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International Energy Agency
Programme of Research, Development and Demonstration on
District Heating and Cooling

ANNEX IV

ADVANCED ENERGY TRANSMISSION FLUIDS

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Introduction

The application of drag reducing additives in district heating networks is a promising technology to improve the competition conditions of those systems. The pressure loss and therefore, the pumping costs of existing networks can be reduced or their capacity can be increased. The pipes and fittings of new planned networks can be designed in smaller diameters. Other possibilities to use the drag reducing effect are the decrease of the supply temperature due to an increasing mass flow while keeping the capacity constant or the integration of further (far away) heat sources which becomes only economic due to decreasing pumping costs. Resources can be saved and the pollution of the environment can be reduced.

During the last 13 years, lots of investigations concerning the application of cationic surfactants as drag reducing additives have been carried out in Canada, Denmark, Finland, Germany, Korea, the Netherlands, Sweden and the United States. Experiments with different kinds of cationic surfactants such as Ethoquate, Habon(-G), Obon(-G), Dobon(-G), C_{16}TASal etc. have been carried out. The effects on pressure drop behaviour in straight pipes, helical tubes as well as the heat transfer behaviour in those geometries have been investigated. The heat transfer of shell and tube, helical tube and plate heat exchangers has been examined. Furthermore, heat meters, fittings, pumps, the corrosion behaviour, environmental aspects and water hammering have been investigated when using drag reducing surfactants.

In laboratory tests and full scale investigations in Denmark, Germany and the Netherlands, the general suitability of cationic surfactants - especially of Habon-G and Dobon-G in combination with Sodiumsalicylate - as drag reducing substances for district heating systems was proven. Based on those results, strategies to apply surfactants in real district heating systems have been developed.

The International Energy Agency (IEA) also undertook research, development and testing of advanced energy transmission fluids for district heating and cooling. An Experts Group made up of member countries representatives was formed 1987 to guide and to support research in this field. Nations participating in this cooperative research programme include Canada, Denmark, Finland, Germany, Korea, the Netherlands, Norway, Sweden, United Kingdom and the United States.

In addition to IEA-sponsored activities a number of government agencies are sponsoring independent research and testing efforts on advanced energy transmission fluids for district heating and cooling. The interest in this R & D-work is apparent from this that in many cases chemical manufacturers, engineering firms and DHC-systems operators participate to share one's experiences.

In the meantime - starting 1983 - four working programmes (called Annex I, II, III and IV) were established, which essentially are attend with the development of additives capable of reducing friction losses and increasing the capacity of district heating and cooling distribution systems.

The first three Annex-programmes with important subjects, such as environmental aspects, operational aspects, corrosion behaviour, simulation etc., have been finished (Annex I (1986), Annex II (1989), Annex III (1992). The results of the projects-activities have been published and disseminated in the participating countries.
In 1993 a further working-programme, so called Annex IV, with a new three-year period was established. Under Annex IV, four projects concerning the field of research „Advanced Transmission Fluids for District Heating and Cooling“, have been supported for which several institutes and companies of different countries have been carried out alone and in cooperation with various partners theoretical and experimental work:

- Project A: „Modelling of the Location and Requirements for Heat Exchangers in District Heating Networks Using Friction Reduction Additives“,
- Project B: „Experiments on the Effects of Friction Reduction Additives on Substations“,
- Project C: „Survey of Environmental Restrictions to the Use of Additives in District Heating and Cooling Systems“ and
- Project D: „Improving of the Heat Transmission Properties of Tube Bundle Heat Exchangers by Installing Obstacles inside the Pipes“.

The project A is dealing with the simulation of the behaviour of comprehensive transport networks with special consideration of heat exchangers which separate the transport system from the distribution network. With this a network simulation program for transport systems with a graphical user interface and a CAD-like function for network design was developed. With the simulation tool the modifications of a district heating network, which are necessary when applying drag reducing additives can be worked out. These are mainly the heat exchangers for the hydraulic separation in a transportation-system containing surfactant solution and several distribution-systems operating with district heating water, the pressure maintenance and the treatment and supply of water for the distribution system. Simulation results for the application of surfactants in a real system are given. Within the context of project A, economic calculations have been carried out. These calculations consider general models and give an overview of the savings in costs which can be expected under certain conditions.

In project B the influence of drag reducing additives in small domestic heat exchangers and on flow meters which are installed in small consumer stations has been determined. Four different kinds of heat exchangers (single wall plate HE, helix shaped multiple double HE, spiral double pipe HE and double wall plate HE) and four different flow meters (magnetic-inductive, mechanical-inductive and mechanical meters) have been investigated under technical conditions (primary and secondary circuit temperatures for summer and winter).

This investigations show the following results:

- No significant changes concerning the pressure drop of the investigated heat exchangers.
- Minor decreases in heat transmission capacity of the investigated heat exchangers when using Habon-G (by that the influence for tertiary heat exchanger is smaller than for secondary),
- The magnetic-inductive and one mechanical heat meter are low influenced by the use of Habon-G. The accuracy of the two other mechanic heat meters decreases strongly with increasing surfactant concentration.
Aim of project C was the collecting of data and information about commercially available drag reducing surfactants and of regulations of different countries concerning the approval of drag reducing additives in district heating systems. Therefore, a questionnaire has been developed, which was handed to all members of the Experts Group „Advanced Transmission Fluids for District Heating and Cooling“ to register the state of conditions of the different IEA member countries.

The questionnaire was answered from nearly all member countries (Canada, Denmark, Finland, Germany, Korea, the Netherlands, Sweden and the United States). The analysis results that an unambiguous conclusion covering the situation in all countries cannot be drawn. In most countries there are no concrete rules related to this new technology. It seems to be clear that a certain reluctance towards the introduction of new additives in general is a common attitude. The technology has not been declined in any of the countries.

When using drag reducing additives inside pipes, the heat transfer from the fluid on the inner pipe area is reduced significantly. Therefore, the last project D has been carried out, in which the improvement of the heat transmission conditions in tube bundle heat exchangers by installing turbulence increasing obstacles (spiral springs) inside the pipes was investigated.

These investigations in laboratory scale (University of Dortmund) have been carried out with water and surfactant solutions at different flow velocities, concentrations and temperatures. The results with pure water agree quite well with literature values. With obstacles a strong improvement of the inner heat transfer coefficient is reached. When using obstacles (prestressed helical springs of stainless steel wire with defined pitches) the heat transfer with surfactant solutions is - under certain conditions - even better than in case of using water without obstacles. On the other hand, measurements of the pressure drop inside the tubes show a significant increase. Compared to water without obstacles an increase of 200 to 800 % can occur, depending on the pitch of the spring.

In full scale tests in district heating exchangers (CHP plant in Heming) pipes containing spiral springs (both conventional and stainless steel) were in operation for one year. The spirals caused no difficulties during operation, light corrosion in the conventional steel and no sign of corrosion in the stainless material.

The conclusion show, that to the use of obstacles, the total heat transfer is improved, that the steam in the CHP-plant can be expanded to a lower level. This implies a higher electricity generations with which the increased pumping demand through the heat exchanger can be compensated. Herewith the installing of optimized obstacles inside the pipes of condensers represents an experimentally supported measures with which the use of suitable drag reducing additives in district heating systems will be economically possible.

The reports and the results of the several projects are presented as independent parts of the total report. They have an independent table of contents and therefore, an independent numbering of pages.
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However the following reports are the result of the knowledge, expertise and work of the particular principal authors and project leaders.

December 1996

Artur Steiff
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Modelling of the Location and Requirements for Heat Exchangers in District Heating Networks Using Friction Reduction Additives

Principal Investigator: Prof. P.-M. Weinspach Thermische Verfahrenstechnik GmbH, Germany

B  Project B:

Experiments on the Effects of Friction Reduction Additives on Substations

Principal Investigator: Delft University of Technology, Faculty of Mechanical Engineering, Laboratory of Thermal Power Engineering, The Netherlands

C  Project C:

Survey of Environmental Restrictions to the Use of Additives in District Heating and Cooling Systems

Principal Investigator: Brunn & Sørensen AS, Denmark

D  Project D:

Improving of the Heat Transmission Properties of Tube Bundle Heat Exchangers by Installing Obstacles inside the Pipes

D1  Investigations of Heat Transfer and Pressure Drop

D2  Testing of Obstacles in an Operating Heat Exchanger and Evaluation of the Overall Effect

Principal Investigator: Brunn & Sørensen AS, Denmark,
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Modelling of the Location and Requirements for Heat Exchangers in District Heating Networks using Friction Reduction Additives

Professor Weinspach

Thermische Verfahrenstechnik GmbH

Dortmund, August 1995
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1. Introduction

The application of drag reducing surfactants in district heating systems is of great importance because the economic viability of those systems can be increased by using these additives. With small amounts of cationic surfactants in district heating water the pressure drop in pipelines can be reduced significantly. Due to this effect the pump energy of existing district heating systems can be decreased or the transport capacity can be raised. In addition, the supply temperature can be decreased significantly when keeping the heat output constant. New district heating networks can be designed with smaller pipe diameters or the maximum economic transmission length can be increased by applying drag reducing additives.

Due to environmental, safety and economic aspects, the use of drag reducing additives is planned for transport systems only. Therefore, indirectly connected consumer systems are necessary. In Germany, for instance, 50% of the district heating networks are indirectly connected. In some cases it will be necessary to separate the consumer stations hydraulically by installing heat transmission stations when planning the application of friction reducing surfactants.

In addition to the positive effect of drag reduction some negative effects occur, due to the change in flow conditions. These are influences on heat exchangers, heat meters, pumps, filters, etc. The most important problem is a significant heat transfer reduction in heat exchangers. In shell and tube heat exchangers with straight long pipes, a heat transfer reduction of about 95% and therefore, a significant reduction of the heat output is possible.

Concerning the supply guarantee, a reduction of heat output can not be tolerated in most cases. Therefore, a lot of investigations have been carried out in laboratory and full scale tests with several kinds of heat exchangers which are typical for district heating systems. Up to now, correlations for most of the typical apparatus have been developed with which it is possible to calculate the conditions when using drag reducing additives.

Aim of this project is the development of a simulation tool to evaluate the effects caused by drag reducing additives in district heating transport systems with special consideration of heat exchangers. In the context of this study, such a calculation tool has been developed.

With this simulation tool, the influence of drag reducing additives on single heat exchangers as well as on complete heat exchanger networks such as district heating transport systems can be calculated for water and drag reducing surfactant solutions as heat carrier. Furthermore, the conditions when changing the operation mode from direct to indirect (hydraulic separation of network parts by installing additional heat exchanger stations) can be calculated.

The simulation program can be used to modify existing heat exchangers or to design additional necessary devices e.g. when installing new heat exchangers for hydraulic separations. The modification of single system elements can be tested by evaluating the complete modified DH-system. By simulating several cases of modified systems and comparing the results of the simulations, an optimum technical solution can be achieved.

Premise of the application of drag reducing additives is the economic viability. Comparing the modified system working with drag reducing additives (that means the optimum technical solution which has been found with the simulation tool) and the original system, it is possible to estimate the savings in cost due to the application of surfactants in district heating networks. Therefore, it is necessary to evaluate cost functions to be able to calculate the proportions and
conditions that are caused by the additives. Cost functions as well as models to estimate the savings have been developed in further projects [23][25]. A short summary about the results of these studies is given at the end of this report (see Appendix - B).
2. Using Friction reducing Additives in DH-Systems

In the following chapters possibilities of using drag reducing additives in district heating systems are presented. Two fundamental possibilities have to be considered:

- The application in existing networks and
- The application in systems that still have to be built.

The application in existing networks can lead to problems because the change of an existing configuration can result in difficulties. Unfavourable conditions can lead to complex modifications before the application of surfactants can be realised.

On the other hand, when building new systems, the application of surfactants can be taken into consideration during the planning stage.

The presented possibilities are borderline cases. The combination of different applications is conceivable as well.

2.1 Application in existing systems

2.1.1 Reducing the pumping costs

In figure 2.1 the application of drag reducing additives for saving pumping costs (reducing the pressure drop) is shown. Under the condition of a constant flow rate, the pressure drop and therefore the pumping costs can be reduced compared to water.

![Fig. 2.1: Reducing pumping costs (pressure drop) [23]](image-url)
With the definition of the drag reduction DR:

$$DR = \frac{\Delta p_w - \Delta p_s}{\Delta p_w},$$

eq. (2.1)

the equation for the pumping capacity $P$:

$$P = \eta_{tot} \cdot \dot{V} \cdot \Delta p,$$

eq. (2.2)

and the assumption that the total efficiency of the pump $\eta_{tot}$ is constant, compared to water the reduction in pumping capacity $PCR$ is:

$$PCR = P_w \cdot DR.$$  

eq. (2.3)

Under favourable conditions a drag reduction and therefore a reduction of pumping energy of about 80 \% is possible.

### 2.1.2 Increasing the capacity

Analog to figure 2.1 in figure 2.2 the increase of the capacity is shown. Prerequisite for this kind of application is that heat producer and consumers have enough capacity.

On the other hand, bottlenecks in systems that work on capacity limits can be eliminated. Investment for the extension of the system can be postponed or even avoided.

![Diagram](image-url)  

**Fig. 2.2:** Increasing the capacity [25]
The characteristic curve for water can be described as follows [6]:

\[ \Delta p_w = K_w \cdot v_w^2. \]  

eq. (2.4)

With the assumption that the drag coefficient is no function of the Reynolds number, the pressure drop of drag reducing surfactant solutions can be calculated as follows:

\[ \Delta p_s = (1 - DR) K_w \cdot v_s^2. \]  

eq. (2.5)

For an increase of heat capacity, the maximum flow rate that can be achieved when applying drag reducing additives is of great importance. With the assumption of a constant pumping capacity and identical efficiencies for water and surfactant solution, the maximum flow rate with surfactant solutions can be calculated with equation 2.6:

\[ v_{s, \text{max}} = v_{w, \text{max}} \cdot (1 - DR)^{-1/3}. \]  

eq. (2.6)

Table 2.1 shows the achievable flow rate as function of drag reduction DR.

<table>
<thead>
<tr>
<th>DR in [%]</th>
<th>factor of flow rate increase</th>
</tr>
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<tbody>
<tr>
<td>50</td>
<td>1.26</td>
</tr>
<tr>
<td>55</td>
<td>1.31</td>
</tr>
<tr>
<td>65</td>
<td>1.42</td>
</tr>
<tr>
<td>75</td>
<td>1.59</td>
</tr>
<tr>
<td>80</td>
<td>1.71</td>
</tr>
</tbody>
</table>

Tab. 2.1: Maximum achievable flow rate

### 2.1.3 Shifting heat generation

Apart from the above described direct possibilities to apply surfactants (chapter 2.1.1 and 2.1.2) an indirect use of the drag reducing effect can be taken into account. The temporal shifting of heat generation for district heating is an indirect application.

Above all in large transport systems, it can be advantageous to use the increase of capacity for shifting the heat capacity to cheaper heat sources (even in greater distances). An example would be the increase of the heat quantity produced in e.g. waste incineration plants (waste heat) and to decrease the heat production in CHP-stations.

On the other hand, cheap heat generation plants in greater distances can be integrated into heat distribution systems (increasing the maximum economic transmission length).

Figure 2.3 qualitatively shows the shifting of heat capacity.
Using Friction reducing Additives in DH-Systems

Fig. 2.3: Shifting of heat capacity/integration of production plants in greater distances

The integration of additional heat production plants or waste heat can lead to a reduction in costs as well as to a relief of environment.

2.1.4 Reducing the supply temperature

Increasing the flow rate by keeping the heat capacity constant makes it possible to reduce the supply temperature and therefore, to reduce heat generation costs in CHP plants. Due to the following equation:

$$\dot{Q} = M \cdot c_p \cdot (T_{\text{sup}} - T_{\text{ret}}),$$

the supply temperature can be reduced when keeping $\dot{Q}, c_p$ und $T_{\text{ret}}$ constant and increasing the mass flow rate $M$.

Table 2.2 gives an example for the possibility to reduce the supply temperature. Assumption for this calculation is an increase of the mass flow rate of 40 % (this is a DR of 63 %). The heat load and return temperature are constant.

<table>
<thead>
<tr>
<th></th>
<th>$\dot{Q}$ [MW]</th>
<th>$M$ [kg/s]</th>
<th>$T_{\text{sup}}$ [°C]</th>
<th>$T_{\text{ret}}$ [°C]</th>
<th>$c_p$ [kJ/(kgK)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>water</td>
<td>100</td>
<td>398,7</td>
<td>130</td>
<td>70</td>
<td>4,18</td>
</tr>
<tr>
<td>surfactant</td>
<td>558,2</td>
<td></td>
<td>113</td>
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Tab. 2.2: Example for reducing the supply temperature [25]

The example shows that a significant reduction of the supply temperature is possible and therefore, the heat generation costs can be reduced, too.
In addition the heat loss will be decreased due to the reduced temperature between the supply and the ambient temperature. Furthermore, the static pressure can be reduced.

2.2 Application in new systems

2.2.1 Reducing the pipe diameter

The optimum pipe diameter is a result of the optimization of the total transport costs of a DH system. The transport costs consist of the investment cost for the pipe system and the pumping costs.

These two parts show a reverse development concerning the dependence on the pipe diameter. The investment for the pipe is reduced by decreasing the diameter - the operating costs rise with decreasing pipe diameter. The optimum diameter results from the minimum of the sum.

Due to the drag reducing effect, the operating costs can be reduced. The cost curve is moved to smaller costs. On the other hand, due to the application of additives, additional investment is necessary. Therefore, the curve for investment is moving to higher values.

If the decrease in operating costs is dominant, the optimum pipe diameter will shift to lower values. This is shown in figure 2.5.
2.3 Requirements for the application of drag reducing additives in DH-systems

Main requirement for the application of cationic surfactants is the perfect functioning under the changed flow conditions. Due to the addition of surfactants no restriction concerning the tasks, aims and advantages (economic and environmental) may occur. The change of the operation behaviour has to be well known and has to be compensated by carrying out different measures. Necessary requirements for the application of drag reducing additives are:

• **supply guarantee**
  due to the drag and heat transfer reducing effect, the operation behaviour of heat exchanger is influenced. Therefore, especially when operating with maximum heat load, problems concerning the supply are conceivable. The influence on heat exchanger has to be known to calculate and compensate for the heat transfer reduction effect. This can be carried out with the simulation program TenSim.

• **Operating safety and systems safety**
  Due to the changed flow condition and physical and chemical properties of the fluid an influence on water hammering, corrosion behaviour, control etc. is conceivable. This problem is discussed in chapter 3.
• Availability of the system
  A lack of possibilities for water treatment can lead to a restriction of the availability e.g. in case of necessary maintenance of the pipe system. For new systems that are designed especially for the application of surfactant solution, the quality of the additive is of great importance.

• Economic viability
  Main requirement for the introduction of a new technology is the economic viability. If this requirement is not fulfilled, the application of drag reducing additives will not be successful.

2.4 Requirements for the additives

Beside the requirements for DH-systems the cationic surfactants have to guarantee the following properties:

• Range of the drag reducing effect
  The range in which the drag reducing effect occurs has to correspond with the operating conditions in DH-systems. Otherwise, a sufficient effect can not be guaranteed.

• Long term stability
  The drag reducing effect has to be long term stable. Irreversible processes would require a permanent addition of additive. This would lead to an enrichment of surfactant in the system or require a permanent treatment.

• Properties of the surfactants
  The physical and chemical properties of the additives may not cause any danger concerning man or environment. The advantage of saving energy and resources may not be exhausted due to negative properties e.g. toxicity of the surfactants.

2.5 Systems for additive application

Up to now the application of drag reducing additives is only useful in transmission pipelines which are hydraulic separated from the distribution system. Reasons are:

• Due to high flow velocities in transport systems, 70 - 90% of the pressure losses occur in these systems. Therefore, the biggest potential for the application of cationic surfactants lies in transport systems.
• Bottlenecks occur in most cases in transport systems.
• The use in distribution systems requires a lot of modifications to realise the application. Therefore, a lot of investment is necessary and on the other hand only a few savings are achievable.
• Additional safety against the penetration of surfactants into drinking water is realisable if an additional circuit is installed.
3. Influence of Cationic Surfactants on System Parts

To fulfill the requirements described in chapter 2.4 modifications of the system that are considered for the application of drag reducing additives are necessary. Therefore, all significant influences have to be known in advance.

In the following chapters the influences on the system elements of district heating systems are described. After that measures to compensate significant influences are presented. Due to their great importance for the supply particular attention has to be spent on heat exchanger.

3.1 Heat exchanger

In addition to the desired drag reduction the effect of reducing turbulence causes a decrease of the heat transfer between the fluid and the wall. This fact does not influence the heat loss in the transport pipeline significantly, but compared with water a considerable decrease of the heat transfer observes in heat exchangers. This can lead to a significant reduction of heat transfer capacity. The formula for calculating the heat flow rate is [38]:

\[ Q = k \cdot A \cdot \Delta T_m \quad \text{eq. (3.1)} \]

with \( k = \) overall heat transfer coefficient,
\( A = \) heat transfer area and
\( \Delta T_m = \) mean temperature difference of the overall heat transmission.

The overall heat transfer resistance coefficient \( 1/k \) is (for plane plates) defined as:

\[ \frac{1}{k} = \frac{1}{\alpha_1} + \frac{s}{\lambda} + \frac{1}{\alpha_2} + R_F. \quad \text{eq. (3.2)} \]

It consists of the heat transfer resistance coefficients \( 1/\alpha \) between the fluid and the wall, the resistance coefficient of thermal conductivity of the wall \( s/\lambda \) and, in case of fouling of the additional resistances (fouling factor) \( R_F \).

The heat transfer coefficient \( \alpha \) in single phase flow with forced convection is given as a correlation equation of the non-dimensional heat transfer coefficient, the Nusselt number, as a function of Reynolds number \( Re \), Prandtl number \( Pr \) and the geometric parameter \( L/D_i \) [26] (\( Nu = f(Re, Pr, L/D_i) \)). The Nusselt number is defined as follows [40]:

\[ Nu = \frac{\alpha \cdot D_i}{\lambda}, \quad \text{eq. (3.3)} \]

with \( D_i = \) inner diameter of the pipe [m] and
\( \lambda = \) thermal conductivity of the fluid [W/(mK)].
The heat transfer reduction HTR can be described as the percental decrease of the heat transfer coefficient $\alpha$ (or Nu) compared to water (index $w = $ water; $s = $ surfactant) [9]:

$$HTR = \frac{\alpha_w - \alpha_s}{\alpha_w} \cdot 100\%.$$  \hspace{1cm} \text{eq. (3.4)}

Comparing the effects of heat transfer reduction and drag reduction (percental decrease of drag reduction DR that is defined analogously to HTR) the heat transfer reduction is always stronger than the drag reduction under equal conditions [10].

The heat transfer reduction only describes the influence on one part of the total heat transfer in a heat exchanger. Therefore, the percental reduction of the overall heat transfer OHR is defined:

$$OHR = \frac{k_w - k_s}{k_w} \cdot 100\%.$$  \hspace{1cm} \text{eq. (3.5)}

This parameter allows more precise statements about behaviour of heat exchangers.

HTR considers only one of the thermal resistances, while OHR considers three and more thermal resistances. Therefore, the overall heat transfer reduction is always smaller than the heat transfer reduction under the same conditions [24].

In the end the most important parameter that has to be considered is the heat load that is caused by the reduction of the heat transfer properties. The heat output reduction HOR is defined as:

$$HOR = \frac{\dot{Q}_w - \dot{Q}_s}{\dot{Q}_w} \cdot 100\%.$$  \hspace{1cm} \text{eq. (3.6)}

An additional effect concerning the influence on heat exchangers is the increase of the temperature difference $\Delta T_m$. The outlet temperature is increasing on the hot side ($T_{1,\text{out}}$) and decreasing on the cold side ($T_{2,\text{out}}$). This is shown in figure 3.1. As a result, the reduction of the heat load is smaller than the reduction of the overall heat transfer.

![Fig. 3.1: Influence on the temperature difference [24]](image-url)
Exemplary values for HTR, OHR and HOR for a real district heating apparatus (plate heat exchanger) could be:

\[
\begin{align*}
\text{HTR} &= 60 \% \\
\text{OTR} &= 40 \% \\
\text{HOR} &= 10 \%.
\end{align*}
\]

A quite large heat transfer reduction is not necessarily causing a large heat output reduction. The overall heat transfer reduction and the heat output reduction are mainly determined by the conditions of the heat transfers on both sides of the heat exchanger and the strength of the heat transfer reduction.

If the decisive heat transfer resistance (small \( \alpha \)-value) is on the side where the application of additives is planned, the decrease of the overall heat transfer (and therefore also of the heat output) caused by the use of surfactants is strong. If, in the opposite way, the reduced heat transfer resistance is not the decisive one, the overall heat transfer and the heat output will not be decreased significantly.

For this reason the reduction of the heat transfer in transport pipes with insulation is of no importance. Only if the dominant heat transfer resistance is on the side of the surfactants, a considerable overall heat transfer and heat load reduction can be expected. As a characterization of the heat transfer conditions the parameter \( \omega \) is introduced. This ratio between the two heat transfer coefficients is defined the following way [25]:

\[
\omega = \frac{\alpha_1}{\alpha_2}.
\]

\( \omega \) describes the conditions under the original conditions (operation with pure water). In case of \( \omega \gg 1 \) the overall heat transfer reduction is quite small, but in the case of \( \omega \ll 1 \) the influence will be very strong (see figure 3.2).

In addition to \( \omega \), the level of heat transfer reduction is a further decisive parameter that is determining the overall heat transfer coefficient and therefore, the heat output reduction.

The range of heat transfer reduction that appears in typical heat exchangers is shown in table 3.1. In addition, the values of \( \omega \) are also given. The strongest heat transfer reduction occurs in shell and tube heat exchangers. HTR values of 95% in maximum can be caused due to the application of cationic surfactants. Especially in long pipes, strong influences are possible [41]. The reduction of heat transfer coefficients in helical tube and plate heat exchangers are smaller.

<table>
<thead>
<tr>
<th>type of HE</th>
<th>medium</th>
<th>( \omega )</th>
<th>HTR</th>
<th>effect (OHR)</th>
</tr>
</thead>
<tbody>
<tr>
<td>shell &amp; tube</td>
<td>water/steam</td>
<td>0.5 - 3.0</td>
<td>75 - 95</td>
<td>strong</td>
</tr>
<tr>
<td></td>
<td>water/water</td>
<td>1.0 - 4.0</td>
<td>75 - 95</td>
<td>strong</td>
</tr>
<tr>
<td></td>
<td>water/gas</td>
<td>25 - 125</td>
<td>75 - 95</td>
<td>small</td>
</tr>
<tr>
<td>helical tube</td>
<td>water/water</td>
<td>0.6 - 4.0</td>
<td>45 - 70</td>
<td>medium</td>
</tr>
<tr>
<td>plate</td>
<td>water/water</td>
<td>0.5 - 2.0</td>
<td>45 - 70</td>
<td>medium</td>
</tr>
</tbody>
</table>

Tab. 3.1: \( \omega \) and HTR range of typical heat exchangers

Combining the parameters \( \omega \) and HTR, strong influences occur in shell and tube condensers and shell and tube heat exchangers that operate with water on both sides.
Figure 3.2 shows an example of the influence of $\omega$ and HTR on the overall heat transfer reduction OHR for typical apparatus under a certain operating condition ($\lambda/s = 24,000 \text{ W/(m}^2\text{K)}$, $\alpha_1 = 5,500 \text{ W/(m}^2\text{K)}$, $R_F = 0 \text{ (m}^2\text{K)/W)}$).

**Fig. 3.2:** OHR as a function of HTR and $\omega$ for typical DH-heat exchanger [25],
($\alpha_1 = 5,500 \text{ W/(m}^2\text{K)}$, $\lambda/s = 24,000 \text{ W/(m}^2\text{K)}$, $R_F = 0 \text{ (m}^2\text{K)/W)}$)

Figure 3.2 confirms the statement of table 3.1. Shell and tube condensers and water/water shell and tube heat exchangers are influenced very strong. On the other hand, shell and tube heat exchangers that are operating with gas and water, the influence is only small, due to the high value of $\omega$. Values of OHR are only about 20% if HTR is about 90% and the values of $\omega$ are relatively low.

However, a general statement about the efficiency of heat exchangers cannot be made without the exact calculation under the regard of operation and construction parameters. Therefore, the simulation TenSim has been developed. With this simulation tool, tube bundle-, plate- and helical tube heat exchangers can be calculated when applying water or drag reducing surfactant solutions [5].

In the following chapters, the results of the investigations of tube bundle-, helical tube- and plate heat exchangers as the basis of the simulation tool are presented.
3.1.1 Tube bundle heat exchangers

The characteristic of the HTR in solutions with additives is strongly dependent on the tube length and the ratio L/D respectively. With increasing tube length the HTR is increasing if the Reynolds number is kept constant. The reason for this behaviour is the percentage of the good heat transfer in the range of the entrance length in the pipes (50 - 100-Di). In very long tubes (L = 600 - 1,000-Di) the heat transfer is nearly determined by a laminar flow. The good heat exchange in the entrance length is only of minor importance for the total heat transfer in the pipe. In comparison to this the HTR in very short pipes is only small, due to the high percentage of turbulent flow.

<table>
<thead>
<tr>
<th>conditions</th>
<th>results</th>
</tr>
</thead>
<tbody>
<tr>
<td>surfactant</td>
<td>Habon</td>
</tr>
<tr>
<td>concentration</td>
<td>1,000 wppm</td>
</tr>
<tr>
<td>Reynolds</td>
<td>20,000</td>
</tr>
</tbody>
</table>

**Tab. 3.1:** Influence of the pipe length on HTR

For the calculation of solutions with drag reducing additives in straight tubes one formula has been developed which exactly describes the technical relevant range [41]:

\[
Nu = \frac{3}{4} \sqrt[3]{3.66^3 + 2.191 \cdot Re \cdot Pr \cdot \frac{D_i}{L} \cdot \left(1 + \frac{L_e}{L}\right)^{0.249}}
\]

with: \( D_i \): inner diameter [m],
\( L \): length of the pipe [m] and
\( L_e \): entrance length [m].

\[
L_e = 0.075 \cdot D_i \cdot Re_W
\]

\( Re_W \) is built with the material properties at wall temperature.

The guilty range of this formula is as follows:
- length of the tube: \( 50 < L/D < 800 \),
- Reynolds number: \( 1,000 < Re < 100,000 \),
- temperature: \( 40 \, ^{\circ}C < T < 180 \, ^{\circ}C \) and
- Prandtl number: \( 0.95 < (Pr/Pr_w) < 1.05 \).

Equation 3.8 and 3.9 are the basis for calculating the behaviour of shell and tube apparatus using drag reducing additives on the pipe side. These formulas are implemented into the simulation program. In the following tables a calculation - carried out with TenSim - for a typical condenser is presented.
The data of the condenser are shown in table 3.3.

<table>
<thead>
<tr>
<th>Design</th>
<th>Nominal Operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>diameter of the pipes</td>
<td>23 mm</td>
</tr>
<tr>
<td>length of the pipes</td>
<td>6,000 mm</td>
</tr>
<tr>
<td>number of pipes</td>
<td>1,800</td>
</tr>
<tr>
<td>heat exchange area</td>
<td>814 m²</td>
</tr>
<tr>
<td>number of pass</td>
<td>1</td>
</tr>
<tr>
<td>appl. of surfactants</td>
<td>pipe side</td>
</tr>
</tbody>
</table>

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>nominal heat load</td>
<td>50 MW</td>
</tr>
<tr>
<td></td>
<td>satur. steam temp. T_s</td>
<td>85.5 °C</td>
</tr>
<tr>
<td></td>
<td>entrance temp. T_in</td>
<td>60 °C</td>
</tr>
<tr>
<td></td>
<td>outlet temp. T_out</td>
<td>68.4 °C</td>
</tr>
<tr>
<td></td>
<td>mass flow (steam)</td>
<td>22.8 kg/s</td>
</tr>
<tr>
<td></td>
<td>mass flow (water)</td>
<td>1,500 kg/s</td>
</tr>
</tbody>
</table>

**Tab. 3.1:** Data of the condenser

In table 3.4 and 3.5 the results and the thermic parameters HTR, OHR and HOR of four operating points are presented.

<table>
<thead>
<tr>
<th>Operating point</th>
<th>Nom.</th>
<th>Q̄_N</th>
<th>HTR</th>
<th>OHR</th>
<th>HOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>50</td>
<td>92%</td>
<td>75%</td>
<td>71%</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>89%</td>
<td>80%</td>
<td>72%</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>20</td>
<td>87%</td>
<td>81%</td>
<td>70%</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>10</td>
<td>82%</td>
<td>76%</td>
<td>61%</td>
<td></td>
</tr>
</tbody>
</table>

**Tab. 3.1:** Results of the calculation, part 1 (thermic parameters)

The calculation clearly shows that the application of drag reducing additives in steam condensers can lead to a significant influence. Values of HOR of more than 60% show that condensers can only be used if measures are carried out to improve the operating behaviour with drag reducing additives.

Using drag reducing additives on the shell side, due to stronger turbulences, the HTR is significantly smaller than on the tube side. An exactly precalculation of the heat transfers which will occur on the shell side using surfactants is not possible due to the complicated flow conditions.
(even with pure water). Coarse estimations by using results that have been achieved during several projects can be carried out. Figure 3.3 shows results of the application on the shell side with and without support plates.

![Graph showing OHR vs flow rate for pipe side and shell side with and without support plates.](image)

**Fig. 3.3:** OHR if using surfactants on the shell side with and without support plates

The OHR on the shell side is significantly smaller than on the pipe side. The curves show a maximum, due to the two different effects of \( \omega \) and HTR. With increasing flow rate the HTR and therefore the OHR increases, too. On the other hand, the increase of \( \omega \) with rising flow rates leads to a decrease of OHR. The parameter \( \omega \) gets dominant with increasing flow rate. That causes the maximum.

Concerning the hydraulic behaviour the pressure loss is slightly decreasing when using surfactant solutions on the pipe side. The value of this drag reduction is in consideration of all pressure losses around 10 to 25 % [41]. If drag reducing additives are used on the shell side there is an increase of the pressure drop because of the increase of the viscosity. In the relevant flow rate range the increase of the pressure loss in plants without cross baffles is around some percentages. In comparison to this the value in plants with support plates is between 20 and 40 %.

### 3.1.2 Helical tube heat exchangers

The HTR in helical tubes is compared with straight pipes under constant conditions significantly smaller. The maximum reduction of HTR in helical tubes is about 80 %. The reason for less influence of the drag reducing additives is the build up of a secondary flow which opposes the behaviour of the fluid to turn laminar. Due to the secondary flow the hydrodynamic and thermal effects in the tube entry of helical tubes can be neglected.
Assuming that the Nusselt numbers at wall shear stresses below the critical value are independent on temperature and concentration of the additive solution and that there is no influence of the curvature ratio $\delta = D_l/D_H$ on the heat transfer in the range of small Reynolds numbers, the heat transfer coefficients of solutions with drag reducing additives can be determined with the following correlations [41]:

$$
\text{Nu} = 0.073 \cdot \text{Re}^{0.694}
$$

for the range of validity: Reynolds number: $3000 < \text{Re} < 10,000$

$$
\text{Nu} = 0.05 \cdot \left( \frac{D_l}{D_H} + 0.1 \right) - 1.69 \cdot \text{Re}^{0.694} + 0.1
$$

for the range of validity: Reynolds number: $10,000 < \text{Re} < 150,000$

curvature ratio: $0.0882 < \delta < 0.105$ and $0.036 < \delta < 0.105$

Analog to shell and tube apparatus the equations 3.10 and 3.11 are implemented into the simulation program. The following tables show examples for the calculation of the conditions in helical tube heat exchangers when using drag reducing additives. Table 3.6 shows the data of the apparatus that has been investigated.

<table>
<thead>
<tr>
<th>design</th>
<th>nominal operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>diameter of the pipes</td>
<td>nominal heat load</td>
</tr>
<tr>
<td>14 mm</td>
<td>310 kW</td>
</tr>
<tr>
<td>length of the pipes</td>
<td>entrance temperature</td>
</tr>
<tr>
<td>2,900 to 3,300 mm</td>
<td>pipe side $T_{in}$</td>
</tr>
<tr>
<td>number of pipes</td>
<td>60 °C</td>
</tr>
<tr>
<td>34</td>
<td>entrance temperature</td>
</tr>
<tr>
<td></td>
<td>shell side $T_{in}$</td>
</tr>
<tr>
<td></td>
<td>100 °C</td>
</tr>
<tr>
<td>height of the bundle</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>430 mm</td>
<td>pipe side $M_1$</td>
</tr>
<tr>
<td>heat transfer area</td>
<td>3.8 kg/s</td>
</tr>
<tr>
<td>4 m$^2$</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>pipe arrangement</td>
<td>shell side $M_2$</td>
</tr>
<tr>
<td>staggered</td>
<td>3.8 kg/s</td>
</tr>
<tr>
<td>appl. of surfactants</td>
<td>pipe side</td>
</tr>
</tbody>
</table>

Tab. 3.1: Data of the helical tube heat exchanger

The results of the simulation for four different operating points are presented in table 3.7 and 3.8. The influence of the surfactant solution is significantly smaller compared to the calculation for the shell and tube heat exchanger with straight pipes. The HTR values are below 60 % for every operating point.

At small heat loads ($0.2 - \dot{Q}_W$) the reduction of the heat output is nearly negligible. However, decisive for the supply guarantee is the nominal heat load. For the nominal value of $\dot{Q}$ the heat output reduction is not negligible. A HOR-value of more that 20 % occurs.

Therefore, analog to the application of drag reducing additives in shell and tube heat exchangers, the operating conditions for helical tube apparatus have to be improved, too.
For the application on the shell side the same statement can be made as for shell and tube apparatus. The influence will be smaller than on the pipe side. Due to the complicated flow condition the exact calculation is not possible. In this case experimental investigations for typical designs have to be carried out.

The influence of surfactant solutions on the hydraulic behaviour in helical tubes is less compared to that in straight tubes. Therefore the reduction of the resistance is smaller for helical tube exchangers referring to the technical relevant scale. Concerning the application on the shell side, the statements for shell and tube heat exchangers can be applied.

### 3.1.3 Plate heat exchangers

Plate heat exchangers are mainly used in transmission stations for the hydraulic separation of transport system from distribution network. Due to their modular design they have compared to other kinds of heat exchangers substantial advantages:

- high heat transfer coefficients due to profiled plates (high turbulence) and small wall thicknesses,
- small heat losses and only a small required place due to the very compact design,
- good adaptability to different requirements because of the possibility to combine different kinds of plates and
- simple design and therefore, simple installation and maintenance.
Due to these advantages in most cases plate heat exchangers are installed in transmission stations to separate the transport system from the distribution network [14]. The influence of drag reducing additives on plate heat exchangers is comparable to that in helical tube apparatus. Due to the turbulence creating profile the maximum HTR is about 60 to 70 % (see figure 3.2). A distinction between the side on which the surfactants are used is not necessary. The geometric conditions are identical on both sides of the apparatus.

The calculation of heat transfer when using cationic surfactants can be carried out as follows [22]:

\[ \text{Nu}_s = K \cdot \text{Re}^m \cdot \text{Pr}^{0.4} \cdot \left( \frac{Pr}{Pr_w} \right) \cdot f_T = \text{Nu}_w \cdot f_T. \quad \text{eq. (3.12)} \]

Equation 3.12 describes the Nusselt number of water corrected with the parameter \( f_T \). The correction parameter \( f_T \) has to be determined experimentally. In the range of small Reynolds numbers \( f_T \) is increasing up to a characteristic Reynolds number (\( \text{Re}_c \)). Above this value \( f_T \) is constant up to the critical Reynolds number (critical wall shear stress). \( \text{Re}_c \) is dependent on the geometry, the kind of plates, the concentration, temperature etc.

For small Reynolds numbers \( \text{Re} < \text{Re}_c \), \( f_T \) can be calculated according to the following equation [22]:

\[ f_T = K_a \cdot \text{Re}^f. \quad \text{eq. (3.13)} \]

Above the characteristic Reynolds number and below the critical value, \( f_T \) can be calculated as follows:

\[ f_T = K_b = 1 - \frac{\text{HTR}}{100}. \quad \text{eq. (3.14)} \]

In this range the characteristic of surfactant solution is parallel to that of pure water until the critical wall shear stress is reached. In this range the maximum heat transfer reduction occurs. Values of HTR from 60 to 70 % in maximum have been determined up to now. For flow rates above the critical velocity the solution shows the same behaviour as water.

In any case, due to the complicated flow conditions, experimental investigations are necessary for a good description of heat transfer and pressure drop. A general description is not possible because of the high number of parameters that influence the heat transfer. Estimations can be done by means of known results for similar kinds of plates.

Concerning the most important parameters that influence the behaviour of plate heat exchangers the HTR increases with the length of the plate and the angle of the profile. Other parameters that are investigated at the moment are the breadth of the plate and the width of the gap between the plates.

By means of a 4 MW-plate heat exchanger that is used for a hydraulic separation in a district heating network simulations of the behaviour when using drag reducing additives have been carried out. The important data are shown in table 3.9.

Table 3.10 and 3.11 present the results of the calculations for 4 different operating conditions. The results show a constant HTR for all operating points. The resulting OHR is about 30 %. Due to the behaviour of the average temperature difference the HOR is below 10 %. 
Due to the advantages mentioned above and the relatively small influence plate heat exchanger should be used when applying drag reducing additives.

