Project A

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

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Contents

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 occure, 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.

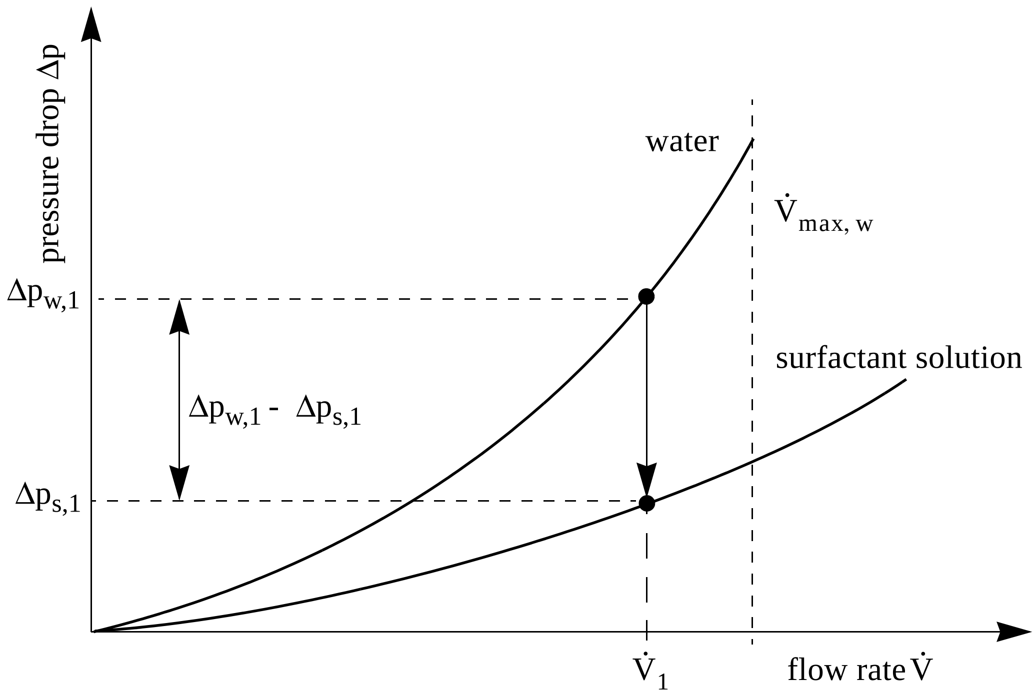


Fig. 2.1: Reducing pumping costs (pressure drop) [23]

With the definition of the drag reduction DR:

$$DR = \frac{\Delta p_w - \Delta p_s}{\Delta p_w}, \qquad \text{eq. (2.1)}$$

the equation for the pumping capacity P:

$$P = \eta_{tot} \cdot \dot{V} \cdot \Delta p, \qquad \text{eq. (2.2)}$$

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

$$PCR = P_w \cdot DR. \qquad \text{eq. (2.3)}$$

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.

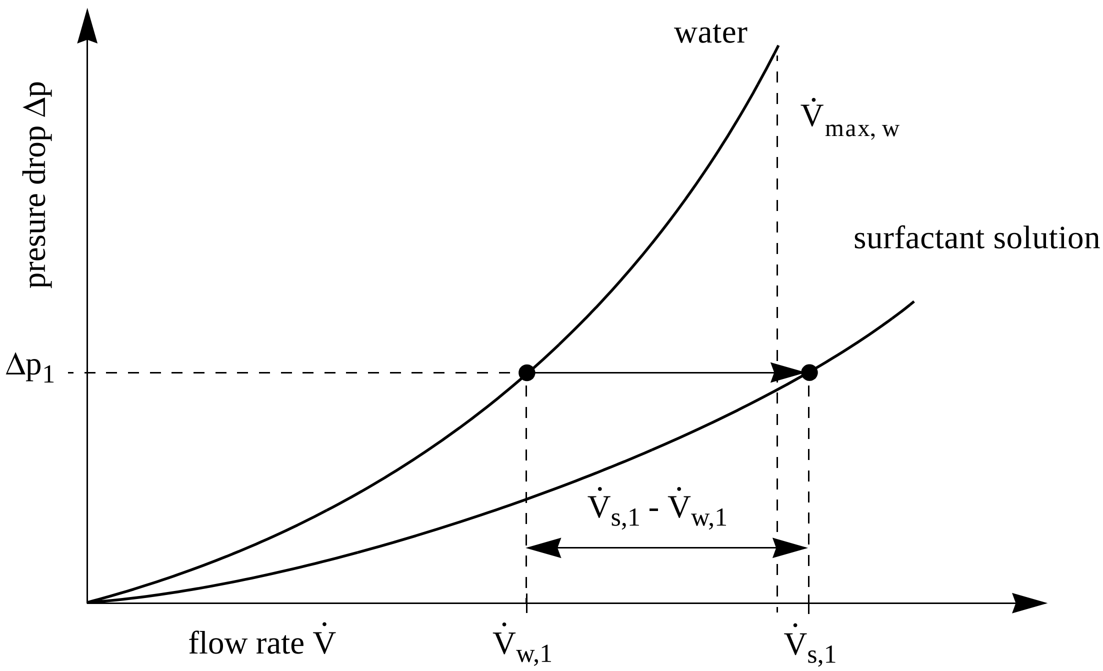


Fig. 2.2: Increasing the capacity [25]

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.

The characteristic curve for water can be described as follows [6]:

$$\Delta p_w = K_w \cdot v_w^2 . \qquad \text{eq. (2.4)}$$

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:

$$\Delta p_s = (1 - DR)K_w \cdot v_s^2 . \qquad \text{eq. (2.5)}$$

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:

$$v_{s,\,max} = v_{w,\,max} \cdot (1 - DR)^{-1/3} . \qquad \text{eq. (2.6)}$$

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

DR in [%]	factor of flow rate increase
50	1.26
55	1.31
65	1.42
75	1.59
80	1.71

Tab. 2.1: Maximum achievable flow rate

2.1.3 Shifting heat generation

Apart from the above described direct possibilities to apply surfactants (chapter 2.1.1and 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.

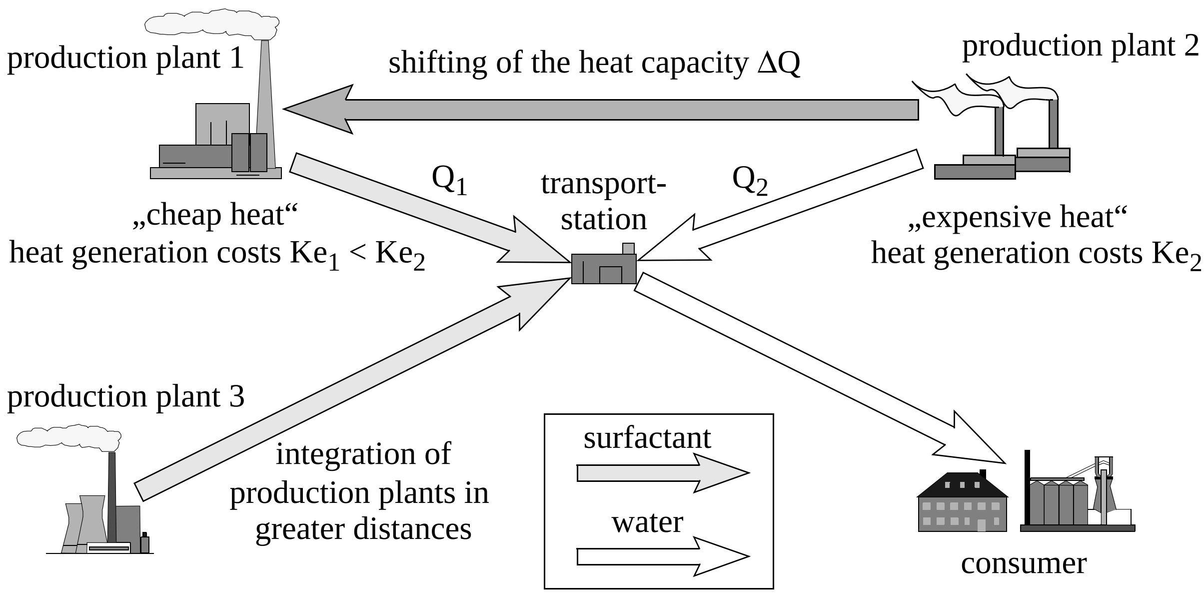


Fig. 2.3: Shifting of heat capacity/integration of production plants in greater distances

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:

$$\dot{Q} = \dot{M} \cdot c_p \cdot (T_{sup} - T_{ret}) \,, \qquad \text{eq. (2.7)}$$

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

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.

	$\dot{Q}$ [MW]	$\dot{M}$ [kg/s]	T_{sup} [°C]	T_{ret} [°C]	c_p [kJ/(kgK)]
water	100	398,7	130	70	4,18
surfactant		558,2	113		

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.

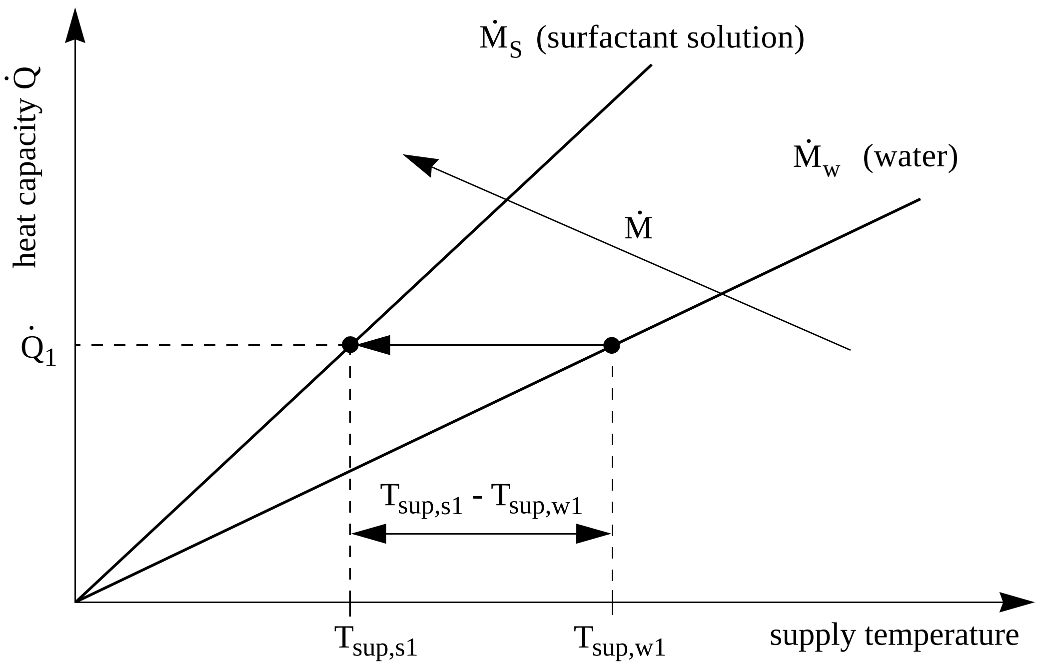


Fig. 2.4: Reducing the supply temperature [25]

In addition the heat loss will be decreased due to the reduced temperature between the supply and the ambient temperatur. 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**.

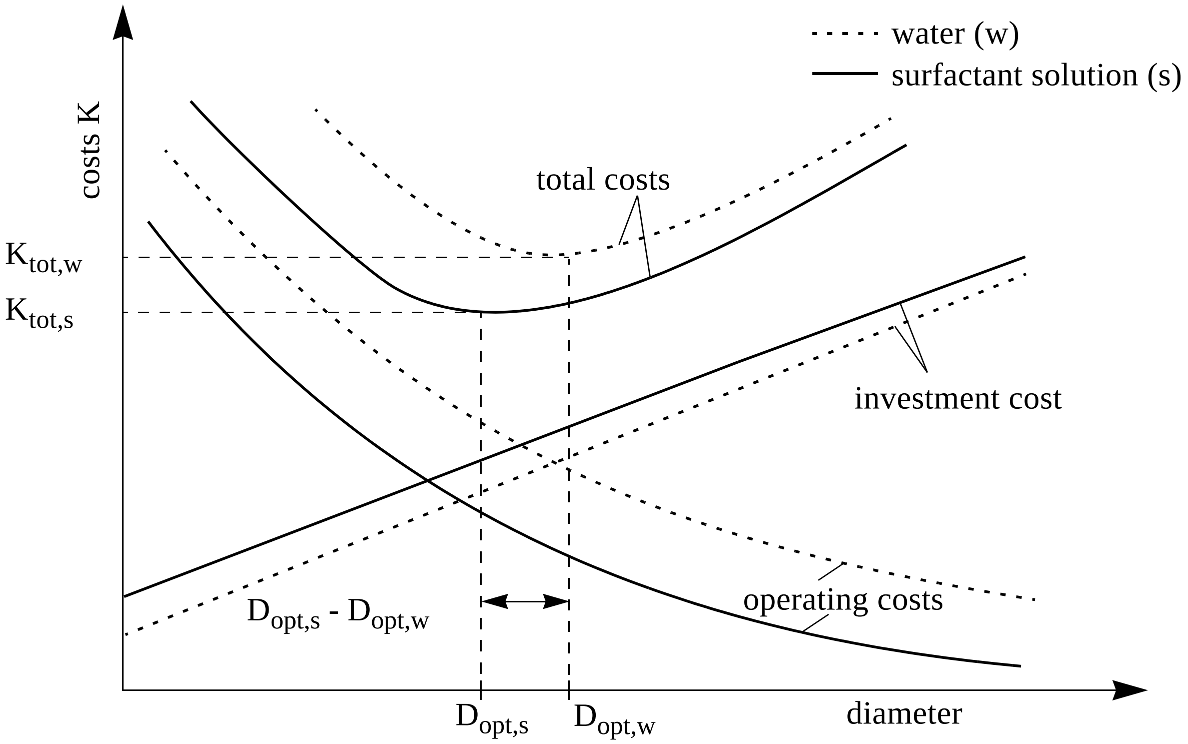


Fig. 2.5: Reducing the pipe diameter [25]

According to the conditions a reduction of 1 or 2 nominal diameter stages is possible [18].

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 sucessful.

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 ressources 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]:

$$\dot{Q} = k \cdot A \cdot \Delta T_m \qquad \text{eq. (3.1)}$$

with k = overall heat transfer coefficient,
A = heat transfer area and
ΔT_m = mean temperature difference of the overall heat transmission.

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

$$\frac{1}{k} = \frac{1}{\alpha_1} + \frac{s}{\lambda} + \frac{1}{\alpha_2} + R_F. \qquad \text{eq. (3.2)}$$

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/λ and, in case of fouling of the additional resistances (fouling factor) R_F.

The heat transfer coefficient α 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]:

$$Nu = \frac{\alpha \cdot D_i}{\lambda}, \qquad \text{eq. (3.3)}$$

with D_i = inner diameter of the pipe [m] and
λ = thermal conductivity of the fluid [W/(mK)].

The heat transfer reduction HTR can be described as the percental decrease of the heat transfer coefficient α (or Nu) compared to water (index w = water; s = surfactant) [9]:

$$\mathrm{HTR} \equiv \frac{\alpha_w - \alpha_s}{\alpha_w} \cdot 100\,\%. \qquad \text{eq. (3.4)}$$

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:

$$\mathrm{OHR} \equiv \frac{k_w - k_s}{k_w} \cdot 100\,\%. \qquad \text{eq. (3.5)}$$

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:

$$\mathrm{HOR} \equiv \frac{\dot{Q}_w - \dot{Q}_s}{\dot{Q}_w} \cdot 100\,\%. \qquad \text{eq. (3.6)}$$

An additional effect concerning the influence on heat exchangers is the increase of the temperature difference ΔT_m. The outlet temperature is increasing on the hot side ($T_{1,out}$) and decreasing on the cold side ($T_{2,out}$). 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.

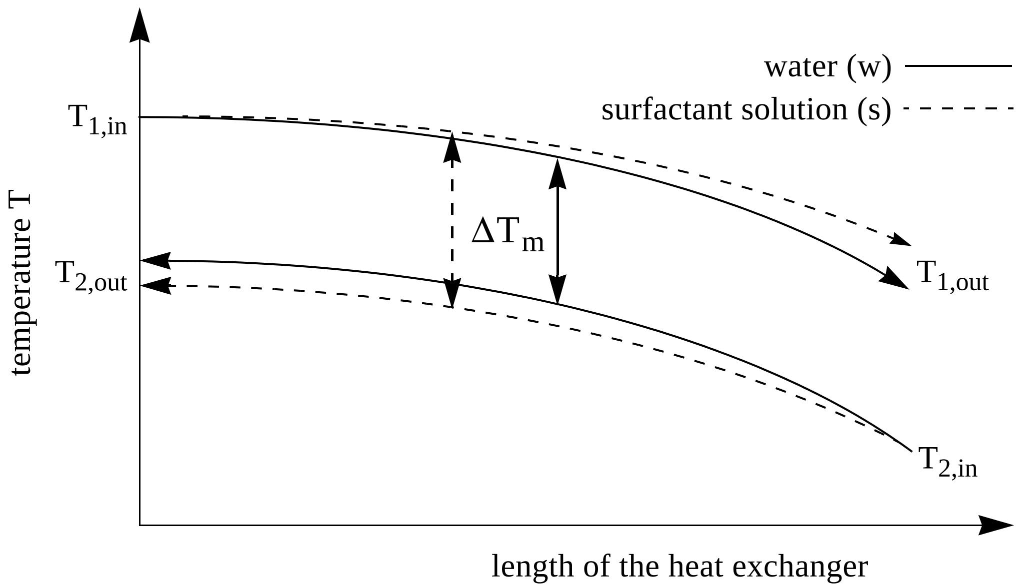


Fig. 3.1: Influence on the temperature difference [24]

Examplary values for HTR, OHR and HOR for a real district heating apparatus (plate heat exchanger) could be:

HTR = 60 % OTR = 40 % HOR = 10 %.

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 α-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 ω is introduced. This ratio between the two heat transfer coefficients is defined the following way [25]:

$$\omega \equiv \frac{\alpha_1}{\alpha_2} \ . \qquad \text{eq. (3.7)}$$

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

In addition to ω, the level of heat transfer reduction is a further decisive parameter that is determinig 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 ω 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.

type of HE	medium	ω	HTR	effect (OHR)
shell & tube	water/steam	0.5 - 3.0	75 - 95	strong
	water/water	1.0 - 4.0	75 - 95	strong
	water/gas	25 - 125	75 - 95	small
helical tube	water/water	0.6 - 4.0	45 - 70	medium
plate	water/water	0.5 - 2.0	45 - 70	medium

Tab. 3.1: ω and HTR range of typical heat exchangers

Combining the parameters ω and HTR, strong influences occure 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 ω and HTR on the overall heat transfer reduction OHR for typical apparatus under a certain operating condition (λ/s = 24,000 W/(m^{2}K), α_1 = 5,500 W/(m^{2}K), R_F = 0 (m^{2}K)/W).

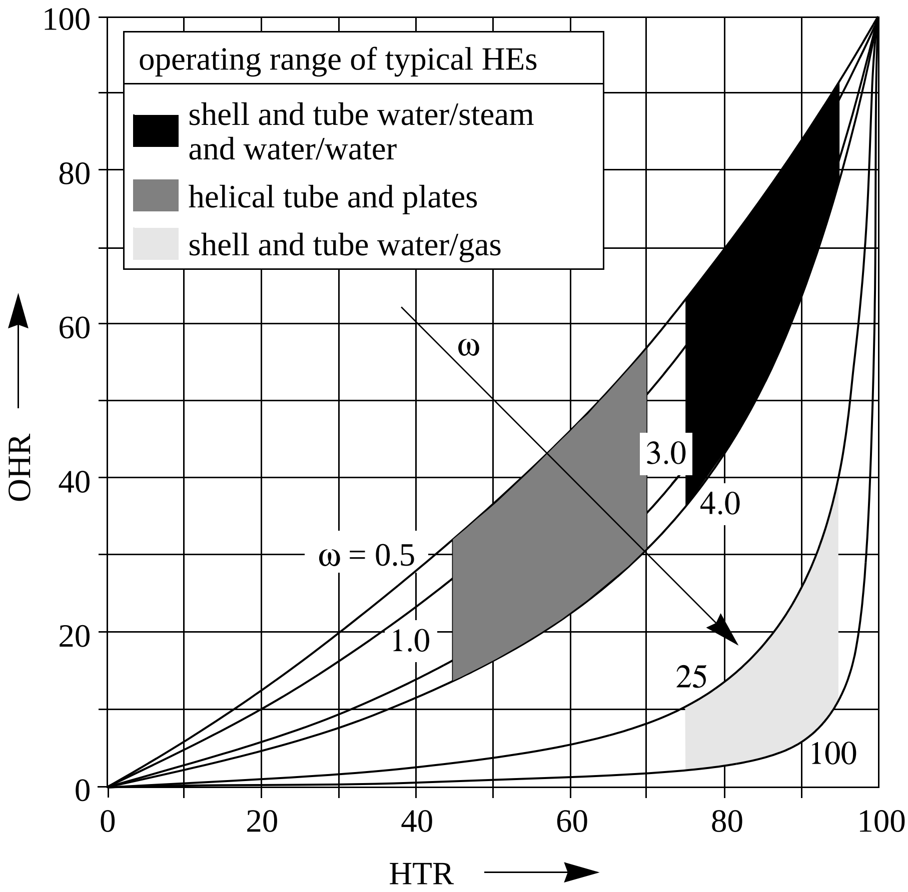


Fig. 3.2: OHR as a function of HTR and ω for typical DH-heat exchanger [25], (α_1 = 5,500 W/(m^{2}K), λ/s = 24,000 W/(m^{2}K), R_F = 0 (m^{2}K)/W)

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 ω. Values of OHR are only about 20 % if HTR is about 90 % and the values of ω 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 dependend 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·D_i). In very long tubes (L = 600 - 1,000·D_i) 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 comparision to this the HTR in very short pipes is only small, due to the high percentage of turbulent flow.

conditions		results	
		L/D_i	HTR
surfactant	Habon	80	77.7
concentration	1,000 wppm	142	83.8
Reynolds	20,000	600	88.5

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]:

$$Nu = \sqrt[3]{3.66^3 + 2.191 \cdot Re \cdot Pr \cdot \frac{D_i}{L}} \cdot \left(1 + \frac{L_e}{L}\right)^{0.249} \qquad \text{eq. (3.8)}$$

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

$$L_e = \dot{0}.075 \cdot D_i \cdot Re_W , \qquad \text{eq. (3.9)}$$

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 °C < T < 180 °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 3.1**.

design		nominal operation	
diameter of the pipes	23 mm	nominal heat load	50 MW
length of the pipes	6,000 mm	satur. steam temp.T_S	85.5 °C
number of pipes	1,800	entrance temp. T_{in}	60 °C
heat exchange area	814 m^2	outlet temp. T_{out}	68.4 °C
number of pass	1	mass flow (steam)	22.8 kg/s
appl. of surfactants	pipe side	mass flow (water)	1,500 kg/s

Tab. 3.1: Data of the condenser

In table 3.1 und 3.1 the results and the thermic parameters HTR, OHR and HOR of four operating points are presented.

operating point		water			surfactant solution		
		α [W/m^2K]	k [$W/(m^2K)$]	$\dot{Q}$ [MW]	α [$W/(m^2K)$]	k [$W/(m^2K)$]	$\dot{Q}$ [MW]
1	$\dot{Q}_N$	10,806	2,863	50	895	714	14.6
2	0.6 $\dot{Q}_N$	4,530	2,180	30	497	425	8.4
3	0.4 $\dot{Q}_N$	2,798	1,702	20	365	320	6.1
4	0.2 $\dot{Q}_N$	1,373	1,039	10	245	219	3.9

Tab. 3.1: Results of the calculation, part 1

oper. point	ω	HTR	OHR	HOR
$\dot{Q}_N$	2.5	92 %	75 %	71 %
0.6 $\dot{Q}_N$	0.9	89 %	80 %	72 %
0.4 $\dot{Q}_N$	0.5	87 %	81 %	70 %
0.2 $\dot{Q}_N$	0.2	82 %	76 %	61 %

Tab. 3.1: Results of the calculation, part 2 (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.

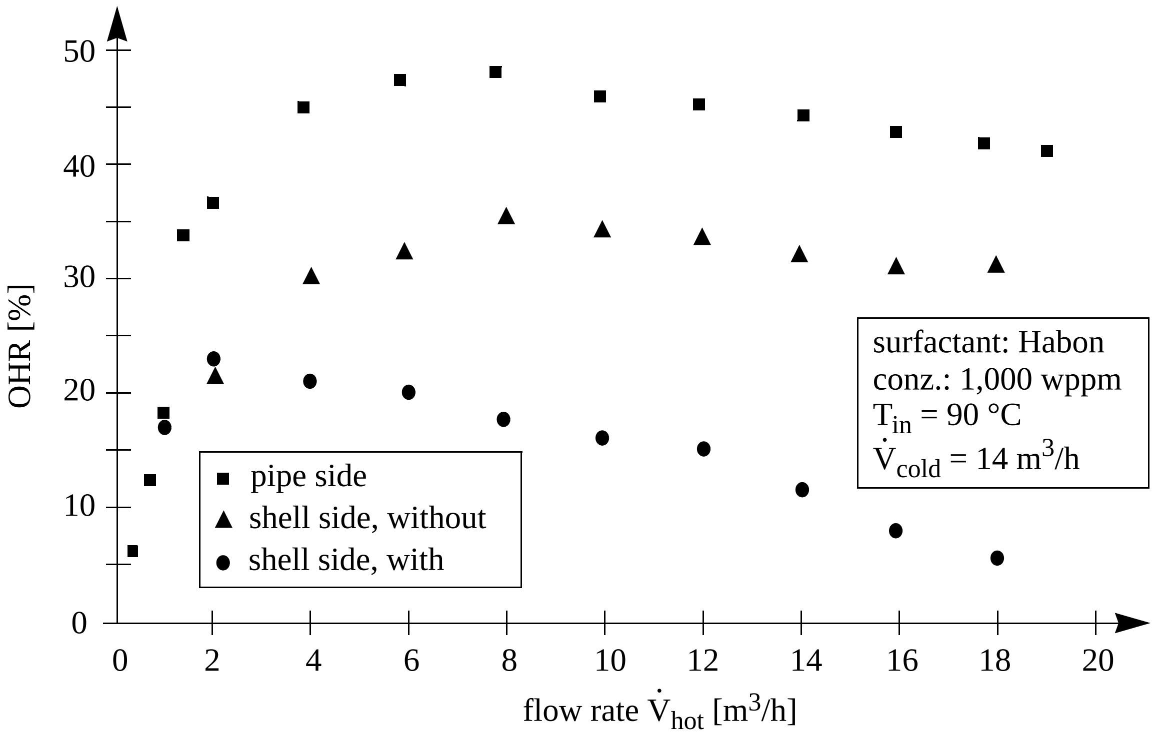


Fig. 3.3: OHR if using surfactants on the shell side with and without support plates [30]

The OHR on the shell side is significantly smaller that on the pipe side. The curves show a maximum, due to the two different effects of ω and HTR. With increasing flow rate the HTR and therefore the OHR increases, too. On the other hand, the increase of ω with rising flow rates leads to a decrease of OHR. The parameter ω 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 $\delta = D_i/D_H$ 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]:

$$Nu = 0.073 \cdot Re^{0.694} \qquad \text{eq. (3.10)}$$

for the range of validity: Reynolds number: $3000 < Re < 10{,}000$
curvature ratio: $0.0882 < \delta < 0.105$ and

$$Nu = 0.05 \cdot \left(\frac{D_i}{D_H} + 0.1\right) - 1.69 \cdot Re^{\sqrt{\frac{D_i}{D_H} + 0.1}} \qquad \text{eq. (3.11)}$$

for the range of validity: Reynolds number: $10{,}000 < Re < 150{,}000$
curvature ratio: $0.036 < \delta < 0.105$

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 3.1** shows the data of the apparatus that has been investigated.

design		nominal operation	
diameter of the pipes	14 mm	nominal heat load	310 kW
length of the pipes	2,900 to 3,300 mm	entrance temperature pipe side T_{in}	60 °C
number of pipes	34	entrance temperature shell side T_{in}	100 °C
height of the bundle	430 mm	mass flow rate pipe side $\dot{M}_1$	3.8 kg/s
heat transfer area	4 m^2		
pipe arrangement	staggered	mass flow rate shell side $\dot{M}_2$	3.8 kg/s
appl. of surfactants	pipe side		

Tab. 3.1: Data of the helical tube heat exchanger

The results of the simulation for four different operating points are presented in table 3.1 and 3.1. 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 $(0.2 \cdot \dot{Q}_N)$ 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 % occures.

