Project D

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

D1	**Investigations of Heat Transfer and Pressure Drop**

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Contents

1. Introduction

The application of drag reducing additives is a promising technology to increase the economic viability of district heating and cooling systems. With small amounts of cationic surfactants in district heating water the pressure drop in pipelines can be reduced significantly. Due to this effect the pump energy of existing district heating systems can be decreased or the transport capacity can be raised clearly. In addition the supply temperature of DH-systems can be decreased significantly when keeping the heat output constant. New networks can be designed with smaller pipe diameters or the maximum economic transmission length can be increased by applying drag reducing additives.

Parallel to the positive effect of drag reduction further effects occure due to the change in flow behaviour (viscoelastic instead of Newtonian). These are influences on heat exchangers, heat meters, pumps, filters, etc. The most important problem is the heat transfer reduction in heat exchangers. In shell and tube heat exchangers with straight long pipes a heat transfer reduction of about 95 % and therefore a significant reduction of heat output is possible. Up to now most of the influences have been investigated in laboratory and full scale tests.

Concerning the application in DH-systems, measures have to be carried out to compensate for all significantly negative effects. Considering the supply guarantee as most important requirement the influence on heat exchangers has to be payed particular attention. One of the most promising measures to improve the heat transmission properties of tube bundle heat exchangers is the installation of turbulence increasing obstacles inside the pipes.

In former investigations [15] wire nettings have been installed to improve the heat transfer coefficient inside straight pipes. The result of these investigations with drag reducing additives was a significant improvement. The values for water without turbulence increasing installations have almost been achieved. On the other hand the increase in heat transmission is accompanied by a strong increase in pressure drop.

Therefore new investigations have to be carried out to find better kinds of installations which show sufficient properties concerning heat transmission and pressure drop behaviour. This is the goal of the proposed study. Different obstacles have been tested to find a proper solution.

In this project, spiral springs of different gradients have been used to increase turbulence inside a double pipe heat exchanger. As influence quantities, the flow velocity, temperatures at the entrances of the heat exchanger as well as the concentrations of the cationic surfactant (HABON-G) have been varied.

2. State of the art

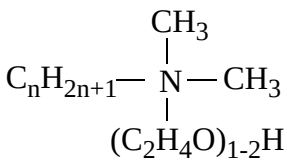
2.1 Behaviour of drag reducing surfactant solutions

Surfactants are low-molecular substances with a low chemical activity and a low solubility. Surfactants consist of a hydrophobic group - as a long-chain alkyl-part in general - and a hydrophilic part which consists of a molecule-part that can be ionized, of a polar or polarizable group or a molecule-part which can build hydrogen bonds.

For an application as drag reducing agents in district heating systems quaternary ammonia-compounds have been taken into consideration (see **figure 2.1**). Concerning the operating conditions of district heating systems the cationic substances HABON-G (n = 16) and DOBON-G (n = 22) together with the additional counter-ion salicylate (added as sodiumsalicylate) have been proved to be sufficient [1].

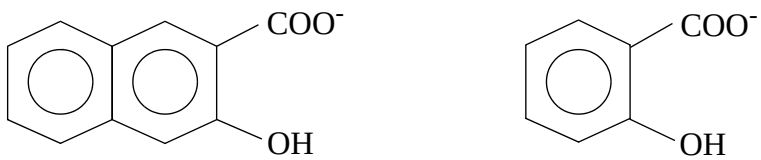
DOBON-G in addition with sodiumsalicylate shows the drag reducing effect in a temperature range of 40 °C < T < 130 °C and critical wall shear stresses below ca. 130 Pa. The achievable drag reduction is about 83 % compared to water [16].

The additive HABON-G shows a comparable drag reducing effect in a temperature-range of 30 °C < T < 105 °C below a wall shear stress of about 80 Pa [9]. Therefore, the conditions under which a drag reducing effect occures correspond quite well with the operating conditions of district heating systems.



n = 16: HABON-G
n = 22: DOBON-G

n-alkyldimethylpolyoxethylammonia-cation



3-hydroxy-2-naphthoate (counter-ion); salicylate (additional counter-ion)

Fig. 2.1: Structure of the considered additives [9]

In diluted solutions surfactant-monomeres form spherical micelles above the critical concentration CMC I (critical micelle concentration). In aqueous solutions the hydrophobic alkyl-chains form the core of the spherical micelles. The hydrophilic groups stick into the solution and build the cover of the sphere. If the concentration is increased drag reducing surfactants form rod like micelles above a second characteristic concentration - the CMC II [1].

The presence of rod like micelles is considered to be a necessary condition for the drag reducing effect. Above a certain shear stress (onset-point) the rod like micelles form a specific structure - the so-called shear induced state (SIS) [1]. The shear induced state is shown in **figure 2.2**. The

micelles align in direction of the flow and reduce the radial turbulences and therefore the pressure loss inside pipes, too [3]. Furthermore the shear induced state causes the viscoelastic properties, due to interlocking and bending of the structure.

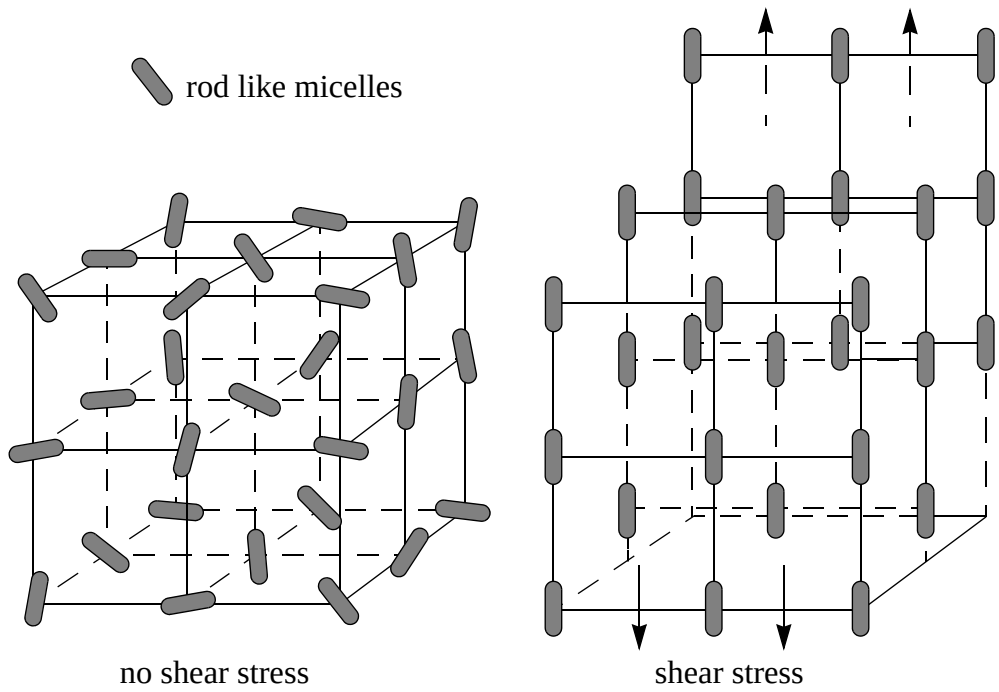


Fig. 2.2: Shear induced state (SIS) [11]

The drag characteristic of a surfactant solution inside a straight pipe compared with that of pure water is presented in **figure 2.3**.

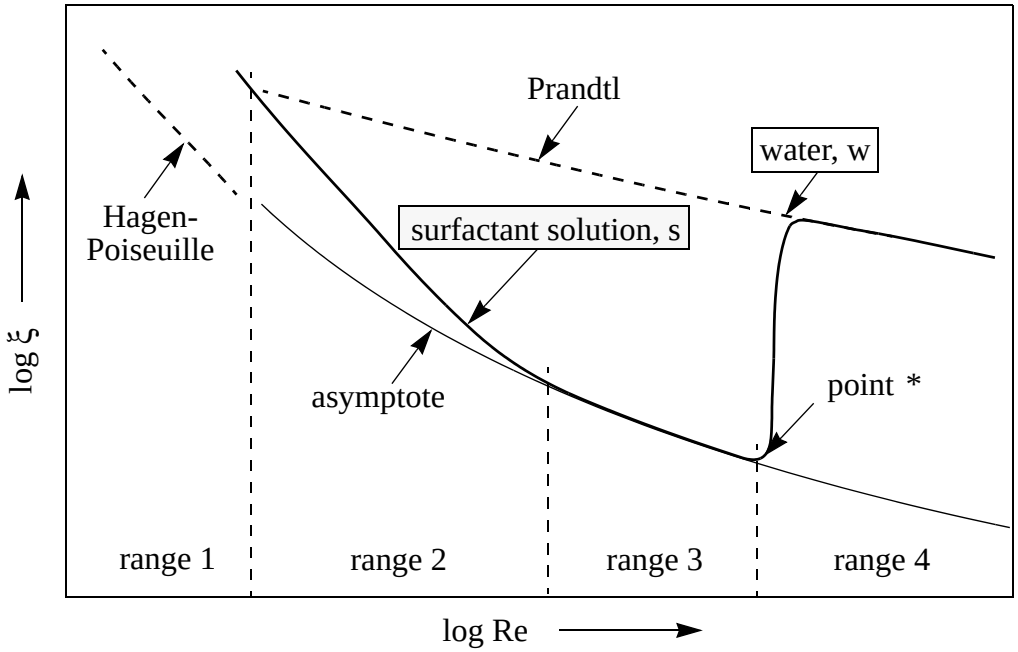


Fig. 2.3: Drag characteristic of a straight pipe for surfactant solution and water [1]

For small Reynolds numbers (range 1) the surfactant solution is moved parallel to higher drag coefficients. This results from the higher viscosity of the surfactant solution. With increasing flow velocity (range 2) the shear induced state is built up and a drag reduction occures. The drag coefficients for surfactant solution are below those for water - the difference is increasing with rising Reynolds number. In range 3 the maximum drag reduction appears. The characteristic runs along

an asymptote [1]. In range 4 the drag reducing effect disappears (point *) and the characteristic reaches the curve for water. The critical value * can be described independent from pipe diameter using the critical wall shear stress. Usually the critical wall shear stress is calculated for water (index w):

$$\tau^*_{W,\,w} \equiv \frac{\Delta p^*_w \cdot d_i}{4 \cdot L} \equiv \frac{\xi^*_w \cdot \rho \cdot w^2}{8}\,. \qquad \text{eq. (2.1)}$$

with $\tau^*_{W,\,w}$: critical wall shear stress for water,

d_i : inner diameter of the pipe,

L: length of the pipe and

Δp^*_w : pressure loss inside the pipe (length „L“) for water.

The drag reduction DR compared to water is characterized by the percental difference of pressure drop between water and surfactant solution:

$$DR \equiv \frac{\Delta p_w - \Delta p_s}{\Delta p_w} \cdot 100\% \equiv \frac{\xi_w - \xi_s}{\xi_w} \cdot 100\% \equiv \frac{\tau_{W,\,w} - \tau_{W,\,s}}{\tau_{W,\,w}} \cdot 100\%\,. \qquad \text{eq. (2.2)}$$

In addition to the desired drag reduction a decrease of heat transfer occures due to the reduction of radial turbulences. This can influence the heat transfer in heat exchangers considerable. In the worst case the reduction of heat transfer can lead to a significant reduction of heat output and therefore, can cause bottle-necks in heat supply.

A typical heat transfer characteristic (Nusselt versus Reynolds number) inside a straight pipe is shown in **figure 2.4**.

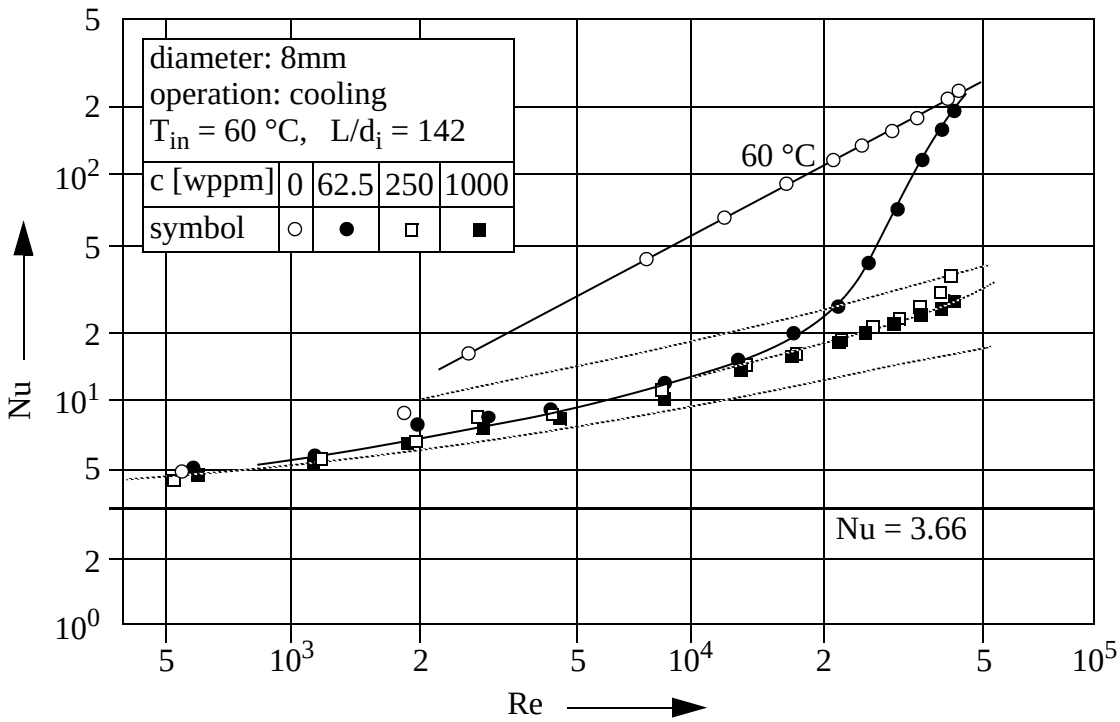


Fig. 2.4: Heat transfer characteristic of a surfactant solution (HABON) [15]

In the laminar region the Nusselt characteristic for surfactant solution is identic to that for water. With increasing Reynolds number the surfactant solution keeps the laminar heat transfer characteristic up to the critical Reynolds number.

