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hiTRAN technology in Single and Two-Phase Flow applications

The Use of hiTRAN Wire Matrix Elements to Improve the Thermal Efficiency of Tubular Heat Exchangers in Single and Two-Phase Flow

Peter Drögemüller

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Over the past decades, innovations in the design of heat exchangers have improved thermal process efficiency. hiTRAN wire matrix technology is applied on the tube side of tubular heat exchangers and has been successfully used in a wide range of application, ranging from sensible heating and cooling to condensing and boiling services. The technology is used for both new designs and in revamp situations where it can increase the effectiveness of existing equipment. This review gives the required information to determine the process conditions for the most efficient use of this technology.

Keywords: Heat transfer enhancement, hiTRAN, Process intensification, Turbulator

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1 Introduction

Increasing energy prices, more stringent emission targets, and regional rising temperature levels are all examples of variables which increase the need for efficient process design in order to reduce plant operating costs. Software tools like process simulators and pinch analysis are used to optimize for maximum throughput and heat recovery. In order to implement the results of such optimization, the selection of suitable heat transfer hardware is important. This has led to an increased use of heat transfer augmentation techniques in industry. Various heat transfer enhancement options for the design of tubular heat exchangers are available. These range from extended and modified tube surfaces, modified baffle design to different types of tube internals such as hiTRAN technology.

Flow conditions inside a tube determine the mechanism of heat transfer from the tube wall to the bulk of the fluid flow. hiTRAN technology fundamentally changes the flow pattern. The changes affect single and two-phase flow in different ways. Starting with single-phase flow, these changes and the impact on heat transfer are explained in this paper.

The technology was developed by Cal Gavin LTD and is backed up by more than 30 years of research and several thousand applications in industry. The hiTRAN inserts consist of a core wire with loops attached at equidistant points as seen in Fig. 1. The insert is manufactured so as to be slightly larger than the inside diameter of the tube. The springiness of the wire ensures that the loops are touching the wall. The number of loops per unit length can be varied continuously in order to coincide with the allowable pressure drop. In addition, other geometrical parameters such as wire size and loop angle are varied to achieve optimum performance.

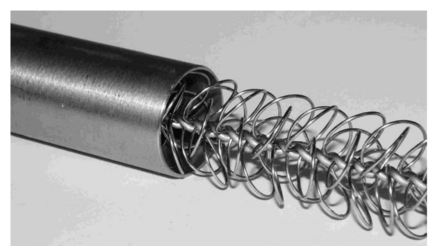

Figure 1. hiTRAN wire matrix element being inserted in tube.

Dr. Peter Drögemüller (peter.droegemueller@calgavin.com), Cal Gavin LTD – R&D, Minerva Mill Innovation Centre, Station Road, Alcester, B495ET, United Kingdom.

2 hiTRAN in Single-Phase Flow

2.1 Hydrodynamic in Adiabatic Pipe Flow with hiTRAN

Under isothermal conditions tube side flow can be characterized as either laminar or turbulent flow. In laminar flow there is no radial fluid mixing between the fluid layers. After a hydrodynamic entrance length, a characteristic Poiseuille velocity profile develops (Fig. 2, dotted line), due to the viscous drag at the tube wall. The parabolic shaped velocity profile can be derived analytically from the Navier-Stokes equation. Even for increased velocities the dimensionless laminar velocity profile does not change and the velocity gradient du/dy at the wall remains constant. According to

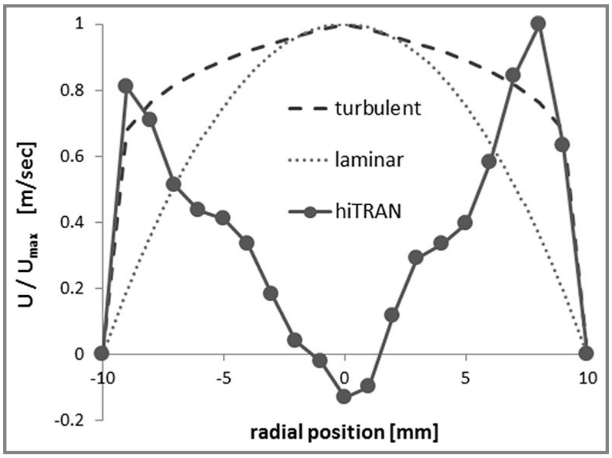

Figure 2. Normalized velocity profile for turbulent (Re ~ 20000) and laminar flow and measured values for hiTRAN at Re = 500.

the original work of Reynolds [1], laminar flow in a pipe becomes unstable if the Reynolds number exceeds a certain critical value. Below this critical value all disturbances will decay and laminar flow will be maintained. In his experiments he found a value of Re_{cr} = 2260. Recent studies by Avila [2] suggest a value of Re_{cr} = 2040. In general, studies by various different researchers have placed the value between $1760 < Re_{cr} < 2300$ for highly disturbed entrance regions [3].

After transition to turbulence, the flow shows local random velocity and pressure fluctuations. Due to these fluctuations, the mixing between the fluid layers is much greater. As a result, the velocity distribution is more uniform in turbulent pipe flow compared to laminar flow. The velocity profile is much steeper compared to laminar flow conditions. The shape can be approximated with the power-law-velocity profile first experimentally derived by Nikuradse [4] (Fig. 2, dashed line). The velocity gradient at the wall increases with increasing Reynolds number or flow velocity causing reduction in boundary layer thickness. This in turn improves heat transfer and generates higher wall shear stress which is beneficial for fouling reduction.

When using hiTRAN wire matrix inserts these conditions change substantially. Smeethe [5] showed in experimental measurements the differences between empty pipe flows and flows influenced by hiTRAN Elements. He used laser doppler velocimetry (LDV) and particle image velocimetry (PIV) technology to measure velocity profiles and entire velocity vector fields.

Both measurement techniques show a much steeper velocity gradient near to the wall. They also indicate that the entrance length required to develop the velocity profile is much shorter compared to tubes without internals. Measurements backed up by CFD simulations show entrance lengths similar to those in turbulent flow conditions. For heat exchanger design the hydrodynamic entrance effect can therefore be neglected. In Fig. 2 it is shown that for a laminar flow condition of Re = 500 velocity gradients are measured which are similar to gradients encountered in fully developed turbulent flow. Over the cross section of the tube two velocity maxima are encountered. Since the center of the tube is occupied by the core wire, the maximum flow velocity is shifted towards the tube wall, resulting in steeper near-wall velocity gradients. Influenced by packing density and Reynolds numbers up to seven times higher velocity gradients were measured, when applying hiTRAN in laminar flow [6]. This can be translated directly into increased wall shear stress, which is calculated as follows:

$$\tau_W = \mu \frac{du}{dy} \qquad (1)$$

From Eq. (1) it can be seen that steeper near-wall velocity gradients induce higher wall shear forces. In fouling application where the removal rate is increased by higher wall shear forces, tube internals can therefore play an important role to reduce fouling [7 – 10].