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.

operating point		water			surfactant solution		
		α [W/(m²K)]	k [W/(m²K)]	$\dot{Q}$ [kW]	α [W/(m²K)]	k [W/(m²K)]	$\dot{Q}$ [kW]
1	$\dot{Q}_N$	8,056	3,606	310	3,363	2,117	246.1
2	0.6 $\dot{Q}_N$	4,949	2,397	186	2,691	1,646	163
3	0.4 $\dot{Q}_N$	3,284	1,711	124	2,267	1,355	115.9
4	0.2 $\dot{Q}_N$	1,813	1,017	62	1,423	867	60.1

Tab. 3.1: Results of the calculation, part 1

operating point	ω	HTR	OHR	HOR
$\dot{Q}_N$	1	58.3 %	41.2 %	20.5 %
0.6 $\dot{Q}_N$	0.9	45.6 %	31.3 %	12.2 %
0.4 $\dot{Q}_N$	0.8	31 %	20.8 %	6.4 %
0.2 $\dot{Q}_N$	0.6	21.5 %	14.7 %	2.9 %

Tab. 3.1: Results of the calculation, part 2

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 und 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]:

$$Nu_s = K \cdot Re^m \cdot Pr^{0.4} \cdot \left(\frac{Pr}{Pr_W}\right) \cdot f_T = Nu_w \cdot f_T \ . \qquad \text{eq. (3.12)}$$

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 (Re_c). Above this value f_T is constant up to the critical Reynolds number (critical wall shear stress). Re_c is dependent on the geometry, the kind of plates, the concentration, temperature etc.

For small Reynolds numbers < Re_c, f_T can be calculated according to the following equation [22]:

$$f_T = K_a \cdot Re^r . \qquad \text{eq. (3.13)}$$

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

$$f_T = K_b = 1 - \frac{HTR}{100} . \qquad \text{eq. (3.14)}$$

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 occures. 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 3.1**.

Table 3.1 and **3.1** 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.

design		nominal operation	
hydr. diameter	6.15 mm	nominal heat load	4 MW
width of the gap	3.6 mm	entrance temperature primary side $T_{1,in}$	100 °C
breadth of the gap	444 mm	entrance temperature secondary side $T_{2,in}$	60 °C
number of plates	83	mass flow rate prim.	30.6 kg/s
heat exchange area	73 m^2	mass flow rate sec.	30.6 kg/s
thickness of the plate	0.7 mm		

Tab. 3.1: Data of the plate heat exchanger

operating point		water			surfactant solution		
		α [W/(m^{2}K)]	k [W/(m^{2}K)]	$\dot{Q}$ [MW]	α [W/(m^{2}K)]	k [W/(m^{2}K)]	$\dot{Q}$ [MW]
1	$\dot{Q}_N$	18,260	6,336	4	8,397	4,493	3.68
2	0.6 $\dot{Q}_N$	11,904	4,603	2.4	5,455	3,153	2.21
3	0.4 $\dot{Q}_N$	8,571	3,541	1.6	3,940	2,379	1.49
4	0.2 $\dot{Q}_N$	4,926	2,191	0.8	2,257	1,433	0.75

Tab. 3.1: Results of the calculation, part 1

operating point	ω	HTR	OHR	HOR
$\dot{Q}_N$	1	54 %	29.1 %	9 %
0.6 $\dot{Q}_N$	1	54.2 %	31.5 %	8 %
0.4 $\dot{Q}_N$	1	54 %	32.8 %	7 %
0.2 $\dot{Q}_N$	1	54.2 %	34.5 %	6 %

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 occures 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 3.1** shows the maximum values for HTR and HOR and in addition the values of the heat transfer coefficient ratio ω.

apparatus	max HTR	ω	max HOR
shell and tube	95 %	1 - 4	70 %
helical tube	80 %	0,6 - 4	30 %
plate	70 %	0,5 - 2	10 %

Tab. 3.1: Maximum HTR, HOR and ω

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 echanger - 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 maintenace, 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.

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/D_i-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 signifcant 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..

kind of meter	influence	suitability	remark
single-jet impeller meter	strong	no	correcture with critical wall shear stress possible
multi-jet impeller meter	strong	no	
Woltmann-flow meter	medium	no - limited	
magnitive inductive	no	yes	best possibility
ultrasonic (one measuring path)	medium	no	
ultrasonic (LLL-principle)	medium - low	no - limited	
ultrasonic (passing the whole flow profile)	low	yes	
orifice plate	medium - low	limited	possible if: $d/D_i < 0.4$
venturi tube	medium - low	limited	possible if: $d/D_i < 0.53$
Pitot tube flow meter	medium	no	

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 whithout restrictions. Orifice plates and venturi tubes can be used under special conditions concerning the aperture ratio. For orifice plates the ratio d/D_i 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 shutt-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.

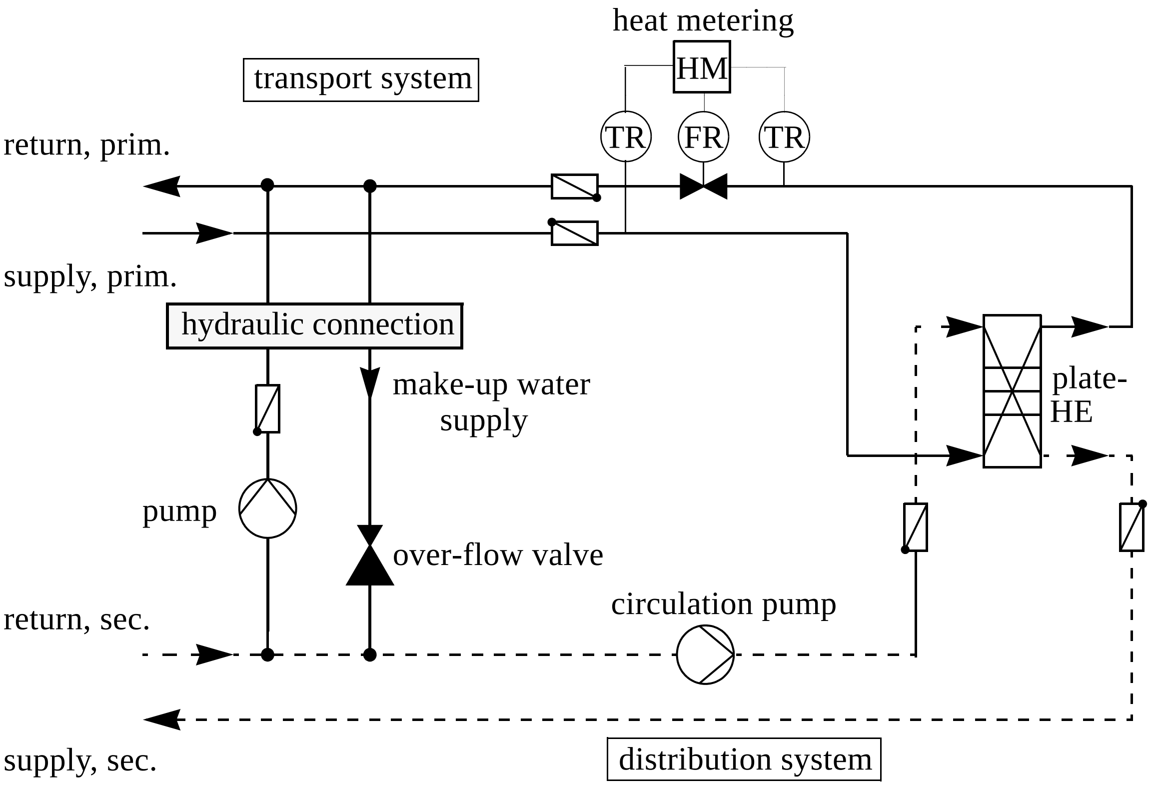


Fig. 3.4: Example of a dynamic pressure maintenance [30]

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 occure. 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 occures.

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 ammoniumcompositions QACs (commercial used surfactants) are shown in table 3.1. 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 (LD_{50} > 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_{50}-value (rat) in mg per kg body weight. The LD_{50}-values are listed in **table 3.1**. 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].

	decomposition		ecological toxicity	
	primary	total	Fish, LC_{50} [mg/l]	Daphnien, EC_{50} [mg/l]
standard value	> 80 %	> 60 %	> 1	> 1
QAC	94 %	100 %	1 - 40	0.1 - 100
Dobon-G	> 90 %	-	7.1	> 1,000

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

quaternary ammoniumcompositions (QAC)	LD_{50} [mg/kg KG]
Dobon-G	> 2,000
Hexadecyltrimethylammoniumbromide (CTAB)	410 - 430
Distearyldimethylammoniumchloride (DSDMAC)	> 5,000
Stearylbenzyldimethylammoniumchloride	1,000
Cetylbenzyldimethylammoniumchloride	250 - 300

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 occures. 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,

    - 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].

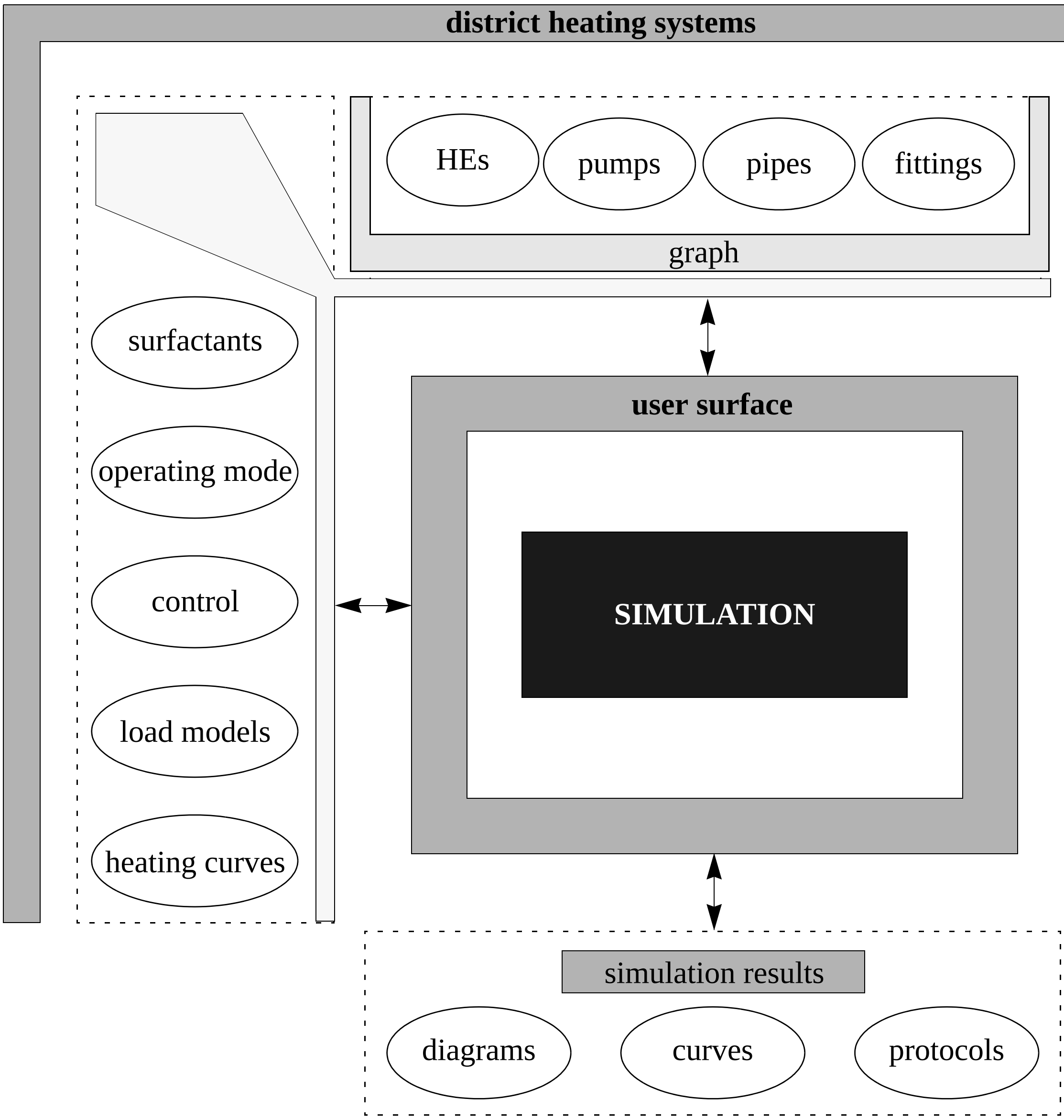


Fig. 4.1: Components of the simulation program

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 hydraulical 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 continous, because there is a path between every pair of nodes.

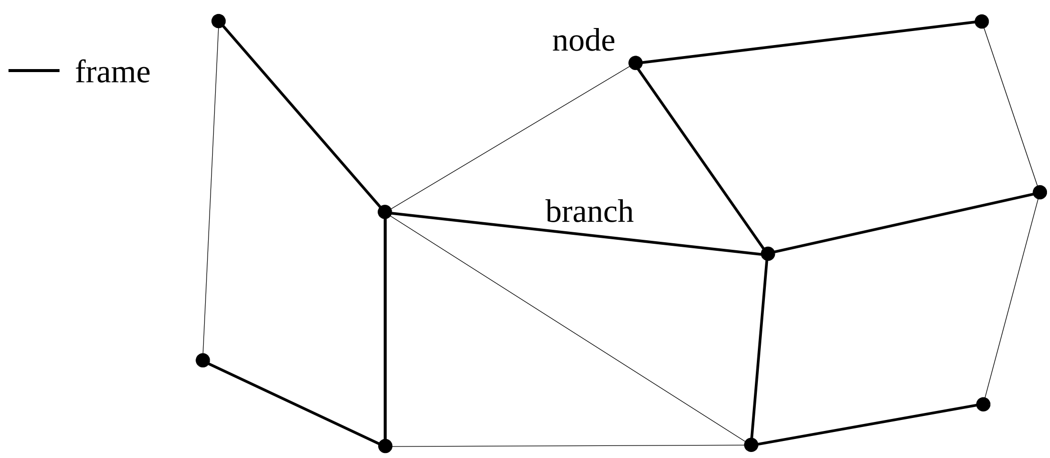


Fig. 4.2: Continous graph and frame

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:

$$N = K_G + 1 . \qquad \text{eq. (4.1)}$$

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:

$$M = (K + 1) - N . \qquad \text{eq. (4.2)}$$

Algorithm for determining the frame

For the simulation program TenSim the BFS-method (**B**readth-**F**irst-**S**earch) 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_{Start} which is marked all adjacent nodes are determined (These are all nodes of the branches which are connected with K_{Start}). If an adjacent node K_N has not been checked (is not marked), the connection between K_{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.).

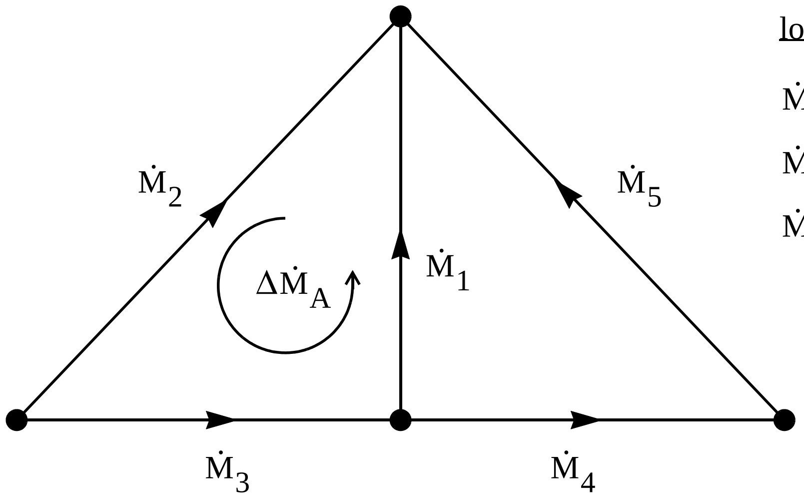
All network calculations use Kirchhoff´s laws as basic (voltage law and current law). The first law - the current law - means that the sum of all mass flow rates in a node is zero:

$$\sum \dot{M} = 0 . \qquad \text{eq. (4.3)}$$

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

$$\sum \Delta p = 0 . \qquad \text{eq. (4.4)}$$

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.



<u>loop-correction:</u>

$$\dot{M}_{1,new} = \dot{M}_{1,old} + \Delta\dot{M}_A$$
$$\dot{M}_{2,new} = \dot{M}_{2,old} - \Delta\dot{M}_A$$
$$\dot{M}_{3,new} = \dot{M}_{3,old} + \Delta\dot{M}_A$$

Fig. 4.3: Kirchhoff´s laws and loop correction

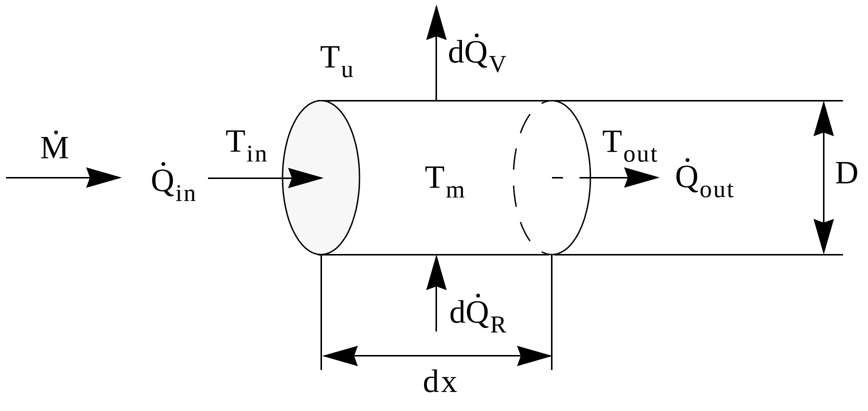
The simplified network in **figure 4.3** shows that the adjacent loop is also influenced. Therefore the calculation of a large system with a large number of loops is not easily to verify.

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**.



index	
in	entrance
out	outlet
R	friction
V	loss
m	average
u	ambient

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:

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

$$\dot{Q}_{out} = \dot{M} \cdot c_p \cdot T_{out}, \qquad \text{eq. (4.6)}$$

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

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

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

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

With the boundary conditions:

$x = 0,\ T_m(x=0) = T_{in}$ and

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

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

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

with the constants α and β:

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

The outlet temperature can be calculated according to equation 4.11:

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

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

$$T_{out} = T_u + (T_{in} - T_u) \cdot \exp\left(-\frac{1}{\alpha} \cdot \frac{L}{D}\right). \qquad \text{eq. (4.12)}$$

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):

$$\Delta T \equiv T_{out} - T_{in} = (T_{in} - T_u) \cdot \left(\exp\left(-\frac{k \cdot \pi \cdot D \cdot L}{c_p \cdot \dot{M}}\right) - 1\right) . \qquad \text{eq. (4.13)}$$

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]:

$$\Delta p = \dot{M} \cdot |\dot{M}| \cdot r - \rho \cdot g \cdot \Delta H . \qquad \text{eq. (4.14)}$$

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:

$$r = \frac{8}{\pi^2 \cdot \rho} \cdot \frac{L}{D^4} \cdot \left[\frac{L}{d} \cdot \xi + \Sigma\, \xi_{add}\right] . \qquad \text{eq. (4.15)}$$

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]:

$$\xi = \frac{64}{Re} . \qquad \text{eq. (4.16)}$$

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

$$\frac{1}{\xi} = -2 \cdot \log\left[\frac{2.51}{Re \cdot \sqrt{\xi}} + \frac{k/D}{3.71}\right] . \qquad \text{eq. (4.17)}$$

Equation 4.17 can only be solved iterative. Due to the fast convergence no problems occure. 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):

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

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

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:

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

For the reference speed n_0 the characteristic is described as follows:

$$\Delta p_0 = a_0(n_0) + a_1(n_0) \cdot \dot{M}_0 + a_2(n_0) \cdot \dot{M}_0^2 \quad . \qquad \text{eq. (4.21)}$$

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:

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

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

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

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

$$\Delta p = a_0(n) + a_1(n) \cdot \dot{M} + a_2(n) \cdot \dot{M}^2$$

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

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

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:

$$\nabla\left[\sum_i (y_i - \hat{y}_i)^2\right] = \vec{O} \; . \qquad \text{eq. (4.25)}$$

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:

$$\sum_i (y_i - \hat{y}_i)^2 = \sum_i ((a + b \cdot x_i + c \cdot x_i^2 - y_i)^2) \; . \qquad \text{eq. (4.26)}$$

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

$$\begin{bmatrix} 2 \cdot a \cdot \Sigma x_i^0 + 2 \cdot b \cdot \Sigma x_i + 2 \cdot c \cdot \Sigma x_i^2 - 2 \cdot \Sigma y_i \\ 2 \cdot a \cdot \Sigma x_i + 2 \cdot b \cdot \Sigma x_i^2 + 2 \cdot c \cdot \Sigma x_i^3 - 2 \cdot \Sigma(x_i \cdot y_i) \\ 2 \cdot a \cdot \Sigma x_i^2 + 2 \cdot b \cdot \Sigma x_i^3 + 2 \cdot c \cdot \Sigma x_i^4 - 2 \cdot \Sigma(x_i^2 \cdot y_i) \end{bmatrix} = \vec{O} \quad \text{or} \qquad \text{eq. (4.27)}$$

$$\begin{bmatrix} \Sigma x_i^0 & \Sigma x_i & \Sigma x_i^2 \\ \Sigma x_i & \Sigma x_i^2 & \Sigma x_i^3 \\ \Sigma x_i^2 & \Sigma x_i^3 & \Sigma x_i^4 \end{bmatrix} \cdot \begin{bmatrix} a \\ b \\ c \end{bmatrix} = \begin{bmatrix} \Sigma y_i \\ \Sigma(x_i \cdot y_i) \\ \Sigma(x_i^2 \cdot y_i) \end{bmatrix} . \qquad \text{eq. (4.28)}$$

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.

$$n = -\left(\frac{a_1}{a_0} \cdot \frac{n_0}{2} \cdot \dot{M}\right) + \sqrt{\left(\frac{a_1}{a_0} \cdot \frac{n_0}{2} \cdot \dot{M}\right)^2 - \left(\frac{a_2 \cdot \dot{M}^2 - \Delta p}{a_0} \cdot n_0^2\right)} . \qquad \text{eq. (4.29)}$$

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:

$$P_V = \dot{Q}_{add} . \qquad \text{eq. (4.30)}$$

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 $\dot{Q}_{add} = \dot{M} \cdot c_p \cdot \Delta T$ 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:

$$\Delta T = \frac{1 - \eta}{\eta} \cdot \frac{\Delta p}{\rho \cdot c_p} . \qquad \text{eq. (4.31)}$$

For an example with the following data:

Δp = 16 bar = 1.6 MPa,
ρ = 975 kg/m^3,
c_p = 4,180 J/(kg K) and
η = 0.75,

the increase of temperature is ΔT = 0.131 K. Assuming a flow rate of 1,000 m^3/h the dissipated energy flow rate of the pump is:

$$\dot{Q}_{add} = \dot{V} \cdot \rho \cdot c_p \cdot \Delta T = 0.148 \ \text{MW} .$$

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.

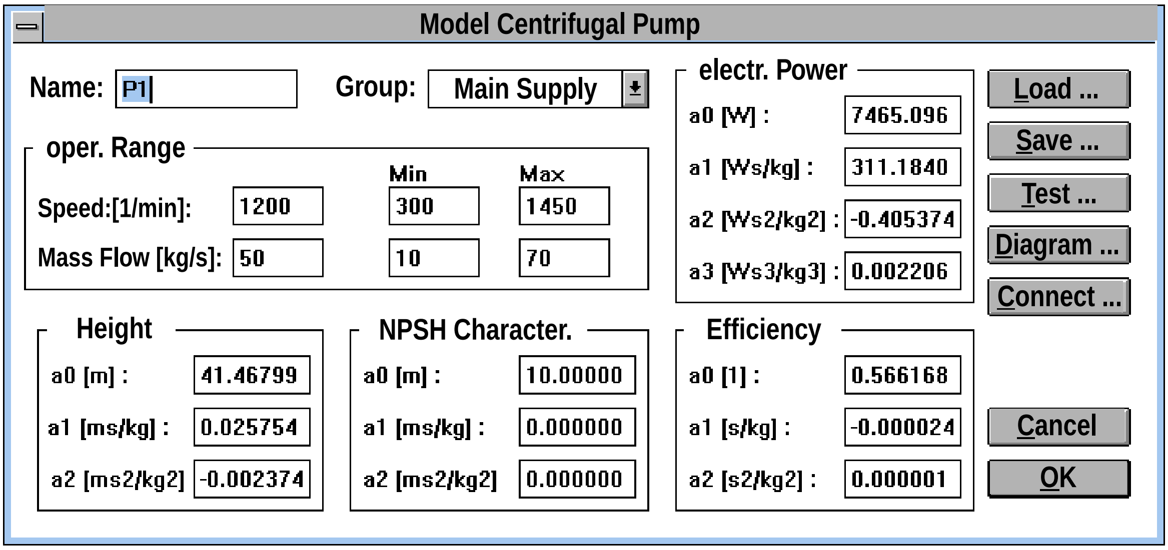


Fig. 4.5: Dialog of the model „centrifugal pump“ (Modell Kreiselpumpe)

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 $\dot{M}_0$ 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 ΔT_m is defined as follows:

$$\Delta T_m \equiv \frac{1}{A} \cdot \int_A (\Delta T_1^+ - \Delta T_2^+) dA \quad \text{eq. (4.32)}$$

ΔT^+ 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:

$$\dot{Q} = \dot{M}_1 \cdot (h_{1in} - h_{1out}) = \dot{M}_2 \cdot (h_{2out} - h_{2in}) \ . \qquad \text{eq. (4.33)}$$

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

$$\dot{Q} = \dot{W}_1 \cdot (T_{1in} - T_{1out}) = \dot{W}_2 \cdot (T_{2out} - T_{2in}) \ , \qquad \text{eq. (4.34)}$$

with the heat capacity flow rate $\dot{W}_i \equiv \dot{M}_i \cdot c_{pi}$.

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 $(T_{1ein} - T_{2ein})$ dimensionless basic numbers can be defined. These are used for the simulations [38]:

non-dimensional average temperature difference Θ:

$$\Theta = \frac{\Delta T_m}{(T_{1in} - T_{2in})} \text{, with } 0 \le \Theta \le 1 \text{,} \qquad \text{eq. (4.35)}$$

non-dimensional average change of mass flow temperature Ψ:

$$\Psi_1 = \frac{T_{1in} - T_{1out}}{T_{1in} - T_{2in}},\ \Psi_2 = \frac{T_{2out} - T_{2in}}{T_{1in} - T_{2in}} \text{, with } 0 \le \Psi \le 1 \text{,} \qquad \text{eq. (4.36)}$$

number of transfer units NTU:

$$NTU_1 = \frac{k \cdot A}{\dot{W}_1}, NTU_2 = \frac{k \cdot A}{\dot{W}_2} \text{, with } 0 \le NTU \le \infty \text{ and} \qquad \text{eq. (4.37)}$$

ratio of heat capacity flow rate R:

$$R_1 = \frac{\dot{W}_1}{\dot{W}_2} = \frac{1}{R_2} \text{, with } 0 \le R \le \infty \text{.} \qquad \text{eq. (4.38)}$$

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

$$\frac{\Psi_1}{\Psi_2} = \frac{NTU_1}{NTU_2} = \frac{1}{R_1} = R_2 \text{ and} \qquad \text{eq. (4.39)}$$

$$\Theta = \frac{\Psi_1}{NTU_1} = \frac{\Psi_2}{NTU_2} \text{.} \qquad \text{eq. (4.40)}$$

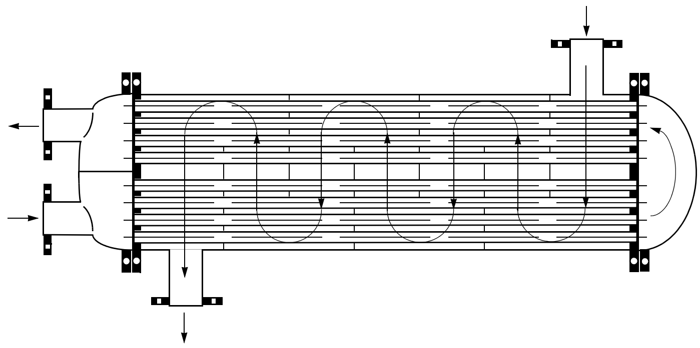
Model „Shell ant Tube Heat Exchanger“ - cell method

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 occure 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.