<table>
<thead>
<tr>
<th>design</th>
<th>nominal operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>hydr. diameter</td>
<td>6.15 mm</td>
</tr>
<tr>
<td>width of the gap</td>
<td>3.6 mm</td>
</tr>
<tr>
<td>breadth of the gap</td>
<td>444 mm</td>
</tr>
<tr>
<td>number of plates</td>
<td>83</td>
</tr>
<tr>
<td>heat exchange area</td>
<td>73 m²</td>
</tr>
<tr>
<td>thickness of the plate</td>
<td>0.7 mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>nominal heat load</th>
<th>entrance temperature primary side $T_{1,in}$</th>
<th>entrance temperature secondary side $T_{2,in}$</th>
<th>mass flow rate prim.</th>
<th>mass flow rate sec.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 MW</td>
<td>100 °C</td>
<td>60 °C</td>
<td>30.6 kg/s</td>
<td>30.6 kg/s</td>
</tr>
</tbody>
</table>

Tab. 3.1: Data of the plate heat exchanger

<table>
<thead>
<tr>
<th>operating point</th>
<th>$\dot{Q}_N$</th>
<th>$\alpha$ [W/(m²K)]</th>
<th>$k$ [W/(m²K)]</th>
<th>$\dot{Q}$ [MW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$\dot{Q}_N$</td>
<td>18,260</td>
<td>6,336</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>0.6 $\dot{Q}_N$</td>
<td>11,904</td>
<td>4,603</td>
<td>2.4</td>
</tr>
<tr>
<td>3</td>
<td>0.4 $\dot{Q}_N$</td>
<td>8,571</td>
<td>3,541</td>
<td>1.6</td>
</tr>
<tr>
<td>4</td>
<td>0.2 $\dot{Q}_N$</td>
<td>4,926</td>
<td>2,191</td>
<td>0.8</td>
</tr>
<tr>
<td>$\alpha$ [W/(m²K)]</td>
<td>$k$ [W/(m²K)]</td>
<td>$\dot{Q}$ [MW]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>18,260</td>
<td>6,336</td>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>11,904</td>
<td>4,603</td>
<td>2.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8,571</td>
<td>3,541</td>
<td>1.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4,926</td>
<td>2,191</td>
<td>0.8</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Tab. 3.1: Results of the calculation, part 1

<table>
<thead>
<tr>
<th>operating point</th>
<th>$\omega$</th>
<th>HTR</th>
<th>OHR</th>
<th>HOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{Q}_N$</td>
<td>1</td>
<td>54 %</td>
<td>29.1%</td>
<td>9 %</td>
</tr>
<tr>
<td>0.6 $\dot{Q}_N$</td>
<td>1</td>
<td>54.2%</td>
<td>31.5%</td>
<td>8 %</td>
</tr>
<tr>
<td>0.4 $\dot{Q}_N$</td>
<td>1</td>
<td>54 %</td>
<td>32.8 %</td>
<td>7 %</td>
</tr>
<tr>
<td>0.2 $\dot{Q}_N$</td>
<td>1</td>
<td>54.2 %</td>
<td>34.5 %</td>
<td>6 %</td>
</tr>
</tbody>
</table>

Tab. 3.1: Results of the calculation, part 2

Concerning the pressure drop the surfactant solutions show no strong influence. For the most kinds of plates the characteristic for additive solution is nearly the same as for pure water. In some cases a small increase of the pressure drop occurs in a range of low Reynolds numbers. With increasing flow velocity the characteristic is getting near the water curve and in some cases crossing it and reaching a range of a small drag reduction. For technical application the influence is not of great importance.
3.1.4 Summary of the behaviour of heat exchangers

The application of drag reducing additives causes a strong influence of the heat transfer properties. This HTR can lead to a significant deterioration of the heat output and therefore to a restriction of the supply guarantee. Table 3.12 shows the maximum values for HTR and HOR and in addition the values of the heat transfer coefficient ratio $\omega$.

<table>
<thead>
<tr>
<th>apparatus</th>
<th>max HTR</th>
<th>$\omega$</th>
<th>max HOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>shell and tube</td>
<td>95 %</td>
<td>1 - 4</td>
<td>70 %</td>
</tr>
<tr>
<td>helical tube</td>
<td>80 %</td>
<td>0,6 - 4</td>
<td>30 %</td>
</tr>
<tr>
<td>plate</td>
<td>70 %</td>
<td>0,5 - 2</td>
<td>10 %</td>
</tr>
</tbody>
</table>

Tab. 3.1: Maximum HTR, HOR and $\omega$

This problem has to be considered for every heat exchanger that is installed in a DH system. Before applying drag reducing additives the influence on every apparatus has to be calculated and compensated if necessary. Therefore the simulation program TenSim has been developed. Heat exchangers - especially plate heat exchangers, which have a lot of advantages in heat transmission stations - can be calculated, measures can be developed and tested.

3.1.5 Measures to improve the heat output behaviour of heat exchangers

An obvious possibility in the exchange of a strongly influenced apparatus against an other heat exchanger - e.g. a plate heat exchanger for a shell and tube bundle apparatus. For the design - especially for plate heat exchanger - the simulation program TenSim can be used. The advantages of that measure is that the new heat exchanger can be especially designed for the application of drag reducing additives. Disadvantages are the additional space demand and the high costs.

Another possibility is the installation of an additional apparatus. According to the operating conditions the additional apparatus can be installed in series (higher pressure drop) or parallel. The advantages and disadvantages are the same as discussed for the exchange of an apparatus apart from the cost of this measure which are smaller.

A hydraulic separation (tertiary circle) has the advantage that the new apparatus can be designed especially for drag reducing surfactant solutions. Great disadvantages are the very high costs, the big space demand that is necessary to build up a tertiary circle with additional pumps, pressure maintenance, heat exchangers and other equipments and the additional temperature difference that is necessary to operate the additional heat exchangers.

Due to the great disadvantages - especially the very high costs due to the new installations - this possibility is not very realistic.
Measures to improve the heat output behaviour of heat exchangers

Furthermore for some heat exchangers it is possible to increase the heat transfer area. Especially the area of plate heat exchangers can be increased very easily by increasing the number of plates. The heat transfer area of shell and tube apparatus can be increased by exchanging the tube bundle for an other one with pipes of a smaller diameter.

This measure is - especially for plate apparatus - very easy to verify. Due to the parallel installation of additional area the flow velocity and the pressure drop are decreased. On the other hand the heat transfer coefficient is decreasing due to lower flow velocities. Therefore the necessary increase of the plate number is more than proportional. The realisation of the exchange of the bundle in shell and tube heat exchangers takes more expenditure. Concerning the operating with a tube bundle of smaller pipes the flow velocity is increasing and therefore the heat transfer coefficient is increasing, too. On the other hand the L/D₁-ratio rises and therefore the improvement of heat transfer is slightly made worse.

The increase of the heat transfer area of helical tube heat exchanger is - in most cases - not possible because normally the shell is welded with the tube bundle. The expenditure would be too great to realise this measure.

Rising the flow velocity of the district heating system leads to higher heat transfer rates. But on the other hand an increase of the flow velocity leads to higher pressure drops and to higher return temperatures. Investigations of a plate heat exchanger have shown that an increase of the mass flow rate of 10 - 20 % is necessary to reach the heat output conditions for water. This is accompanied by an increase in temperature of 3 - 6 degrees [23].

Due to this significant change of the operating conditions this measure should not be applied.

A further possibility could be an increase in flow velocity only in the heat exchanger. This could be realized e.g. by turning a single pass apparatus into three pass heat exchanger (for shell and tube apparatus as well as for plate heat exchanger). When exceeding the critical wall shear stress the conditions for water could be reached. A dangerous disadvantage exists in times of low heat loads (in summer). In this case the flow velocities are very small. Than the flow velocity will fall below the critical value. This can lead to alternating operating points (drag reduction - no drag reduction) and therefore to dangerous oscillations of the whole system. Due to this complicated operation this measure should not be applied.

A possibility that is also investigated in the frame of Annex IV is the increase of turbulence. An increase of turbulence can be reached by installing obstacles inside the pipes or exchanging plates for plates with greater angles of the profile (hard plates). Using hard plates can be combined with the possibility to increase the number of plates. Considering the shell side, an increase in turbulence can be achieved by installing cross baffles.

In first investigations with spiral springs that have been pulled through a pipe of a double pipe heat exchanger, the heat transfer could be improved significantly. The heat transfer coefficients with spiral springs and drag reducing additives were in some cases significantly better than with pure water without obstacles.

A great disadvantage of all these measures to increase turbulence is the very strong increase in pressure drop. Comparing with the case for water without turbulence increasing measures an increase of pressure drop of 100 - 700 % can occure - especially when installing obstacles inside pipes.

Furthermore the installation of obstacles inside pipes with a great number of long pipes with a small diameter is of enormous expenditure. A question that has to be answered is the mechanical stability of the pipes when obstacles are inserted.
The heat load can be increased when rising the average temperature difference (e.g. increasing the saturation temperature in steam condensers). This is a severe change of operating conditions and on the other hand a measure that causes a strong increase of heat generation costs in CHP plants. Finally calculations with TenSim have shown that the necessary increase of e.g. the saturation temperature is too strong to be able to realize this measure without getting big problems. Therefore an increase of the average temperature difference between the two sides of a heat exchanger should not be considered.

A further possibility is the change of the two sides of an apparatus. Changing the shell and tube side leads to better heat transfer conditions. A problem is the requirement that the operation conditions have to fit. On the other hand the realization will not be very easy if big heat exchangers with supply pipes of large diameters are considered. Furthermore the heat transfer conditions can not be improved sufficiently because the heat transfer reduction can not be suppressed completely. Therefore additional measures have to be carried out.

Considering the statements above the best possibility to improve the behaviour of heat exchangers is (for plate heat exchangers) the increase of plate numbers in combination with the use of hard plates. With the simulation TenSim this measure can be calculated if the influence of drag reducing additives can be described. For the other apparatus the installation of an additional heat exchanger should be considered. After having finished the investigations concerning the installation of obstacles inside pipes, a statement about the quality of this measure can be made.

### 3.2 Pumps

The KSB AG Frankenthal has carried out special investigations about the behaviour of centrifugal pumps in systems that operate with drag reducing additives. Several pumps that are typical for district heating systems have been examined.

The result was that there is no significant direct influence on centrifugal pumps concerning the characteristic curve, the efficiency, corrosion properties or NPSH values. The only effect is an indirect influence due to the shifting of the characteristic curve of the district heating system. Therefore the operating point of the pump is shifting, too. Under bad conditions this can lead to a small decrease of the efficiency. Up to now no decrease of the efficiency could be observed in several field tests.

Because there is no significant influence on centrifugal pumps there is no necessity in carrying out modifications or measures. If - under very unfavourable conditions - a significant influence due to the shifting of the operating point to lower pressure drops occurs a change of the impeller is a possibility to compensate unwanted effects.
3.3 Heat meters

Heat meters consists of a flow meter, a thermometer and an arithmetic unit. Due to the change of flow conditions an influence on flow meters and thermometers (HTR) is conceivable. **Flow meters** that are used in district heating systems are the following:

- impeller meters (single- and multi-jet),
- Woltmann flow meters,
- meters that make use of the backpressure (dynamic pressure) or the active pressure method (venturi tubes, orifice plates, jets etc.),
- ultrasonic meters
  - ultrasonic meters that operate with a signal that only passes a small area of the flow-profile,
  - ultrasonic meters that operate with a signal that passes the complete area or a certain part of the flow-profile (e.g. the LLL - Lambda Locked Loop principle) and
- magnetic inductive meters.

The accuracy on heat meters has been investigated at the University of Dortmund. The strongest influence show impeller meters. They have a negative inaccuracy of 80 % [23]. The following table shows the tested heat meters and the results.

<table>
<thead>
<tr>
<th>kind of meter</th>
<th>influence</th>
<th>suitability</th>
<th>remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>single-jet impeller meter</td>
<td>strong</td>
<td>no</td>
<td>correction with critical wall shear stress possible</td>
</tr>
<tr>
<td>multi-jet impeller meter</td>
<td>strong</td>
<td>no</td>
<td></td>
</tr>
<tr>
<td>Woltmann-flow meter</td>
<td>medium</td>
<td>no - limited</td>
<td></td>
</tr>
<tr>
<td>magnetic inductive</td>
<td>no</td>
<td>yes</td>
<td>best possibility</td>
</tr>
<tr>
<td>ultrasonic (one measuring path)</td>
<td>medium</td>
<td>no</td>
<td></td>
</tr>
<tr>
<td>ultrasonic (LLL-principle)</td>
<td>medium - low</td>
<td>no - limited</td>
<td></td>
</tr>
<tr>
<td>ultrasonic (passing the whole flow profile)</td>
<td>low</td>
<td>yes</td>
<td>possible if: ( d/D_t &lt; 0.4 )</td>
</tr>
<tr>
<td>orifice plate</td>
<td>medium - low</td>
<td>limited</td>
<td>possible if: ( d/D_t &lt; 0.53 )</td>
</tr>
<tr>
<td>venturi tube</td>
<td>medium - low</td>
<td>limited</td>
<td></td>
</tr>
<tr>
<td>Pitot tube flow meter</td>
<td>medium</td>
<td>no</td>
<td></td>
</tr>
</tbody>
</table>

**Tab. 3.1:** Influence on flow meters [23][25][34]
For the flow measuring of surfactant solutions only magnetic inductive and ultrasonic meters which use a pass over the whole flow profile can be used without restrictions. Orifice plates and venturi tubes can be used under special conditions concerning the aperture ratio. For orifice plates the ratio \( d/D_h \) has to be below 0.4 and for venturi tubes below 0.53.

Measures to guarantee a correct heat metering that is the condition for a correct heat accounting are the exchange of influenced meters for meters that also work correctly with drag reducing additives. Another possibility is the change of an influenced meter to the hydraulic separated side that works with water. Therefore the operating conditions have to be suitable. The last possibility is the installation of an (influenced) device on the water side.

Due to the decrease of heat transfer coefficient an influence on thermometers is conceivable. This influence consists in an increase of indication time. This can lead to dynamic errors of measurement. But compared to the slow rates of change of the total system this influence on the indication time is negligible. Therefore the influence on thermometers is not significant and no measures have to be carried out.

### 3.4 Other influences

Concerning the application of drag reducing additives the following system elements and general influences have been investigated:

- pipes, angles and fittings,
- tanks
- pressure maintenance
- water hammering
- corrosion behaviour
- environmental aspects
- effects caused by physical properties and
- dosing system.

In the following chapters the influences and measures that can be carried out to compensate negative effects are presented.

#### 3.4.1 Pipes, angles and fittings

In straight pipes a drag reduction up to 85 % has been found when using drag reducing additives. Considering angles like knees and elbows, fittings etc. the pressure drop with cationic surfactants can be considered as constant compared to water (no influence) [10]. Due to this behaviour elements like angles, fittings etc. reduce the reachable drag reduction in pipe systems. Concerning the maximum drag reduction system with straight, long pipes have to be preferred.
Therefore, the calculation of the pressure drops of knees, elbows, fittings etc. is carried out for pure water in the simulation program.

In district heating systems fittings are used as shut-off devices or control valves. Due to the change in flow condition a change of the characteristic curve of control valves is conceivable. Therefore measurements have been carried out. The characteristic curve of several typical control valves have been determined under the conditions of drag reducing additives.

A significant effect that could influence the control properties could not be found. Neither in laboratory tests nor in full scale investigations effects on the operating conditions of control valves could be observed.

Therefore measures are unnecessary.

### 3.4.2 Tanks

Tanks are used in DH-systems - especially in systems with CHP plant production [14]- to decouple the production of heat and electricity and to improve the supply guarantee. Due to the application of surfactants two effects are conceivable:

- influence of the convective heat transfer inside the tank. An increase of the rate of convection would lead to an increase of the mixing between the hot and the cold layer and therefore to a decrease of the efficiency of the tank and
- an effect on the operation behaviour due to precipitation of surfactants if the temperature is below the Krafft temperature.

A further problem is the amount of surfactants that have to be dosed in addition due to volume of the tank. This can be bigger than the volume of the complete pipe system. This leads to additional costs for a quantity of additives that is not in action. A possibility to solve this problem is a hydraulic separation of the tank from the rest of the system. How far this measure is sensible has to be shown in an all-embracing analysis including a cost calculation.

Measurements of the heat exchange properties of tanks which contained surfactant solutions did not show any significant influence. The boundary layer between the hot and cold fluid was as stable as for pure water. Therefore the efficiency of tanks is not been influenced, too.

Concerning the precipitation the agglomerates that are forming below Krafft temperature are like cotton waddings and very easy soluble if temperature rises above Krafft point again. A significant influence is not to expect.

### 3.4.3 Pressure maintenance

Pressure maintenances are divided into static and dynamic equipments. The components of pressure maintenances are not influenced (compare chapter 3.2 and 3.4.1). Concerning the membranes of some static maintenances no chemical effects have to be expected.
The only problem is - in some cases - the realization of dynamic pressure maintenances. If the realization is like shown in figure 3.4, the hydraulic connection between transport- and distribution system has to be cut off. Otherwise surfactants would penetrate into the distribution system. This has - up to now - to be avoided.

Fig. 3.4: Example of a dynamic pressure maintenance [30]

To avoid a penetration of surfactants into the distribution system different measures can be carried out. A simple solution is the installation of an autarkic pressure maintenance. Another possibility is the treatment of the surfactant solution. Therefore an ultrafiltration facility can be installed [16]. With cross flow ultrafiltration the concentration of the permeate can reach the minimum of CMC I. The CMC I at temperatures around 70 °C is about 20 wppm for Habon-G and below 20 wppm for Dobon-G. This means that a certain concentration (far below the effective concentration) has to be tolerated in the distribution system [17].

How far this can be tolerated has to be shown in a complete analysis of the system under consideration of the boundary conditions.

3.4.4 Water hammering

Due to unforeseeable breakdowns unsteady processes and therefore water hammering can occur. This means a great danger for district heating systems. Processes that can lead to water hammering are [28]:

- control processes,
- breakdown of the pressure maintenance,
- sudden shut-down of shut-off devices,
• breakdown of circulation pumps and
• big leakages or pipe ruptures.

Measurements concerning the behaviour of water hammering in systems that operate with drag reducing additives have been carried out at the University of Dresden [21]. No negative effects could be established on both sides of a quick-action gate valve. On the back side of the valve even a significant decrease of the water hammering due to fast condensing processes could be observed. Therefore an improvement of water hammering conditions occurs.

A negative effect is the increasing risk of water hammering due to the increase of flow velocity if applying surfactant solutions for rising the capacity (see chapter 2.1.2). But this is a general problem and not caused by drag reducing additives.

Measures that can be carried out are the general measures (the same as for pure water). These are:
• increasing the shutting time of valves,
• increasing the inertia of the moving parts of circulation pumps,
• building a bypass at the circulation pump or
• building an antifluctuator.

3.4.5 Corrosion behaviour

Typical materials for district heating systems have been investigated at the DECHEMA, Germany [8]. All investigated materials show no increase of the corrosion rate. Up to now no problems concerning corrosion properties could be observed under normal operating conditions. Rather a significant decrease of the corrosion rates of pumps could be established [27].

Therefore no measures to compensate an influence of surfactant solutions on the corrosion behaviour have to be carried out.

3.4.6 Environmental aspects

Standard values of decomposition and aquatic toxicity compared with real data of Dobon-G and other quaternary ammonium compositions QACs (commercial used surfactants) are shown in table 3.14. The standard values have been set from the German „Hauptausschuß Detergentien“ [15].

The data for Dobon-G are all above the standard values. Compared with the commercial used surfactants Dobon-G has a significant lower toxicity to bacteria (LD₅₀ > 1,000 mg/l). The total decomposition of the surfactants which are suitable as drag reducing additives has not been investigated up to now. For cationic surfactants of comparable structures the proof of total decomposition has already been carried out.
The most important parameter concerning the toxicity is the LD\textsubscript{50}-value (rat) in mg per kg body weight. The LD\textsubscript{50}-values are listed in table 3.15. Apart from the softener DSDMAC, the danger due to the oral toxicity of Dobon-G is smaller compared to the other commercial used substances [1].

<table>
<thead>
<tr>
<th>decomposition</th>
<th>ecological toxicity</th>
</tr>
</thead>
<tbody>
<tr>
<td>primary</td>
<td>total</td>
</tr>
<tr>
<td>standard value</td>
<td>&gt; 80 %</td>
</tr>
<tr>
<td>QAC</td>
<td>94 %</td>
</tr>
<tr>
<td>Dobon-G</td>
<td>&gt; 90 %</td>
</tr>
</tbody>
</table>

**Tab. 3.1:** Decomposition rates and toxical data of Dobon-G and other QACs [19]

<table>
<thead>
<tr>
<th>quaternary ammoniumcompositions (QAC)</th>
<th>LD\textsubscript{50} [mg/kg KG]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dobon-G</td>
<td>&gt; 2,000</td>
</tr>
<tr>
<td>Hexadecyltrimethylammoniumbromide (CTAB)</td>
<td>410 - 430</td>
</tr>
<tr>
<td>Distearyldimethylammoniumchloride (DSDMAC)</td>
<td>&gt; 5,000</td>
</tr>
<tr>
<td>Stearylbenzyldimethylammoniumchloride</td>
<td>1,000</td>
</tr>
<tr>
<td>Cetylbenzyldimethylammoniumchloride</td>
<td>250 - 300</td>
</tr>
</tbody>
</table>

**Tab. 3.1:** Oral toxicity of QACs [19][31]

The values of Habon-G are quite similar to those of Dobon-G. Therefore it is abstained from a presentation of these data.

Concerning the counterions naphthoate and salicylate there is no fear of an additional danger, despite of a relative bad biological decomposition [4].

Due to their toxic behaviour no additional regulations have to be followed in Germany. Additional installations etc. have not to be carried out. Concerning the other member countries of the IEA experts group advanced fluids the regulations can be gathered from the report of B&S „Survey of Environmental Restrictions to the Use of Additives in District Heating and Cooling Systems“.

3.4.7 **Effects caused by physical properties**

The most important physical properties of cationic surfactants that can be used as drag reducing additives in district heating and cooling systems are due to their chemical structure the high adsorption and dispersing capability.
Adsorption behaviour

Due to the strong affinity to surfaces and therefore the high adsorption capability a loss of concentration after adding the surfactants occurs. Due to the coating of every free surface the amount of additive has to be bigger corresponding to that.

Therefore the concentration has to be controlled in short time intervals and the quantity of surfactant has to be increased until the adsorption equilibrium is reached. Systems which have a very large free surface (high roughness, high degree of fouling) can show an adsorption loss of ca. 50%. In systems which have a high degree of fouling the water content should be exchanged.

Dispersing properties

Surfactants detach loose precipitations. This can lead to a blocking of filters shortly after the first dosing of additive. It can be necessary to clean filters due to the dispersion of loose particles. This effect has to be considered positively because the cross section of the pipe is enlarged, the system is cleaned and fouling effects e.g. in heat exchangers are prevented.

3.4.8 Dosage device

Apart from the modification of the system a new dosing device has to be installed. A good consistence for the additive that has to be dosed is about 10% for Dobon-G and 50% for Habon-G. The diluted additive should be pumped with a reciprocating pump, a screw pump etc. on the suction side of the circulation pump. For diluting the additive hot water from the return pipe can be used. Another possibility is the adding of the surfactant solution from a tanker.
4. The Simulation Program

Chapter 3 shows that the basis for the simulation - the significant influences due to drag reducing surfactants - are known. The most significant influences are the hydraulic behaviour in pipes (pressure loss) and the heat transfer reduction in heat exchangers. Therefore the calculation is considering especially these components.

Of primary interest for the simulation is the steady-state of the system because in this case the most significant changes in behaviour are expected. The dynamic behaviour will not be influenced significantly. The highest priority has the calculation of the supply guarantee. Therefore the modelling of heat exchanger plays a decisive role.

A further aim of the program is the testing of modifications that have been projected to compensate the negative influences due to the application of drag reducing additives. Comparing the origin state with the modified system the efficiency of measures can be assessed.

In the following chapter the simulation program TenSim and its single moduls are described. In chapter 4 the closest attention is paid to the mathematical description. How to work with the simulation program TenSim is described in an extra handbook.

4.1 Description of the program

The simulation program has to fulfill the following requirements:

- thermo-hydraulic simulation of the steady-state of transport systems,
- the simulation of single components of D.H. networks; especially the possibility to simulate heat exchangers and to test modifications,
- the possibility to change network configuration in order to simulate modifications of the system,
- the possibility to simulate different control devices because the application of drag reducing additives can require a new control strategy,
- a sufficient presentation of the results of total systems and single components
  and
- the possibility of a graphic presentation for the most important parameters as pressure, temperature etc. as basis for the comparison of different operating modes (e.g. water - surfactant solution or different variations operating with surfactant solutions).

The realization of the above mentioned requirements can be described with figure 4.1 [5]. The first step is the making of physical-mathematical models to describe the single components of district heating systems. Concerning the application of additives models for the following system elements have to be built:

- heat exchanger,
  - plate heat exchanger,
The Simulation Program

- shell and tube heat exchanger,
- helical tube heat exchanger,
- centrifugal pumps and
- pipes and fittings.

In a second step the models of the single components have to be connected to a network. Therefore the graph theory is applied [7].

**Fig. 4.1:** Components of the simulation program

Further important models are controllers. They determine the steady-state. The following control devices are implemented:

- pressure maintenance,
- operating conditions of the pumps (pressure difference, node pressure, height of the pump, speed),
- consumer stations (minimum supply temperature) and
- heat producer (constant supply temperature, constant heat load).
The reference values are fixed manually. For consumer stations and heat producer the reference values can be received from sufficient models (if available). The following models can be used:

- heating curves for controlling the producer (supply temperatures)
- load models for the consumers.

The last important kind of model is the mathematical description of the behaviour of drag reducing surfactants. Therefore the mathematical equations for the calculation of the hydraulic behaviour as well as those for the calculation of the heat exchange have to be implemented into the program.

The realization of the simulation program TenSim has been carried out object oriented in turbo pascal for windows. Due to this kind of programming, the design of the program is very clear and a strictly modular realizable.

The operation under windows makes it possible to use the typical menues and keyboard operations. The surface, which includes a display of a flow sheet permits a simple access to single elements and the total system. Any operating conditions of elements or of the total system can be received or presented.

### 4.1.1 The model „District Heating Network“

The modeling of network structures is a problem whose solution can be lead back to Kirchhoff. For electric networks the graph theory is the mathematical basic [7].

A nondirectional graph is composed of a certain number of nodes and branches (see figure 4.2). A branch is a connection between two nodes. In relation to a district heating system the system elements (pipes, heat exchangers, pumps etc) represent the branches and the connections between the elements are the nodes. The degree of a node is the number of branches which are connected with it.

A path is a part of the graph which connects two nodes, where every node and every branch is different. If the two nodes are identical a closed path is received which is named loop. The graph of a district heating system is continuous, because there is a path between every pair of nodes.

![Continuous graph and frame](image-url)
Concerning the mathematical calculation of a graph the "tree" and the "frame" are of great importance. A tree is a continous graph without loops. If the tree is including all nodes of a graph it is described as a frame. Tree and frame are characterized by the fact that there is only one path which connects two nodes. Therefore every branch of the graph that is not a part of the frame is forming loops.

Figure 4.2 shows a continous graph with 10 nodes and 15 branches. A possible frame is characterized through bold marked lines. It is originated from cancelling 6 branches. When adding one more branch a new node is also added to the frame. Therefore the number of nodes $N$ of a frame is by 1 higher that the number of the branches $K_G$ that build the frame:

$$N = K_G + 1.$$  

eq. (4.1)

The adding of a branch into a frame which connects two existing nodes is building a loop. If the number of nodes $N$ and branches $K$ is known the number of loops of a continous graph can be calculated as follows:

$$M = (K + 1) - N.$$  

eq. (4.2)

Algorithm for determining the frame

For the simulation program TenSim the BFS-method (Breadth-First-Search) has been applied. Characteristic of the BFS-method is that a frame is determined which has a short path between two nodes concerning the number of branches.

Beginning with an arbitrary node $K_{\text{Start}}$ which is marked all adjacent nodes are determined (These are all nodes of the branches which are connected with $K_{\text{Start}}$). If an adjacent node $K_n$ has not been checked (is not marked), the connection between $K_{\text{Start}}$ and $K_n$ is added to the frame. After that, all adjacent nodes $K_n$ which are not marked are the starting nodes and are written in a list of starting nodes. The process is finished, if all nodes of the graph have been checked (then all nodes are marked to be checked and no more node is on the list). The flow chart of the BFS-algorithm is shown in the appendix.

The algorithm for determining a tree is the same as for determining the frame in principle. To get trees of special qualities, additional conditions have to be set to determine the branches. An important secondary condition e.g. is the search in direction of the flow. If the search over heat consumers is prohibited, the existing tree is separating the district heating system in supply and return. The determined trees are the basic for the presentation of diagramms (pressure, temperature etc.).

All network calculations use Kirchhoff’s laws as basic (voltage law and current law). The first law - the current law - means that the sum of all mass flow rates in a node is zero:

$$\sum M = 0.$$  

eq. (4.3)

The second law - the voltage law - means, that the sum of the pressure losses in a closed path (loop) is zero:

$$\sum \Delta p = 0.$$  

eq. (4.4)
Figure 4.3 shows a simplified network with 4 nodes and 5 branches. The number of loops is \( M = (K + 1) - N = (5 + 1) - 4 = 2 \). If despite of fulfilling equation 4.3 the voltage law is not fulfilled (e.g. the sum of pressure losses in loop A is not zero) a loop correction has to be carried out. Therefore the mass flow rates of a loop \( \dot{M} \) have to be corrected with \( \Delta \dot{M}_A \) and new pressure losses in the branches have to be calculated.

\[
\begin{align*}
\dot{M}_{1,\text{new}} &= \dot{M}_{1,\text{old}} + \Delta \dot{M}_A \\
\dot{M}_{2,\text{new}} &= \dot{M}_{2,\text{old}} - \Delta \dot{M}_A \\
\dot{M}_{3,\text{new}} &= \dot{M}_{3,\text{old}} + \Delta \dot{M}_A 
\end{align*}
\]

**Fig. 4.3:** Kirchhoff's laws and loop correction

The simplified network in figure 4.3 shows that the adjacent loop is also influenced. Therefore the calculation of a large system with a large number of loops is not easily to verify.

Transport systems that are considered for the application of drag reducing additives normally have no or only a few loops. Therefore, the application of the BFS-algorithm is sufficient. The implementation of other algorithm for determining frames and trees into the program is possible without great expenditure.

### 4.1.2 The model „Pipe“

**Thermodynamic model**

The course of temperature inside a pipe can be calculated by the use of a differential element of a pipe [13]. The differential element is shown in figure 4.4.

**Fig. 4.4:** Energy balance for differential pipe element
Assuming that the specific heat capacity $c_p$ is constant the enthalpy flow rates can be calculated as follows:

\[ \dot{Q}_{in} = M \cdot h_{in} = M \cdot c_p \cdot T_{in}, \quad \text{eq. (4.5)} \]

\[ \dot{Q}_{out} = M \cdot c_p \cdot T_{out}, \quad \text{eq. (4.6)} \]

\[ d\dot{Q}_V = k \cdot \pi \cdot D \cdot dx \cdot (T_m - T_u) = k \cdot dA \cdot \Delta T, \quad \text{eq. (4.7)} \]

\[ d\dot{Q}_R = V \cdot dp = \frac{\pi \cdot D^2}{4} \cdot w \cdot \xi \cdot \frac{\rho}{2} \cdot w^2 \cdot \frac{dx}{D}. \quad \text{eq. (4.8)} \]

With equation 4.5 to 4.8 the following differential equation can be derived:

\[ \frac{w \cdot \rho \cdot c_p}{4k} \cdot \frac{T_m - T_u - \xi \cdot \frac{\rho}{2} \cdot w^3 \cdot \frac{1}{4k}} \cdot dT_m = \frac{1}{D} \cdot dx. \quad \text{eq. (4.9)} \]

With the boundary conditions:

\[ x = 0, \ T_m(x=0) = T_{in} \ 	ext{and} \]

\[ x = L, \ T_m(x=L) = T_{out} \]

the equation can be solved and transformed into the integral form:

\[ T_{rel.} \int_{T_{in}}^{T_{out}} \frac{w \cdot \rho \cdot c_p}{4k} \cdot \frac{T_m - T_u - \xi \cdot \frac{\rho}{2} \cdot w^3 \cdot \frac{1}{4k}} \cdot dT_m = \frac{L}{D}. \quad \text{eq. (4.10)} \]

with the constants $\alpha$ and $\beta$:

\[ \alpha = \frac{w \cdot \rho \cdot c_p}{4k}, \ \beta = \xi \cdot \frac{\rho}{2} \cdot w^3 \cdot \frac{1}{4k} \ \text{(friction term)}, \]

The outlet temperature can be calculated according to equation 4.11:

\[ T_{out} = (T_u + \beta) + [T_{in} - (T_u + \beta)] \cdot \exp\left(\frac{1}{\alpha} \cdot \frac{L}{D}\right). \quad \text{eq. (4.11)} \]

For practical calculations the friction term $\beta$ can be neglected. Despite of increasing expenditure no significant increase in accuracy can be expected, if $\beta$ is taken into consideration. If the friction term is neglected ($\beta = 0$) the result is:

\[ T_{out} = T_u + (T_{in} - T_u) \cdot \exp\left(\frac{1}{\alpha} \cdot \frac{L}{D}\right). \quad \text{eq. (4.12)} \]
Equation 4.12 in a transformed form is applied in TenSim to calculate the course of temperature. In the end, the following equation has been implemented into the program to describe the thermodynamic behaviour (the loss in temperature on the length):

$$\Delta T = T_{\text{out}} - T_{\text{in}} = (T_{\text{in}} - T_{\text{o}}) \cdot \left( \exp\left( -\frac{k \cdot \pi \cdot D \cdot L}{c_p \cdot M} \right) - 1 \right).$$  

**eq. (4.13)**

**Hydraulic model**

The course of pressure inside a pipe as a function of the mass flow rate, the geometry of the pipe, the friction and other resistances can be described as follows [20]:

$$\Delta p = M \cdot |M| \cdot r - \rho \cdot g \cdot \Delta H.$$  

**eq. (4.14)**

The pressure loss is composed of the dynamic part, described by the product of the pipe-constant $r$ and the square of the mass flow rate and a static part, given by the difference of the height. The constant $r$ is defined as follows:

$$r = \frac{8}{\pi^2} \cdot \frac{L}{\rho \cdot D^4} \cdot \left[ \frac{L}{d} \cdot \xi + \Sigma \xi_{\text{add}} \right].$$  

**eq. (4.15)**

It describes the pressure loss of the straight part of a pipe and that one caused by additional resistances as e.g. fittings, knees etc. The values of the additional resistances can be gathered from different tables [14][36]. On the other hand, additional resistances can be considered by using an additional length.

For laminar flow ($Re < 2.300$) the law of Hagen-Poiseuille is used [35]:

$$\xi = \frac{64}{Re}.$$  

**eq. (4.16)**

In the turbulent range different equations to calculate the drag coefficient can be used [43]. TenSim uses the equation of Colebrook-White:

$$\frac{1}{\xi} = -2 \cdot \log \left[ \frac{2.51}{Re \cdot \sqrt{\xi}} + \frac{k}{3.71 \cdot D} \right].$$  

**eq. (4.17)**

Equation 4.17 can only be solved iterative. Due to the fast convergence no problems occur. After 4 loops, the mistake is below 1.2 %.

For the calculation, the following assumption are made:

- the fluid is incompressible,
- the calculation is carried out as a function of temperature; therefore a change in density and viscosity is taken into consideration,
- the energy change due to the change in volume is not considered; an average temperature between entrance and outlet is used.
In the course of the calculation the mass flow rate is changing. Due to this behaviour a new outlet (and average) temperature has to be calculated before calculating the pressure drop. Deviating from this TenSim uses the last calculated outlet temperature because the influence of the change in temperature is small. In the cause of the iteration the change in temperature gets smaller until the steady state is reached.

4.1.3 The model „Centrifugal Pump“

In district heating systems centrifugal pumps with speed control are applied. Analog to the model for pipes the hydraulic and thermodynamic model are described. For the simulation the thermodynamic model is only of secondary importance.

Hydraulic model

Determining of the characteristic curves

Centrifugal pumps are characterized by the following characteristic parameters:

- height,
- efficiency,
- pumping capacity and
- NPSH-value (Net Positive Suction Head).

The characteristic curves results from the representation of the above mentioned parameters in a diagramm versus the flow rate. It is the basis of the mathematical model for centrifugal pumps.

For a certain flow rate the height of a pump is controlled by the speed. In the range of the reference speed \( n_0 \) the following equation is valid [32] (affinity law):

\[
\frac{\dot{M}}{\dot{M}_0} = \frac{n}{n_0} \quad \text{with} \quad \Delta p = \text{const. and eq. (4.18)}
\]

\[
\frac{\Delta p}{\Delta p_0} = \left( \frac{n}{n_0} \right)^2 \quad \text{with} \quad \dot{M} = \text{const. eq. (4.19)}
\]

Therefore the conversion of the characteristic from the reference speed \( n_0 \) on the speed \( n \) is possible. The characteristic can be calculated in general with the following equation:

\[
\Delta p = a_0(n) + a_1(n) \cdot \dot{M} + a_2(n) \cdot \dot{M}^2. \quad \text{eq. (4.20)}
\]

For the reference speed \( n_0 \) the characteristic is described as follows:

\[
\Delta p_0 = a_0(n_0) + a_1(n_0) \cdot \dot{M}_0 + a_2(n_0) \cdot \dot{M}_0^2. \quad \text{eq. (4.21)}
\]
If the coefficients $a_0(n_0), a_1(n_0), a_2(n_0)$ are known the height of a pump can be calculated by using equations 4.18, 4.19 and 4.21:

\[
\Delta p \cdot \left( \frac{n}{n_0} \right)^2 = a_0(n_0) + a_1(n_0) \cdot \frac{n}{n_0} \cdot \dot{M} + a_2(n_0) \cdot \left( \frac{n}{n_0} \right)^2 \cdot \dot{M}^2 \quad \text{or} \quad \text{eq. (4.22)}
\]

\[
\Delta p = a_0(n_0) \cdot \left( \frac{n}{n_0} \right)^2 + a_1(n_0) \cdot \frac{n}{n_0} \cdot \dot{M} + a_2(n_0) \cdot \dot{M}^2. \quad \text{eq. (4.23)}
\]

Comparing the coefficients of equations 4.23 and 4.20, the result is:

\[
\Delta p = a_0(n_0) \cdot \left( \frac{n}{n_0} \right)^2 + a_1(n_0) \cdot \frac{n}{n_0} \cdot \dot{M} + a_2(n_0) \cdot \dot{M}^2
\]

and the coefficients for the characteristic at the speed $n$ are received:

\[
a_0(n) = a_0(n_0) \cdot \left( \frac{n}{n_0} \right)^2, \quad a_1(n) = a_1(n_0) \cdot \frac{n}{n_0} \quad \text{and} \quad a_2(n) = a_2(n_0).
\quad \text{eq. (4.24)}
\]

Therefore, to determine the coefficients $a_0(n_0), a_1(n_0), a_2(n_0)$, three values $\Delta p, \dot{M}$ have to be known to adjust the coefficients with a regression analysis. The solution of the resulting system of equations can be carried out by using the Gauß-algorithm.

**Linear regression of the characteristic**

The linear regression of the characteristic curve of the pump is carried out with the method of the smallest error squares. The square of the differences between the calculated values $\hat{y}$ and the measured values $y$ has to be minimized:

\[
\nabla \left[ \sum (y_i - \hat{y}_i)^2 \right] = \vec{0}.
\quad \text{eq. (4.25)}
\]

With the general equation for the characteristic $\hat{y} = a + b \cdot x + c \cdot x^2$, the sum of the error squares which has to be minimized is:

\[
\sum (y_i - \hat{y}_i)^2 = \sum ((a + b \cdot x_i + c \cdot x_i^2 - y_i)^2).
\quad \text{eq. (4.26)}
\]

Forming the derivation in respect to $a, b$ and $c$ the following system is received:

\[
\begin{bmatrix}
2 \cdot a \cdot \Sigma x_i^0 + 2 \cdot b \cdot \Sigma x_i + 2 \cdot c \cdot \Sigma x_i^2 - 2 \cdot \Sigma y_i \\
2 \cdot a \cdot \Sigma x_i^1 + 2 \cdot b \cdot \Sigma x_i^2 + 2 \cdot c \cdot \Sigma x_i^3 - 2 \cdot \Sigma (x_i \cdot y_i) \\
2 \cdot a \cdot \Sigma x_i^2 + 2 \cdot b \cdot \Sigma x_i^3 + 2 \cdot c \cdot \Sigma x_i^4 - 2 \cdot \Sigma (x_i^2 \cdot y_i)
\end{bmatrix}
= \vec{0}
\quad \text{or} \quad \text{eq. (4.27)}
\]
The solution can be carried out with the Gauß-algorithm or other methods for linear systems of equations.

**Calculation of the speed**

In the course of the network calculation it is necessary to determine the speed of the circulation pumps. The flow rate and height are pre-set values (regulation variables). The speed is resulting from transforming equation 4.23.

\[
\begin{align*}
n &= -\left( \frac{a_1}{a_0} \cdot \frac{n_0}{2} \cdot M \right) + \sqrt{\left( \frac{a_1}{a_0} \cdot \frac{n_0}{2} \cdot M \right)^2 - \left( \frac{a_2 \cdot M^2 - \Delta p}{a_0} \cdot n_0^2 \right)}.
\end{align*}
\]

**Thermodynamic model**

This model is based on the assumption that the total dissipated energy of the pump \( P_V \) is converted into energy \( \dot{Q}_{add} \) which is heating the heat transfer medium:

\[
P_V = \dot{Q}_{add}.
\]

With the dissipated energy \( P_V = (1 - \eta) \cdot P_{add} \), the pumping capacity \( P_p = \eta \cdot P_{add} \), the enthalpy flow rate difference \( \dot{Q}_{add} = M \cdot c_p \cdot \Delta T \) and the equation for the pumping capacity \( P_p = \Delta p \cdot \dot{V} \) it is possible to calculate the increase of temperature due to the dissipated energy:

\[
\Delta T = \frac{1 - \eta}{\eta} \cdot \frac{\Delta p}{\rho \cdot c_p}.
\]

For an example with the following data:

- \( \Delta p = 16 \text{ bar} = 1.6 \text{ MPa} \),
- \( \rho = 975 \text{ kg/m}^3 \),
- \( c_p = 4,180 \text{ J/(kg K)} \) and
- \( \eta = 0.75 \),

the increase of temperature is \( \Delta T = 0.131 \text{ K} \). Assuming a flow rate of 1,000 m\(^3\)/h the dissipated energy flow rate of the pump is:

\[\dot{Q}_{add} = \dot{V} \cdot \rho \cdot c_p \cdot \Delta T = 0.148 \text{ MW}.\]
Heat exchanger models

The ratio of the dissipated energy to the transported heat is decisive for the accuracy of the simulation. With the assumption of a temperature difference between supply and return of 40 °C the ratio is 270. Therefore the dissipated energy has no significant influence on the calculation.