In cases where the tube side fluid shows shear-thinning, pseudoplastic behavior, higher shear rates at the wall, and also at the insert wires are beneficial to improve the fluid movement. Oliver [11] investigated the potential for hiTRAN under these conditions. His measurements suggest benefits in terms of reduced pressure drop when compared to the use in Newtonian flow behavior. The PIV velocity field measurements undertaken by Smeethe [5] also suggest a velocity component from the wall towards the centerline of the tube.

Those findings are backed up by dye stream experiments. Colored ink with similar properties compared to the bulk flow is injected into the flow near to the tube wall (Fig. 3). In the area of plain empty tube flow (Fig. 3a) the ink is not mixed with the bulk flow and remains at the tube wall. Once the dye hits the first loop of the insert wire, the color fluid is deflected towards the bulk flow (Fig. 3b) indicating a change in flow direction on a macro scale.

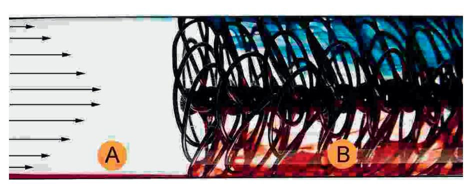

Figure 3. Red and blue ink injected near the tube wall in laminar flow (A) and mixed (B).

This radial fluid mixing improves the rate of mass and heat transfer from the tube wall and between the fluid layers. It also shortens the fluid residence time near to the wall, reducing the time the fluid is exposed to high or low wall temperatures. This benefits applications where fouling and product degradation are caused by excessive wall temperatures. The change in flow hydrodynamics also has an impact on the overall residence time distribution by changing laminar flow conditions to plug flow behavior as demonstrated later. These are the conditions for isothermal flows; an additional layer of complexity is added when investigating the flow involving heat transfer.

2.2 Heat Transfer Characteristics in Viscous Flow

When considering flows involving heat transfer one has to differentiate between turbulent, transitional and laminar flow conditions. In turbulent flow forced convection is the dominant factor and the heat transfer can be determined as a function of the Reynolds and Prandtl number. In heat exchanger design transitional and laminar flow are usually only encountered in cases where turbulent flow is not achievable, e.g., with high fluid viscosities or at very small tube geometries. Under these conditions it is very important to predict the heat transfer coefficient accurately because a low value may become the controlling resistance in the design. Calculations are more complex than under turbulent conditions. For short tube lengths entrance effects to develop the velocity and temperature profiles have to be taken into account. A distinction should also be made between heat flux boundary (uniform heat flux, UHF) and temperature boundary conditions (uniform wall temperature, UWT). The wall correction factor which in general accounts for radial variations of fluid viscosity with temperature can be very significant in laminar flow. It is common to use the viscosity correction of Sieder and Tate [12] in laminar flow

$$\frac{Nu_m}{Nu_c} = \left(\frac{\mu_b}{\mu_W}\right)^{0.14} \tag{2}$$

or as recommended in the VDI-Wärmeatlas, an expression taking into account the change in Prandtl number between bulk and wall flow [13].

In Eq. (2) there is no difference of the exponent between heating and cooling. Due to the poor heat transfer between wall and bulk flow, the differences between wall and bulk temperature and therefore also the differences between the corresponding viscosities can be significantly higher compared to turbulent flow. Corrections of 20 % to 50 % are common for viscous laminar applications.

Furthermore, in laminar flow the heat transfer can be dominated by either forced or mixed convection. In mixed convection flow conditions, a secondary flow profile caused by density differences is superimposed on the forced velocity profile in the flow direction. The dominant mechanism depends on the conditions and physical properties of the fluid being heated or cooled. Oliver [14] points out that it is important to realize that mixed convection heat transfer can be significantly different and its magnitude several times higher compared to heat transfer rates in pure forced convection. The effect was first quantitatively examined by Colburn [15]. In order to determine when buoyancy effects do have to be taken into account, different flow maps have been proposed by Ghajar [16] to identify the dominant effect based on dimensionless numbers. The most commonly used is the map from Matais and Eckert [17] they proposed to differentiate between forced and mixed convection when the mixed convection heat transfer deviates by more than 10 % from the pure forced convection heat transfer. In Fig. 4 extracts of the flow map are shown and the dotted line represents the separation between forced and mixed convection. The diamond markers show experimental tube side heat transfer results (see also Fig. 6) with increasing Reynolds numbers where the heat transfer mechanism shifts from mixed convection to forced convection (A). In addition typical cross sectional flow pattern calculated with computational fluid dynamic (CFD) for the experimental conditions are displayed.

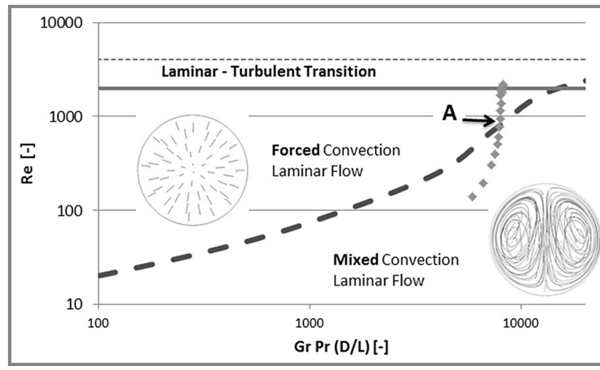

Figure 4. Section of horizontal laminar flow map [17], with border between mixed and forced convection.

Results of calculations with CFD, when verified with experimental data, have been very useful in describing the underlying complex flow patterns [18]. Simulations show that the change between the flow regimes is gradual. Evaluation of typical industrial Prandtl and Grasshoff number ranges under laminar conditions indicates that the majority of applications operate in the mixed convection regime.

Under these conditions, natural convection superimposed on the main flow causes a rise of less dense fluid towards the upper tube region, whereas the more dense fluid accumulates at the bottom of the tube. The direction of movement within the tube depends on whether the tube is heated or cooled. For a mixed convection example case in Fig. 5, CFD is employed to investigate the flow behavior of heat transfer oil entering a 22 mm tube at 70 °C (Fig. 5b). The oil is cooled over a tube length of about 2.5 m with a uniform wall temperature of 7 °C. The flow condition is laminar with a Reynolds number of 250. Since the oil is cooled, it becomes denser at the tube wall compared to the bulk and therefore moves along the wall towards the bottom of the tube. This is shown in the cross sectional flow pattern in Fig. 5a. As a result, the temperature over the cross section of the tube becomes stratified (Fig. 5c), with the highest temperature at the top and the lowest towards the bottom of the tube.

Fig. 5b represents the condition at tube inlet before cooling and Fig. 5c the temperature distribution at tube outlet.