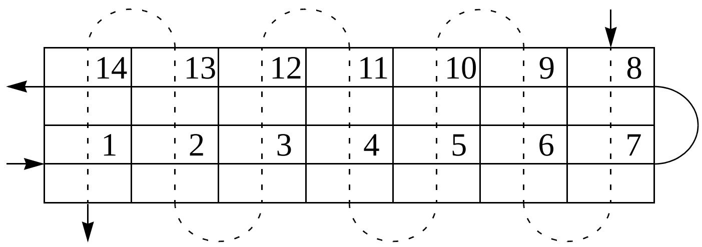
To calculate the influence of geometry on the heat transfer capacity of shell and tube heat exchangers the apparatus is devided into several cells (see **figure 4.6)**. Considering the arrangement of the connection pieces (4 different possibilities) the flow direction and the number of passes (odd/even-numbered) 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 totaly mixed (ideal vessel) and
- the flow on the pipe side is not mixed.

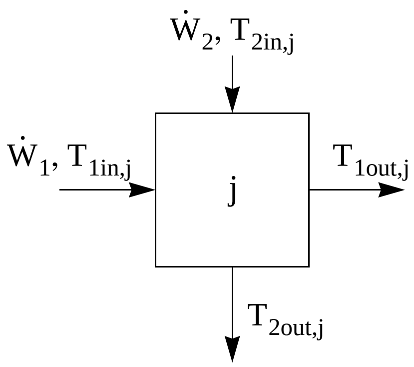


a) shell and tube heat exchanger



b) ideal apparatus (cell-method)

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



index 1: pipe side,
index 2: shell side
index j: number of the cell

Fig. 4.7: Single cell of a heat-exchanger

The apparatus is devided into $(N + 1) \cdot (M + 1)$ cells (N = number of passes on the tube side, M = 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:

$$T_{2out,j} = T_{2in,k} \text{ and} \qquad \text{eq. (4.41)}$$

$$T_{1in,j} = T_{1out,j-1}\,. \qquad \text{eq. (4.42)}$$

The cell j is in direction of the flow of $\dot{W}_1$ and the cell k in direction of $\dot{W}_2$.

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

$$T_{1out} = (1 - \Psi_1) \cdot T_{1in} + \Psi_1 \cdot T_{2in} \text{ and} \qquad \text{eq. (4.43)}$$

$$T_{2out} = (1 - \Psi_2) \cdot T_{2in} + \Psi_2 \cdot T_{1in}\,. \qquad \text{eq. (4.44)}$$

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 identic and that it can be calculated as follows:

$$NTU_{1.Z} = \frac{NTU_{1.tot}}{\text{number of cells}} \text{ and} \qquad \text{eq. (4.45)}$$

$$NTU_{2.Z} = \frac{NTU_{2.ges}}{\text{number of cells}}\,. \qquad \text{eq. (4.46)}$$

Mean wall temperatures can be determined with equation 4.47:

$$\Delta T_{i,W} = \frac{k \cdot A}{\alpha_i \cdot A_i} \cdot \Delta T_m\,. \qquad \text{eq. (4.47)}$$

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 identic.

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 occure 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 occure.

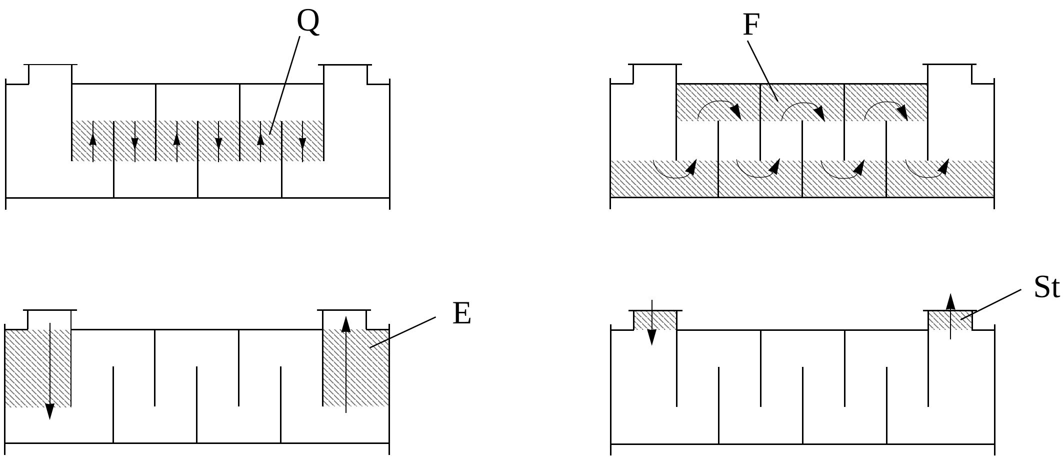


Fig. 4.8: Division of a heat exchanger in different zones [38]

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

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

This is nearly the complete dynamic pressure.

$$\Delta p_{R,\,in} = \xi_{R,\,in} \cdot n_R \cdot \frac{\rho}{2} \cdot w^2, \text{ with } \xi_{R,\,in} \approx 0.25 - 0.5. \qquad \text{eq. (4.49)}$$

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

$$\Delta p_U = \xi_U \cdot (Z - 1) \cdot \frac{\rho}{2} \cdot w^2, \text{ with } \xi_U \approx 2.5, \qquad \text{eq. (4.50)}$$

$$\Delta p_{R,\,out} = \xi_{R,\,out} \cdot n_{tube} \cdot \frac{\rho}{2} \cdot w^2\text{, with } \xi_{R,\,out} \approx 1 \text{ and} \qquad \text{eq. (4.51)}$$

$$\Delta p_{St,\,out} = \xi_{out} \cdot \frac{\rho}{2} \cdot w^2\text{, with } \xi_{out} \approx 0.25 - 0.5\,. \qquad \text{eq. (4.52)}$$

Der geodetic pressure drop can be calculated as follows:

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

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.

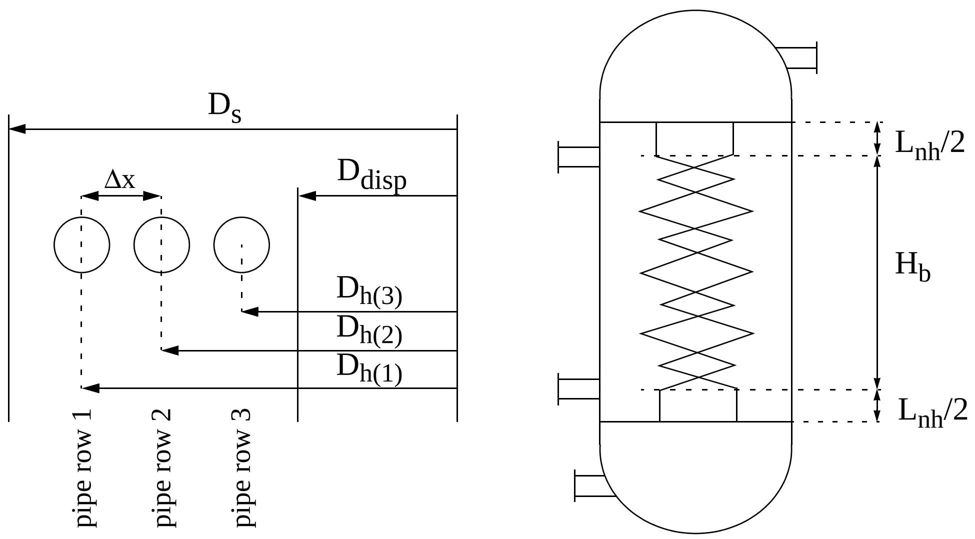


Fig. 4.9: Important geometric quantities for helical tube heat exchangers

The calculation requires a lot of assumptions and simplifications. It is assumed that the pipe-rows are equidistant. The distance Δx 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_{win}, the gradient of the helical tube bundle s_h and its average diameter of curvature D_{mh}.

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

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

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

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

$$D_{mh} = D_h \cdot \left(1 + \left(\frac{s_h}{\pi \cdot D_h}\right)^2\right).$$ 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:

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

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

The gradient of the pipes which leads to a transverse flow is neglected. A significant influence occures at gradients above 40 °. In this case a reduction of the heat transfer of more than 5 % occures [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 devided 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 occures. This effect is not considered, too.

Despite of this simplifications calculations of real condensers showed a good correspondance 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 neclected. 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_T [22]. The particular parameters to build the correlations can be implemented into the program by using the dialog shown in **figure 4.10**

To simplfy 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:

$$Eu = C_1 \cdot Re^{Z_1} . \qquad \text{eq. (4.61)}$$

The Euler number is defined as follows:

$$Eu = \frac{\Delta p}{\rho \cdot w_C^2} . \qquad \text{eq. (4.62)}$$

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.

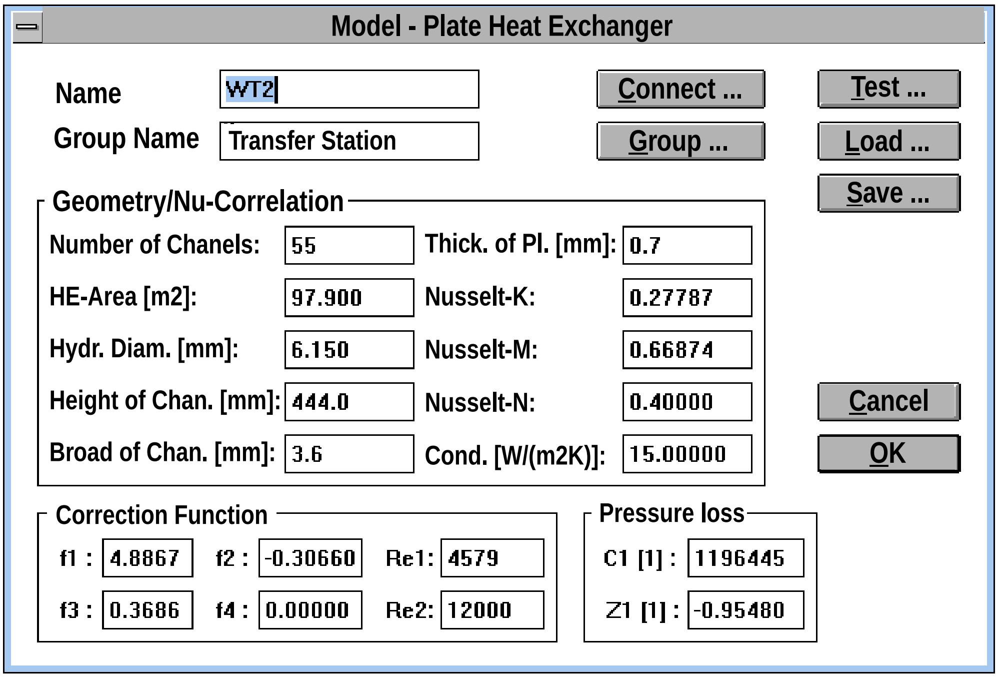


Fig. 4.10: Dialog for the model „plate heat exchanger“

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 Δp_{cp} and half of the pressure loss of any path (ap) to the other reference node p_{K2}:

$$p_{K1} = \Delta p_{cp} + \frac{\Delta p_{ap}}{2}. \qquad \text{eq. (4.63)}$$

After the calculation of the network the laws of Kirchhoff are fulfiled. 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.

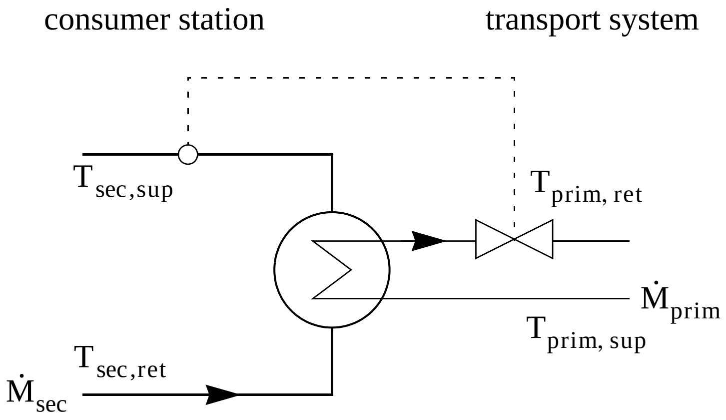


Fig. 4.11: Scheme of a temperature control of a consumer station

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 distric 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 $T_{prim,\ sup}$. 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.

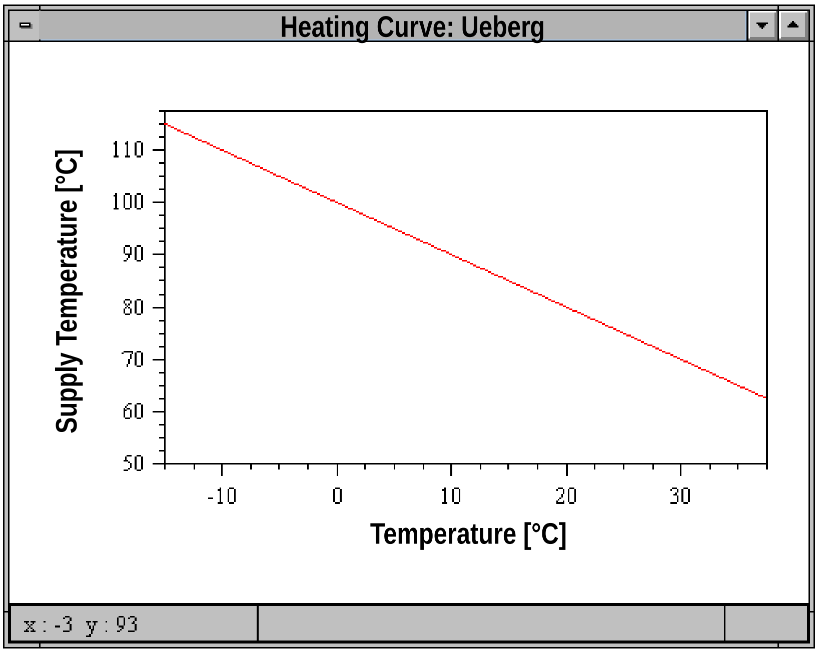


Fig. 4.12: Heating curve

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 dependece 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 °C < T < 15 °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:

$$\text{Polynomial: } y = a_0 + a_1 \cdot T + a_2 \cdot T^2 + a_3 \cdot T^3, \qquad \text{eq. (4.64)}$$

$$\text{minimum: } T_{min} = -\frac{a_2}{3 \cdot a_3} + \sqrt{\left(\frac{a_2}{3 \cdot a_3}\right)^2 - \frac{a_1}{3 \cdot a_3}} \text{ and} \qquad \text{eq. (4.65)}$$

$$\text{turning point: } T_{TP} = -\frac{a_2}{3 \cdot a_3}. \qquad \text{eq. (4.66)}$$

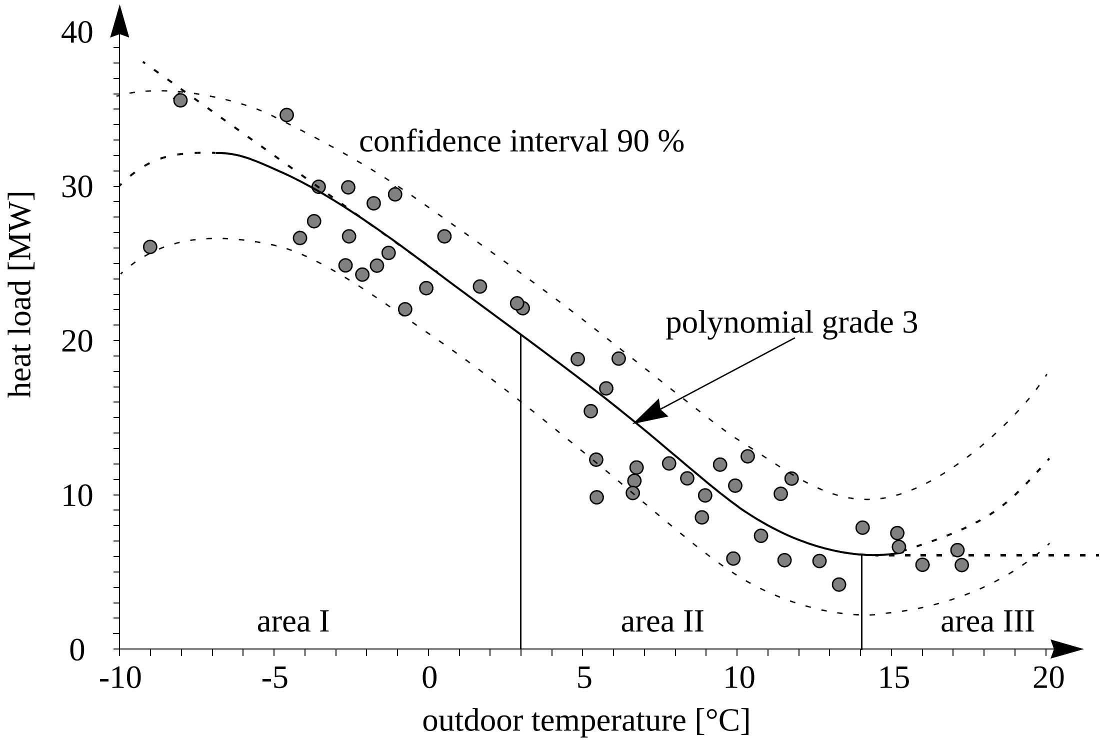


Fig. 4.13: Heat load as function of average outdoor temperature

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

$$\text{area 1: } y = y(T_{TP}) + \left(\frac{dy}{dT}\right)_{TP} \cdot (T - T_{TP}) \quad . \qquad \text{eq. (4.67)}$$

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:

$$\text{area 3: } y_{min} = a_0 + a_1 \cdot T_{min} + a_2 \cdot T_{min}^2 + a_3 \cdot T_{min}^3 = \text{const}. \qquad \text{eq. (4.68)}$$

In area 2 the heat load can be described with equation 4.64.

Concerning the dependence on the time of the day an accuracy of the model of 90 % for the heat load in the morning has been achieved (due to the higher unsteadiness) and 96 % for the evening hours. 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.

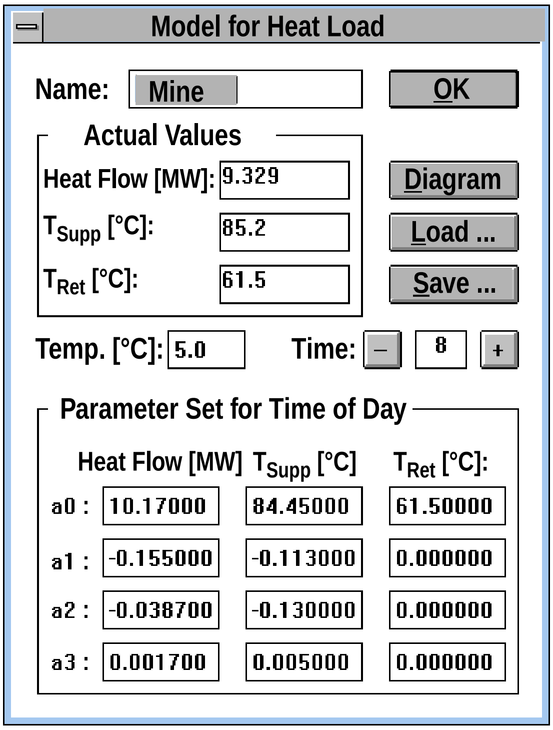


Fig. 4.14: Dialog for the heat load model

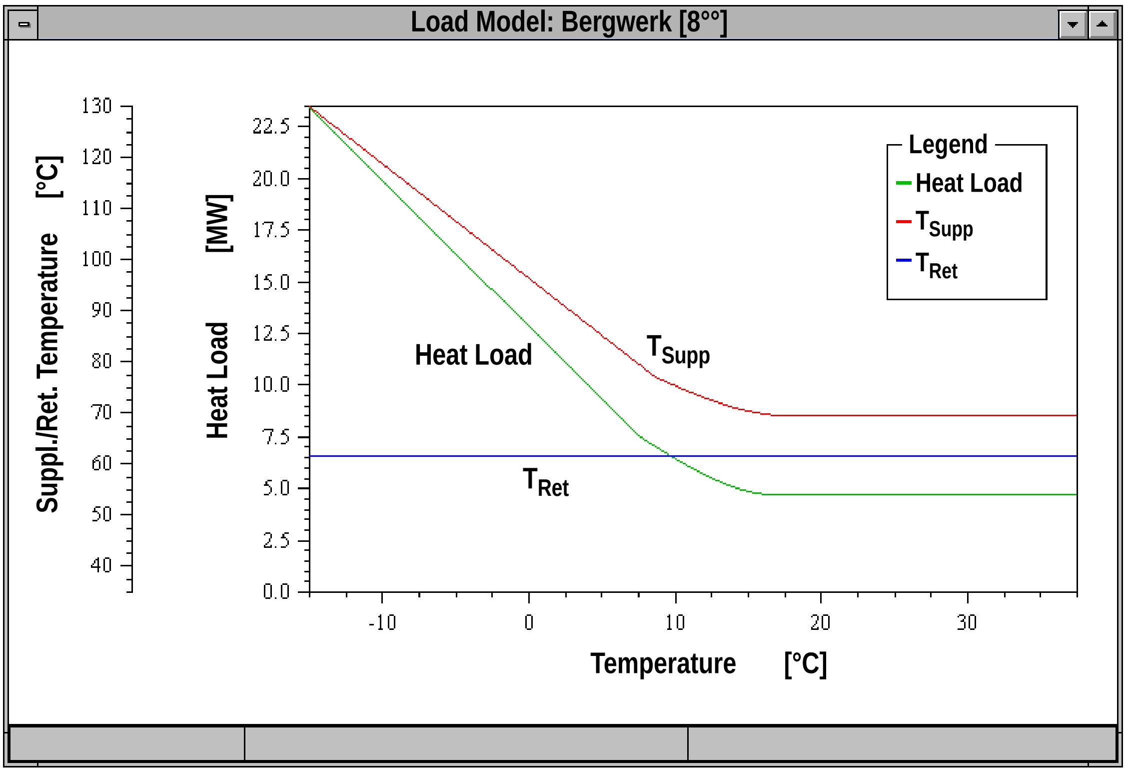


Fig. 4.15: Diagram for the dialog heat load model

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

„return temperature“ 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 identificated 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 choosen 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 τ_{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 τ_{opt} and T_{opt} have to be determined [10]:

$$T_{opt} = T_0 \cdot c_A^a , \qquad \text{eq. (4.69)}$$

$$\tau_{opt} = \tau_0 + b \cdot c_A \text{, in the area of application:} \qquad \text{eq. (4.70)}$$

$$c_1 \le c \le c_2 \quad [\text{wppm}] \quad \text{and} \quad T_1 \le T \le T_2 \quad [°C] . \qquad \text{eq. (4.71)}$$

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 :

$$\tau_{rel} = \frac{\tau_{W,w}}{\tau_{opt}} . \qquad \text{eq. (4.72)}$$

In the same way the temperatures are related as difference T - T_{opt} to the optimal temperature:

$$T_{rel} = \frac{T - T_{opt}}{T_{opt} + 273.15}, \text{ T in [°C].} \qquad \text{eq. (4.73)}$$

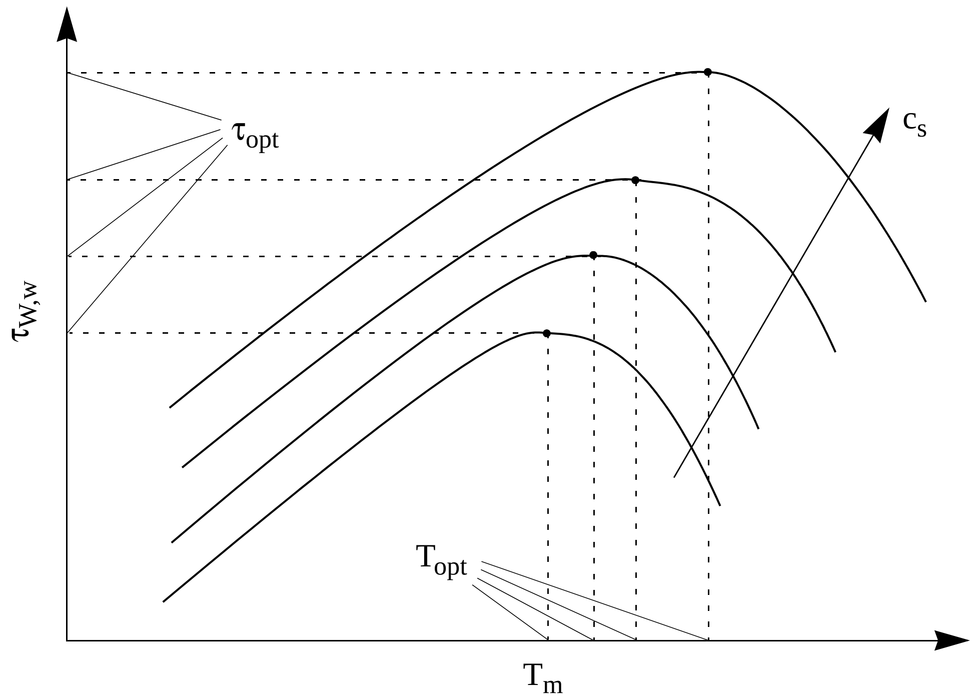


Fig. 4.16: Critical wall shear stress of surfactant solutions as function of temperature and concentration

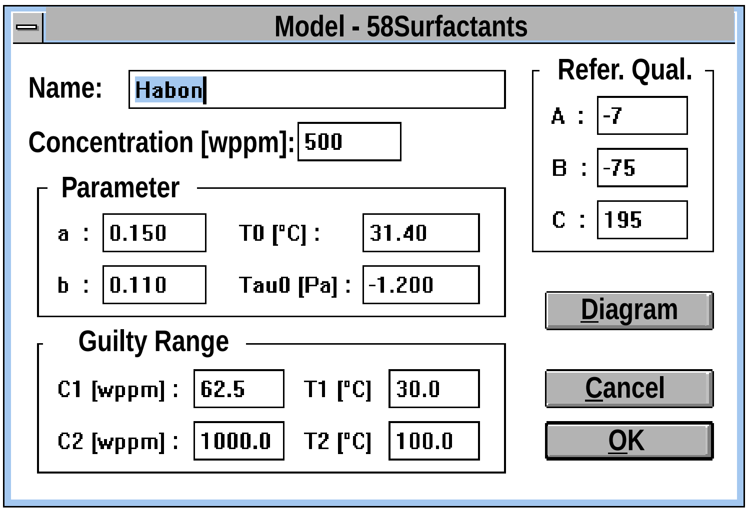


Fig. 4.17: Dialog for the model „surfactants“

The relation between τ_{rel} and T_{rel} can be described as fraction of polynomial second degree:

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

The coefficients of the functions 4.69, 4.70 und 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 indepentend from the concentration is preferred due to its simple construction:

$$\tau_{W,s} = 0.35 \cdot (\tau_{W,w})^{0.78} . \qquad \text{eq. (4.75)}$$

The dialog for the model „surfactant“ is shown in **figure 4.17**. The above mentioned parameters for the equations 4.69, 4.70 und 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 fulfil 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 consistance 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 consistance 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 correcture 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 fulfil 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 than 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 guaranties 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

Symbol	Description	Unit
a	coefficient	[1]
a_n	annuity	[1/a]
A	heat exchange area	[m^2]
b	coefficient	[1]
c	coefficient	[1]
c	concentration	[mol/l]
c_p	specific heat capacity	[J/(kgK)]
C_1	coefficient	[1]
d	diameter of the cross section for measuring	[m]
D	diameter	[m]
f_{loss}	factor of water losses	[1]
f_T	correction term	[1]
f_{ni}	share of not influenced pressure losses	[1]
g	gravitation constant	[m/s^2]
h	specific enthalpie	[J/kg]
h	height	[m]
H	height	[m]
I	cost	[DM]
k	absolute height of roughness	[m]
k	overall heat transfer coefficient	[$W/(m^2K)$]
k_T	specific transportation cost	[DM/Wh]
K	number of branches	[1]
K	cost	[DM]
K	parameter for regression	[1]
Ke	heat generation cost	[DM]
L	length	[m]
L_h	entrance length	[m]
m	parameter for regression	[1]
M	number of loops	[1]
$\dot{M}$	mass flow rate	[kg/s]
n	number	[1]
n	speed	[1/s]
N	number of nodes	[1]
p	pressure	[Pa]
Δp	pressure loss	[Pa]
P	capacity	[W]

