Effect of Baffle Orientation on Heat Transfer and Pressure Drop of Shell and Tube Heat Exchangers with and without Leakage Flows

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EFFECT OF BAFFLE ORIENTATION ON HEAT TRANSFER AND PRESSURE DROP OF SHELL AND TUBE HEAT EXCHANGERS WITH AND WITHOUT LEAKAGE FLOWS

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ABSTRACT The effect of baffle orientation on the heat transfer and pressure drop of shell and tube heat exchangers in the domain of turbulent flow is investigated numerically using the commercial CFD code FLUENT. The segmentally baffled shell and tube heat exchangers considered follow the TEMA standards and consist of 76 and 660 plain tubes respectively, with fixed outside diameter and arranged in a triangular layout. Air, water and engine oil with Prandtl numbers in the range of 0.7 to 206 are used as shell-side fluids. For horizontally and vertically orientated baffles, simulations are performed using different flow velocities at the inlet nozzle. A shell-side gain factor suitable for the assessment of shell and tube heat exchangers is introduced as ratio of the shell side heat transfer coefficient to the shell-side pressure drop. To facilitate the decision between horizontal and vertical baffle orientation, a performance factor Φ is used as ratio of the gain factor for horizontally orientated baffles to the gain factor for vertical baffle orientation.

The simulation results show a significant influence of the baffle orientation on the shell-side pressure drop and heat transfer of shell and tube heat exchangers. In the shell and tube heat exchanger with leakage flows the vertical baffle orientation seems to be more advantageous than the horizontal orientation. The benefit of vertical baffle orientation on horizontal baffle orientation is more noticeable for gases. Contrariwise, the simulation results for shell and tube heat exchangers without leakages show the advantage of the horizontal baffle orientation over the vertical orientation, particularly in the inlet and outlet zone for all investigated shell-side fluids.

The comparison of calculation results with and without leakage flows presents different behaviour and underlines the importance of a consideration of tube-baffle leakage and bypass streams for the prediction of the performance factor of technical heat exchangers.

NOMENCLATURE

$Nu_{0,AW}$	Modified shell side Nusselt number	*Greek symbols*	
$Re_{\psi,l}$	Modified shell side Reynolds number	ψ	Void ratio of tube bank
		Γ	Gain factor
Abbreviation		Φ	Performance factor
HEX	Heat exchanger		

INTRODUCTION

Numerous methods are cited in literature for the design of shell and tube heat exchangers, which consider the effect of different geometrical factors like leakages, baffle spacing, tube layout and arrangement and baffle cut on heat transfer and pressure loss:

Colburn [1933] suggested a correlation for the shell-side heat transfer coefficient, limited to staggered tube layout and based on the assumption that shell-side flow behaves similarly to flow across ideal tube banks. Grimison [1937] modified Colburn´s correlation by considering non-isothermal effects. Donohue [1949] and Kern [1965] published shell-side methods based on overall data from baffled heat exchangers which assumed that the baffles are used to direct the shell fluid in the tube bundle perpendicularly to the bundle. Due to the limited number of available data only an insufficient variation of basic geometrical parameters like baffle spacing, baffle cut and tube layout was presented. To overcome this deficiency, safety factors were introduced which lead to reduced accuracies for the prediction of shell-side heat transfer coefficient and pressure drop. Tinker [1958] suggested a schematic flow pattern where the shell-side flow is divided into a number of individual streams and the pressure drop of the main effective cross-flow stream acts as driving force for the other streams. This early analysis of shell-side flow was extended by Palen and Taborek [1969]. Heat Transfer Research, Inc. (HTRI) [1969] developed several methods for the for the shell-side flow pressure drop and heat transfer based on stream analysis methods. Bell [1963] published the so-called Delaware method which implies that the specific heat exchanger is described completely geometrically and the process specifications for all streams are given. The VDI [2002] recommends the calculation of the pressure loss and the mean heat transfer coefficient on the shell side according to the Delaware method and studies published by Gnielinski [1977,1978,1983] and Gaddis [1977,1978,1983]. Using the VDI method the heat transfer coefficient and pressure drop for pure cross-flow in an ideal tube bank is calculated and then the effective average shell-side heat transfer is evaluated by use of different stream-flow correction factors. The overall shell-side pressure loss is the sum of the pressure drop for cross-flow between the baffle tips, the pressure drop in the end zones, the pressure drop in the baffle windows and the pressure drop in both the inlet and outlet nozzle. This method was checked by Gnielinski and Gaddis against a large number of measurements from the literature. A maximum error of ±15% for the overall heat transfer coefficient and ± 35% for the pressure loss compared to the measurements was found. Mohammadi et. al. [2006] investigated numerically the effect of baffle orientation and baffle cut on heat transfer and pressure drop of a shell and tube heat exchanger. In order to perform the calculations for a heat exchanger with 660 tubes the authors had to neglect leakage flows.

In summary, it can be noted that well-established calculation methods consider the effect of different baffle orientations on heat transfer and pressure loss not or only insufficiently. Nowadays available hard- and software for the calculation of heat transfer and pressure loss in complex heat exchangers simplifies the investigation of such geometrical variations since no cumbersome experiments are required. However, the effect of simplifying assumptions for the numerical treatment, like the neglection of leakage flows (cf. Mohammadi et. al. [2006]) on the performance of the heat exchanger has to be evaluated and assessed critically. Hence, in the context of the present study, the performance of shell and tube heat exchangers with different baffle orientation (horizontal and vertical) as well as with and without leakage flows are investigated numerically in more detail.

NUMERICAL METHOD

Decription of problem Two standard shell and tube heat exchangers consisting of 76 tubes (HEX 1) and 660 tubes (HEX 2) are considered. Each heat exchanger consists of different flow sections such as the inlet zone, several intermediate zones located between adjacent baffles and the outlet zone. As an example a simplified sketch of the geometry of HEX 1 in which the tube bank is not depicted for simplicity is shown in Figure 1. Both heat exchangers are single pass E type shell with equally-sized inlet, outlet and central baffle spacing as well as single segmental plate baffles with different orientation. The geometrical data are based on TEMA standard and listed in Table 1 for HEX 1 and Table 2 for HEX 2. The baffle cut for the heat exchangers HEX 1 and HEX 2 is chosen in such a way that the ratio of the heat transfer area of the tubes in the baffle window to the heat transfer area of the tubes in one baffle spacing zone is similar for both of the two heat exchangers. Clearance between tube outside diameter and the baffle holes as well as clearance between the shell and the baffles and the associated leakage flows are considered for HEX 1. No leakage flows are considered for HEX 2. Geometrical aspects of the tube bank are shown in Figure 2.

Table 1
Geometrical Data of HEX 1

shell diameter D_{shell}	254.00 mm	tube pitch l_{tp}	23.81 mm
tube outside diameter d_{tube}	19.05 mm	tube pitch angle	60 degree
nozzle inside diameter D_{nozzle} at inlet	81.20 mm	baffle spacing L_{bs}	112.06 mm
nozzle inside diameter D_{nozzle} at outlet	105.56 mm	baffle cut (in percent of D_{shell})	20 %
nozzle length L_{nozzle}	113.00 mm	baffle orientation	hor./ver.
baffle thickness	3.20 mm	number baffle spacing zones	4
radial clearance between inside shell and baffle	1.13 mm	radial clearance between tube outside and baffle hole	0,19 mm