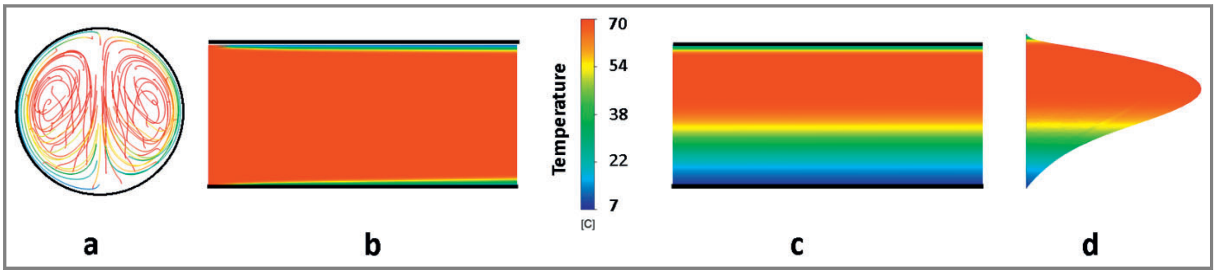

Figure 5. CFD simulation of flow in horizontal tube at Re = 250. a) Flow pattern; b) temperature distribution (inlet); c) temperature distribution (outlet); d) velocity profile outlet.

The color scheme represents different temperatures as indicated in the legend. The CFD calculated temperature results were compared to measurements and showed a maximum deviation of 0.5 % from the measured value. In this case, the measured mixed outlet temperature was 61.8 °C and the calculated value was 62 °C. The color indicates that the majority of the fluid passing through the upper part of the tube travels with hardly any change in temperature (red). Since in general the viscosity of liquids increases at lower temperatures, the velocity profile becomes asymmetric with the highest velocity towards the top of the tube and almost stagnant flow near to the bottom (Fig. 5d). As a consequence, parts of the fluid remain much longer in the tube, which impacts product degradation and fouling behavior. For example, fluids with tendency of wax formation at a certain temperatures will be affected adversely by the low heat transfer. In this example, a layer of fluid at the same temperature as the cooling fluid is formed at the bottom area of the tube. This indicates a loss of temperature driving force with very poor heat transfer in this area of the tube. Under these conditions it is very difficult to predict overall exchanger performance. In a vertical tube arrangement, where gravity forces act parallel to the flow velocity, buoyancy effects are much less pronounced. Heat transfer is dominated by forced convection and is in general lower compared to horizontal tube arrangements. Measured heat transfer rates in laminar flow are in general more than one order of magnitude lower than turbulent flow. This is demonstrated in Fig. 6. There the measured dimensionless heat transfer coefficient is shown as a function of Reynolds number and compared with theoretical predictions from literature. The measurements were performed with heat transfer oil (Prandtl ~160) and the flow regimes cover laminar to the onset of turbulent flow conditions. Within the laminar regime, the heat transfer mechanism changes from mixed convection to forced (point A). This is evident when displaying the corresponding dimensionless numbers, seen as triangles, in the flow map Fig. 4.

As Bergles [19] points out, text book solutions for laminar heat transfer coefficients do in general only consider forced convection with entrance effects at constant fluid properties neglecting buoyancy effects. The VDI correlation [13] for laminar flow is such an example. This type of equation under predicts heat transfer rates in mixed convection laminar flow regimes considerably, in this example by more than 40 %. Once forced convection dominates from point A onwards, the prediction of laminar flow condition is accurate. Correlations which take into account buoyancy and

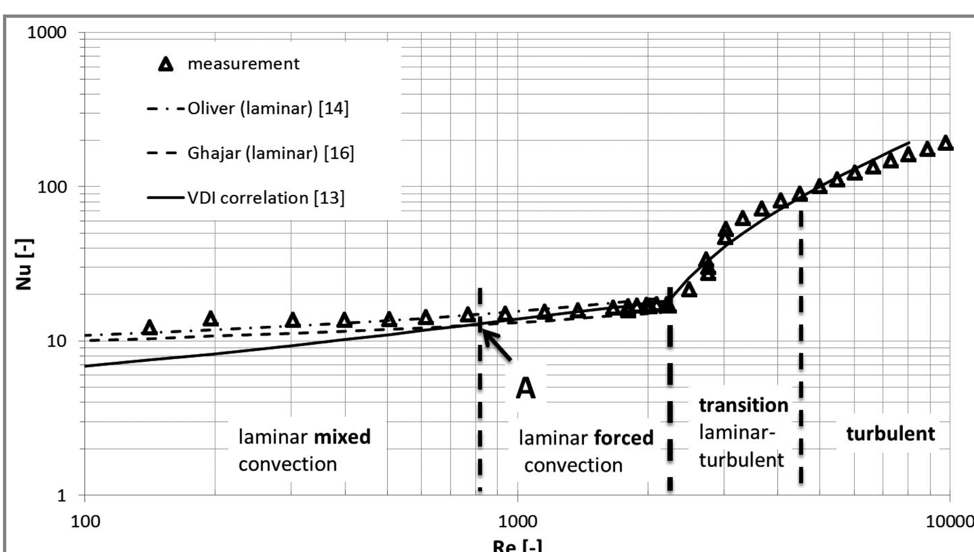

Figure 6. Plain empty tube side heat transfer measurements and comparison with theoretical data.

forced convection effects do predict the heat transfer in laminar flow reasonably well as demonstrated with the correlations from Oliver [14] and Ghajar [16].

In laminar flow the increase of heat transfer with Reynolds number is only modest; typically a doubling of the velocity gives just a 10 % to 20 % increase in the heat transfer rate. For that reason, using multiple tube passes to increase the heat transfer coefficient is much less effective than in turbulent flow where the heat transfer almost doubles with a doubling of the fluid velocity. It is also evident from the graph that once the transition to turbulent flow is reached, the increase in heat transfer is very sudden and steeper compared to the laminar and even fully developed turbulent region. In the region from ~2300 < ~Re < ~3000 the increase of heat transfer with Reynolds Number is about 10 times greater compared to fully turbulent flow, and therefore, extremely sensitive to a change in flow or physical properties. Calculation for heat transfer rates in the region is generally done by pro rata interpolation between turbulent and laminar conditions, which leads to uncertainties. The extent of transitional flow depends also on geometrical conditions and entrance effects. Small changes in process or property conditions when operating in this region can have a profound impact on heat exchanger performance.

In summary, tube side laminar and transitional flow can be characterized as follows:

- In general, very low heat transfer compared to turbulent flow conditions;
- Large number of influencing variables to determine heat transfer (Reynolds, Prandtl, Grasshof, entrance effects, tube orientation, etc.). Therefore, more uncertainty when designing heat exchangers;
- Possible flow stratifications due to buoyancy forces in mixed convection laminar flow conditions; long wall residence times at bottom of the tube under those conditions;
- Heat transfer depends on tube orientation;
- Wall correction effects are more pronounced compared to turbulent flow;
- Large variation of heat transfer with Reynolds when designing exchanger in transition flow region.