In figure 4.5 the dialog of the model „centrifugal pump“ is presented. The pump gets a name e.g. P1 and allocated to a group of pumps. The operating range is determined by the input of a minimum and a maximum speed at the corresponding mass flow rate. Furthermore the reference speed \( n_0 \) and the nominal mass flow rate \( M_0 \) have to be given as reference values for the calculation of the characteristic of the pump.

A data record can be saved on a data medium and also be loaded from it. To verify the data records they can be presented in the form of a diagram. With the switch „Connect“ the pump can be allocated to a node or the allocation can be checked.

The model „centrifugal pump“ is a tool with which any centrifugal pumps can be simulated within the complete program TenSim.

### 4.1.4 Heat exchanger models

The heat flow rate in a heat exchanger can be calculated with equation 3.1, \( \dot{Q} = k \cdot A \cdot \Delta T_m \).

In this equation the assumption is used that the average overall heat transfer coefficient \( k \) is constant. The average temperature difference \( \Delta T_m \) is defined as follows:

\[
\Delta T_m = \frac{1}{A} \int_A (\Delta T_1^+ - \Delta T_2^+) \, dA.
\]

\( \Delta T^+ \) are hypothetic temperature differences between the two fluids, \( A \) is the heat transfer area.
For the steady state equation 4.33 can be used if heat losses and kinetic and potential energy differences are neglected:

\[ \dot{Q} = M_1 \cdot (h_{1\text{in}} - h_{1\text{out}}) = M_2 \cdot (h_{2\text{out}} - h_{2\text{in}}). \]  

eq. (4.33)

If physical properties are assumed as constant (negligible influence of temperature and pressure) equation 4.33 can be transformed to:

\[ \dot{Q} = \dot{W}_1 \cdot (T_{1\text{in}} - T_{1\text{out}}) = \dot{W}_2 \cdot (T_{2\text{out}} - T_{2\text{in}}), \]  

eq. (4.34)

with the heat capacity flow rate \( \dot{W}_j = M_j \cdot c_p \).

The physical properties are determined for the average temperature between entrance- and outlet temperature. With equation 4.34 and 3.1 and division by the largest temperature difference \( (T_{1\text{ein}} - T_{2\text{ein}}) \) dimensionless basic numbers can be defined. These are used for the simulations [38]:

non-dimensional average temperature difference \( \Theta \):

\[ \Theta = \frac{\Delta T_m}{(T_{1\text{in}} - T_{2\text{in}})}, \]  

with \( 0 \leq \Theta \leq 1 \), eq. (4.35)

non-dimensional average change of mass flow temperature \( \Psi \):

\[ \Psi_1 = \frac{T_{1\text{in}} - T_{1\text{out}}}{T_{1\text{in}} - T_{2\text{in}}}, \Psi_2 = \frac{T_{2\text{out}} - T_{2\text{in}}}{T_{1\text{in}} - T_{2\text{in}}}, \]  

with \( 0 \leq \Psi \leq 1 \), eq. (4.36)

number of transfer units NTU:

\[ \text{NTU}_1 = \frac{k \cdot A}{\dot{W}_1}, \text{NTU}_2 = \frac{k \cdot A}{\dot{W}_2}, \]  

with \( 0 \leq \text{NTU} \leq \infty \) and eq. (4.37)

ratio of heat capacity flow rate R:

\[ R_1 = \frac{\dot{W}_1}{\dot{W}_2} = \frac{1}{R_2}, \]  

with \( 0 \leq R \leq \infty \). eq. (4.38)

The following relation between the non-dimensional numbers can be stated:

\[ \frac{\Psi_1}{\Psi_2} = \frac{\text{NTU}_1}{\text{NTU}_2} = \frac{1}{R_1} = R_2 \] and eq. (4.39)

\[ \Theta = \frac{\Psi_1}{\text{NTU}_1} = \frac{\Psi_2}{\text{NTU}_2} \]. eq. (4.40)
Model „Shell and Tube Heat Exchanger“ - cell method

The calculation of water/water operating shell and tube heat exchanger is carried out with the cell model [40]. By using the cell model overlappings of temperatures which can occur in multi-pass apparatus can be considered [11][12]. In the case of overlapping the direction of the heat flow rate is changing. This leads to a reduced efficiency. Taking the increase of passes for a compensation of heat output reduction into consideration (see chapter 3) the overlapping of temperatures can be of importance.

To calculate the influence of geometry on the heat transfer capacity of shell and tube heat exchangers the apparatus is divided into several cells (see figure 4.6). Considering the arrangement of the connection pieces (4 different possibilities) the flow direction and the number of passes (odd/even-numbered) 16 different kinds of structural shapes are possible [33].

For a single cell the following assumptions are made due to the geometry of shell- and tube side and their effect on the flow:

- the flow on the shell side is totally mixed (ideal vessel) and
- the flow on the pipe side is not mixed.

![Cell method for shell and tube apparatus](image)

**Fig. 4.6:** Cell method for shell and tube apparatus (2 passes, 6 cross baffles) [11]

![Single cell of a heat-exchanger](image)

**Fig. 4.7:** Single cell of a heat-exchanger
The apparatus is divided into \((N + 1) \cdot (M + 1)\) cells \((N = \text{number of passes on the tube side, } M = \text{number of passes on the shell side})\) which are linked with their temperatures. In figure 4.7 a single cell is presented.

The relation between the single cells is as follows:

\[
T_{2\text{out},j} = T_{2\text{in},k} \quad \text{and} \quad \text{eq. (4.41)}
\]

\[
T_{1\text{in},j} = T_{1\text{out},j-1} \quad \text{eq. (4.42)}
\]

The cell \(j\) is in direction of the flow of \(W_j\) and the cell \(k\) in direction of \(W_2\).

The outlet temperatures of the cells can be calculated with a heat balance and the definitions of the non-dimensional numbers:

\[
T_{1\text{out}} = (1 - \Psi_1) \cdot T_{1\text{in}} + \Psi_1 \cdot T_{2\text{in}} \quad \text{eq. (4.43)}
\]

\[
T_{2\text{out}} = (1 - \Psi_2) \cdot T_{2\text{in}} + \Psi_2 \cdot T_{1\text{in}} \quad \text{eq. (4.44)}
\]

The linking all cells to the total apparatus leads to a system of equations that has to be solved iteratively. Therefore the outlet temperatures are calculated in direction of the flow. At the beginning of the calculation the inlet temperatures are the start values for the outlet temperatures (cell efficiency = 0).

The course of the calculation is shown in the appendix. First step is fixing the start values (temperatures and pressures). After this heat transfer coefficients and pressure losses are calculated. Next step is the calculation of overall heat transfer coefficients and NTU-values of the total heat exchanger. It is assumed that the NTU for all cells is identical and that it can be calculated as follows:

\[
\text{NTU}_{1,z} = \frac{\text{NTU}_{1,\text{tot}}}{\text{number of cells}} \quad \text{and} \quad \text{eq. (4.45)}
\]

\[
\text{NTU}_{2,z} = \frac{\text{NTU}_{2,\text{ges}}}{\text{number of cells}} \quad \text{eq. (4.46)}
\]

Mean wall temperatures can be determined with equation 4.47:

\[
\Delta T_{k,w} = \frac{k \cdot A}{\alpha_i \cdot A_i} \cdot \Delta T_m \quad \text{eq. (4.47)}
\]

Therefore the following assumptions have been made:

- the average total heat transfer coefficient \(k\) is the same for every cell as for the total apparatus and
- the physical properties can be determined for an average temperature between entrance- and outlet temperature. Therefore in every cell the physical properties are identical.
The calculation of heat transfer coefficients for water can be carried out according to literature [12][38]. The coefficients for surfactant solution can be calculated according to the equations given in chapter 3. TenSim uses the equation of Gnielinski for the calculation of heat transfer coefficients in straight pipes for pure water [38].

The total pressure loss inside the pipes can be calculated by summing up the different parts of pressure losses. These are:

- friction inside the pipe,
- the pressure loss due to a change in the cross section of the flow,
- deviations of the flow and
- the geodetic pressure loss.

The pressure loss due to friction can be calculated with the equations of chapter 4.1.2. Pressure losses due to a change in the cross section occur in the connection pieces and the entrance and outlet of the pipes. In multi-pass apparatus an additional pressure drop is resulting, due to the deviation. Figure 4.8 shows the zones in which the pressure drops occur.

Figure 4.8: Division of a heat exchanger in different zones [38]

The calculation of the particular pressure drops can be carried out with the following equations [39]:

\[
\Delta p_{\text{St, in}} = \xi_{\text{in}} \cdot \frac{\rho}{2} \cdot w^2, \text{ with } \xi_{\text{in}} = 1. \quad \text{eq. (4.48)}
\]

This is nearly the complete dynamic pressure.

\[
\Delta p_{\text{R, in}} = \xi_{\text{R, in}} \cdot n_{\text{R}} \cdot \frac{\rho}{2} \cdot w^2, \text{ with } \xi_{\text{R, in}} = 0.25 - 0.5. \quad \text{eq. (4.49)}
\]

Therefore, an edged pipe entrance is assumed. In other cases the drag coefficients can differ (see [40]).

\[
\Delta p_U = \xi_U \cdot (Z - 1) \cdot \frac{\rho}{2} \cdot w^2, \text{ with } \xi_U = 2.5. \quad \text{eq. (4.50)}
\]
The simulation program \(^{PR} \text{out} = \xi_{R, \text{out}} \cdot \frac{\rho}{2} \cdot w^2, \text{ with } \xi_{R, \text{out}} = 1 \) and \( \Delta p_{R, \text{out}} = \Delta p_{\text{out}} = \xi_{\text{out}} \cdot \frac{\rho}{2} \cdot w^2, \text{ with } \xi_{\text{out}} = 0.25 - 0.5. \) eq. (4.52)

Der geodetic pressure drop can be calculated as follows:

\[ \Delta p_{\text{geo}} = \rho \cdot g \cdot (h_{\text{in}} - h_{\text{out}}). \] eq. (4.53)

By operating with drag reducing additives a reduction of the pressure drop due to friction of 50% is assumed [41]. Other pressure drops are calculated with the relations for pure water.

For the calculation of the pressure drop inside the shell the equations for tube bundles which are flown transverse are used. By using coefficients of correction non ideal states can be taken into consideration. Analog to the tube side the geodetic pressure loss is added. For the exact calculation it is referred to literature [40].

Concerning the dialog it is referred to figure 4.10 which shows the dialog for plate heat exchanger.

**Model „Helical Tube Heat Exchanger“**

In chapter 3 the influence of surfactant solutions on helical tube heat exchanger is presented. Equations for the calculation of the heat transfer inside helical tubes with drag reducing additives are given. The calculation of heat transfer in helical tubes requires some geometrical values which have to be calculated and which can not depicted directly from construction drawings.

**Figure 4.9** shows the most important geometric quantities.

![Fig. 4.9: Important geometric quantities for helical tube heat exchangers](image)

The calculation requires a lot of assumptions and simplifications. It is assumed that the pipe-rows are equidistant. The distance \( \Delta x \) of the rows can determined with equation 4.54.
Other important quantities are the mean diameter of the helical tubes $D_w$, the number of windings $n_{\text{win}}$, the gradient of the helical tube bundle $s_h$ and its average diameter of curvature $D_{mh}$.

$$\Delta x = \frac{D_s - D_{\text{disp}}}{2 \cdot Z_{RR} \cdot \pi}, \quad \text{eq. (4.54)}$$

$$D_h = 2 \cdot \left( \frac{D_h}{2} - Z_{RR} \cdot \Delta x \right), \quad \text{eq. (4.55)}$$

$$n_{\text{win}} = \sqrt{\frac{(L_R - L_{nh})^2 - H_h^2}{D_h \cdot \pi^2}}, \quad \text{eq. (4.56)}$$

$$s_h = \frac{H_h}{n_{\text{win}}} \quad \text{and} \quad \text{eq. (4.57)}$$

$$D_{mh} = D_h \cdot \left( 1 + \left( \frac{s_h}{\pi \cdot D_h} \right)^2 \right), \quad \text{eq. (4.58)}$$

The transverse and longitudinal pitch of the bundle $s_t$ and $s_l$ are determined as a function of the arrangement of the pipes (in normal cases the arrangement is staggered). The following equations are resulting:

staggered arrangement: $s_t = \frac{1}{2} \cdot \frac{s_h}{n_R}, \quad s_l = 2 \cdot \Delta x \quad \text{and} \quad \text{eq. (4.59)}$

arrangement in alignment: $s_t = \frac{s_h}{n_R}, \quad s_l = \Delta x. \quad \text{eq. (4.60)}$

The gradient of the pipes which leads to a transverse flow is neglected. A significant influence occurs at gradients above 40°. In this case a reduction of the heat transfer of more than 5% occurs [41]. In helical tube heat exchangers which are used in district heating systems, a gradient of 12° is typical.

The simulation program is operating with the above described cell method to calculate helical tube heat exchangers. In addition the geometric values have to be determined. For every adjacent pipe row of helical tubes the pressure loss and the heat transfer coefficients are calculated and an average value is determined.

Compared with the shell and tube model the same assumptions are made. The apparatus is for instance divided into three cells. This requires two intermediate values of temperatures. At the beginning of the calculation the temperatures and pressures for every cell are estimated. The wall temperatures are defined as average values within the cells. After the calculation of the heat transfer coefficients and pressure drops the overall heat transfer coefficients, the NTU values and the efficiency of the cells are determined for every single cell.
If the outlet temperatures and wall temperatures of the cells are below a limiting value (error limit) the heat load (sum of the particular heat loads of every cell) can be determined. The exact course of calculation can be seen in the appendix.

**Model „Shell and Tube Condenser“**

Shell and tube condensers are applied in district heating systems for heat extraction from the producer into the transport systems. Concerning the application of drag reducing additives the influence due to the reducing of heat transmission is of importance to be able to make statements concerning the supply guarantee.

The most important value that has to be calculated is the outlet temperature. With the outlet temperature the transmitted heat load can be calculated, too. In addition a simulation of the pressure drop inside the pipes is carried out.

Basing on the known entrance values on the shell and on the tube side the parameters that have to be calculated are estimated in a first step. Afterwards the results are calculated iteratively. The exact course of calculation can be seen in the appendix.

With this module within the total simulation program modifications like the increase of saturation temperature of the steam etc. can be investigated. Furthermore the installation of obstacles can be calculated if equations are available which can be implemented into the program.

Due to the physical conditions in condensers some assumptions have to be made. The calculation is carried out without the consideration of a possible shear stress of the condensing steam. In most cases this simplification leads to no significant influences. On the other hand the flooding of the pipe section and therefore the cooling of the condensate is not considered.

Due to the flow of the condensate from pipe row to pipe row a certain turbulence which increases the heat transfer occurs. This effect is not considered, too.

Despite of this simplifications calculations of real condensers showed a good correspondence to real operation. The existing errors are below 2 % (real operating point for nominal heat load).

**Model „Plate Heat Exchanger“**

Due to the advantages mentioned in chapter 3 plate heat exchanger are favouredly installed in transmission stations. Usually they are operating in reverse flow and are designed in U-shape.

The thermodynamic calculation can be carried out with the cell method. In this case the number of cells is 1. Due to the effect that the first and the last plate have only the half heat exchange area compared to the other plates a non ideal behaviour results. This behaviour leads to a decrease in heat load. The effect is decreasing with increasing number of plates [26]. Above a certain number of plates the effect can be neglected. Usually apparatus which are installed in district heating system have a sufficient number of plates so the effect can be neglected.

To calculate the heat transfer Nusselt characteristics are used which are specific for every apparatus. The application of drag reducing additives is simulated according to chapter 3.1.3. The Nusselt numbers for surfactant solution are calculated by using the correlation for water and a correction term f_r [22]. The particular parameters to build the correlations can be implemented into the program by using the dialog shown in figure 4.10.
To simplify the calculation the assumption is made that the flow is distributed equally over the channels between the plates. Due to the pressure loss in the inlet channel the flow rate is decreasing with increasing number of plates (or channels). Usually this effect is not significant in typical plate heat exchangers which are used in district heating systems.

Analog to calculation the hydraulic simulation requires experimental investigations, too. Exact equations to calculate the pressure drop as a function of all significant parameters do not exist up to now.

For the usual apparatus the following characteristic - Euler as a function of Reynolds - can be used to determine the pressure loss characteristic:

\[
\text{Eu} = C_1 \cdot \text{Re}^{Z_1}.
\]

The Euler number is defined as follows:

\[
\text{Eu} = \frac{\Delta p}{\rho \cdot w_c^2}.
\]

The coefficients \(C_1\) and \(Z_1\) have to be determined experimentally. Due to the strong change of the behaviour of the flow the correlations can not be used for surfactant solutions. Therefore additional experiments have to be carried out to determine the coefficients for additive solution.

![Model - Plate Heat Exchanger](image)

**Fig. 4.10:** Dialog for the model „plate heat exchanger“

Analog to the correction factors for the Nusselt characteristics the coefficients for determining the hydraulic behaviour can be implemented into the program over the dialog (see figure 4.10). The structure of the dialog is analog to that one for pumps. It is described exactly in chapter 5. All necessary data can be implemented, changed, saved and loaded. Furthermore the model can be used to simulate single apparatus without being forced to calculate the complete district heating system.
4.1.5 The model „Control“

The most important parameters which have to be controlled in district heating systems are pressure (or pressure difference) and temperature. In the following chapters the implemented controllers are described.

Pressure maintenance

In some transport systems an average pressure (control point) between two reference nodes is controlled. In a first step the pressure loss over any path from the reference node \( p_{K1} \) to reference node \( p_{K2} \) is determined. The pressure of the reference node \( p_{K1} \), which is the starting point for the calculation of the pressure of all nodes, results from the sum of the control point \( \Delta p_{cp} \) and half of the pressure loss of any path \( (ap) \) to the other reference node \( p_{K2} \):

\[
p_{K1} = \Delta p_{cp} + \frac{\Delta p_{ap}}{2}.
\]

eq. (4.63)

After the calculation of the network the laws of Kirchhoff are fulfilled. However, the absolute pressure level has not been fixed.

The absolute pressure maintenance can be realised by fixing the control point for a certain node and implementing it over a dialog into the program. The pressure for this node is fixed, all other pressures can be calculated dependent on this value.

Temperature control of the consumer stations

For the consumer stations a definite control point for the temperature \( T_{sec, sup} \) is required. The control point is - in most cases - a function of the outdoor temperature and is given by heating curves (see figure 4.12).

If the heat demand is known, which can be taken out of load models, (see chapter 4.1.6) or if the mass flow rate is given all parameters to describe the side of the consumer stations completely are given. In figure 4.11 the scheme of a typical temperature control of consumer stations is shown.

![Fig. 4.11: Scheme of a temperature control of a consumer station](image-url)
If the secondary supply temperature decreases below the control point the primary mass flow rate is increased by opening the control valve. If in the other way the temperature is too high the valve is closed and therefore the heat transmission capacity is reduced.

**Temperature control of suppliers**

The heat producer considered in the simulation program are analog to figure 4.11 hydraulic separated from the transport system. For district heating networks the following two operating modes are usual:

- Control of the supply temperature of the transport system:
  The heat producer is heating up the heat carrier for a given mass flow rate and a given return temperature to the primary supply temperature $T_{\text{prim, sup}}$. The control point is reached by controlling the mass flow rate or the parameters of the steam (in condensers).

- If more than one heat producer are supplying the complete system usually all except one operate with constant heat load. The variant operating producer is controlling the heat load and the supply temperature.

Heat curves or manual instructions can be used to fix the control points and forwarding them to the heat exchanger modules. With that the calculation for the appropriate operating conditions can be carried out.

Heat curves can be implemented over the model „heat curve“ into the program. Control points in shape of diagrams „supply temperature versus outdoor temperature“ can be implemented over a dialog into the program. **Figure 4.12** shows an example of a heating curve which can be received from the simulation program.

![Heating Curve: Ueberg](image)

**Fig. 4.12:** Heating curve

For the simulation the user spares to fix the control points manual by using the heating curves.
4.1.6 The model „Load Forecasting“

The hydraulic and thermodynamic behaviour of district heating systems is determined significantly by the heat demand of the consumers and the temperatures of the distribution systems. Using sufficient models for load forecasting of single distribution systems the real operating behaviour can be simulated ideally. Therefore these models have been developed for the program. The characteristic parameters and therefore the models can be implemented over dialogs (see figure 4.12).

Any operating conditions (control points) are available by using those models. The number of manual inputs is reduced significantly.

In contrast with the load forecasting of electric networks the development of models for heat forecasting are in its infancy. A simple transfer from the proceeding of electric load forecast is not possible due to the different properties of both systems.

District heating systems show a large heat storage capacity and due to their slow transport velocity of the heat carrier they have high dead times. For the heat forecast parameters concerning the weather, especially the outdoor temperature, are much more significant that for the forecast of electric load. The heat consumption is directly determined by these parameters.

By means of operation data of different district heating distribution systems over a period of more than 1 year the behaviour of heat load, supply- and return temperature are analysed by using the multiple linear regression [2].

The evaluation of the data lead to the following general results:

- A dependence on the weekday has to be considered. This is caused substantially by the structure of the consumers. If only appartment buildings are supplied, no dependence of the weekday has to be considered. If the structure is characterized by industrial consumers, schools etc. a dependence of the weekday occures.
- Due to the behaviour of the consumers the time of the day has to be considered (morning showers, lowering the temperature during the night...).
- The load can be described sufficiently in dependence on the average outdoor temperature (the influences of the wind, the humidity of the air etc. lead to small deviations but compared with the outdoor temperature they are negligible).
- Load models can be described mathematical with a polynomial of the grade 3 (see figure 4.13).

In figure 4.13 the heat load of a typical distribution system is shown as a function of the outdoor temperature for 1:00 am. In addition the confidence interval for 90 % is presented. The heat load shows a maximum at low temperatures and decreases with increasing outdoor temperature until a minimum value is reached. Outside the interval \(-5^\circ C < T < 15^\circ C\) the calculated values base on only a few data. Due to this fact deviations are considerable at the edge of the interval. Therefore the curve is devided into three areas which are restricted by the minimum and the turning point of the polynomial.

With the following equation and values correlations have been developed with those load models for district heating systems can be described sufficiently:

\[
\text{Polynomial: } y = a_0 + a_1 \cdot T + a_2 \cdot T^2 + a_3 \cdot T^3, \quad \text{eq. (4.64)}
\]
The model „Load Forecasting“

minimum: \( T_{\text{min}} = -\frac{a_2}{3 \cdot a_3} + \sqrt{\left(\frac{a_2}{3 \cdot a_3}\right)^2 - \frac{a_1}{3 \cdot a_3}} \) and

eq (4.65)

turning point: \( T_{\text{TP}} = -\frac{a_2}{3 \cdot a_3} \)

eq (4.66)

**Fig. 4.13:** Heat load as function of average outdoor temperature

For outdoor temperatures smaller than \( T_{\text{TP}} \) (area 1) a linear extrapolation with the gradient of the turning point is carried out:

\[ y = y(T_{\text{TP}}) + \left(\frac{dy}{dT}\right)_{\text{TP}} \cdot (T - T_{\text{TP}}) \]  

\( y(T_{\text{TP}}) \) is the heat load at the turning point. The gradient is given by equation (4.67).

At extremely low outdoor temperatures the humidity of the air usually is very small. Therefore the heat loads calculated with equation 4.67 are too high. Concerning the supply guarantee equation 4.67 describes the „safe side“.

In area 3 the load is constant (base load). It results from hot-water generation substantially:

\[ y_{\text{min}} = a_0 + a_1 \cdot T_{\text{min}} + a_2 \cdot T_{\text{min}}^2 + a_3 \cdot T_{\text{min}}^3 = \text{const.} \]

\( y_{\text{min}} \) is the minimum heat load. The coefficients \( a_0, a_1, a_2, a_3 \) are determined by the regression method. This can be considered as a very good result as district heating systems have due
to heat accumulation and heat transmission a non-determinable component in the models. For
the intention of the simulation of the application of surfactants this unsteadiness is of no
importance.

Fig. 4.14: Dialog for the heat load model

Fig. 4.15: Diagram for the dialog heat load model

The modelling of the supply and return temperature can be performed analog to the heat load
model. The analysis of the parameter „supply temperature“ shows, that the cubic equation can
be used without restrictions. In comparison to the supply temperature the prognosis of the
The model „Surfactants“ is more difficult. In most cases the return temperature fluctuates around a constant value. For this reason the coefficients $a_1$ to $a_3$ in the cubic statement are set to zero and only a constant value of $a_0$ is used.

The quality of the model of the supply temperature is between 90% and 95% (in the case of the examined data this is equal to a standard deviation of 2 to 4 °C). The quality of the return temperature is between 80 and 90% (this corresponds to a standard deviation of 1 to 2 °C).

The simulation program uses the dialog presented in figure 4.14. In the array „Name“ the name of the model can be entered. Due to this the name can be identified in other dialogs. The input of the set of parameter of thermal load, supply and return temperature curves is entered with the polynomial coefficients $a_0$ to $a_3$ for any time of the day. In the array „actual numerical values“ the results of the simulation can be displayed as function of time and outside temperature. Furthermore the switch „Diagram“ allows a graphical presentation of any load and temperature curves for chosen conditions in order to verify the parameters of the polynomial (see figure 4.15).

### 4.1.7 The model „Surfactants“

For the calculation of the thermal and hydraulic behaviour the effects of the surfactant solutions as well as the mechanical load limit have to be calculable [10]. The breakdown of drag reduction and height of the efficiency from the applied additive depends on the temperature and concentration. The curves of the critical wall shear stress are similar by different concentrations. Only the position and height of the maximum of the critical shear stress (in the following discussion called as optimum value $\tau_{opt}$) clearly depends on the temperature (see figure 4.16).

The critical wall shear stress can be calculated in the following way. At first the optimum value of $\tau_{opt}$ and $T_{opt}$ have to be determined [10]:

$$T_{opt} = T_0 \cdot c^a_{\alpha},$$

$$\tau_{opt} = \tau_0 + b \cdot c_{\alpha}, \text{ in the area of application:}$$

$$c_1 \leq c \leq c_2 \ [\text{wppm}] \ \text{and} \ T_1 \leq T \leq T_2 \ [^{\circ}\text{C}].$$

For a representation which is independent of concentration the values of the critical wall shear stress are related to an optimum value of a certain concentration:

$$\tau_{rel} = \frac{\tau_{w,w}}{\tau_{opt}}.$$
In the same way the temperatures are related as difference $T - T_{\text{opt}}$ to the optimal temperature:

$$T_{\text{rel}} = \frac{T - T_{\text{opt}}}{T_{\text{opt}} + 273.15} \cdot T \text{ in } ^{\circ}\text{C}. \quad \text{eq. (4.73)}$$

![Graph showing critical wall shear stress of surfactant solutions as function of temperature and concentration](image)

**Fig. 4.16:** Critical wall shear stress of surfactant solutions as function of temperature and concentration

![Dialog for the model "surfactants"](image)

**Fig. 4.17:** Dialog for the model "surfactants"

The relation between $\tau_{\text{rel}}$ and $T_{\text{rel}}$ can be described as fraction of polynomial second degree:

$$\tau_{\text{rel}} = \frac{1 + A \cdot T_{\text{rel}} + B \cdot T_{\text{rel}}^2}{1 + A \cdot T_{\text{rel}} + C \cdot T_{\text{rel}}^2}. \quad \text{eq. (4.74)}$$

The coefficients of the functions 4.69, 4.70 and 4.74 have been determined for Habon, Obon and Dobon as well as independent from the additive [10].
With equation 4.74 the drag reducing effect can be calculated. Although equation 4.74 is implemented into the program, equation 2.5 which is independent from the concentration is preferred due to its simple construction:

$$\tau_{w,s} = 0.35 \cdot (\tau_{w,w})^{0.78}.$$  

The dialog for the model „surfactant“ is shown in figure 4.17. The above mentioned parameters for the equations 4.69, 4.70 and 4.74 can be implemented with it. Equation 4.75 does not need any dialog. The field „Name“ is used analog to the above described functions. With the switch „Diagramm“ the critical wall shear stress can be displayed as function of temperature [23].

### 4.1.8 Network calculation

The calculation of the steady state of the network with TenSim has to fulfill the requirement of small calculation times. With increasing number of system elements (number of loops) the calculation time is rising significantly. Up to now the considered networks are transport systems and therefore they have not a lot of loops (or no loop). The calculation of structures without loops can be carried out without using analysis that take a great deal of time. In this case the calculation times are in the range of some minutes. Otherwise in case of systems with a lot of loops much time will be necessary.

The course of calculation for the network calculation is shown in the appendix. Assumption for the calculation is that all elements which build up the system are given (and implemented into the program) to create the configuration of the network. The calculation can be described as follows:

In a **first step** the user has to set up the conditions for control. After that the calculation can be started. The program carries out a check of consistence concerning the validity of the users input. This is mainly the test of the connection of the single elements (inadmissible connections, complete control instructions...). This is e.g. the existence of a pressure maintenance. If the test of consistence is positive the real calculation can be carried out:

The first step of the calculation of a transport system is the estimation of start values for flow rates of the side of the consumers. Therefore the control instructions for consumer stations and supply temperatures are used because these parameters determine the temperatures and heat loads of the consumers (and the flow rate of the consumers).

In a **second step** the primary flow rates and the primary return temperature (the supply temperature is known) are calculated. Therefore the calculation of the heat exchangers as connection between the primary and the secondary side are carried out. With the results of this calculation the loop correction is carried out. The result of this process is the flow rate of the transport system.

In the **third step** the calculated flow rates are distributed equally among different groups of pumps, heat exchangers, etc. which are usually connected parallel (and are equally designed) and fulfill common control instructions.

During the **fourth step** the calculation of the heat producer is carried out. Therefore an equal distribution of different heat producer is intended. The supply temperatures are adjusted due to the control instructions of the heat producer.
In the fifth step the thermodynamic calculation of the network is carried out. Therefore the program creates a "topological list". In this list the order of the thermodynamic calculations of all system elements is determined. The list is created in this way that the elements of the supply are calculated at first and then the elements of the return line (from the producer to the consumers).

During the sixth step, the hydraulic calculation is carried out. Analog to the thermodynamic calculation a "topological list of nodes" is created to determine the pressure in every node. At the same time the hydraulic control of the system is carried out. After that it is checked whether the pressure is constant in all nodes. Otherwise the iteration is continued until the pressure is constant in every node or until a certain number of iterations has been carried out.

If the steady state - concerning the thermodynamic and hydraulic conditions - is reached the calculation is interrupted. Otherwise it is continued until the maximum number of iterations is reached or the calculation is interrupted by the user.

4.1.9 Presentation of the results

The program TenSim offers several possibilities of presenting the results to the user. A simple but intricate kind of presentation is the output of all parameters of every system element in a file. The advantage of this presentation is that all information of any element are available.

The program is presenting a flow sheet of the system (see appendix). In this flow sheet, any information can be seen on-line. Over a dialog any parameters can be received or indicated permanently.

The flow chart (or any selected part of the flow chart) and the containing informations can be saved on date media. Therefore, they can be used for further simulations.

The steady state of a district heating system can be described clearly as temperature- and pressure diagramms. In these diagramms the temperature and pressure of every node is shown as a function of their position in the network. The position of a node results from the distance from a reference node in direction of the flow (real position). The elements of the supply and return pipe are presented in different colours. Therefore the results can be grasped easily. A disadvantage of using real coordinates is the splitting of the system so that the circuit can not be pursued easily.

Therefore a presentation which uses a fictitious position (coordinate) of the nodes has been implemented, too. This presentation requires a lot of expenditure concerning the calculation of the coordinates but it allows a very clear presentation of the results. The elements of the supply and return pipe are also presented in different colours. The direction of the flow is indicated and the system elements are presented, too, if there is space enough. Examples for those diagrams are shown in the appendix.

In addition, all implemented models can be presented graphically over dialogs (control, pumps, heat exchangers, etc.).

Chapter 4 gives an overview of the function of the program, the models which are used to describe the system elements, the assumptions, boundary conditions and other necessities. An example for the simulation of a real district heating system is given in the appendix. Furthermore the flow sheets which describes the sequences of calculation of the different models are presented in the appendix.
The additional made manual for TenSim describes how to use the program. In this manual an example-calculation is given. The single dialogs are explained and it is described how to implement new transport systems and models.
5. Summary and Outlook

In the context of this study a simulation program for calculating the behaviour of district heating systems operating with drag reducing additives has been developed. The behaviour of district heating transport systems as well as of single components - especially typical heat exchangers such as plate, shell and tube and helical tube heat exchangers - can be calculated with the program „TenSim“ when applying drag reducing additives.

The simulation program can be used to modify existing networks and create new district heating systems to realize the operation with surfactant solutions. Single system parts (existing and additional necessary devices) - especially heat exchangers - can be designed or modified to achieve a design which guarantees a well working operating mode.

By simulating several cases of modified systems and comparing the results of the simulations an optimum technical solution can be achieved.

In an example calculation (see Appendix - A, chapter 8.1) the simulation program has been tested. The test system (the system Völklingen Luisenthal) has also been used for a long term full scale test (application of Dobon-G/Sodiumsalicylate), so all technical data (data for apparatus like pumps, heat exchangers, pipes, geographical data etc.) were available as well as results for the operation with drag reducing additives. Therefore simulation results could be compared with results of a real application. The comparison showed that the simulation results calculated with „TenSim“ reproduce the real results sufficiently.

A necessary condition for the application of drag reducing additives is the economic viability. Comparing the modified system working with drag reducing surfactants (that means the optimum technical solution which has been found with the simulation tool) and the original system operating with pure water it is possible to estimate the savings in cost due to the application of surfactants.

Therefore cost functions have to be evaluated in further studies to be able to calculate the investigations that are caused by the additives. Those functions for german conditions have been developed in several studies carried out at the University of Dortmund [18][23][25][29]. Some examples of cost functions are given in the Appendix - B.

Furthermore economics calculations have been carried out. In this studies a general model has been used to estimate the potential in saving costs on principle. The model and some results are also given in Appendix - B. With the results shown in Appendix - B a first estimation can be carried out. Using the developed cost functions, a more precise estimation can be carried out.

Next step concerning the application of drag reducing additives in district heating systems should be the simulation of concrete transport systems with „TenSim“ - including the modifications. Furthermore economics calculations (estimations) should be carried out for real systems to get the necessary informations about the economic aspects of the application of drag reducing additives in existing district heating networks.
# 6. Symbols

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<th>Symbol</th>
<th>Description</th>
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### Symbols

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<td>[1]</td>
</tr>
<tr>
<td>(\xi)</td>
<td>drag coefficient</td>
<td>[1]</td>
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<tr>
<td>(\Psi)</td>
<td>average change of medium temperature</td>
<td>[1]</td>
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### Indices

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<tr>
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<tr>
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<tr>
<td>ap</td>
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<td>b</td>
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<td>Special symbols</td>
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<td>-----------------</td>
<td>-------------------------------------------------------------------------</td>
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<tr>
<td>aKW</td>
<td>altes Kraftwerk (old power plant)</td>
</tr>
<tr>
<td>BFS</td>
<td>Breadth First Search</td>
</tr>
<tr>
<td>BWL</td>
<td>Bergwerk Luisenthal (mine Luisenthal)</td>
</tr>
<tr>
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<td>Computer Aided Design</td>
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<tr>
<td>CMC I/II</td>
<td>first/second Critical Micelle Concentration</td>
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<tr>
<td>DM</td>
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<tr>
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<td>Nominal Diameter</td>
</tr>
<tr>
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<td>Drag Reduction</td>
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<td>Heat Output Reduction</td>
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<td>Lethal Dosis</td>
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<td>Lambda Locked Loop</td>
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<tr>
<td>MID</td>
<td>Magnetive-Induktive Flowmeter</td>
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<td>NTU</td>
<td>Number of Transfer Units</td>
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<td>OHR</td>
<td>Overall Heat Transfer Reduction</td>
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<tr>
<td>PCR</td>
<td>Pumping Cost Reduction</td>
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<tr>
<td>QAV</td>
<td>Quaternary Ammonium Compound</td>
</tr>
<tr>
<td>SAIS</td>
<td>Siedlung Altenkesseler Straße</td>
</tr>
<tr>
<td>SIS</td>
<td>Shear Induced State</td>
</tr>
<tr>
<td>SoSal</td>
<td>Sodiumsalicylate</td>
</tr>
<tr>
<td>TDM</td>
<td>Thousand Deutsche Mark</td>
</tr>
<tr>
<td>wppm</td>
<td>weight parts per million</td>
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### Non Dimensional Parameters

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</tr>
<tr>
<td>Nu</td>
<td>( \text{Nu} = \frac{\alpha \cdot D}{\lambda} )</td>
</tr>
<tr>
<td>Pr</td>
<td>( \text{Pr} = \frac{\nu}{\alpha} = \frac{\eta \cdot c_p}{\lambda} )</td>
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<tr>
<td>Eu</td>
<td>( \text{Eu} = \frac{\Delta p}{\rho \cdot u^2} )</td>
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8. Appendix - A

8.1 Example for a simulation

For a real system - the system Völklingen Luisenthal in Germany - simulations have been carried out for several operating conditions. In the system, a part of a large district heating network, a full scale test has been carried out [24]. Therefore all necessary information have been available. The gathering of information as basis for the simulation is a big problem. In this case, due to planning and carrying out the full scale test, this problem did not appear.

Before applying drag reducing additives a lot of modifications have to be carried out. Most important measure was the hydraulic separation of the test system from the transport system „Schiene Saar“. Figure 8.2 and 8.3 show the system in the origin state and after changing the operating mode from direct to indirect. With TenSim the modifications - especially the new designed plate heat exchangers for the hydraulic separation - can be tested.

8.1.1 The system „Völklingen Luisenthal“

Figure 8.1 shows the plan of the system with the most important data. The transmission station (from „Schiene Saar“ to the system „Völklingen Luisenthal“) is installed in the „altes Kraftwerk“. The consumer stations „altes Kraftwerk“ (aKW) with 0.5 MW, a garage („Kfz-Werkstatt“) with 1.48 MW, a mine, the „Bergwerk Luisenthal“ (BWL) with 7.2 MW and a estate („Siedlung Altenkesseler Straße“, 11 houses) with a total load of 0.87 MW are connected to the system. The pipe diameter is from DN 250 from „Schiene Saar“ to DN 25 to the last house. More data are presented in figure 8.1.

The consumer stations aKW, Kfz-Werkstatt and BWL are equipped with two equal helical tube heat exchangers each. In the houses plate heat exchangers are installed. The supply temperature of the „Schiene Saar“ in controlled by an admixing from the return. Therefore additional pumps are necessary.

To guarantee a sufficient flow rate the pressure difference over the station BWL is controlled by controlling the speed of the circulation pumps. The control point is 800 mbar. If the pressure difference of 800 mbar is not sufficient to supply the estate additional pumps (P_d/P_s in figure 8.2) for increasing the pressure are necessary.

To realize the application of the surfactant system Dobon-G/sodiumsalicylate the modifications shown in figure 8.3 (if compared with figure 8.2) have been carried out. The most important measure was the installation of two plate heat exchangers to separate the system Luisenthal from the „Schiene „Saar“. The installed apparatus have been designed for drag reducing surfactant solutions and therefore the heat exchange area has been increased of about 30%. The following results show to what extend this measure has compensates the effect of heat transfer reduction.
Furthermore an additional pressure maintenance and installations for make-up water had to be installed. Additional circulation pumps and control devices was necessary for the test system Luisenthal. Some heat meters have been exchanged due to the influence of cationic surfactant solution on their accuracy.

pipes: DN 200-DN 25  
nominal pressure: PN 25  
length of the pipe: 2,600 m  
temperature: 60-110 °C  
flow velocity: 0.6-2 m/s  
total heat load: 10 MW

Fig. 8.1: Plan of the system Luisenthal [24]

The heat exchangers of the consumer stations have not been modified. The increase of the return temperature due to the drag reducing effect in the consumers’ heat exchanger has been tolerated in the frame of the full scale test. For technical applications it has to be checked clearly to what extend such an increase can be tolerated.

Concerning the planning of the modifications the design of the two additional plate heat exchangers was of special interest. In a first step the heat exchangers have been designed for pure water. Table 8.1 shows the results for this design.

In a second step, basing on the results of laboratory tests, the same calculation has been carried out for the application of drag reducing additives as for pure water. Therefore the Nusselt characteristics that have been found in laboratory tests for small apparatus of similar geometry have been used. The results of the calculation are presented in table 8.2.
The system "Völklingen Luisenthal"

Fig. 8.2: Schematic flow sheet of the origin system

Fig. 8.3: Schematic flow sheet of the modified system (until "altes Kraftwerk")
The heat exchange area for the plate heat exchangers which operate with surfactant solution are designed 34% larger than those for pure water. Therefore the number of plates has been increased from 83 to 111.

<table>
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<tr>
<th>requirements</th>
<th>primary side</th>
<th>secondary side</th>
</tr>
</thead>
<tbody>
<tr>
<td>fluid</td>
<td>water</td>
<td>water</td>
</tr>
<tr>
<td>flow rate</td>
<td>82 m³/h</td>
<td>132.5 m³/h</td>
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<tr>
<td>entrance temp.</td>
<td>130 °C</td>
<td>60 °C</td>
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<tr>
<td>outlet temp.</td>
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<td>100 °C</td>
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<td>nominal heat load</td>
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<tr>
<td>area</td>
<td>73 m²</td>
<td></td>
</tr>
<tr>
<td>number of plates/type</td>
<td>83/85</td>
<td></td>
</tr>
<tr>
<td>height of the plates</td>
<td>2.496 mm</td>
<td></td>
</tr>
<tr>
<td>breadth of the plates</td>
<td>850 mm</td>
<td></td>
</tr>
<tr>
<td>pressure loss</td>
<td>300 mbar</td>
<td>750 mbar</td>
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</table>

**Tab. 8.1:** Design of the plate heat exchangers for water [42]

<table>
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<tr>
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<tbody>
<tr>
<td>fluid</td>
<td>water</td>
<td>surfactant</td>
</tr>
<tr>
<td>area</td>
<td>97.9 m²</td>
<td></td>
</tr>
<tr>
<td>number of plates/type</td>
<td>111/85</td>
<td></td>
</tr>
<tr>
<td>pressure loss</td>
<td>200 mbar</td>
<td>450 mbar</td>
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</table>

**Tab. 8.2:** Design of the plate heat exchangers for surfactant solution [42]

How far this increase of the number of plates had compensated the influence is shown in the following simulations which use the results of the measurements that have been carried out in the frame of the full scale test (e.g. Nusselt characteristics).
**8.1.2 Simulation**

Two examples for the simulation of the system under different conditions are presented:

- Operation with pure water and design for the heat exchangers according table 8.1 (design for water) and
- Operation with surfactant solution and the special designed heat exchangers according table 8.2.