Table 2
Geometrical Data of HEX 2

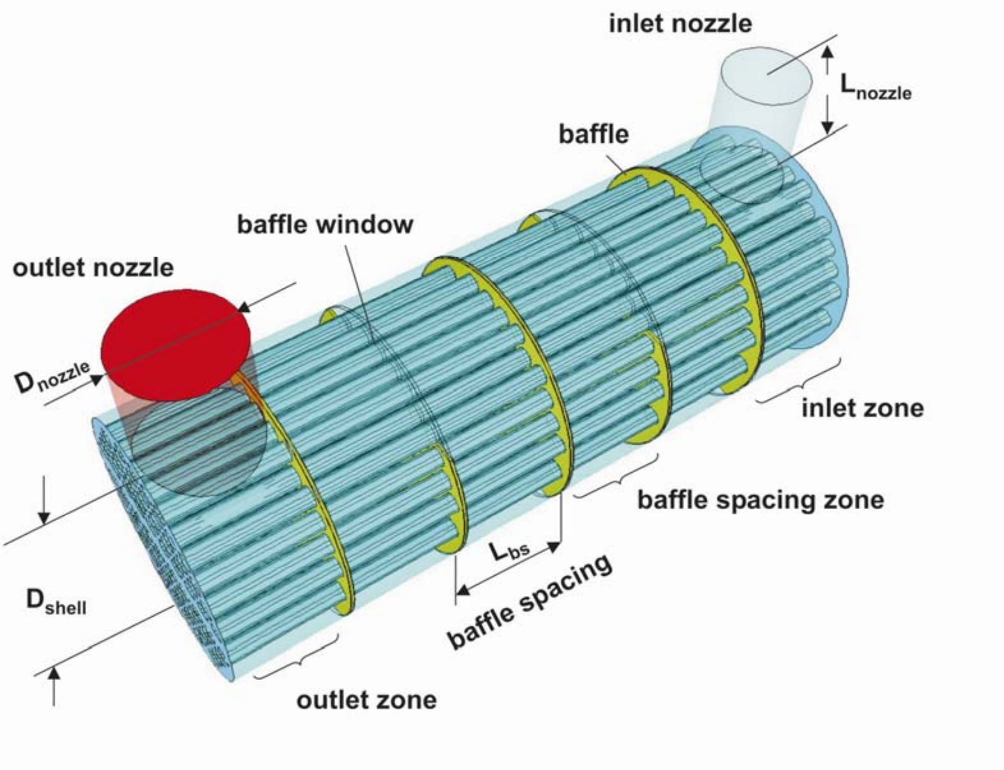
shell diameter D_{shell}	590.93 mm	tube pitch l_{tp}	20.64 mm
tube outside diameter d_{tube}	15.88 mm	tube pitch angle	60 degree
nozzle inside diameter D_{nozzle} at inlet	154.18 mm	baffle spacing L_{bs}	262.41 mm
nozzle inside diameter D_{nozzle} at outlet	154.18 mm	baffle cut (in percent of D_{shell})	24 %
nozzle length L_{nozzle}	192.72 mm	baffle orientation	horiz./vertic.
baffle thickness	6.35 mm	number baffle spacing zones	6

For both heat exchangers the shell fluid has constant temperature T_{inlet} and uniform velocity u_{inlet} at the inlet nozzle. All tubes in the bundle are at constant wall temperature T_{tube} as it occur during condensation or evaporation inside the tubes. Only one mode of operation is considered: The shell fluid is heated if $T_{tube} > T_{inlet}$ is valid.

In order to define the baffle orientation of the heat exchanger, reference planes and vectors are introduced as given in Figure 2.3:

- The *baffle orientation plane* is parallel to the tube-bundle axis and touches the baffle edge,
- the *inlet (outlet) plane* contains the inlet (outlet) area of the inlet (outlet) nozzle,
- *face vectors* are normal to the planes considered and directed to the centre of the shell,
- the *baffle vector* is normal to the baffle orientation plane and directed towards the outside of the shell.

Heat exchangers with horizontal baffle orientation may then be characterized by a counter-clockwise angle between the baffle vector and the face vector of the inlet plane equal to 0° (at the inlet zone) and 180° (at the first intermediate zone), respectively. Similarly, heat exchangers with vertical baffle orientation show a value of 270° (at the inlet zone) and 90° (at the first intermediate zone).



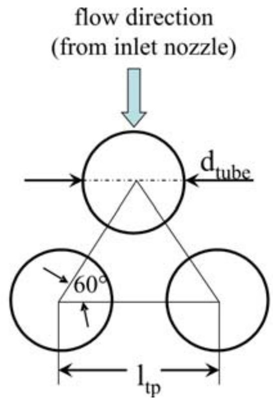


Figure 1. Geometrical aspects of HEX 1 (horizontal baffle orientation)

Figure 2. Geometrical aspects of the tube bank

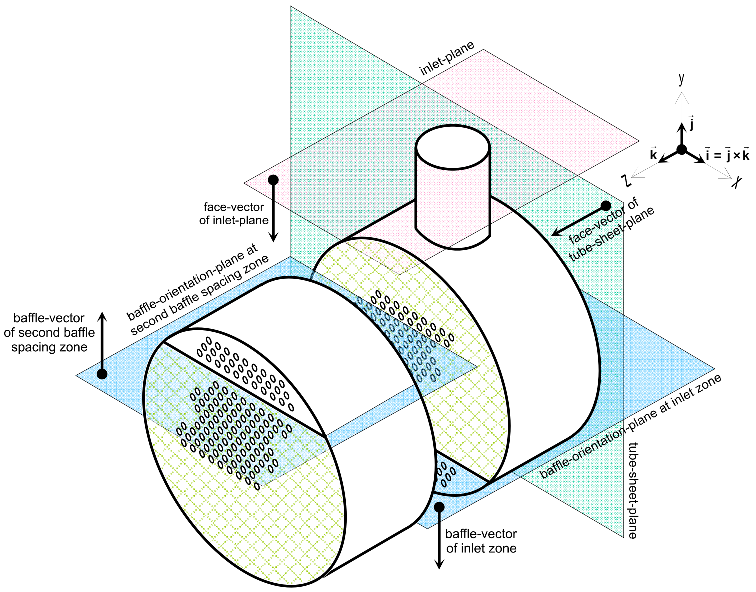


Figure 3. Plane und vector system for the definition of the baffle orientation

Geometry and mesh structure The heat exchangers are subdivided into zones as depicted in Figure 1 exemplarily for HEX 1: One inlet zone, several baffle spacing zones and one outlet zone. Each zone has been built from around 200 (HEX 1) and 2.000 (HEX 2) single volumes, respectively. Roughly 80% of these volumes are longitudinal prisms. The combination of neighbouring prisms builds the fluid volume around each tube. A set of these conjugated structures, looking like a honeycomb, represents the fluid bulk around the tube bank. Due to the constant tube wall temperature, no additional grid is required to mesh the tube-side fluid volume. All 3D elements applied for the mesh schemes are hexahedral. In total about 1,200,000 mesh elements are used for each of the two heat exchangers HEX 1 and HEX 2. Almost 99% of the 3D elements have aspect ratio less than 15. About 97% of all mesh elements are less-skewed elements with values less than 0.4 for the skewness factor.

CFD Model For both heat exchangers HEX 1 and HEX 2 a velocity inlet boundary condition is used to describe the flow conditions at the inlet nozzle. Steady uniform velocities u_{inlet} normal to the inlet surface of the inlet nozzle are considered which result in Reynolds numbers $Re_{nozzle} = \rho_{inlet} \cdot u_{inlet} \cdot D_{nozzle} / \mu_{inlet}$. The relevant data for the description of the boundary condition at the inlet nozzle are listed in Table 3. Temperature dependent polynomial functions are used to describe the physical properties of the shell fluids at the given operating pressure of 7.9 bar.

Table 3
Boundary Condition at the Inlet Nozzle

	HEX 1	HEX 2
T_{inlet}	370 K	
$T_{tube\ wall}$	400 K	
Re_{nozzle}	2.000 - 139.000	10.000 - 305.000
Turbulence specifications	Turbulence intensity: $0.16 \cdot Re_{nozzle}^{-1/8}$ Length scale : $0.07 \cdot D_{nozzle}$	

Due to leakage flows which are considered for HEX 1, interactions between the different flow sections (inlet and outlet zones, intermediate baffle spacing zones) may occur up-stream and downstream. Hence HEX 1 is simulated as a whole, consisting of different flow sections with an outflow boundary condition at the outlet nozzle. All other boundaries of HEX 1 are adiabatic.

In contrast to HEX 1, no leakage flows are considered for HEX 2. In this case it is expected that no influence of different flow sections up-stream occurs. Each flow section can be treated separately. Flow conditions at the exit of each flow section of HEX 2 are outflow boundary conditions. The outflow is used as inlet boundary condition of the adjacent zone downstream. All other boundaries of HEX 2 are adiabatic. The procedure of the numerical treatment of HEX 2 is depicted in Figure 4, in which the tube bank is not illustrated for reason of clearness and a horizontal baffle orientation is assumed.