2.3 Enhanced Heat Transfer and Flow Distribution in hiTRAN Flow

As outlined, flow conditions for heat transfer and flow distributions are not ideal in laminar flow. The use of passive enhancement technology such as hiTRAN thermal systems can improve these conditions. Tube side heat transfer measurements carried out at Cal Gavin LTD, the manufacturer of hiTRAN Wire Matrix Inserts, show significant differences to the plain empty tube behavior. Typical curves are shown in Fig. 7 for hiTRAN Elements with low and high packing densities and compared with plain empty tube heat transfer rates. The dimensionless heat transfer is calculated here as Nusselt divided by Prandtl. Typically, the elements are designed in such a way that the entire allowable frictional tube side pressure drop for the exchanger design is used. This is done by varying the packing density of the elements. Since this geometry parameter can be varied continuously by variation of the amount of loops attached to the core wire, the induced heat transfer and pressure drop varies accordingly.

The dotted lines in Fig. 7 show some of various additional insert geometries possible. Single-phase heat transfer and pressure drop measurement results for more than 600 different insert geometries with respect to packing density, wire size, loop angle and tube diameter were correlated. These correlations are implemented for all insert variations in hiTRAN.SP, a free software tool available from Cal Gavin [20].

From the graph it is noted that the heat transfer increases continuously at a constant rate with increased flow velocity and Reynolds number. For the Reynolds numbers investigated, there is no appreciable difference between laminar and turbulent flow conditions. Uncertainty of the plain empty tube performance in the transition region is therefore removed. Compared to plain empty tube conditions, an increase of heat transfer rate of up to 16 times is possible. It can be seen that the enhancement levels achieved are greatest in laminar and transitional flow in the Reynolds range from 50 to 10 000. As a rule of thumb for the majority of

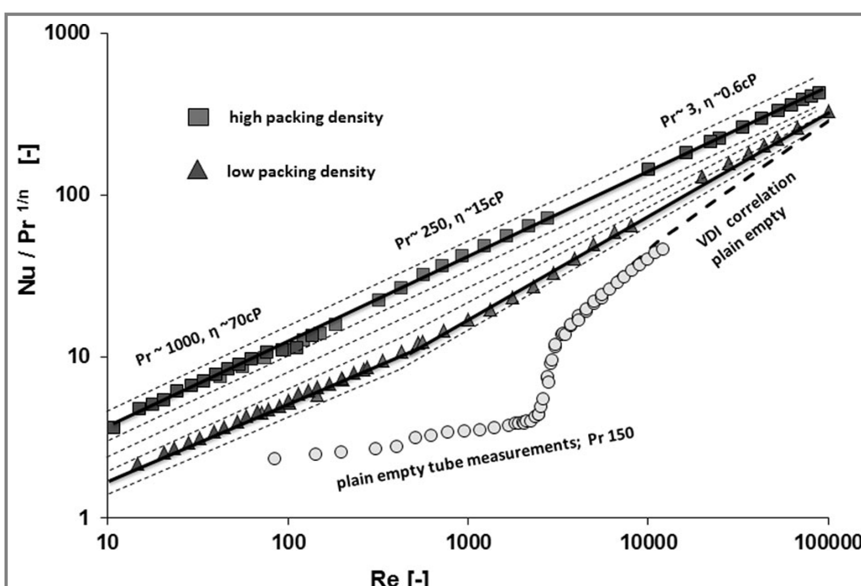

Figure 7. Dimensionless tube side heat transfer as function of Reynolds for plain empty tube and different hiTRAN packing densities.

industrial applications this equates to a viscosity range from about 2 cP to 200 cP.

Unlike to laminar flow, heat transfer in turbulent flow is a strong function of flow velocity. Therefore, the same levels of heat transfer enhancement under those flow conditions can often be achieved with higher flow velocities in a multi pass design without the need for inserts. For this reason, hiTRAN applications in turbulent flow are where a multi pass design is not possible, e.g., in situations of temperature cross. Under these conditions hiTRAN offers up to 5 times the plain empty tube heat transfer. At the lower end, hiTRAN offers benefit in terms of heat transfer for Reynolds numbers as low as 1.

The use of hiTRAN increases the associated pressure drop compared to the pressure drop for a plain empty tube at the same flow velocity. In order to compensate for this increase, hiTRAN enhanced exchangers are designed to operate at low flow velocities. This typically equates to a single or two-pass heat exchanger design maintaining up to 10 times tube side heat transfer at similar or even lower pressure drop compared to plain empty tube exchangers with the same heat transfer area.

This is demonstrated in Fig. 8 where the tube side heat transfer and pressure drop is shown for a plain empty and enhanced hiTRAN design as a function of flow velocity. The different markers represent additional tube passes in the exchanger design, from one to eight and the corresponding tube side velocities from 0.2 m s^{-1} (one pass exchanger) to 1.5 m s^{-1} (eight pass exchanger). In this example the optimum design for the plain tube exchanger yields an eight pass exchanger (Re ~1700) for the allowable pressure drop of 1 bar with a tube side heat transfer coefficient of 160 W m^{-2}K^{-1}. The tube side fluid is a viscous oil (μ = 18 cP). As outlined before, under laminar flow conditions an increase in flow velocity only adds a limited amount of heat transfer but considerably increases the pressure drop as shown in the graph. With hiTRAN installed, the exchanger has to be designed for a lower tube side velocity in order to operate with identical tube side pressure drop. From Fig. 8 it can be seen that a two pass hiTRAN design yields the same pressure drop with a much improved tube side heat transfer coefficient of about 1400 W m^{-2}K^{-1}. A one pass hiTRAN design would still increase the heat transfer coefficient more than 5-fold with only 1/5 of the plain empty tube eight-pass pressure drop.

In a tube side controlled scenario the duty of the enhanced heat exchanger would increase accordingly. Guidelines for plain empty tube heat exchanger design recommend a minimum flow velocity for different fluids; this is based on a minimum wall shear stress in order to reduce fouling. As shown in Fig. 2, hiTRAN technology can increase the wall shear considerably. In the present example, the hiTRAN induced wall shear for the two pass exchanger is calculated to be 5.6 Pa. This is similar to the wall-shear stress of the plain empty tube eight pass design.

As shown in Fig. 5, buoyancy effects in laminar flow can lead to flow stratifications; this has an impact on the predictability of heat transfer and also of the residence time in these stagnant areas. Again, CFD simulation was employed in order to investigate the impact of fluid mixing on fluid stratification with hiTRAN under heating and cooling conditions. The simulations were validated by comparing the calculated and measured outlet temperatures. In general, all simulation results were within ± 3 % of the measured value. For the same tube inlet and cooling conditions at the wall shown in Fig. 5 for the plain empty tube simulations/experiments were repeated with hiTRAN. The simulation results presented in Fig. 9 show a flow pattern over the cross section of the tube which is substantially different to the plain empty tube. The flow streamlines show that the fluid movement is captured within the loops of the insert (Fig. 9a). Due to the mixing action of the insert temperature, differences between adjacent fluid layers are much smaller. For that reason, the driving force for natural convection, which causes flow stratification, is diminished.