Corresponding to the drag reduction (DR, equation 2.2) the heat transfer reduction HTR can be defined as the percentage decrease in heat transfer coefficient k (or Nusselt number) compared to that one of water (index w = water; s = surfactant) [4]:

$$\mathrm{HTR} \equiv \frac{k_w - k_s}{k_w} \cdot 100\,\% \equiv \frac{Nu_w - Nu_s}{Nu_w} \cdot 100\,\%. \qquad \text{eq. (2.3)}$$

Comparing the effects of heat transfer reduction and drag reduction the heat transfer reduction is always stronger than the drag reduction under otherwise equal conditions [4].

The heat transfer reduction HTR describes only the influence on one side of the heat exchangers. The overall heat transfer k_0 is determined by several thermal resistances. Therefore the percentage reduction of the overall heat transfer OHR has been defined:

$$\mathrm{OHR} \equiv \frac{k_{0w} - k_{0s}}{k_{0w}} \cdot 100\,\%. \qquad \text{eq. (2.4)}$$

This parameter allows more precise statements about the behaviour of overall heat transmission. OHR considers all thermal resistances of a heat exchanger. Due to the fact that only one resistance is reduced the overall heat transfer reduction OHR is always smaller than the heat transfer reduction HTR under the same conditions [3].

Finally the most important parameter which has to be considered in this connection is the heat load. Due to the influence on heat transfer the heat load is reduced compared to water. The heat output reduction HOR is defined as:

$$\mathrm{HOR} \equiv \frac{\dot{Q}_w - \dot{Q}_s}{\dot{Q}_w} \cdot 100\,\%. \qquad \text{eq. (2.5)}$$

An additional effect concerning the influence on heat exchangers is as a rule the increase of the effective temperature difference ΔT_m. The outlet temperature is increasing on the hot side ($T_{1,out}$) and decreasing on the cold side ($T_{2,out}$). It can be easily shown that as a result the reduction of the heat load $\dot{Q}_i$ is smaller than that of the overall heat transfer coefficient k_{0i}.

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

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

As a characterisation of the heat transfer behaviour the parameter ω has been introduced. This parameter is defined as the ratio between two heat transfer coefficients [11]:

$$\omega \equiv \frac{k_1}{k_2}. \qquad \text{eq. (2.6)}$$

The parameter ω describes the behaviour under the original conditions (operation with pure water). In case of ω >> 1 the overall heat transfer reduction is quite small but in the opposite case of ω << 1 the side on which surfactants should be used will be influenced very strong (see **figure 2.5**). In addition to ω the level of heat transfer reduction is a further decisive parameter that is determinig the overall heat transfer coefficient and also the heat output reduction.

The range of heat transfer reduction and the values of ω that appear in typical heat exchangers are shown in **figure 2.5**. This figure shows the influence of ω and HTR on the overall heat transfer reduction OHR for a typical apparatus as example with the following operating conditions [5][12][13][14]: (λ/s = 24,000 W/(m^{2}K), k_1 = 5,500 W/(m^{2}K), R_f = 0 (m^{2}K)/W).

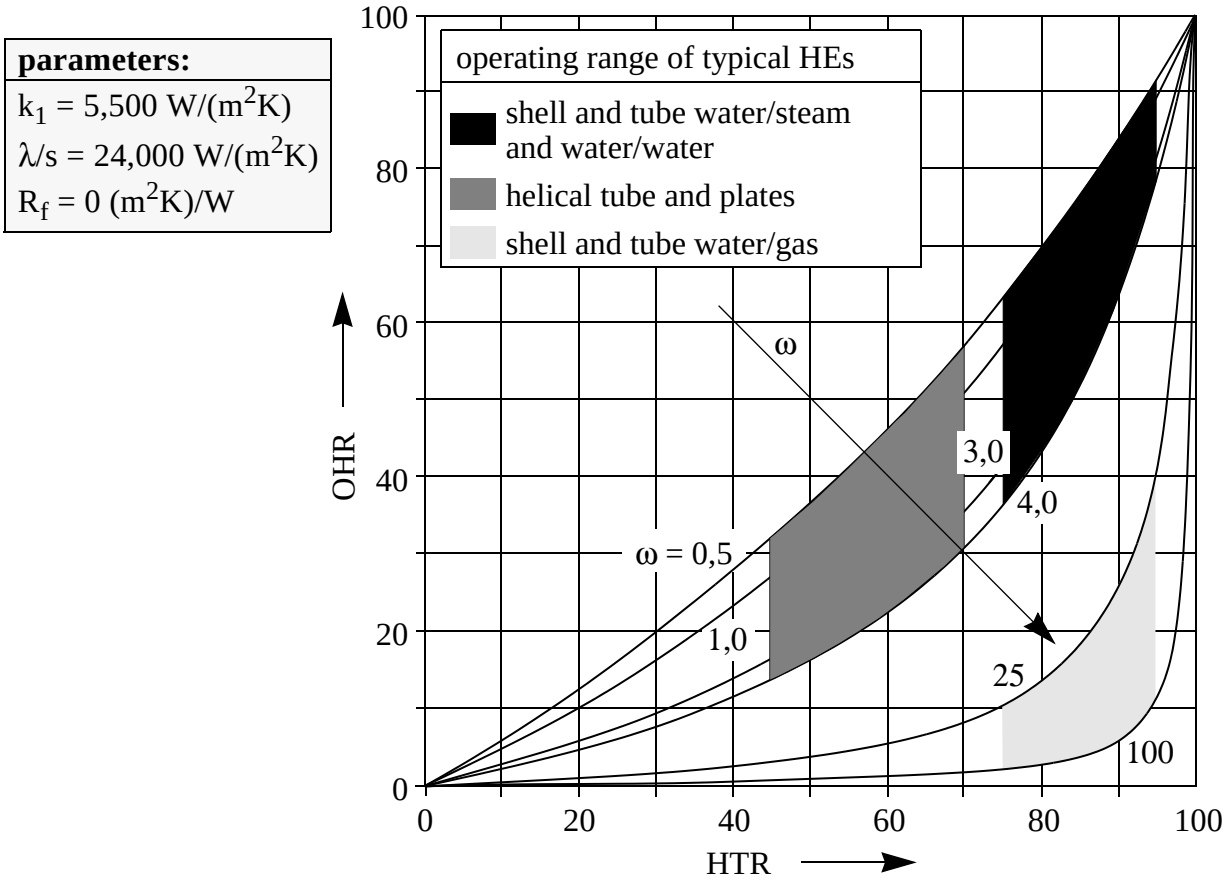


Fig. 2.5: OHR as a function of HTR and ω for typical DH-heat exchanger [11]

The strongest heat transfer reduction occurs in shell and tube heat exchangers. HTR values of 95 % in maximum can be caused due to the application of cationic surfactants. Especially in long pipes, strong influences are possible [15]. Therefore, it is necessary to develop measures to improve the heat transmission properties inside the pipes of these apparatus when planning the application of drag reducing additives. Otherwise problems concerning the supply guarantee can occure.

The reduction of heat transfer coefficients in helical tube and plate heat exchangers are smaller. 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. On the other hand the influence on shell and tube heat exchangers that are operating with gas and water is small due to a high value of ω. Values of OHR are only about 20 % if HTR is about 90 % and the values of ω are relatively high.

The Nusselt characteristic of straight pipes operating with surfactant solutions strongly depends on the tube length and the ratio L/d_i respectively [15]. With increasing tube length the HTR is increasing if the Reynolds number is kept constant. The reason for this behaviour is the percentage of the good heat transfer in the range of the entrance length of pipes ($(50 \div 100) \cdot d_i$). In long tubes ($L = (600 \div 1{,}000) \cdot d_i$) the heat transfer is nearly determined by a laminar flow. In this case the good heat exchange on the entrance length is only of minor importance for the total heat transfer inside the pipe. In comparision to this the HTR in very short pipes is only small due to the high percentage of turbulent flow and good heat transfer respectively.

Table 2.1 shows the experimental results of the heat transfer reduction of a straight pipe for different pipe lengths and values for L/d_i respectively. The significant influence of the pipe length is obvious.

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

Tab. 2.1: Influence of the pipe length on HTR

For the calculation of heat transfer in straight pipes with drag reducing additive solutions, a formula has been developed which exactly describes the technical relevant range [15]:

$$Nu = \sqrt[3]{3.66^3 + 2,191 \cdot Re \cdot Pr \cdot \frac{d_i}{L}} \cdot \left(\frac{L + L_e}{L}\right)^{0.249} \qquad \text{eq. (2.7)}$$

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

$$L_e = 0.075 \cdot d_i \cdot Re_W , \qquad \text{eq. (2.8)}$$

Re_W is calculated with the material properties of water at wall temperature.

The validity range of this formula can be gathered from [15].

In **table 2.2** two example calculations which have been carried out with the simulation program TenSim [2] are presented. The results of the characteristic values HTR, OHR and HOR for different Reynolds numbers are shown.

The values decrease in the above mentioned order: HTR > OHR > HOR.

Re	HTR	OHR	HOR
24,000	86 %	79 %	69 %
61,000	91 %	80 %	74 %

Tab. 2.2: Simulation results for HTR, OHR and HOR

Table 2.2 impressively shows the necessity of improving the heat transmission properties when using drag reducing additives in existing shell and tube heat exchangers. A reduction of heat output of about 70 % can not be tolerated.

Therefore this study to the improvement of heat transfer inside straight pipes by installing obstacles has been carried out.

In preliminary investigations from Weber at the University of Dortmund positive results of an artificial increase in turbulence to improve heat transfer have been received in principle. Wire nettings which have been installed inside pipes have been used as turbulence increasing obstacles. In **figure 2.6** some results of these investigations are presented.

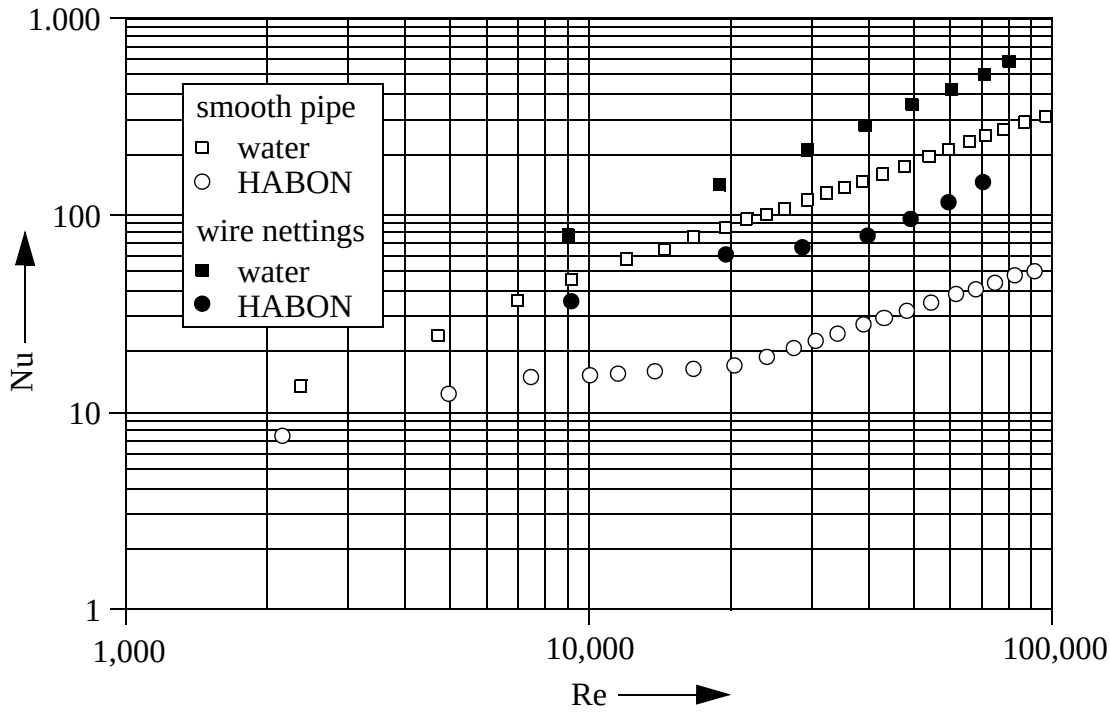


Fig. 2.6: Improvement of heat transfer of a 1000 wppm-HABON solution by wire nettings installed inside pipes

Compared with the smooth pipe without obstacles a significant increase can be achieved. The heat transmission reduction can be reduced from 81 - 84 % to 30 - 44 %, due to the installation of a wire netting.