In both cases the configuration of the system shown in figure 8.3 is the basis for the calculations. Before starting the simulation the steady state has to be defined by fixing the control points and conditions (e.g. the heat load of the consumers, configuration of the pressure maintenance). The pressure is fixed on the suction side of the circulation pumps ($P_1/P_2$). The control point is 4.5 bar (in the flow chart of the system which is created with the CAD module of the program, the point where the pressure is controlled is node No. 22). The control point and the node where the pressure has to be controlled can be implemented with a dialog.

The circulation pumps ($P_1/P_2$) of the system Luisenthal have to ensure the pressure difference of 800 mbar over the installations of the „Bergwerk Luisenthal“ (this means a pressure difference of 800 mbar between node No. 15 and 4).

The additional pumps $P_4/P_5$ operate with a constant speed of 900 rpm in case of operating conditions which require more than 800 mbar for the supply of the estate. The parameters of these pumps can be changed in any way over a dialog.

Control input for the heat supply from the „Schiene Saar“ is the supply temperature of the system Luisenthal. The supply temperature is controlled with the speed of the pumps $P_A$ and $P_B$. The control point can be pre-determined with heating curves (as function of outdoor temperature) or implemented manually.

The consumers are characterized by the heat load, the supply- and the return temperature. Therefore the flow rate of the consumers is fixed, too. The flow rate is adjusted with valves behind the heat exchangers. The control points for the heat load and the temperatures can be given over load models or implemented manually. Load models for the system are available.

In the following list the operating condition for the simulation of the two different configurations are shown. Therefore the nominal heat load is assumed:

- Total heat load: 9.62 MW (4.81 MW for each plate heat exchanger)
- supply temperature: primary: 110 °C, secondary: 100 °C.

The results from the simulation can be taken directly from the flow sheets. Furthermore the diagrams for the pressure (fictitious coordinates) are shown. For the simulation with pure water two different cases are considered:

- $P_4/P_5$ in operation and
- $P_4/P_5$ not in operation.

The results in figure 8.4 and 8.5 show that the supply of the „Siedlung Altenkesseler Straße“ without $P_4/P_5$ cannot be realized. In the estate the pressure difference between supply and return becomes zero. During the simulation the user gets an alert in such cases.
Fig. 7.4: Flow sheet with the results of the simulation „water without P₄/P₅“
Pie 7.5: Diagram for operation with and without platelet.
Fig. 7.6: Flow sheet with the results of the simulation „water with $P_4/P_5$“
Fig. 7.7: Diagram for operation with „water with P.g/P.s“
Fig. 7.8: Flow sheet with the results of the simulation „surfactants without P₄/P₅“
Fig. 7.9: Diagram for operation with "surfactants without $P_4/P_5$".
The supply can only be guaranteed if the pumps \( P_4/P_5 \) are in operation. In **figure 8.6** and **8.7** the results of the simulation with pumps \( P_4/P_5 \) in operation are presented.

In this case the supply of the complete estate is safe. Conspicuous of these results is the high pressure difference at the end of the „Siedlung Altenkesseler Straße“ that results from the operation of \( P_4/P_5 \) with constant speed. Reducing the speed, the pressure difference could be decreased. This would lead to a more economic operation.

Using drag reducing surfactants (see **figure 8.8** and **8.9**), no additional pumps are necessary. Due to the drag reducing effect the gradient of the pressure curve is decreasing so the control point for the „Bergwerk Luisenthal“ (800 mbar) is sufficient for the supply of the complete estate. Considering the pressure difference behind the last consumer of the „Siedlung Altenkesseler Straße“ the value of 800 mbar could be decreased significantly. The most important results of all simulations are presented in table 8.3.

<table>
<thead>
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<th>result</th>
<th>place</th>
<th>fluid</th>
<th>difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure loss</td>
<td>pump P1</td>
<td>water</td>
<td>3.35 bar</td>
</tr>
<tr>
<td></td>
<td>pump P4</td>
<td>surfactants</td>
<td>1.00 bar</td>
</tr>
<tr>
<td></td>
<td>sum</td>
<td></td>
<td>4.35 bar</td>
</tr>
<tr>
<td>return temperature</td>
<td>primary</td>
<td>water</td>
<td>71.5 °C</td>
</tr>
<tr>
<td></td>
<td>secondary</td>
<td>surfactants</td>
<td>59.7 °C</td>
</tr>
<tr>
<td>flow rate</td>
<td>primary</td>
<td></td>
<td>212 m³/h</td>
</tr>
<tr>
<td></td>
<td>secondary</td>
<td></td>
<td>215 m³/h</td>
</tr>
</tbody>
</table>

**Tab. 8.3:** Results of the simulation

The example clearly shows the possibility to increase capacity or to prevent bottle necks. Due to the drag reducing additives the capacity can be increased so far that in cases of high loads no additional pumps are necessary to guarantee the supply.

If no additional pumps had been installed the application of drag reducing additives would be a possibility to ensure the supply without building a new pump station.

Comparing the results of the simulation (see table 8.3) a significant increase of the return temperatures and mass flow rates is obvious. This is a reaction of the system to the heat transfer reduction. The flow rate is rising because better heat transfer coefficients are resulting. The flow rate is rising until the demanded heat load can be realized.

The increase of the flow rate leads to an increase of the return temperature. These two combined effects should not occur when applying surfactants. For technical applications the heat transfer reduction has to be compensated sufficiently.

With the simulation program TenSim sufficient modifications can be developed. As an example the new installed heat exchangers are simulated to calculate the necessary design. That means in worst case that the operation with surfactants at maximum heat load has to be identical to that one of the apparatus that is designed for water.
Simulations of several apparatus have shown that the condition „identical operating parameters for the maximum heat load“ are sufficient. Points of lower loads fulfill the requirements in any case due to the fact that the heat transfer reduction decreases with the flow rate.

For the following simulations, the parameters presented in table 8.1 are used. In addition the Nusselt characteristic of the apparatus for the surfactant Dobon-G/SoSal is available. This has been developed in the frame of the field test. Compared to the laboratory tests the influence on the heat exchangers has been stronger.

The simulation for the application of surfactants in the two heat exchangers is carried out in the following way:
- Considering the maximum heat load,
- calculation of the conditions for the apparatus that has been designed for water and
- increasing the number of plates until the required conditions are reached.

In figure 8.10 the results are presented. Temperatures, heat load and the required values are outlined as function of number of plates (or the heat exchange area). The entrance temperatures and flow rates are given according to table 8.1.

![Figure 8.10: Example for the design of a plate heat exchanger operating with drag reducing additives](image)

Figure 8.10 clearly shows that the increase of the heat exchange area of ca. 30 % in not sufficient to compensate the effect due to drag reducing additives. A complete compensation can be reached if the number of plates is 194. Therefore, the heat exchange area has to be more than doubled compared to the design for water (83 plates).
Considering the installed apparatus (111 plates) the nominal heat load is 5.72 MW instead of 6.00 MW. The secondary supply temperature is about 2 ° too low and the primary return temperature 3 ° too high. The required correspondence with the values for water are not reached. But the differences are only small and therefore, problems concerning the supply did not appear.

With TenSim single system elements - especially heat exchangers - as well as complete district heating system can be calculated when using friction reducing additives. If the necessary informations are available the effects can be simulated and measures to compensate for these effects can be developed and tested.

8.2 Description of the most important modules

8.2.1 Flowsheet of the BFS-Algorithm.

Fig. 8.11: Flow sheet of the BFS-Algorithm [5]
8.2.2 Flow chart for the calculation of network

- Network calculation
  - Set start values for the heat exchangers
  - Calculate heat extraction at the consumer stations
  - Fix new mass flow for system (loop correction)
  - Calculate heat input (make dispatch)
  - Create a topological list for thermal calculations
  - Thermodynamic calculation of the system

Temperature of all nodes stationary? or max. number of iterations exceeded?
- No
- Yes
Fig. 8.12: Flow chart of the system calculation [5]
8.2.3 Flow charts for the calculation of heat exchangers

**Shell and tube heat exchangers**

Start values:
- Design data
- Operation data
- Entrance pressure and temperature
- Water mass flow rates

Calculate:
- Geometry

Start values:
- Outlet temp. = arithm. mean of entrance temperatures
- Mean fluid temp. = arithm. mean of entrance and outlet temperature.
- Wall temp. = mean fluid temp.
- Pressures = entrance pressures

Calculate shell side:
- Physical properties

Calculate shell side:
- Pressure drop
- New pressure

Shell side pressure constant? No

Calculate shell side:
- Surface heat transfer coefficient

Yes
Calculate:
- Physical properties
  tube side

Surfactants?

Yes

Calculate: critical Re-number for the decrease of efficiency

No

Calculate for tube side:
- Pressure drop and from this new reference pressure
  (Calculation for water!)

Re(tube) < Re(critical)?

Yes

Calculation for water or surfactants?

No

Calculate for tube side:
- Pressure drop and from this new reference pressure
  (Drag reducing effect!)

Tube side pressure constant?

No

Yes
Flow charts for the calculation of heat exchangers

Calculate for tube and shell side:
- Relation between heat flow capacities
- NTU overall, NTU cell
- Non-dimensional change of temperature

Calculate:
- Overall heat transfer coefficient

Calculate for tube and shell side:
- Surface heat transfer coefficient (calculation for water!)

Calculate:
- Non-dimensional temperatures (Calculations of cells)

Calculate:
- Outlet temperatures (tube)
- Outlet temperatures (shell)
Calculate:
- Mean degree of the efficiency of the cells
- Non-dimensional mean difference of temperature
- Mean difference of temperature

Calculate for tube side:
- Wall temperature

Tube wall-temperature constant?  
Yes

Calculate for tube side:
- Mean fluid temperature

Fluid temperature in the tube constant?  
Yes

Calculate for shell side:
- Wall temperature

Shell wall-temperature constant?  
Yes
Calculate for shell side:
- Fluid temperature

Fluid temperature in the shell constant?

Yes
- Calculate:
  - Transferred heat load

Output of results

No

Fig. 8.13: Flow charts for the calculation of heat exchanger [5]
**Helical tube heat exchanger**

Start values:
- Design data
- Operation data
- Entrance pressure and temperature of water
- Water mass flow rates

Calculate:
- Geometry

Start values:
- Fluid- and wall temp. = entrance temp.
- Pressures = entrance pressures

Calculate for tube side:
- Flow speed, \( w_p \)
- Reynolds number

\[ w_p > w_{p, \text{max}} \]

Yes

Warning

No

Cell

Yes

Calculate: critical Re-number for the decrease of the efficiency

No
Flow charts for the calculation of heat exchangers

1. Tube row
   - surfactants?
     - Yes: Re(tube) < Re(critical)?
       - Yes: Calculation for water or surfactants?
         - No: Calculation for tube side: pressure drop and from this new reference pressure (Drag reducing effect!)
       - No: Next tube row, until all rows are calculated
     - No: Calculation for tube side: pressure drop and from this new reference pressure (Calculation for water!)
   - Next tube row, until all rows are calculated

2. Calculate for tube side:
   - Mean pressure drop
   - Outlet pressure

3. Calculate for shell side:
   - Pressure drop
   - Outlet pressure
Calculate for tube side:
- Heat transfer coefficient of the surface
  (Calculation for water!)

Calculate for tube side:
- Mean heat transfer coefficient of the surface

Calculate for shell side:
- Heat transfer coefficient of the surface
Calculate:
- Heat transfer coefficient of the surface

Calculate:
- Relation between heat flow capacities
- NTU-values
- Degrees of efficiency of the cells

Next cell, until all cells are calculated

Calculate:
- Outlet- and intermediate temperatures

Cells

Calculate:
- Non-dimensional changes in temperature
- Mean difference of temperatures
- Wall temperatures
- Heat flow rate

Next cell, until all cells are calculated
Abb. 8.14: Flow chart for the calculation of helical tube heat exchangers
**Shell and tube bundle condenser**

Start values:
- Design data
- Operating data
- Entrance pressure and temperature of water
- Water mass flow rates

Start values tube side:
- Outlet temp. = arithm. mean of entrance- and steam temperature
- Mean fluid temp. = arithm. mean of entrance- and outlet temperature
- Wall temp. = mean fluid temperature

Start value on the shell side:
- Wall temp. = arithm. mean of steam- and wall temperature on the tube side

Calculate for shell side:
- Physical properties
- Heat transfer coefficient of the surface

Calculate for tube side:
- Physical properties
Calculate for tube side:
- Pressure drop and from this new reference pressure
  (Calculation for water!)

1. Calculate for tube side:
- Pressure drop and from this new reference pressure
  (Drag reduction effect!)

Tube side pressure constant?

Yes

2

No

B
Flow charts for the calculation of heat exchangers

Calculate for tube side:
- Heat transfer coefficient of the surface
  (Calculation for water!)

Calculate for tube side:
- Overall heat transfer coefficient

Calculate for tube side:
- Outlet temperature

Calculate for tube side:
- Non-dimensional, mean difference of temperatures
- Wall temperature of the tube

Wall temperature on the tube side constant?

Re(tube) < Re(critical)?

Calculation for water or surfactants?

Calculation for tube side:
- Surface heat transfer coefficient
  (drag reduction effect!)

Yes

No

3

2

No

Yes

0

1

B
Calculate for the tube side:
- Mean fluid temperature

Mean fluid temperature on the tube side constant?

Yes

Calculate for shell side:
- Non-dimensional mean difference of temperature
- Wall temperature of the tube

Tube wall temperature on the shell side constant?

Yes

Calculate:
- Transferred heat flow

Output of the results

Abb. 8.15: Flow chart for the calculation of shell and tube bundle condensers
9. Appendix - B

In this chapter a short overview is given about several studies from the University of Dortmund. Main subjects of these studies are cost functions concerning the application of cationic surfactants and general models to estimate the savings in cost for several possibilities of application.

9.1 Cost Functions concerning the Application of FRAs

Within the scope of the above mentioned studies cost functions of the following system elements have been evaluated:

- centrifugal pumps,
- heat exchangers (shell and tube, plate heat exchangers),
- pressure maintenances,
- water treatment equipment,
  - reverse osmosis,
  - ultrafiltration and degasification,
- containers,
- buildings,
- pump stations,
- transmission stations,
- pipes,
- dosing devices and surfactants,
- energy and water,
- heat generation cost in:
  - heat plants,
  - combined heat and power plants,
  - waste heat and further cost functions.

In this chapter examples for cost functions of important system parts are given. An overview of all functions is given in [23]. The functions are describing german conditions and have to be transformed into the corresponding conditions of other countries if necessary.
Important system parts of district heating systems are heat exchangers. Typical for district heating systems are plate, shell and tube and helical tube heat exchangers. Figure 9.1 shows the cost function (and the prices which have been achieved by industry investigations). This cost function can be described with equation 9.1 for apparatus from 70 to 450 m² (this corresponds to a heat load of ca. 5 to 40 MW).

Equation 9.1 describes to pure price of the apparatus and not the total cost including transport, installation, insurance etc. The total cost can be calculated by considering special factors $F_{bm}$ (bm - bare module), $F_{tm}$ (tm - total module) and $F_{gr}$ (gr - grass root) [37].

$$K_{\text{PlateHE}} = 1,371.4 \cdot A^{0.66} \text{ [DM, 1992]}, \text{ with } A \text{ in } [m^2].$$

**eq. (9.1)**

![Figure 9.1: Cost of plate heat exchangers (equation 9.1, 1992)](image)

The cost of plate heat exchangers do not show the typical increase in prices for system parts of chemical plants [29]. At the end of the eighties and in the beginning of the nineties a strong pressure of competition occured. Due to the competition in this branch of industry the prices were decreasing. The prices in 1992 were below those for 1978 [37].

The cost for shell and tube and helical tube heat exchanger can be gathered from [37]. The cost for centrifugal pumps, water treatment equipment, containers, buildings, pump stations and pipes can also be gathered from literature. Equations for calculating heat generation cost, dosing devices and additives are available in [25] and [29]. Energy and water cost must be determined according to the existing conditions.

Due to the fact that the application of drag reducing additives is considered only in transport systems which are hydraulically separated from the distribution network a cost function of entire transmission stations has been developed [29].

In figure 9.2 the specific cost of entire transmission stations are shown. Values from literature [3] are compared with own calculations which have been carried out with the created cost functions. The assumptions (temperatures, pressure etc.) which have been made can be looked up in [29]. The values from literature are significant higher than those from own calculations. The main reason for this behaviour ist the conversion of the literature values from 1975 to actual values (1992) with the typical index. Due to the progress in technology (e.g. the applica-
tion of plate heat exchangers instead of helical tube heat exchangers) the increase in prices has been smaller than for typical components of chemical industry. Therefore the application of the typical index leads to values which are too high.

![Graph showing cost of entire transmission stations](image)

**Fig. 9.2:** Cost of entire transmission stations (1992)

For transmission stations with a nominal heat load above 50 MW, an appropriate estimation with 40,000 DM/MW can be carried out (for the assumptions the calculation is based on - see [29]).

## 9.2 Economics Calculation

### 9.2.1 The model for economics calculation

Calculations to estimate the possible savings have been carried out using the model shown in **figure 9.3**. Basis of this model is the application of drag reducing additives in the transport system only. In the transport system basic heat which is generated in CHP-plants is transported to a transmission station. The peak load is generated in a heat plant which directly supplies the distribution network.

Due to the restriction that the application of drag reducing additives is only foreseen in transport systems two different cases are distinguished in principle:

- a transmission station has to be installed and
- a transmission station is already existing.
In case that a transmission station has to be built a large amount of investment cost have to be spent to realize the condition of hydraulic separation. Further information about boundary conditions assumptions etc. can be gathered from [25] and [29].

![Diagram of transmission station](image)

**Fig. 9.3:** Model for economics calculation [25]

The use of this general model gives only an estimate because no general district heating systems exist. The following considerations and calculations show the range in which possible savings in cost can be expected.

### 9.2.2 The cost model for „saving pumping cost“

Savings in cost can be achieved if the following condition is fulfilled:

\[-a_n \cdot K(\dot{Q}, \Delta T_{\text{max}}, L) + r(\dot{Q}, \Delta T_{\text{max}}, L) > 0.\]  

**eq. (9.2)**

The first part of unequation 9.2 describes the annual cost due to necessary investment (e.g. cost for additional heat exchangers, new water treatment plants etc.) to realize the application of drag reducing additives. The factor $a_n$ is defined as follows:

\[a_n = \frac{p_k \cdot (1 + p_k)^n}{(1 + p_k)^n - 1}.\]  

**eq. (9.3)**

The parameter $p_k$ describes the interest rate.

The second part of equation 9.2 describes the annual savings (or losses) in cost which result from the application of cationic surfactants.
The cost model for "saving pumping cost"

\[ r (\dot{Q}, \Delta T_{\text{max}}, L) \text{ can be calculated with the following equation:} \]

\[ r = DR_m \cdot \frac{\Delta p}{\Delta L} \cdot \frac{V_N}{\eta_p} \cdot s_{st} \cdot T_P \cdot L - \frac{\pi}{4} \cdot D^2 \cdot L \cdot f_{\text{loss}} \cdot k_s - \]

\[ i_{\text{main}} \cdot (K - I_s) - V_{ss} \cdot f_{\text{loss}} \cdot k_{\text{treat}}. \]

The symbols are defined as follows:

- \( DR_m \) = average drag reduction \([1]\),
- \( \Delta p/\Delta L \) = pressure gradient \([\text{Pa/m}]\),
- \( V_N \) = nominal flow rate \([\text{m}^3/\text{s}]\),
- \( \eta_p \) = efficiency of the pump; \( \eta_p = 0.75 \) (speed controlled pump),
- \( s_{st} \) = cost for electricity; \( s_{st} = 0.00018 \text{ DM/Wh} \),
- \( T_P \) = pumping duration in \([\text{h/a}]\):

\[ P_{\text{max}} \cdot T_P = \int_0^{8.760} P(t) \, dt, \]

\( L \) = length of the pipe \([\text{m}]\),
\( D \) = pipe diameter \([\text{m}]\),
\( f_{\text{loss}} \) = annual water losses: \( f_{\text{loss}} = V_{\text{loss}}/V_{ss} = 0.2 \) \([\text{a}^{-1}]\),
\( k_s \) = price for surfactants; \( k_s = 50 \text{ [DM/m}^3\text{water]} \) (1993),
\( i_{\text{main}} \) = annual maintenance rate; \( i_{\text{main}} = 0.025 \) \([\text{a}^{-1}]\) (incl. insurance),
\( K \) = capital cost due to the application of additives \([\text{DM}]\),
\( I_s \) = cost for additive at the beginning of application:

\[ I_s = \frac{\pi}{4} \cdot D^2 \cdot L \cdot 1.5 \cdot k_s, \text{ in [DM]}; \]

the factor 1.5 considers adsorption losses of the first dosing.

\( V_{ss} \) = volume of subsystem \([\text{m}^3]\) and
\( k_{\text{treat}} \) = specific operating cost for water treatment of the subsystem \([\text{DM/m}^3]\).

For a certain maximum temperature difference and a maximum heat load the necessary size of a system (that means the pipe length \( L \)) to reach economic viability can be calculated with the condition:

\[ -a_n \cdot K (\dot{Q}, \Delta T_{\text{max}}, L) + r (\dot{Q}, \Delta T_{\text{max}}, L) = 0, a_n [1/\text{a}]. \]

With the following assumptions the "economic length" can be calculated:

- \( \Delta T_{\text{max}} = 70 \text{ °C} - (T_{\text{ret,min}} = 60 \text{ °C}, T_{\text{supp,max}} = 130 \text{ °C}) \),
- \( DR_m \) has been calculated with an assumed heating curve, \( DR_m \) can be assumed to 60% (pessimistic estimation),
- \( \Delta p/\Delta L \) and \( D \) can be calculated with an optimization calculation or - for concrete examples - the real values can be used,
• $T_P$ has been calculated with the following equation:

$$T_P = 1.244 \cdot \left( \frac{\Delta T_{\text{max}}}{0.7} - 44 \right)^{0.35}, \quad \text{eq. (9.8)}$$

• $K$ can be calculated by adding all investment cost which are necessary due to the application of additives. These are for example:
  • cost for dosing devices,
  • cost for additional heat exchanger areas (or complete heat exchangers),
  • cost for water treatment plants,
  • cost for additives and
  • eventually the cost for a complete transmission station.

If - on the other hand - the length or certain reference transport cost respectively are assumed the absolute savings in cost can be calculated. The results for the calculation of the „economic length“ according equation 9.7 and of the absolute savings when assuming reference transportation cost of 20 DM/MWh are presented in the following chapter.

### 9.2.3 Results for the possibility „savings in pumping cost“

In figure 9.4 the „economic length“ for a pay off time of 20 years is shown. Considering the results the significant influence of the investment cost of a transmission station is evident. The length that is necessary in case of an already existing transmission station are realistic compared to real systems. Considering the „economic length“ which is necessary if a transmission station has to be built (to realize the application of drag reducing additives) the values are relatively high compared to real systems.

![Fig. 9.4: „Economic length“ for the application „pumping cost reduction“ [29]](image-url)
The two cases shown in figure 9.4 are borderline cases. The conditions of real district heating systems are - in most cases - between the presented borderline cases. This means e.g. an existing separation of transport system from distribution network - and therefore an existing transmission station - but a hydraulic connection between both systems to realize water supply or pressure maintenance. Such an hydraulic connection can not be tolerated because in this case surfactant solution would penetrate from the transport system into the distribution network.

To realize the application of drag reducing additives for this conditions a new pressure maintenance as well as an independent water supply to compensate for water losses has to be installed.

The two considered cases - complete transmission station has to be built/is already existing - are the upper and lower limits. The real conditions are (in most cases) between those limits.

Figure 9.5 presents the results for the assumption that certain reference transportation cost are fixed. The reference transportation cost are 20 DM/MWh for the origin operation with water.

![Figure 9.5: Savings in transportation cost (reference value for water: 20 DM/MWh) [29]](image)

It can be seen that in case of the necessity of the installation of a complete transmission station no savings in cost can be achieved. In the other case maximum savings of about 2 DM/MWh are possible.

In a sensitivity analysis several parameters have been varied to find out which parameters have the most important influence. In [23], [25] and [29] those analysis are shown for different cases of application (drag reduction, increase in capacity and reduction of pipe diameter). The results of the sensitivity study for the case - „savings in pumping cost - the installation of a complete transmission station is not necessary“ - are presented in figure 9.6.

The varied parameters are (the values for the centre are put in brackets):

- $\text{DR}_m$ - the average drag reduction (66 %),
- $f_{ni}$ - share of not influenced pressure losses (0.2),
- $\text{HOR}$ - heat output reduction (30 % - for condensers),
- $\Delta p/\Delta L$ - pressure gradient (109 Pa/m),
• K - investment cost (according to [23] - equation 5.53 in [DM/MWh]),
• \( s_{st} \) - price for electric power (0.18 DM/kWh),
• \( k_T \) - specific transportation cost (20 DM/MWh) and
• \( \Delta V \) - water losses (0.2 \( \cdot V_{net} \)).

The strongest influence show the parameters \( \Delta p/\Delta L \), DR, \( s_{st} \) and \( k_T \). For the assumptions which have been made a negative result only occurs under bad conditions. These are a low pressure gradient, a low drag reduction, low specific transportation cost and a low price for electric power.

**Fig. 9.6:** Results of the sensitivity analysis for the application: savings in pumping cost - the installation of a complete transmission station is not necessary

Under good conditions savings of 1 to 2 DM/WMh can be achieved.

If a transmission station is necessary a positive result can not be achieved - even under favourable conditions. In this case the specific cost of a transmission station of about 2.4 DM/MWh have to be subtracted from the results shown in **figure 9.6**. This leads to negative results in any case.

Additional information can be gathered from [23], [25] and [29].
9.2.4 The cost model for „increasing the capacity“

In contrast to the possibility to save pumping cost (rationalization measure) the increase in supply capacity (or the elimination of bottle-necks) is an extension measure. Necessary condition is a sufficient heat demand. The calculation is also based on the model shown in figure 9.3. Analog to chapter 9.2.2, equation 9.2 has to be fulfilled to achieve a positive economic result. The savings (losses) can be calculated as follows:

\[
r = (f_{incr} - 1) \cdot k_T \cdot \dot{Q}_{max} \cdot T_b \cdot \frac{\pi \cdot D^2 \cdot L \cdot f_{loss} \cdot k_s}{4} \cdot i_{main} \cdot (K - I) - \quad \text{eq. (9.9)}
\]

\[
V_s \cdot f_{loss} \cdot k_{treat} \cdot r \quad \text{[DM/a]}
\]

mit: \( f_{incr} \) = factor which describes the ratio between the flow rate with and without surfactants for constant pumping capacity. It can be calculated with the following equation:

\[
f_{incr} = \left( \frac{V_s}{V_w} \right) = \left( \frac{\xi_{sw}}{\xi_s} \frac{1}{(1 - DR) \cdot (1 - f_{ni}) + f_{ni}} \right)^{1/3} \quad \text{eq. (9.10)}
\]

\( f_{ni} \) describes the part of pressure losses which can not be reduced due to the application of additives (e.g.: elbows, valves etc.). It is defined as:

\[
f_{ni} = \frac{(b - 1)}{b} \quad \text{with } b \equiv 1 + \frac{\Delta L_e}{L} \quad \text{eq. (9.11)}
\]

\( \Delta L_e \) = equivalent pipe length to consider additional flow resistances.

\( \xi_{sw}, \xi_s \) = drag coefficient for water/surfactant solution [1].

\( k_T \) = specific transportation cost of the origin system [DM/Wh]

\( T_b \) = load utilization period of maximum demand [h/a].

The other parameters have to be determined analog to chapter 9.2.2.

In addition to equation 9.2 another condition has to be fulfilled. This is the demand that the application of surfactants is more economic than the installation of new pumps or pumping stations. Otherwise the installation of pumps would be preferred. The savings (losses) due to the installation of new pumps can be calculated with equation 9.12.

\[
r = (f_{incr} - 1) \cdot k_T \cdot \dot{Q}_N \cdot T_b \cdot i_{main} \cdot K - \quad \text{eq. (9.12)}
\]

\[
s_{st} \cdot T_p \cdot \dot{V} \cdot \frac{\Delta p}{\Delta L_0} \frac{L}{\eta_p} \left( \frac{\Delta p}{\Delta L_1} \cdot f_{incr} - 1 \right),
\]

with the index: „0“ for the origin state without increase in capacity and „1“ for the case of increased capacity.
9.2.5 Results for the possibility „increase in capacity“

In figure 9.7 analog to figure 9.4 the economic length - the length of the pipe which is necessary to achieve a positive result according to equation 9.2 - is shown. The two borderline cases „transmission station existing“ and transmission station necessary“ are considered.

![Economic length diagram](image)

**Fig. 9.7:** „Economic length“ for the application „increase in capacity“ [29]

Compared to figure 9.4 the necessary transport length is significant smaller for both cases. If a transmission station is already existing the application of drag reducing additives is economic for a pipe length below 5 km. Real district heating systems (systems with more than ca. 10 MW) fulfil this requirement. Only in a few cases the pipe lengths of real systems are below this value.

If a transmission station has to be built, the economic length for a pay off time of 20 years is increasing considerably. The lengths are between 5 and 35 km. Compared with the results for the application to save pumping cost these values are significantly smaller and are in the range of real system lengths.

In figure 9.8 the necessary pipe length for a pay off time of 20 years for the case „increase in heat capacity“ due to the application of drag reducing additives is compared with an increase in capacity due to the installation of new pumps. Therefore the assumption is made that a transmission station is already existing.

For nominal heat loads below 80 MW the necessary pipe lengths are smaller when installing new pumps. Above 80 MW the conditions are reverse. The economic lengths for the variant „installation of new pumps“ are increasing superproportionally for values above 100 MW and reach unrealistic length above circa 230 MW.

The reason for this behaviour is the dependence of the investment cost from the maximum heat load. This dependence is in case of installing new pumps substantially higher than in case of applying surfactant solutions.
Considering the results in figure 9.8 it has to be taken into account that a smaller economic length is not synonymous with a higher profitability. Systems with pipe length above both curves (the grey range in figure 9.8) fulfill the requirement for both cases - installation of new pumps and application of additives. Which possibility is more economic cannot be seen from figure 9.8. The only range where a concrete comparison is possible is between both curves. E.g. in the range below 80 MW, only the installation of additional pumps is economic.

**Fig. 9.8:** Economic length for a pay off time of 20 years for an increase in heat capacity when using drag reducing additives (transmission station existing) compared with the installation of new pumps

**Fig. 9.9:** Specific transportation cost for an application of drag reducing additives/new pumps; reference value: 20 DM/MWh (for water as heat carrier)
Figure 9.9 shows the transportation cost for an increase in capacity analog to figure 9.5. The possibilities „application of drag reducing additives“ (transmission station existing/ necessary) are compared with the „installation of new pumps“. The reference value for the transportation cost are again 20 DM/MWh.

Due to the application of drag reducing additives an economic result can be achieved in the complete range of nominal heat load. The installation of additional pumps is profitable only for values below circa 230 MW (superproportional dependence of investment cost from nominal heat load).

Analog to figure 9.6 a sensitivity analysis has been carried out for the application to increase capacity. The centre and varied parameters are identical to those described in chapter 9.2.3. For the calculations shown in figure 9.6 the assumption is made that a transmission station is existing. If a transmission station has to be built the corresponding cost (ca. 2.4 DM/MWh) have to be subtracted from the results.

The most significant parameter are the drag reduction, the transportation cost, the pressure gradient and the not influenced pressure losses. If a transmission station has to be installed the investment cost also show a high significance [25].

Fig. 9.10: Sensitivity analysis for the application: increase in heat capacity - the installation of a complete transmission station is not necessary

Comparing figure 9.6 with 9.10 it is evident that only the application for increasing capacity is significantly influenced by the not influenced pressure losses. Reason for this is the direct influence of $f_n$ on the increase in capacity and therefore also on the savings in cost. A significant influence of the water losses is not present. Compared to the other investment the cost due to water losses are small.

If no transmission station has to be installed the application of drag reducing additives is profitable in any case. Under the made assumptions average savings in cost of about 4.30 DM/MWh are possible (centre). Under easy terms, savings of even 6 DM/MWh are possible.
Comparing the „saving of pumping cost“ with the „increase in capacity“ the increase in capacity is a much more profitable possibility to improve the economic viability. The reason for this is the small share of the pumping cost of the total transportation cost. Therefore, only a small part of total cost can be reduced.

### 9.2.6 General results for economics calculation

The results for the most important kinds of application are summarized in figure 9.11. Therefore the borderline case - no installation of a transmission station is necessary - is considered. If a transmission station has to be installed due to the application of drag reducing additives approximately 2.4 DM/MWh have to be subtracted from the presented results.

![Diagram showing savings in cost](image)

**Fig. 9.11:** Average potential in saving cost (the installation of a transmission station due to the application of FRAs is not necessary)

The very promising possibilities are the applications to reduce pipe diameter (and simultaneously the supply temperature), the shifting of heat load and the increase in output capacity.

Concerning the presented results the assumptions which have been made have to be considered. The results show the fundamental range in which the savings in cost can be expected. For completely deviating assumptions a deviation from the presented results probably will occure.

**Figure 9.11** shows that the application of drag reducing additives contains a high potential of saving cost in principle. Due to the fact that distinct heating systems show extremely different boundary conditions, operating conditions, geographical conditions, design etc. an individual examination of any system has to be carried out before the application can be realized.
Project B

The Effect of Surfactants on Domestic Heat Exchangers for Hot Water Supply and Heat Flow Meters in D/H Systems

EV-1670

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1. Introduction

In the electricity supply system in the Netherlands, combined heat and power generation is becoming increasingly important. More and more units are decentralized, and this creates new opportunities for district heating.

An important feature of district heating systems is that the heat needs to be transported over relatively long distances. Research has been under way for many years now, especially in Germany, the U.S.A. and Canada, to develop suitable additives for reducing friction losses in pipelines.

These efforts have been successful in that additives are now available that reduce friction losses by up to 70%.

The amount of friction in the transport system dictates the pipe diameters and the pump power the be installed for heat circulation. The pipelines and the pumps, together with the pump energy, form a major cost item and any reduction achieved here will add to the profitability of district heating. In existing systems, the use of suitable additives allows the flow rate, and so the maximum load, to be increased without any further investment. In addition, such additives can reduce pump losses.

The International Energy Agency (IEA) coordinates an international research and development programme aimed at developing improved additives. It has assigned NOVEM to act as operating agent for the Netherlands.

Habon-G is an additive specially designed for use in the water mains of district heating systems. Since Habon-G reduces turbulence, it reduces not only friction losses but also the heat transfer. These effects can only determined through experiment, as no models are as yet available for theoretical determination. Up until now, only the effect of Habon-G on friction losses and heat transfer has been assessed in large industrial systems and in demonstration projects.

Extensive fundamental research into the effects of additives has been carried out in a doctoral research project at the University of Dortmund, Germany [1]. Chapter 2 of this report is largely based on the results of that project.

Going on where the German work leaves off, the present investigation aims to assess the performance of Habon-G, through measurements, in small district heating systems and the heat exchangers and heat flow meters for the domestic water supply in homes. This investigation was ordered by NOVEM.
2. Fluid flow and friction

The flow condition in a pipe is normally described by means of the Reynolds number Re:

\[ \text{Re} = \frac{v \cdot D}{v}. \quad \text{eq. (2.1)} \]

Depending on the value of Re, the flow is either laminar or turbulent. If conditions in the entry flow region are ignored, a laminar flow is characterized by the absence of fluid exchange perpendicularly to the direction of flow. Turbulence, on the other hand, is characterized by complete mixing in the fluid flow, with the velocity varying in both magnitude and direction. The occurrence of eddies is another remarkable feature.

The resistance to flow in a pipe can be expressed as the pressure difference per unit pipe length using the dimensionless friction factor \( \xi \)

\[ \xi = \frac{\Delta p}{L \cdot \rho \cdot v^2}. \quad \text{eq. (2.2)} \]

For laminar flow, at Reynolds numbers not exceeding 2320, the friction factor is independent of the roughness of the pipe wall and equals:

\[ \xi = \frac{64}{\text{Re}}. \quad \text{eq. (2.3)} \]

![Friction factor for hydraulically smooth and technically rough pipes](image-url)
At high Reynolds numbers the flow is turbulent, in which case it is no longer possible to theoretically determine the friction factor. If the flow passes the transition region between laminar and turbulent, the friction factor obeys the following formula, which was derived by Prandtl and improved by Nikuradse through experiment. This formula is valid only for smooth pipe walls.

\[ \frac{1}{\sqrt{f}} = 2 \cdot \log (Re \cdot \sqrt{f}) - 0.8. \quad \text{eq. (2.4)} \]

Both formulas are graphically represented in figure 2.1.

2.1 Possible additives

For many years now, efforts have been made to reduce friction losses in pipelines. Apart from using smooth-wall pipes, two options are available, i.e. wall dampening and the use of additives.

In wall dampening, the pipe is lined with either a smooth, elastic and dampening coating or a highly elastic liner.

A wide variety of products, mostly polymer-based, are available or under development for reducing the friction losses. Polymers, however, because of their high thermal sensitivity, generally are less suitable for use in district heating systems.

Another class of additive are surfactants. These are surfactants, such as soap, with a relatively low molecular weight. Under certain conditions, surfactants form micelles (small clusters) that have a friction-reducing effect. They differ from polymers in that the micelles, but not the surfactants, degrade under heavy mechanical loads. This degradation mechanism is reversible - the micelles are restored on relief of the load.

\[ \begin{align*}
\text{CH}_3 \\
C_{n}H_{2n+1} - N - \text{CH}_3 \\
(C_2H_4O)_{1.2}H
\end{align*} \]

\( n = 16: \text{Habon-G} \quad n = 18: \text{Obon-G} \quad n = 22: \text{Dobon-G} \)

n-Alcyldimethylpolyoxethylammonium-Cations

Fig. 2.2: Chemical structure of surfactants
A number of surfactants developed under the auspices of IEA exhibit increased corrosiveness to metals, for which reason they are less suitable. Three surfactants developed by Hoechst AG neither exhibit such corrosiveness nor degrade to any appreciable extent.

These three surfactants (Habon, Obon, Dobon) were specially developed by Hoechst for district heating service. They have different temperature ranges and, so, different areas of application. Depending on their chemical structure, the upper limit of the temperature range is between 90 and 140 °C at flow velocities up to about 4 m/s. The temperature range of Habon extends to 95 °C. Accordingly, for the purposes of the present investigation, Habon would seem to be most suitable.

The Hoechst surfactants are surfactants comprising a hydrophilic head and a hydrophobic tail consisting of carbon chains. Their chemical structure is shown in figure 2.2.

Friction losses are reduced by preventing turbulent flow from occurring. In this way, friction loss can be reduced by about 80 %. As well as friction loss, the heat transfer is reduced also. Both effects are advantageous in district heating systems except that a lower heat transfer is a disadvantage for the heat exchangers included in such systems.

### 2.2 Operating principle of Habon

A turbulent flow can be broken down into a laminar boundary layer, an intermediate buffer layer and a turbulent main stream flow. The laminar layer is adjacent to the wall and is very thin. It is followed by the buffer layer, where most of the irreversible energy dissipation takes place as a result of the vehement exchange of impulses. The layer at the centre, the turbulent main stream flow, is characterised by strong eddies but accounts for only a small portion of the overall energy dissipation.

![Diagram showing the operating principle of Habon](image-url)
The friction-reducing effect of surfactants in aqueous solutions depends on the formation and the shape of micelles. Micelles are clusters of surfactant molecules, which can take any of a variety of shapes, such as spheres, bars or disks. Their hydrophilic heads form the boundary layer relative to the water whilst the tails combine to fill the micelle volume. Bar-shaped micelles should be used for reducing the friction loss.

If a given concentration is exceeded, the critical micelle concentration (CMC₁), the surfactants will form spherical micelles. This concentration is hardly temperature-dependent. If the concentration is increased still further, the number of surfactant molecules per micelle will increase until the micelle volume is completely filled with carbon chains. When another critical concentration (CMC₁₁) is exceeded, some surfactants, including Habon, produce other micelle shapes, such as bars, because these boundary faces are more favourable energy-wise. The CMC₁ concentration is strongly temperature dependent (see figure 2.3).

Surfactant solutions that form spherical micelles behave in the same way as the solvent, in this case water, even at high concentrations and the viscosity becomes somewhat higher than that of the water.

At Habon concentrations that are only little higher than CMC₁₁, the surfactant solutions in which bar-shaped micelles have formed exhibit a favourable viscoelastic behaviour. Such cells become oriented by the pulse loads of turbulent flow and form a viscoelastic network which expands the buffer layer and reduces the layer of the turbulent main stream flow (figure 2.4).

Fig. 2.4: Viscoelastic network, shear induced structure
2.3 Operating range of surfactants

At the maximum reduction in friction loss that can be attained with polymers or surfactants - in other words, when the laminar buffer layer extends to the centre of the flow - the friction factor may be determined using Virk's equation [1]:

\[
\frac{1}{\sqrt{\varepsilon}} = -16.2 + 9.5 \cdot \log\left(\frac{Re \cdot \sqrt{\varepsilon}}{2}\right).
\]

(2.5)

If the flow is within the operating range of the surfactant, the friction loss of the (aqueous) surfactant solution is almost independent of temperature and concentration. In that case, the friction loss of a solution of Habon in water approaches this theoretical minimum resistance according to Virk's equation. This flow condition is also known as pseudolaminar flow. The figure below shows the range in which the friction factor may vary on addition of Habon. The possible behaviour of such a fluid is given by way of example.

![Behaviour of a surfactant solution](image)

Fig. 2.5: Behaviour of a surfactant solution

Both the upper limit and the lower limit are dependent on the Habon concentration, the temperature, the flow velocity and the geometry.