Under mixed convection flow conditions (see Fig. 4), even at single figure Reynolds numbers, no stratification of the flow is observed. The temperature distribution over the cross section of the tube at the tube outlet is almost uniform (Fig. 9 c) indicated as yellow. Due to the considerably higher heat transfer the measured/simulated mixed outlet temperature is much lower with 49.9 °C compared to 61.8 °C for the plain empty tube case. The velocity profile (Fig. 9d) shows several maxima over the plane which compares well with the LDV measurements presented in Fig. 2. The simulation indicates

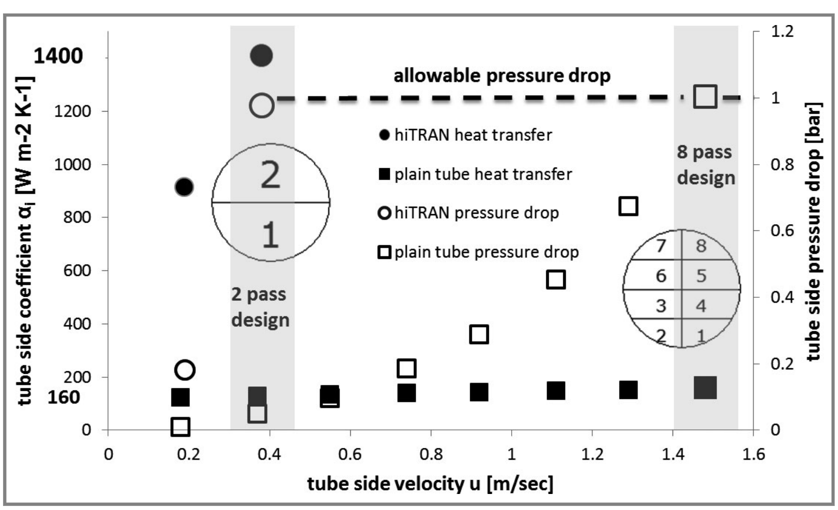

Figure 8. Heat transfer and pressure drop as function of tube velocity (pass arrangement) for a typical hiTRAN design.

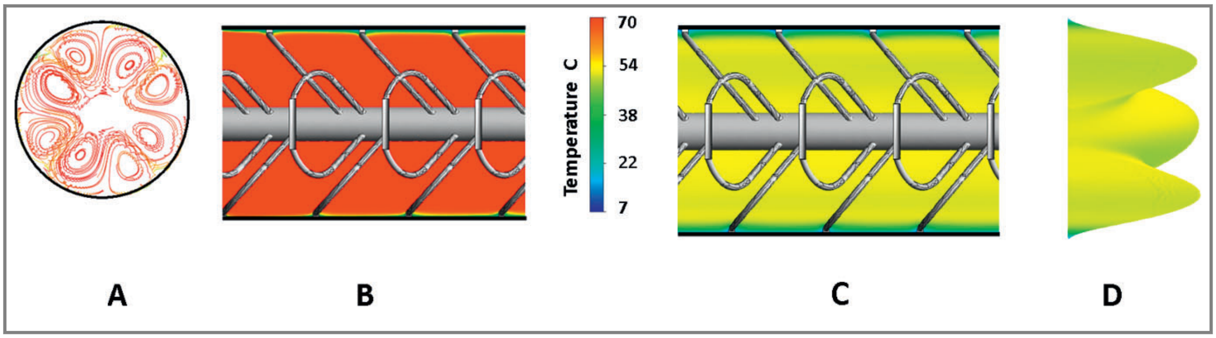

Figure 9. CFD simulation of flow in horizontal tube at Re = 250 with hiTRAN. a) Flow pattern; b) temperature distribution (inlet); c) temperature distribution (outlet); d) velocity profile outlet.

four velocity maxima distributed over the tube cross section with zero velocity in the center. No stagnant velocity zones towards the bottom of the tube are present when operating with hiTRAN. This also has implications on the residence time distribution which is much narrower than in case of the plain empty tube flow.

The residence time distribution calculated with CFD (Fig. 10) reflects the velocity profiles in Fig. 5d and Fig. 9d. It can be seen that the residence time distribution with inserts is much narrower compared to the plain empty tube distribution. This is beneficial for applications which are sensitive to long residence times at cooled or heated surfaces and might suffer degradation under such conditions.

Not only the fluid distribution within a single tube, but also the distribution in a tube bundle is influenced by the use of hiTRAN. Ellerby [21] has shown that the ratio between momentum pressure drop at the inlet nozzle and bundle frictional pressure determine the fluid distribution in the bundle. In cases where the pressure loss in the bundle accounts for less than 75 % of the total pressure drop, maldistribution is possible. Tube internals can be used in order to improve the fluid distribution by increasing the frictional pressure drop.

The advantages of using hiTRAN in single-phase flow applications can be summarized as follows:

- Up to 16 times the tube side heat transfer. This can be utilized in order to reduce the size in new design or to improve the exchanger duty in retrofit situations with existing equipment;
- Prevention of flow stratification;
- Shorter residence at the tube wall with impact on product quality and fouling behavior;
- No sudden change in heat transfer during the transition from laminar to turbulent flow. Improved exchanger behavior under partial load, start up and turn down conditions;
- All available tube side pressure drop is utilized, hiTRAN design meets pressure drop requirements;
- Better fluid distribution in the tube bundle.

3 hiTRAN in Two-Phase Applications

hiTRAN enhancement technology is also widely used in two-phase applications where the tube side heat transfer is limiting. In the following, an overview is given about the type of applications which can benefit from enhancement.

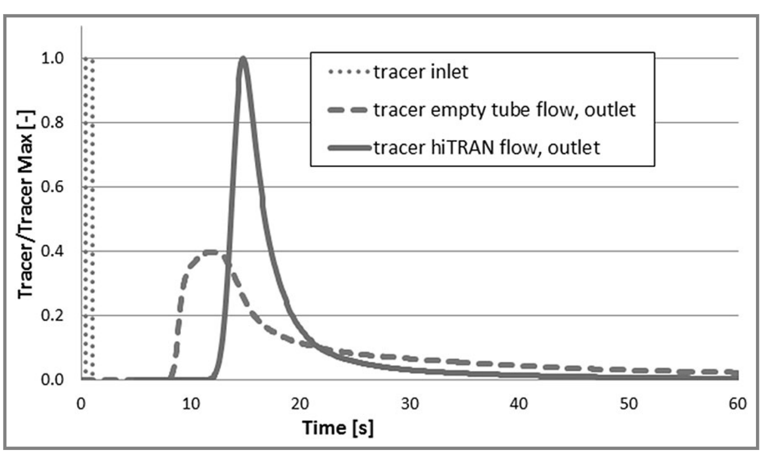

Figure 10. Calculated residence time distribution in plain empty tube and tube equipped with hiTRAN at Re = 250.