On the other hand measurements of the pressure drop showed a tremendous increase of 200 to 300 % compared to the operation with water without obstacles. Therefore the increase of pressure drop is much stronger than the improvement in heat transfer.

Assuming a heat transfer reduction of 40 % (instead of 81 - 84 %) an example calculation has been carried out for a condenser. The following parameters - geometry and operating conditions - have been assumed:

Geometry:	Length of pipe:	12 m,
	inner diameter:	12.6 mm,
	number of pipes:	4,600.
Operation:	max. heat output:	87 MW,
	condensation temp.:	76.7 °C,
	max. flow velocity:	2.1 m/s.

For the origin state - smooth pipe/no obstacles/surfactant solution - the simulation of the condenser shows the following results:

HTR: 95 % and
HOR: 72 %.

Assuming the improvement of heat transfer due to the installation of obstacles of a HTR of 40 % (which is a pessimistic estimation), the resulting heat output reduction is:

HOR: 6.4 %.

The decrease in heat output can be reduced of about 90 % due to the increase in turbulence. The HOR of 6.4 % can be tolerated concerning the supply guarantee of real district heating systems.

The example impressively shows the potential of the installation of obstacles to improve the heat transfer (and output) behaviour of shell and tube heat exchangers.

3. Method

3.1 The test plant

The experiments have been carried out in the test plant presented in **figure 3.2**. Main parts of the plant are the two separated circles - the hot circle and the cold one. Both can be filled up with surfactant solutions. The drag reducing surfactants can be added over a special dosage device (B3 and P3). Both circles are equipped with the necessary measurement and control devices. Therefore the required entrance conditions (temperatures, flow rates) can be realized.

The centrifugal pump P1 is transporting the heat carrier from B1 through the heating circle. The water/surfactant solution is pumped through a flow heater with a maximum electric power of 190 kW. The required temperature is controlled by the control unit TIC 7. Behind the flow heater the heat carrier reaches a by-pass which is installed for a better temperature control. One part of the flow is transported back to B1 and the other part reaches the test heat exchanger HE. The flow rate of the hot circuit is controlled by the unit FRC 1. The realizable flow range is from 0.05 m^3/h to 20 m^3/h. The pressure of the pump is 5 bar.

The cold circle is identical to the hot one in principle. Instead of a flow heater two series connected plate heat exchangers are installed to cool the heated solution. Further informations can be gathered from **figure 3.2**.

If stationary conditions are reached, the four temperatures, the two flow rates and the two pressure differences are measured and registered.

As test heat exchanger a special developed double pipe heat exchanger has been installed. The apparatus has a length of 4,000 mm, an inner diameter d_i of 17.3 mm and an outer diameter d_a of 28.5 mm (see **figure 3.1)**.

The investigated obstacles are shown in **figure 3.3**. The spiral springs have an outer diameter of 17.5 mm. Their diameter is bigger than the inner diameter of the inner pipe of the heat exchanger to reach a certain tension between the spring and the wall of the pipe.

Due to this tension the spiral springs stay on their location at the inner pipe wall even when high flow rates have been adjusted.

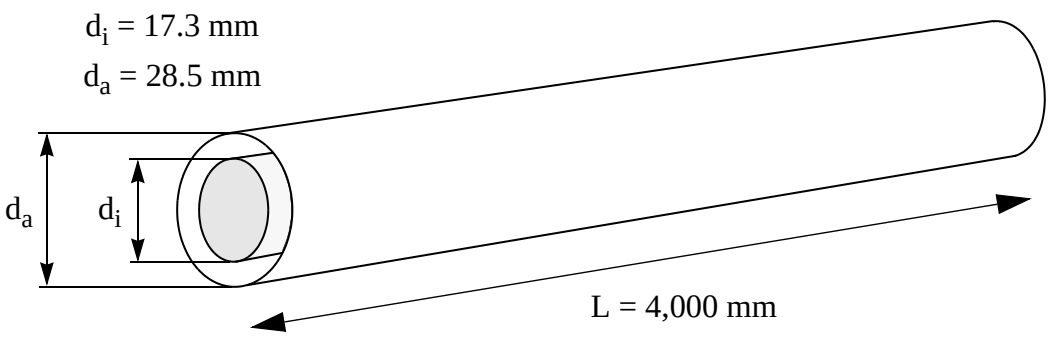


Fig. 3.1: Investigated double pipe heat exchanger

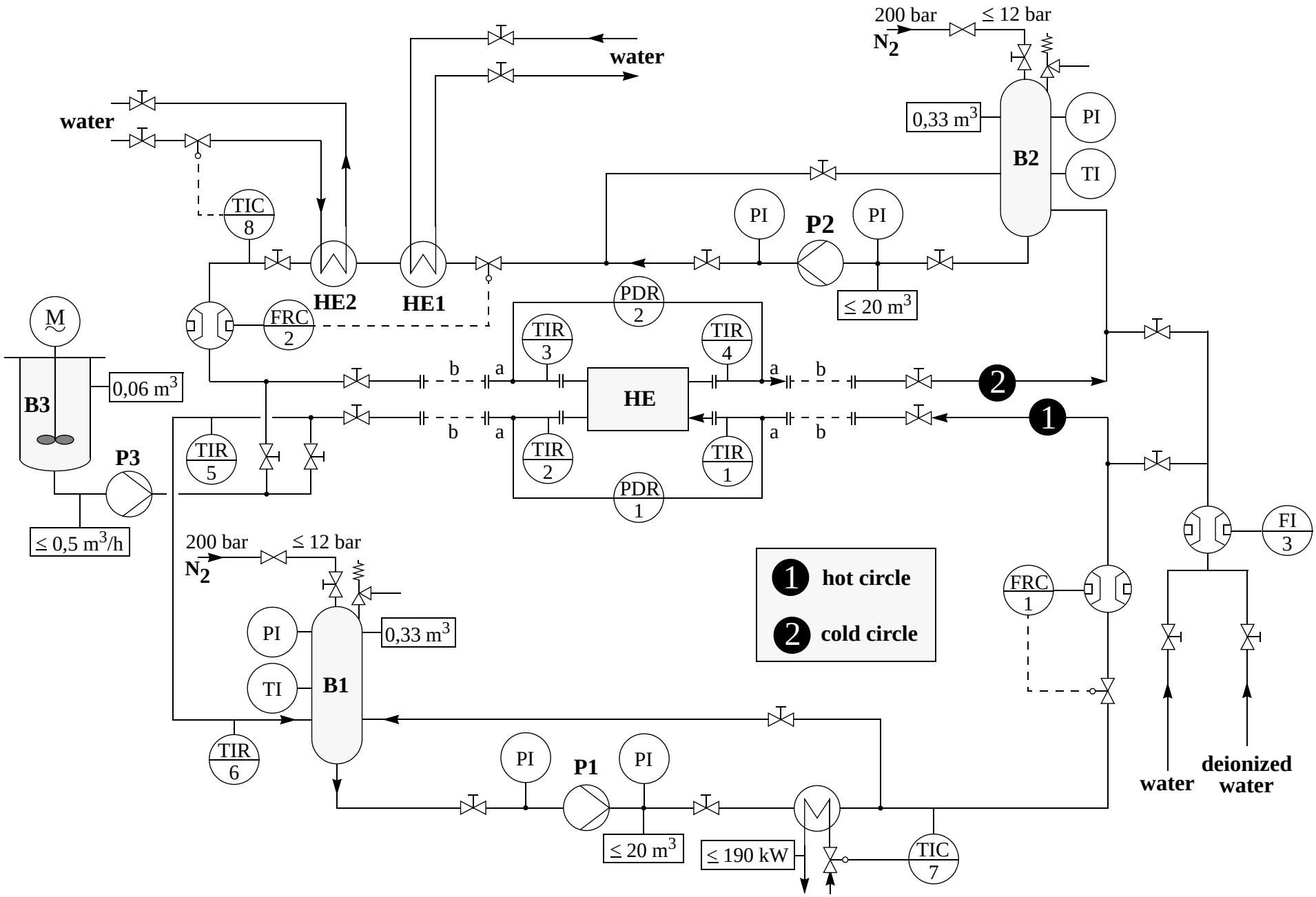


Fig. 3.2: The test plant

The thickness of the wire has been 1 mm for any different kind of spiral spring.

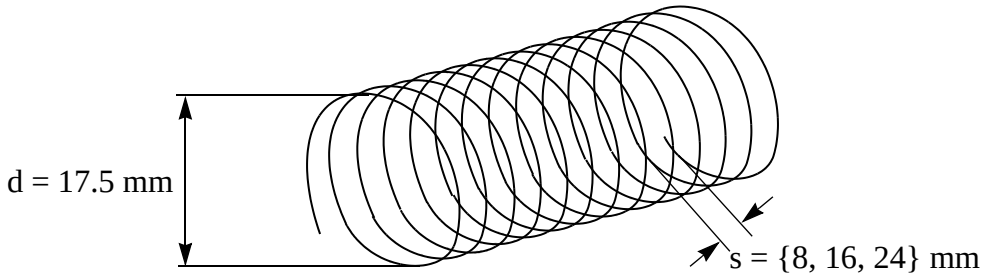


Fig. 3.3: Spiral springs

During the investigations, the pitch s has been varied. Springs with pitches of 8 mm, 16 mm and 24 mm have been installed and investigated.

3.2 Program

For the investigations the cationic surfactant HABON-G has been applied (see **figure 2.1)**. The varied parameters have been:

- Temperature,
- concentration,
- flow velocity and
- pitch of the spiral springs.

Due to the fact that the results of the laboratory tests shall be used for the installation of spiral springs inside a real condenser the operating conditions (temperatures and flow velocities) have been adjusted to the operating conditions of the full scale apparatus. To reach a nearly constant temperature on the hot side the flow rate of the hot circle has always been relatively high. The maximum flow rate was limited by the essential required accuracy of the thermometers (temperature difference). The different entrance temperatures are presented in table 3.1.

Number	entrance temperature cold circle (surfactant)	entrance temperature hot circle (water)
1	55 °C	77 °C
2	70 °C	94 °C
3	55 °C	84 °C
4	82 °C	90 °C
5	30 °C	60 °C
6	60 °C	90 °C

Tab. 3.1: Entrance temperatures of hot and cold circle

After having carried out the reference measurements with water, the concentrations:

- 125 wppm (weight parts per million),
- 250 wppm and
- 500 wppm

have been adjusted.

The flow rate of the cold medium (surfactant) has been varied between 0.4 m^3/h and 2.0 m^3/h. 18 different flow velocities between the minimum and maximum values have been investigated to get sufficient results for heat transfer and pressure drop.

4. Results

For the presentation of the results the non-dimensional quantities Re, ξ and Nu are preferred. Drag characteristics (ξ/Re-diagrams) and Nusselt characteristics (Nu/Re-diagrams) are used to describe the hydrodynamic and the heat transfer behaviour of water and surfactant solutions inside straight pipes with and without spiral springs.

The non-dimensional quantities are defined as follows:

$$Re \equiv \frac{d \cdot w \cdot \rho}{\eta}, \qquad \text{eq. (4.1)}$$

$$\xi \equiv \Delta p \cdot \frac{\rho}{2} \cdot w^2 \cdot \frac{L}{d} \text{ and} \qquad \text{eq. (4.2)}$$

$$Nu \equiv \frac{k \cdot d}{\lambda}. \qquad \text{eq. (4.3)}$$

In these definitions the physical properties of pure water are used. This is the usual procedure to describe the behaviour of drag reducing solutions [1][4][15].

Due to the installation of the obstacles inside the pipes the hydraulic diameter is decreasing. Calculations of the hydraulic diameters and comparisons between these values and the normal diameter d_i (see **figure 3.1**) show that the influence of the decrease in hydraulic diameter is neglectable. The inaccuracy when simply using the inner diameter is 1.1 % in maximum (for the smallest pitch s = 8 mm).