The governing equations for mass, momentum and energy conservation are solved numerically using the finite-volume-method-based CFD code FLUENT. A segregated solution method solves the governing equations in implicit formulation sequentially. For the pressure-velocity coupling the SIMPLE algorithm is applied. The RNG k-ε model is selected as turbulence model. A standard wall function proposed by Launder and Spalding [1974] bridges the viscosity-affected region between the wall and the fully turbulent region. A check of the dimensionless sublayer-scaled distance y^+ confirmed the correctness of the mesh structure and of the applied semi-empirical function to treat the near wall regions.

An analysis of the iterative numerical procedure showed a maximum average convergence error of less than 0.004% for properties like continuity, velocity, energy, kinetic energy and dissipation rate of kinetic energy of turbulence.

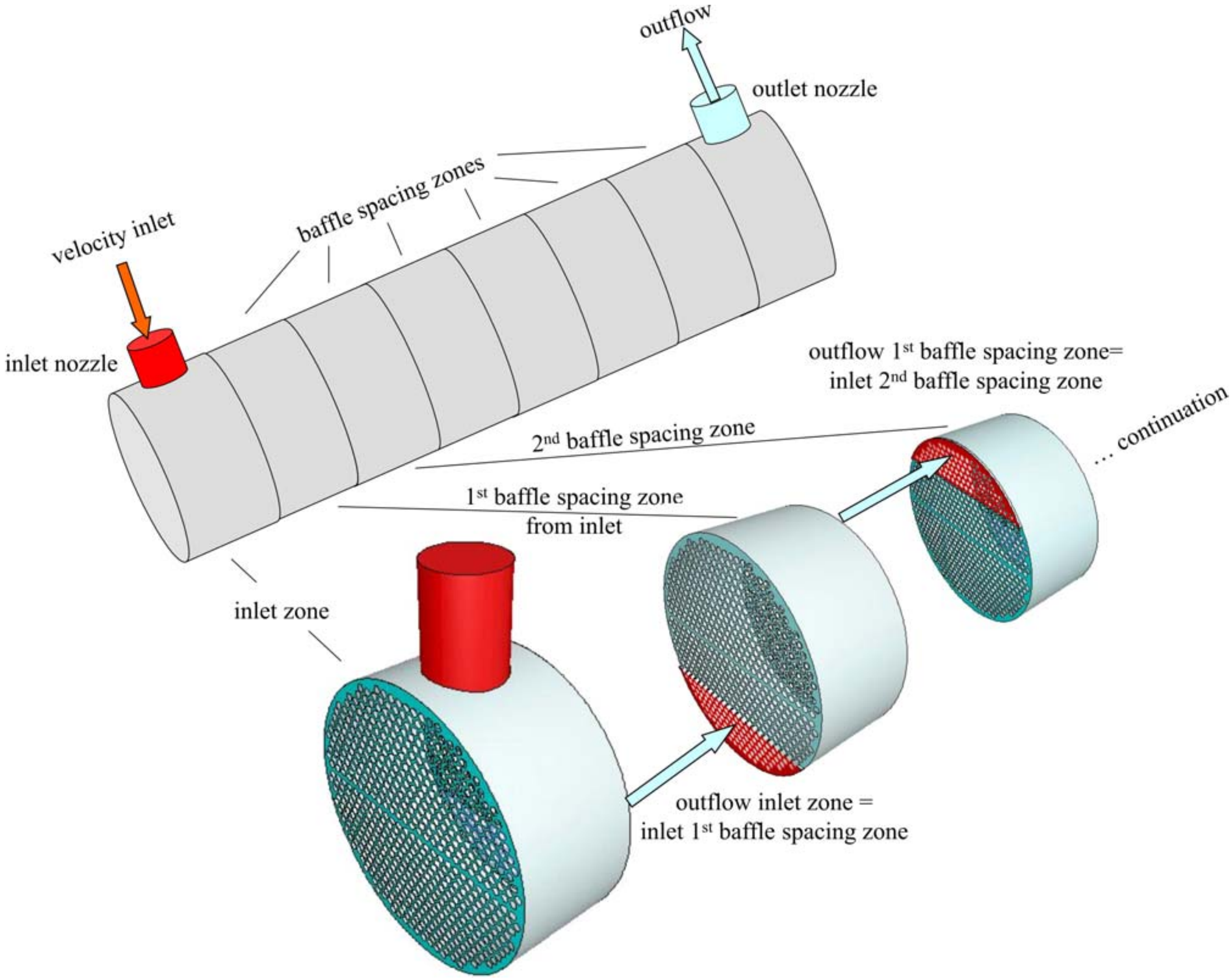


Figure 4. Zoning of HEX 2

VALIDATION

Example 1 The numerical scheme and mesh structure have been validated using experimental data for four ideal tube banks with triangular tube layout published by Kays et. al. [1954]. For this purpose simulated values for the dimensionless pressure loss and heat transfer coefficient were compared with the available experimental data. The Fanning friction factor f and the modified Colburn j-factor are used to characterize the pressure drop and heat transfer:

$$f = \frac{2 \cdot r_h \cdot \Delta p_{\text{tube bank}}}{\rho \cdot u_{max}^2 \cdot L} \quad (1) \qquad j_H = St_o \cdot Pr^{2/3} = \frac{Nu}{Re_o \cdot Pr^{1/3}} \quad (2) \qquad Re_o = \frac{\rho \cdot u_{max} \cdot d_{tube}}{\mu} \quad (3)$$

u_{max} denotes the maximum velocity of the shell fluid within the tube bank and d_{tube} the tube outside diameter. The geometrical property r_h is defined as $r_h = A_c \cdot L / A_h$ with A_C as minimum free flow area, L as effective flow length of the tube bank ($L = l_1 \cdot n_1$) and A_h as heat transfer area.

The comparison between the simulated results and the experimental data is given in Figure 5 and shows good agreement. A maximum deviation of ±10 % is found between simulated and measured data.

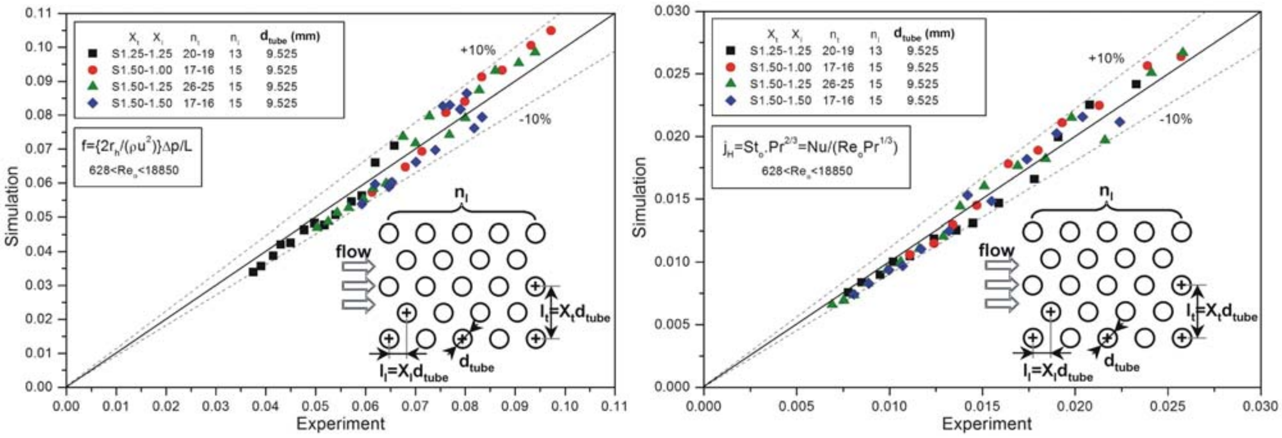


Figure 5. Comparison of CFD results and experimental data published by Kays et. al. [1954] for pressure loss (left) and shell-side heat transfer (right) of tube banks

Example 2 For a second validation of the numerical scheme the shell-side Nusselt number of a heat exchanger comparable to HEX 1 with horizontal and vertical baffle orientation was calculated numerically and compared with values obtained with the VDI method (cf. VDI Wärmeatlas [2002]). Geometrical data of the heat exchanger are given in Table 1. The Nusselt number is defined as $Nu_{shell} = \left(h_{shell} \pi \cdot d_{tube} / 2 \cdot k\right) \cdot \left(Pr_{wall} / Pr_{shell}\right)^m$ with m= 0.11 for cooling and m=0.25 for heating. Water, air and engine oil serve as shell fluids. Properties of the shell fluids are calculated at the average shell-side temperature $(T_{inlet}+T_{outlet})/2$, while Pr_{wall} has to be calculated at the tube wall temperature T_{wall}.