3.1 hiTRAN in Film Flow Applications

Heat transfer equipment is often designed in such a way that the liquid is flowing as gravity or shear controlled film towards the exit of the equipment. This is the case in falling film evaporators and also in vertical in tube condensing applications.

3.1.1 Falling Film Evaporators

Depending on the liquid load and the fluid viscosity, both the characteristics of the film and the heat transfer conditions change. The film characteristic can be described using the Film-Reynolds number:

$$\mathrm{Re_F} = \frac{4\,\Gamma}{\mu} \quad (3)$$

For Film-Reynolds numbers above 1600 one can expect transitional and turbulent film conditions [22]. These are characterized by good tube side heat transfer coefficient and short residence times in the tubes. For lower Film-Reynolds numbers the film becomes wavy laminar with surface waves flowing with higher velocity over a laminar film at the tube wall. In wavy laminar conditions, which typically represent film viscosities in the range from 2 to 200 cP, the heat transfer rate is much reduced compared to turbulent conditions. The thickness of the film also increases at higher viscosities. Therefore, there is little or no mixing of fluid in the film. When evaporating multicomponent mixtures, the more volatile component will evaporate first and the surface of the film will enrich in the less volatile components. If the film is not mixed, there will be a decline in the driving temperature difference between the wall and the saturation temperature of the liquid which will result in a lower evaporation rate.

Falling film evaporators are often used to evaporate temperature sensitive products. In general, the residence time is very short and the distribution narrow. When evaporating more viscous fluids, the residence time becomes longer with liquid hold up in the laminar sub layer near to the wall. Visual observations and residence time measurements indicate a change in flow behavior when applying hiTRAN under these conditions. In Fig. 11 it can be seen how a smooth liquid film becomes agitated after contacting the wires of the element.

Residence time experiments were undertaken with water-glycerol solutions under isothermal conditions. At the inlet of a vertical 2.5-m-long tube with an internal diameter of 50 mm, a highly concentrated NaCl-water-glycerol solution with a similar viscosity to the main fluid was injected with a syringe. The electrical response at the inlet and outlet was measured. The experiments were conducted under plain empty tube conditions and with inserts of different packing densities.

Under turbulent flow conditions the shapes of the inlet and outlet responses are similar, indicating a narrow residence time distribution. When measuring the residence time distribution under wavy laminar flow conditions the plain empty tube shows a long response time for the signal at the tube exit (Fig. 12). This indicates fast flowing surface waves with rapid response, followed by a long tail, indicating that part of the fluid is held back in the laminar sub layer near to the wall. Under these conditions the mixing within the film is limited. When hiTRAN is used under these flow conditions the distribution becomes more even, which is similar to the response under turbulent flow conditions. This indicates that surface waves penetrate the laminar sub layer. Refreshing the surface and mixing fluid with the laminar sub layer is important when evaporating as it evens out the concentration differences in the film when evaporating mixtures with different saturation temperatures. This leads to a larger driving temperature difference between the heating medium and the evaporating fluid.

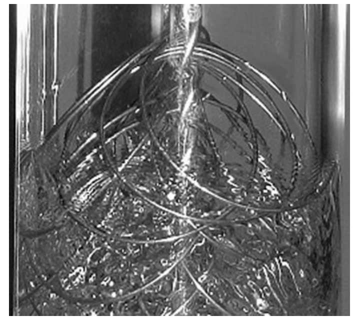

Figure 11. Induced turbulence in film flow when disturbed by the hiTRAN loop wires.

Falling film evaporators are sensitive to the liquid distribution over the circumference of the tube and in order to achieve the calculated performance, the distribution has to be uniform over the whole circumference. Measurements show that the wires of the insert redistribute the liquid in case of maldistribution at the tube entrance.

Heat transfer measurements were undertaken at a steam-heated single tube test facility at the University of Bremen in order to investigate the impact of changed hydrodynamics on heat transfer. The experiments were carried out with water and water-glycerol mixtures in order to cover an industrially relevant range of Film-Reynolds numbers. The results show an increase of tube side heat transfer with Film-Reynolds number.

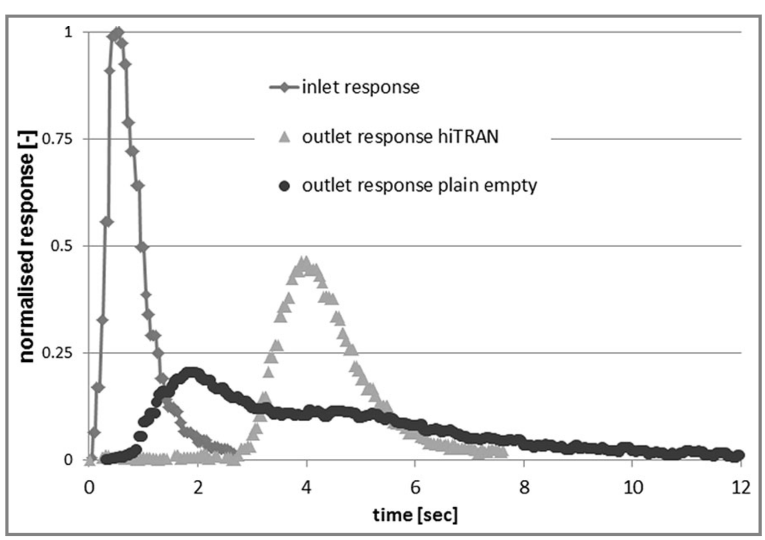

Figure 12. Measured residence time distribution with hiTRAN Insert and plain empty tube at $\mathrm{Re_{Film}} = \sim 700$ in wavy laminar flow.

For turbulent flow with water an increase in tube side performance of up to 30 % was reported. A more pronounced effect occurs when measuring the heat transfer coefficient of a more viscous water-glycerol film under wavy laminar and transitional flow conditions. Fig. 13 shows an up to 80 % increase in tube side film coefficients when applying hiTRAN enhancement under these conditions. This increased heat transfer often results in a reduced subcooled length at the tube entrance, leaving more surface area for the evaporation process.

Feedback from retrofit installations in industry confirms the improved tube side performance but more interestingly the benefits in terms of better product quality after evaporation, which can be associated with the improved residence time distribution.

3.1.2 Tube Side Condensers

Similar conditions to those in falling film evaporators can be found in vertical tube side condensers, where the condensing vapor forms a liquid film on the inside of the tube. The overall condensing heat transfer coefficient depends on the flow conditions of the film and vapor phase. In general, condensation coefficients are high and are not the controlling heat transfer resistance. This can often change when condensing multicomponent mixtures or if the condensation takes place with inert components. In these situations additional mass transport limitations between the liquid-vapor interfaces can have an important impact on the condensing process. The condensation temperature also reduces under these conditions along the condensation path, and therefore, the vapor has to be cooled so as to adapt to the changing condensing temperatures. The amount of vapor decreases along the condensation path, resulting in low vapor velocities with low Reynolds numbers and associated low heat transfer rates. The film thickness increases as more vapor is being condensed, with the thickest film towards the exit of the condenser. In laminar and laminar-wavy film flow the condensing heat transfer coefficient decreases as the film thickness increases.