4.1 Heat transfer

4.1.1 Results without spiral springs

Before installing the spiral springs reference measurements without obstacles have been carried out to compare the results to those of former investigations. In **figure 4.1** the Nusselt characteristics for water and surfactant solutions of different concentrations and entrance temperatures are shown.

As known from former measurements the Nusselt numbers for drag reducing surfactants are significantly below those for pure water if the Reynolds number is below a critical value (or better: if the wall shear stress is below the critical wall shear stress). Behind the critical wall shear stress the characteristics rise and reach the values of water. The critical wall shear stress is dependent on temperature and concentration.

With increasing concentration the point on which the characteristic rises to that one for water is moving to higher Reynolds numbers. The increase in temperature leads to the same effect because both temperatures are below the maximum value $T_{opt} = T(\tau_{opt})$ with $\tau_{opt} = \tau^*_{W,\,max}$ (see [4]).

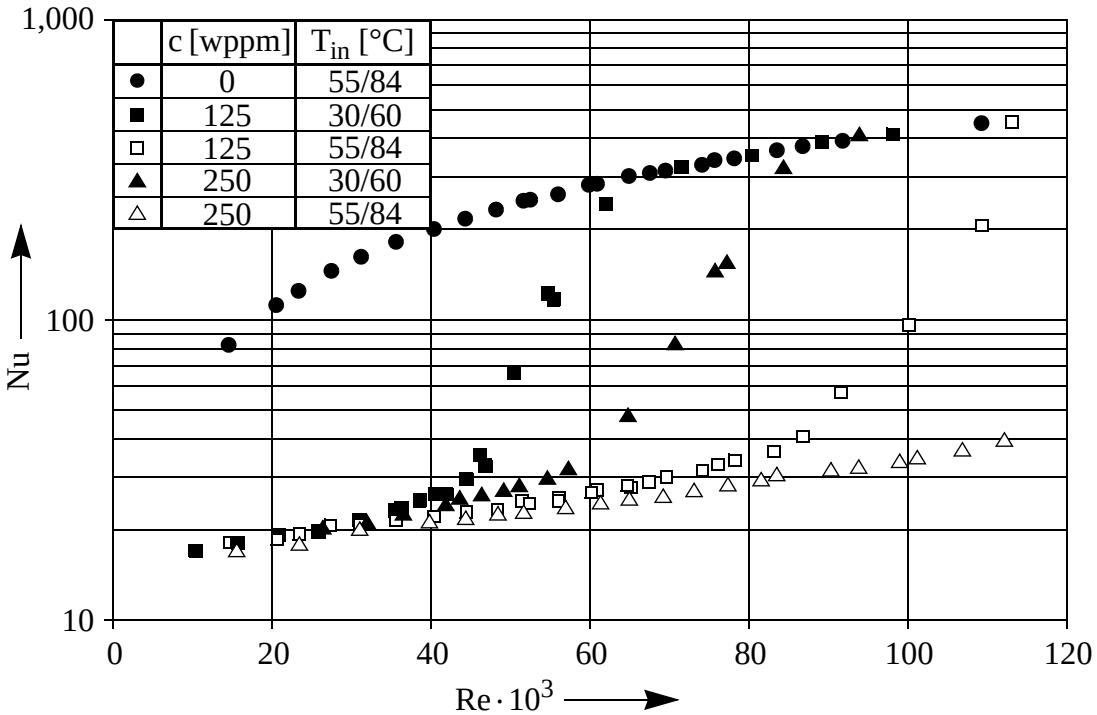


Fig. 4.1: Measurements of heat transfer with water and surfactant solutions of different concentration and entrance temperatures without obstacles

The qualitative behaviour is identical to former results as well as the scale of the values. The heat transfer reduction (HTR) in accordance with equation (2.3) reaches values above 90 %. For Reynolds numbers below the critical value the heat transfer coefficients for surfactant solutions can be calculated with the equation of Weber [15] (equation (2.7)) with sufficient precision.

4.1.2 Results with spiral springs

4.1.2.1 Results with water

In **figure 4.2** the results of the heat transfer inside the pipe with a spiral spring for water are shown. The pitch of the spring was 8 mm and the entrance temperature has been 55 °C.

It can be seen impressively that the heat transfer is increasing enormously. The values of the Nusselt number with obstacles are about twice the corresponding values without the spiral spring. The reason for that is the increase in turbulence due to the disturbance of the flow near the wall behind the wire of the spring. For the following description of the heat transfer behaviour the heat transfer improvement (HTI) is defined:

$$HTI \equiv \frac{k_S - k_w}{k_w} \cdot 100\ \%\ \text{or}\ HTI \equiv \frac{Nu_S - Nu_w}{Nu_w} \cdot 100\ \%, \qquad \text{eq. (4.4)}$$

with: Index S: „Spring“ and
index w: „water (without spiral springs)“.

The definition HTI uses the „origin state“ of the heat exchanger (water, without obstacles) as reference value to determine the heat transfer improvement. The values k_S or Nu_S can stand either for water or for surfactant solution. Thus, for the application of drag reducing additives, the value of HTI has to be above 0 to reach the value of water without springs.

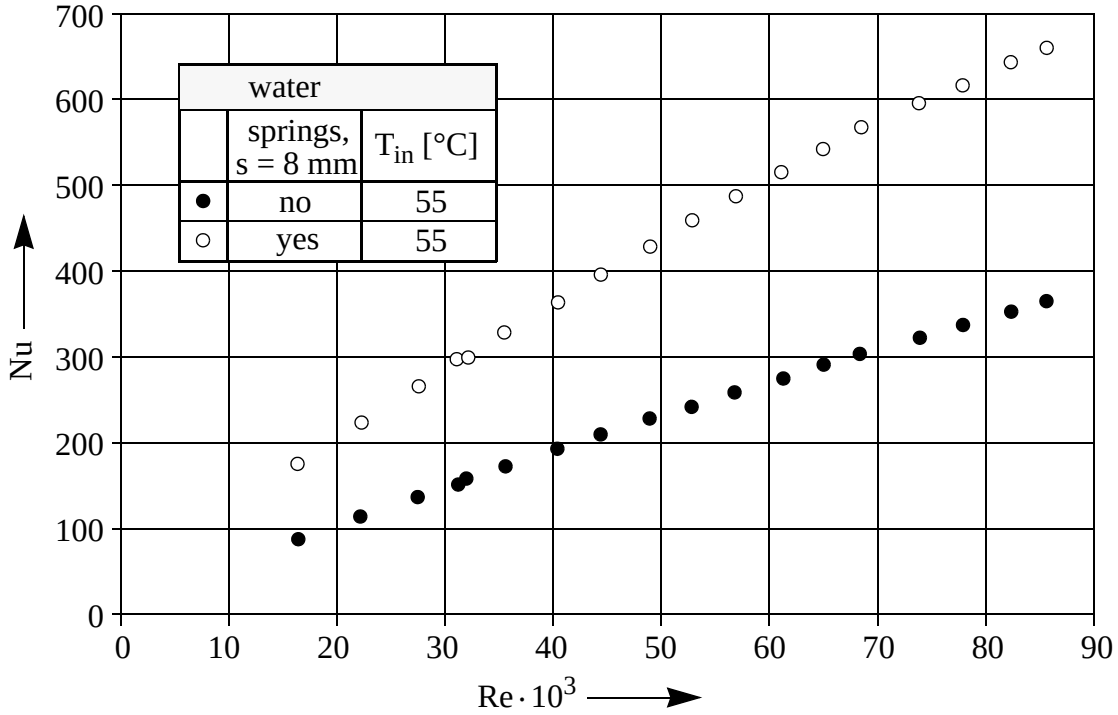


Fig. 4.2: Nusselt characteristic for water with spiral springs (s = 8 mm, T_{in} = 55 °C)

Considering the results in **figure 4.2**, the heat transfer improvement reaches values between 85 % and 100 %. Similar results can be received, if the temperature has been varied.

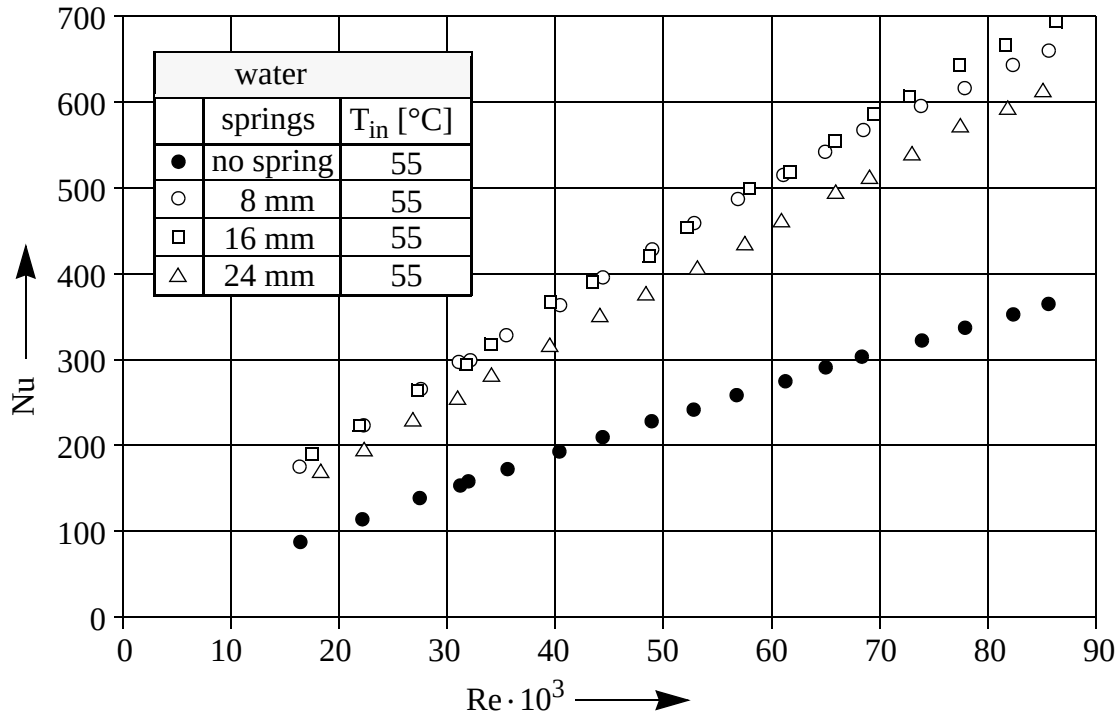


Fig. 4.3: Nusselt characteristic for water with springs of different pitches (T_{in} = 55 °C)

The influence of the pitch of the spiral spring is shown in **figure 4.3**. The values of the 8 mm and the 16 mm spring are almost identical. Only at high Reynolds numbers the characteristic of the 8 mm spring is bending to lower values.

The characteristic of the 24 mm spring runs significantly below the two other curves. The medium heat transfer improvement of this spring is above 70 % in contrary to the 8 mm and the 16 mm spring which show an avarage improvement of about 90 %.

The explanation of this behaviour could be an „effective distance“ on which an optimum heat transfer improvement occures due to the increase in turbulence. In contrary to these „effective distance“ there are areas on which those turbulences and therefore, the heat transfer improvement are reduced.

The „effective distance“ begins behind the spring (after a specific length) and ends after a certain distance. Directly behind the wire the turbulences have to develop to improve the heat transfer conditions near the wall. After a certain length, the turbulences are becoming calm. Both the beginning and the end of the effective distance - the range of maximum heat transfer improvement - are dependent on the Reynolds number. In **figure 4.4** and **4.5** a qualitative description of the possible behaviour of the turbulences is given.