Figures 6 and 7 show the comparison of the CFD-based results with the values obtained by the VDI method. The comparison shows a satisfying level of agreement for the prediction of the shell-side Nusselt number and pressure drop with horizontally orientated baffles. However, the comparison indicates that the VDI method undersizes a shell and tube heat exchanger with vertical baffle orientation, significantly.

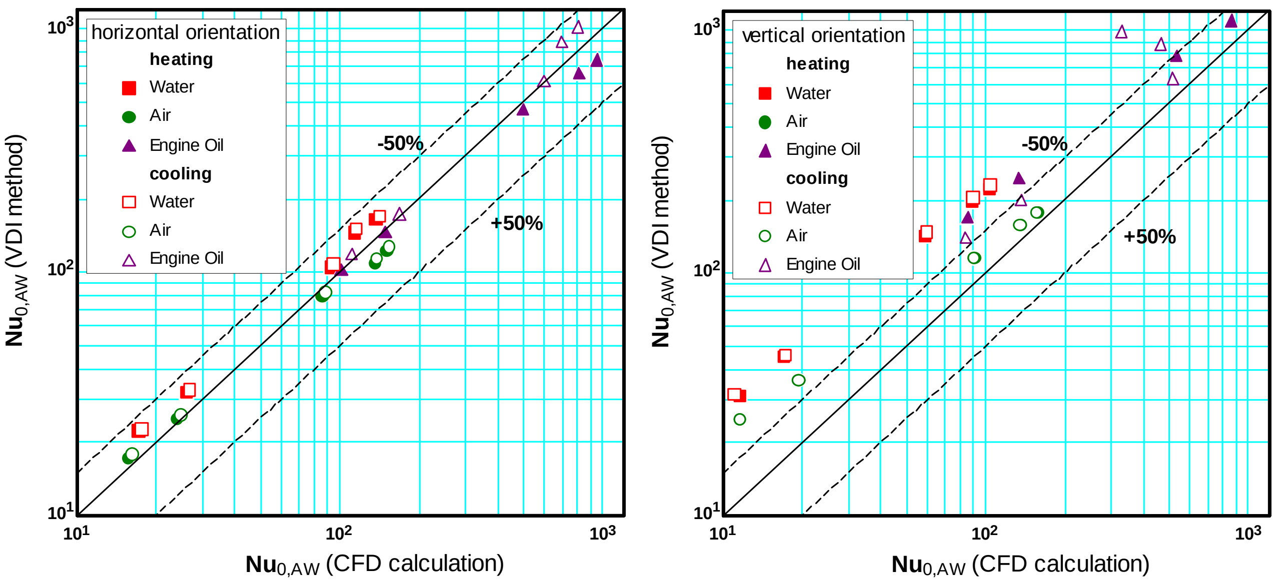


Figure 6. Comparison of shell-side Nusselt numbers, CFD calculation versus VDI-method, left: horizontal baffle orientation, right: vertical orientation

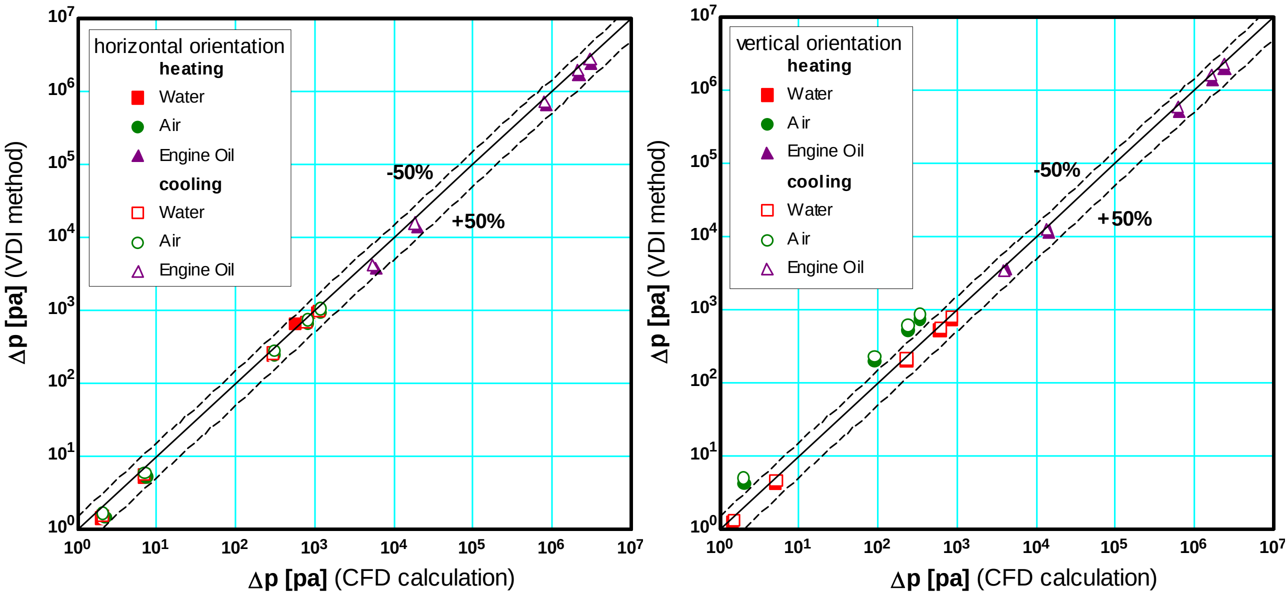


Figure 7. Comparison of shell-side pressure drop, CFD calculation versus VDI method, left: horizontal baffle orientation, right: vertical baffle orientation

RESULTS AND DISCUSSION

Parameters and Definition In the following the effect of different baffle orientations on the thermo-fluid-dynamic behaviour of the shell-side fluid in shell and tube heat exchangers is discussed. Two heat exchangers HEX 1 (cf. Table 1) and HEX 2 (cf. Table 2), one direction of heat flow (heating), two directions of baffle orientation (horizontal, vertical) and three shell fluids (air, water and engine oil) in the range of 0.7 to 206 for the Prandtl number - based on thermo-physical properties at operating temperature and pressure - were considered for the numerical investigations. In order to assess the simplifying assumption to neglect leakage flows, the calculation of HEX 1 is performed with leakage flows and the calculation of HEX 2 without leakage flows. Heat transfer and pressure drop of the heat exchangers are characterized by the overall Nusselt number Nu_{shell} (cf. previous chapter) and the dimensionless Karman number defined as

$$Nk_{shell} = \rho \cdot D^3_{shell} \cdot \Delta p / (\mu^2 \cdot L_{baffle spacing}) \tag{4}$$

Since the heat transfer coefficient relates to the energy recovered by the heat exchanger and the pressure drop refers to the work which is necessary to maintain the shell-side fluid flow, a shell-side gain factor suitable for the assessment of shell and tube heat exchangers may be introduced as ratio of the shell-side heat transfer coefficient to the shell-side pressure drop:

$$\Gamma_{shell} = \frac{Nu_{shell}}{Nk_{shell}} \sim \frac{h_{shell}}{\Delta p_{shell}} \tag{5}$$

To facilitate the decision between horizontal and vertical baffle orientation, a performance factor Φ is defined:

$$\Phi = \frac{(\Gamma_{shell})_{horizontal}}{(\Gamma_{shell})_{vertical}} \tag{6}$$

A performance factor Φ greater than one indicates that a heat exchanger with horizontally orientated baffles is more desirable than one with vertical baffle orientation. Rationally, no advantage exists between different baffle orientations when the performance factor is equal or near to unity.