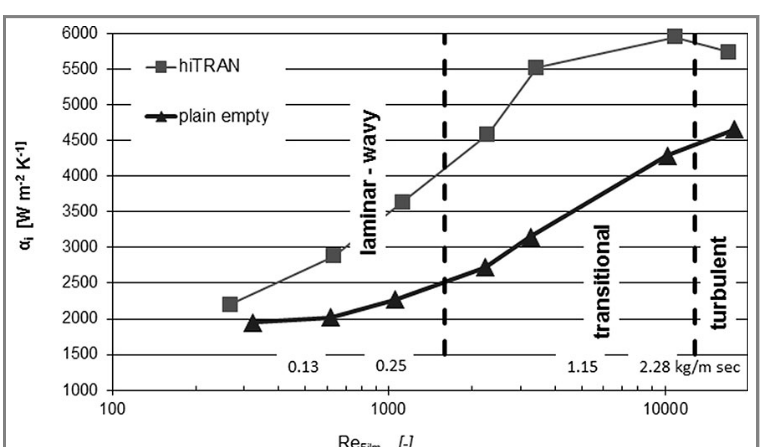

Figure 13. Measured tube side heat transfer in falling film evaporators with and without hiTRAN.

From these considerations it can be seen that the use of hiTRAN technology improves different aspects of the tube side condensation process. The wires produce additional turbulence in the condensate film similar as shown in Fig. 14. In condensation experiments conducted by Briggs [23], with refrigerants R-113 as single component liquid in turbulent flow, where the film resistance is the major component for the condensing coefficient, an increase of approximately 35 % in tube side coefficient is obtained compared to the plain empty tube conditions. Due to the inclined direction of the loop wire, part of the fluid is directed towards the center of the fluid.

Figure 14. hiTRAN alignment in film flow with flow deflection to the center of the tube.

This contributes to a reduction in film thickness but more importantly it breaks up the vapor-liquid interface and increases the mass transfer between the phases. Additional film and vapor mixing contribute to a more even concentration profile in the vapor and the condensing film. This therefore reduces the concentration of less volatile components at the liquid- vapor interface. The sensible cooling of the vapor often accounts for the main heat transfer resistance in the condensation process. It is therefore beneficial to apply hiTRAN in regions with low vapor velocities, here similar improvements of the vapor phase coefficients are possible as reported for single-phase heat transfer improvement.

Condensation applications are often very sensitive to changes in absolute pressure. Under identical vapor content and flow velocities tube internals increase the frictional pressure drop compared to the plain empty tube. Since the condensation rate is increased when using enhancement, more vapor is able to condense, hence, the remaining vapor velocity reduces. The pressure drop changes with the square of the velocity; therefore, this effect reduces the frictional pressure drop when using inserts. In order to keep the frictional pressure drop as low as possible, hiTRAN Elements are often installed towards the exit of the condenser tube only. In this area the potential for improvements is greatest since here the vapor velocities are low and the condensation film thickness is high. Depending on the process and property conditions, the overall condensation coefficient can be substantially increased when applying enhancement.

In the case of horizontal in tube condensation liquid accumulates at the bottom of the tube. In the entrance region this liquid layer is thin and high vapor velocities are present which agitate and move the film. Towards the end of the tube the film becomes very thick and can cover large sections of the tube. Since the film can be slow moving, it may often be of a laminar nature with poor heat transfer coefficient, almost forming an insulating layer between the cooling medium and the condensing vapor. Apart from the aspects discussed earlier, in those scenarios hiTRAN improves the liquid film heat transfer coefficient considerably. It can also be observed that with sufficient vapor flows part of the fluid is redistributed by the wire structure towards the top of the horizontal tube, thus, improving mixing between vapor and liquid phase. Again it is often advised to install only part of the tube towards the exit with enhancement.

When condensing often further sub cooling is required. If this is done in the same equipment as the condensation itself, low flow velocities are encountered. Under these circumstances hiTRAN enhancement offers possibilities to improve heat transfer rates in such services.

3.2 hiTRAN in Vaporizers

The heat transfer of tube side boiling, therefore also the opportunity of enhancement, is very much a function of the observed flow pattern and boiling mechanism. A typical curve is shown in Fig. 15, where the heat transfer is shown qualitatively over the tube length for a fluid which is fully evaporated and super-heated.

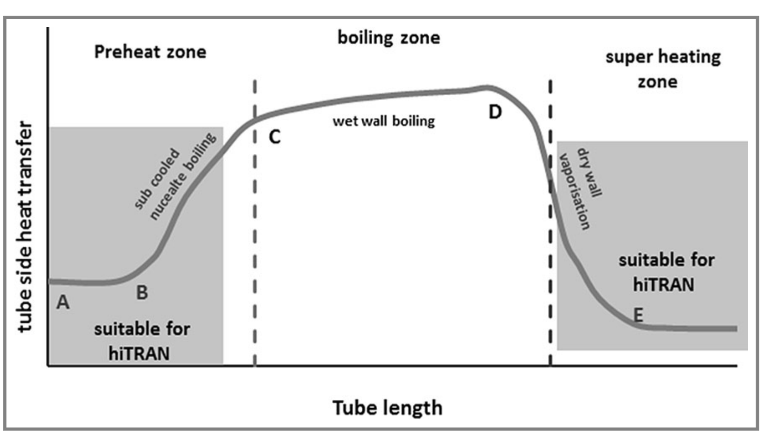

Figure 15. Qualitative tube side heat transfer as a function of tube length in a vaporizer.

In general, the liquid enters the vaporizer sub-cooled below saturation temperature; therefore, it has to be preheated to its saturation temperature. After single-phase heating (A–B) without nucleation, the first nucleation sites appear at the tube wall, where the temperature difference is sufficient. This preheat region is characterized by relatively poor heat transfer coefficients. Once the fluid is heated up to saturation temperature, so called wet wall boiling starts (C). Here, boiling mechanisms like nucleate and convective boiling can be observed. The flow patterns in this area of two-phase flow change as the quality changes and they also depend on the orientation of the tube. In this zone the fluid is in contact with the wall and is highly agitated and in general the heat transfer is high. When the vapor content increases further, there is a point when wall dry out begins (D). The supply of liquid to the wall by impingement of drops is less than the rate of evaporation. In a vertical tube this transition is sudden, whereas in a horizontal tube the change to dry wall starts at the top of the tube and proceeds gradually to the bottom. A vaporizer should be designed in a way that dry-out only occurs at high vapor qualities when it is not preventable. In the situation where the heat flux exceeds a critical value, dry wall conditions may occur at any vapor quality. In these scenarios, a vapor layer is formed between the liquid bulk flow and the wall; this can also be described as inverted annular flow.