2.3.1 Lower limit of operating range

First of all, if bar-shaped micelles are to be produced, it is necessary that a certain limit concentration, \( \text{CMC}_H \), be exceeded. The flow will not be completely viscous but rather slightly viscoelastic. This viscoelasticity ensures that the bar-shaped micelles will become oriented on
exposure to pulse loads. When the Reynolds number is increased, the pulse load from turbulence will become so high as to cause the bar-shaped micelles to form permanently oriented networks. These dampen the turbulence and eventually prevent the main stream flow.

If the concentration is only just above \( \text{CMC}_{\text{II}} \), only few, relatively large micelles will be formed. These are not well capable of forming an oriented network, which is why their friction-reducing effect is only small.

### 2.3.2 Upper limit of operating range

The networks are destroyed when a given flow velocity and temperature are exceeded. The upper limit is determined by the mechanical load resulting from friction. These frictional forces exert a shear stress on the networks. When the maximum shear stress of the networks is exceeded they disappear and the flow becomes viscous again.

The maximum shear stress is a function of the temperature and the concentration.

---

**Fig. 2.6:** Critical shear stress as a function of temperature

It is also possible that on a temperature rise, \( \text{CMC}_{\text{II}} \) rises to exceed the concentration used. When that happens the bar-shaped micelles are replaced by spherical ones and the flow turns viscous again.
2.4 Effects of temperature and concentration

To each flow velocity belongs a lower temperature limit at which the solution begins to be effective. It has been shown that the upper temperature limit is dependent on the micelle concentration $CMC_{II}$ and that the effectiveness of Habon rapidly diminishes above this temperature limit.

Fig. 2.7: Effect of temperature on flow behaviour
The length of bar-shaped micelles increases, but their number decreases, with decreasing temperature. When the lower temperature limit is exceeded, the solution is no longer able to build a proper micelle structure from the few large micelles that are available, so that the friction-reducing effect ceases. When the velocity, and, so, the mechanical load, is increased the lower temperature limit becomes higher and the upper temperature limit becomes lower.

**Figure 2.7** shows how the micelle structure, and so the friction-reducing effect, eventually collapses under the mechanical load.

When the temperature is increased the upper limit rises from the operating range to a maximum value. When the temperature is increased still further the upper limit drops to a lower value. At the same time, the friction-reducing effect diminishes, that is, the flow changes from pseudolaminar to attenuated turbulent.

As the figure shows, at low flow loads and high Habon concentrations, an increase in temperature causes the effectiveness in the attenuated turbulent flow range to diminish. Eventually, pseudolaminar flow is attained at higher Reynolds numbers, with significantly lower friction (**figure 2.8**).

An increase in concentration ensures that formation of the effective micelle structure remains possible, so extending the operating range.

**Fig. 2.8:** Effect of Habon concentration on flow behaviour
2.5 Effects of hydraulic diameter and flow velocity

The Reynolds number is not a suitable means of designating the mechanical limit load. All other conditions being equal, different critical Reynolds numbers are found for different hydraulic diameters (whereupon the flow becomes turbulent again).

![Graph: Effect of pipe diameter on flow behaviour](image)

**Fig. 2.9:** Effect of pipe diameter on flow behaviour

The critical Reynolds number increases with increasing hydraulic diameter. At the same time, at lower flow velocities (within the attenuated turbulence area), the turbulence-dampening effect decreases with increasing hydraulic diameter.
2.6 Effect of wall shear stress

In the earlier investigation it was found that the wall shear stress can be used for determining the turning point to designate the thermal and mechanical load limits. Figure 2.10 shows the wall shear stresses for the three friction curves indicated in figure 2.9 as a function of the Reynolds number.

![Wall shear stress vs Reynolds number](image)

**Fig. 2.10:** Wall shear stress vs Reynolds number

The turning point can be identified as a sudden change in shear stress. Both the limit shear stress and the turning point are about the same for all pipe diameters.

2.7 Outlook for present investigation

In the light of the theoretical backgrounds described in the preceding sections an attempt may be made to predict some results. The heat exchangers under test are relatively small in diameter and contain many tube bends. Thus, the internal friction and the flow velocity will be large. Also, the Reynolds number will be high, with completely turbulent flow.

If the internal diameter is known, it is possible to determine with [1] what concentration of Habon G needs to be added in order to change the flow regime in a heat exchanger from turbulent to pseudolaminar. It is also possible, then, to predict by calculation the reduction in friction loss and in heat transfer (see Appendix A).

The following should be noted here.
1. The investigation referred to in [1] largely dealt with isothermal flow whilst the present investigation deals with flow regimes involving a radial and an axial temperature gradient. The radial temperature gradient is especially important in as much as the effective bar-shaped micelles need to be built up from the wall if they want to have any friction-reducing effect. The effect of such a gradient is unknown but may be significant.

2. Some of the measurements in [1] were made on bench-scale systems with all flows fully developed and the hydraulic entry effect specially eliminated in the interest of accuracy. Others were made on district heating systems some kilometers long, which presumably were designed and constructed in such a way as to minimize the entry effect. The present investigation covers four small and compact heat exchangers with thin-wall tubes and a large number of bends. In such heat exchangers, the hydraulic entry effect has a major impact. This being so, it will probably be necessary to add more Habon-G than the calculations in Appendix A suggest in order for pseudolaminar flow to develop within the heat exchangers.
3. Description of test facility

As far as their heat demand is concerned, district heating systems are dependent on weather conditions and the user's need for comfort. This dependence generally changes with the seasons and is manifested by varying water temperatures. In the present investigation, this variation has been compensated for by assuming two particular temperatures that are representative of summer and winter conditions.

The domestic water supply must always be as hot as the user pleases, regardless of the season. The lower temperature limit should be high enough to kill bacteria and dissolve fat whilst the upper temperature limit should not cause excessive scaling or skin burns. In practice, this is accomplished by varying the flow rate of the system water.

In the present investigation, the flow rates of both the domestic water supply and the transport lines in the district heating system were varied in a realistic way.

This is because flow rate variations over as wide as practicable a range yield more results from which to determine the effectiveness of Habon-G.

Domestic water supplies operating in conjunction with a district heating system must meet the following requirements:

- The temperature of the domestic water supply should be between 50 and 60 °C.
- At a flow rate of 5 l/min and an inlet temperature of 10 °C, the domestic water must be able to be heated at least 45 K.
- The return temperature of the system water must be as low as possible, certainly lower than 50 °C.

Given the wide variations in the flow rates of both the domestic water supply and the system water, these requirements have been ignored in the measurements, but they are taken account of in interpreting the measuring results.

The heat exchangers used for domestic water supply come in all sorts of design and are supplied by many different vendors. The same applies to the heat flow meters. Therefore, in consultation with NOVEM, four heat exchangers and heat flow meters of different design were chosen. The vendors are felt to be immaterial and no further details of them are given here.

In order to be able to assess the effect of adding Habon-G, a reference measurement (without addition to the system water) was carried before and after the Habon-G measurements.

An essential parameter, apart from temperature and flow rate, is the concentration of Habon-G. Earlier work [1][2][4] has shown that the effects of Habon-G depend on many factors such as the size and shape of the heat exchanger, temperatures and flow rates. Theoretical considerations cannot be made without making a large number of assumptions and simplifications. As opposed to the other three heat exchangers, the internal dimensions of unit C were known, so enabling broad theoretical calculations (Appendix A). From those calculations and the results of earlier work, a concentration range for Habon-G of 100 -1000 wppm (eight parts per million) was chosen.
3.1 Automatic control

In keeping with general laboratory practice and given the large number of measurements that had to be taken, the test facility was automated wherever practicable. Instruments with analogue 4 - 20 mA inputs and outputs were used where appropriate and linked to a process computer. For data acquisition, the methods of data storage and processing available in the laboratory were opted for.

As a result of these departure points, the inlet temperature of the primary system, the primary flow rate and the secondary flow rate were controlled by a programmable logical controller. A PC was used for controlling the PLC, for sampling the points of measurement and for data acquisition.

3.2 Test circuit

A facility was assembled at the Technical University of Delft enabling the domestic water supply with heat input from a district heating system to be simulated. The flow diagram is shown below. Further details and a photo are given in Appendix B.

![Fig. 3.1: Flow diagram of test circuit](image_url)

The heat exchangers are connected to headers having a larger diameter than the mains. This makes it easier to connect or replace any number of heat exchangers and reduces the number of instruments needed. The temperature in, and the pressure difference between, the headers are representative of all heat exchangers inserted between them. Thus, only one thermometer and differential pressure gauge were needed.
The primary circuit (the district heating section) includes an electric heater rated 20 kW, used for heating the system water, a pneumatic control valve for adjusting the primary flow rate, a circulation pump, a flow meter, two Pt 100 temperature elements (one being placed upstream of the inlet and the other downstream of the outlet of the heat exchanger) a pressure vessel and an automatic vent valve.

The primary circuit was 10.9 m long, with a capacity of 50 litres. An array of ball valves allowed any one of the four heat exchangers and any one of the four flow meters to be inserted into the circuit. A differential pressure gauge was installed across the pump and the two headers in the primary circuit.

A fill vessel was connected to the primary circuit via ball valves. This was used for adding Habon-G and for pressurizing the primary circuit with nitrogen.

Water was fed to the secondary circuit (the domestic water supply section) from a closed booster system with a capacity of 50 m³. The secondary system included a flow meter, two Pt-100 elements, a pneumatic control valve and a hand-operated vent valve.

The primary and secondary circuits, the heat exchangers and the heater were lagged to standards normally applied in the industry [5].

### 3.2.1 Heat exchangers

Four counter-current heat exchangers could be inserted into the primary circuit:

- Heat exchanger A was a compact, single-wall plate-type heat exchanger. The manufacturer's specifications indicated that the maximum hot-water capacity was 5 l/min. The header-to-header length was 1.15 m.
- Heat exchanger B was of a helix-shaped, multiple double-pipe design with a maximum hot-water capacity of 7 l/min (specified by manufacturer). The header-to-header length was 1.90 m.
- Heat exchanger C was of the spiral double-pipe design with a maximum hot-water capacity of 5 l/min. The header-to-header length was 1.40 m.
- Heat exchanger D was a coiled double-wall plate-type unit with a maximum hot-water capacity of 5 l/min. Its header-to-header length was 1.30 m.

### 3.2.2 Heat flow meters

The four heat flow meters were arranged in series just downstream of the heater outlet. Each measured the inlet and outlet temperatures with the aid of Pt-100 elements; these were all connected in the same way wherever possible. Each heat flow meter measured the heat content that passed through it. Three meters (A, C and D) gave readings in gigaJoules and one (B) in MWh. In addition, meters A and B showed the total volume passing through them in m³.

The flow meter of heat flow meter C was of the magnetic-inductive type; those of the other heat flow meters were of the mechanical or mechanical-inductive type.
The percentage error of the heat flow meters depends on the accuracy with which the temperatures and flow rates are measured. In most flow meters the accuracy decreases with decreasing flow, so that the error of the heat flow meter, too, will decrease with decreasing flow.

In the present investigation, no attempt has been made to establish the relation between the flow and the errors, because the heat flow meters indicated the energy levels in too large quantities (MJ or kWh). This would make any such attempt extremely time-consuming.

### 3.2.3 Habon-G concentration

The Habon-G was supplied by the client as a concentrated liquid. It is viscous at room temperature with poor miscibility with water. As a result, as appeared in a number of preliminary experiments, it is difficult and takes long to prepare a liquid having the desired concentration. Also, Habon-G is known to readily oxidize. Therefore, a closed fill system with a nitrogen connection was added to the system.

### 3.3 Measuring programme

The measurement setting is dictated by three parameters: the primary inlet temperature, the primary flow rate and the secondary flow rate.

Measurements were conducted at primary inlet temperatures of 70 and 90 °C so as to simulate summer and winter conditions. The two flow rates were varied over as wide as practicable a range: from 4 to 12 l/min on the primary side and from 2 to 8 l/min on the secondary side, both increasing in increments of 2 l/min. These set points yield a measuring programme comprising 40 settings.

The number of attainable settings is limited, however, by the pump capacities on the primary and the secondary side and the power rating of the electric heater. The setting with a secondary flow rate of 8 l/min, for instance, was not always attainable because either the pump power or the heater power was inadequate to maintain the primary inlet temperature with water being withdrawn at that rate. In the end, because of these constraints, a suitable measuring programme was devised comprising 28 different settings.

### 3.4 Control

The system was to be in equilibrium if correct measuring results were to be obtained. A preliminary study [6] indicated that true equilibrium was made impossible by the noise, which interfered with the setting in various ways. External factors such as the environmental climate, other users of the water supply booster and the stability of the measuring instruments all affected the
readings. In order to suppress these effects as much as possible it was decided to control, and take samples from, the system by means of a PLC. A PC operating in tandem with the PLC took care of the controller set points and data storage.

The system exhibited a non-linear behaviour because the heat transfer to the surrounding area and to the secondary system was not proportional to the primary temperature. On changing the set point, it took from 25 minutes to two hours until a fairly stable condition was attained. Control performance was improved by changing the control action (PI or PID) of the controller and entering different control parameters for each heat exchanger and each setting in the measuring programme.

It was decided to adjust the controller once and for all to values believed to be optimum for all four heat exchangers.

### 3.5 Accuracies

The accuracy of the measurements was limited by two factors:
- The finite accuracy of the measuring instruments and
- Systematic errors.

Given the importance of the measuring accuracy in the present investigation, a number of preliminary studies were carried out. These are discussed below.

#### 3.5.1 Finite accuracy of measuring instruments

The accuracies as specified by the manufacturer are listed in Appendix C. As can be seen, the accuracy of each instrument is equal to or better than 1 % of full-scale deflection. Some experiments were conducted for a more accurate determination of these accuracies:

- **Temperature sensing elements**
  
  Since the temperature differences across the inlet and outlet of a circuit are much more important here than the temperatures *per se*, a preliminary study [6] was made to assess the relative accuracy of the temperature sensors. This was done by simultaneously submerging the sensors pair by pair in a water-ice bath. The results are given in Appendix C.

- **Flow meters**
  
  Earlier work [1] had shown that Habon-G has an adverse effect on the accuracy of certain types of flow meter. Flow meters of the magnetic-inductive type have appeared not to be affected by Habon-G so that this type was used in both the primary and the secondary circuit without any further investigation on our part.
3.5.2 Systematic errors

The systematic errors can be broken down as follows:

- **Stability of the setting**

  The controllers mentioned earlier ensure optimum stability of the system at a particular setting. However, the primary inlet temperature was found to slightly oscillate with an amplitude of about 0.15 K and at a frequency of about 0.0006 Hz (T_{cycle} approx. 30 minutes). This proved to be independent of the set values and the control parameters. This condition arose only some time after the set points for the primary temperature and primary flow rate and the secondary flow rate were adjusted. This delay is due to the large (heat) buffer capacity of the system.

  Preliminary study [5] showed it is safe to assume that a stable condition is attained when the deviation of the primary inlet temperature is less than 0.1 K and the rate of change of the primary inlet and outlet temperatures is less than 0.05 K/min. The 10-minute measurement is started when these requirements are met.

- **Heat losses to the surrounding area**

  Because of the design of the headers to which the heat exchangers are connected, the temperature sensors may be placed rather far away from the heat exchangers. The amounts of heat delivered on the primary side and taken up on the secondary side that are calculated from this temperature measurement are different. A preliminary study [5] indicates that this difference is too large to ignore. Indeed, it constitutes a sizeable error because it is not exactly known where these "losses" occur. However, the measured values can well be corrected if the heat losses are assumed to be proportional to the pipe length and the temperature difference between the pipeline and the surrounding area. The correction used in this investigation is detailed in Appendix E.

- **Differences in secondary inlet temperature**

  These differences are not really an error but do distort the basis on which the measured values are compared. With the aid of the thermal efficiency, however, the values can be converted to a fixed inlet temperature. This conversion is given in Appendix G.

- **Pressure drop across heat exchangers**

  Because of the header design, the measured pressure drop equals the pressure drop across the heat exchanger plus the pressure drop across the connections to and from the headers. Some heat exchangers had more bends because of the various connection possibilities. This may cause the measurements to yield a distorted picture. However, since these additional bends result from the geometry and the construction of the heat exchanger, the connections have been regarded as forming an integral part of the heat exchanger.
Systematic errors

- **Fouling in the test facility**

  As time went by, the heat transfer and the pressure drop across a heat exchanger may have been affected by fouling in the tubes. This effect is not likely to have been appreciable given the duration of the investigation and the use of Habon-G. All the same, this effect will be assessed by conducting a second reference measurement without Habon-G after all measurements have been taken.

- **Degradation of Habon-G**

  Habon-G readily oxidizes. Thus, any oxygen entering the test facility would affect the measurements. Therefore, it was decided to use demineralized water as the system water. It was circulated through the system for some days and at elevated temperature so as to rid the system of oxygen. The concentration of Habon-G was measured at regular intervals.
4. Results of measurements

Besides the parameters that dictate a setting (the primary inlet temperature and primary flow rate and the secondary flow rate), the following parameters were measured:

- the primary outlet temperature
- the secondary inlet and outlet temperatures
- the pressure drop across the heat exchanger and the pressure rise across the pump
- the reading of the heat flow meter.

The secondary inlet temperature is dependent here on a number of external factors such as other uses of the water supply booster. The other parameters are dependent on the setting. The settings and the measured values were stored in a computer because of the accuracy and further processing of the raw data.

Once stable conditions were established in the facility, the raw data were stored for ten minutes and processed to obtain a single mean value per measuring period. These values are given in Appendix H and are rearranged in this chapter as follows for ease of interpretation.

- Habon-G concentration
- heat exchangers
- heat flow meters
- pressure measurements
- reference measurements

The example given in Appendix I shows how the measured values have been further processed and corrected.

4.1 Habon-G concentration

Starting from three concentrations between 100 and 1000 wppm, attempts were made to take measurements at 100, 500 and 1000 wppm. In all instances, the calculated required amount of Habon-G was added from the fill system. Representative samples were taken only after extensive mixing and heating. The samples were then analysed by the method described in Appendix D. The average Habon-G concentration for at least five samples are as follows.

<table>
<thead>
<tr>
<th>measurement</th>
<th>concentration of Habon-G [wppm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>concentration 1</td>
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</tr>
<tr>
<td>concentration 2</td>
<td>520</td>
</tr>
<tr>
<td>concentration 3</td>
<td>1470</td>
</tr>
</tbody>
</table>

*Tab. 4.1: Average Habon-G concentrations*
In view of the deviation between the 'set' concentration and the measured concentration, a large number of samples taken for a particular concentration at different points in the measuring period were analysed so as to check the analysis.

4.2 Heat exchangers

As mentioned earlier, the measured values needed to be corrected because of the heat losses (Appendix E) and converted to a fixed secondary inlet temperature (Appendix G). This was necessary to bring the measured values to comparable values that are representative of the heat exchanger performance. The results for each heat exchanger are presented in two different ways:

- In order to be able to assess the effect of Habon-G on heat exchanger performance the power transferred in each heat exchanger has been plotted against the reference measurement and the highest habon-G concentration at the various secondary flow rates.
- Since the secondary outlet temperature is of prime concern to the consumer whilst the primary outlet temperature is of prime concern to the operator, these two temperatures have been plotted for a number of measuring points that are relevant to the domestic water supply. These points were chosen with the following considerations in mind:
  - The design of domestic water supply systems assumes that the user will turn the hot-water tap wide open (about 5 l/min) and will then add cold water until the desired temperature is reached. That is why the secondary flow rates of 2 and 8 l/min were dropped as being representative of the requirements in a district heating system.
  - Assuming tap water needs to be heated 45K during winter, the primary and the secondary flow rates need to be about equal. In summer, however, the primary flow rate will need to be about twice as large as the secondary flow rate.

In the light of these considerations, the measuring points Nos. 8, 15, 23 and 26 in the measuring programme are considered relevant. No. 14 has been included in the presentation inasmuch as, in practice, optimization will take place at this particular point.
4.2.1 Heat exchanger A

The primary and the secondary outlet temperature of heat exchanger A are plotted below against the five settings and against various measured Habon-G concentrations.

![Graph showing outlet temperatures of heat exchanger A](image)

Fig. 4.1: Outlet temperatures of heat exchanger A
Results of measurements

Plotted below are the power levels transferred at various secondary flow rates against the reference measurement and the highest Habon-G concentrations that were measured.

![Graph showing power levels transferred at various secondary flow rates](image)

**Fig. 4.2:** Power levels transferred at various secondary flow rates
4.2.2 Heat exchanger B

The primary and the secondary outlet temperature of heat exchanger B are plotted below against the five settings and against various measured Habon-G concentrations.

Fig. 4.3: Outlet temperatures of heat exchanger B
Results of measurements

The power levels transferred in heat exchanger B are plotted below against the reference measurement and the highest Habon-G concentration.

*Fig. 4.4:* Power levels transferred in heat exchanger B
4.2.3 Heat exchanger C

The primary and the secondary outlet temperature of heat exchanger C are plotted below against the five settings and against various measured Habon-G concentrations.

Fig. 4.5: Outlet temperatures of heat exchanger C
The power levels transferred in heat exchanger C are plotted below against for the reference measurement and the highest Habon-G concentrations.

Fig. 4.6: Power levels transferred in heat exchanger C
4.2.4 Heat exchanger D

The primary and the secondary outlet temperature of heat exchanger D are plotted below against the five settings and against various measured Habon-G concentrations.

![Heat Exchanger D](image)

**Fig. 4.7:** Outlet temperatures of heat exchanger D
The power levels transferred in heat exchanger D are plotted below against for the reference measurements and the highest Habon-G concentration.

![Graph showing power levels transferred in heat exchanger D](image)

**Fig. 4.8:** power levels transferred in heat exchanger D
4.3 Pressure drop measurements

4.3.1 Heat exchangers

The figures below show the effect of Habon-G on the average decrease in pressure drop across each heat exchanger in relation to the reference measurement.

**Fig. 4.9:** Pressure drop over heat exchangers A & B
Results of measurements

Fig. 4.10: Pressure drop over heat exchangers C & D
4.3.2 Pump delivery head

The pump delivery heads shown below have been derived from the measured values. The pressure is the average pressure measured throughout the programme across each of the heat exchangers at 70 and 90 °C.

Arranging the data according to temperature, it appears that at 70 °C pump performance diminishes by max. 1.3 % and improves at the highest concentration by max. 2.6 %. At 90 °C, pump performance diminishes at the first concentration by max. 1.6 % and rises to the original level at higher concentrations.

4.4 Heat flow meters

Measurements on the heat flow meters were conducted simultaneously with those on the heat exchangers. Thus, the results given in Appendix J are the averages of the values found in the measuring programme, during which the flow rate was varied between 4 and 12 l/min.

The values measured by the PLC/PC needed to be corrected since the temperature sensors of the PLC were located at some distance from the sensors of the heat flow meters. For that reason, the results of the energy measurements by the heat flow meter and the PLC were inherently different. The method of correction is based on the assumption that the heat loss is proportional to the pipe length and the difference between the metal temperature and ambient. The method of calculation is described in Appendix F; one of the measurements is worked out in Appendix I by way of example.
4.4.1 Flow meters

Flow meters A and B indicated not only the energy levels but also the volumes passing through them. The figure shown below gives the accuracy of the measurement of flow of these two heat flow meters at various Habon-G concentrations.

![Accuracy of Heat Meters](image)

**Fig. 4.12:** Accuracy of heat meters (flow measurement)
4.4.2 Heat flow measurements

When using system water without additives, i.e. demineralised water, the percentage error of the heat flow meters ranges from less than 1% to, occasionally, 7%.

The results for various Habon-G concentrations are summarized below.

![Accuracy of Heat Meters](image)

**Fig. 4.13:** Accuracy of heat meters (energy measurement)

4.5 Reference measurements

Initially, all measurements were taken with demineralized water being used as system water, without any addition of Habon-G. The results of these measurements serve as a reference for those taken with addition of Habon-G.

On completion of all Habon-G measurements the reference measurements were repeated for three heat exchangers. This was done on the one hand to determine the repeatability of the measurements and on the other to estimate the fouling from rust formation, if any.

The raw data from the second reference measurement is to be found in Appendix K. This data, too, has been adjusted to take account of heat losses (as described earlier) and converted to the same secondary inlet temperature. The adjusted primary and secondary outlet temperatures so obtained have been compared with the results of the first reference measurement. The discrepancies found are summarized as follows:
Secondary outlet temperature: mean difference: +0.3% = 0.2 K, standard deviation: 0.8% = 0.4 K and maximum values: -1.5 to 1.9%.

Primary outlet temperature: mean difference: +0.5% = 0.2 K, standard deviation: 1.0 = 0.5 K and maximum values: -4.9 to 0.7%.
5. Interpretation

5.1 Comparative evaluation of heat exchanger performance in the reference measurements

Important parameters of heat exchangers in district heating systems are the power transferred, the primary return temperature and the pressure drop. The figures below show the results of the reference measurements for these parameters.

![Graph showing exchanged heat](image)

**Fig. 5.1:** Exchanged heat
Fig. 5.2: Pressure drop over heat exchangers

Fig. 5.3: Primary outlet-temperature (secondary flow = 4 l/min)
The return temperatures and heat transfer are practically the same for all heat exchangers, despite their different designs, geometries and prices. The differences in friction loss are substantial, mostly because of the different designs.

Heat exchanger C gives the best performance in terms of heat transfer (during the reference measurements), closely followed by unit A. Units D and B give about the same performance. All heat exchangers amply meet the requirement that the return temperature must be less than 50 °C on heating from 10 to 55 °C and at a secondary flow rate of 5 l/min.
5.2 Effect of Habon-G on flow regime and heat transfer

Considering the effects of Habon-G as observed in earlier work [2], i.e. laminarization of the flow at higher Reynolds numbers, one might expect the flow to pass a transition point at which, above a particular primary flow rate, the pressure drop and the transferred power strongly increase. These two effects are compared below. No transition region is observed.

**Heat exchanger A**

**Reduction of pressure drop**

<table>
<thead>
<tr>
<th>Concentration (ppm)</th>
<th>Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>119</td>
<td>70</td>
</tr>
<tr>
<td>520</td>
<td>70</td>
</tr>
<tr>
<td>1470</td>
<td>70</td>
</tr>
<tr>
<td>119</td>
<td>90</td>
</tr>
<tr>
<td>520</td>
<td>90</td>
</tr>
<tr>
<td>1470</td>
<td>90</td>
</tr>
</tbody>
</table>

**Decrease in transferred heat**

![Graph showing reduction of pressure drop and decrease in transferred heat](image)

**Fig. 5.5:** Reduction of pressure drop & decrease in transferred heat (heat exchanger A)
Effect of Habon-G on flow regime and heat transfer

Heat exchanger B

Reduction of pressure drop

<table>
<thead>
<tr>
<th>Concentration</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>119 wppm</td>
<td>70 °C</td>
</tr>
<tr>
<td>520 wppm</td>
<td>70 °C</td>
</tr>
<tr>
<td>1470 wppm</td>
<td>70 °C</td>
</tr>
<tr>
<td>119 wppm</td>
<td>90 °C</td>
</tr>
<tr>
<td>520 wppm</td>
<td>90 °C</td>
</tr>
<tr>
<td>1470 wppm</td>
<td>90 °C</td>
</tr>
</tbody>
</table>

Decrease in transferred heat

<table>
<thead>
<tr>
<th>Concentration</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>119 wppm</td>
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<tr>
<td>520 wppm</td>
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<tr>
<td>1470 wppm</td>
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<tr>
<td>119 wppm</td>
<td>90 °C</td>
</tr>
<tr>
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<td>90 °C</td>
</tr>
<tr>
<td>1470 wppm</td>
<td>90 °C</td>
</tr>
</tbody>
</table>

Fig. 5.6: Reduction of pressure drop & decrease in transferred heat (heat exchanger B)
Interpretation

**Heat exchanger C**

Reduction of pressure drop

![Graph showing reduction of pressure drop](image)

<table>
<thead>
<tr>
<th>concentr.</th>
<th>temp.</th>
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<tbody>
<tr>
<td>119 wppm</td>
<td>70 °C</td>
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<td>520 wppm</td>
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</tr>
<tr>
<td>1470 wppm</td>
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Decrease in transferred heat

![Graph showing decrease in transferred heat](image)

<table>
<thead>
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<th>concentr.</th>
<th>temp.</th>
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<td>119 wppm</td>
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<tr>
<td>520 wppm</td>
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</tr>
<tr>
<td>1470 wppm</td>
<td></td>
</tr>
</tbody>
</table>

**Fig. 5.7:** Reduction of pressure drop & decrease in transferred heat (heat exchanger C)
Effect of Habon-G on flow regime and heat transfer

At the highest concentration, the power transferred in three heat exchangers decreased by 7% relative to the reference measurements. This was at 70 °C and a primary flow of 4 l/min. On increasing the flow to 6 l/min, this value dropped in two heat exchangers to approx. 3% and did not drop any further at higher flows. This decrease might be indicative of a laminar/turbulent transition point. This change does not occur, however, at 90 °C. Apparently, this temperature lies outside the operating range of Habon-G at the concentrations used.
Interpretation

The pressure drop across the heat exchangers changes when the heat transfer changes. However, no clear relation could be established that might indicate the existence of a transition point.

The maximum decrease in heat transfer lies at the border of the practical area of application of heat exchangers in a district heating system. It does not constitute any impediment, since the primary flow rate can be increased without the primary return temperature exceeding 50 °C.

Calculations made on one of the heat exchangers indicate that, under the test conditions, the flow in it is turbulent (33,000 < Re < 80,000) but that it is within the region where it should have been influenced even by low Habon-G concentrations. Apparently, a laminar flow is prevented by the "hydraulic entry effect". The path length of the undisturbed flow is too short for laminar flow to develop, given the many bends and twists in heat exchangers. This is a deliberate choice made by the designer so as to produce a compact unit and to increase the degree of turbulence and so the heat transfer.

5.3 Effect of Habon-G on pump curve

The performance of, i.e. the pressure rise across, a pump depends on the viscosity of the fluid, which in turn depends on the fluid temperature. Thus, in determining the differences in pump performance, the fluid temperature had to be the same during all measurements.

In the test facility, the pump was placed next to the heat exchangers. Accordingly, the fluid temperature in the pump was the primary outlet temperature. This temperature is strongly dependent on the secondary flow rate. The plotted values are the averages for each heat exchanger of all secondary flow rate measurements at a particular primary flow rate. Thus, differences in the primary return temperatures have been averaged for these measurements. The viscosity was expected to be comparable therewith at the plotted values.

The pump curve appeared to be hardly affected by the presence of Habon-G. The figure given elsewhere shows that the pump performance decreases slightly at the first two concentrations but that it increases at the highest concentration. At no time were the differences in pump performance greater than 3 %; in actual fact, all percentage figures were below the threshold, or within the dead space, of the meters.

5.4 Effect of Habon-G on heat flow meters

The figures in section 3.4 indicate that the accuracy of the flow meter in heat flow meter A is good (better than 1 %). The accuracy of the flow meter in heat flow meter B is disappointing and is affected by Habon-G. As regards the measurement of the heat content, the following is noted:

- The results for heat flow meter A are remarkable in that the flow measurement is accurate but the overall percentage error is disappointing. Apparently, the temperature measurement is inaccurate.
• The accuracy of the flow measurement in heat flow meter B is about 3%. The accuracy diminishes in the presence of Habon-G. The overall accuracy ranges from 3 to 9%.

• In the case of heat flow meter C (which incorporates a magnetic-inductive flow meter), the errors that were measured are below the threshold. No change in percentage error was observed on adding Habon-G.

• Heat flow meter D is somewhat less accurate than C, but here again the error is below the threshold. Its accuracy is not affected by Habon-G in any material way.

5.5 Repeatability of measurements

Comparison of the first series of measurements (using demin. water without additives) and the last series of reference measurements indicates that, on average, the differences in outlet temperature are less than 0.5% (= 0.4 K). Thus, the repeatability is satisfactory. Given that there has been a slight improvement rather than a deterioration, there has not been any serious fouling.

The repeatability of the concentration measurement has also been assessed (Appendix D). The standard deviation after repeated titration is less than 2 wppm.
6. Diligence, conclusions and recommendations

The care exercised in the present investigation will be evident from a number of sections in this report and from the results presented. The repeatability figures should be proof of the reproducibility of the raw data whilst the correction and conversion methods given in the appendices render the results legitimate and suitable for meaningful comparison.

Extraneous influences have been accounted for suitably and reliably by the methods of measurement, correction and conversion.

The test facility used for the investigation was operated for 1200 hours. Over that period, a total of 12 MWh of electrical power was consumed; the volumes passed through the primary and secondary circuits totalled 450 and 200 m$^3$, respectively. The measurements described herein lasted 700 hours in all. The remaining time was taken up by testing of parts and components and the assembly, the preliminary experiments and setting of controllers.

The results of the reference measurements indicate that:

- The heat exchangers give practically the same performance as far as power transfer is concerned.
- The friction losses across the heat exchangers show major differences.
- Two of the heat flow meters are unreliable.

The measurements with Habon-G added indicate that:

- The concentration of Habon-G is difficult to adjust.
- The heat transfer in the heat exchangers slightly deteriorates (not more than 2% on average), but without impairing proper performance.
- The addition of Habon-G has no appreciable effect on friction losses in the heat exchangers.
- Pump performance is not affected by adding Habon-G.
- Two of the four heat flow meters are sensitive to the use of Habon-G.

6.1 Recommendations for further investigation

Given the importance of the temperature measurements and of proper interpretation of their results, it would seem useful to measure the temperatures nearer to the heat exchangers. The temperature sensors could be placed at the inlet and outlet of each heat exchanger. In that case, the method of correction now used to compensate for the distance between the heat exchangers and temperature sensors could be omitted. Such a modification would be fairly drastic, would reduce the flexibility of the installation and would call for more sensors.

Prior thereto, the accuracy of the measuring results could be improved by basing the method of correction on a logarithmic co-current heat dissipation rather than the linear (average) heat dissipation now used and by applying an iterative method for correction of the outlet temperatures.
Changes are most distinct at the highest Habon-G concentrations. A higher concentration may be needed in this type of small-scale installations because the hydraulic entry effect is more pronounced. This makes it more difficult for laminar flow to be attained. The highest concentration that was tested might coincide with the lower limit of the range in which Habon-G is effective. This ought to be established in a follow-up investigation.

As to the percentage error of the heat flow meters it would be useful to initiate long-term tests one aim of which should be to determine the deviations now found in relation to the volume passing through.
7. Summary

This report discusses the results of a NOVEM-supervised investigation aimed at assessing in how far the surfactant Habon-G can reduce friction losses in domestic water supply systems utilizing heat from district heating systems. Earlier work has shown that Habon-G, when added to heat transport systems, has a beneficial effect on the required pump capacity. Habon-G produces a laminar flow, and this reduces not only the friction losses in pipelines but also the heat transfer.

This report discusses the effect of Habon-G on both parameters as observed in four different heat exchangers and heat flow meters used for domestic water supply systems.

For the purposes of the present investigation a test facility was made in which the district heating circuit is simulated by a closed loop which included an adjustable electric heater. The domestic water circuit was connected to a high-capacity water supply booster. The heat exchangers and heat flow meters were integrated in the facility and could be inserted one at a time into either circuit by means of ball valves.

Measurements made on the heat exchangers indicate that all four heat exchangers meet the specified requirements, although there are some differences. The flow in them still seems to be turbulent, despite the laminising effect of Habon-G.

The heat flow meters do not all meet the requirements. Two were found to be at error and their inaccuracy increased with increasing Habon-G concentrations. The percentage errors of the other two meters were below the threshold and were not affected by the presence of Habon-G.

As observed earlier, adding Habon-G to the water in district heating systems has a beneficial effect on the flow resistance and reduces the heat losses in the feed lines. The present investigation shows that addition of Habon-G does not affect the heat transfer in the domestic water supply system in any material way. However, a critical assessment of the type of heat flow meter to be used is called for.
# 8. Symbols

<table>
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<th>Unit</th>
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<tr>
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<td>concentration</td>
<td>[wppm]</td>
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<tr>
<td>(c_p)</td>
<td>heat capacity</td>
<td>[J/kgK]</td>
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<tr>
<td>D</td>
<td>diameter</td>
<td>[m]</td>
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<tr>
<td>L</td>
<td>length</td>
<td>[m]</td>
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<tr>
<td>(\tau)</td>
<td>shear stress</td>
<td>[Pa]</td>
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9. Literature

[1] Fankhünel, M.:  
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Bruun & Sorensen Energiteknik AS, Risskov, Danmark

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TU-Delft, WbMT, EV 1637, Delft, 1992

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TU-Delft, WbMT, EV-1630, Delft 1992

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[8] Feijen, J.J.:  
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Nijgh en Van Ditmar, druk 2, 1990
10. APPENDIX A
Calculation of Habon-G Concentration

The method cited in [1] is used to determine the concentration needed to influence the flow condition at various primary flow velocities.

The temperature is assumed to be constant as the flow passes through the heat exchanger. If the Habon-G concentration is high enough to influence the flow condition at that inlet temperature, it is also high enough to have an effect at lower temperatures elsewhere in the heat exchanger.

The figure on the next page showing the maximum shear stresses has been derived from experiments at constant temperature. It is not clear in how far this information is valid for non-isothermal flows.

Formulae used:

\[ v = \frac{Q}{A} = \frac{Q}{\frac{1000}{1000}} \left( \frac{1}{60} \right) \left( \frac{1}{\pi} \right) \left( \frac{1}{D^2} \right) \text{ [m/s]}, \] Gl. (10.1)

\[ Re = \frac{v \cdot D \cdot \rho}{\eta}, \] Gl. (10.2)

\[ \frac{1}{\sqrt[\xi]{\xi}} = 2 \cdot \log (Re \cdot \sqrt[\xi]{\xi}) - 0.8 \text{ and } \] Gl. (10.3)

\[ \tau_w = \frac{\rho \cdot v^2}{8}, \] Gl. (10.4)

Knowns: \[ D = 10.35 \text{ [mm]}, \]
\[ \eta = 0.40 \cdot 10^{-3} \text{ [Pa-s] (70 °C)} \text{ and } \]
\[ \eta = 0.35 \cdot 10^{-3} \text{ [Pa-s] (90 °C)} \]
APPENDIX A Calculation of Habon-G Concentration

Solvent: demin. water, pH 7
Additive: Habon
Method of operation: isothermal

Course of $\tau_{w,LM} (T_m)$

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<tr>
<th>$c_A$</th>
<th>Symbol</th>
<th>$D_i$ [mm]</th>
</tr>
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<td></td>
<td>8</td>
</tr>
<tr>
<td>125</td>
<td></td>
<td>15</td>
</tr>
<tr>
<td>250</td>
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<td>25</td>
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<td>500</td>
<td>△</td>
<td>50</td>
</tr>
<tr>
<td>1000</td>
<td>▼</td>
<td>100</td>
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</table>

Fig. 10.1: Critical wall shear stress of a Habon/water mixture vs temperature and concentration [1]

<table>
<thead>
<tr>
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Tab. 10.1: Operating conditions
APPENDIX B Measurement setup

Fig. 11.2: Photo of the test rig
# 12. Appendix C

## Measurement plan

<table>
<thead>
<tr>
<th>Point</th>
<th>Primary flow [l/min]</th>
<th>Secondary flow [l/min]</th>
<th>Primary entrance-temperature [°C]</th>
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<tr>
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<td>8</td>
<td>90</td>
</tr>
</tbody>
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Tab. 12.1: Measurement plan
13. APPENDIX D
Accuracy of measuring instruments

- Flow meters
  - Picomag II magnetic-inductive type, supplier: Endress & Hauser of Naarden
    - primary circuit: range 0-30 l/min, accuracy 1 %
    - secondary circuit: range 0-12 l/min, accuracy 1 %

Differential pressure gauge
- 843 dP cell, supplier: Foxboro Nederland of Soest
  - heat exchanger: range 0-750 mbar, accuracy 0.25 %
  - pump: range 0-750 mbar, accuracy 0.25 %

Temperature sensors
- Pt-100 elements, supplier: Thermo-Electric B.V. of Pijnacker
  - primary circuit: range 0-100 °C, accuracy 0.1 K
  - secondary circuit: range 0-100 °C, accuracy 0.1 K

Relative accuracy of Pt-100 (in comparison with one another) measured by submersion pair by pair in water/ice bath:
  - Pt-100 elements in primary circuit: < 0.03 K
  - Pt-100 elements in secondary circuit: < 0.04 K

These figures amply meet the requirements.
14. APPENDIX E
Measurement of Habon-G concentration by the titration method

According to information from the supplier of Habon-G, concentrations of Habon-G in water in the range we are dealing with here can be measured by two analytical methods. One is based on infrared technology using an IR-Photometer. A major drawback of this method is that it takes long to calibrate the equipment. Given the small number of samples to be analysed, we therefore preferred to use the other method, a two-phase titration technique. That method, too, requires a competent operator but is not difficult for any laboratory analyst.

The titration technique

The method involves two steps: a sample is divided into two phases, one above the other, with the aid of a number of liquids, one of which is an indicator solution.

In the second step, a titration solution is slowly added until the lower phase in the sample reaches a transition point. The amount of titration solution used is a measure of the concentration.

The indicator solution

This solution is prepared by dissolving in a 250-ml graduated flask 0.4 gramme of dimidiumbromide (3.8-diamino-5-methyl-6-phenylphenantridinium bromide) and 0.2 gramme of disulfin blue in 250 ml of 10-% ethanol/water (2.5 ml of ethanol in 247.5 ml of distilled water).

As the solution is light-sensitive, it should be stored in homogenised condition in a brown bottle.

The titration solution

The titration solution is prepared by dissolving 1.442 grammes of sodium laurylsulphate in some distilled water in a 1000-ml graduated flask. Foam develops during the dissolution process, which is eliminated by adding 60 ml of n-butanol and a few drops of concentrated sulphuric acid. The flask is topped up with distilled water to 1000 ml.