Q	heat quantity	[Ws]
$\dot{Q}$	heat load	[W]
r	constante	[1]
r	return flow	[DM/a]
R	ratio of heat load	[1]
R_f	fouling factor	[1]
s	thickness of the wall	[m]
s_{st}	price for electric power	[DM/kWh]
T	temperature	[T]
T_P	pumping duration	[h/a]
ΔT_m	average temperature difference	[K]
ΔT_{ln}	logarithmic temperature difference	[K]
v	flow velocity	[m/s]
V	volume	[m^3]
$\dot{V}$	flow rate	[m^3/s]
w	flow velocity	[m/s]
x	coordinate	[1]
$\hat{y}$	calculated value	[1]
Z_1	coefficient	[1]
α	heat transfer coefficient	[W/(m^2K)]
δ	ratio of diameters	[1]
Δ	difference	[1]
η	efficiency	[1]
λ	heat conductivity	[W/(mK)]
ρ	density	[kg/m^3]
Θ	non dimensional avarage temperature difference	[1]
Σ	sum	[1]
τ	shear stress	[Pa]
ω	ratio of heat transfer coefficient	[1]
ξ	drag coefficient	[1]
Ψ	average change of medium temperature	[1]

Indices	**Meaning**
add	additional
ap	any path
b	bundle
c	characteristic
c	channel
cp	control point
d	displacer
G	frame

h	helical tube
i	inner
in	entrance
m	average
max	maximum
min	minimum
new	new value
nh	not helical
N	nominal value
old	old value
opt	optimum
prim	primary
P	pump
ret	return
R	friction term
R	pipe
s	surfactant solution
sec	secondary
start	start
sup	supply
S	steam (saturation point)
S	shell
St	connection piece
t	transverse
tot	total
tp	turning point
u	ambient
U	deviation
V	loss
w	water
win	winding
*	critical value
0	origin state
1	state 1
1	primary side/pipe side
2	secondary side/shell side
2	state 2
3	state 3

Special symbols	**Meaning**
aKW	altes Kraftwerk (old power plant)
BFS	Breadth First Search
BWL	Bergwerk Luisenthal (mine Luisenthal)
CAD	Computer Aided Design
CMC I/II	first/second Critical Micelle Concentration
DM	Deutsche Mark
DN	Nominal Diameter
DR	Drag Reduction
EC_{50}	Effective Concentration
HE	Heat Exchanger
HM	Heat Meter
HOR	Heat Output Reduction
HTR	Heat Transfer Reduction
LC_{50}	Lethal Concentration
LD_{50}	Lethal Dosis
LLL	Lambda Locked Loop
MID	Magnetive-Induktive Flowmeter
NTU	Number of Transfer Units
OHR	Overall Heat Transfer Reduction
PCR	Pumping Cost Reduction
QAV	Quarternary Ammonium Compound
SAlS	Siedlung Altenkesseler Straße
SIS	Shear Induced State
SoSal	Sodiumsalicylate
TDM	Thousand Deutsche Mark
wppm	weight parts per million

Non Dimensional Parameters		**Definition**
Re	Reynolds number	$Re = \frac{\bar{u} \cdot D}{\nu}$
Nu	Nusselt number	$Nu = \frac{\alpha \cdot D}{\lambda}$
Pr	Prandtl number	$Pr = \frac{\nu}{a} = \frac{\eta \cdot c_p}{\lambda}$
Eu	Euler number	$Eu = \frac{\Delta p}{\rho \cdot u^2}$

7. List of References

[1] Abeck, W.:
Erstellung eines Konzepts zur Aufbereitung wässeriger Tensidlösungen,
Diplomarbeit, Universität Dortmund, 1991

[2] Beisemann, D.:
Modellierung der verbrauchergesteuerten Leistungscharakteristik von Fernheizsekundärnetzen,
Diplomarbeit, Universität Dortmund, 1991

[3] BMFT:
Gesamtstudie Fernwärme - TA 4 Wärmetransport und Verteilung,
Bonn, 1977

[4] Borchert, R.:
Toxikologische und ökotoxikologische Bewertung kationischer Tenside,
Studienarbeit, Universität Dortmund, 1990

[5] Bräuer, S.:
Simulation des Betriebsverhaltens einer Fernwärmeschiene,
Diplomarbeit, Universität Dortmund, 1992

[6] Brauer, H.:
Grundlagen der Einphasen- und Mehrphasenströmung,
Verlag Sauerländer, Aarau, 1971

[7] Christofides, N.:
Graph Theory,
Academic Press, New York, 1975

[8] DECHEMA:
Abschlußbericht zum Forschungsvorhaben 0328710H, 1993,
Korrosionsverhalten von Werkstoffen in tensidhaltigen Kreislaufwässern unterschiedlicher Qualität

[9] Fankhänel, M.:
Druckverlust und Wärmeübergang in Fernwärmenetzen bei Einsatz von mizellaren Widerstandsverminderern,
Dissertation, Universität Dortmund, 1989

[10] Fankhänel, M.:
Druckverlust und Wärmeübergang in Fernwärmeleitungen bei Zugabe von Reibungsminderern,
Abschlußbericht zum BMFT-Vorschungsvorhaben 03E5539 B, 1989

[11] Gaddis, E.S.; Schlünder, E.U.:
Temperaturverlauf und übertragene Wärmemenge in Röhrenkesselapparaten mit Umlenkblechen,
Verfahrenstechnik 9, 1975 12, S. 2671-2679

[12] Gaddis, E.S.:
Über die Berechnung des Austauscherwirkungsgrades und der mittleren Temperaturdifferenz in mehrgängigen Rohrbündelwärmeaustauschern mit Umlenkblechen,
Verfahrenstechnik 12, 1978 3, S. 144-149

[13] Gross, L.:
Temperaturverlauf in einer Heißwasserleitung,
in *Fernwärme international* Jg. 6, 1977, Heft 4

[14] Hakansson, K.:
Handbuch der Fernwärmepraxis,
Vulkan-Verlag, Essen, 3. Aufl. 1986

[15] Hauptausschuß Detergentien U III 3:
Sachstandsbericht: Tenside in Wasch- und Reinigungsmitteln,
Umwelt 2, 1987, S. 71-73

[16] Helmig, J.:
Untersuchungen zum Einsatz der Ultrafiltration für die Abtrennung kationischer Tenside aus Fernheizwasser,
Dissertation, Universität Dortmund, 1992

[17] Helmig, J.:
Entwicklung von Aufarbeitungsverfahren für tensidhaltige Lösungen,
Abschlußbericht zum BMFT-Vorschungsvorhaben 03 287 10F, Teil A, 1994

[18] Heese, R.:
Grundlagen der Projektierung und Beurteilung der Wirtschaftlichkeit von Fernwärmeversorgungssystemen beim Einsatz widerstandsvermindernder Additive,
Diplomarbeit, Universität Dortmund, 1993

[19] Hoechst AG:
DIN-Sicherheitsblätter für Dobon-G, Frankfurt, August 1988

[20] Idelchik, I.E.:
Handbook hydraulic Resistance,
Springer Verlag, 1986

[21] Kleindl, M.:
Abschlußbericht zum Forschungsvorhaben 0328710L, 1993,
Der Einfluß reibungsmindernder Zusätze auf instationäre hydraulische Vorgänge und die Betriebssicherheit von Fernwärmenetzen

[22] Kleuker, H.-H.:
Wärmeübergang und Druckverlust wässeriger Tensidlösungen in Plattenwärmeübertragern,
Dissertation, Universität Dortmund, 1992

[23] Klöpper, K.:
Erstellung eines Gesamtkonzeptes zur Anwendung widerstandsvermindernder Additive in Fernwärmentzen,
Abschlußbericht zum BMFT-Forschungsvorhaben 03 287 10F, 1995

[24] Klöpper, K.; Althaus, W.; Wefels, P.; Besch, H.:
Langzeiteinsatz von Reibungsminderern im Fernwärmenetz Völklingen-Luisenthal,
in *Fernwärme International*, Jg. 25, 1995, Heft 4/5, S. 412

[25] Klöpper, K.:
Einsatz widerstandsvermindernder Additive in Fernwärmesystemen,
Dissertation, Universität Dortmund, 1996

[26] Martin, H.:
Wärmeübertrager,
Georg Thieme Verlag, Stuttgart - New York, 1988

[27] Mollenkopf, G.:
Einfluß von Tensidlösungen in Fernheizsystemen auf das Betriebs- und Regelverhalten von Kreiselpumpen,
Abschlußbericht zum Forschungsvorhaben 0328710 G, 1991

[28] Munser, H.:
Fernwärmeversorgung,
VEB Deutscher Verlag für Grundstoffindustrie, Leipzig, 1983

[29] Noeres, P.:
Einsatzmöglichkeiten von Reibungsminderern in Fernwärmetransportsystemen,
Diplomarbeit, Universität Dortmund, 1993

[30] Reher, U.:
Rechnergestützte Systemanalyse der Fernwärmeschiene Saar zur Beurteilung der Wirtschaftlichkeit von Reibungsminderern,
Diplomarbeit, Universität Dortmund, 1994

[31] Rippen, G.:
Handbuch Umweltchemikalien,
Ecomed Verlagsgesellschaft mbH, Landsberg/Lech, 2. Aufl. 1987

[32] Schulz, H.:
Die Pumpen,
Springer Verlag, 13. Auflage, Berlin, New York, 1977

[33] Schwarz, M.:
Simulation von Rohrbündel- und Rohrwendelwärmeaustauschern beim Einsatz kationischer Reibungsminderer,
Studienarbeit, Universität Dortmund, 1993

[34] Stahr, D.:
Einfluß reibungsmindernder Tensidlösungen auf die Meßgenauigkeit von Woltmann-Zählern, Staudruck- und Wirbeldurchflußmeßzählern,
Diplomarbeit, Universität Dortmund, 1992

[35] Strauß, K.:
Vorlesung Strömungsmechanik II,
Universität Dortmund 1993

[36] Troskiewicz, J.:
Optimierung von Fernwärmesystemen bei großen Transportentfernungen,
in *Fernwärme international* 15, 1986 Nr. 2, S. 86

[37] Ullrich, G.D.:
A guide to chemical engineering process design,
John Wiley & Sons inc., 1984

[38] VDI-Wärmeatlas:
Berechnungsblätter für den Wärmeübergang,
VDI-Verlag GmbH, 5. Aufl., Düsseldorf, 1988.

[39] VDI-Wärmeatlas:
Berechnungsblätter für den Wärmeübergang,
VDI-Verlag GmbH, 6. Aufl., Düsseldorf, 1991

[40] Verfahrenstechnische Berechnungsmethoden:
Teil 1: Wärmeübertrager,
VCH Verlagsgesellschaft mbH, Weinheim, 1987

[41] Weber, M.:
Wärmeübergang und Druckverlust wässeriger Tensidlösungen in Rohren und Rohrwendeln,
Dissertation, Universität Dortmund, 1990

[42] Wefels, P.; Besch, H.; Klöpper, K.; Bräuer, S.; Althaus, W.; Steiff, A.; Weinspach, P.-M.:
Anwendung und Erprobung mizellarer Reibungsminderer in Fernwärmesystemen,
Abschlußbericht zum BMFT Forschungvorhaben 03 287 10 B, 1994

[43] Werner, U.:
Mechanische Verfahrenstechnik,
Skriptum zur Vorlesung, Unversität Dortmund, 1991

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_4/P_5 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 compensated 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.

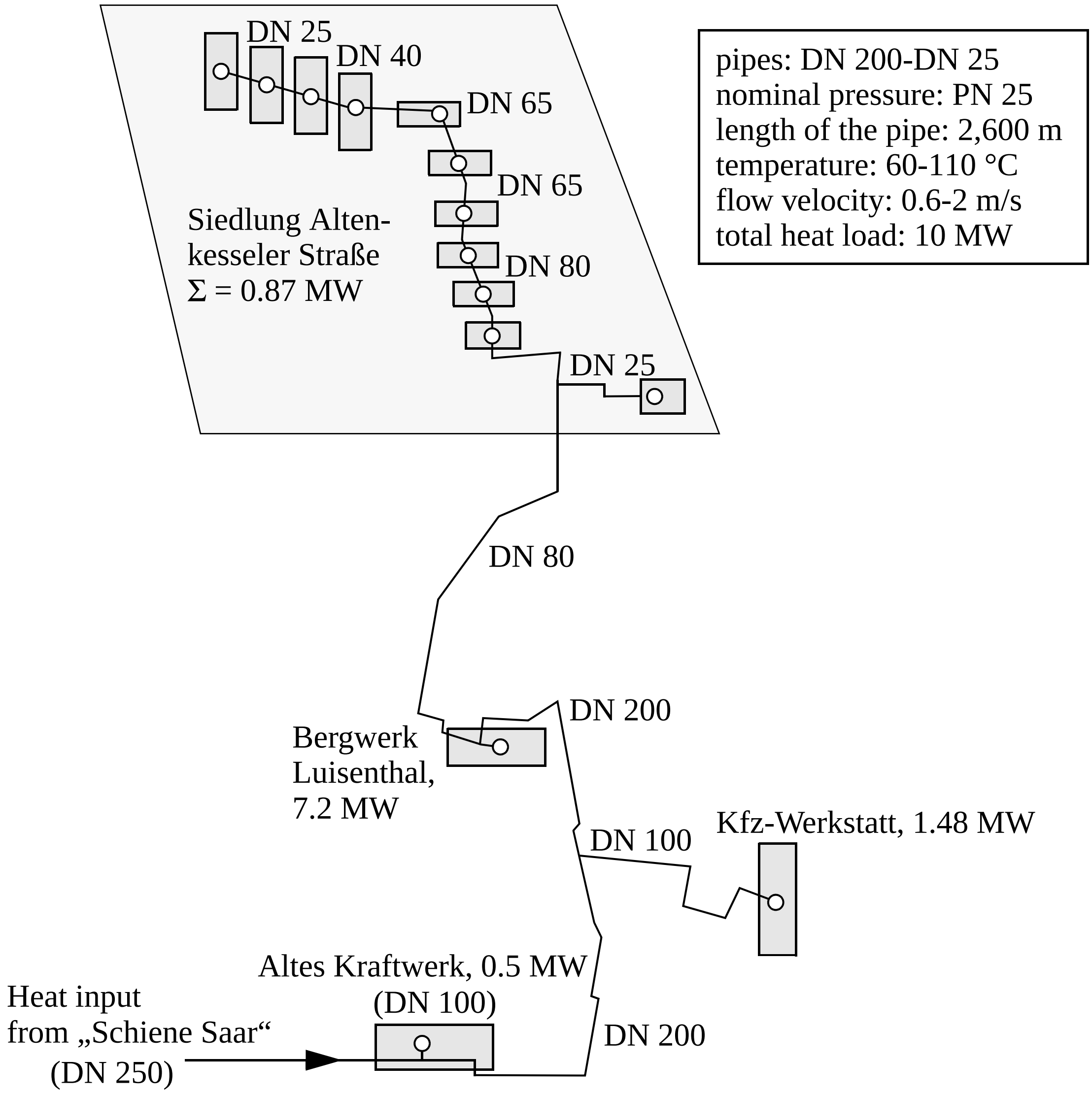


Fig. 8.1: Plan of the system Luisenthal [24]

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**.

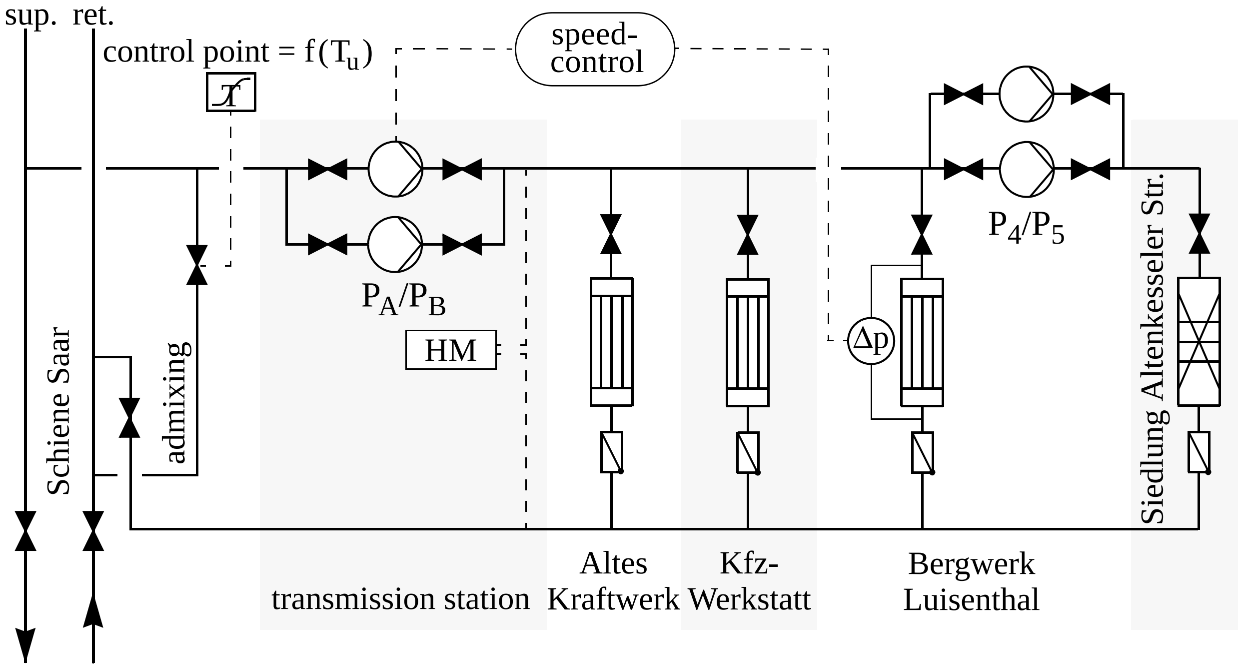


Fig. 8.2: Schematic flow sheet of the origin system

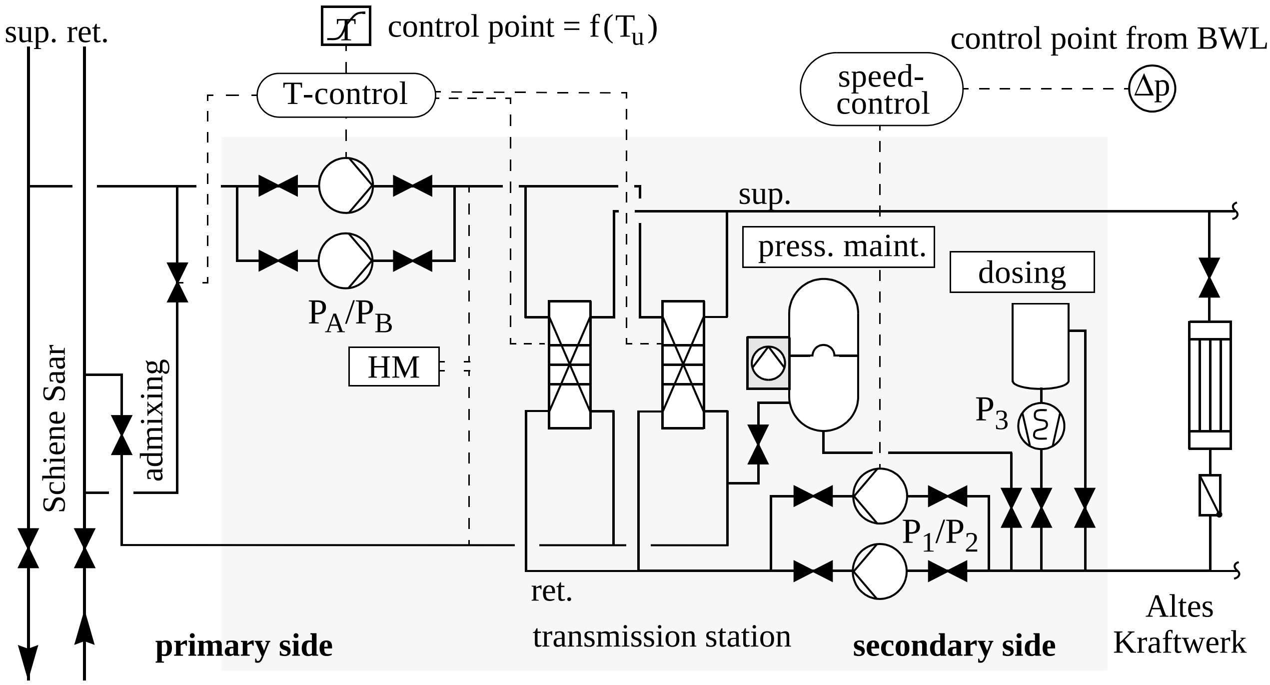


Fig. 8.3: Schematic flow sheet of the modified system (until „altes Kraftwerk")

The heat exchange area for the plate heat exchangers which operate with surfactant solution are designed 34 % larger then those for pure wtaer. Therefore the number of plates has been increased from 83 to 111.

		primary side	secondary side
requirements	fluid	water	water
	flow rate	82 m^3/h	132.5 m^3/h
	entrance temp.	130 °C	60 °C
	outlet temp.	65 °C	100 °C
	nominal heat load	6.0 MW	
results	area	73 m^2	
	number of plates/type	83/σ85	
	height of the plates	2.496 mm	
	breadth of the plates	850 mm	
	pressure loss	300 mbar	750 mbar

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

		primary side	secondary side
	fluid	water	surfactant
results	area	97.9 m^2	
	number of plates/ type	111/σ85	
	pressure loss	200 mbar	450 mbar

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 Luisenhal. 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 diagramms 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.

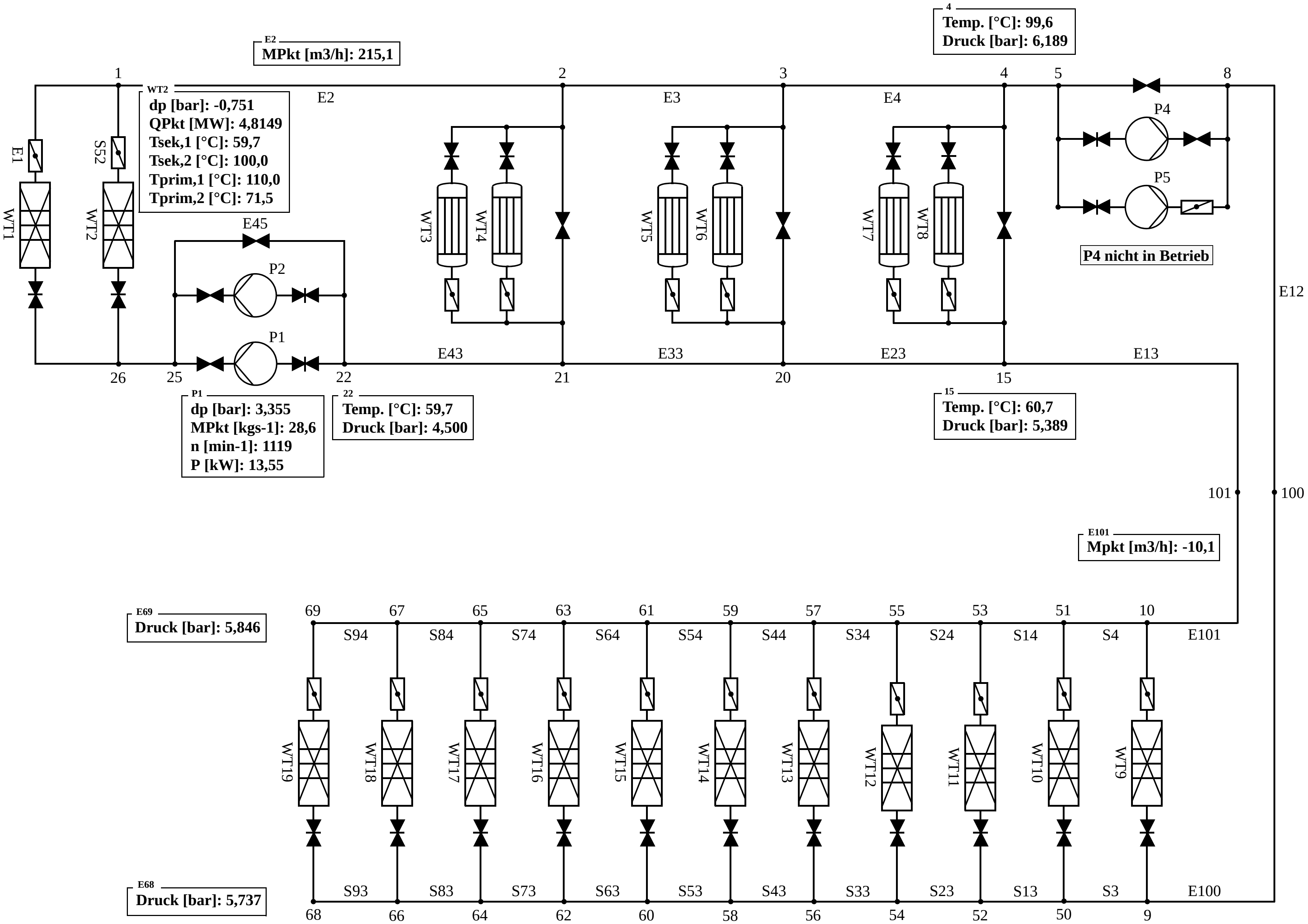


Fig. 7.4: Flow sheet with the results of the simulation „water **without** P_4/P_5“