Local behaviour of HEX 1 Figures 8-1 to 8-3 show the performance factor of HEX 1 (with leakage flows) at different zones (1.. inlet zone, 2-5 … baffle spacing zones, 6 … outlet zone) for the shell-side fluids air, water and engine oil. The Reynolds number at the inlet nozzle $Re_{nozzle} = \rho_{inlet} \cdot u_{inlet} \cdot D_{nozzle} / \mu_{inlet}$ with D_{nozzle} in accordance with Table 1 is used as a parameter and covers the range $2,000 \leq Re_{nozzle} \leq 139,000$.

It can be noticed that the local performance factor for HEX 1 is almost always lower than one and decreases downstream along the baffle spaces. This effect is more noticeable for lower Reynolds numbers. The decay of the performance factor becomes greater for lower Prandtl numbers.

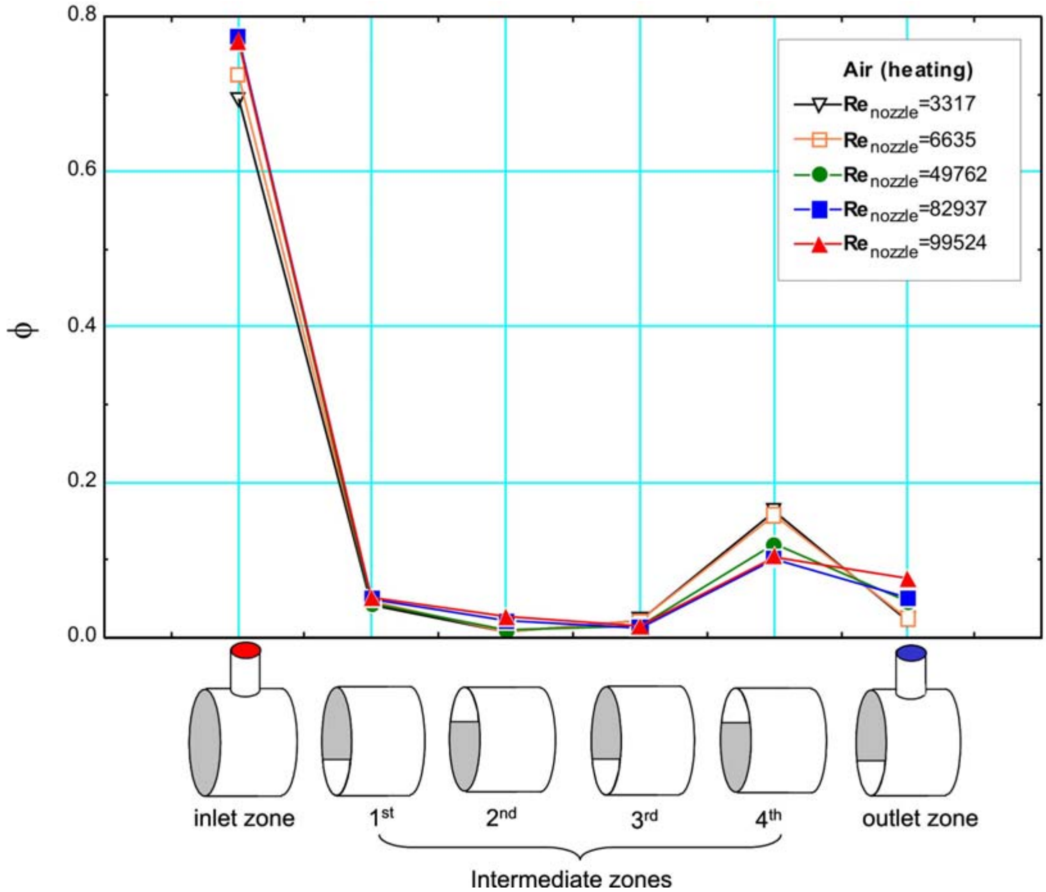


Figure 8-1. Local performance factor of HEX 1 for air

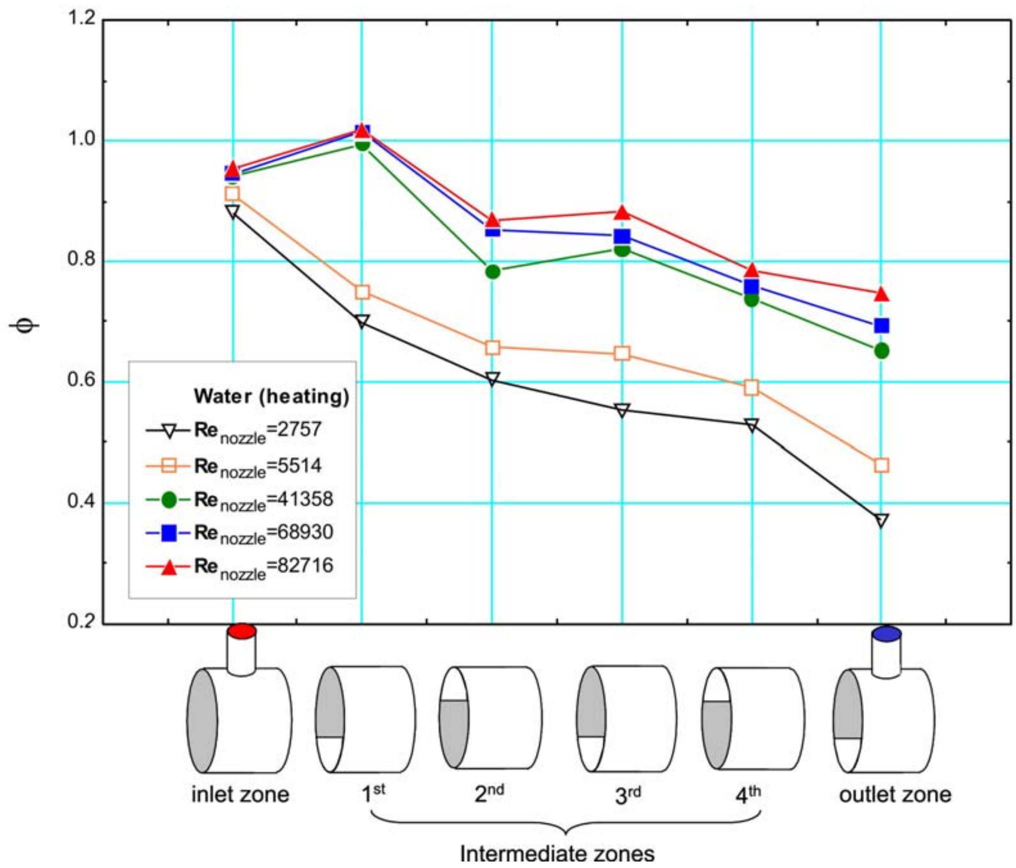


Figure 8-2: Local performance factor of HEX 1 for water

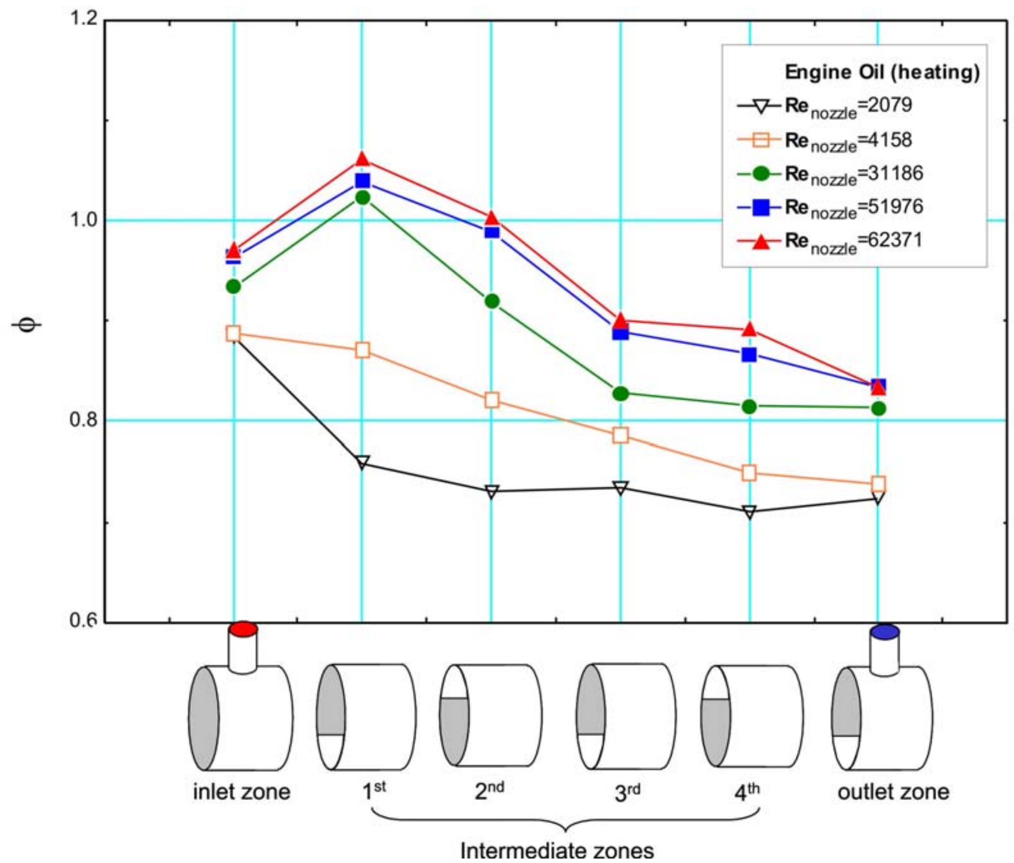


Figure 8-3: Local performance factor of HEX 1 for engine oil

Local behaviour of HEX 2 The study for HEX 2 was performed without considering leakage flows. Figures 9-1 to 9-3 show the simulation results for the performance factor at different baffle spacing zones for the investigated – in case of HEX 2 – two shell fluids (Pr=0.7, Pr=1.6). The Reynolds number at the inlet nozzle $Re_{nozzle} = \rho_{inlet} \cdot u_{inlet} \cdot D_{nozzle} / \mu_{inlet}$ (D_{nozzle} see Table 2) is used as parameter and covers the range $10,000 \leq Re_{nozzle} \leq 305,000$.

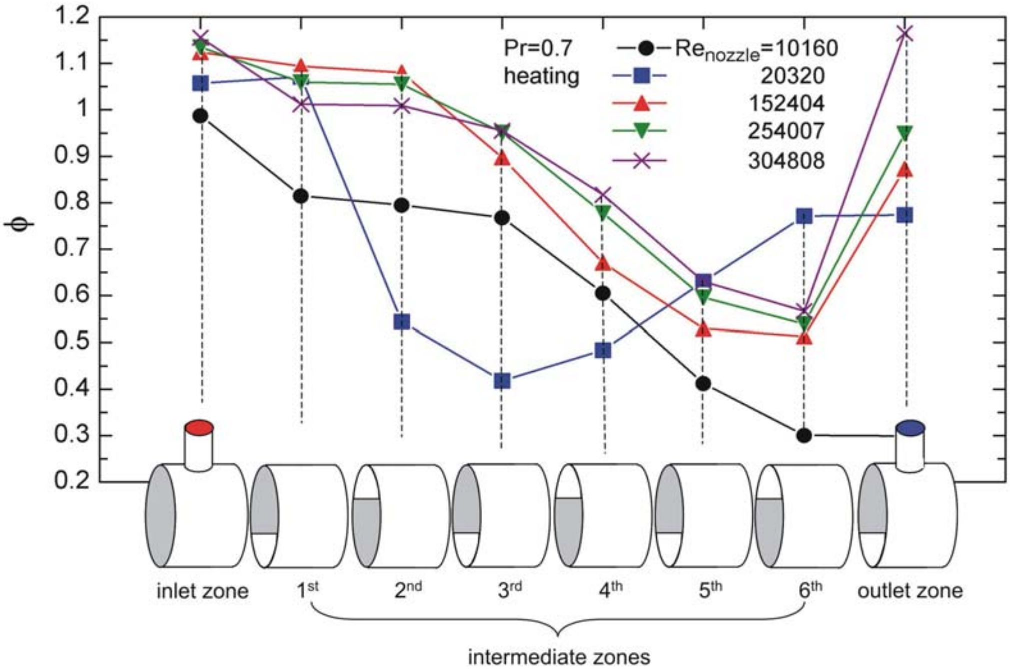


Figure 9-1. Local performance factor at each baffle spacing zone of HEX 2 for air (Pr=0.7)

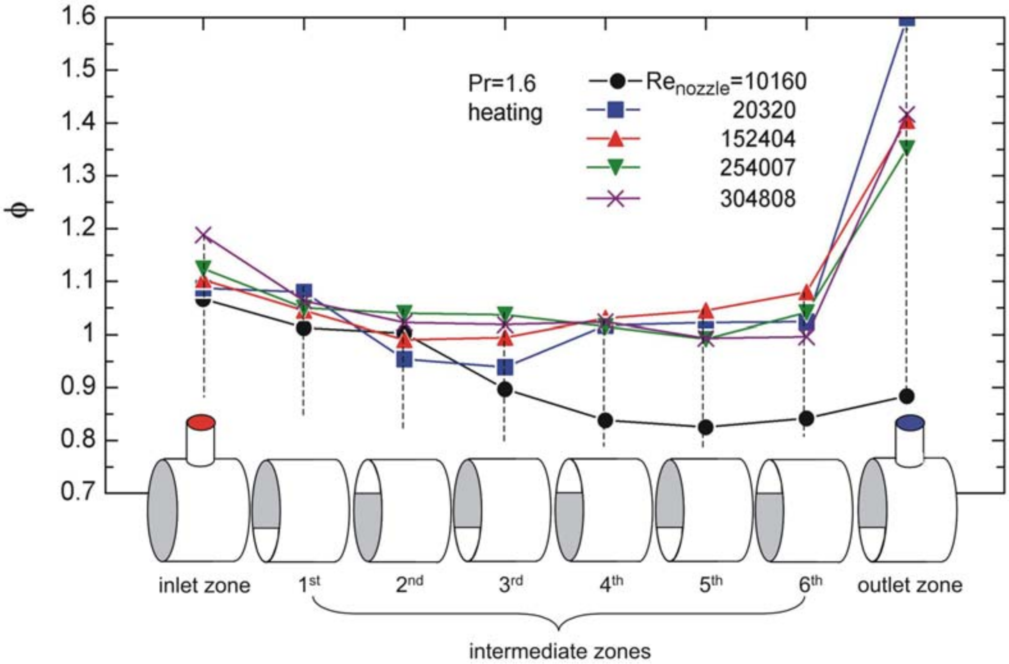


Figure 9-2. Local performance factor at each baffle spacing zone of HEX 2 for water (Pr=1.6)