Procedure

The concentration of a surfactant in a solution is determined by transferring 20 ml of the solution into a sealable 250-ml graduated flask togher with 30 ml of chloroform and 10 ml of indicator solution. On brief shaking, this mixture divides into two phases: a non-polar upper phase and a polar lower phase. Next, sulphuric acid should be added with a pipette, while constantly slewing the flask, until the upper phase turns light brown and the lower phase a greenish blue.
The titration is carried out by adding a small, known amount of titration solution to the solution. This should be repeated, while constantly shaking, until the lower phase strongly discolours to light pink. The amount of titration solution added until that happens is a measure of the surfactant concentration in the sample. The concentration can finally be calculated using the following formula.

\[
c_{\text{Habon-G}} \text{[wppm]} = A \text{[ml]} \cdot M_{\text{Habon-G}} \cdot 0.25,
\]

with \( M_{\text{Habon-G}} = 524 \text{ gram/mole} \).

**The measurements**

In order to have some certainty as to the reliability of the analytical method and its procedure, we took a large number of samples at the first concentration and at different points in time. The results of the titrations are tabulated below.

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Tab. 14.1: Concentration measurements

Mean concentration = 119 wppm (deviation \( \sigma = 1.45 \text{ wppm} \)
15. APPENDIX F

Summary of heat losses

The overall heat loss to the surrounding area is defined as the difference between the primary power, that is, the power determined from the two Pt-100 elements and the flow meter in the primary circuit, and the secondary power (ditto in the secondary circuit).

A distribution key has been devised on the basis of the temperature difference with the surrounding area and the pipe length. The assumptions made are realistic inasmuch as the amount of heat transmitted through the pipe wall is largely determined, or limited, by the lagging (some centimetres, $\lambda = 0.04$ W/mK) and natural convection ($\alpha_{\text{outside}} = 25$ W/m$^2$K) and to a much lesser degree by the steel pipe wall (some millimetres, $\lambda = 50$ W/mK) the flow velocity ($\alpha_{\text{inside}} = 15$ kW/m$^2$K, possibly affected by the Habon-G concentration. Other compounding factors such as radiation and thermal leaks from one pipe to another have been ignored. In this way, the overall heat loss is distributed to correct the temperatures.

The overall heat loss is the sum total of the following losses:

- **Loss 1** Primary, upstream of the heat exchanger inlet, between the Pt-100 element and the heat exchanger inlet (pipe length $L_1$, temperature $T_1$)
- **Loss 2** Primary, downstream of heat exchanger outlet, between the heat exchanger and the Pt-100 element (line length $L_2$, temperature $T_2$).
- **Loss 3** Secondary, downstream of the heat exchanger outlet, between the heat exchanger and the Pt-100 element (pipe length $L_3$, temperature $T_3$).

This data is tabulated below.

<table>
<thead>
<tr>
<th>Heat Exchanger</th>
<th>$L_1$ [m]</th>
<th>$L_2$ [m]</th>
<th>$L_3$ [m]</th>
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<tr>
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<td>2.44</td>
<td>1.26</td>
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<td>B</td>
<td>2.6</td>
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<tr>
<td>C</td>
<td>1.25</td>
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<td>D</td>
<td>0.81</td>
<td>0.61</td>
<td>2.47</td>
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Tab. 15.1: Length of heat exchanger

Any heat loss in the secondary circuit ahead of the heat exchanger inlet is negligible since the temperature is virtually ambient.

If the overall heat loss $HL$ is equated to the difference between the primary power dissipation and the secondary power consumption, the loss can be distributed among the various line stretches using the following formula:

$$\text{Loss 1} = \frac{L_1 \cdot (T_1 - T_{\text{amb}})}{(L_1 \cdot (T_1 - T_{\text{amb}})) + L_3 \cdot (T_3 - T_{\text{amb}})} \cdot HL.$$  

Gl. (15.1)

Loss 2 and Loss 3 can be calculated analogously.

Now that the overall heat loss has been apportioned to the three stretches, we can calculate the temperature difference resulting from the heat loss as follows.
$\Delta T = \frac{\text{Loss}_n}{m \cdot c_p}$  \hspace{1cm} \text{Gl. (15.2)}$

Depending on the direction of flow, this temperature difference should be added to or deducted from the primary and the secondary inlet and outlet temperatures.
16. APPENDIX G
Determination of energy flow at heat flow meters

As the temperature sensors of the heat flow meters and those of the PLC were rather far apart, the energy measurements are not as a matter of course in agreement because of the heat losses. Therefore, the energy flow measured by the PLC has been corrected for meaningful comparison.

Appendix F gives a break-down of the overall energy loss for the various lines. Since we now know the energy loss in the primary feed and return line, and the pipe length of these two lines, we can determine the **average energy loss per metre**.

The distance between the Pt-100 elements of the heat flow meter and those of the test facility is known. Multiplying the average energy loss per meter and the pipe length between the Pt-100 elements of the PLC and those of the heat flow meter yields the energy loss between the sensor of the PLC and the heat flow meter. This has been done for both the feed line and the return line.

In the case of the feed line to the heat exchanger, the temperature is measured first by the heat flow meter and next by the PLC. On the return line from the heat exchanger, the temperature is measured first by the PLC and next by the heat flow meter. Consequently, the two calculated losses should be added to the energy flow recorded by the PLC.
17. APPENDIX H
Conversion of secondary inlet temperature

During the measurements the secondary inlet temperature slowly varied between 18.3 and 25.8 °C. For a meaningful comparison of the heat exchangers (performance-wise) and the measured values of each heat exchanger, the data needs to be converted to a fixed inlet temperature. We have opted to convert to an average value of 20 °C.

The thermal efficiency [7] of the heat exchanger in this situation has been calculated using the temperatures corrected for the heat loss (Appendix F). The thermal efficiency is the ratio of the transferred power to the power transferred by an ideal heat exchanger. An ideal heat exchanger should here be understood to mean one of infinite length, so that the primary and secondary outlet temperatures are equal. The new outlet temperatures are calculated using the thermal efficiency and assuming a secondary inlet temperature of 20 °C. The following formulae have been used:

- Where the primary flow rate is greater than the secondary flow rate:

  \[
  \eta = \frac{T_{sec, out} - T_{sec, in}}{T_{prim, in} - T_{sec, in}},
  \]

  \[
  T_{sec, out, new} = T_{sec, in, new} + \eta \cdot (T_{prim, in} - T_{sec, in}) \text{ and}
  \]

  \[
  T_{prim, out} = T_{prim, in} - \frac{\Phi_{sec}}{\Phi_{prim}} \cdot (T_{sec, out} - T_{sec, in}).
  \]

- Where the secondary flow rate is greater than the primary flow rate:

  \[
  \eta = \frac{T_{prim, in} - T_{prim, out}}{T_{prim, out} - T_{sec, in}},
  \]

  \[
  T_{prim, out} = T_{prim, in} + \eta \cdot (T_{sec, in} - T_{prim, in}) \text{ and}
  \]

  \[
  T_{sec, out} = T_{sec, in} - \frac{\Phi_{prim}}{\Phi_{sec}} \cdot (T_{prim, in} - T_{prim, out}).
  \]
## 18. APPENDIX I
Results Heat Meters

### Reference

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<thead>
<tr>
<th>Type of heat meter</th>
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<th>according PLC/PC</th>
<th>PLCV/PC corrected</th>
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<td>MWh</td>
<td>m³</td>
</tr>
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<td>C</td>
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<tr>
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Tab. 18.1: Reference

### Habon-G concentration = 119 wppm

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<th>PLCV/PC corrected</th>
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Tab. 18.2: Habon-G concentration = 119 wppm

### Habon-G concentration = 520 wppm

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<td>m³</td>
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Tab. 18.3: Habon-G concentration = 520 wppm
### Habon-G concentration = 1470 wppm

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Tab. 18.4: Habon-G concentration = 1470 wppm
19. APPENDIX J

Example calculation using method of correction

This example shows how the measured values from a random point of measurement are corrected to allow meaningful comparison.

The correction consists of three parts:

- correction for heat loss
- conversion to new inlet temperatures
- conversion of the power measured at the heat flow meter

**Data used:**

Measured values: Heat exchanger B, reference measurement, point of measurement 15

\[
\begin{align*}
T_{\text{prim},i} &= 69.8 \degree\text{C}, \\
T_{\text{prim},\text{out}} &= 49.2 \degree\text{C}, \\
\Phi_{\text{prim}} &= 6.0 \text{ l/min}, \\
T_{\text{sec},i} &= 21.6 \degree\text{C}, \\
T_{\text{sec},\text{out}} &= 57.1 \degree\text{C}, \\
\Phi_{\text{sec}} &= 6.0 \text{ l/min},
\end{align*}
\]

Distance Sensor \(T_{\text{prim},i}\) and entrance heat exchanger = 2.60 m

Distance Sensor \(T_{\text{prim},\text{out}}\) and entrance heat exchanger = 1.60 m

Distance Sensor \(T_{\text{sec},\text{out}}\) and entrance heat exchanger = 1.74 m

**Correction Heat Loss:**

\[
\begin{align*}
P_{\text{prim}} &= \Phi_{\text{prim}} \cdot \rho \cdot c_p \cdot \Delta T = 17.9 \text{ kW}, \\
P_{\text{sec}} &= \Phi_{\text{sec}} \cdot \rho \cdot c_p \cdot \Delta T = 14.84 \text{ kW}, \\
P_{\text{loss}} &= 17.9 \text{ kW} - 14.84 \text{ kW} = 2.35 \text{ kW}.
\end{align*}
\]

\[
\begin{align*}
\text{Loss}_1 &= \frac{L_1 \cdot (T_{\text{prim},\text{in}} - T_{\text{amb}})}{L_1 \cdot (T_{\text{prim},\text{in}} - T_{\text{amb}}) + L_2 \cdot (T_{\text{prim},\text{out}} - T_{\text{amb}}) + L_3 \cdot (T_{\text{sec},\text{out}} - T_{\text{amb}})}, \\
\text{Loss}_1 &= 1.26 \text{ W}, \\
\text{Loss}_2 &= 0.46 \text{ W}, \\
\text{Loss}_3 &= 0.63 \text{ W}.
\end{align*}
\]

The corrections for the temperatures are:
\( \Delta T_1 = \frac{P}{\Phi_{\text{prim}} \cdot \rho \cdot c_p} = 1.5 \text{ K}, \)  
Gl. (19.5)

\( \Delta T_2 = \frac{P}{\Phi_{\text{prim}} \cdot \rho \cdot c_p} = 0.5 \text{ K}, \)  
Gl. (19.6)

\( \Delta T_3 = \frac{P}{\Phi_{\text{prim}} \cdot \rho \cdot c_p} = 1.5 \text{ K}, \)  
Gl. (19.7)

The corrected temperatures are:

\[ T_{\text{prim,in}} = T_{\text{prim,in}} - \Delta T_1 = 68.3 \, ^\circ\text{C}, \]  
Gl. (19.8)

\[ T_{\text{prim,out}} = T_{\text{prim,out}} + \Delta T_2 = 49.8 \, ^\circ\text{C}, \]  
Gl. (19.9)

\[ T_{\text{sec,out}} = T_{\text{sec,out}} + \Delta T_3 = 58.6 \, ^\circ\text{C}. \]  
Gl. (19.10)

**Calculation to standard temperatures:**

\[ T_{\text{prim,i}} = 70 \, ^\circ\text{C} \div T_{\text{sec,i}} = 20 \, ^\circ\text{C}, \]

The efficiency is:

\[ \eta = \frac{T_{\text{sec,out}} - T_{\text{sec,in}}}{T_{\text{prim,in}} - T_{\text{sec,in}}} = 0.79 \]  
Gl. (19.11)

The corrected standard output temperatures at \( T_{\text{prim,i}} = 70 \, ^\circ\text{C} \) and \( T_{\text{sec,i}} = 20 \, ^\circ\text{C} \) become:

\[ T_{\text{sec,out}} = T_{\text{sec,in}} + \eta \cdot (T_{\text{prim,in}} - T_{\text{sec,in}}) = 59.7 \, ^\circ\text{C} \text{ and} \]  
Gl. (19.12)

\[ T_{\text{prim,out}} = T_{\text{prim,in}} + \frac{\Phi_{\text{sec}}}{\Phi_{\text{prim}}} \cdot (T_{\text{sec,out}} - T_{\text{sec,in}}) = 50.2 \, ^\circ\text{C}. \]  
Gl. (19.13)

**Heat Flow at Heat Meter:**

Average heat loss pro meter:

\[ P_{\text{av.loss,in}} = \frac{1.26}{2.6} = 0.48 \frac{\text{kW}}{\text{m}}, \]  
Gl. (19.14)

\[ P_{\text{av.loss, out}} = \frac{0.46}{1.6} = 0.29 \frac{\text{kW}}{\text{m}}. \]  
Gl. (19.15)

The heat flow through the heat meter is:

\[ P_{\text{heatmeter}} = P_{\text{prim}} + P_{\text{loss,in}} + P_{\text{loss,out}} = 17.79 \text{ kW}. \]  
Gl. (19.16)
Table with the measured and calculated temperatures:

<table>
<thead>
<tr>
<th>Temperatures</th>
<th>$T_{\text{measured}}$ [$^\circ$C]</th>
<th>$T_{\text{corrected}}$ [$^\circ$C]</th>
<th>$T_{\text{calculated}}$ [$^\circ$C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{prim,i}}$</td>
<td>69.8</td>
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<td>70</td>
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<td>$T_{\text{prim,out}}$</td>
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<td>$T_{\text{sec,i}}$</td>
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<td>21.6</td>
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</tr>
<tr>
<td>$T_{\text{sec,out}}$</td>
<td>57.1</td>
<td>58.6</td>
<td>59.7</td>
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</table>

Tab. 19.1: Table with the measured and calculated temperatures
### 20. APPENDIX K

**Heat meters:**

**A: ICM RV 81.A**

<table>
<thead>
<tr>
<th>Temperature:</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>ΔT:</td>
<td>1 °C .. 100 °C</td>
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<tr>
<td>Temp. range:</td>
<td>20 °C .. 150 °C</td>
</tr>
<tr>
<td>Temp. sensor:</td>
<td>PT 100, DIN 43760</td>
</tr>
</tbody>
</table>

**Flow:**

| Type | 430 |
| Qn | 0.6 m³/h |
| DN | 15 |
| PN | 16 |
| meter class | C |
| Impuls | 1/l |

**B: Thermiflo**

<table>
<thead>
<tr>
<th>Temperature:</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>ΔT:</td>
<td>3 °C .. 70 °C</td>
</tr>
<tr>
<td>Temp. range:</td>
<td>20 °C .. 90 °C</td>
</tr>
<tr>
<td>Temp. sensor:</td>
<td>PT 100, IEC 751</td>
</tr>
</tbody>
</table>

**Flow:**

| Type | 430 |
| Qn | 1 m³/h |
| DN | 15 |
| PN | 16 |
| meter class | C/H-B/V |
| Impuls | 1/l |
### C: Clorius combimeter E 25

**Temperature:**
- ΔT: 5 °C .. 80 °C
- Temp. range: 20 °C .. 120 °C
- Temp. sensor: PT 100, ITL 21

**Flow:**
- Type: 6300015
- Q<sub>n</sub>: 0.03 .. 1.5 m<sup>3</sup>/h
- DN: 3/4"
- PN: 16
- meter class: C QIML Class 4
- Impuls: 1/1

### D: Pollux polluSonic-K

**Temperature:**
- ΔT: 3 °C .. 100 °C
- Temp. range: 0 °C .. 150 °C
- Temp. sensor: PT 100, ITL 21

**Flow:**
- Type: DWU 90
- Q<sub>n</sub>: 0.6 m<sup>3</sup>/h
- DN: 15
- PN: 16
- meter class: C/B
- Impuls: 1/1
21. APPENDIX L

Raw heat exchanger data

**HEAT EXCHANGER A Reference with 0 wppm**

<table>
<thead>
<tr>
<th>point</th>
<th>primary system</th>
<th>secondary system</th>
<th>pressure drop</th>
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<td>$T_{out}$</td>
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<td>[°C]</td>
<td>[°C]</td>
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Tab. 21.1: Heat exchanger A - reference with 0 wppm
Tab. 21.2: Heat exchanger A - Habon-G concentration = 119 wppm
### HEAT EXCHANGER A - Habon-G concentration = 520 wppm

<table>
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<th>primary system</th>
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<th>pressure drop</th>
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</table>

Tab. 21.3: Heat exchanger A - Habon-G concentration = 520 wppm
HEAT EXCHANGER A - Habon-G concentration = 1470 wppm

| point | primary system | | | secondary system | | | pressure drop |
|-------|----------------|----------------|------------------|----------------|----------------|----------------|
|       | flow [l/min]   | \(T_{in} \) [°C] | \(T_{out} \) [°C] | flow [l/min] | \(T_{in} \) [°C] | \(T_{out} \) [°C] | pump [mbar] | exch. [mbar] |
| 1     | 4.0            | 70.2            | 47.8             | 2.0          | 24.7            | 61.4            | 727          | 160         |
| 2     | 6.0            | 70.0            | 54.4             | 2.0          | 24.9            | 63.3            | 708          | 170         |
| 3     | 8.0            | 70.0            | 58.0             | 2.0          | 25.1            | 63.8            | 687          | 184         |
| 4     | 10.0           | 70.0            | 60.3             | 2.0          | 25.2            | 64.2            | 673          | 202         |
| 5     | 12.0           | 70.1            | 61.8             | 2.0          | 25.2            | 64.3            | 648          | 224         |
| 6     | 4.0            | 70.0            | 37.6             | 4.0          | 23.9            | 51.7            | 735          | 160         |
| 7     | 6.0            | 70.0            | 44.0             | 4.0          | 23.8            | 56.6            | 712          | 170         |
| 8     | 8.0            | 70.1            | 49.0             | 4.0          | 23.9            | 59.0            | 698          | 184         |
| 9     | 10.0           | 70.1            | 52.6             | 4.0          | 23.8            | 60.4            | 681          | 202         |
| 10    | 12.0           | 70.1            | 55.2             | 4.0          | 23.8            | 61.2            | 664          | 224         |
| 11    | 4.0            | 70.0            | 33.5             | 6.0          | 23.3            | 44.6            | 738          | 160         |
| 12    | 6.0            | 70.1            | 38.9             | 6.0          | 23.2            | 50.1            | 719          | 170         |
| 13    | 8.0            | 70.0            | 43.5             | 6.0          | 23.1            | 53.3            | 704          | 184         |
| 14    | 10.0           | 70.0            | 47.3             | 6.0          | 23.0            | 55.3            | 682          | 202         |
| 15    | 12.0           | 70.0            | 50.3             | 6.0          | 23.0            | 56.8            | 662          | 224         |
| 16    | 4.0            | 70.0            | 31.1             | 8.0          | 22.6            | 39.8            | 737          | 160         |
| 17    | 6.0            | 70.1            | 35.5             | 8.0          | 22.6            | 45.3            | 717          | 170         |
| 18    | 8.0            | 70.1            | 39.7             | 8.0          | 22.5            | 49.1            | 694          | 184         |
| 19    | 10.0           | 70.2            | 43.4             | 8.0          | 22.5            | 51.8            | 682          | 202         |
| 20    | 12.0           | 69.9            | 46.4             | 8.0          | 22.5            | 53.4            | 661          | 224         |
| 21    | 4.0            | 89.9            | 57.7             | 2.0          | 24.6            | 78.5            | 706          | 161         |
| 22    | 6.0            | 90.0            | 67.5             | 2.0          | 24.8            | 80.7            | 678          | 171         |
| 23    | 4.0            | 90.1            | 43.0             | 4.0          | 23.6            | 64.1            | 724          | 160         |
| 24    | 6.0            | 89.9            | 53.0             | 4.0          | 23.7            | 70.9            | 701          | 170         |
| 25    | 4.0            | 90.2            | 37.8             | 6.0          | 23.4            | 53.9            | 733          | 160         |
| 26    | 6.0            | 90.0            | 46.0             | 6.0          | 23.6            | 61.5            | 711          | 170         |
| 27    | 4.0            | 90.0            | 35.0             | 8.0          | 23.1            | 47.6            | 736          | 160         |
| 28    | 6.0            | 90.0            | 41.4             | 8.0          | 22.9            | 54.8            | 710          | 170         |

Tab. 21.4: Heat exchanger A - Habon-G concentration = 1470 wppm
### APPENDIX L Raw heat exchanger data

**HEAT EXCHANGER B - Reference with 0 wppm**

<table>
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<tr>
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<th>flow [l/min]</th>
<th>primary system</th>
<th>secondary system</th>
<th>pressure drop</th>
</tr>
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<tbody>
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<td>$T_{out}$ [°C]</td>
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Tab. 21.5: Heat exchanger B - reference with 0 wppm
### HEAT EXCHANGER B - Habon-G concentration = 119 wppm

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<th>secondary system</th>
<th>pressure drop</th>
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Tab. 21.6: Heat exchanger B - Habon-G concentration = 119 wppm
**HEAT EXCHANGER B - Habon-G concentration = 520 wppm**

| point | flow [l/min] | T\textsubscript{in} [°C] | T\textsubscript{out} [°C] | flow [l/min] | T\textsubscript{in} [°C] | T\textsubscript{out} [°C] | pressure drop pump [mbar] | pressure drop exch. [mbar] |
|-------|--------------|----------------|----------------|--------------|----------------|----------------|----------------|----------------|------------------------|
| 1     | 4.0          | 70.0           | 47.7           | 2.0          | 24.2           | 60.4           | 717            | 171           |
| 2     | 6.0          | 70.0           | 53.8           | 2.0          | 24.2           | 62.6           | 695            | 193           |
| 3     | 8.0          | 70.0           | 57.6           | 2.0          | 24.3           | 63.5           | 680            | 223           |
| 4     | 10.0         | 70.2           | 60.0           | 2.0          | 24.2           | 63.8           | 658            | 259           |
| 5     | 12.0         | 70.1           | 61.6           | 2.0          | 24.2           | 64.2           | 629            | 304           |
| 6     | 4.0          | 69.9           | 38.3           | 4.0          | 23.3           | 50.8           | 730            | 172           |
| 7     | 6.0          | 70.1           | 45.3           | 4.0          | 23.6           | 55.6           | 709            | 195           |
| 8     | 8.0          | 70.3           | 50.0           | 4.0          | 24.0           | 58.5           | 691            | 226           |
| 9     | 10.0         | 69.8           | 53.0           | 4.0          | 24.0           | 59.7           | 674            | 263           |
| 10    | 12.0         | 69.9           | 55.5           | 4.0          | 24.0           | 60.7           | 642            | 308           |
| 11    | 4.0          | 70.1           | 33.9           | 6.0          | 23.2           | 44.6           | 728            | 173           |
| 12    | 6.0          | 70.0           | 39.6           | 6.0          | 23.3           | 49.9           | 713            | 196           |
| 13    | 8.0          | 70.1           | 44.1           | 6.0          | 23.2           | 53.3           | 696            | 226           |
| 14    | 10.0         | 70.0           | 47.3           | 6.0          | 23.1           | 55.9           | 671            | 264           |
| 15    | 12.0         | 69.9           | 50.1           | 6.0          | 23.1           | 57.5           | 637            | 309           |
| 16    | 4.0          | 70.2           | 30.1           | 6.0          | 22.4           | 40.0           | 728            | 173           |
| 17    | 6.0          | 70.2           | 35.1           | 8.0          | 22.2           | 45.3           | 709            | 196           |
| 18    | 8.0          | 69.9           | 39.3           | 8.0          | 22.2           | 49.4           | 690            | 227           |
| 19    | 10.0         | 69.8           | 42.9           | 8.0          | 22.5           | 52.2           | 668            | 264           |
| 20    |              |                |                |              |                |                |                |                |
| 21    | 4.0          | 89.9           | 58.5           | 2.0          | 24.9           | 77.1           | 718            | 172           |
| 22    | 6.0          | 89.9           | 67.7           | 2.0          | 25.0           | 79.5           | 689            | 195           |
| 23    | 4.0          | 90.0           | 44.2           | 4.0          | 23.7           | 63.1           | 726            | 172           |
| 24    | 6.0          | 90.3           | 54.0           | 4.0          | 24.4           | 70.7           | 711            | 195           |
| 25    | 4.0          | 90.0           | 37.5           | 6.0          | 23.1           | 53.7           | 732            | 173           |
| 26    | 6.0          | 90.2           | 45.6           | 6.0          | 23.3           | 62.2           | 709            | 195           |
| 27    | 4.0          | 89.9           | 33.5           | 8.0          | 22.8           | 47.6           | 732            | 173           |

Tab. 21.7: Heat exchanger B - Habon-G concentration = 520 wppm
**HEAT EXCHANGER B - HABON-G CONCENTRATION = 1470 wppm**

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**Tab. 21.8:** Heat exchanger B - Habon-G concentration = 1470 wppm
HEAT EXCHANGER C - Reference with = 0 wppm

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Tab. 21.9: Heat exchanger C - reference with = 0 wppm
HEAT EXCHANGER C - Habon-G concentration = 119 wppm

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Tab. 21.10: Heat exchanger C - Habon-G concentration = 119 wppm
### HEAT EXCHANGER C - Habon-G concentration = 520 wppm

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Tab. 21.11: Heat exchanger C - Habon-G concentration = 520 wppm
HEAT EXCHANGER C - Habon-G concentration = 1470 wppm

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Tab. 21.12: Heat exchanger C - Habon-G concentration = 1470 wppm
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Tab. 21.13: Heat exchanger D - reference with 0 wppm
**HEAT EXCHANGER D - Habon-G concentration = 119 wppm**

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Tab. 21.14: Heat exchanger D - Habon-G concentration = 119 wppm
HEAT EXCHANGER D - Habon-G concentration = 520 wppm

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Tab. 21.15: Heat exchanger D - Habon-G concentration = 520 wppm
**HEAT EXCHANGER D - Habon-G concentration = 1470 wppm**

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Tab. 21.16: Heat exchanger D - Habon-G concentration = 1470 wppm
## 22. APPENDIX M
Result second reference measurement

### HEAT EXCHANGER B - Reference 2

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Tab. 22.17: Heat exchanger B - reference 2
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Tab. 22.18: Heat exchanger C - reference 2
HEAT EXCHANGER D - Reference 2

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Tab. 22.19: Heat exchanger D - reference 2
Project C

Survey of Environmental Restrictions to the Use of Additives in District Heating and Cooling Systems

BRUUN & SØRENSEN GROUP AS
CONSULTING ENGINEERS,
ECONOMISTS AND PLANNERS

May 1995
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1. Introduction

The purpose of this project was to provide a survey of restrictions and approval procedures for the use of additives such as friction reducing agents and phase-change materials in district heating and cooling systems in various countries.

The survey was based on a questionnaire distributed to the members of the IEA Advanced Fluids Expert Group at the beginning of 1995.

The target group for the survey is:
- The potential users of advanced fluids, i.e. district heating and cooling utilities,
- The manufacturers of additives,
- National, regional and local authorities involved in the approval of the use of the additives,
- The expert group.

1.1 Background

For more than 10 years an extensive research and test programme concerning the use of additives for district heating and cooling systems has been carried out in various countries.

As an example technical subjects related to the use of friction reducing agents Habon-G and Dobon-G have been examined in laboratory and field tests as well as theoretical studies with the conclusion that there are no serious technical problems in relation to the use of these additives in district heating systems.

One main issue, however, remains to be clarified before a full-scale application and commercial availability can be foreseen, namely the environmental restrictions related to the use of these additives. The reason for this is that the additives are slightly toxic although not more harmful than many surfactants used in industry and households.

The greatest concern to the use of additives is attached to the possible leakage to the surroundings (drinking water resources) and the risk of contaminating consumers’ hot sanitary water. These subjects therefore are some of the main obstacles for the use of additives in district heating and cooling systems.

Already at an early stage it was therefore recognized by the expert group that the use of friction reducing agents in district heating systems should be limited to primary heat transmission systems, which are hydraulically isolated from distribution networks, only.

It was therefore considered important to examine the attitude of the authorities toward this subject in various countries in order to evaluate the future possibilities for using this type of additives at a large scale.

On this background Bruun & Sørensen Group AS presented a proposal to the expert group on the execution of this survey and was appointed to carry out this task. The report was written by Mr. Flemming Hammer.
2. The survey

A questionnaire was distributed to all members of the expert group, and replies have been received from the following countries: Canada, Finland, Denmark, Germany, Korea, The Netherlands, Sweden and the USA. A copy of the questionnaire with the answers given by B&S for Denmark is in the appendix.

2.1 Basis

The first questions relate to the basis on which the replies are given.

Most of the information is based on general knowledge or informal contacts between individual members of the group and representatives from local, regional or national authorities.

In Germany and Denmark where concrete full-scale tests have been conducted in a total of six cases, which demanded approval from authorities, concrete experience has been gained, and is reflected in the answers.

Each country indicated the following basis of their answers:

- General knowledge - no contact with authorities: Korea, USA
- Some informal exchange of information: Canada, Finland, The Netherlands, Sweden
- Application under preparation: Sweden
- Application delivered: -
- Permission with restrictions obtained: Denmark, Germany
- Permission obtained without restrictions: -

2.2 Relevant additives

A number of substances have been used for small-scale testing in Canada, Germany, The Netherlands, Sweden and the USA. Large-scale field test were only made with the additives from Hoechst AG; Frankfurt, Germany (Habon), Habon-G and Dobon-G, the latter in conjunction with sodium salicylate. These additives are for district heating.

Habon-G (n-alkyldimethylpolyoxethylammonium, n=16) fits to the temperature conditions used in Herning, Denmark (40-100°C). The concentration used in order to obtain the maximum effect has been 150-250 ppm.

Dobon-G mixed with sodium salicylate n-alkyldimethylpolyoxethylammonium-3-hydroxy-2-naphthoate, n=20-22) was used in Völklingen, Germany, where the temperature level is 40-130°C. Concentration of Dobon-G 1500 ppm + sodium salicylate 700 ppm.
These additives are registered chemical substances in the EU-countries.

The following additives for district cooling have been submitted for approval to be imported or manufactured in Canada. They are active in the temperature range of approx. 0-30°C:

- Octadecyltrimethylammonium chloride**
- Cetyltrimethylammonium bromide (hexadecyltrimethylammonium bromide)*
- Dodecyltrimethylammonium bromide (or chloride)*
- Octadecyl-bis-(2-hydroxyethyl)-methylammonium chloride**
- 9-octadecen-bis-(2-hydroxyethyl)-methylammonium chloride**
- Myristiltrimethylammonium bromide (or chloride)**

The following additives, to be used as counter ions at a concentration of some 400 ppm have temporary approval for import/manufacture but will have to go through a more detailed review in the next year before being finally approved:

- 2,6-dihydroxybenzoic acid (α-resorcylic acid)*
- 3-methylosalicylic acid (2-hydroxy-3-methyl-benzoic acid or cresotic acid)*

* Available from Aldrich Chemical Co., Milwaukee, Wisconsin, USA
** Available from Akzo Chemicals Ltd., Toronto, Ontario, Canada

### 2.3 Transport, storage and handling of the additives

Hoechst AG has submitted EC safety data sheets complying with 91/155/EEC for Habon-G (Hoe 4089) and Dobon-G (Hoe 3987) (Appendix).

### 2.4 Authorities

These questions refer to the formalities which have been met by the contacts to the relevant authorities.

#### 2.4.1 Rules reported

In laboratory tests and/or small scale in-house tests no restrictions nor difficulties have been reported in any of the countries. In the Netherlands, however, the additive was treated as chemical waste after use.
Time limited large scale, realistic demonstration plants have been carried out in two countries only:

- In Germany approval was obtained for 3 projects of 2, 2 and 12 months duration, respectively.
- In Denmark it was approved to apply the additives in 3 projects with a duration of 18 months each.

In both countries it was possible to dispose of the tensid enriched after water use. In Denmark it could be led to the local sewage system under certain conditions (temperature below 35°C and low flow rate). The water was led to three step (mechanical, biological, chemical) water treatment plant, and no difficulties in this respect were reported. In Germany it was approved to dispose of the enriched water into a river after it had been treated with sodium bentonite. The bentonite separated from the water had to be treated as hazardous waste, and water led to the river had to be checked for fish toxicity, COD and BOD.

In Sweden a maximum of 50 mg/l (50ppm) is allowed in water disposed to sewers. The concentrations used in Herning and Völklingen, respectively, were approx 3 and 30 times higher than this limit.

Considerations of full-scale use, unlimited in time, are made in the Netherlands, Germany and Denmark. In Germany a concrete application for the use in a DH-network with a water volume of 78 m³ has been handed in to the authorities. No reactions have been obtained at present. In Denmark the national authorities have asked for further documentation on specific items before general permissions will be considered.

### 2.4.2 Level of authorities

Generally, both national, regional and local authorities are involved in the approval of additives in all countries. The local authorities will mainly be involved in matters related to the disposal, while approval for use is mostly dealt with by regional and/or national authorities.

### 2.4.3 Criteria

The criteria on which time-limited permissions were given in Germany and Denmark were almost equal:

- Use of heat exchangers, i.e. no direct contact to consumer installations,
- Leak indicating systems,
- Current documentation of the additives from the manufacturer,
- Documentation of the ability to decompose under anaerobic conditions,
- No introduction of tenside into surface water (Germany),
- The plants were fairly new and therefore considered safe (Denmark).
2.4.4 Demands still to be fulfilled

Subjects still to be documented in Denmark are related to which substances will be formed by decomposition of the additives and to which method to use in order to detect additives at concentrations of less than 10 μg/l.

2.4.5 Status of permission

In Germany the following projects have been approved:

- Permission for 2 months of operation with Dobon/sodiumsalicylate in a 1.2km $2^* \text{DN} \ 450 \text{ mm pipeline having a fluid volume of approximately} \ 500 \text{ m}^3$.
- Permission for 12 months of operation with Dobon-G/sodiumsalicylate in a system with a nominal diameter of DN 25 mm - DN 200 mm having a fluid volume of approximately 70 m$^3$.

In Denmark:

- Permission for 18 months of operation with Habon in a 2.8 km $2 \times \text{DN} \ 200 \text{ mm pipeline with concentration up to} \ 1000 \text{ ppm. Optimum effect was achieved at approx.} \ 220 \text{ ppm}$.
- Same with Habon-G.
- Same with Habon-G in one 7.5 km DN 125 mm pulsating pipeline + two storage tanks (in total 1000 m$^3$). Concentration for the time being is 150 ppm. Ongoing.

2.5 Conclusion

When this survey was launched, it was clear that the level of information and the technical background was very different from country to country. This is also reflected in the answers obtained, whereby an unambiguous conclusion covering the situation in all countries cannot be drawn.

In most countries there are, however, no concrete rules related to this new technology. It seems to be clear that a certain reluctance towards the introduction of new additives in general is a common attitude. The technology has not been declined in any of the countries.
3. Appendix

3.1 Questionnaire

Survey of environmental restrictions to the use of additives for advanced fluids in district heating and cooling systems.

Please fax or mail your answers to BRUUN & SØRENSEN GROUP AS, attn.: Flemming Hammer, P.O. Box 2151, DK-8240 Risskov

Fax + 45 86 17 39 88

ANSWERS:

1. Status

1.1 Please indicate your basis on which the answers are given:
   a. General knowledge - no contact with authorities
   b. Some informal exchange of information
   c. Application under preparation
   d. Application delivered
   e. Permission with restrictions obtained
   f. Permission obtained without restrictions
   e. Yes - in 3 cases!

2. Relevant additives

2.1 Which additive(s) have formally been presented to some authority with the purpose of obtaining approval for application?

Habon and Habon-G

Hoechst AG, Frankfurt
Att.: Frank-Peter Lang
phone +49 69 305 7516

2.2 Please give brief description of the relevant additive(s): Type, concentration by application etc.

FRA, tenside. Habon 250 ppm
n-Alkyltrimethylammonium
n = 16

FRA, tenside. Habon-G 150 - 250 ppm
n-Alkyldimethylpolyoxethylammonium
n = 16
2.3 Please indicate the same information for other relevant additives, which are expected to be introduced and may already have been subject of discussion with authorities.

2.4 Is the additive(s) a registered and approved chemical substance?

Yes

3. Transport, storage and handling of the additive(s)

3.1 Give information on rules in force for transport, storage and handling. This could be in the shape of a copy of the formal rules as submitted by the manufacturer.

Please refer to enclosure 1

4. Authorities

4.1 What type of system/activity have been considered/applied for/approved:
   a. plants for laboratory tests,
   b. small scale in house demonstration plants
   c. time limited large scale, realistic demonstration plants,
   d. full scale application without limitations of time,
   e. disposal of additive enriched water.

   c. Approval for 18 month in 3 projects.
   d. Considerations. Authorities very reluctant.
   e. Approval for disposal to sewage systems.

4.2 Which authority(ies) shall approve the application? Please indicate their level: National (N), Regional (R) or Local (L).

Ringkøbing Country (R) backed by Danish Environmental Protection Agency (N). Local municipality shall approve of disposal.

- The plants were fairly new and therefore considered safe.
- Use of heat exchangers, i.e. no direct contact to consumer installations.
- Alarm system in pipes and drainage underneath.
- Current documentation of the additive from the manufacturer.
- Documentation of the ability to decompose under anaerobic conditions
4.4 Specific demands to be expected before extended use.

- Documentation on substances formed by decomposition.
- Description of a method of analysis with a low detection value (less than 10 μg/l).

4.5 Please describe formal rules in respect of district heating/cooling water (with or without advanced fluids additives) in respect to:
   a. leakages from the central station,
   b. controlled leakages from the pipeline network,
   c. uncontrolled leakage from the network,
   d. leakage to consumer installations,
   e. disposal of additive enriched water.

   Before a D.H. system can be commissioned, it has to be approved by the relevant Country according to the law for "particular polluting activities". The possible pollution must be described and the Country will evaluate the information and approve it. Specifically:
   a. and e. water to be led to the sewage must be below 35 °C.
   b. and c. The pipeline network must be designed and laid according to the rules in force for underground storage tanks.

4.6 Status of permission(s) and indication of plant size.

   1. Permission for 18 month of operation with Habon in 2.8 km in 2 x DN 200 mm pipeline with concentration up to 1000 ppm.
   2. Same with Habon-G
   3. Same with Habon-G in one 7.5 km DN 125 mm pulsating pipeline + two storage tanks (in total 1000 m³). Concentration p.t. is 150 ppm.
   Ongoing.

4.7 Additional remarks

Please note that you are welcome to include enclosures referring to each item.

Flemming Hammer
30 November 1994
**1. Identification of the substance/preparation and company**

Product details

**Trade name**
Hoe S 3987

Supplier details:

**Firm**
HOECHST AG
D/65926 Frankfurt am Main
Telephone no.: 069/3050

Information provided by:

**Division:** D  Fine chemicals and colours

Emergency telephone number: 069-305-6418

**2. Composition/information on ingredients**

**Chemical characterization**
Fettalkyldimethylpolyoxethylammoniumsalz
(Dobon G, 35 %ig)

**UN number:** 1993

**Hazardous ingredients**

- **Isopropanol**
  - **Concentration:** 5 %
  - **CAS number:** 67-63-0
  - **Hazard symbols:** F 11
  - **R phrases:** 38

- **Fatty alkyldimethylpolyoyethylammonium salt**
  - **Concentration:** 35 %
  - **Hazard symbols:** Xi 38

**3. Hazards identification**

Flammable.
Irritating to skin.

**4. First aid measures**

**General information**
Remove soiled or soaked clothing immediately

after contact with skin
In case of contact with skin wash off immediately with soap and water

after contact with eyes
In case of contact with eyes rinse thoroughly with plenty of water and seek medical advice
5. Fire-fighting measures

Suitable extinguishing media
- water spray jet
- foam
- sand
- carbon dioxide
- dry powder

6. Accidental release measures

Methods for cleaning up/taking up
- Mechanisch aufnehmen. Reste mit Wasser verdünnen and mit
  flüssigkeitsbindendem Material (z.B.: Sägemehl, Natriumbentonit,
  Universalbinder) aufnehmen.

7. Handling and storage

Handling
Advice on safe handling
- Provide good ventilation when handling large quantities.

Advice on protection against fire and explosion
- Traces of flammable substances can collect in the vapour space of
  closed systems, therefore keep sources of ignition away.

Storage
- Storage class: 3A

*8. Exposure controls/personal protection

Ingredients with occupational exposure limits to be monitored

TRGS 900 / TRGS 905
Type and origin:
- MAK Maximum work place concentration
CAS number: 67-63-0
markings
- 2-Propanol
Limit value
- 400 ml/m³ Year: 1993
- 980 mg/m³

Extreme value limit category II,1

Personal protective equipment

General protective measures
- Avoid contact with skin
- Avoid contact with eyes

Hand protection:
- Gloves
9. Physical and chemical properties

Appearance
- Form: Pasty
- Colour: beige
- Odour: of isopropanol

Date relevant to safety

Changes in physical state

Pourpoint
- Approx. method: DIN/ISO 3016
- Pourpoint: 18 °C

Flash point
- Method: DIN 51755
- Flash point: 36 °C

Density
- Method: DIN 51757
- Density: 0.98 g/cm³ at 60 °C

Solubility in water
- Soluble, turbid

pH value
- 6 to 7
- (at 10 g/l H₂O)

Combustion number: BZ1
- Does not catch fire

10. Stability and reactivity

11. Toxicological information

Acute oral toxicity (LD₅₀)
- Species: rat
- Method: OECD 401
- LD₅₀: > 2000 mg/kg

Irritant effect on skin
- Specie: rabbit
- Method: OECD 404
11. Irritant effect on eyes
   non-irritant
   Species: rabbit eye
   Method: OECD 405

12. Ecological information

   Data on elimination (persistence and degradability):
   Biodegradability > 90 %
   Method: OECD confirmatory test

   Ecotoxic effect
   Fish toxicity (LC50)
   Duration of exposure: 8.5 mg/l
   Species: zebra fish
   Method: OECD 203

   Bacteria toxicity (EC50)
   Method: OECD 209

13. Disposal considerations

   Product
   In accordance with local authority regulations, take to special waste incineration plant

*14. Transport information

   Road transport
   ADR: 3/31C
   GGVS: 3/31C
   RID: 3/31C
   GGVE: 3/31C

   Product characteristic
   ENTZUENDBARER FLUESSIGER STOFF, N.A.G. (ISOPROPANOL (ISOPROPYLALKOHOL))

   Hazard no.: 30
   Substance number: 1993

   Inland waterways transport
   ADNR: 3/31C

   Product characteristic
**EC safety data sheet**

Safety data sheet in accordance with 91/155/EEC  
Trade name  
Hoe S 3987  
Product no. 06 FBON145  
Substance code: 36995  
Version 4  
Position 11.01.95  
Page: 5 (5)

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<th>3-07 *</th>
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<thead>
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<tr>
<td>Correct technical name</td>
<td>FLAMMABLE LIQUID, N.O.S. (ISOPROPYL ALCOHOL)</td>
<td></td>
</tr>
</tbody>
</table>

Further information  
Dispatch by post Not permitted.