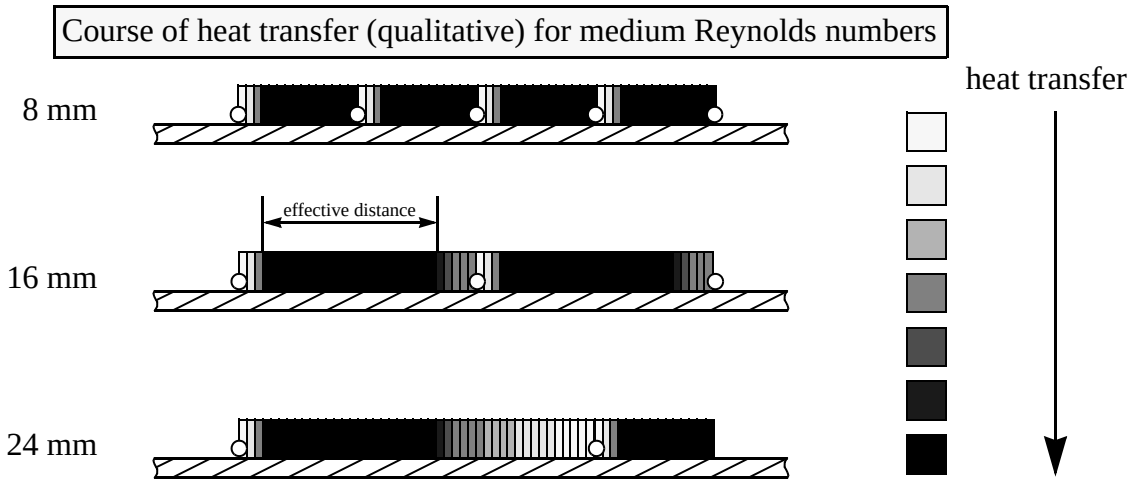


Fig. 4.4: Qualitative description of turbulence imrpovement at medium Reynolds numbers

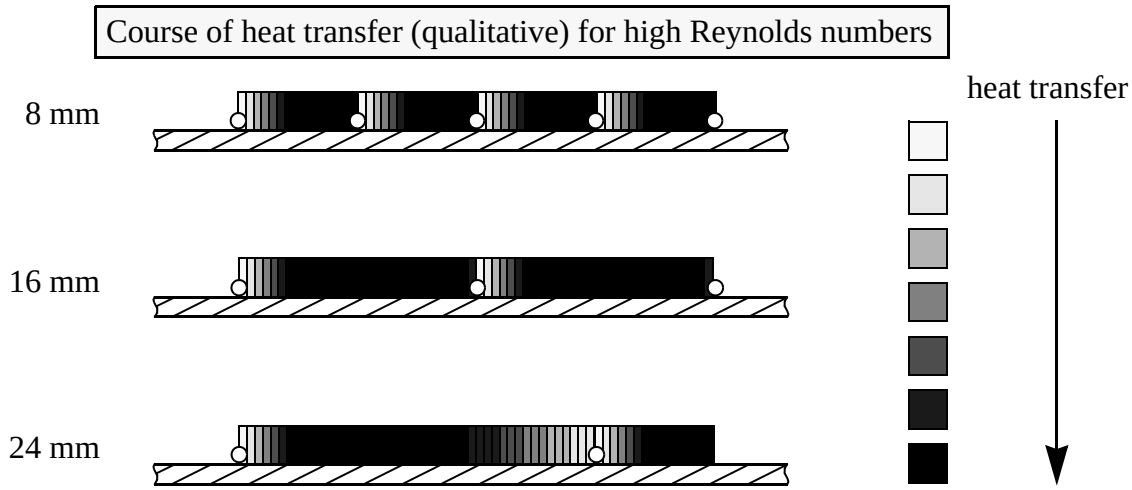


Fig. 4.5: Qualitative description of turbulence imrpovement at higher Reynolds numbers

At medium Reynolds numbers the characteristic for the 8 mm and the 16 mm springs are nearly identical (see **figure 4.3**). On the one hand the effective distance of the 16 mm spring is - compared to the 8 mm spring - shorter than its pitch. On the other hand in case of the 8 mm spring twice as much windings are existing. Behind these windings the turbulences has to be developed first before improving the heat transfer near the wall. These effects compensate for each other. In case of the 24 mm spring the pitch is significantly larger than the effective distance (The disturbance of the flow is not strong enough to increase the turbulences between two windings of the springs). Therefore the heat transfer improvement is much smaller compared to the other two cases.

With increasing Reynolds number the distance which is necessary to form the turbulences is increasing, too (see **figure 4.5**). This is a disadvantage for the 8 mm spring because the effective distance is larger than the pitch anyway. Thus, the range in which the turbulence is increased in maximum, is smaller compared to lower Reynolds numbers. The linear course turns into an underproportional. In contrast to the 8 mm spring the effective distance for the 16 mm spring can still be increased. Due to these reverse effects - increase of the area of flow development behind the wire and increase of effective length - the linear course of its characteristic continues. The same effect occures with the 24 mm spring, only on a lower level.

The observed heat transfer behaviour suggests that in case of applying pure water, no general optimum design concerning the pitch of the springs can be reached. The optimum value is dependent on Reynolds number. Therefore a spring that is optimal over the complete range of flow velocity can not be found.

In this theory the increase in heat exchange area due to the spring is neglected. This probably will improve the heat transfer behaviour. Considering the marginal area (or better: volume) of the spring and the fact that the spring only touches the wall with an infinitesimal part of its area, this effect can be neglected.

4.1.2.2 Results with surfactant solutions

Figure 4.6 shows the Nusselt characteristics of Habon-G solutions of different concentrations and for water with an entrance temperature of 55 °C. These measurements have been carried out with the 8 mm spring. Considering the concentrations 125 and 250 wppm the typical behaviour of drag reducing additives can be observed. At low Reynolds numbers a heat transfer reduction (compared to water with obstacles) can be seen. In this range the characteristic is almost identical to that one for water without obstacles. With increasing flow velocity the Nusselt numbers are rising and reach - at a critical Reynolds number - the values for water with obstacles. This critical Reynolds number is increasing with the concentration.

At high Reynolds numbers (above 70,000) the values for 125 and 250 wppm are slightly above those for water with obstacles. This effect may be result from the micellar structures in the solution which influence the turbulence near the wall.

Considering the surfactant solution of 500 wppm the behaviour is completely different from the other solutions. The characteristic runs ca. 15 % to 25 % below the curve for pure water without obstacles. A critical value on which the characteristic rises to the values for water with obstacles can not be seen.

The stability of the shear induced state is increasing with the concentration [8]. It is possible that the critical wall shear stress is not be reached. Considering the last measuring point which shows a tendency to increase to higher Nusselt numbers the critical wall shear stress can be expected slightly behind this point.

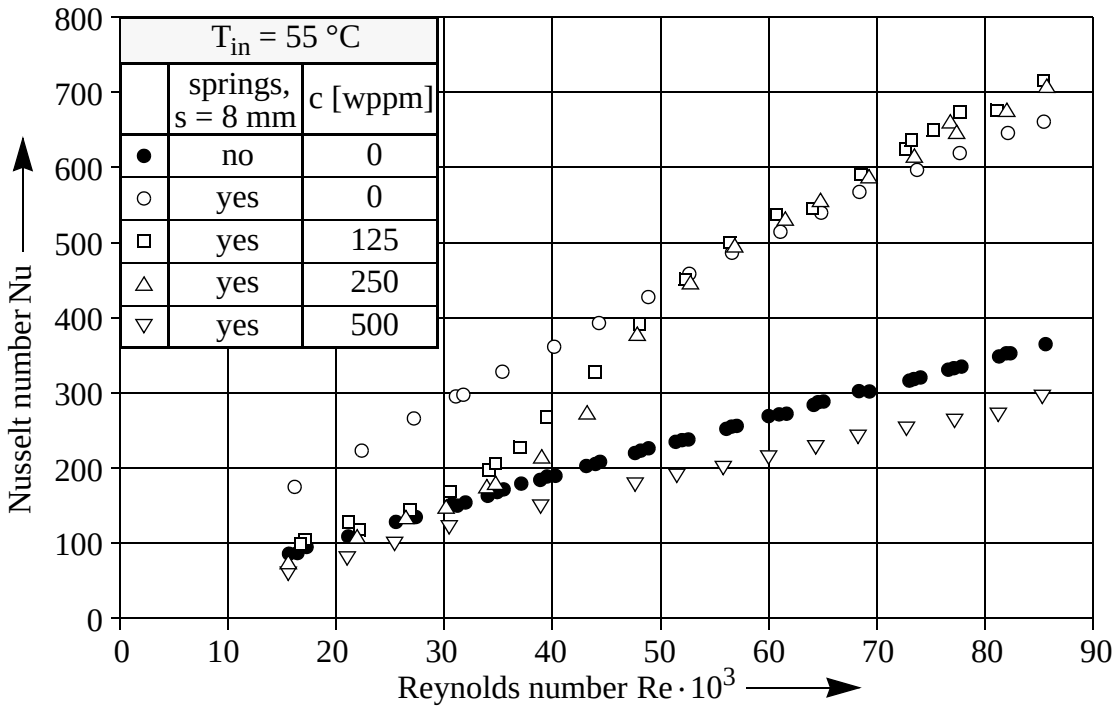


Fig. 4.6: Nusselt characteristic for water and drag reducing surfactant solutions of different concentrations with spiral springs (s = 8 mm)

An other explanation could be the flexibility of the rod like micelles. Due to the fact that the rod like micelles grow with increasing concentration [3] their flexibility - and thus their ability to produce turbulences near the wall - will decrease.

To find out whether a critical wall shear stress will be reached measurements at higher flow velocities have to be carried out. With the above described test rig, those measurements can not be carried out without modifications.

Figure 4.7 shows the dependence of the Nusselt characteristic for different temperatures. The measurements have been carried out with the 24 mm spring for a 125 wppm solution. With increasing temperature the characterisitics are moved to lower Nusselt numbers - or better: are moved to higher Reynolds numbers. This effect is mostly caused due to the physical properties which are used to calculate the Reynolds number. With increasing temperature, the viscosity is decreasing much stronger than the density of water. Thus the Reynolds numbers for identical flow velocities are increasing with the temperature.

In **figure 4.7** the typical course of the characteristics (see also figure 4.6) can be seen. At a certain Reynolds number the characteristic is bending to higher values and reaches the corresponding curve of water (this can only be seen for the entrance temperature of 70 °C - for the reason of clarity, the water curves for the other temperatures have not been shown).

The characteristic for the temperature of 82 °C runs parallel to the water curve. Compared to the corresponding water curve for 82 °C, the characteristics are almost identical. In this case the critical wall shear stress is reached and no effect due to the drag reducing additives occures.

Comparing **figure 4.7** with **figure 4.6** it is obvious that the critical Reynolds number is moved to higher values, if the pitch of the spiral springs is increased. The comparison of the different pitches is given in **figure 4.8**.

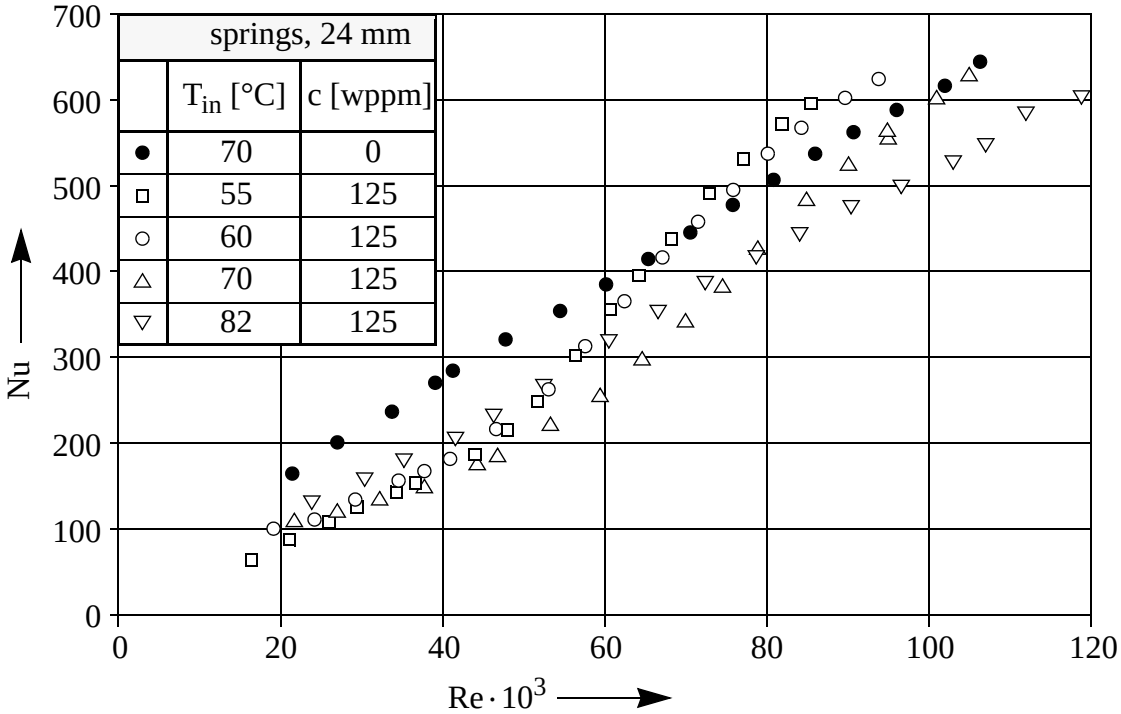


Fig. 4.7: Nusselt characteristic for water and drag reducing surfactant solutions of different temperatures with spiral springs (c = 125 wppm, s = 24 mm)