The simulation results for the performance factor show values which are significantly higher than one in the inlet zone followed by a decay of the performance factor when reaching and passing the intermediate baffle spacing zones further downstream. The decay of the performance factor becomes greater with decreasing Prandtl number. At the outlet zone a remarkable increase of the performance factor is recognizable.

Comparative evaluation of simulation results with and without leakage flows A comparison of the simulation results achieved with (Figures 8-1 to 8-3) and without (Figures 9-1 to 9-3) leakage flows shows that the model without leakage flows overestimates continuously the local performance factor compared to the results achieved with leakages. In single zones like the inlet zone, the first three intermediate baffle spacing zones and in particular in the outlet zone a performance factor Φ greater than one occurs which indicates the advantage of horizontal baffle orientation compared to the vertical baffle orientation.

In contrast to that the model with leakage flows predicts virtually always values for the performance factor lower than unity which shows that a heat exchanger with vertically orientated baffles is more desirable than one with horizontal baffle orientation. In particular the benefit of vertical baffle orientation over horizontal baffle orientation is more noticeable for gases.

Both models agree in the prediction of the qualitative behaviour of the heat exchanger in the intermediate baffle spacing zones far away from the end zones (inlet, outlet): The performance factor is constant or decreases. The latter is more noticeable for low Reynolds numbers.

In summary, it can be noted that the results of the numerical simulations with and without leakage flows present different quantitative behaviour of the investigated heat exchangers. In the following section a brief explanation of the different behaviour of the heat exchangers is given. In particular the flow behaviour at the end zones (inlet, outlet) is discussed, since these zones showed the biggest differences for the performance factor if calculated with leakage and without leakage flows, respectively.