* 15. Regulatory information

Labelling in accordance with GefStoffV/EC  
hazard warning labelling compulsory

Hazard symbols  
Xi Irritant

Hazardous component(s) to be indicated on label  
contains  
- Isopropanol  
- Fatty alkyldimethylpolyoxethylammonium salt

R phrases  
10 Flammable.  
38 Irritating to skin.

S phrases  
16 Keep away from sources of ignition — No smoking.  
26 In case of contact with eyes, rinse immediately with plenty of water and seek medical advice.  
28.2 After contact with skin, wash immediately with water and soap.  
37/39 Wear suitable gloves and eye/face protection.

National regulations :  
Water Hazard Class (Germany) : 1 (self-classification)

16. Other information

This information is based on our present state of knowledge. It should not therefore be construed as guaranteeing specific properties of the products described or their suitability for a particular application.

Chapter which has been changed in respect of its previous version is marked with * * *.  

Date of printing : 24.04.95
Project D

Improving the Heat Transmission Properties of Tube Bundle Heat Exchangers by Installing Obstacles inside the Pipes

D1 Investigations of Heat Transfer and Pressure Drop

D2 Testing of Obstacles in an Operating Heat Exchanger and Evaluation of the Overall Effect

December 1996
Improving the Heat Transmission Properties of
Tube Bundle Heat Exchangers by Installing
Obstacles inside the Pipes

Professor Weinspach

Thermische Verfahrenstechnik GmbH

December 1996
1. Introduction

The application of drag reducing additives is a promising technology to increase the economic viability of district heating and cooling systems. With small amounts of cationic surfactants in district heating water the pressure drop in pipelines can be reduced significantly. Due to this effect the pump energy of existing district heating systems can be decreased or the transport capacity can be raised clearly. In addition the supply temperature of DH-systems can be decreased significantly when keeping the heat output constant. New networks can be designed with smaller pipe diameters or the maximum economic transmission length can be increased by applying drag reducing additives.

Parallel to the positive effect of drag reduction further effects occur due to the change in flow behaviour (viscoelastic instead of Newtonian). These are influences on heat exchangers, heat meters, pumps, filters, etc. The most important problem is the heat transfer reduction in heat exchangers. In shell and tube heat exchangers with straight long pipes a heat transfer reduction of about 95% and therefore a significant reduction of heat output is possible. Up to now most of the influences have been investigated in laboratory and full scale tests.

Concerning the application in DH-systems, measures have to be carried out to compensate for all significantly negative effects. Considering the supply guarantee as most important requirement the influence on heat exchangers has to be payed particular attention. One of the most promising measures to improve the heat transmission properties of tube bundle heat exchangers is the installation of turbulence increasing obstacles inside the pipes.

In former investigations [15] wire nettings have been installed to improve the heat transfer coefficient inside straight pipes. The result of these investigations with drag reducing additives was a significant improvement. The values for water without turbulence increasing installations have almost been achieved. On the other hand the increase in heat transmission is accompanied by a strong increase in pressure drop.

Therefore new investigations have to be carried out to find better kinds of installations which show sufficient properties concerning heat transmission and pressure drop behaviour. This is the goal of the proposed study. Different obstacles have been tested to find a proper solution.

In this project, spiral springs of different gradients have been used to increase turbulence inside a double pipe heat exchanger. As influence quantities, the flow velocity, temperatures at the entrances of the heat exchanger as well as the concentrations of the cationic surfactant (HABON-G) have been varied.
2. State of the art

2.1 Behaviour of drag reducing surfactant solutions

Surfactants are low-molecular substances with a low chemical activity and a low solubility. Surfactants consist of a hydrophobic group - as a long-chain alkyl-part in general - and a hydrophilic part which consists of a molecule-part that can be ionized, of a polar or polarizable group or a molecule-part which can build hydrogen bonds.

For an application as drag reducing agents in district heating systems quaternary ammonia-compounds have been taken into consideration (see figure 2.1). Concerning the operating conditions of district heating systems the cationic substances HABON-G \((n = 16)\) and DOBON-G \((n = 22)\) together with the additional counter-ion salicylate (added as sodiumsalicylate) have been proved to be sufficient [1].

DOBON-G in addition with sodiumsalicylate shows the drag reducing effect in a temperature range of \(40 ^\circ C < T < 130 ^\circ C\) and critical wall shear stresses below ca. 130 Pa. The achievable drag reduction is about 83 % compared to water [16].

The additive HABON-G shows a comparable drag reducing effect in a temperature-range of \(30 ^\circ C < T < 105 ^\circ C\) below a wall shear stress of about 80 Pa [9]. Therefore, the conditions under which a drag reducing effect occurs correspond quite well with the operating conditions of district heating systems.

\[
\begin{align*}
C_n & \text{H}_{2n+1} - N - CH_3 & n = 16: \text{HABON-G} \\
(C_2H_4O)_1 & \text{H} & n = 22: \text{DOBON-G}
\end{align*}
\]

\(n\)-alkyldimethylpolyoxethylammonia-cation

\[
\begin{align*}
\text{3-hydroxy-2-naphthoate (counter-ion); salicylate (additional counter-ion)}
\end{align*}
\]

Fig. 2.1: Structure of the considered additives [9]

In diluted solutions surfactant-monomeres form spherical micelles above the critical concentration \(\text{CMC I (critical micelle concentration)}\). In aqueous solutions the hydrophobic alkyl-chains form the core of the spherical micelles. The hydrophilic groups stick into the solution and build the cover of the sphere. If the concentration is increased drag reducing surfactants form rod like micelles above a second characteristic concentration - the \(\text{CMC II [1]}\).

The presence of rod like micelles is considered to be a necessary condition for the drag reducing effect. Above a certain shear stress (onset-point) the rod like micelles form a specific structure - the so-called shear induced state (SIS) [1]. The shear induced state is shown in figure 2.2. The
micelles align in direction of the flow and reduce the radial turbulences and therefore the pressure loss inside pipes, too [3]. Furthermore the shear induced state causes the viscoelastic properties, due to interlocking and bending of the structure.

**Fig. 2.2:** Shear induced state (SIS) [11]

The drag characteristic of a surfactant solution inside a straight pipe compared with that of pure water is presented in **figure 2.3**.

**Fig. 2.3:** Drag characteristic of a straight pipe for surfactant solution and water [1]

For small Reynolds numbers (range 1) the surfactant solution is moved parallel to higher drag coefficients. This results from the higher viscosity of the surfactant solution. With increasing flow velocity (range 2) the shear induced state is built up and a drag reduction occurs. The drag coefficients for surfactant solution are below those for water - the difference is increasing with rising Reynolds number. In range 3 the maximum drag reduction appears. The characteristic runs along
an asymptote \([1]\). In range 4 the drag reducing effect disappears (point \(*\)) and the characteristic reaches the curve for water. The critical value \(*\) can be described independent from pipe diameter using the critical wall shear stress. Usually the critical wall shear stress is calculated for water (index \(w\)):

\[
\tau_{w,w}^* = \frac{\Delta p_w^* \cdot d_i}{4 \cdot L} = \frac{\xi_{w}^* \cdot \rho \cdot w^2}{8},
\]

with \(\tau_{w,w}^*\) : critical wall shear stress for water,
\(d_i\) : inner diameter of the pipe,
\(L\) : length of the pipe and
\(\Delta p_w^*\) : pressure loss inside the pipe (length "\(L"\) ) for water.

The drag reduction \(DR\) compared to water is characterized by the percental difference of pressure drop between water and surfactant solution:

\[
DR = \frac{\Delta p_w - \Delta p_s}{\Delta p_w} \cdot 100\% = \frac{\xi_{w} - \xi_{s}}{\xi_{w}} \cdot 100\% = \frac{\tau_{w,w} - \tau_{w,s}}{\tau_{w,w}} \cdot 100\%.
\]

In addition to the desired drag reduction a decrease of heat transfer occurs due to the reduction of radial turbulences. This can influence the heat transfer in heat exchangers considerable. In the worst case the reduction of heat transfer can lead to a significant reduction of heat output and therefore, can cause bottle-necks in heat supply.

A typical heat transfer characteristic (Nusselt versus Reynolds number) inside a straight pipe is shown in figure 2.4.

**Fig. 2.4:** Heat transfer characteristic of a surfactant solution (HABON) [15]
In the laminar region the Nusselt characteristic for surfactant solution is identical to that for water. With increasing Reynolds number the surfactant solution keeps the laminar heat transfer characteristic up to the critical Reynolds number.

Corresponding to the drag reduction (DR, equation 2.2) the heat transfer reduction HTR can be defined as the percentage decrease in heat transfer coefficient $k$ (or Nusselt number) compared to that one of water (index $w =$ water; $s =$ surfactant) [4]:

$$HTR = \frac{k_w - k_s}{k_w} \cdot 100\% = \frac{Nu_w - Nu_s}{Nu_w} \cdot 100\%. \quad \text{eq. (2.3)}$$

Comparing the effects of heat transfer reduction and drag reduction the heat transfer reduction is always stronger than the drag reduction under otherwise equal conditions [4].

The heat transfer reduction HTR describes only the influence on one side of the heat exchangers. The overall heat transfer $k_0$ is determined by several thermal resistances. Therefore the percentage reduction of the overall heat transfer OHR has been defined:

$$OHR = \frac{k_{0w} - k_{0s}}{k_{0w}} \cdot 100\%. \quad \text{eq. (2.4)}$$

This parameter allows more precise statements about the behaviour of overall heat transmission. OHR considers all thermal resistances of a heat exchanger. Due to the fact that only one resistance is reduced the overall heat transfer reduction OHR is always smaller than the heat transfer reduction HTR under the same conditions [3].

Finally the most important parameter which has to be considered in this connection is the heat load. Due to the influence on heat transfer the heat load is reduced compared to water. The heat output reduction HOR is defined as:

$$HOR = \frac{Q_w - Q_s}{Q_w} \cdot 100\%. \quad \text{eq. (2.5)}$$

An additional effect concerning the influence on heat exchangers is as a rule the increase of the effective temperature difference $\Delta T_m$. The outlet temperature is increasing on the hot side $(T_{1,\text{out}})$ and decreasing on the cold side $(T_{2,\text{out}})$. It can be easily shown that as a result the reduction of the heat load $\dot{Q}_i$ is smaller than that of the overall heat transfer coefficient $k_{0i}$.

A quite large heat transfer reduction is not necessarily causing a large heat output reduction. The overall heat transfer reduction and the heat output reduction are mainly determined by the conditions of heat transfer on both sides of the apparatus and the strength of HTR.

If the decisive heat transfer resistance (small $k$-value) is on the side where the application of additives is planned the decrease of the overall heat transfer (and therefore also of the heat output) caused by the use of surfactants instead of water is strong. If in the opposite way the reduced heat transfer resistance is not the decisive one, the overall heat transfer and the heat output will not decrease distinctly.
As a characterisation of the heat transfer behaviour the parameter $\omega$ has been introduced. This parameter is defined as the ratio between two heat transfer coefficients [11]:

$$\omega = \frac{k_1}{k_2},$$  \hspace{1cm} \text{eq. (2.6)}

The parameter $\omega$ describes the behaviour under the original conditions (operation with pure water). In case of $\omega >> 1$ the overall heat transfer reduction is quite small but in the opposite case of $\omega << 1$ the side on which surfactants should be used will be influenced very strong (see figure 2.5). In addition to $\omega$ the level of heat transfer reduction is a further decisive parameter that is determining the overall heat transfer coefficient and also the heat output reduction.

The range of heat transfer reduction and the values of $\omega$ that appear in typical heat exchangers are shown in figure 2.5. This figure shows the influence of $\omega$ and HTR on the overall heat transfer reduction OHR for a typical apparatus as example with the following operating conditions [5][12][13][14]: ($\lambda/s = 24,000 \text{ W/(m}^2\text{K)}$, $k_1 = 5,500 \text{ W/(m}^2\text{K)}$, $R_f = 0 \text{ (m}^2\text{K)/W}$).

**parameters:**
- $k_1 = 5,500 \text{ W/(m}^2\text{K)}$
- $\lambda/s = 24,000 \text{ W/(m}^2\text{K)}$
- $R_f = 0 \text{ (m}^2\text{K)/W}$

![Diagram](image)

**Fig. 2.5:** OHR as a function of HTR and $\omega$ for typical DH-heat exchanger [11]

The strongest heat transfer reduction occurs in shell and tube heat exchangers. HTR values of 95% in maximum can be caused due to the application of cationic surfactants. Especially in long pipes, strong influences are possible [15]. Therefore, it is necessary to develop measures to improve the heat transmission properties inside the pipes of these apparatus when planning the application of drag reducing additives. Otherwise problems concerning the supply guarantee can occur.
The reduction of heat transfer coefficients in helical tube and plate heat exchangers are smaller. Combining the parameters \( \omega \) and HTR strong influences occur in shell and tube condensers and shell and tube heat exchangers that operate with water on both sides. On the other hand the influence on shell and tube heat exchangers that are operating with gas and water is small due to a high value of \( \omega \). Values of OHR are only about 20 % if HTR is about 90 % and the values of \( \omega \) are relatively high.

The Nusselt characteristic of straight pipes operating with surfactant solutions strongly depends on the tube length and the ratio \( L/d_i \) respectively [15]. With increasing tube length the HTR is increasing if the Reynolds number is kept constant. The reason for this behaviour is the percentage of the good heat transfer in the range of the entrance length of pipes \((50 \div 100) \cdot d_i\). In long tubes \((L = (600 \div 1,000) \cdot d_i)\) the heat transfer is nearly determined by a laminar flow. In this case the good heat exchange on the entrance length is only of minor importance for the total heat transfer inside the pipe. In comparison to this the HTR in very short pipes is only small due to the high percentage of turbulent flow and good heat transfer respectively.

**Table 2.1** shows the experimental results of the heat transfer reduction of a straight pipe for different pipe lengths and values for \( L/d_i \) respectively. The significant influence of the pipe length is obvious.

<table>
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<tr>
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<th>results</th>
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<tr>
<td>surfactant</td>
<td>Habon</td>
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<td>concentration</td>
<td>1,000 wppm</td>
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<tr>
<td>Re</td>
<td>20,000</td>
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<table>
<thead>
<tr>
<th>L/di</th>
<th>HTR</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>77.7</td>
</tr>
<tr>
<td>142</td>
<td>83.8</td>
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<tr>
<td>600</td>
<td>88.5</td>
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</table>

**Tab. 2.1:** Influence of the pipe length on HTR

For the calculation of heat transfer in straight pipes with drag reducing additive solutions, a formula has been developed which exactly describes the technical relevant range [15]:

\[
Nu = \sqrt[3]{3.66^3 + 2,191 \cdot \frac{d_i}{L} \cdot \frac{L + L_e}{L}}^{0.249} \quad \text{eq. (2.7)}
\]

with: \( d_i \) : inner diameter [m],
\( L \) : length of the pipe [m] and
\( L_e \) : entrance length [m].

\[
L_e = 0.075 \cdot d_i \cdot Re_w \quad \text{eq. (2.8)}
\]

\( Re_w \) is calculated with the material properties of water at wall temperature.

The validity range of this formula can be gathered from [15].

In **table 2.2** two example calculations which have been carried out with the simulation program TenSim [2] are presented. The results of the characteristic values HTR, OHR and HOR for different Reynolds numbers are shown.
The values decrease in the above mentioned order: HTR > OHR > HOR.

<table>
<thead>
<tr>
<th>Re</th>
<th>HTR</th>
<th>OHR</th>
<th>HOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>24,000</td>
<td>86%</td>
<td>79%</td>
<td>69%</td>
</tr>
<tr>
<td>61,000</td>
<td>91%</td>
<td>80%</td>
<td>74%</td>
</tr>
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</table>

**Tab. 2.2:** Simulation results for HTR, OHR and HOR

Table 2.2 impressively shows the necessity of improving the heat transmission properties when using drag reducing additives in existing shell and tube heat exchangers. A reduction of heat output of about 70% can not be tolerated.

Therefore this study to the improvement of heat transfer inside straight pipes by installing obstacles has been carried out.

In preliminary investigations from Weber at the University of Dortmund positive results of an artificial increase in turbulence to improve heat transfer have been received in principle. Wire nettings which have been installed inside pipes have been used as turbulence increasing obstacles. In **figure 2.6** some results of these investigations are presented.

![Improvement of heat transfer of a 1000 wppm-HABON solution by wire nettings installed inside pipes](image)

**Fig. 2.6:** Improvement of heat transfer of a 1000 wppm-HABON solution by wire nettings installed inside pipes

Compared with the smooth pipe without obstacles a significant increase can be achieved. The heat transmission reduction can be reduced from 81 - 84% to 30 - 44%, due to the installation of a wire netting.

On the other hand measurements of the pressure drop showed a tremendous increase of 200 to 300% compared to the operation with water without obstacles. Therefore the increase of pressure drop is much stronger than the improvement in heat transfer.
Assuming a heat transfer reduction of 40 % (instead of 81 - 84 %) an example calculation has been carried out for a condenser. The following parameters - geometry and operating conditions - have been assumed:

**Geometry:**
- Length of pipe: 12 m,
- Inner diameter: 12.6 mm,
- Number of pipes: 4,600.

**Operation:**
- Max. heat output: 87 MW,
- Condensation temp.: 76.7 °C,
- Max. flow velocity: 2.1 m/s.

For the origin state - smooth pipe/no obstacles/surfactant solution - the simulation of the condenser shows the following results:

- **HTR:** 95 % and
- **HOR:** 72 %.

Assuming the improvement of heat transfer due to the installation of obstacles of a HTR of 40 % (which is a pessimistic estimation), the resulting heat output reduction is:

- **HOR:** 6.4 %.

The decrease in heat output can be reduced of about 90 % due to the increase in turbulence. The HOR of 6.4 % can be tolerated concerning the supply guarantee of real district heating systems.

The example impressively shows the potential of the installation of obstacles to improve the heat transfer (and output) behaviour of shell and tube heat exchangers.
3. Method

3.1 The test plant

The experiments have been carried out in the test plant presented in figure 3.2. Main parts of the plant are the two separated circles - the hot circle and the cold one. Both can be filled up with surfactant solutions. The drag reducing surfactants can be added over a special dosage device (B3 and P3). Both circles are equipped with the necessary measurement and control devices. Therefore the required entrance conditions (temperatures, flow rates) can be realized.

The centrifugal pump P1 is transporting the heat carrier from B1 through the heating circle. The water/surfactant solution is pumped through a flow heater with a maximum electric power of 190 kW. The required temperature is controlled by the control unit TIC 7. Behind the flow heater the heat carrier reaches a by-pass which is installed for a better temperature control. One part of the flow is transported back to B1 and the other part reaches the test heat exchanger HE. The flow rate of the hot circuit is controlled by the unit FRC 1. The realizable flow range is from 0.05 m$^3$/h to 20 m$^3$/h. The pressure of the pump is 5 bar.

The cold circle is identical to the hot one in principle. Instead of a flow heater two series connected plate heat exchangers are installed to cool the heated solution. Further informations can be gathered from figure 3.2.

If stationary conditions are reached, the four temperatures, the two flow rates and the two pressure differences are measured and registered.

As test heat exchanger a special developed double pipe heat exchanger has been installed. The apparatus has a length of 4,000 mm, an inner diameter $d_i$ of 17.3 mm and an outer diameter $d_o$ of 28.5 mm (see figure 3.1).

The investigated obstacles are shown in figure 3.3. The spiral springs have an outer diameter of 17.5 mm. Their diameter is bigger than the inner diameter of the inner pipe of the heat exchanger to reach a certain tension between the spring and the wall of the pipe.

Due to this tension the spiral springs stay on their location at the inner pipe wall even when high flow rates have been adjusted.

$$d_i = 17.3 \text{ mm}$$
$$d_o = 28.5 \text{ mm}$$

![Fig. 3.1: Investigated double pipe heat exchanger](image-url)
The thickness of the wire has been 1 mm for any different kind of spiral spring.

\[ d = 17.5 \text{ mm} \]

\[ s = \{8, 16, 24\} \text{ mm} \]

**Fig. 3.3:** Spiral springs

During the investigations, the pitch \( s \) has been varied. Springs with pitches of 8 mm, 16 mm and 24 mm have been installed and investigated.

### 3.2 Program

For the investigations the cationic surfactant HABON-G has been applied (see figure 2.1). The varied parameters have been:

- Temperature,
- concentration,
- flow velocity and
- pitch of the spiral springs.

Due to the fact that the results of the laboratory tests shall be used for the installation of spiral springs inside a real condenser the operating conditions (temperatures and flow velocities) have been adjusted to the operating conditions of the full scale apparatus. To reach a nearly constant temperature on the hot side the flow rate of the hot circle has always been relatively high. The maximum flow rate was limited by the essential required accuracy of the thermometers (temperature difference). The different entrance temperatures are presented in table 3.1.

<table>
<thead>
<tr>
<th>Number</th>
<th>entrance temperature cold circle (surfactant)</th>
<th>entrance temperature hot circle (water)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>55 °C</td>
<td>77 °C</td>
</tr>
<tr>
<td>2</td>
<td>70 °C</td>
<td>94 °C</td>
</tr>
<tr>
<td>3</td>
<td>55 °C</td>
<td>84 °C</td>
</tr>
<tr>
<td>4</td>
<td>82 °C</td>
<td>90 °C</td>
</tr>
<tr>
<td>5</td>
<td>30 °C</td>
<td>60 °C</td>
</tr>
<tr>
<td>6</td>
<td>60 °C</td>
<td>90 °C</td>
</tr>
</tbody>
</table>

**Tab. 3.1:** Entrance temperatures of hot and cold circle
After having carried out the reference measurements with water, the concentrations:

- 125 wppm (weight parts per million),
- 250 wppm and
- 500 wppm

have been adjusted.

The flow rate of the cold medium (surfactant) has been varied between 0.4 \text{ m}^3/\text{h} and 2.0 \text{ m}^3/\text{h}. 18 different flow velocities between the minimum and maximum values have been investigated to get sufficient results for heat transfer and pressure drop.
4. Results

For the presentation of the results the non-dimensional quantities Re, ξ and Nu are preferred. Drag characteristics (ξ/Re-diagrams) and Nusselt characteristics (Nu/Re-diagrams) are used to describe the hydrodynamic and the heat transfer behaviour of water and surfactant solutions inside straight pipes with and without spiral springs.

The non-dimensional quantities are defined as follows:

\[ \text{Re} = \frac{d \cdot w \cdot \rho}{\eta}, \]  
\[ \xi = \frac{\Delta p \cdot \rho}{2 \cdot w^2 \cdot \frac{L}{d}} \] and  
\[ \text{Nu} = \frac{k \cdot d}{\lambda}. \]

In these definitions the physical properties of pure water are used. This is the usual procedure to describe the behaviour of drag reducing solutions [1][4][15].

Due to the installation of the obstacles inside the pipes the hydraulic diameter is decreasing. Calculations of the hydraulic diameters and comparisons between these values and the normal diameter \( d_i \) (see figure 3.1) show that the influence of the decrease in hydraulic diameter is neglectable. The inaccuracy when simply using the inner diameter is 1.1% in maximum (for the smallest pitch \( s = 8 \text{ mm} \)).

4.1 Heat transfer

4.1.1 Results without spiral springs

Before installing the spiral springs reference measurements without obstacles have been carried out to compare the results to those of former investigations. In figure 4.1 the Nusselt characteristics for water and surfactant solutions of different concentrations and entrance temperatures are shown.

As known from former measurements the Nusselt numbers for drag reducing surfactants are significantly below those for pure water if the Reynolds number is below a critical value (or better: if the wall shear stress is below the critical wall shear stress). Behind the critical wall shear stress the characteristics rise and reach the values of water. The critical wall shear stress is dependent on temperature and concentration.
With increasing concentration the point on which the characteristic rises to that one for water is moving to higher Reynolds numbers. The increase in temperature leads to the same effect because both temperatures are below the maximum value $T_{\text{opt}} = T (\tau_{\text{opt}})$ with $\tau_{\text{opt}} = \tau_{w,\text{max}}$ (see [4]).

The qualitative behaviour is identical to former results as well as the scale of the values. The heat transfer reduction (HTR) in accordance with equation (2.3) reaches values above 90%. For Reynolds numbers below the critical value the heat transfer coefficients for surfactant solutions can be calculated with the equation of Weber [15] (equation (2.7)) with sufficient precision.

### 4.1.2 Results with spiral springs

#### 4.1.2.1 Results with water

In figure 4.2 the results of the heat transfer inside the pipe with a spiral spring for water are shown. The pitch of the spring was 8 mm and the entrance temperature has been 55 °C.

It can be seen impressively that the heat transfer is increasing enormously. The values of the Nusselt number with obstacles are about twice the corresponding values without the spiral spring. The reason for that is the increase in turbulence due to the disturbance of the flow near the wall behind the wire of the spring. For the following description of the heat transfer behaviour the heat transfer improvement (HTI) is defined:

$$HTI = \frac{k_S - k_w}{k_w} \cdot 100 \% \quad \text{or} \quad HTI = \frac{\text{Nu}_S - \text{Nu}_w}{\text{Nu}_w} \cdot 100 \%,$$

**eq. (4.4)**
with:  Index S: „Spring” and
       index w: „water (without spiral springs)“.

The definition HTI uses the „origin state“ of the heat exchanger (water, without obstacles) as reference value to determine the heat transfer improvement. The values $k_S$ or $N_u_S$ can stand either for water or for surfactant solution. Thus, for the application of drag reducing additives, the value of HTI has to be above 0 to reach the value of water without springs.

\[ \text{Fig. 4.2: Nusselt characteristic for water with spiral springs (s = 8 mm, } T_{in} = 55 \, ^\circ C) \]

Considering the results in figure 4.2, the heat transfer improvement reaches values between 85 % and 100 %. Similar results can be received, if the temperature has been varied.

\[ \text{Fig. 4.3: Nusselt characteristic for water with springs of different pitches (} T_{in} = 55 \, ^\circ C) \]
The influence of the pitch of the spiral spring is shown in figure 4.3. The values of the 8 mm and the 16 mm spring are almost identical. Only at high Reynolds numbers the characteristic of the 8 mm spring is bending to lower values.

The characteristic of the 24 mm spring runs significantly below the two other curves. The medium heat transfer improvement of this spring is above 70% in contrary to the 8 mm and the 16 mm spring which show an average improvement of about 90%.

The explanation of this behaviour could be an „effective distance“ on which an optimum heat transfer improvement occurs due to the increase in turbulence. In contrary to these „effective distance“ there are areas on which those turbulences and therefore, the heat transfer improvement are reduced.

The „effective distance“ begins behind the spring (after a specific length) and ends after a certain distance. Directly behind the wire the turbulences have to develop to improve the heat transfer conditions near the wall. After a certain length, the turbulences are becoming calm. Both the beginning and the end of the effective distance - the range of maximum heat transfer improvement - are dependent on the Reynolds number. In figure 4.4 and 4.5 a qualitative description of the possible behaviour of the turbulences is given.

**Fig. 4.4:** Qualitative description of turbulence improvement at medium Reynolds numbers

**Fig. 4.5:** Qualitative description of turbulence improvement at higher Reynolds numbers
At medium Reynolds numbers the characteristic for the 8 mm and the 16 mm springs are nearly identical (see figure 4.3). On the one hand the effective distance of the 16 mm spring is - compared to the 8 mm spring - shorter than its pitch. On the other hand in case of the 8 mm spring twice as much windings are existing. Behind these windings the turbulences has to be developed first before improving the heat transfer near the wall. These effects compensate for each other. In case of the 24 mm spring the pitch is significantly larger than the effective distance (The disturbance of the flow is not strong enough to increase the turbulences between two windings of the springs). Therefore the heat transfer improvement is much smaller compared to the other two cases.

With increasing Reynolds number the distance which is necessary to form the turbulences is increasing, too (see figure 4.5). This is a disadvantage for the 8 mm spring because the effective distance is larger than the pitch anyway. Thus, the range in which the turbulence is increased in maximum, is smaller compared to lower Reynolds numbers. The linear course turns into an underproportional. In contrast to the 8 mm spring the effective distance for the 16 mm spring can still be increased. Due to these reverse effects - increase of the area of flow development behind the wire and increase of effective length - the linear course of its characteristic continues. The same effect occurs with the 24 mm spring, only on a lower level.

The observed heat transfer behaviour suggests that in case of applying pure water, no general optimum design concerning the pitch of the springs can be reached. The optimum value is dependent on Reynolds number. Therefore a spring that is optimal over the complete range of flow velocity can not be found.

In this theory the increase in heat exchange area due to the spring is neglected. This probably will improve the heat transfer behaviour. Considering the marginal area (or better: volume) of the spring and the fact that the spring only touches the wall with an infinitesimal part of its area, this effect can be neglected.

4.1.2.2 Results with surfactant solutions

Figure 4.6 shows the Nusselt characteristics of Habon-G solutions of different concentrations and for water with an entrance temperature of 55 °C. These measurements have been carried out with the 8 mm spring. Considering the concentrations 125 and 250 wppm the typical behaviour of drag reducing additives can be observed. At low Reynolds numbers a heat transfer reduction (compared to water with obstacles) can be seen. In this range the characteristic is almost identical to that one for water without obstacles. With increasing flow velocity the Nusselt numbers are rising and reach - at a critical Reynolds number - the values for water with obstacles. This critical Reynolds number is increasing with the concentration.

At high Reynolds numbers (above 70,000) the values for 125 and 250 wppm are slightly above those for water with obstacles. This effect may be result from the micellar structures in the solution which influence the turbulence near the wall.

Considering the surfactant solution of 500 wppm the behaviour is completely different from the other solutions. The characteristic runs ca. 15 % to 25 % below the curve for pure water without obstacles. A critical value on which the characteristic rises to the values for water with obstacles can not be seen.
The stability of the shear induced state is increasing with the concentration [8]. It is possible that the critical wall shear stress is not be reached. Considering the last measuring point which shows a tendency to increase to higher Nusselt numbers the critical wall shear stress can be expected slightly behind this point.

![Graph](image)

**Fig. 4.6:** Nusselt characteristic for water and drag reducing surfactant solutions of different concentrations with spiral springs ($s = 8 \text{ mm}$)

An other explanation could be the flexibility of the rod like micelles. Due to the fact that the rod like micelles grow with increasing concentration [3] their flexibility - and thus their ability to produce turbulences near the wall - will decrease.

To find out whether a critical wall shear stress will be reached measurements at higher flow velocities have to be carried out. With the above described test rig, those measurements can not be carried out without modifications.

**Figure 4.7** shows the dependence of the Nusselt characteristic for different temperatures. The measurements have been carried out with the 24 mm spring for a 125 wppm solution. With increasing temperature the characteristics are moved to lower Nusselt numbers - or better: are moved to higher Reynolds numbers. This effect is mostly caused due to the physical properties which are used to calculate the Reynolds number. With increasing temperature, the viscosity is decreasing much stronger than the density of water. Thus the Reynolds numbers for identical flow velocities are increasing with the temperature.

In **figure 4.7** the typical course of the characteristics (see also figure 4.6) can be seen. At a certain Reynolds number the characteristic is bending to higher values and reaches the corresponding curve of water (this can only be seen for the entrance temperature of 70 °C - for the reason of clarity, the water curves for the other temperatures have not been shown).

The characteristic for the temperature of 82 °C runs parallel to the water curve. Compared to the corresponding water curve for 82 °C, the characteristics are almost identical. In this case the critical wall shear stress is reached and no effect due to the drag reducing additives occurs.
Comparing figure 4.7 with figure 4.6 it is obvious that the critical Reynolds number is moved to higher values, if the pitch of the spiral springs is increased. The comparison of the different pitches is given in figure 4.8.

**Fig. 4.7:** Nusselt characteristic for water and drag reducing surfactant solutions of different temperatures with spiral springs (c = 125 wppm, s = 24 mm)

**Fig. 4.8:** Nusselt characteristic for water and drag reducing surfactant solutions for spiral springs of different pitches (c = 250 wppm, T\textsubscript{in} = 55 °C)
In figure 4.8 the dependence of heat transfer of cationic surfactant solutions (Habon-G, 250 wppm) on the pitch of the installed spiral springs is shown. As expected the heat transfer improvement is decreasing with increasing pitch of the obstacles. Furthermore the critical Reynolds number is increasing with increasing pitch. Thus for the 24 mm spring the values for water are not reached.

The explanation for this behaviour is that the ability to build and keep up the shear induced state is increasing with the pitch. Due to the higher turbulences and therefore the higher wall shear stress in case of applying springs with smaller pitch, the critical wall shear stress will be reached at lower flow velocities.

For the results in figure 4.8 ($T_{in} = 55 \, ^{\circ}C$, $c = 250 \, \text{wppm}$) the heat transfer improvement HTI according to equation 4.4 has been calculated and presented in figure 4.9. As mentioned above the function with obstacles and surfactants must reach at least the heat transfer of water without obstacles to fulfil the necessary requirement for a technical application. That means the heat transfer improvement has to be at least 0 or bigger. This is characterized by the hatching.

![Graph showing heat transfer improvement for drag reducing surfactant solutions with spiral springs of different pitches](image)

**Fig. 4.9:** Heat Transfer Improvement for drag reducing surfactant solutions with spiral springs of different pitches ($c = 250 \, \text{wppm}, T_{in} = 55 \, ^{\circ}C$)

In the investigated range of flow velocity the values of heat transfer improvement for the 24 mm spring are always below the demanded value of 0. On the other hand an avarage heat transfer reduction of 30 % (or a negative heat transfer improvement of -30 %) does only result in a reduction of overall heat transfer of 15 % to 20 % (see figure 2.5) and in an even smaller heat output reduction [10]. But nevertheless, the operating conditions for drag reducing additives may not be worse compared to those for water - especially on the field of heat transfer.

Considering the other obstacles the conditions for the 8 mm spring are significantly better. For Reynolds numbers above 26,000 the HTI is above 0 and therefore the requirement concerning the heat transfer is fulfilled. For the 16 mm spring this condition is fulfilled for Reynolds numbers bigger than 56,000.
Concerning the technical application only the 8 mm spring can be used for a concentration of 250 wppm if the Reynolds number is above 20,000 (the heat transfer reduction of 2% to 4% in the range between 20,000 and 26,000 can be neglected).

Considering the concentration of 500 wppm the heat transfer is always worse than that for water without spiral springs. A compensation for the effect of heat transfer reduction cannot be achieved.

In figure 4.10 the Nusselt characteristics for 125 wppm for different pitches are compared to the reference values for water without springs.

Fig. 4.10: Nusselt characteristic for drag reducing surfactant solutions for spiral springs of different pitches (c = 125 wppm, T$_{in}$ = 55 °C) compared with water without obstacles

In this case the conditions are quite better compared to the results for 250 wppm. For 8 mm the heat transfer is better than the required reference values over the whole range of Reynolds numbers. Even the 16 mm does almost fulfill the required condition. Only at Re < 27,000 the values are slightly below the reference curve (HTI > -5%).

Considering the 24 mm spring the requirements are fulfilled for Re < 47,000. For smaller flow velocities a HTI of -15% in maximum appears. Whether this heat transfer reduction - compared to the origin state - can be tolerated has to be investigated in exact calculations.

Furthermore for a technical application, a concentration of 125 wppm could be too small to reach an optimum drag reduction in the pipe system. The results presented in figure 4.7 show that the effect of the surfactants is decreasing significantly with the temperature. For an entrance temperature of 82 °C, the effect has almost disappeared.

Another important aspect is the behaviour of pressure drop inside pipes with obstacles. An increase in heat transfer can only be achieved by an increase in pressure drop. A technical application of the spiral springs is only meaningful if the increase in heat transfer does not lead to an unacceptable pressure drop. In chapter 4.2 the results of these investigations are presented.
4.2 Pressure drop

Analogous to the measurements of the heat transfer reference measurements without spiral springs have been analysed. In figure 4.11 results of the pressure drop behaviour are presented. In this figure drag characteristics inside the smooth pipe as well as with the 8 mm spring are presented for water and drag reducing surfactant solutions (Habon-G, 250 wppm).

Fig. 4.11: Drag characteristic for water with spiral springs (s = 8 mm)

Considering the pipe without obstacles a significant drag reduction of more than 70% can be observed. The critical wall shear stress could not be reached. Altogether the typical behaviour of drag reducing surfactant solutions inside straight pipes can be observed.

Comparing the measurements for water with and without obstacles a significant increase of drag coefficient can be regarded. The pressure drop is 600 to 700% higher due to the installation of the 8 mm spring.

This confirms the expectation of a strong increase in pressure drop. The increase of heat transfer of about 90% to 100% is accompanied by a superproportional increase in pressure drop. Analogous to the heat transfer improvement, the pressure drop increase (PDI) is defined:

\[
PDI = \frac{\Delta p_S - \Delta p_w}{\Delta p_w} \cdot 100 \% 
\]

or

\[
PDI = \frac{\xi_S - \xi_w}{\xi_w} \cdot 100 \%,
\]

with: Index S: „Spring“ and
index w: „water, (no obstacles)“.

The definition PDI also uses the „origin state“ of the heat exchanger (water, no obstacles) as reference value. The value \(\Delta p_S\) or \(\xi_S\) can stand either for water or for surfactant solution.
The influence of the concentration on the drag characteristic for a 8 mm spring is presented in figure 4.12. Analogous to the heat transfer behaviour the drag reducing effect is extended to higher Reynolds numbers with increasing concentration (increase in critical wall shear stress). The critical value is the local minimum of the drag characteristic at low Reynolds numbers (e.g. 17,000 for 125 wppm, 20,000 for 250 wppm).

Considering the curves at lower Reynolds numbers the drag-coefficients are increasing with the concentration. This effect results from the increase in viscosity with increasing concentration. For Reynolds numbers above the critical value, the characteristics of the 125 wppm and the 250 wppm are approximately identical. They are 10 % to 15 % above the values for water. This corresponds with the higher Nusselt numbers of the surfactant solutions in this range [6].

The drag coefficients of the 500 wppm solutions are in the range Re < 30,000 above the curve for water. This is caused by the higher viscosity of this solution. For Re > 30,000, the drag reducing effect (compared to the case: water with springs) can be seen. The values are below those of water and reach a drag reduction (with the water value with springs as reference) of 20 % in maximum.

Figure 4.13 shows the drag characteristics for the 24 mm spring for different entrance temperatures and a concentration of 125 wppm. The same measurements which are presented in figure 4.7 have been evaluated. It can be seen that the effect of drag reduction is moved to higher Reynolds numbers with increasing temperature. The same effect that has been received in the Nusselt characteristic can be seen in the drag characteristic for a temperature of 82 °C. According to the relatively low concentration the critical wall shear stress is low - the drag reducing effect is not very distinctive.

---

**Fig. 4.12:** Drag characteristic for water and drag reducing surfactant solutions of different concentrations with spiral springs (s = 8 mm)

Analogous to the heat transfer behaviour the critical values are - compared to the 8 mm spring in figure 4.12 - moved to higher Reynolds numbers. The increase in turbulence and therefore the increase in wall shear stress due to the spiral springs is decreasing with rising pitch. Due to this, the critical wall shear stress moves to higher Reynolds numbers when increasing the pitch.
In figure 4.14 the drag characteristics for springs of different pitches for water and surfactant solutions of 250 wppm at an entrance temperature of 55 °C are presented. Analogous to the Nusselt characteristic (decrease in Nusselt with increase in pitch) the drag coefficients decrease with the pitch of the spiral spring. The critical values are as well increasing with the pitch. The critical value of the 24 mm spring could not be reached. The local minimum of the 16 mm spring is not very distinctive.

**Fig. 4.13:** Drag characteristic for water and drag reducing surfactant solutions of different temperatures with spiral springs (c = 250 wppm, s = 24 mm)

**Fig. 4.14:** Drag characteristic for water and drag reducing surfactant solutions for spiral springs of different pitches (c = 250 wppm, T_{in} = 55 °C).
Analog to figure 4.9 in figure 4.15 the pressure drop increase (PDI) is given as a function of Reynolds number. The increase in pressure drop is between 200% (24 mm spring) and more than 800% (8 mm spring). Therefore the increase in pressure drop is superproportional compared to the improvement of the heat transfer.

**Fig. 4.15:** Pressure drop increase for drag reducing surfactant solutions with spiral springs of different pitches (c = 250 wppm, T\textsubscript{in} = 55 °C)

**Fig. 4.16:** Pressure drop increase (PDI) and heat transfer improvement (HTI) for drag reducing surfactant solutions with spiral springs of different pitches (c = 250 wppm, T\textsubscript{in} = 55 °C)
In figure 4.16 the pressure drop increase (PDI) is compared with the heat transfer improvement (HTI). At higher Reynolds numbers (Re > 40,000) the superproportional increase of pressure drop which has to be spent to improve the heat transfer can be seen impressively. Even for the 24 mm spring the increase in drag reduction is up to 400 % although the „increase“ in heat transfer is negative (-30 %).

At lower Reynolds numbers (Re < 40,000) the behaviour is the other way round. The heat transfer improvement is increasing while the pressure drop increase (PDI) is decreasing. This is an effect of higher viscosity at lower Reynolds numbers.

Considering the values of pressure drop increase which correspond to a heat transfer improvement of more than 0 (requirement for technical application), the increase of pressure drop is at least 500 % (16 mm spring at Reynolds = 57,500).

Instead a drag reduction of about 70 % (see figure 4.11) in case of the straight pipe without obstacles, an increase of more than 500 % (up to 850 % for the 8 mm spring, \( c = 250 \ \text{wppm} \) and \( T_{in} = 55 ^\circ \text{C} \)) has to spend to reach the desired heat transfer behaviour.

### 4.3 Importance concerning technical application

Concerning the technical application of spiral springs to improve the heat transfer in case of using drag reducing additives the aim - the increase in heat transfer to values above those for water without turbulence increasing obstacles - can be reached, if applying the right obstacle.

Considering the condensers in Herning where the measurements for mechanical stability and corrosion have been carried out (see part two of this report) the range of Reynolds numbers is between 25,000 and 75,000. Assuming that a concentration of 250 wppm has to be used to reach a sufficient drag reduction the 8 mm spring has to be installed.