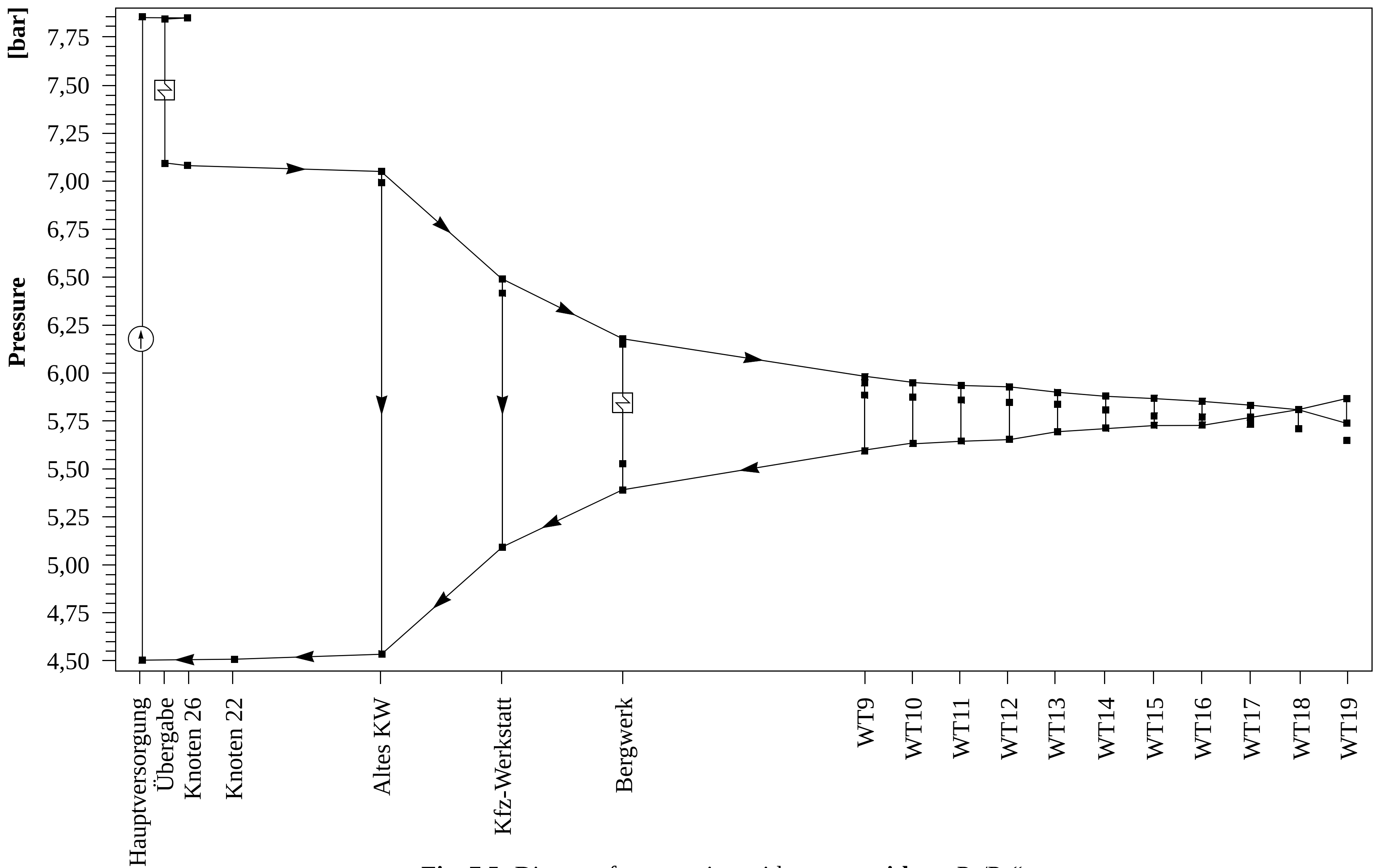


Fig. 7.5: Diagram for operation with „water **without** P_4/P_5“

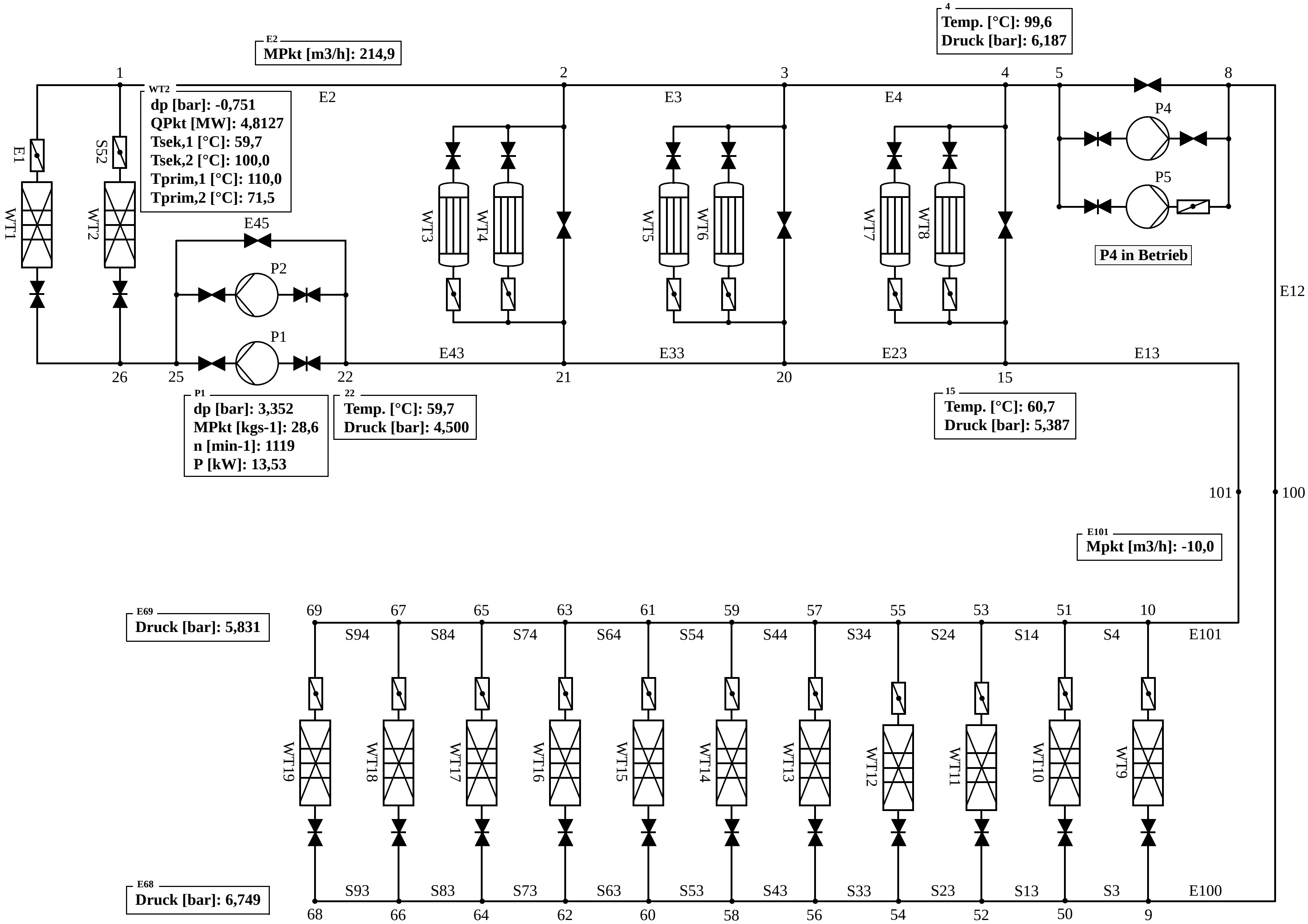


Fig. 7.6: Flow sheet with the results of the simulation „water **with** P_4/P_5“

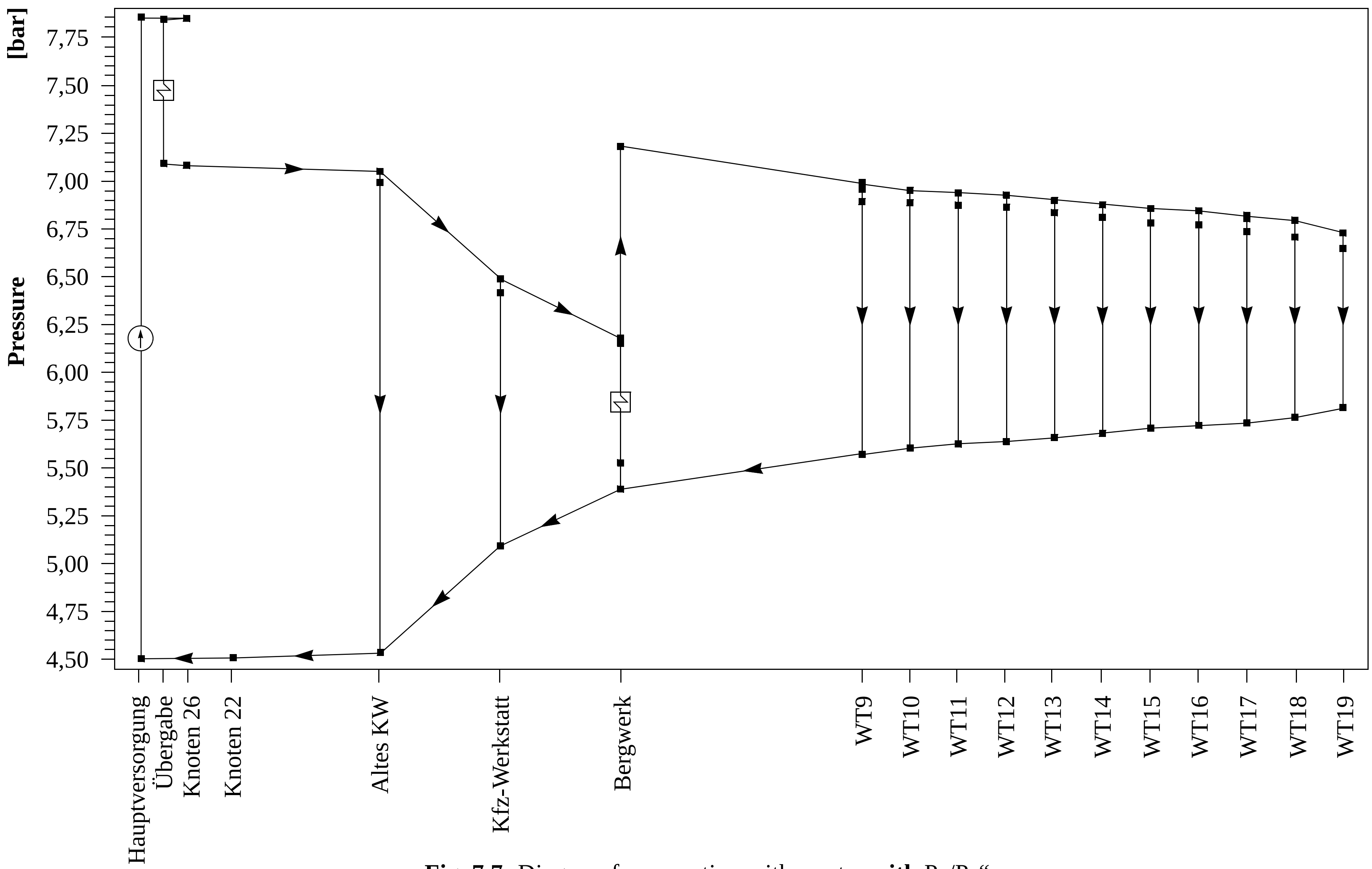


Fig. 7.7: Diagram for operation with „water **with** P_4/P_5“

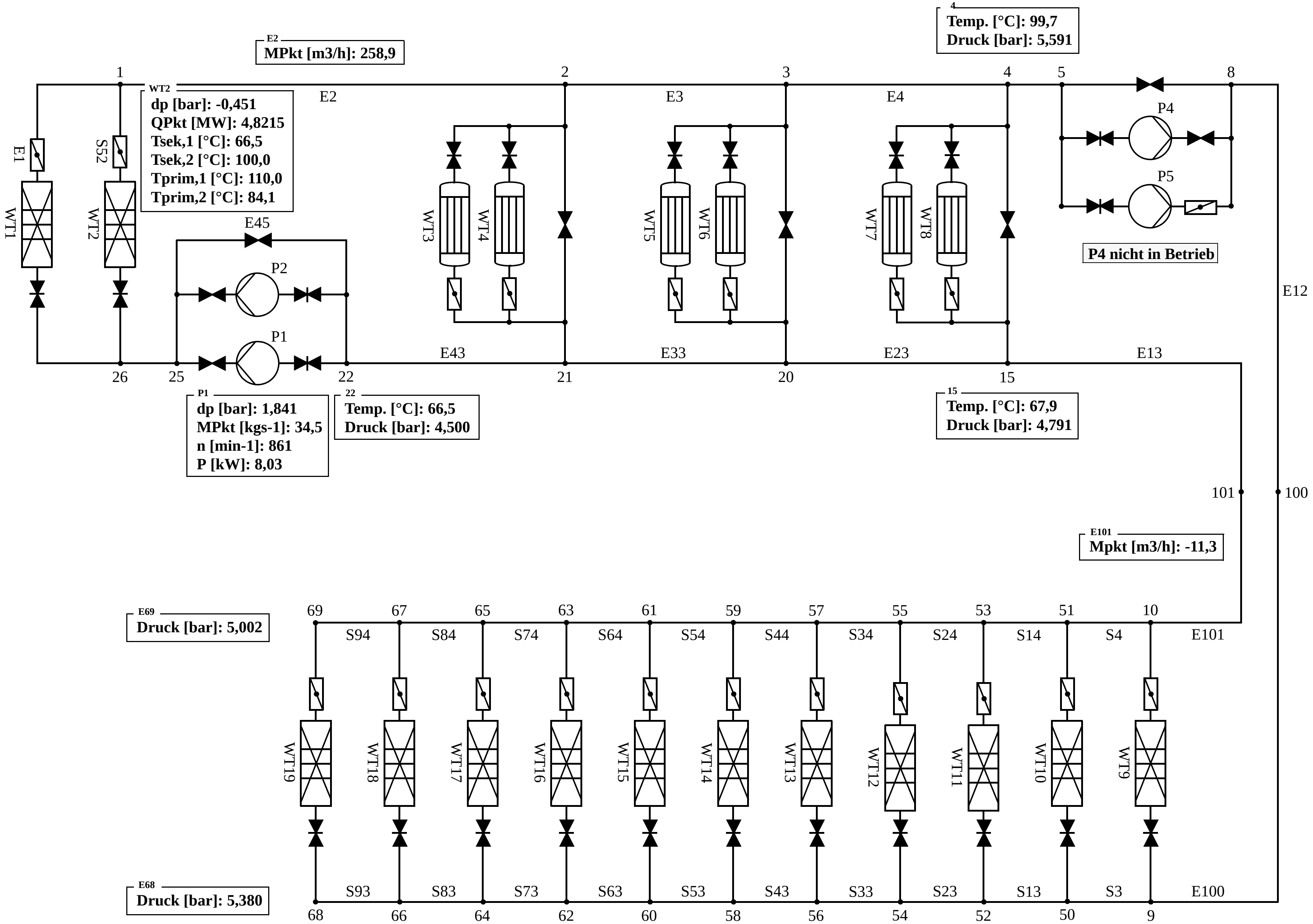


Fig. 7.8: Flow sheet with the results of the simulation „surfactants **without** P_4/P_5“

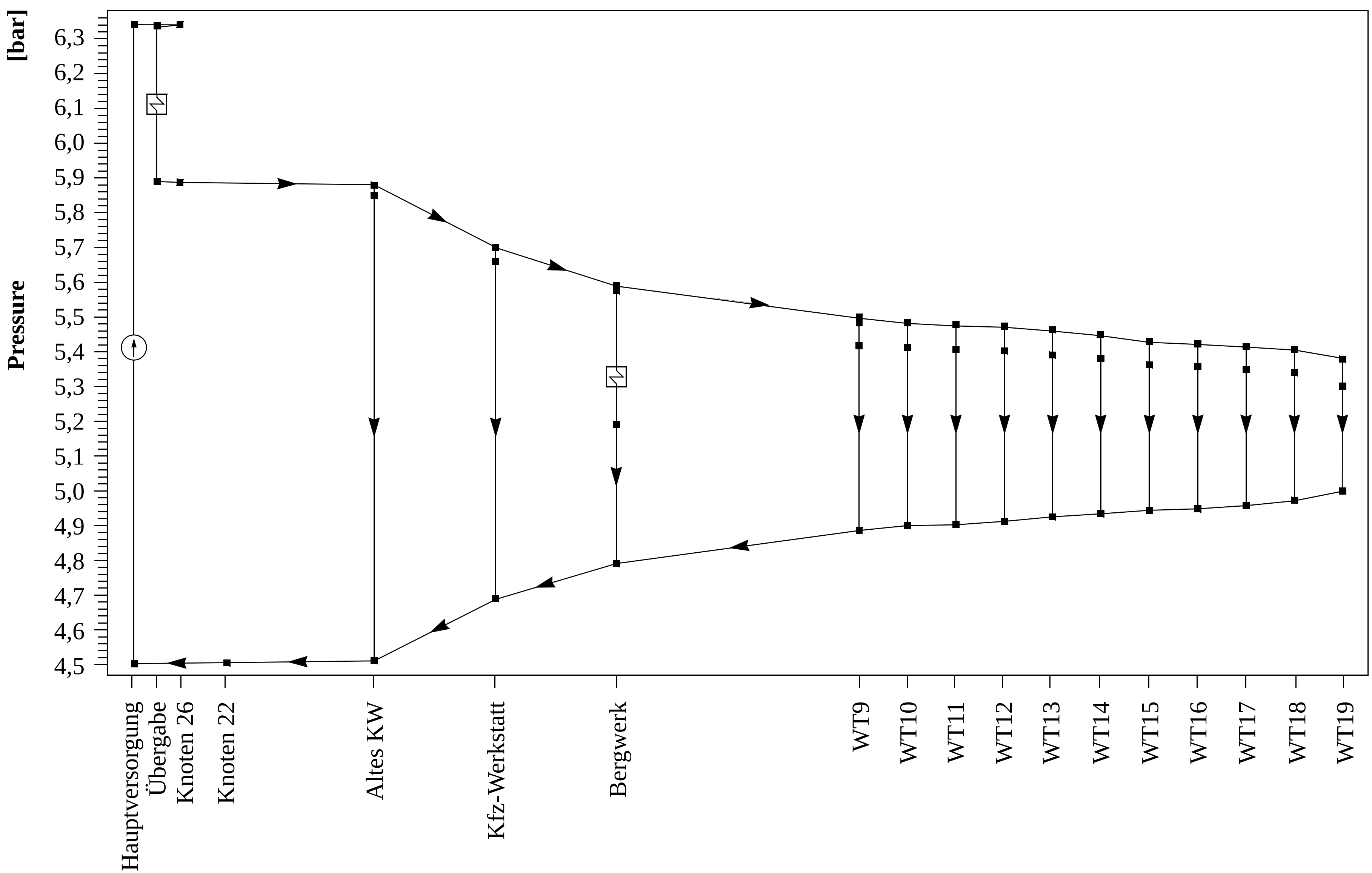


Fig. 7.9: Diagram for operation with „surfactants **without** P_4/P_5“

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.

result	place	fluid		difference
		water	surfactants	
pressure loss	pump P1	3.35 bar	1.84 bar	1.51 bar
	pump P4	1.00 bar	-	1.00 bar
	sum	4.35 bar	1.84	2.51 bar
return temperature	primary	71.5 °C	84.1 °C	12.6 °C
	secondary	59.7 °C	66.5 °C	6.8 °C
flow rate	primary	212 m^3/h	330,0 m^3/h	118 m^3/h
	secondary	215 m^3/h	258,9 m^3/h	43,9 m^3/h

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 occure 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.

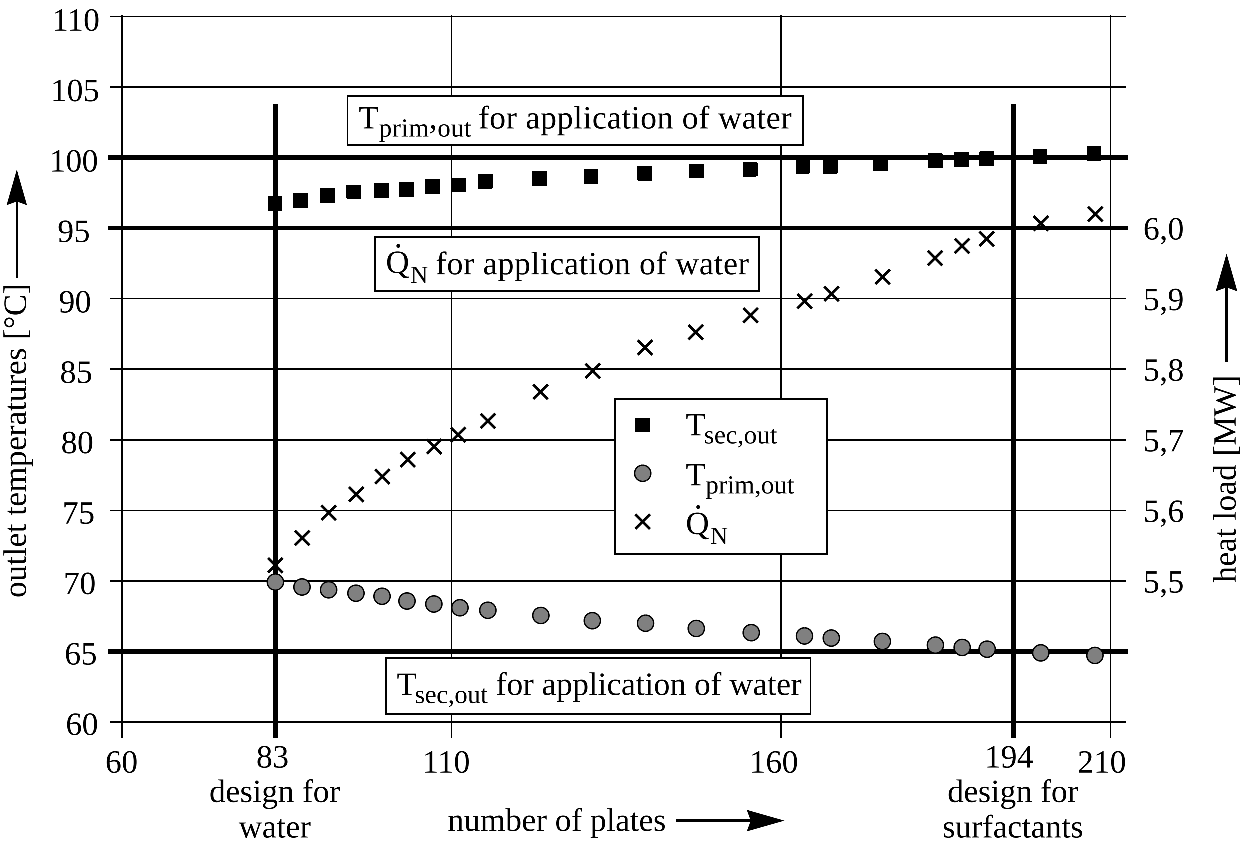


Fig. 8.10: Example for the design of a plate heat exchanger operating with drag reducing additives

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 correspondance 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.

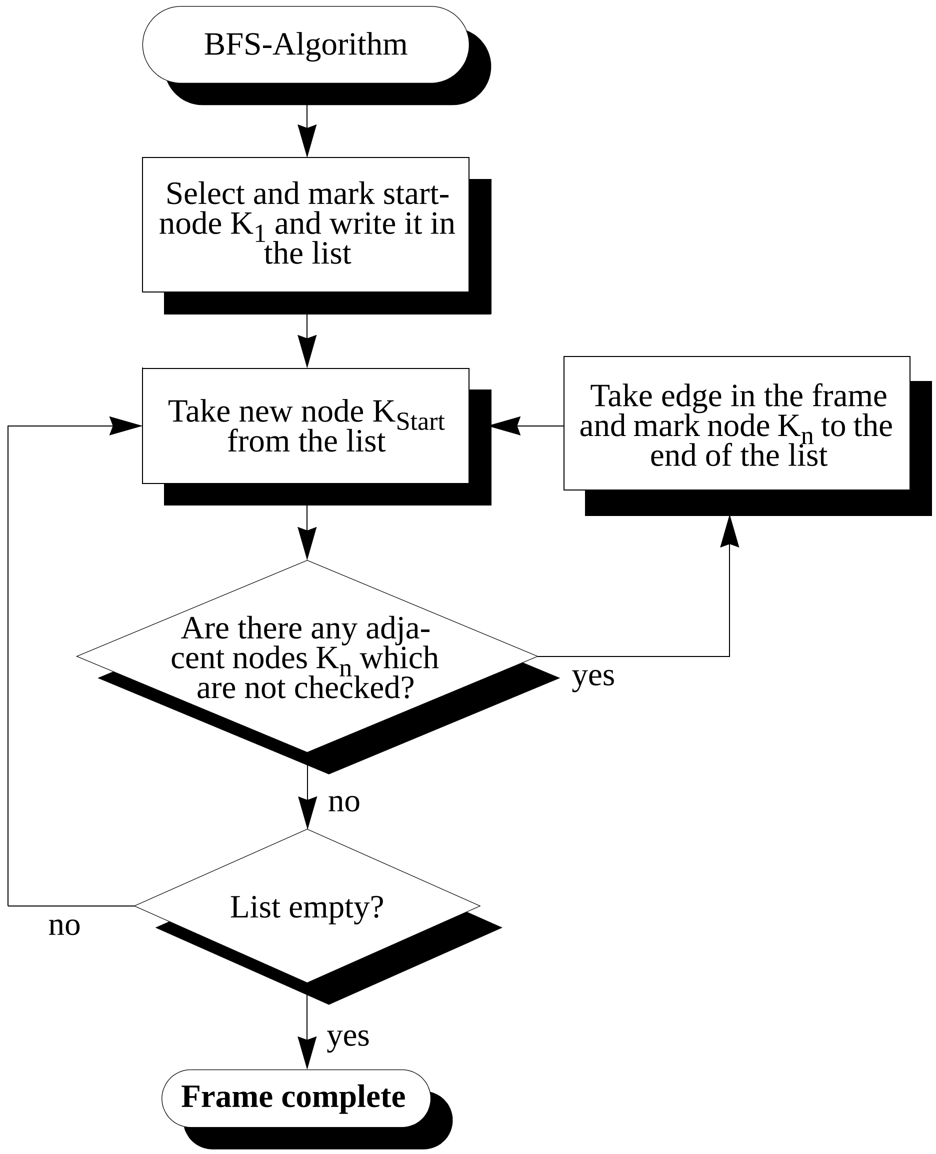
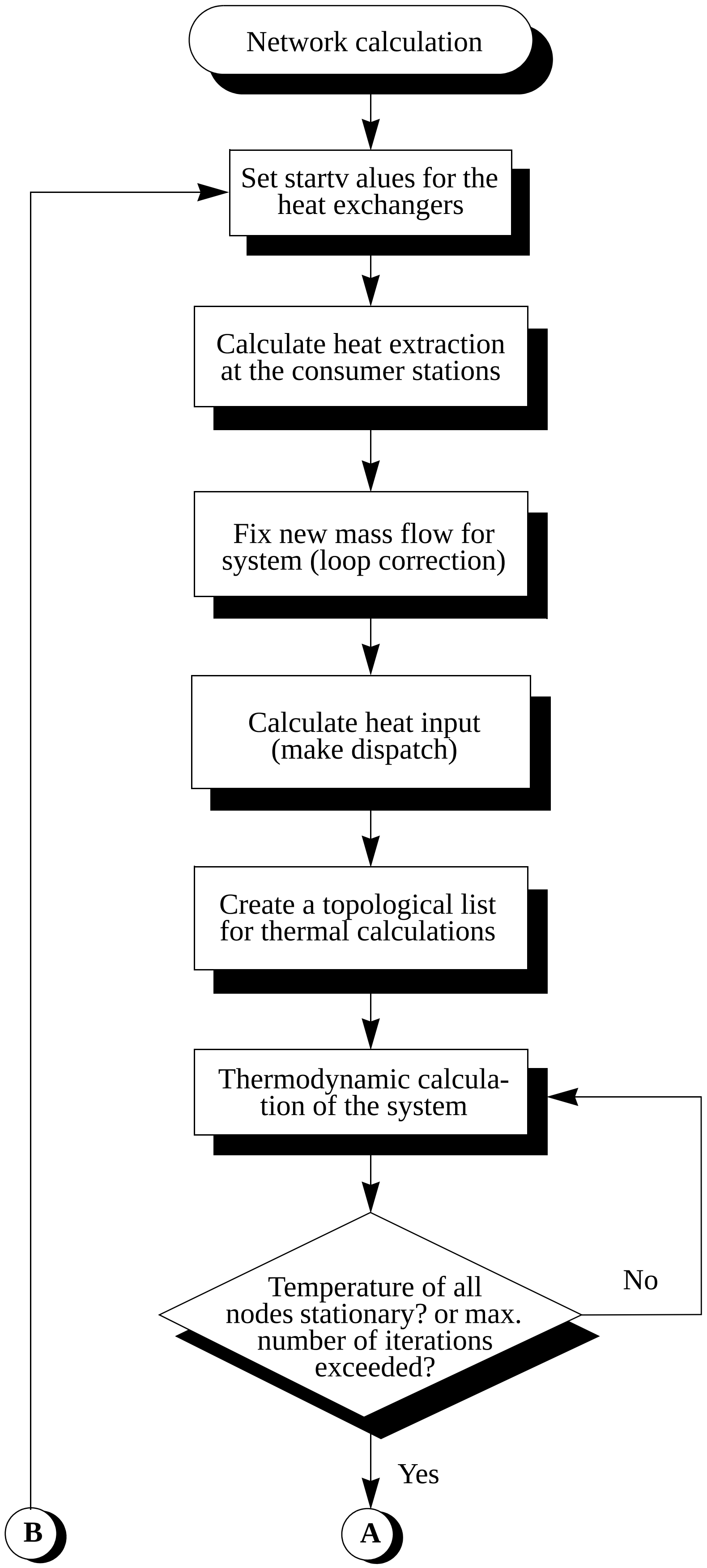


Fig. 8.11: Flow sheet of the BFS-Algorithm [5]