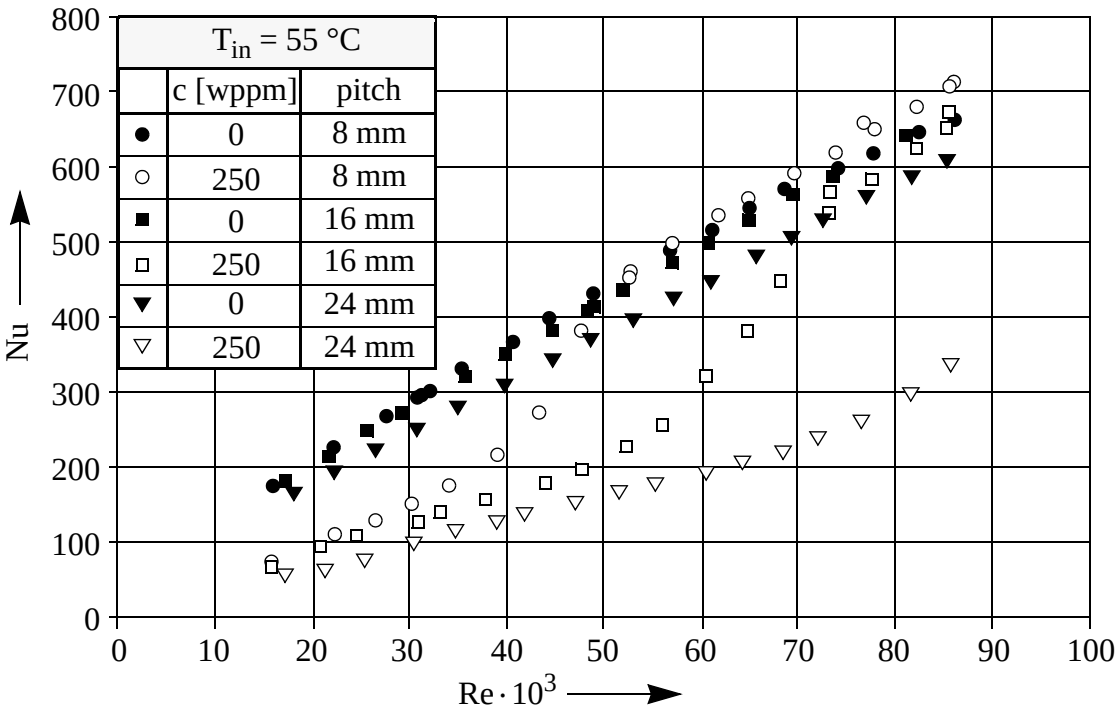


Fig. 4.8: Nusselt characteristic for water and drag reducing surfactant solutions for spiral springs of different pitches (c = 250 wppm, T_{in} = 55 °C)

In **figure 4.8** the dependence of heat transfer of cationic surfactant solutions (Habon-G, 250 wppm) on the pitch of the installed spiral springs is shown. As expected the heat transfer improvement is decreasing with increasing pitch of the obstacles. Furthermore the critical Reynolds number is increasing with increasing pitch. Thus for the 24 mm spring the values for water are not reached.

The explanation for this behaviour is that the ability to build and keep up the shear induced state is increasing with the pitch. Due to the higher turbulences and therefore the higher wall shear stress in case of applying springs with smaller pitch, the critical wall shear stress will be reached at lower flow velocities.

For the results in **figure 4.8** (T_{in} = 55 °C, c = 250 wppm) the heat transfer improvement HTI according to equation 4.4 has been calculated and presented in **figure 4.9**. As mentioned above the function with obstacles and surfactants must reach at least the heat transfer of water without obstacles to fulfil the necessary requirement for a technical application. That means the heat transfer improvement has to be at least 0 or bigger. This is characterized by the hatching.

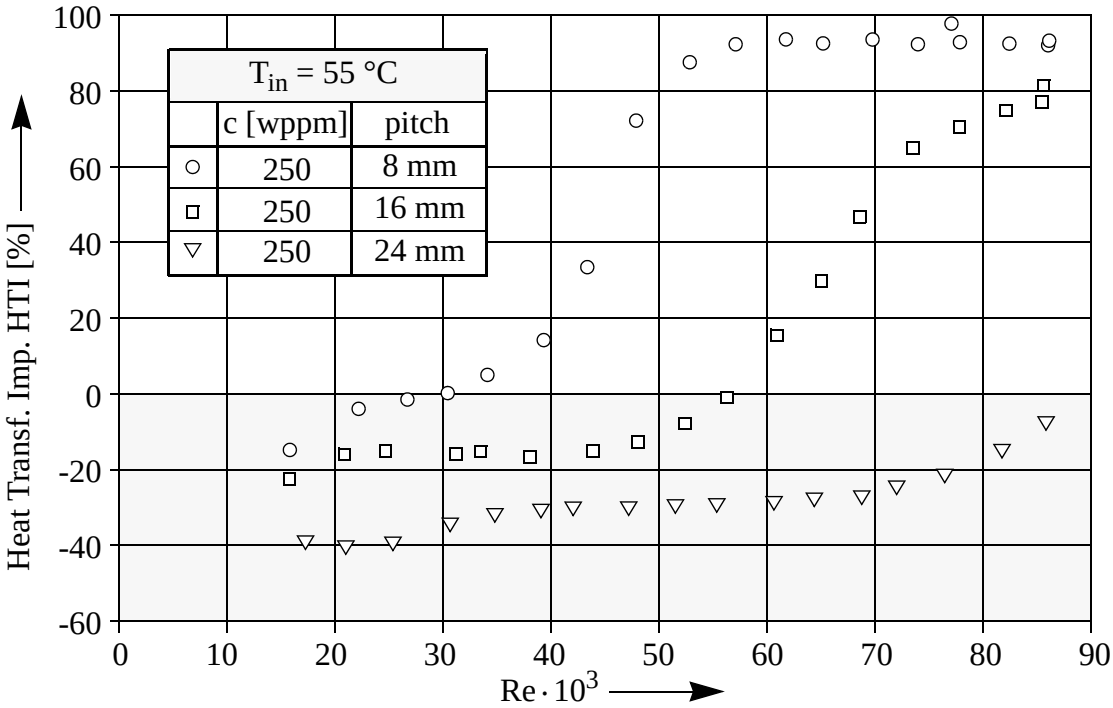


Fig. 4.9: Heat Transfer Improvement for drag reducing surfactant solutions with spiral springs of different pitches (c = 250 wppm, T_{in} = 55 °C)

In the investigated range of flow velocity the values of heat transfer improvement for the 24 mm spring are always below the demanded value of 0. On the other hand an avarage heat transfer reduction of 30 % (or a negative heat transfer improvement of -30 %) does only result in a reduction of overall heat ransfer of 15 % to 20 % (see **figure 2.5**) and in an even smaller heat output reduction [10]. But nevertheless, the operating conditions for drag reducing additives may not be worse compared to those for water - especially on the field of heat transfer.

Considering the other obstacles the conditions for the 8 mm spring are significantly better. For Reynolds numbers above 26,000 the HTI is above 0 and therefore the requirement concerning the heat transfer is fulfilled. For the 16 mm spring this condition is fulfilled for Reynolds numbers bigger than 56,000.

Concerning the technical application only the 8 mm spring can be used for a concentration of 250 wppm if the Reynolds number is above 20,000 (the heat transfer reduction of 2 % to 4 % in the range between 20,000 and 26,000 can be neglected).

Considering the concentration of 500 wppm the heat transfer is always worse than that for water without spiral springs. A compensation for the effect of heat transfer reduction cannot be achieved.

In **figure 4.10** the Nusselt characteristics for 125 wppm for different pitches are compared to the reference values for water without springs.

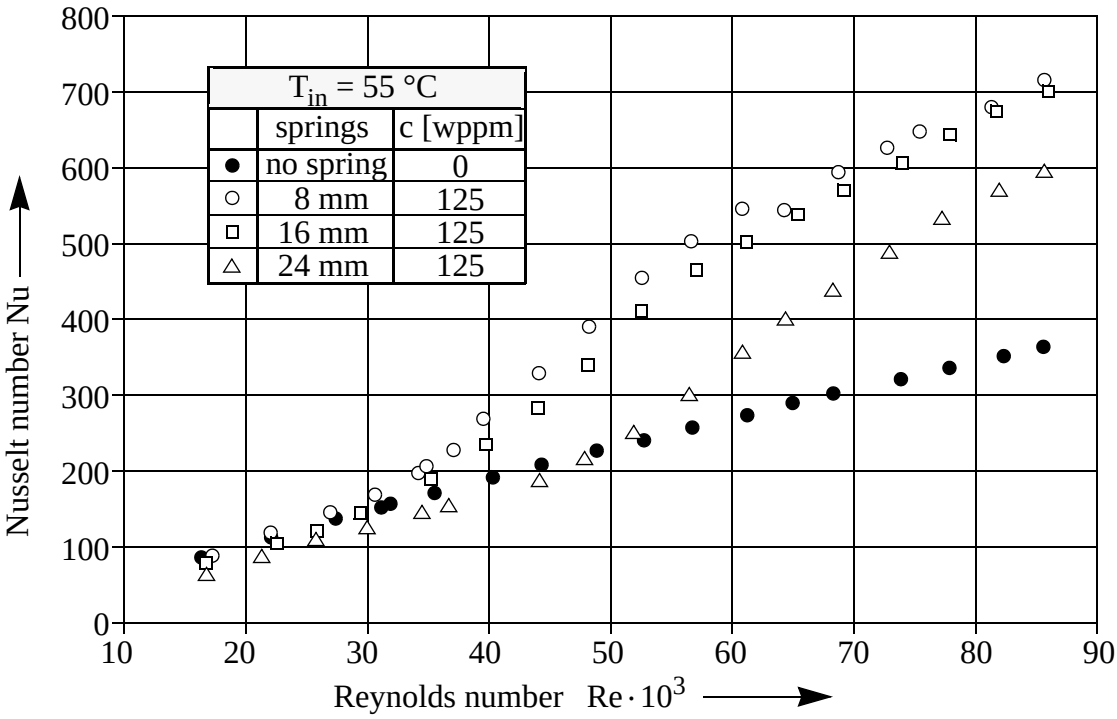


Fig. 4.10: Nusselt characteristic for drag reducing surfactant solutions for spiral springs of different pitches (c = 125 wppm, T_{in} = 55 °C) compared with water without obstacles

In this case the conditions are quite better compared to the results for 250 wppm. For 8 mm the heat transfer is better than the required reference values over the whole range of Reynolds numbers. Even the 16 mm does almost fulfill the required condition. Only at Re < 27,000 the values are slightly below the reference curve (HTI > -5 %).

Considering the 24 mm spring the requirements are fulfilled for Re < 47,000. For smaller flow velocities a HTI of -15 % in maximum appears. Whether this heat transfer reduction - compared to the origin state - can be tolerated has to be investigated in exact calculations.

Furthermore for a technical application, a concentration of 125 wppm could be too small to reach an optimum drag reduction in the pipe system. The results presented in **figure 4.7** show that the effect of the surfactants is decreasing significantly with the temperature. For an entrance temperature of 82 °C, the effect has almost disappeared.

Another important aspect is the behaviour of pressure drop inside pipes with obstacles. An increase in heat transfer can only be achieved by an increase in pressure drop. A technical application of the spiral springs is only meaningful if the increase in heat transfer does not lead to an unacceptable pressure drop. In chapter 4.2 the results of these investigations are presented.

4.2 Pressure drop

Analogous to the measurements of the heat transfer reference measurements without spiral springs have been analysed. In **figure 4.11** results of the pressure drop behaviour are presented. In this figure drag characteristics inside the smooth pipe as well as with the 8 mm spring are presented for water and drag reducing surfactant solutions (Habon-G, 250 wppm).

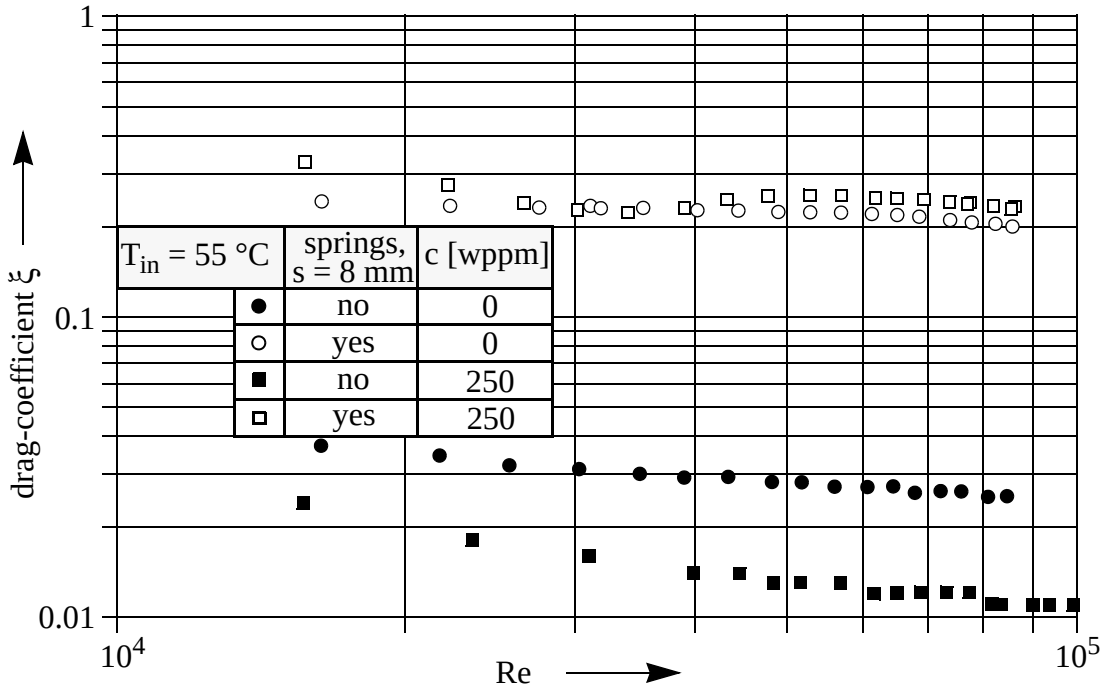


Fig. 4.11: Drag characteristic for water with spiral springs (s = 8 mm)

Considering the pipe without obstacles a significant drag reduction of more than 70 % can be observed. The critical wall shear stress could not be reached. Altogether the typical behaviour of drag reducing surfactant solutions inside straight pipes can be observed.