Flow behaviour without leakage flows A look at the geometrical situation may be helpful for the following discussions.

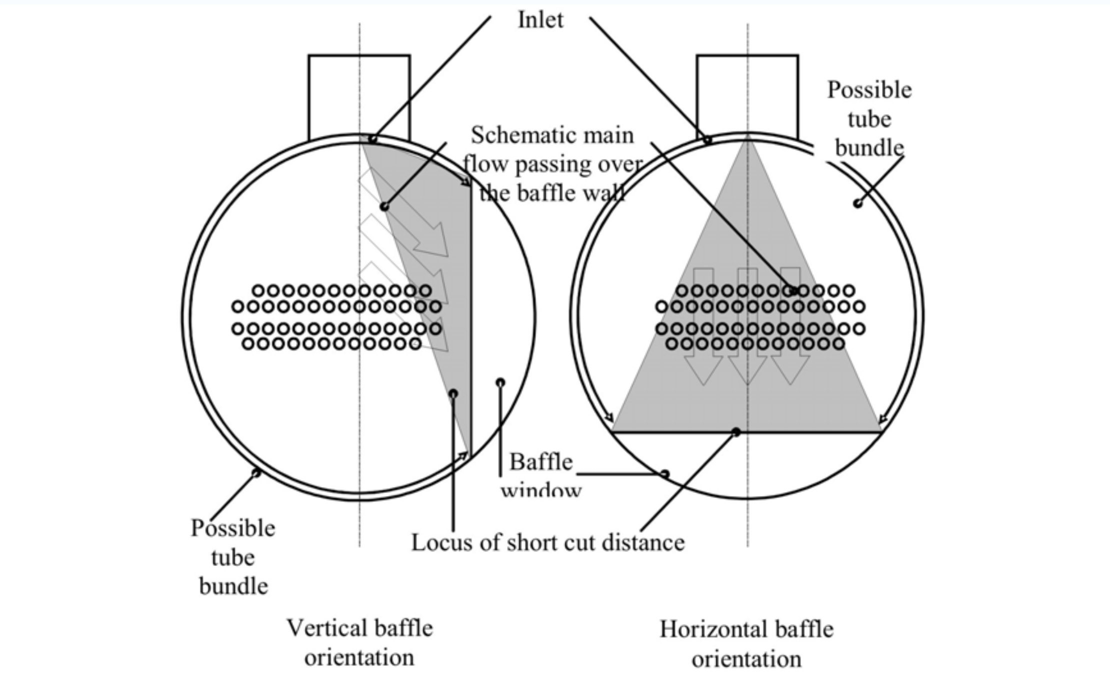


Figure 10. Schematic illustration of geometrical and fluidic aspects of the shell-side fluid in the inlet

nozzle zone for vertical and horizontal baffle orientation.

The average short cut distance between the inlet nozzle and the first baffle window - as well as the last baffle window and the outlet nozzle - is longer in a shell and tube heat exchanger with horizontal baffle orientation compared to the same one with vertical baffle orientation as depicted schematically in Figure 10.

For a heat exchanger with horizontal baffle orientation the velocity profile at the first baffle window is symmetrical. Contrariwise, a heat exchanger with vertical baffle orientation shows an irregular velocity distribution which increases the residence time of the shell fluid and with it the heat transfer. Therefore, the performance factor decreases. At the outlet zone of the heat exchanger the average distance from the baffle window to the outlet nozzle is larger for horizontally orientated baffles than for vertical baffle orientation (cf. Figure 10), consequently the performance factor has to ascend.

Flow behaviour with leakage flows The baffled shell-side flow can not be adequately expressed by simple approaches due to its complexity. In fact a stream analysis has to be carried out. Only parts of the fluids take the desired path along the tube bundle, whereas potentially a substantial portion flows through the leakage and the bypass area between the tube bundle and the shell wall. Individual streams originally introduced by Tinker [1958] and extended by Palen and Taborek [1969] have to be distinguished (cf. Figure 11): Stream A is the leakage stream in the orifice formed by the clearance between the baffle hole and tube wall, stream B describes the main effective cross-flow stream, which can be related to flow across ideal tube banks, stream C is the tube bundle bypass stream in the gap between the bundle and the shell wall, stream E presents the leakage stream between the baffle edge and the shell wall. The pressure drop of the cross-flow stream B acts as a driving force for the other streams, forcing parts of the flow through the leakage and bypass clearances. Thus, the pressure drop distribution developed by stream B in the region of the baffles plays an important role in the explanation of the effect of leakages on the performance of shell-and-tube heat exchangers with vertical and horizontal baffle orientations. For the following discussion numerical results are used which were obtained for HEX 1 using water as shell-side fluid (Pr=1.6).

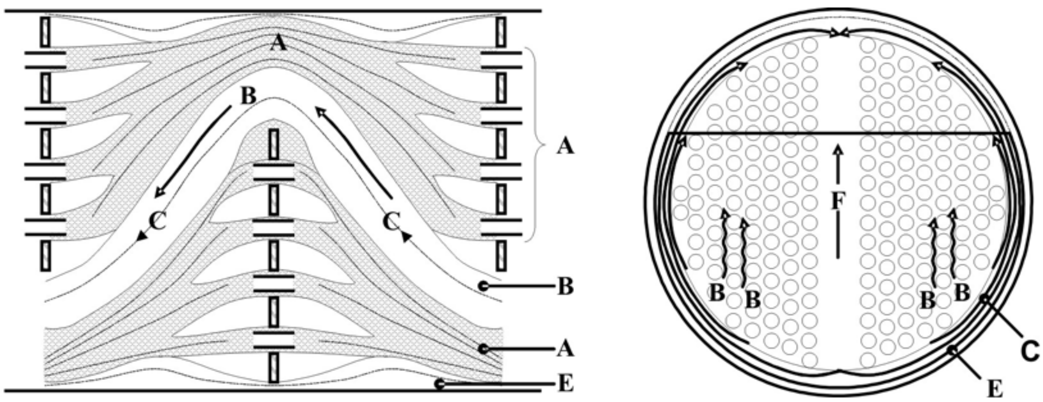


Figure 11. Flow path of streams through the shell of a cross-baffled shell and tube heat exchanger.

Figure 12 illustrates the ratio of the shell-side heat transfer coefficient $h_{horizontal}/h_{vertical}$ at each baffle spacing as a function of Reynolds number Re_{nozzle}. It can be seen that a shell and tube heat exchanger with horizontal baffle orientation has predominately a greater shell-side heat transfer coefficient than a shell and tube heat exchanger with vertical baffle orientation in the region of high Reynolds numbers ($Re_{nozzle} > 4.9\times10^4$). In general the values of $h_{horizontal}/h_{vertical}$ increase with increasing Reynolds number. The advantage of horizontal baffle orientation in heat transfer compared to vertical baffle orientation decreases along the heat exchanger such that the ratio of $h_{horizontal}/h_{vertical}$ reaches a value less than one for low Reynolds numbers. The unfavorable effect of tube bundle bypass and tube baffle leakage streams on the heat transfer is responsible for the fact that the mean ratio $h_{horizontal}/h_{vertical}$ is smaller than one for low Reynolds numbers ($Re_{nozzle} < 7\times10^3$) and only about 20% greater than one for high Reynolds numbers. The descending trend of $h_{horizontal}/h_{vertical}$ along the heat exchanger is caused by the influence of the main effective cross-flow stream B on the other flow streams.