8.2.2 Flow chart for the calculation of network



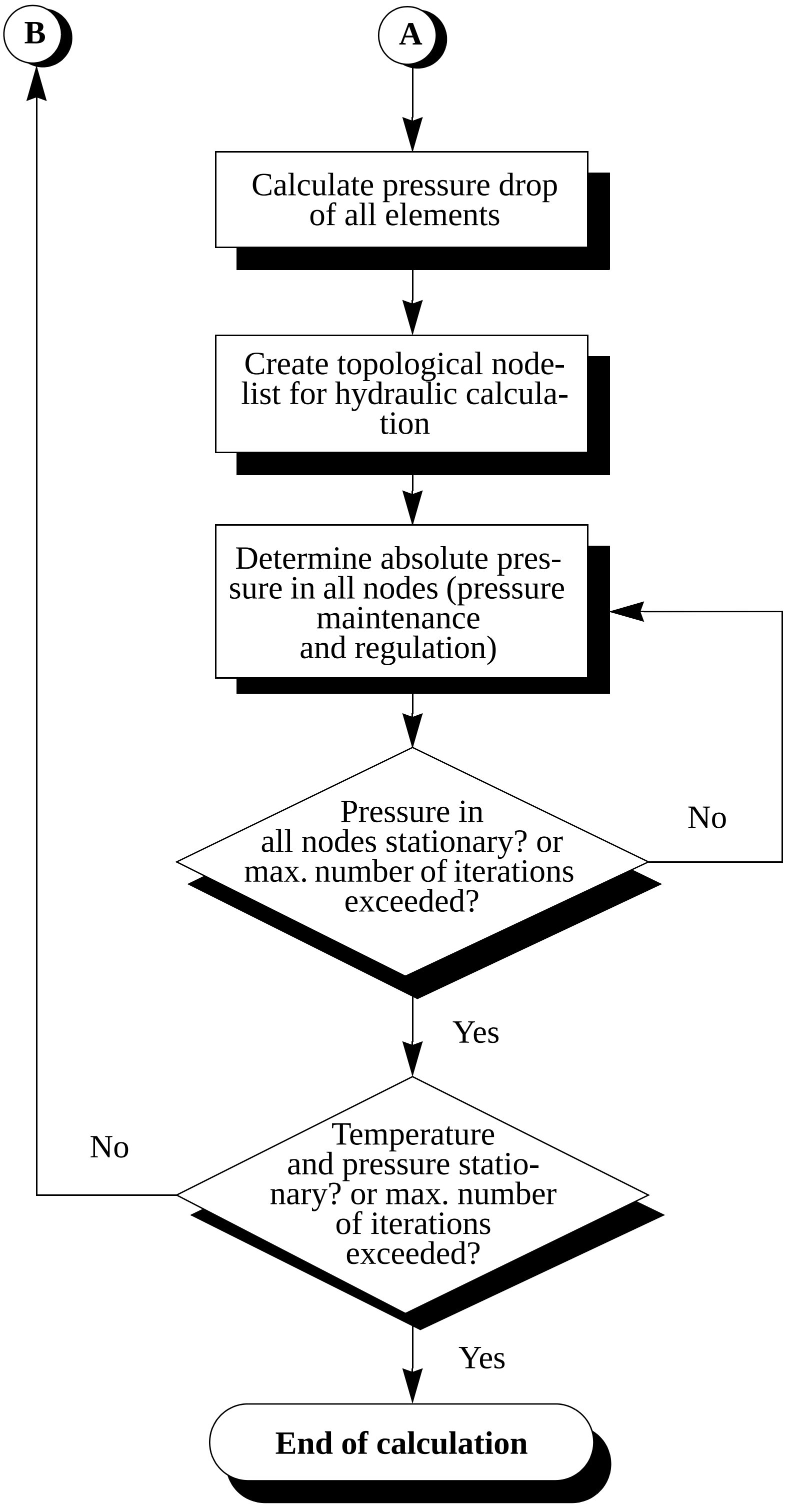
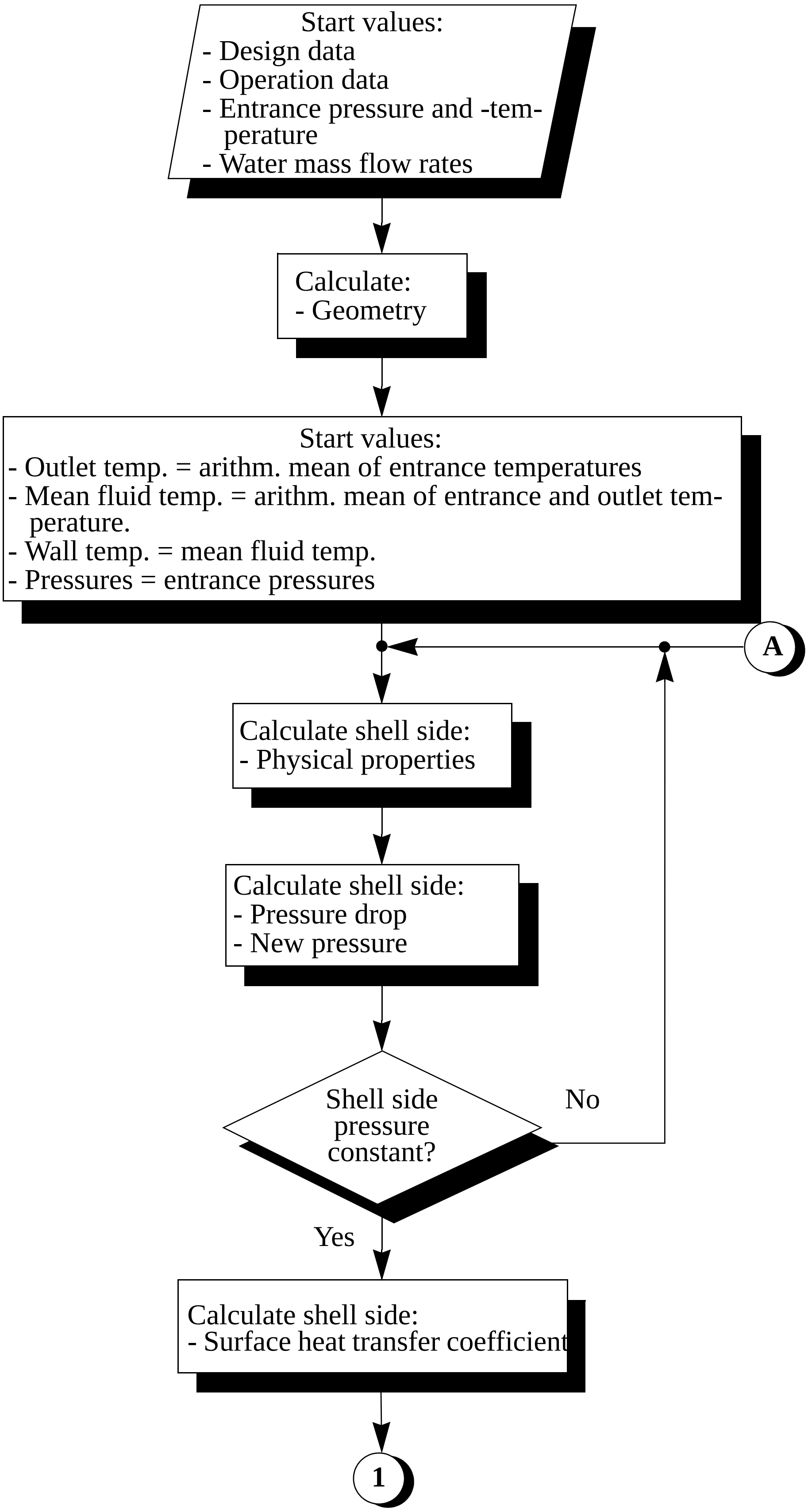
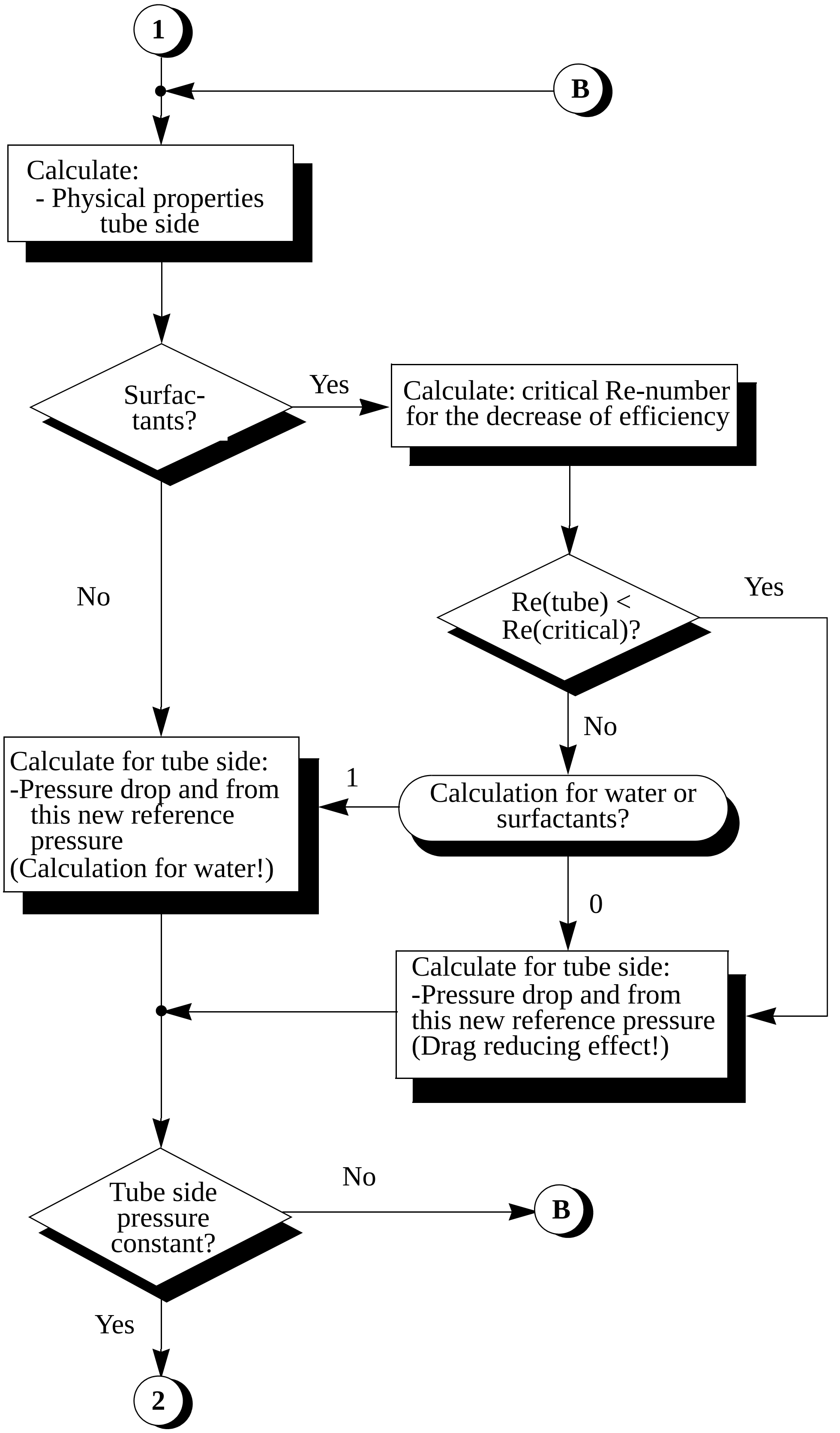


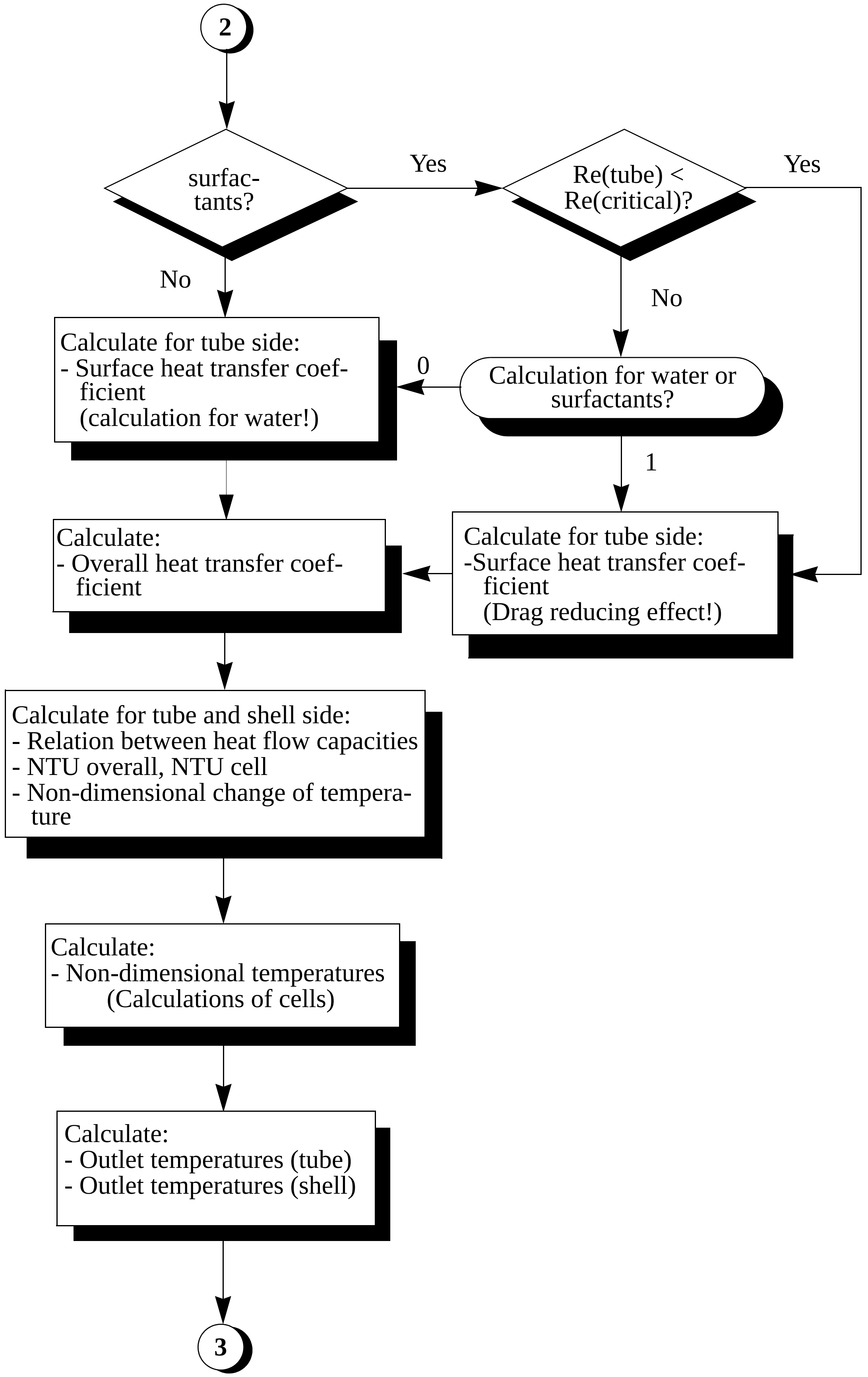
Fig. 8.12: Flow chart of the system calculation [5]

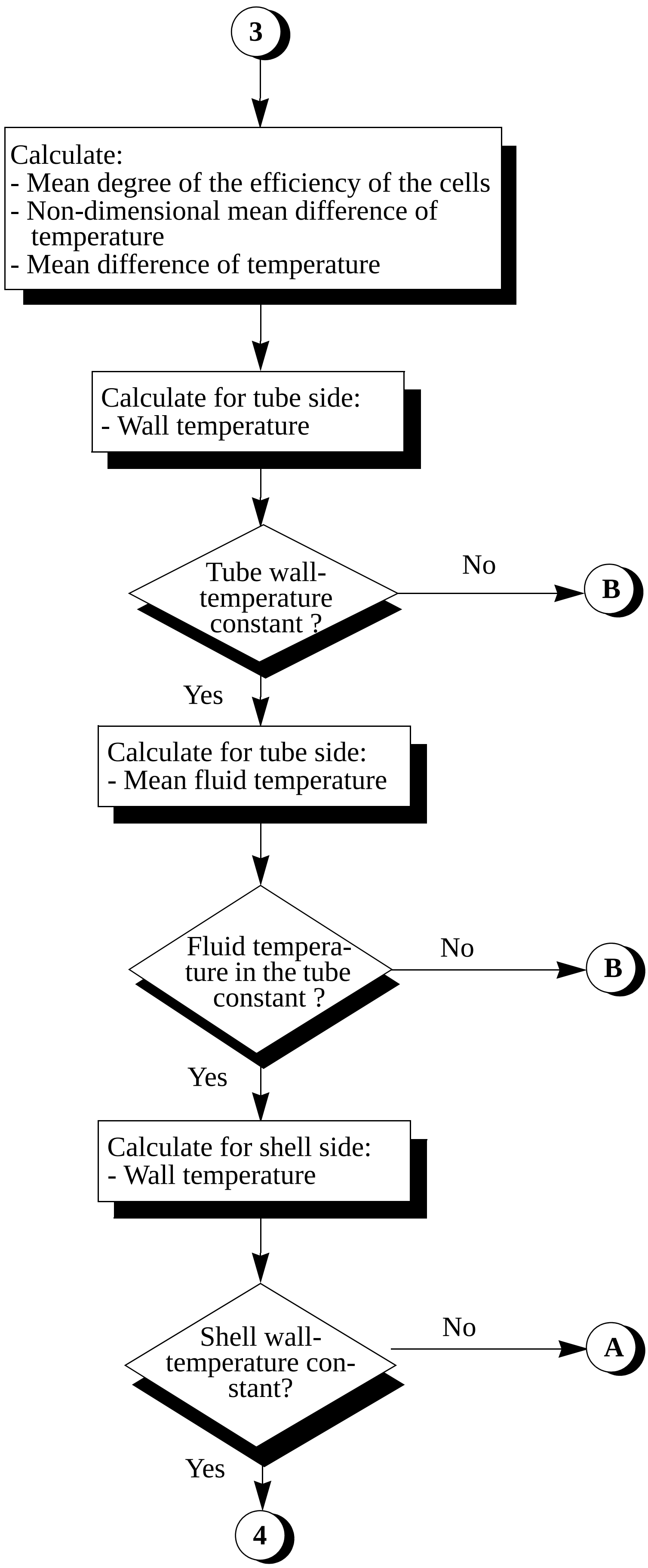
8.2.3 Flow charts for the calculation of heat exchangers

Shell and tube heat exchangers




1
B
Calculate:
- Physical properties tube side
Surfac-tants?
Yes
Calculate: critical Re-number for the decrease of efficiency
No
Re(tube) < Re(critical)?
Yes
No
Calculate for tube side:
-Pressure drop and from this new reference pressure
(Calculation for water!)
1
Calculation for water or surfactants?
0
Calculate for tube side:
-Pressure drop and from this new reference pressure
(Drag reducing effect!)
Tube side pressure constant?
No
B
Yes
2


2
surfac-
tants?
Yes
Re(tube) <
Re(critical)?
Yes
No
No
Calculate for tube side:
- Surface heat transfer coef-
ficient
(calculation for water!)
0
Calculation for water or
surfactants?
1
Calculate:
- Overall heat transfer coef-
ficient
Calculate for tube side:
-Surface heat transfer coef-
ficient
(Drag reducing effect!)
Calculate for tube and shell side:
- Relation between heat flow capacities
- NTU overall, NTU cell
- Non-dimensional change of tempera-
ture
Calculate:
- Non-dimensional temperatures
(Calculations of cells)
Calculate:
- Outlet temperatures (tube)
- Outlet temperatures (shell)
3


3
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 ?
No
B
Yes
Calculate for tube side:
- Mean fluid temperature
Fluid tempera-ture in the tube constant ?
No
B
Yes
Calculate for shell side:
- Wall temperature
Shell wall-temperature con-stant?
No
A
Yes
4

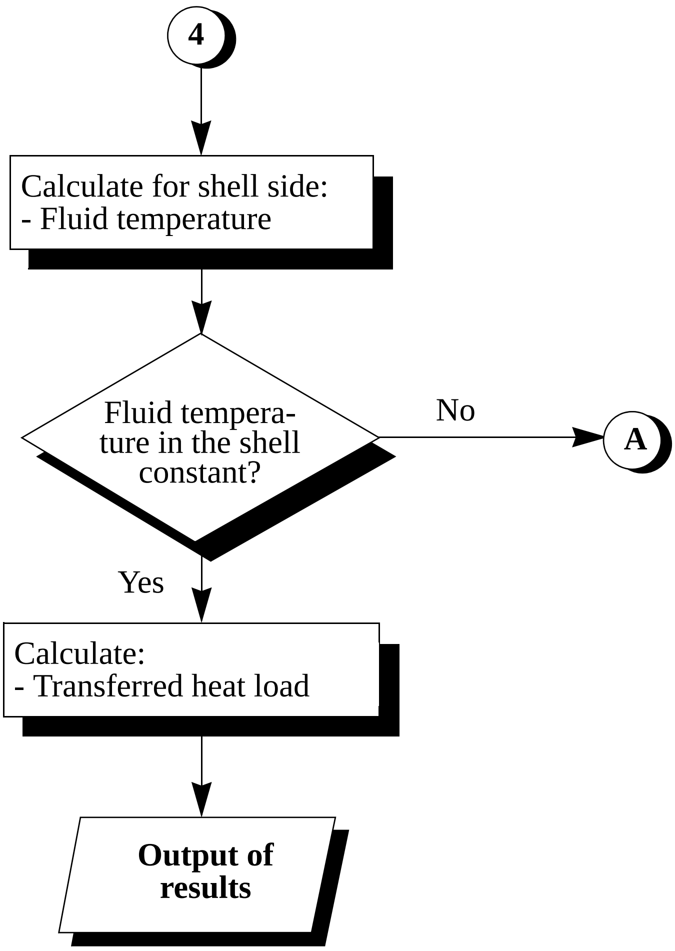
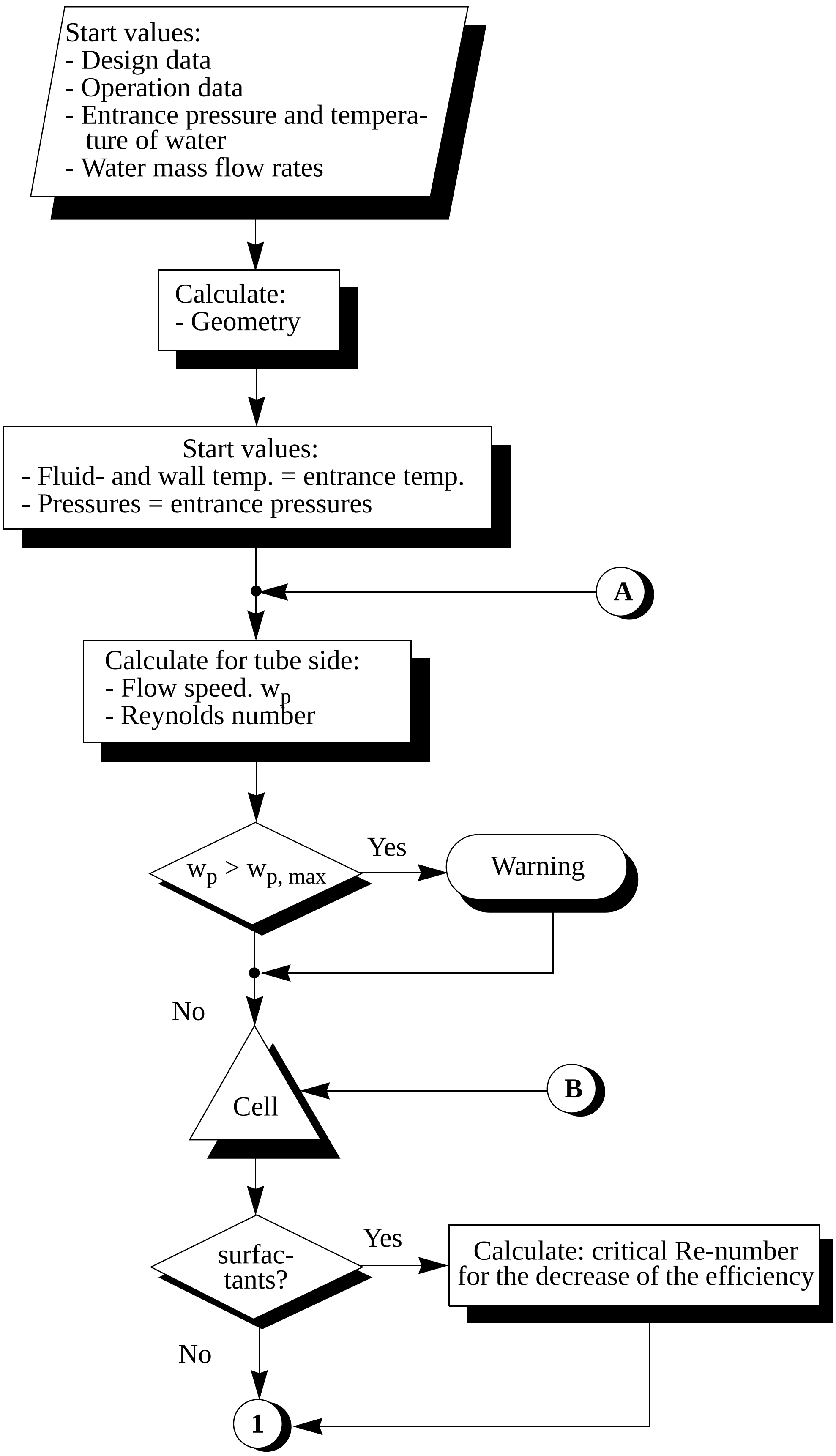
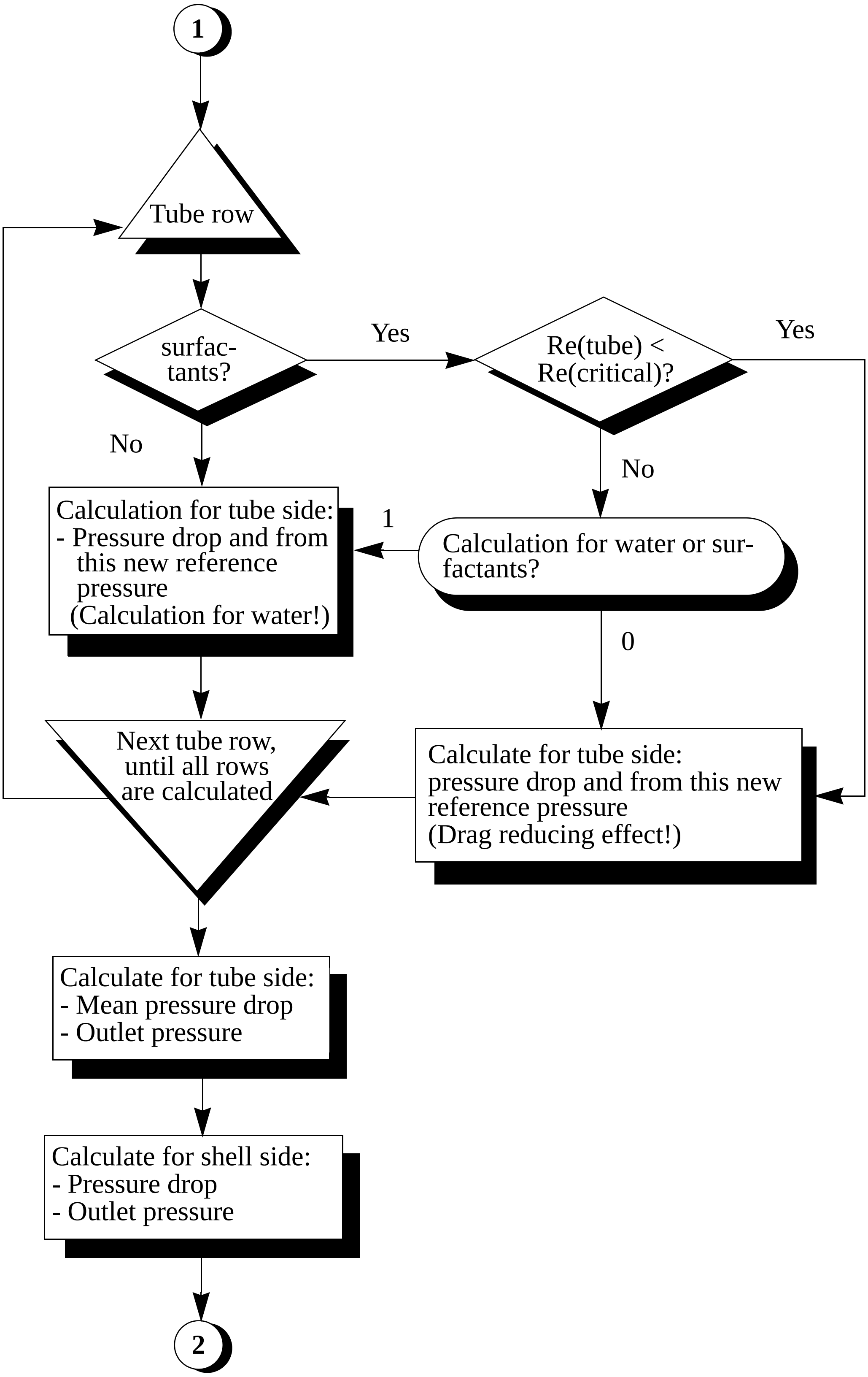
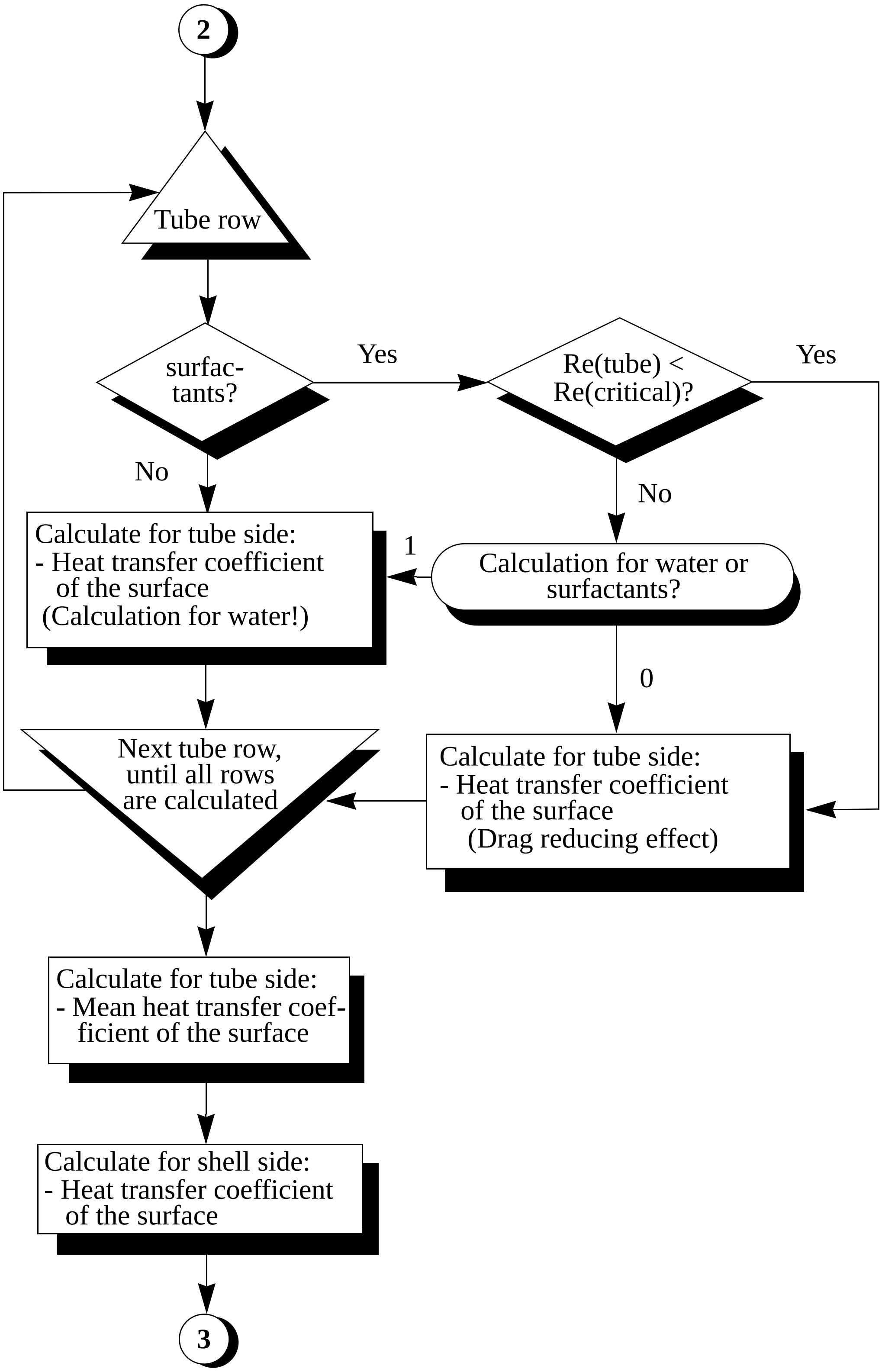