This means - in case of nominal heat load - that the pressure drop will increase of about 600 % to 850 %. The pressure drops for nominal heat load of the origin state is 0.8 bar (condenser 1) and 0.95 bar (condenser 2). If a 8 mm spring will be installed and 250 wppm Habon-G will be used, the pressure drop will increase enormously (6.8 bar apparatus 1 and 8.075 bar apparatus 2).

Even if a solution is accepted that does not fulfil the required conditions properly - e.g. a 24 mm spring and a concentration of 125 wppm, the increase in pressure drop will be about 250 %. Which solution will be chosen has to be determined in dependence of the system and its operating conditions.
In this study the heat transfer and pressure drop of drag reducing surfactant solutions inside straight pipes with obstacles has been investigated. Therefore an existing test rig, mainly consisting of two closed loops (one for cooling and one for heating) has been modified. As cationic surfactant Habon-G (hexadecyldimethylpolyoxethylammonia-cation and 3-hydroxy-2-naphthoate as counter-ion) has been applied.

Spiral springs of different pitches have been used as obstacles to increase turbulence and therefore the heat transfer which ordinarily is significantly decreasing when applying drag reducing additives (up to 95%). The springs consist of wire made of stainless steel of a diameter of 1 mm. The diameter of the spring has been a little bit higher than the inner diameter of the pipe to guarantee a certain support.

For pure water the installation of the obstacles leads to an increase in heat transfer of nearly 100% in maximum. The behaviour is dependent on the pitch. The characteristic for the 8 mm and the 16 mm springs are almost identical while the Nusselt numbers of the 24 mm spring are smaller.

For surfactant solutions the heat transfer behaviour strongly depends on concentration, temperature and pitch. For 500 wppm the improvement in heat transfer is only small. The conditions of the origin state (water without obstacles) - which is the requirement for a suitable operation with surfactants - cannot be reached with this concentration. The Nusselt characteristic is parallel below that one for water without springs.

In contrast to the 500 wppm solution, the 125 wppm and 250 wppm show a completely different behaviour. At low Reynolds numbers the characteristics start in the range of that one for water without obstacles and rise - at a certain Reynolds number - up to the values for water with obstacles. This critical value is dependent on concentration, temperature and pitch of the spring. With increasing pitch the critical point is moving to higher Reynolds numbers as well as with increasing concentration. With increasing temperature the critical values also move to higher flow velocities until a certain temperature which is dependent on concentration is reached. Behind this point, the values are moving to lower Reynolds numbers and the characteristic is moving to higher Nusselt numbers because the drag reducing effect is weakening.

For a concentration of 250 wppm the heat transfer conditions of the origin state can be reached with the 8 mm spring above Reynolds = 25,000 and with the 16 mm spring above 56,000. For 125 wppm the conditions of the origin state can be reached with all investigated springs (for the 24 mm spring the condition Re > 47,000 has to be fulfilled).

Therefore the aim to reach the heat transfer coefficients of pure water without obstacles could be reached under certain conditions.

On the other hand measurements of the pressure drop showed a significant increase. This increase is - compared to the increase in heat transfer - superproportional. The strongest increase could be observed for the 8 mm spring for 250 wppm. In this case the pressure drop is increasing of about 850%.
The course of the drag characteristic mainly shows the same dependencies as the Nusselt characteristics. The most important influence has the pitch of the spring. Comparing the 8 mm and the 24 mm spring for a 250 wppm surfactant solution the pressure drop increase of the 8 mm spring is 850 % in maximum compared to 200 % in maximum for 24 mm.

Considering the technical application and assuming that the heat transfer has to be at least as large as for water without obstacles an enormous pressure drop increase has to be spent. Thus for the technical application detailed calculations of the changed conditions of the complete system have to be carried out in order to check the installation of spiral springs inside pipes as a measure to improve the behaviour of tube bundle heat exchangers when using drag reducing additives.
### 6. Symbols

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<tr>
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<td>surfactant solution</td>
</tr>
<tr>
<td>S</td>
<td>Spring</td>
</tr>
<tr>
<td>w</td>
<td>water</td>
</tr>
<tr>
<td>W</td>
<td>wall</td>
</tr>
<tr>
<td>*</td>
<td>critical value</td>
</tr>
<tr>
<td>1</td>
<td>primary side/pipe side</td>
</tr>
<tr>
<td>2</td>
<td>secondary side/shell side</td>
</tr>
</tbody>
</table>

**Special symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>CMC I/II</td>
<td>first/second Critical Micelle Concentration</td>
</tr>
<tr>
<td>DH</td>
<td>district heating</td>
</tr>
<tr>
<td>DR</td>
<td>Drag Reduction</td>
</tr>
<tr>
<td>HE</td>
<td>Heat Exchanger</td>
</tr>
<tr>
<td>HOR</td>
<td>Heat Output Reduction</td>
</tr>
<tr>
<td>HTI</td>
<td>Heat Transfer Improvement</td>
</tr>
<tr>
<td>HTR</td>
<td>Heat Transfer Reduction</td>
</tr>
<tr>
<td>MID</td>
<td>Magnetic-Inductive Flowmeter</td>
</tr>
<tr>
<td>OHR</td>
<td>Overall Heat Transfer Reduction</td>
</tr>
<tr>
<td>PDI</td>
<td>Pressure Drop Increase</td>
</tr>
<tr>
<td>SIS</td>
<td>Shear Induced State</td>
</tr>
<tr>
<td>SoSal</td>
<td>Sodium Salicylate</td>
</tr>
<tr>
<td>wppm</td>
<td>weight parts per million</td>
</tr>
</tbody>
</table>

**Non-dimensional Parameters**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Re</td>
<td>( \frac{u \cdot d}{v} )</td>
</tr>
<tr>
<td>Nu</td>
<td>( \frac{\alpha \cdot d}{\lambda} )</td>
</tr>
<tr>
<td>Pr</td>
<td>( \frac{v}{a} = \frac{\eta \cdot c_p}{\lambda} )</td>
</tr>
</tbody>
</table>
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Project D

Improving the Heat Transmission Properties of Tube Bundle Heat Exchangers by Installing Obstacles inside the Pipes

D2 Testing of Obstacles in an Operating Heat Exchanger and Evaluation of the Overall Effect

ELSAMPROJEKT A/S
BRUUN & SØRESEN GROUP AS

December 1996
Preface

This report is part 2 of the project D “Improving the Heat Transmission Properties of Tube Bundle Heat Exchangers by Installing Obstacles inside the Pipes”. The previous part is written by Prof. Weinspach Thermische Verfahrenstechnik GmbH.

This part 2 contains the general introduction to the project, the reporting of the activities made in Denmark and the overall conclusion. Paragraph 3 has been written by Mr. Henning Andersen, Elsamprojekt while the other paragraphs were written by Flemming Hammer of Bruun & Sørensen Group AS.
1. Introduction

1.1 Purpose of the project

The purpose of this project was to state the effects of using turbulators in the shape of steel springs in the water tubes of tube heat exchangers applying drag reducing additives in the water - so called "smooth water".

Tube heat exchangers are generally used for the generation of district heating in combined heat and power plants. A previous theoretical project showed that the reduction of the heat transfer properties by the use of smooth water in such heat exchangers is very considerable and must be compensated for. It also indicated promising possibilities for the improvement by means of the springs applied in this project although increase of the pressure loss must be encountered.

Based on this knowledge, three main subjects have been focused on: Measurement of the effects on heat transfer properties and pressure loss, long time testing of metallurgic effects in a heat exchanger operating with normal water and evaluation of the overall effects on the combined generation of heat and power.

1.2 Summary

An existing test rig mainly consisting of a one tube heat exchanger and two closed loops at the University of Dortmund was modified to host the first part of this project. The additive Habon G, developed by Hoechst AG, which have been applied successfully in three cases in Herning, was used for the tests.

Spiral springs made of 1 mm stainless steel of different pitches were used as obstacles to increase the turbulence and hence the heat transfer. The diameter of the springs was made a bit larger than the inner diameter of the tube in order to achieve the fixation of the springs.

For pure water, the increase in heat transfer was at its maximum 100 % depending on the pitch. For smooth water the behaviour strongly depends on pitch, concentration of additive and temperature. For a concentration of 250 ppm, which was successfully used in previous demonstrations in Herning, the original state of heat transfer can be reached with the 8 mm spring at water velocities normally prevailing in the CHP-plant in Herning. However, the pressure drop in this case increased to some 850 %.

In one of the two heat exchangers used for generating district heating in Herning since 1982, 5 spirals were installed during the scheduled summer stop in 1995. After one year of operation - with normal water - the pipes containing the spirals were dismounted during the regular revision of the plant in August 1996. regarding pipes and springs both conventional and stainless steel was used.
The spirals had caused no difficulties during operation, they showed no signs of displacement in the tubes, and the stated light corrosion in the conventional steels were equally distributed. The stainless materials showed no sign of corrosion.

Two concrete combined heat and power plants were used as models for the calculations, so that the effects of the use of obstacles and smooth water could be quantified. The thermal couplings of these plants were slightly changed by "installing" a pump to handle the pressure loss through the heat exchangers only. Using the results achieved from the experiments in Dortmund a number of model calculations have been carried out, and are shown in the enclosures.

1.3 Conclusion

In the previous work, the fact that the pressure loss increases disproportionally was seen as a serious drawback of the use of turbulators.

The outcome of this project have shown that due to dramatically improvement of the heat resistance on the tube inside due to the use of obstacles, the total heat transfer is so much improved, that the steam can be expanded to a lower level, implying a higher electricity generation. This higher output of electricity is generally speaking able to compensate for the increased pumping demand through the heat exchangers.

It was also found that such obstacles in the shape of simple steel springs can be manufactured, installed and operated without any significant problems.

Therefore, the overall conclusion is that in respect of changing to the use of smooth water operation almost status quo as to the tube heat exchanger in CHP-stations is achieved!

An ongoing project will make use of this and all other results achieved during 10 years of activity within demonstration and development of smooth water for district heating operation. The project will comprise a cost/benefit analysis and is planned to be terminated by the end of 1997.

1.4 Project management

The project was carried out in a co-operation among the following three partners:

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Heinrich-Sträter-Straße 12
D-44229 Dortmund
Phone +49 0231 / 7 35 000
The practical tests of the springs were made in the combined heat and power plant in Herning, Denmark. The positive interest and readiness of the operational manager Mr. Bent Haurballe and his staff is highly appreciated.

I/S Vestkraft
Herningværket
Energivej
DK-7400 Herning
Phone +45 97 22 35 00

### 1.5 Financing

This project was co-financed by the following two programs:

IEA Advanced Transmission Fluids for District Heating and Cooling (R&D / IEA, Annex IV) and ELSAMs R&D programme.
2. Background

In the late 1980s the first full scale demonstrations of the use of friction reducing additives in district heating systems took place in Völklingen, Germany and Herning, Denmark. The reduction of the friction was considerable. Pumping power was reduced to approx. 50% and 25% respectively in the two systems [1, 2, 3]. This was a result of the reduced turbulence of the water.

However, it was also found that the heat transfer in the applied plate heat exchangers was reduced implying a higher flow rate in order to keep up with the heat output of the system. A project was set up in order to analyse this problem and to find ways of designing such heat exchangers in order to avoid or reduce this negative effect of smooth water. It was concluded that it is possible to compensate partly or almost completely by re-shaping the plate heat exchangers [4].

Use of smooth water only has an effect of significance in large straight pipes with a high load factor. This is normally not the case in distribution systems, but certainly in superior transmission systems, conveying large amounts of base load heat from i.e. combined heat and power plants to local distribution networks. Therefore smooth water will also have to circulate through tube bundle heat exchangers normally used in CHP-plants to cool the exhaust steam from the turbines by means of district heating water being heated in this way.

In a foregoing project [5] the expected effects on heat transfer conditions applying smooth water in such exchangers were investigated. It was found that without compensation, the reduction of the heat transfer would forbid the use of smooth water. It has the effect that the necessary transfer of heat from the condensation of the steam to the district heating water cannot take place, whereby the pressure in the exhaust casing will rise to a prohibitive level.

It was found that by the introduction of turbulators in the shape of webs or springs inside the water pipes, turbulence can be created and in this way increase the heat transfer rate sufficiently. The negative effects would be, however, that the pressure loss through the water tubes would increase.

In order to have these results verified and to test whether springs can be installed and will make no harm inside the tubes, this project was proposed.
3. Activities in Denmark

3.1 Testing of obstacles at "Herningværket"

General

Installation of built-in helical springs inside condenser tubes is a simple and economic attractive method to improve the total heat transmission of an existing shell and tube condenser. The only modifications this rebuilt implies, is the insertions of pre-stressed helical springs into the condenser tubes. That means, the original condenser design can be reestablished simply by removing the installed springs and this is an important factor for full scale testing on an operational plant.

The heat transfer resistance in condensers are often much greater on the internal waterside than on the external condensing surface. Reducing internal heat transfer resistance implies a substantial improvement of the overall heat transfer, because the thermal resistance are serial coupled.

If the internal fluid is a surfactant solution (smooth water), this internal heat transfer resistance is increased by a factor of 5-10 compared to a pure water solution. Therefore it is of utmost importance to improve the internal heat transmission to get an acceptable heat transfer performance.

The intention with this work is to demonstrate practical design possibilities and long term effects from built-in springs in a full scale test. The Heat and Power plant in Herning, Denmark has been chosen as the test plant and the built-in springs was planned to be installed in a limited number of tubes in district heating condenser no 1. The design of this condenser is shown in fig 3.1 and the tubes to be installed with helical springs are shown in fig 3.2.

All the tubes were originally made as black tubes in St 35.8, but because of wear and abrasion problems arising on the external tube surfaces, it has been necessary to replace the tubes in the two external boundary layers with tubes made of austenitic steel (Werkstoff 1.4462). With these two different tube materials it was obvious to install springs made of standard hard-drawn spring steel (Werkstoff nr 1.0600) and austenitic steel (Werkstoff nr 1.4310) to expose eventual corrosion problems in the widest sense.

The mounting and dismounting of the helical springs in full tube length (10300 mm) has to be unproblematic. The fixations of the installed springs have to be safe to avoid any operational problems caused by spring-detachment. An obvious way to solve these problems was to insert correct pre-stressed springs inside the tubes.

Spring-tube calculations

Simple calculations of pre-stressed built-in springs were initially carried out based on standard formulas for design of helical springs exposed to torsion. The pre-stressed springs have to match the condenser tube dimensions $d_i \times L = 12.6 \times 10300 \text{ mm}$. 
Applying material data for standard spring steel according to DIN 17223, the simple calculations gave a first-hand figure of the maximum allowable radial deflection achieved by twisting the spring. The calculations indicated, that it was possible to achieve a sufficient radial deflection to be able to mount and dismount the springs inside the tubes.

The simple helical spring calculations are only valid for small axial pitches. The spring pitch has most properly a strong influence on the internal heat transfer and pressure losses. It is possible to optimise the pre-stressing procedure by combining a twisting and a stretching of the springs simultaneously. Finally it is very important to be able to maximise the radial deflections without overloading and damaging the springs.

These reasons justify a refined accurate model of the pre-stressed spring-tube system. The refined model was developed based on a full 3D double curved beam element exposed to axial forces and torsion moment [7].

At present the model has been used successfully in the design phase of the spring-tube system. The model predicts accurately the necessary twisting numbers and eventually stretching measure to accomplish a specified radial gap. The model calculates the stress level and ensures that the spring is not overloaded. The model gives finally an estimate of the pre-stressed forces between the spring windings and the internal tube surface. This figure is used to evaluate the stability of the spring-tube grip under district heat operation.

**Selected springs, mounting and dismounting**

A parametric study encountering various spring thread diameters, pitches and thread material according to DIN 17223-17224 has been accomplished. In consultation with the helical spring manufacturer DK FJEDRE A/S, two spring systems have been selected to the planned experiments in Herning CHP-plant.

We have chosen a 1.0 mm hard-drawn spring steel wire Class D (Werkstoff nr. 1.0600) and a 1.0 mm stainless steel wire (Werkstoff nr. 1.4310) to be built in a tube with an internal diameter of 12.6 mm. The unloaded external spring diameter is 13.0 mm and the pitch is chosen to 12 mm. The external diameter of the pre-stressed spring is calculated to 11.75 mm before insertion - see fig 3.3.

A special tool was designed to pre-stress the springs correctly. The tool consists of a stiff adjoining helical spring attached to a drilling machine in one end and welded to a hold with a notch in the other end. The tool is inserted into the spring to be mounted. The spring to be installed is fixed to the notch in one end and counterbalanced in the other end at the drilling machine. The drilling machine, equipped with a revolution counter, twists the tool and thereby the installation spring in a pre-calculated number of revolutions. The installation spring has now the correct external diameter and are ready to be inserted into the condenser tube. The tool is shown in fig 3.3.. Dismounting is simple worked out in an opposite sequence.

**Results from the long term tests**

The springs were mounted in the summer revision 1995 and dismounted in the following summer revision 1996. After the condenser was opened the springs were found intact and correctly fixed. The springs and the related tubes were dismounted and subdivided into inlet, centre and outlet parts. The test material were marked and delivered to Mr. Knud Erik Poulsen from the Material Testing Department in the Danish Technology Institute.
The testing period was approximately 13 months with a normal district heating production. That means with reduced load in the summer period and up to max. load in the winter period (max. water velocity 2.1 m/s). The black tubes (St 38.8) have been exposed for corrosion since the start of the power plant in 1983. The austenitic tubes have first been installed in 1990.

The results from Mr K.E. Poulsen's corrosion examination [6] are restored below.

<table>
<thead>
<tr>
<th>Tube position</th>
<th>Tube material</th>
<th>Spring material</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. A inlet zone</td>
<td>Stainless steel W 1.4462</td>
<td>Stainless steel W 1.4310</td>
</tr>
<tr>
<td>2. A centre zone</td>
<td>Stainless steel W 1.4462</td>
<td>Stainless steel W 1.4310</td>
</tr>
<tr>
<td>3. C inlet zone</td>
<td>Steel St 35.8</td>
<td>Stainless steel W 1.4310</td>
</tr>
<tr>
<td>4. C centre zone</td>
<td>Steel St 35.8</td>
<td>Stainless steel W 1.4310</td>
</tr>
<tr>
<td>5. D inlet zone</td>
<td>Steel St 35.8</td>
<td>Spring steel W 1.0600</td>
</tr>
<tr>
<td>6. D centre zone</td>
<td>Steel St 35.8</td>
<td>Spring steel W 1.0600</td>
</tr>
<tr>
<td>7. D outlet zone</td>
<td>Steel St 35.8</td>
<td>No spring</td>
</tr>
</tbody>
</table>

The tubes have been examined for corrosions with the following results:

<table>
<thead>
<tr>
<th>Tube position</th>
<th>Weight of corrosion products</th>
<th>Weight of tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>A inlet+spring</td>
<td>~0 g/m</td>
<td>425.0 g/m</td>
</tr>
<tr>
<td>A center+spring</td>
<td>~0 g/m</td>
<td>425.0 g/m</td>
</tr>
<tr>
<td>C inlet+spring</td>
<td>8.55 g/m</td>
<td>384.7 g/m</td>
</tr>
<tr>
<td>C center+spring</td>
<td>12.45 g/m</td>
<td>388.7 g/m</td>
</tr>
<tr>
<td>D inlet+spring</td>
<td>11.38 g/m</td>
<td>379.3 g/m</td>
</tr>
<tr>
<td>D center+spring</td>
<td>18.86 g/m</td>
<td>375.7 g/m</td>
</tr>
<tr>
<td>D outlet and no spring</td>
<td>11.41 g/m</td>
<td>380.1 g/m</td>
</tr>
</tbody>
</table>

The weight loss of black tubes made of St 35.8 corresponds to an approximate average thickness on 0.1 mm and cover existing gabcorrosions on the internal and external tube surfaces. The maximum gabcorrosion amounts 0.25 mm.

The size of corrosions on St 35.8 tubes are practical similar. It is not possible to see any significant differences of corrosions at the inlet, centre or outlet tube position. The spring materials reveal no visible differences in the corrosion pattern.

The corrosion of springs made of hard-drawn spring steel is modest.

There is not found any corrosions on the stainless springs and the stainless tubes.

**Conclusion**

The long term test demonstrates no operational and fixation problems. The tubes were found in exactly the same positions as they were mounted.

The results from the corrosion tests show no corrosion attacks on springs and tubes made of stainless steel. The hard-drawn spring-steel indicates a minor corrosion attack.

The black tubes show no significant corrosion pattern related to the spring assembly. The major corrosion attacks are most properly caused by the long time exposure to the water in district heating system and the extraction steam. These tubes have been installed since 1983.
It has been demonstrated that long springs in oversize \( (d_e = 13 \text{ mm}, L = 10300 \text{ mm}) \) can be installed into long condenser tubes \( (d_i = 12.6 \text{ mm}, L = 10300 \text{ mm}) \) without any serious problems.

We must conclude that stainless steel springs are the best solution, because corrosion is negligible. Even if the tubes are made of ferritic steel (St 35.8) it is still recommendable to use springs in stainless steel.

It is important that the pre-stressing of the springs before and after insertion is correctly carried out. Any operational problems caused by detached springs are unacceptable.

### 3.2 Application of results obtained at the test rig in Dortmund

**Models**

Two typical Heat and Power Plants are simulated to operate with drag reducing additives in district heating water and with helical springs installed inside tubes in district heating condensers - these systems are called modified. The original heat-balances are established, so it is possible to make comparisons and evaluations between the modified and the original systems.

Original and modified power plant systems are described below.

1. **Plant:** MKS unit 3/4, Århus, Denmark (Reference Plant).
   - Design: Extraction type with separate seawater cooled condenser.
   - District heating system: Original system applying pure water.

2. **Plant:** MKS unit 3/4, Århus, Denmark.
   - Design: Extraction type with separate seawater cooled condenser.
   - District heating system: Modified system applying drag reducing additives.
   - Internal helical springs installed.

3. **Plant:** Herning CHP-plant, Denmark (Reference Plant).
   - Design: Backpressure type (electricity and heat stiffly bound).
   - District heating system: Original system applying pure water.

4. **Plant:** Herning CHP-plant.
   - Design: Backpressure type.
   - District heating system: Modified system applying drag reducing additives internal helical springs installed.

The thermal couplings of the power plants are shown as heat-balances found in the appendix. The couplings are slightly changed compared to the original ones. It has been necessary to add an extra district heating pump "PDH" to be able to compare the results. The pump "PDH" equalises the district heating outlet pressure with the inlet one and the pump-power is then added to the internal power consumption. That means net electric powers, net efficiencies and \( \text{Cm/Cv} \)-values are direct comparable.
The results from part D1 in this report "Investigations of heat transfer and pressure drop" have been used to describe internal heat transfer and pressure drop in the modified district heating condensers.

According to chapter 4 in part D1 it should be possible to increase the internal heat transfer coefficients of the modified district heating system with approximately 100% compared to the original systems. It is recommended to use a formula from weber [8] to compute the internal heat transfer coefficients for condensers using drag reducing additives and installed with helical springs.

The drawbacks we get by provoking turbulence inside the tubes is a substantial pressure drop. In chapter 4, part D1 it is demonstrated that the pressure drops in the modified system amounts 700 - 800% compared to the pressure drops in the original system. We have to consider these substantial pressure drops in the power plant models, so therefore we simply multiply the original pressure drop by a factor of 8. The original pressure drop is determined by means of Colebrook's formula.

**Results for Power Plant MKS unit 3/4.**

Power Plant MKS unit 3/4 were originally designed for high district heating temperatures of 70-125 °C. In the meantime the tendencies have been to lower the district heating temperature level to improve efficiencies. A typical lower level district heating production is carried out at temperatures 55-115 °C.

A number of heat-balances at 100% load have been carried out for both the original and the modified model of the power plant MKS unit 3/4. The results are represented as graphical heat-balances in the appendix. The main key-data for these calculations are shown in table 3.2.1 (low district heating temperatures 55-115 °C) and in table 3.2.2 (high district heating temperatures 70-125 °C).

At high district heating temperatures the calculations show positive results. The net electrical power as function of district heating is generally improved with a max. value of 430 kW. The Cv-value (defined in the enclosure) is generally reduced with a max. value of 1%.

At low district heating temperatures the calculations show negative results. The net electrical power as function of district heating is generally reduced with a maximum value of 1280 kW. The Cv-value is generally increased with a max. value of 1.5%.

The reason to this negative shift in low temperature region is due to increased losses in the outlet ends of IP turbine. The turbine was original designed to relative high extraction pressures and temperatures to feed the district heating system. By decreasing the district heating temperature level the volume flows are increased and the last stages in the turbine are additional loaded. This tendency is increased further by using helical springs in district heaters, because of the improved heat transmissions.

It has been discussed to replace a number of the last stages in the IP turbine to meet the demand for operating with even lower district heating temperatures. In this case it is most properly, that the plant will show the same positive or neutral tendencies as we found for operating at high temperature level.

It is interesting to see that the additional pumping power caused by the great pressure drop in the modified system, is regained in an increased expansion in the turbines primary due to the improved heat transmissions in district heaters.
Results for the CHP-plant in Herning.

The plant in Herning was designed for district heating temperatures of 55-90 °C. Today the plant operates at lower temperature levels - typical 42.6-85.6 °C.

A number of heat-balances have been carried out for both the original and the modified model of the plant. The results are represented as graphical heat-balances in the appendix. The main key-data for these calculations are shown in table 3.2.3 (low district heating temperatures 42.6-85.6 °C) and in table 3.2.4 (high district heating temperatures 55-90 °C).

At high district heating temperatures the calculations show positive results at 25%, 50% and 75% boiler-load. At 100% boiler-load the results are negative. The net electrical power as function of gross boiler-load shifts between 200 kW extra and 350 kW lesser power. The Cm-value (defined in the enclosure) shifts between +0.7% and -0.6%. The pressure drop is increased from 1.1 bar to 8.6 bar at 100% boiler-load.

At low district heating temperatures the calculations show exclusively positive results. The net electrical power as function of gross boiler-load is improved at all gross boiler-loads. The max. increase of net electrical power amounts 363 kW. The Cm-value is raised to a maximum of 1.1%. The pressure drop is increased from 0.7 bar to 5.5 bar.

The success here is due to the increased temperature difference and thereby reduced massflow in district heating condensers. The pressure difference over the modified condensers are reduced from 8.6 bar at high temperature operation to 5.5 bar at low temperature operation at 100% boiler-load.

A thorough study of heat-balances shows modest flow-losses in the Herning Plant. This indicates a greater robustness against improved condenser capacity.

The turbine has no symmetric cylinders and is exclusively balanced by means of an axial piston. If the pressures in the turbine are changed perceptible it is necessary to consult the turbine manufacturer.

Conclusions

Despite the great pressure drops and thereby extra pumping power we experience, when helical springs are installed into the district heating condenser tubes, it is surprising that the improved heat transmission cause such an extra expansion in the turbines, that the overall effects are practical neutral. The calculations demonstrate increased efficiencies at low temperature district heating and especially here we have to consider the turbines carefully if we try to expand the steam further.

The calculations are based on the work described in part D1. The experiments here deal with a single helical spring with fixed geometrical measures. It is most likely that the heat transmissions and pressure drops can be optimised by choosing another geometry of the obstacle.

The results from the present investigation indicates, that it should be possible to improve power plant efficiency, if we use an optimized obstacle inside the district heating condenser tubes. Anyway if we neutralize the power plant efficiency as demonstrated here, we still reduce the district heating transmission losses with 70-80% compared to traditional district heating systems.
Finally if the pressure drop become too great, it is obvious to check the possibility for raising the
district heating temperature difference and thereby reduce the massflow and then the pressure
drop. This step can very well improves the efficiency as indicated under "Results for Heat and
Power Plant Herning".
4. Outlook

The most conspicuous effect of smooth water is the reduced demand for pumping energy. However, it has to be assumed to be even more important that the capacity of existing networks can be increased due to the possibility of increasing the flow rate. This is a valuable alternative to the building of new expensive networks in those cities having a fully utilized network and being in need of increased capacity.

Another aspect is the possible reduction of the forward temperature in order to reduce heat loss from the network and to improve the efficiency of power production from CHP-plants. The water velocity in the network, which necessarily has to be increased if the differential temperature falls, will be a limiting factor. The application of smooth water has the potential of solving this particular problem.

In case of building new networks, savings may be obtained by using smaller pipe dimensions.

In connection with restructuring of old systems and implementation of CHP on regional levels, it will be likely to build new transmission systems to supply old distribution systems. Also in these cases - which will be relevant in many Central and Eastern European city areas - the utilization of smooth water is an obvious chance of reducing initial costs.

A number of R&D and demonstration projects have taken place during the past 10 years aiming at the goal that friction reducing additives should be applied in hydraulically isolated heat transmission networks in order to save energy for pumping.

In a co-operation between Herning Kommunale Værkert (the public utilities of Herning), CTR (the Metropolitan Copenhagen Heating Transmission Company) and Bruun & Sørensen Group AS a project is now under development, which will apply the results of previous projects and set up models for the exploitation of smooth water in the transmission networks of Herning and Copenhagen respectively. In Herning a reduction of pumping costs is aimed at, while in Copenhagen an extension of the output of the present network is the objective. It is the intention to set up a feasibility study in the two cases and also discuss the effects as to energy consumption and advantages / risks to the environment. This project is co-financed by the participants and the R&D-programme of the Danish Ministry of Energy.

This project will be terminated by the end of 1997.
5. References

1. Einsatz mizellarer Reibungsminderer in Fernwärmesystemen
   Abschlussbericht 1. Versuchsphase Sommer 1988 bei der Fernwärme Verbund Saar GmbH

2. Reduction of friction and pumping power in a district heating network applying surfactants to the circulating water.
   EC Energy demonstration project (no. EC 261 / 89 DK-DE). The report was submitted in June 1990.

3. Reduction of friction and pumping power in a district heating network applying surfactants to the circulating water II.

4. Performance of plate heat exchangers operating with friction reducing additives.
   Carried out by Bruun & Sørensen Group AS under the IEA Programme of Research, Development and Demonstration of District Heating and Cooling, Annex III. The final report was submitted by Novem (NL) in March 1993.


6. Undersøgelse af turbolatorer i glatvandsanlæg
   K.E. Poulsen, DTI, Århus, Denmark

7. Matrix Methods of Structural Analysis, 2. edition, Chapter 3.6
   P.K. Livesly, Pergamon Press

8. Wärmeübergang und Druckverlust wässriger Tensidlösungen in Rohren un Rohrwendeln.
6. Appendix

(District heating condenser, calculated key data and heat-balances for MKS Power plant unit 3/4, Århus and Heat and Power Plant, Herning, Denmark)
Number of passes: 1
Number of tubes: 4600
Tube dimensions: Ø 15 x 1.2
Tube material: St 35.8 og St 37.8
A, B : STAINLESS STEEL TUBES, STAINLESS SPRING
C : BLACK TUBE ST. 35.8, STAINLESS SPRING
D, E : BLACK TUBES ST. 35.8, SPRING STEEL
Prestressed spring to be installed

Initial unloaded spring

Final mounted prestressed spring

Feed torsion spring.
External diameter ø 8
Wire diameter ø 2

The drawing is for assembly tools for internal springs and is not available to the public. It must not be used, copied, or handed to any third party without written permission from ELSAMPROJEKT A/S.
Number of passes: 2
Number of tubes: 3892
Tube dimensions: Ø y 18 x 0.8
Tube material: St S 18.8
### TABLE 3.2.1. CALCULATED KEY-DATA FOR MKS POWER-PLANT UNIT 3/4.

**Livesteam:** Load 100%, 286 kg/s, 240 bar, 535 °C.
**District heat:** Load 0-450 MJ/s at 55-115 °C.

---

Normal district heat operation - pure (desalinated) water.

- **Q-gross**: Gross boiler heat production.
- **Q-dh**: District heat production.
- **P-gross**: Gross electric production.
- **P-netto**: Netto electric production.
- **Cv**: Cv value is defined as the ratio between netto electric power-loss and district heat.
  \[ Cv = \frac{(P_{\text{netto}}(0) - P_{\text{netto}}(dh))}{Q_{\text{dh}}} \]
- **δp-dh**: Internal pressure loss over district heaters.
- **n-netto**: Power plant netto efficiency.

\[ n_{\text{netto}} = \frac{P_{\text{netto}}}{Q_{\text{gross}}} \]

<table>
<thead>
<tr>
<th>Q-dh (MJ/s)</th>
<th>P-gross (MW)</th>
<th>P-netto (MW)</th>
<th>Cv (MW/(MJ/s))</th>
<th>δp-dh (Bar)</th>
<th>n-netto (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>368.074</td>
<td>354.802</td>
<td>0.00</td>
<td>0.00</td>
<td>43.201</td>
</tr>
<tr>
<td>100.00</td>
<td>352.041</td>
<td>339.009</td>
<td>0.158</td>
<td>0.030</td>
<td>41.262</td>
</tr>
<tr>
<td>200.00</td>
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<td>321.234</td>
<td>0.168</td>
<td>0.140</td>
<td>39.071</td>
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<td>250.00</td>
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<td>0.172</td>
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<td>301.363</td>
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<td>0.500</td>
<td>36.630</td>
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<td>1.120</td>
<td>32.889</td>
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</table>

Modified district heat operation - smooth water and spirals.

<table>
<thead>
<tr>
<th>Q-dh (MJ/s)</th>
<th>P-gross (MW)</th>
<th>P-netto (MW)</th>
<th>Cv (MW/(MJ/s))</th>
<th>δp-dh (Bar)</th>
<th>n-netto (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>368.074</td>
<td>354.802</td>
<td>0.00</td>
<td>0.00</td>
<td>43.201</td>
</tr>
<tr>
<td>100.00</td>
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<td>0.157</td>
<td>0.230</td>
<td>41.273</td>
</tr>
<tr>
<td>200.00</td>
<td>334.041</td>
<td>321.112</td>
<td>0.168</td>
<td>1.070</td>
<td>39.058</td>
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<td>37.870</td>
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Absolute and relative key-data differences taken from modified district heat operation (smooth water spirals) and normal district heat operation (pure water).

<table>
<thead>
<tr>
<th>Q-dh (MJ/s)</th>
<th>δP-gross (MW)</th>
<th>δP-netto (MW)</th>
<th>δn-netto (%)</th>
<th>δCv (%)</th>
</tr>
</thead>
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<td>1.282</td>
<td>-0.486</td>
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</table>
### TABLE 3.2.2. CALCULATED KEY-DATA FOR MKS POWER-PLANT UNIT 3/4

Livesteam: Load 100%, 286 kg/s, 240 bar, 535 °C.
District heat: Load 0-440 MJ/s at 70-125 °C.

<table>
<thead>
<tr>
<th>Normal district heat operation - pure (desalinated) water.</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Q-gross</strong> : Gross boiler heat production.</td>
<td><strong>P-gross</strong> : Gross electric production.</td>
</tr>
<tr>
<td><strong>Q-dh</strong> : District heat production.</td>
<td><strong>P-netto</strong> : Netto electric production.</td>
</tr>
<tr>
<td><strong>Cv</strong> : Cv value is defined as the ratio between netto electric power-loss and district heat</td>
<td><strong>δp-dh</strong> : Internal pressure loss over district heaters.</td>
</tr>
<tr>
<td></td>
<td><strong>n-netto</strong> : Power plant netto efficiency.</td>
</tr>
<tr>
<td>$\text{Cv} = \frac{\text{P-netto}(O) - \text{P-netto}(dh)}{\text{Q-dh}}$</td>
<td>$\text{n-netto} = \frac{\text{P-netto}}{\text{Q-gross}}$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Q-dh (MJ/s)</th>
<th>P-gross (MW)</th>
<th>P-netto (MW)</th>
<th>Cv (MW/(MJ/s))</th>
<th>δp-dh (Bar)</th>
<th>n-netto (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>.000</td>
<td>368.074</td>
<td>354.802</td>
<td>.000</td>
<td>.000</td>
<td>43.201</td>
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<td>.040</td>
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<td>.810</td>
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<td>1.060</td>
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<td>280.216</td>
<td>267.270</td>
<td>.199</td>
<td>1.280</td>
<td>32.504</td>
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<table>
<thead>
<tr>
<th>Modified district heat operation - smooth water and spirals.</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Q-dh</strong> (MJ/s)</td>
<td><strong>P-gross</strong> (MW)</td>
</tr>
<tr>
<td>.000</td>
<td>368.074</td>
</tr>
<tr>
<td>100.000</td>
<td>351.344</td>
</tr>
<tr>
<td>200.000</td>
<td>333.801</td>
</tr>
<tr>
<td>236.300</td>
<td>327.386</td>
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<td>324.259</td>
</tr>
<tr>
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<td>312.766</td>
</tr>
<tr>
<td>350.000</td>
<td>301.251</td>
</tr>
<tr>
<td>400.000</td>
<td>290.206</td>
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<tr>
<td>440.000</td>
<td>282.449</td>
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<table>
<thead>
<tr>
<th>Absolute and relative key-data differences taken from modified district heat operation (smooth water spirals) and normal district heat operation (pure water).</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Q-dh</strong> (MJ/s)</td>
<td><strong>δP-gross</strong> (MW)</td>
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<td>100.000</td>
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<tr>
<td>200.000</td>
<td>.121</td>
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<td>250.000</td>
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<td>1.503</td>
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<td>1.887</td>
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<tr>
<td>440.000</td>
<td>2.233</td>
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</table>
TABLE 3.2.3. CALCULATED KEY-DATA FOR HERNING HEAT & POWER-PLANT.
Livesteam data: 110 bar, 525 °C.
District heat temperatures: 42.6-85.6 °C.

<table>
<thead>
<tr>
<th>Normal district heat operation - pure ( desalinated ) water.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q-gross : Gross boiler heat production.</td>
</tr>
<tr>
<td>Q-dh : District heat production.</td>
</tr>
<tr>
<td>P-gross : Gross electric production.</td>
</tr>
<tr>
<td>P-netto : Netto electric production.</td>
</tr>
<tr>
<td>Cm : Cm value is defined as the ratio between netto electric power and district heat</td>
</tr>
<tr>
<td>( Cm = \frac{P{-}\text{netto}}{Q{-}\text{dh}} )</td>
</tr>
<tr>
<td>( \delta p{-}\text{dh} : ) Internal pressure loss over district heaters.</td>
</tr>
<tr>
<td>n-netto : Power plant netto efficiency.</td>
</tr>
<tr>
<td>n-netto = ( \frac{P{-}\text{netto}}{Q{-}\text{gross}} )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Q-gross (MJ/s)</th>
<th>Q-dh (MJ/s)</th>
<th>P-gross (MW)</th>
<th>P-netto (MW)</th>
<th>Cm</th>
<th>( \delta p{-}\text{dh} )</th>
<th>n-netto (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>142.952</td>
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<td>44.166</td>
<td>42.210</td>
<td>.471</td>
<td>.188</td>
<td>29.528</td>
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<tr>
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<td>68.427</td>
<td>.525</td>
<td>.396</td>
<td>31.911</td>
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<td>.691</td>
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</table>

<table>
<thead>
<tr>
<th>Modified district heat operation - smooth water and spirals.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q-gross (MJ/s)</td>
</tr>
<tr>
<td>----------------</td>
</tr>
<tr>
<td>142.952</td>
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<tr>
<td>214.428</td>
</tr>
<tr>
<td>285.904</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Absolute and relative key-data differences taken from modified district heat operation ( smooth water spirals ) and normal district heat operation ( pure water ).</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q-gross (MJ/s) ( % )</td>
</tr>
<tr>
<td>---------------------</td>
</tr>
<tr>
<td>71.476 ( 25)</td>
</tr>
<tr>
<td>142.952 ( 50)</td>
</tr>
<tr>
<td>214.428 ( 75)</td>
</tr>
<tr>
<td>285.904 (100)</td>
</tr>
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</table>
TABLE 3.2.4. CALCULATED KEY-DATA FOR HERNING HEAT & POWER-PLANT.
Livesteam data: 110 bar, 525 °C.
District heat temperatures: 55 - 90 °C.

Normal district heat operation - pure (desalinated) water.

<table>
<thead>
<tr>
<th>Q-gross</th>
<th>Q-dh</th>
<th>P-gross</th>
<th>P-netto</th>
<th>Cm</th>
<th>δp-dh</th>
<th>n-netto</th>
</tr>
</thead>
<tbody>
<tr>
<td>MJ/s</td>
<td>MJ/s</td>
<td>MW</td>
<td>MW</td>
<td></td>
<td>Bar</td>
<td>%</td>
</tr>
<tr>
<td>71.469</td>
<td>49.107</td>
<td>17.423</td>
<td>15.997</td>
<td>.326</td>
<td>.086</td>
<td>22.383</td>
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<tr>
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<td>91.024</td>
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<td>.293</td>
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<td>91.288</td>
<td>.525</td>
<td>1.071</td>
<td>31.933</td>
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</table>

Modified district heat operation - smooth water and spirals.

<table>
<thead>
<tr>
<th>Q-gross</th>
<th>Q-dh</th>
<th>P-gross</th>
<th>P-netto</th>
<th>Cm</th>
<th>δp-dh</th>
<th>n-netto</th>
</tr>
</thead>
<tbody>
<tr>
<td>MJ/s</td>
<td>MJ/s</td>
<td>MW</td>
<td>MW</td>
<td></td>
<td>Bar</td>
<td>%</td>
</tr>
<tr>
<td>71.469</td>
<td>49.064</td>
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<td>16.039</td>
<td>.327</td>
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Absolute and relative key-data differences taken from modified district heat operation (smooth water spirals) and normal district heat operation (pure water).

<table>
<thead>
<tr>
<th>Q-gross</th>
<th>δQ-dh</th>
<th>δP-gross</th>
<th>δP-netto</th>
<th>δn-netto</th>
<th>δCm</th>
</tr>
</thead>
<tbody>
<tr>
<td>MJ/s</td>
<td>MJ/s</td>
<td>MW</td>
<td>MW</td>
<td>%</td>
<td>%</td>
</tr>
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<td>71.469</td>
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<td>.067</td>
<td>.042</td>
<td>.259</td>
<td>.350</td>
</tr>
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<td>.695</td>
</tr>
<tr>
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<td>.112</td>
<td>.604</td>
<td>.103</td>
<td>.154</td>
<td>.240</td>
</tr>
<tr>
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<td>.803</td>
<td>.356</td>
<td>.391</td>
<td>.587</td>
</tr>
</tbody>
</table>