [m^3/s]	0	0.000549	0.00123
Measured volume flow 5 [m^3/s]	0	0.000561	0.00118
Measured volume flow 6 [m^3/s]	0	0.000501	0.00125
Measured volume flow 7 [m^3/s]	0	0.000555	0.00128
Measured volume flow 8 [m^3/s]	0	0.000555	0.00119
Measured volume flow 9 [m^3/s]	0	0.000543	0.00125
Measured volume flow 10 [m^3/s]	0	0.000560	0.00126