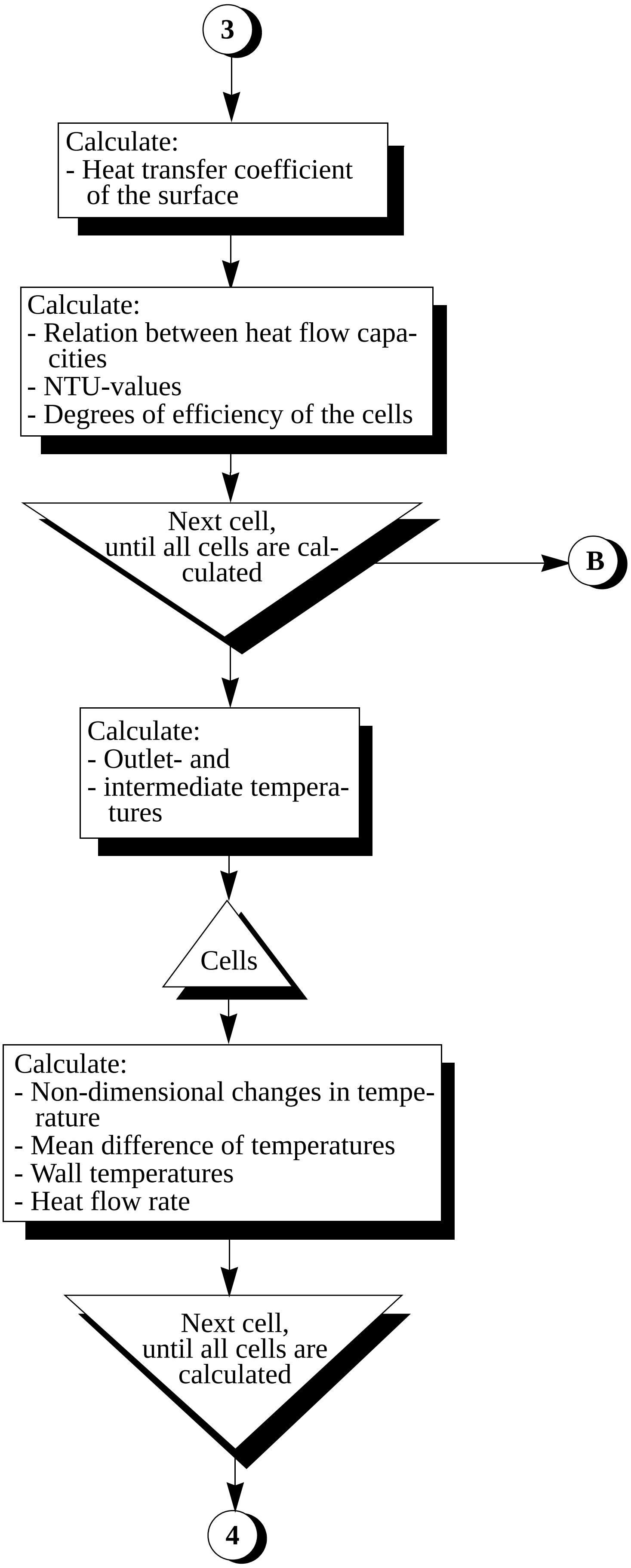
Fig. 8.13: Flow charts for the calculation of heat exchanger [5]

Helical tube heat exchanger




1
Tube row
surfac-
tants?
Yes
Re(tube) <
Re(critical)?
Yes
No
No
Calculation for tube side:
- Pressure drop and from
this new reference
pressure
(Calculation for water!)
1
Calculation for water or sur-
factants?
0
Next tube row,
until all rows
are calculated
Calculate for tube side:
pressure drop and from this new
reference pressure
(Drag reducing effect!)
Calculate for tube side:
- Mean pressure drop
- Outlet pressure
Calculate for shell side:
- Pressure drop
- Outlet pressure
2


2
Tube row
surfac-
tants?
Yes
Re(tube) <
Re(critical)?
Yes
No
No
Calculate for tube side:
- Heat transfer coefficient
of the surface
(Calculation for water!)
1
Calculation for water or
surfactants?
0
Next tube row,
until all rows
are calculated
Calculate for tube side:
- Heat transfer coefficient
of the surface
(Drag reducing effect)
Calculate for tube side:
- Mean heat transfer coef-
ficient of the surface
Calculate for shell side:
- Heat transfer coefficient
of the surface
3


3
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
B
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
4

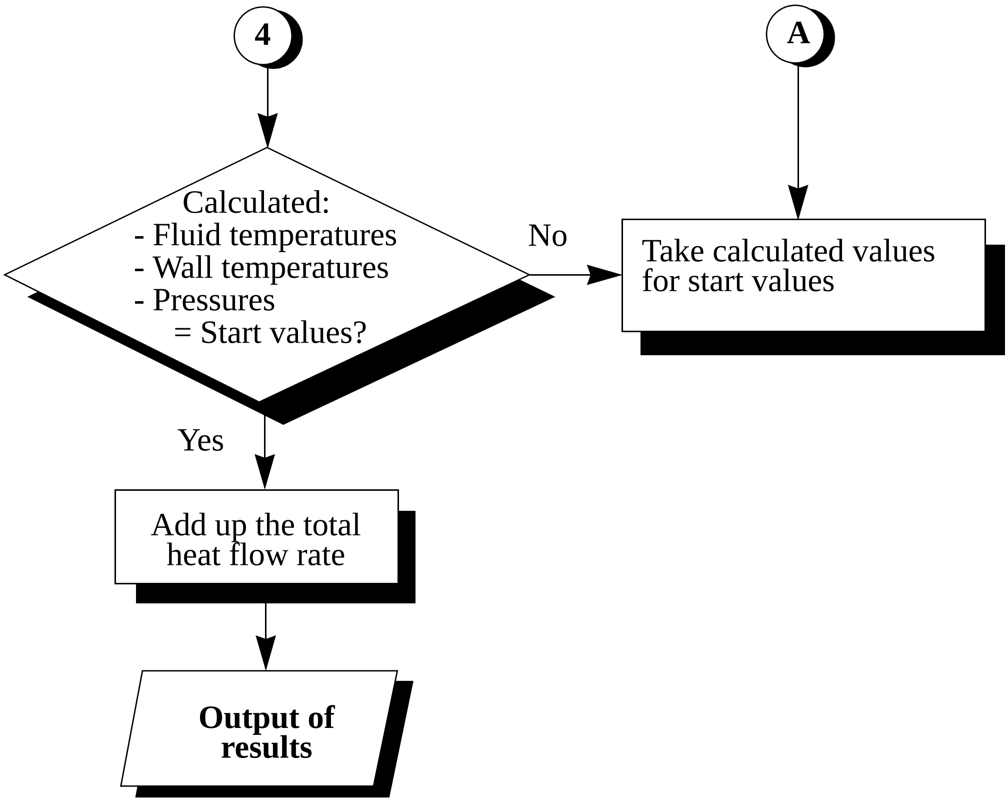
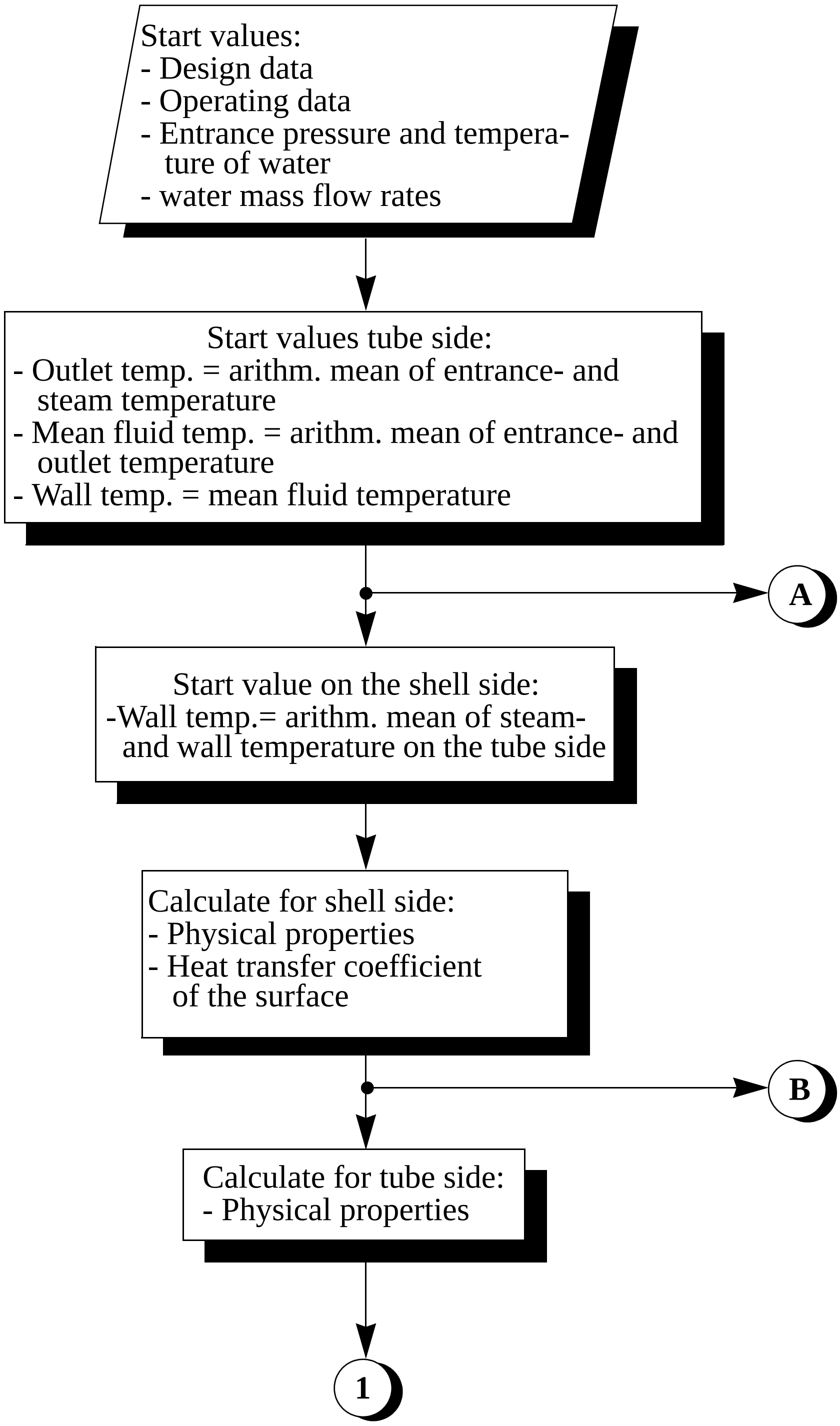
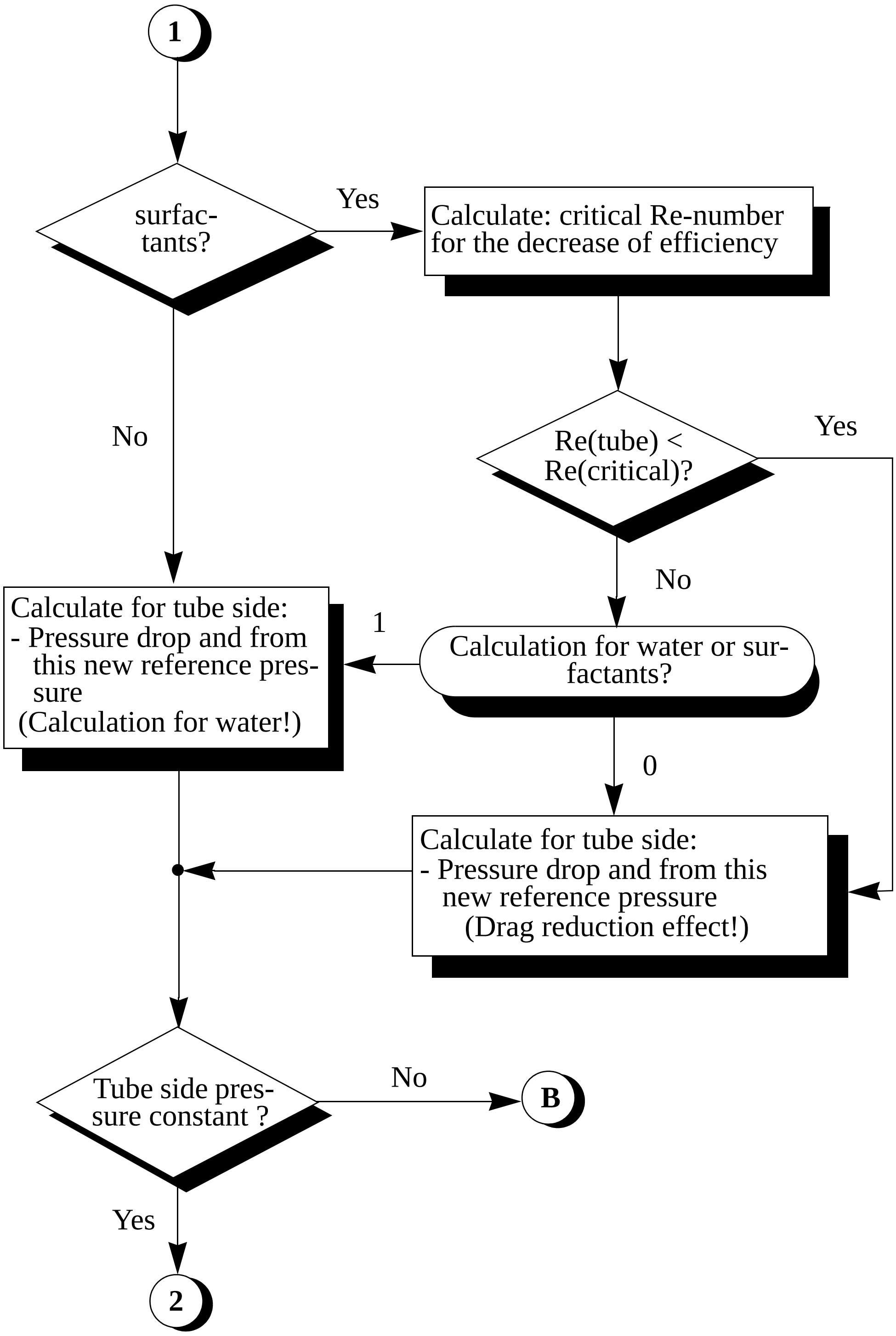
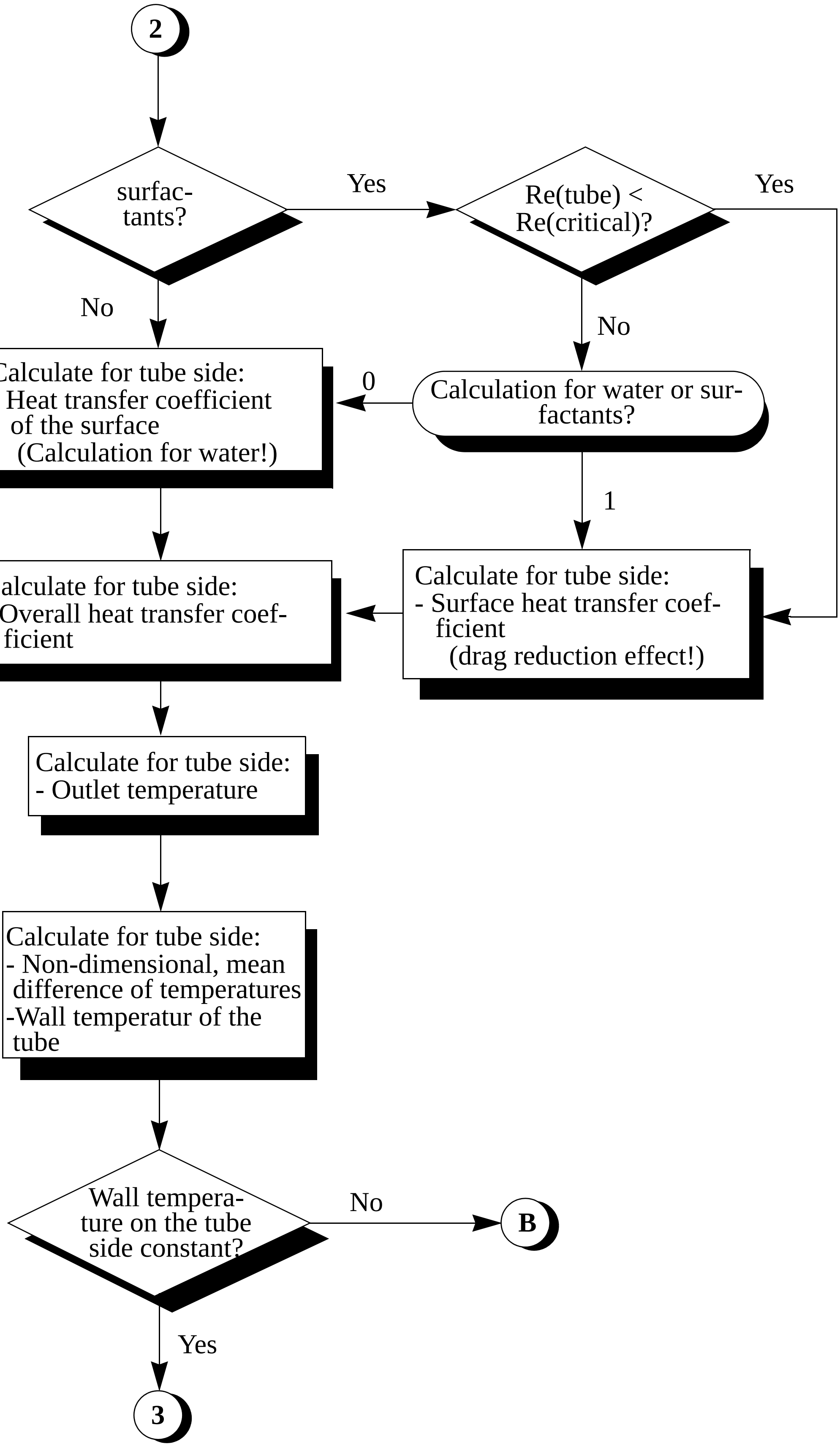


Abb. 8.14: Flow chart for the calculation of helical tube heat exchangers

Shell and tube bundle condenser




1
surfac-
tants?
Yes
Calculate: critical Re-number
for the decrease of efficiency
No
Re(tube) <
Re(critical)?
Yes
No
Calculate for tube side:
- Pressure drop and from
this new reference pres-
sure
(Calculation for water!)
1
Calculation for water or sur-
factants?
0
Calculate for tube side:
- Pressure drop and from this
new reference pressure
(Drag reduction effect!)
Tube side pres-
sure constant ?
No
B
Yes
2


2
surfac-
tants?
Yes
No
Re(tube) <
Re(critical)?
Yes
No
Calculate for tube side:
- Heat transfer coefficient
of the surface
(Calculation for water!)
0
Calculation for water or sur-
factants?
1
Calculate for tube side:
- Overall heat transfer coef-
ficient
Calculate for tube side:
- Surface heat transfer coef-
ficient
(drag reduction effect!)
Calculate for tube side:
- Outlet temperature
Calculate for tube side:
- Non-dimensional, mean
difference of temperatures
-Wall temperatur of the
tube
Wall tempera-
ture on the tube
side constant?
No
B
Yes
3

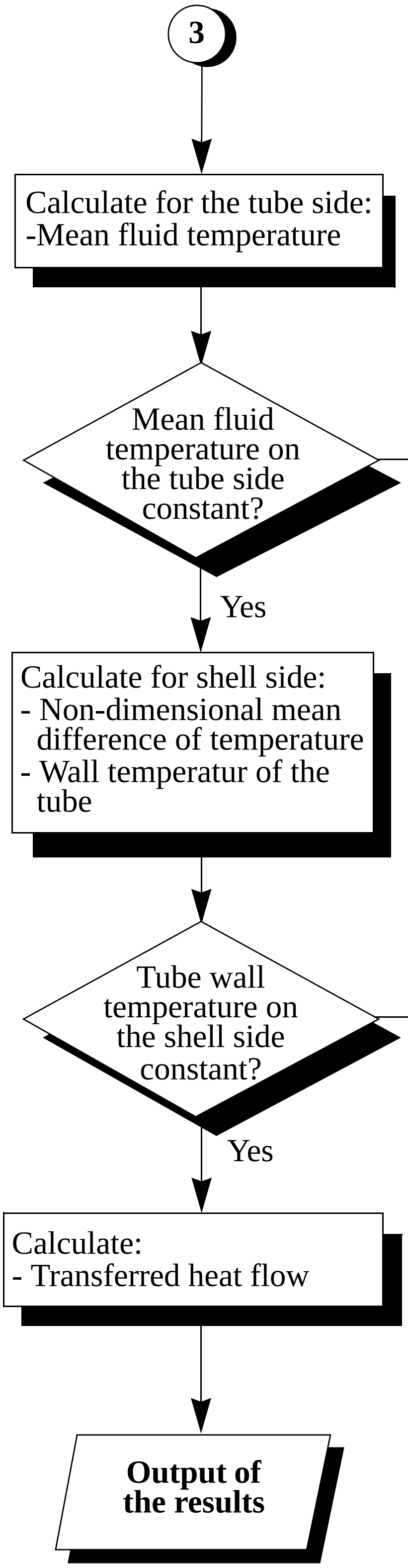


Abb. 8.15: Flow chart for the calculation of shell and tube bundle condensers

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 m^2 (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].

$$K_{PlateHE} = 1{,}371.4 \cdot A^{0.66} \,[DM, 1992], \text{ with } A \text{ in } [m^2] \,. \qquad \text{eq. (9.1)}$$

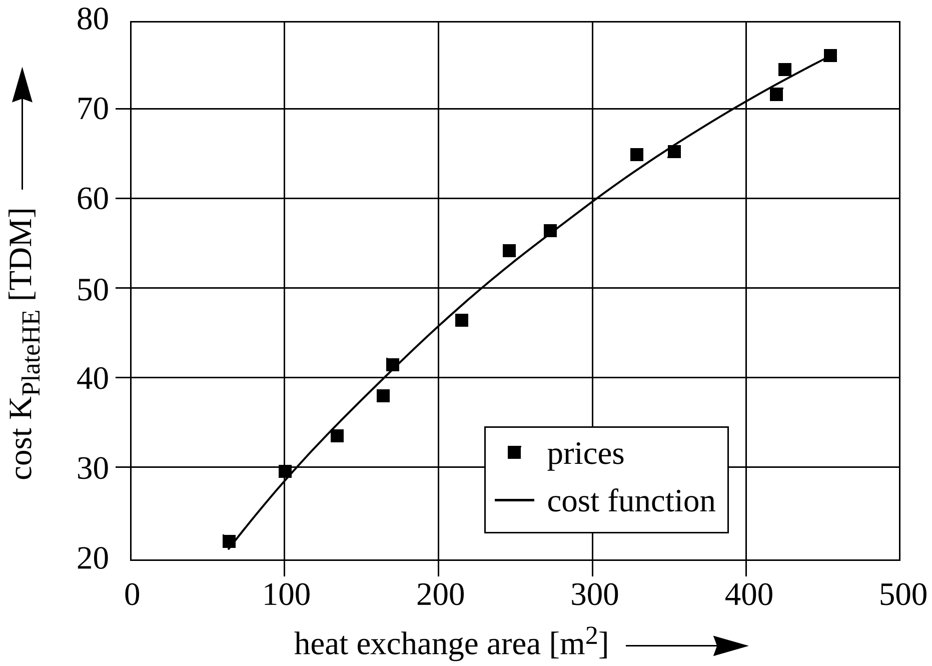


Fig. 9.1: Cost of plate heat exchangers (equation 9.1, 1992)

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 application 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.