Comparing the measurements for water with and without obstacles a significant increase of drag coefficient can be regarded. The pressure drop is 600 to 700 % higher due to the installation of the 8 mm spring.

This confirms the expectation of a strong increase in pressure drop. The increase of heat transfer of about 90 % to 100 % is accompanied by a superproportional increase in pressure drop. Analogous to the heat transfer improvement, the pressure drop increase (PDI) is defined:

$$\mathrm{PDI} \equiv \frac{\Delta p_S - \Delta p_w}{\Delta p_w} \cdot 100\ \% \text{ or } \mathrm{PDI} \equiv \frac{\xi_S - \xi_w}{\xi_w} \cdot 100\ \%, \qquad \text{eq. (4.5)}$$

with: Index S: „Spring“ and
index w: „water, (no obstacles)“.

The definition PDI also uses the „origin state“ of the heat exchanger (water, no obstacles) as reference value. The value Δp_S or ξ_S can stand either for water or for surfactant solution.

The influence of the concentration on the drag characteristic for a 8 mm spring is presented in **figure 4.12**. Analogous to the heat transfer behaviour the drag reducing effect is extended to higher Reynolds numbers with increasing concentration (increase in critical wall shear stress). The critical value is the local minimum of the drag characteristic at low Reynolds numbers (e.g. 17,000 for 125 wppm, 20,000 for 250 wppm).

Considering the curves at lower Reynolds numbers the drag-coefficients are increasing with the concentration. This effect results from the increase in viscosity with increasing concentration. For Reynolds numbers above the critical value, the characteristics of the 125 wppm and the 250 wppm are approximately identical. They are 10 % to 15 % above the values for water. This corresponds with the higher Nusselt numbers of the surfactant solutions in this range [6].

The drag coefficients of the 500 wppm solutions are in the range Re < 30,000 above the curve for water. This is caused by the higher viscosity of this solution. For Re > 30,000, the drag reducing effect (compared to the case: water with springs) can be seen. The values are below those of water and reach a drag reduction (with the water value with springs as reference) of 20 % in maximum.

Figure 4.13 shows the drag characteristics for the 24 mm spring for different entrance temperatures and a concentration of 125 wppm. The same measurements which are presented in **figure 4.7** have been evaluated. It can be seen that the effect of drag reduction is moved to higher Reynolds numbers with increasing temperature. The same effect that has been received in the Nusselt characteristic can be seen in the drag characteristic for a temperature of 82 °C. According to the relatively low concentration the critical wall shear stress is low - the drag reducing effect is not very distinctive.

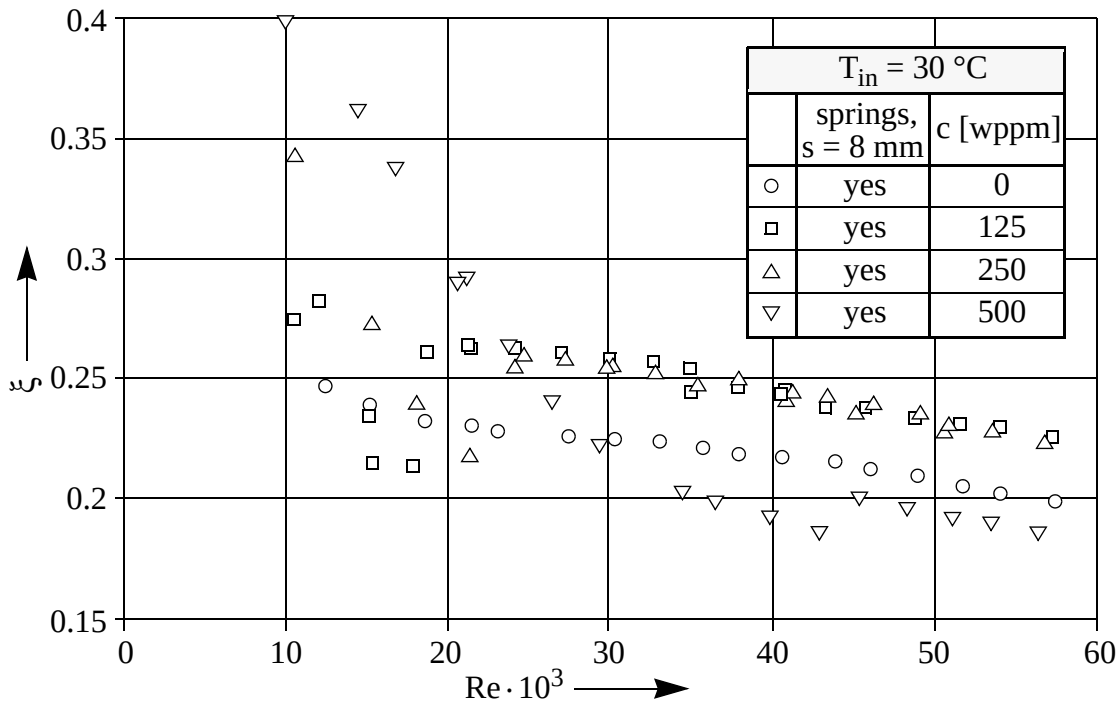


Fig. 4.12: Drag characteristic for water and drag reducing surfactant solutions of different concentrations with spiral springs (s = 8 mm)

Analogous to the heat transfer behaviour the critical values are - compared to the 8 mm spring in **figure 4.12** - moved to higher Reynolds numbers. The increase in turbulence and therefore the increase in wall shear stress due to the spiral springs is decreasing with rising pitch. Due to this, the critical wall shear stress moves to higher Reynolds numbers when increasing the pitch.

In **figure 4.14** the drag characteristics for springs of different pitches for water and surfactant solutions of 250 wppm at an entrance temperature of 55 °C are presented. Analogous to the Nusselt characteristic (decrease in Nusselt with increase in pitch) the drag coefficients decrease with the pitch of the spiral spring. The critical values are as well increasing with the pitch. The critical value of the 24 mm spring could not be reached. The local minimum of the 16 mm spring is not very distinctive.

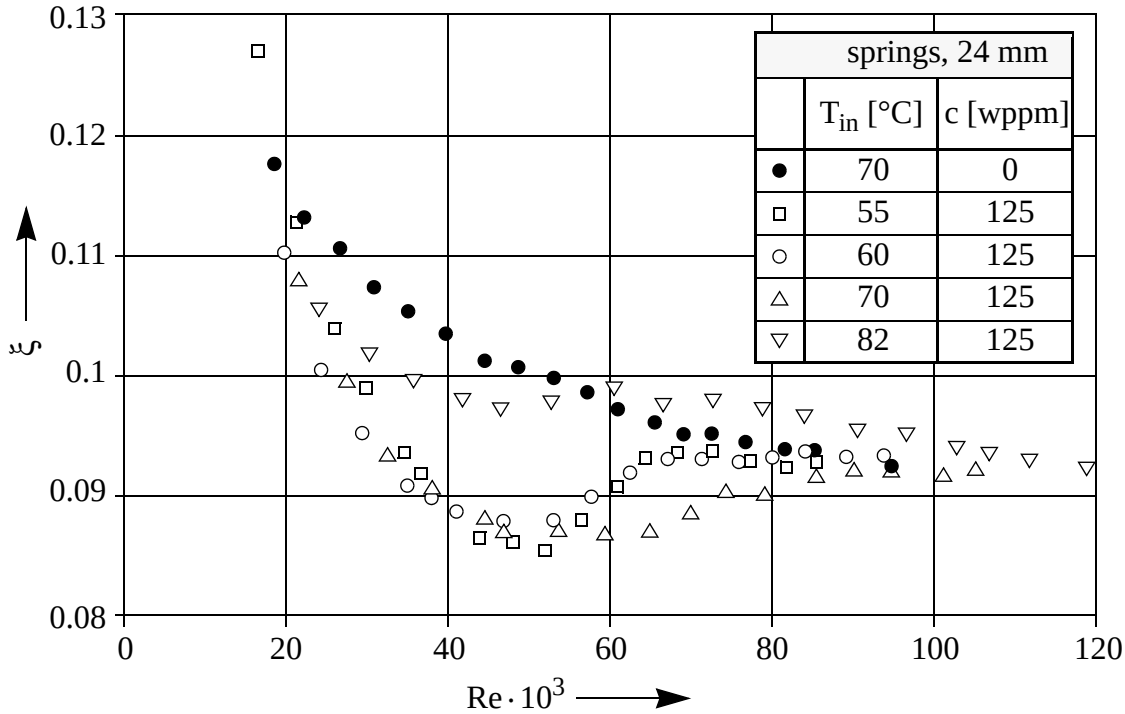


Fig. 4.13: Drag characteristic for water and drag reducing surfactant solutions of different temperatures with spiral springs (c = 250 wppm, s = 24 mm)

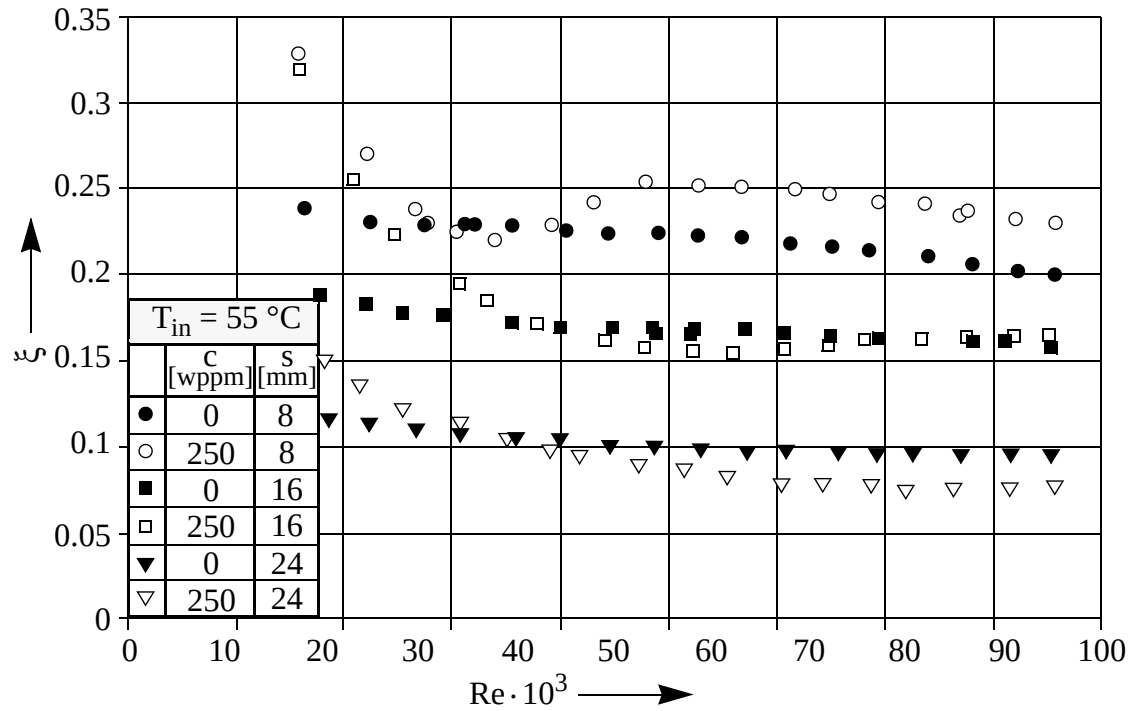


Fig. 4.14: Drag characteristic for water and drag reducing surfactant solutions for spiral springs of different pitches (c = 250 wppm, T_{in} = 55 °C).

Analog to **figure 4.9** in figure **figure 4.15** the pressure drop increase (PDI) is given as a function of Reynolds number. The increase in pressure drop is between 200 % (24 mm spring) and more than 800 % (8 mm spring). Therefore the increase in pressure drop is superproportional compared to the improvement of the heat transfer.