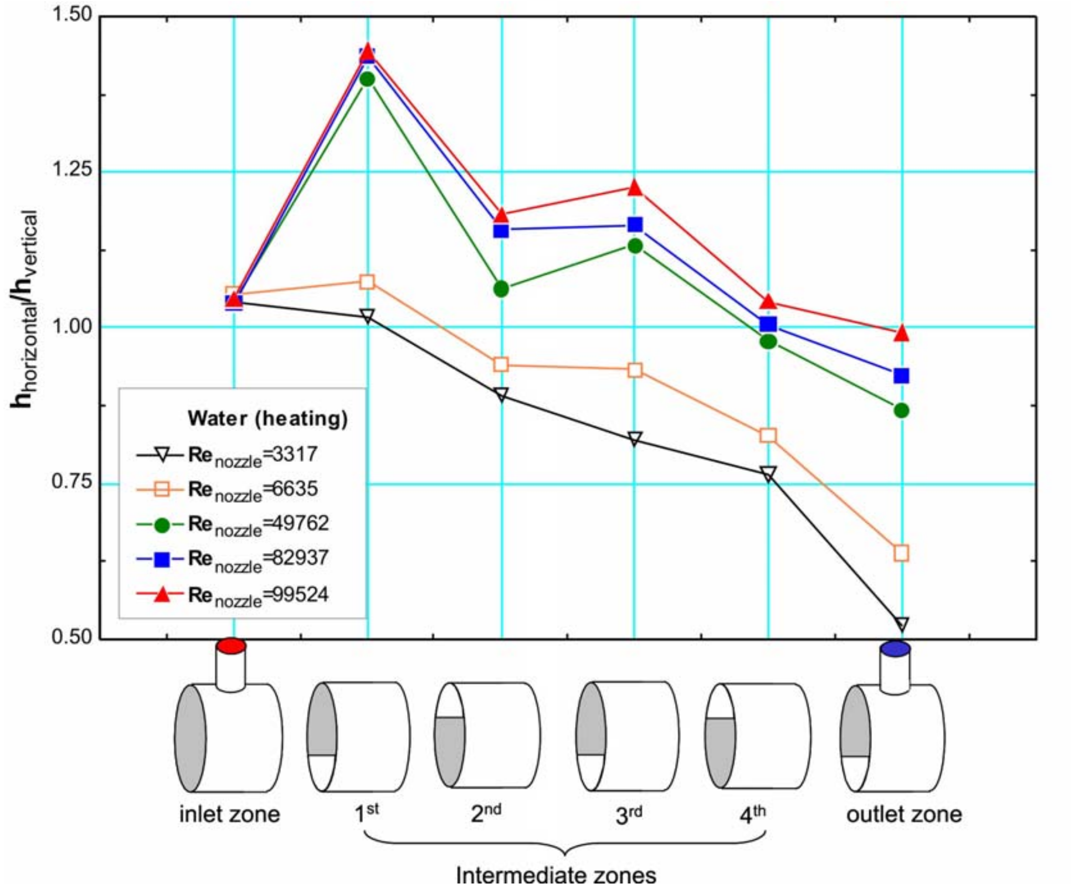


Figure 12. Ratio $h_{horizontal}/h_{vertical}$ at each baffle spacing zone

As already mentioned the average short cut distance between the inlet / outlet and the first /last baffle window is longer in a shell and tube heat exchanger with horizontal baffle orientation compared to the same one with vertical baffle orientation (cf. Figure 10). Hence the portion of fluid moving along the baffle wall in horizontal baffle orientation is greater than the portion of fluid passing along the baffle wall in vertical baffle orientation. This increases the possibility of an increased flow of the shell-side fluid through the tube-baffle clearances (bypass stream E) in the heat exchanger with horizontal baffle orientation compared to configurations with vertical baffle orientation. As a result the remaining main effective cross flow stream B and with it the heat transfer coefficient in the baffle spacing zones is lower for the heat exchanger with horizontally oriented baffles compared to the heat exchanger with vertical baffle orientation.

Figure 13 shows the ratio of the pressure drop for the horizontal baffle orientation to the pressure drop for the vertical baffle orientation $\Delta p_{horizontal}/\Delta p_{vertical}$ for the baffle zones as a function of the Reynolds number. The effect of tube-baffle and baffle-shell leakages on the pressure drop is the main source that increases the overall pressure drop in horizontally baffled heat exchanger compared to the overall pressure drop in vertically baffled shell and tube heat exchangers. The interaction between the different streams decreases the value of the performance factor (cf. Figure 8-2).

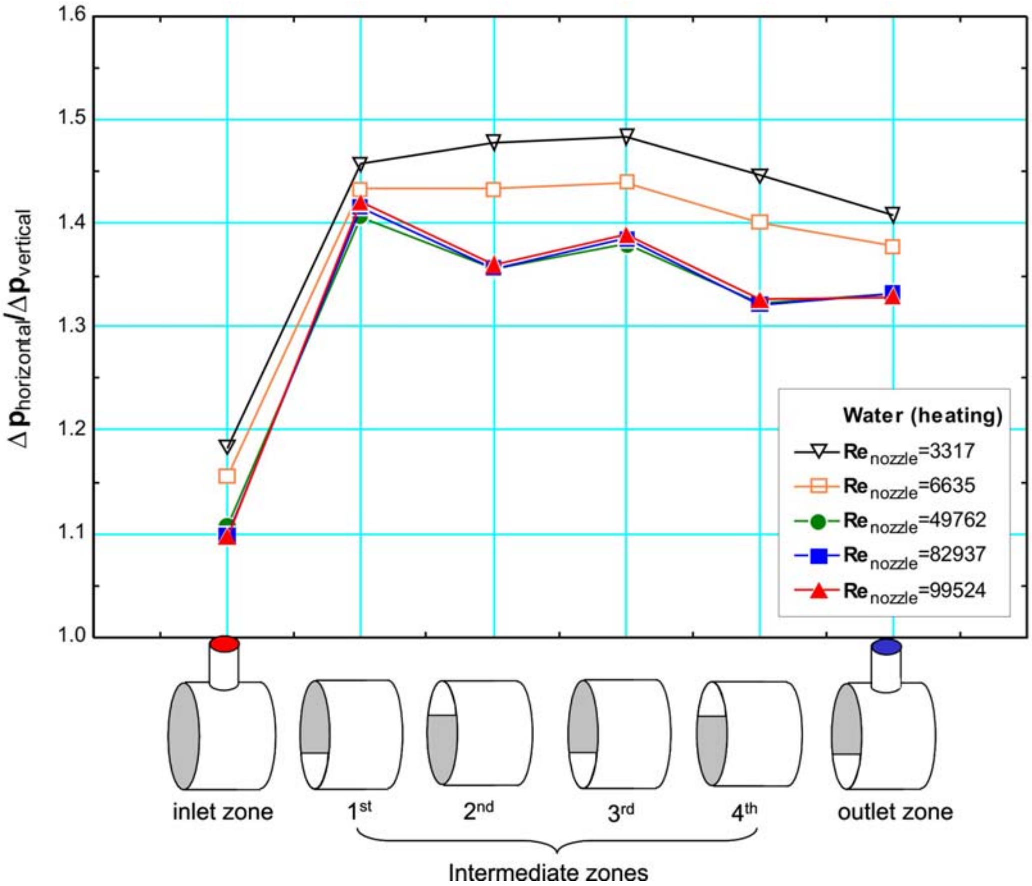


Figure 13. Local ratio $\Delta p_{horizontal}/\Delta p_{vertival}$ for baffle space

CONCLUSIONS

In the context of the present study, it could be shown that the orientation of baffles has a significant influence on the shell-side pressure drop and heat transfer of heat exchangers. By introducing a performance factor, the effects of horizontally and vertically orientated baffles on pressure drop and heat transfer could be compared and assessed. The tube-baffle leakages and bypass streams play an important role in the explanation of the performance factor of segmentally baffled shell and tube heat exchangers. The comparison of calculation results with and without leakage flows presents different behaviour especially at the end zones of the heat exchanger and underlines the importance of a consideration of tube-baffle leakage and bypass streams for the prediction of the performance factor of technical heat exchanger. For all shell-side fluids (air, water, engine oil), which have been considered in a heat exchanger with leakage flows, the vertical baffle orientation seems to be more advantageous than the horizontal orientation. It was found, that the horizontal baffle orientation produces up to 250% higher pressure drop compared to the pressure drop in vertical baffle orientation. The heat transfer coefficient is up to 20% higher than the heat transfer coefficient for vertical orientation. The local and overall behaviour of the performance factor for liquids as shell-side fluid is comparable and reaches a value of about 0.85 at high Reynolds numbers. With air as shell-side fluid a value of about 0.3 for the overall performance factor was obtained at high Reynolds numbers. The benefit of vertical baffle orientation over horizontal baffle orientation is more noticeable for gases since the dissipation rate in gases is much higher than in *liquids*.

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KEYWORDS

Shell and tube heat exchanger, baffle orientation, numerical investigation, CFD, shell-side, performance factor