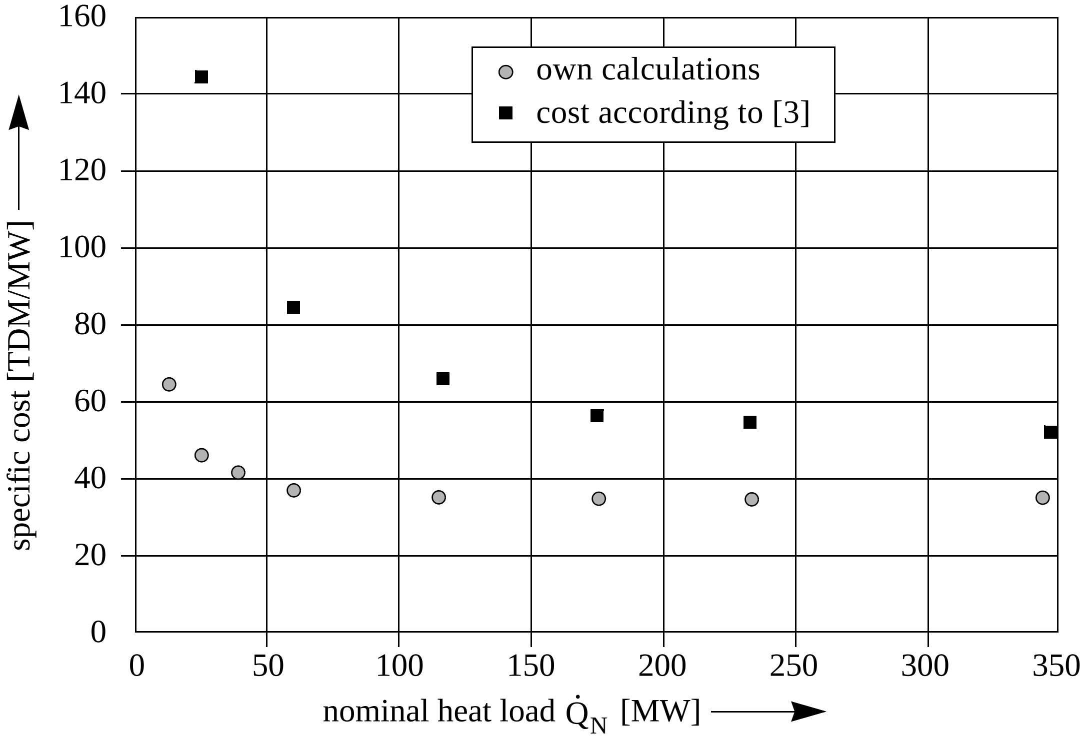


Fig. 9.2: Cost of entire transmission stations (1992)

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].

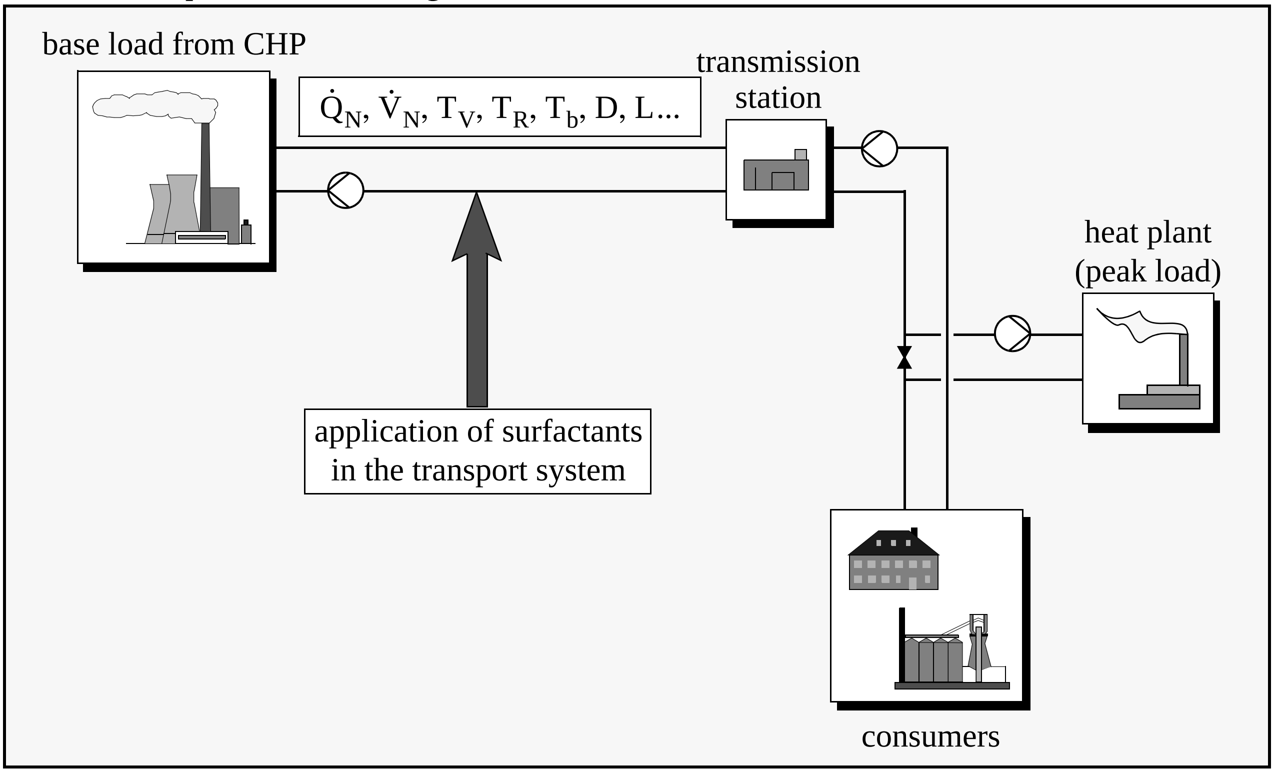


Fig. 9.3: Model for economics calculation [25]

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 fulfiled:

$$-a_n \cdot K(\dot{Q}, \Delta T_{max}, L) + r(\dot{Q}, \Delta T_{max}, L) > 0 \quad . \qquad \text{eq. (9.2)}$$

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:

$$a_n = \frac{p_k \cdot (1 + p_k)^n}{(1 + p_k)^n - 1} \quad . \qquad \text{eq. (9.3)}$$

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.

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

$$r = DR_m \cdot \frac{\Delta p}{\Delta L} \cdot \frac{\dot{V}_N}{\eta_P} \cdot s_{st} \cdot T_P \cdot L - \frac{\pi}{4} \cdot D^2 \cdot L \cdot f_{loss} \cdot k_s - \quad \text{eq. (9.4)}$$

$$i_{main} \cdot (K - I_s) - V_{ss} \cdot f_{loss} \cdot k_{treat} \; .$$

The symbols are defined as follows:

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

$$P_{max} \cdot T_P = \int_0^{8.760} P(t)dt \; , \quad \text{eq. (9.5)}$$

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

$$I_s = \frac{\pi}{4} \cdot D^2 \cdot L \cdot 1.5 \cdot k_s \text{, in [DM];} \quad \text{eq. (9.6)}$$

the factor 1.5 consideres adsorption losses of the first dosing,

V_{ss} = volume of subsystem [m^3] and
k_{treat} = specific operating cost for water treatment of the subsystem [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:

$$-a_n \cdot K(\dot{Q}, \Delta T_{max}, L) + r(\dot{Q}, \Delta T_{max}, L) = 0 \;, \; a_n \text{ [1/a].} \quad \text{eq. (9.7)}$$

With the following assumptions the „economic length“ can be calculated:

- $\Delta T_{max} = 70$ °C - ($T_{ret,min} = 60$ °C, $T_{supp,max} = 130$ °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:

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

- 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 in 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.

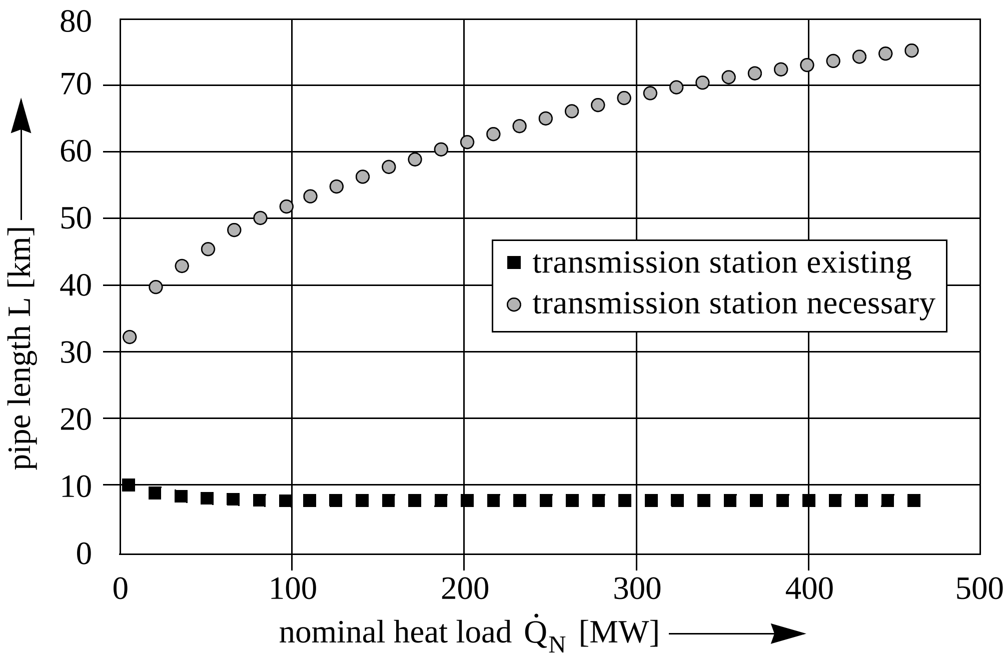


Fig. 9.4: „Economic length“ for the application „pumping cost reduction“ [29]

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.

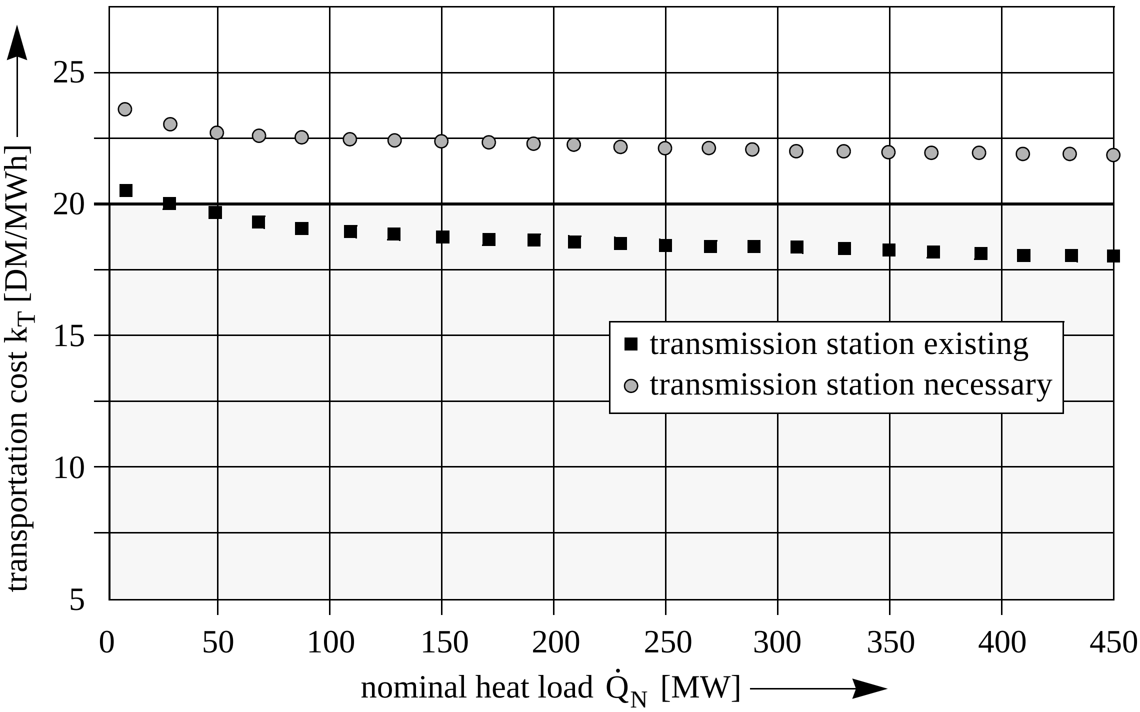


Fig. 9.5: Savings in transportation cost (reference value for water: 20 DM/MWh) [29]

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):

- DR_m - the average drag reduction (66 %),
- f_{ni} - share of not influenced pressure losses (0.2),
- 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
- ΔV - 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 occures under bad conditions. These are a low pressure gradient, a low drag reduction, low specific transportation cost and a low price for electric power.

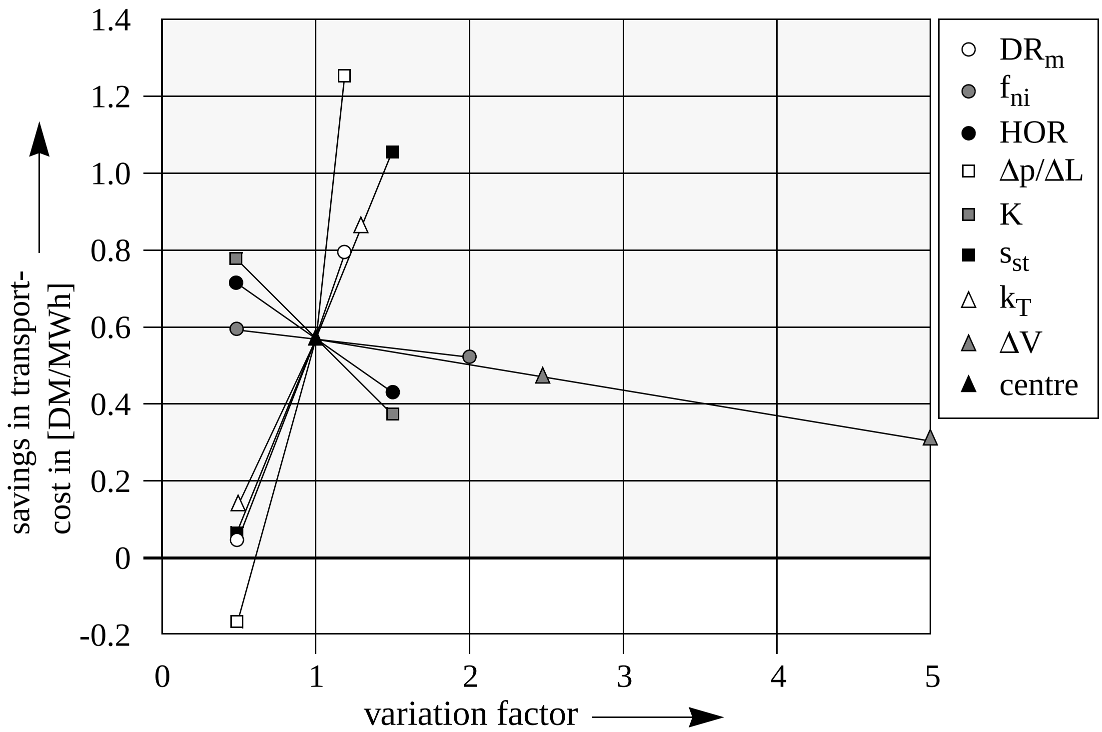


Fig. 9.6: Results of the sensitivity analysis for the application: savings in pumping cost - the installation of a complete transmission station is not necessary

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:

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

$$V_{ss} \cdot f_{loss} \cdot k_{treat}, \; r \; [DM/a]$$

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:

$$f_{incr} = \frac{\dot{V}_s}{\dot{V}_w} = \left(\frac{\xi_w}{\xi_s} \cdot \frac{1}{(1 - DR) \cdot (1 - f_{ni}) + f_{ni}}\right)^{1/3} . \quad \text{eq. (9.10)}$$

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:

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

ΔL_e = equivalent pipe length to consider additional flow resistances.

ξ_w, ξ_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 fulfiled. 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 prefered. The savings (losses) due to the installation of new pumps can be calculated with equation 9.12.

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

$$s_{st} \cdot T_P \cdot \dot{V} \cdot \frac{\Delta p}{\Delta L_0} \cdot \frac{L}{\eta_P} \cdot \left(\frac{\frac{\Delta p}{\Delta L_1}}{\frac{\Delta p}{\Delta L_0}} \cdot f_{incr} - 1\right),$$

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 exisiting“ and transmission station necessary“ are considered.

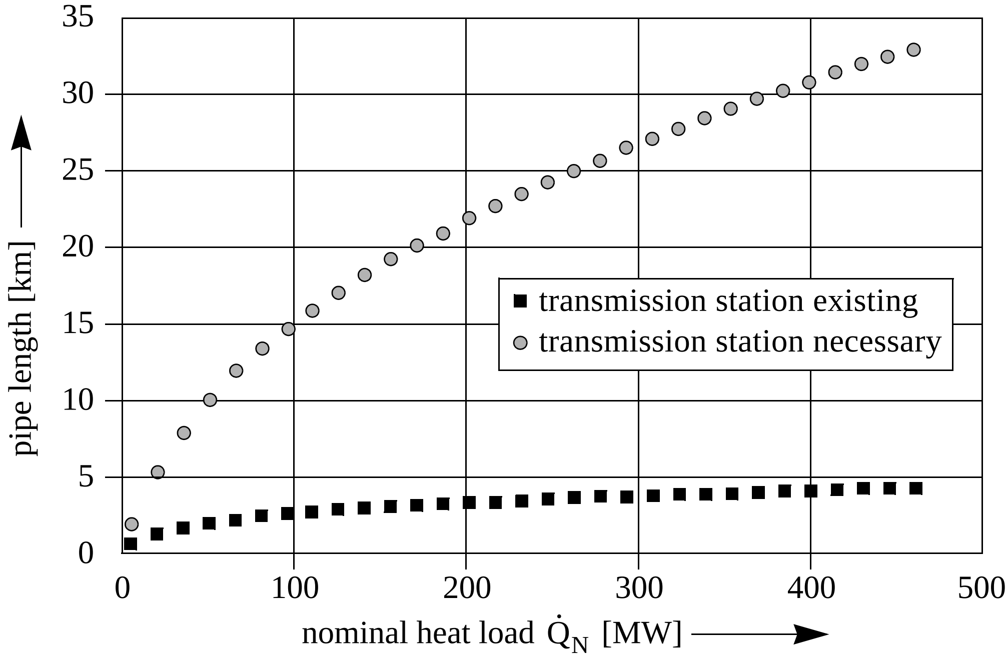


Fig. 9.7: „Economic length“ for the application „increase in capacity“ [29]

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**) fulfil the requirement for both cases - installation of new pumps and application of additives. Which possibility is more economic can not 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.

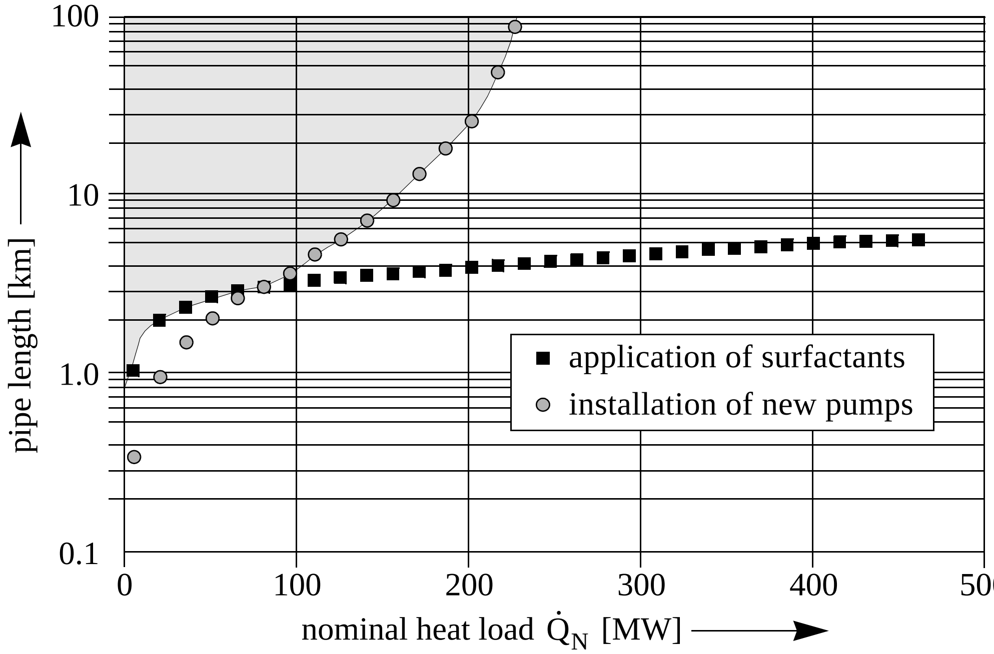


Fig. 9.8: Economic length for a pay off time of 20 years for an increase in heat capacity when using drag reducing additives (transmission station existing) compared with the installation of new pumps

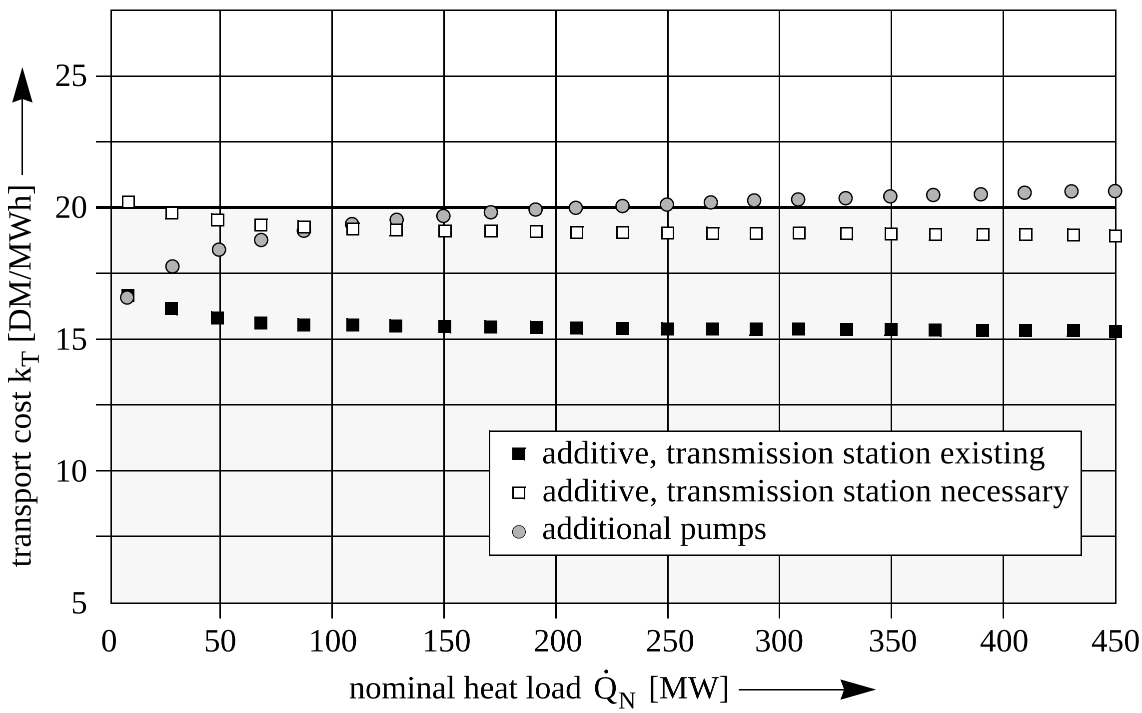


Fig. 9.9: Specific transportation cost for an application of drag reducing additives/ new pumps; reference value: 20 DM/MWh (for water as heat carrier)

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].

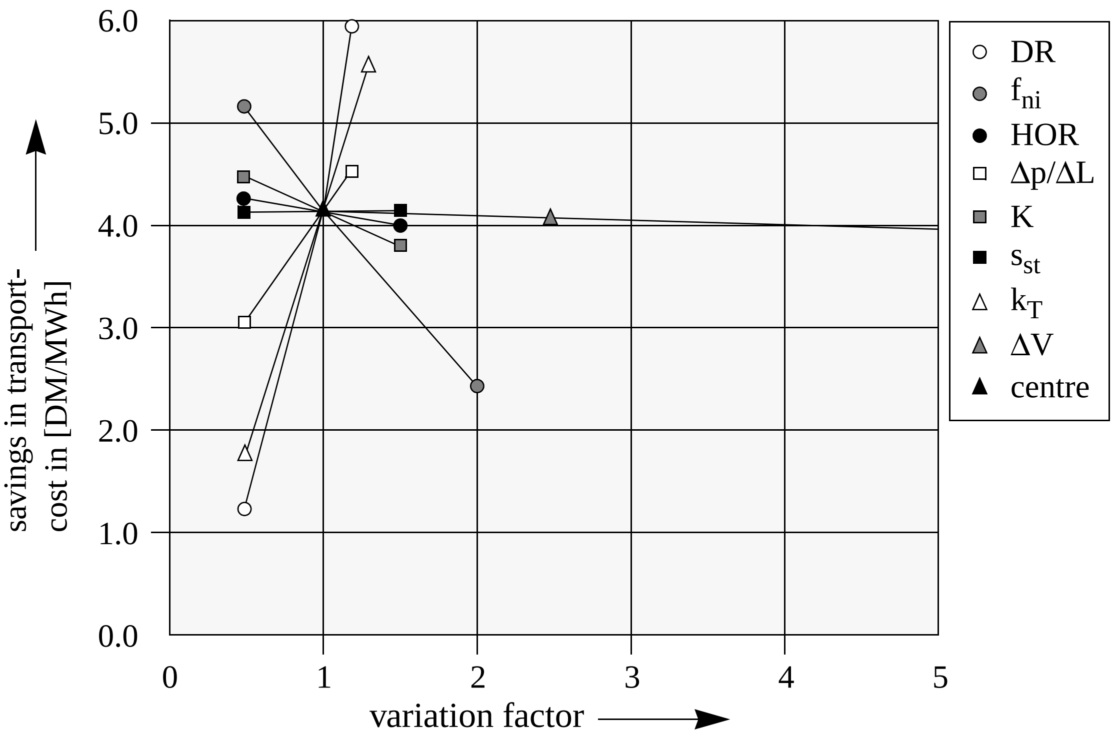


Fig. 9.10: Sensitivity analysis for the application: increase in heat capacity - the installation of a complete transmission station is not necessary

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_{ni} 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.

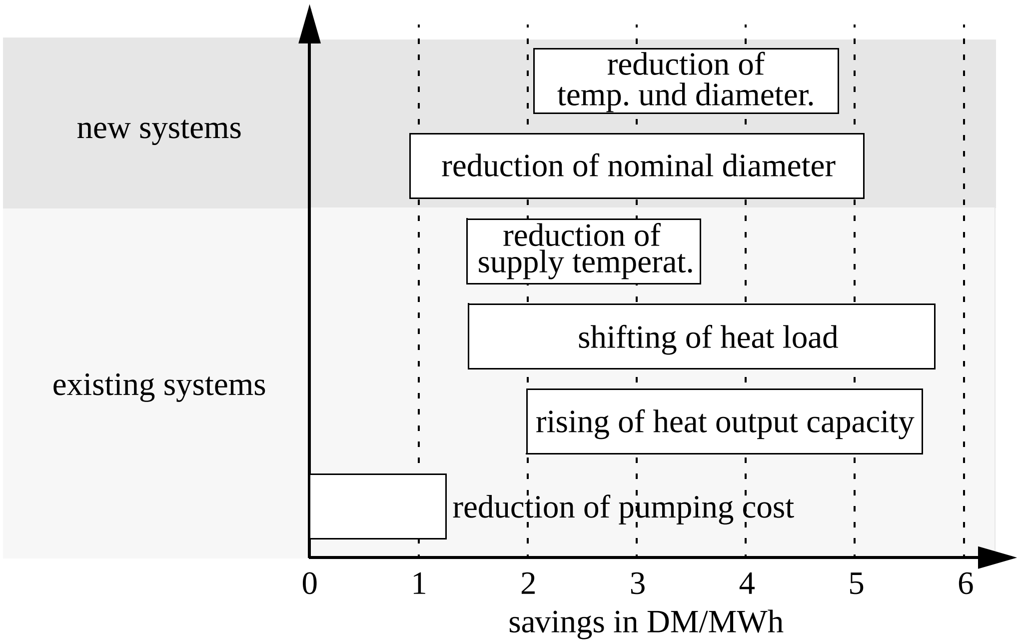


Fig. 9.11: Average potential in saving cost (the installation of a transmission station due to the application of FRAs is not necessary)

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 distict 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.