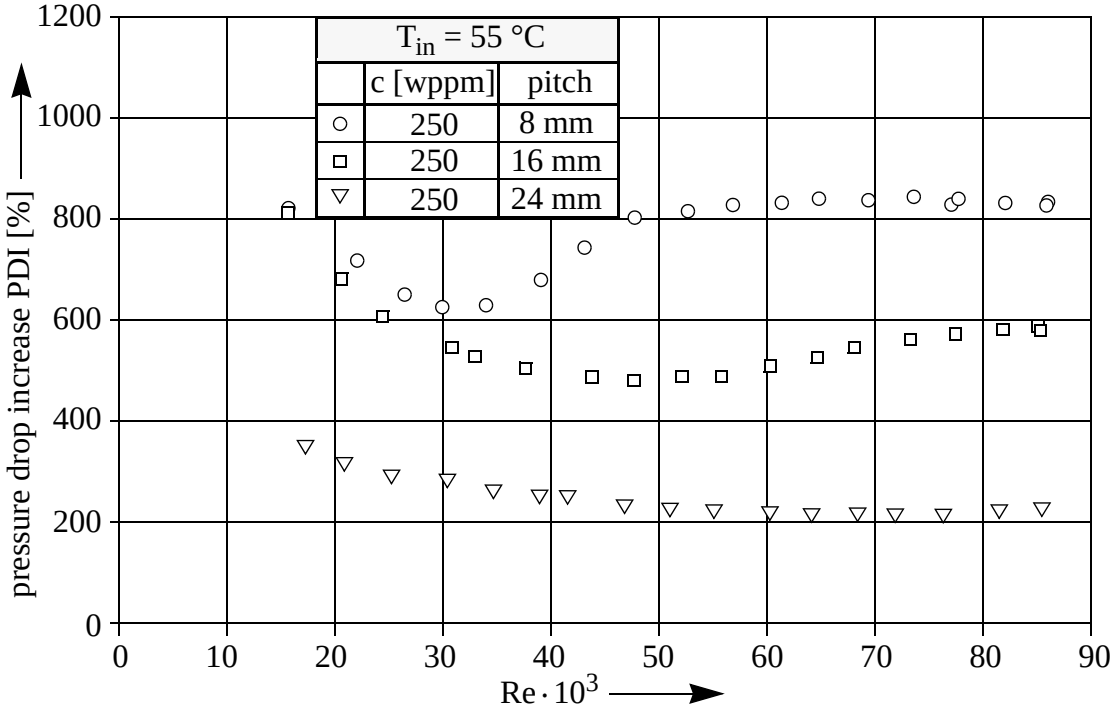


Fig. 4.15: Pressure drop increase for drag reducing surfactant solutions with spiral springs of different pitches (c = 250 wppm, T_{in} = 55 °C)

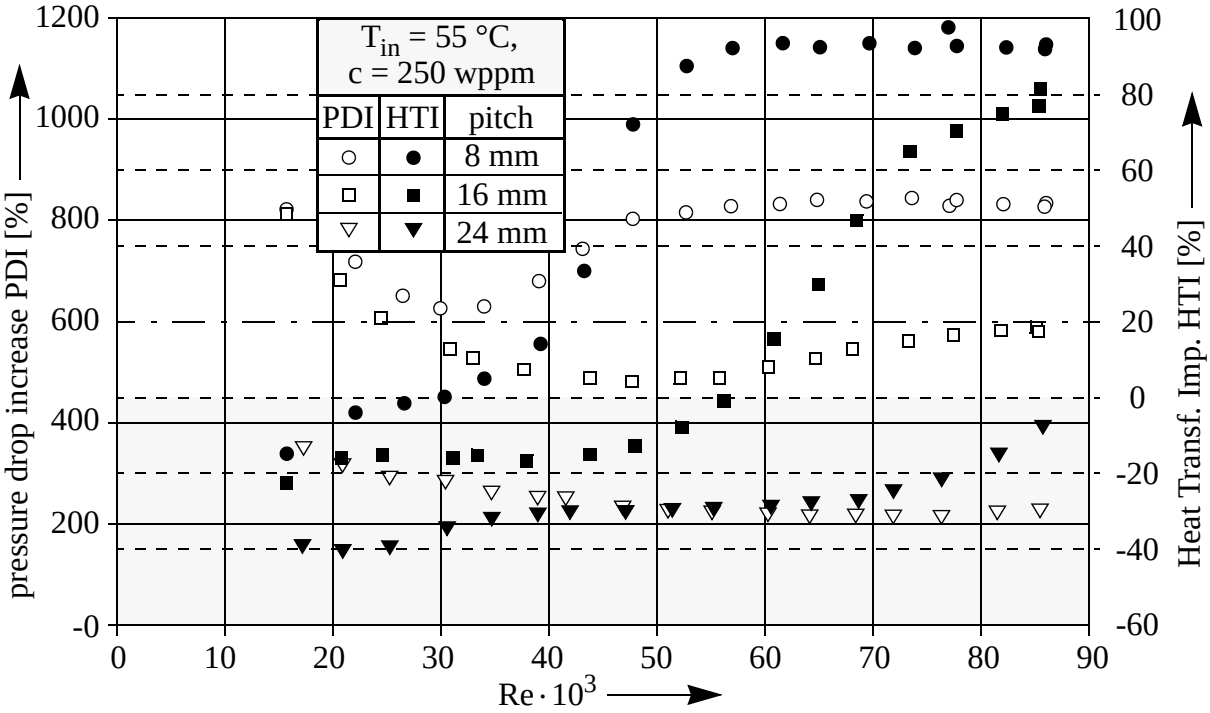


Fig. 4.16: Pressure drop increase (PDI) and heat transfer improvement (HTI) for drag reducing surfactant solutions with spiral springs of different pitches (c = 250 wppm, T_{in} = 55 °C)

In **figure 4.16** the pressure drop increase (PDI) is compared with the heat transfer improvement (HTI). At higher Reynolds numbers (Re > 40,000) the superproportional increase of pressure drop which has to be spent to improve the heat transfer can be seen impressively. Even for the 24 mm spring the increase in drag reduction is up to 400 % although the „increase“ in heat transfer is negative (-30 %).

At lower Reynolds numbers (Re < 40,000) the behaviour is the other way round. The heat transfer improvement is increasing while the pressure drop increase (PDI) is decreasing. This is an effect of higher viscosity at lower Reynolds numbers.

Considering the values of pressure drop increase which correspond to a heat transfer improvement of more than 0 (requirement for technical application), the increase of pressure drop is at least 500 % (16 mm spring at Reynolds = 57,500).

Instead a drag reduction of about 70 % (see **figure 4.11**) in case of the straight pipe without obstacles, an increase of more than 500 % (up to 850 % for the 8 mm spring, c = 250 wppm and T_{in} = 55 °C) has to spend to reach the desired heat transfer behaviour.

4.3 Importance concerning technical application

Concerning the technical application of spiral springs to improve the heat transfer in case of using drag reducing additives the aim - the increase in heat transfer to values above those for water without turbulence increasing obstacles - can be reached, if applying the right obstacle.

Considering the condensers in Herning where the measurements for mechanical stability and corrosion have been carried out (see part two of this report) the range of Reynolds numbers is between 25,000 and 75,000. Assuming that a concentration of 250 wppm has to be used to reach a sufficient drag reduction the 8 mm spring has to be installed.

This means - in case of nominal heat load - that the pressure drop will increase of about 600 % to 850 %. The pressure drops for nominal heat load of the origin state is 0.8 bar (condenser 1) and 0.95 bar (condenser 2). If a 8 mm spring will be installed and 250 wppm Habon-G will be used, the pressure drop will increase enormously (6.8 bar apparatus 1 and 8.075 bar apparatus 2).

Even if a solution is accepted that does not fulfil the required conditions properly - e.g. a 24 mm spring and a concentration of 125 wppm, the increase in pressure drop will be about 250 %. Which solution will be chosen has to be determined in dependence of the system and its operating conditions.

5. Summary and outlook

In this study the heat transfer and pressure drop of drag reducing surfactant solutions inside straight pipes with obstacles has been investigated. Therefore an existing test rig, mainly consisting of two closed loops (one for cooling and one for heating) has been modified. As cationic surfactant Habon-G (hexadecyldimethylpolyoxethylammonia-cation and 3-hydroxy-2-naphthoate as counter- ion) has been applied.

Spiral springs of different pitches have been used as obstacles to increase turbulence and therefore the heat transfer which ordinarily is significantly decreasing when applying drag reducing additives (up to 95 %). The springs consist of wire made of stainless steel of a diameter of 1 mm. The diameter of the spring has been a little bit higher than the inner diameter of the pipe to guarantee a certain support.

For pure water the installation of the obstacles leads to an increase in heat transfer of nearly 100 % in maximum. The behaviour is dependent on the pitch. The characteristic for the 8 mm and the 16 mm springs are almost identical while the Nusselt numbers of the 24 mm spring are smaller.

For surfactant solutions the heat transfer behaviour strongly depends on concentration, temperature and pitch. For 500 wppm the improvement in heat transfer is only small. The conditions of the origin state (water without obstacles) - which is the requirement for a suitable operation with surfactants - cannot be reached with this concentration. The Nusselt characteristic is parallel below that one for water without springs.

In contrast to the 500 wppm solution, the 125 wppm and 250 wppm show a completely different behaviour. At low Reynolds numbers the characteristics start in the range of that one for water without obstacles and rise - at a certain Reynolds number - up to the values for water with obstacles. This critical value is dependent on concentration, temperature and pitch of the spring. With increasing pitch the critical point is moving to higher Reynolds numbers as well as with increasing concentration. With increasing temperature the critical values also move to higher flow velocities until a certain temperature which is dependent on concentration is reached. Behind this point, the values are moving to lower Reynolds numbers and the characteristic is moving to higher Nusselt numbers because the drag reducing effect is weakening.

For a concentration of 250 wppm the heat transfer conditions of the origin state can be reached with the 8 mm spring above Reynolds = 25,000 and with the 16 mm spring above 56,000. For 125 wppm the conditions of the origin state can be reached with all investigated springs (for the 24 mm spring the condition Re > 47,000 has to be fulfilled).

Therefore the aim to reach the heat transfer coefficients of pure water without obstacles could be reached under certain conditions.

On the other hand measurements of the pressure drop showed a significant increase. This increase is - compared to the increase in heat transfer - superproportional. The strongest increase could be observed for the 8 mm spring for 250 wppm. In this case the pressure drop is increasing of about 850 %.

The course of the drag characteristic mainly shows the same dependencies as the Nusselt characteristics. The most important influence has the pitch of the spring. Comparing the 8 mm and the 24 mm spring for a 250 wppm surfactant solution the pressure drop increase of the 8 mm spring is 850 % in maximum compared to 200 % in maximum for 24 mm.

Considering the technical application and assuming that the heat transfer has to be at least as large as for water without obstacles an enormous pressure drop increase has to be spent. Thus for the technical application detailed calculations of the changed conditions of the complete system have to be carried out in order to check the installation of spiral springs inside pipes as a measure to improve the behaviour of tube bundle heat exchangers when using drag reducing additives.

6. Symbols

Symbol	Description	Unit
A	heat exchange area	[m^2]
c	concentration	[wppm]
d	diameter	[m]
k	heat transfer coefficient	[$W/(m^2K)$]
k_0	overall heat transfer coefficient	[$W/(m^2K)$]
L	length	[m]
L_e	entrance length	[m]
$\dot{M}$	mass flow rate	[kg/s]
p	pressure	[Pa]
Δp	pressure loss	[Pa]
$\dot{Q}$	heat load	[W]
R_f	fouling factor	[1]
s	thickness of the wall	[m]
s	pitch	[mm]
T	temperature	[T]
ΔT_m	mean temperature difference	[K]
ΔT_{ln}	logarithmic temperature difference	[K]
$\dot{V}$	flow rate	[m^3/s]
w	flow velocity	[m/s]
Δ	difference	[1]
η	dynamic viscosity	[Pa·s]
λ	heat conductivity	[W/(mK)]
ρ	density	[kg/m^3]
τ	shear stress	[Pa]
ω	ratio of heat transfer coefficients	[1]
ξ	drag coefficient	[1]

Indices	Meaning
a	outer pipe
i	inner pipe
in	entrance
max	maximum
min	minimum
opt	optimum

out	outlet
s	surfactant solution
S	Spring
w	water
W	wall
*	critical value
1	primary side/pipe side
2	secondary side/shell side

Special symbols	**Meaning**
CMC I/II	first/second Critical Micelle Concentration
DH	district heating
DR	Drag Reduction
HE	Heat Exchanger
HOR	Heat Output Reduction
HTI	Heat Transfer Improvement
HTR	Heat Transfer Reduction
MID	Magnetic-Inductive Flowmeter
OHR	Overall Heat Transfer Reduction
PDI	Pressure Drop Increase
SIS	Shear Induced State
SoSal	Sodium Salicylate
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	$Pr = \frac{\nu}{a} = \frac{\eta \cdot c_p}{\lambda}$

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