Under these conditions heat transfer rates may be reduced by an order of magnitude. At point (E) all the liquid is evaporated and heat transfer is identical to single-phase vapour heat transfer.

3.2.1 Shortened Preheat Zone with hiTRAN

Vertical forced and thermo siphon reboilers with boiling on the tube side should be designed to operate under wet wall conditions. Recirculation rates are designed high enough in order to provide good tube side heat transfer, ratios greater than 4 are recommended by Kern [24]. According to Arneth [25] under those conditions the vapor content towards the exit of the exchanger is low, in general not exceeding 20 weight percent, which prevents dry out conditions. Due to the static head at the inlet of a reboiler, the liquid enters the reboiler below the saturation temperature and needs to be preheated to the boiling temperature (C in Fig. 15). The heat transfer in the zone before the onset of nucleate boiling is calculated from the corresponding single-phase correlations. Since vertical reboilers are designed as single pass exchangers, the flow velocity in the preheat section is often low, yielding poor heat transfer rates (Fig. 15). The recirculation is mainly driven by the density change in the boiling zone, therefore thermosiphon reboilers with large preheat length suffer from poor recirculation rates. For these reasons the aim is to design the reboiler in a way to keep this length as short as possible. This "subcooled length" depends on the tube side process and property conditions. Under atmospheric pressure the length of this zone is typically 20 – 50 % of the entire tube length. A large subcooled length of the total tube length can be found when processing viscous liquids due to the poor single-phase heat transfer coefficient and also may occur when operating under vacuum conditions. At high vacuum services the length of the heating zone can approach more than 90 % of the tube length [25]. The use of hiTRAN heat transfer enhancement to shorten the in-effective subcooled length has been investigated theoretically and experimentally. Reddy [26] developed a computer model for the performance of thermosiphon reboilers, measured water data were predicted well. He extended the model to in-corporate hiTRAN wire matrix inserts installed over the calculated sub cooled length. When applying the inserts in this way, the recirculation rate decreased by typically about 30 – 40 %. At the same time, the sub cooled length is shortened with substantial higher tube side coefficient, leaving more tube length for the more effective two-phase boiling regime. Overall, an increase of 50 % to 150 % of tube side performance was reported. A more recent study by Reza [27] gave similar results. His computer model without enhancement was in good agreement with the HTFS computation results for cyclohexane, which was used in his evaluation as the process fluid. He then simulated the performance with hiTRAN installed in the calculated pre-heating region. The results show up to ~60 % reduction of pre heating length and an overall heat transfer improvement of up to 29 %, when using hiTRAN Turbulators.

Experiments to investigate the behavior of thermosiphon reboilers were conducted by Hammerschmidt [28]. For these experiments the insert was installed over the whole tube length of a single tube reboiler. Experiments were conducted with water and more viscous water-glycerol mixtures. Applying a sophisticated measurement regime it was possible to measure the bulk temperature of the evaporating liquid over the tube length. In Fig. 16 bulk temperature profiles along the center of the tube with and without hiTRAN are displayed. In the pre heat region the temperature increases linearly with tube length. Due to the higher heat transfer and a reduction in flow velocity when using hiTRAN, the pre heat zone is considerably shorter when using hiTRAN. Once the saturation temperature is reached, the bulk temperature decreases since the saturation temperature decrease with reduced static head towards the top of the tube. The experimental results reflect well the simulation results discussed earlier.

As expected, due to the higher frictional pressure drop and due to the fact that the inserts were installed over the whole tube length, a reduction in recirculation flow to about 1/3 of the plain empty tube value was measured. Even with this lower flow rate the overall heat transfer coefficient was about 10 % higher for water. More importantly, for viscous water-glycerol mixtures the tube side heat transfer increased about 2.5 times compared to the plain empty tube.

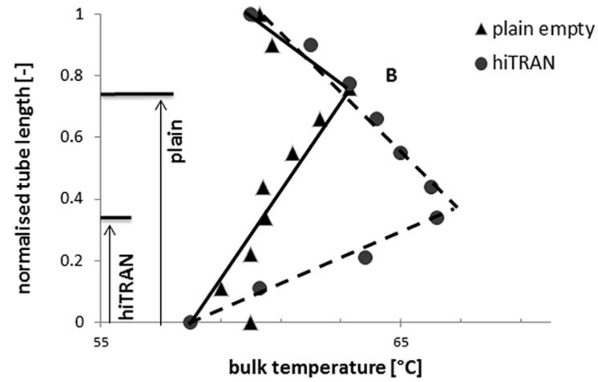

Figure 16. Bulk temperature profile in a thermosiphon reboiler tube with and without hiTRAN [28]. Water-glycerol mixtures, 90 % apparent liquid head, pressure above liquid head ~0.2 bar, temperature driving force ~15 K.

In industrial applications the elements are in general designed in such a way that they are only installed over the calculated preheat length; the feedback from these applications is in line with the research results presented.

3.2.2 Reduced Flow Instabilities with hiTRAN

Unsteady flow in reboiler applications can cause operational and performance problems. The causes for these instabilities are complex. Instabilities in two-phase flows are sensitive to pressure changes in the system. Low system pressures tend to show a higher tendency towards instabilities. Under thermosiphon operation conditions density wave oscillations are the most common form of fluid fluctuations. The so called "Ledinegg instability" is the other major form of instability in vaporizers. They are more commonly experienced when both wet and dry boiling mechanisms occur over the tube length. Under these conditions the incremental pressure drop can decrease substantially over the flow length and cause instabilities. According to Boure [29], this type of instability can be avoided by installing a throttle valve upstream of the exchanger. This steepens the effective pump characteristic enough to avoid flow excursion and move operation into a stable region. Parametric studies to reduce flow instabilities were carried out in the same paper. It was concluded that inlet restriction increases single-phase flow friction, which is in phase with the change in inlet flow. Therefore it provides a dampening effect on increasing flow, hence, an inlet restriction increases flow stability. According to Arneth [25] it is common practice to restrict the flow with a suitable orifice or gate valve in the inlet piping of the reboiler to manage fluctuating flow behavior. In a more recent review paper Nayak and Vijayan [30] report always positive effects in a forced circulation reboiler in case of additional pressure drop in the subcooled single-phase region. In general, this is also the case for thermosiphon reboiler applications. Investigation of flow stability in parallel tubes has also shown beneficial effects when the pressure drop in the single-phase region was increased.

Crowley [31] reported a positive effect on flow stability when the heated length was reduced. This was observed in forced and in natural convection applications. When designing a reboiler in such a way that hiTRAN is installed only in the sub-cooled region additional pressure drop is induced into this region. This has similar effects as a flow restriction in the inlet of the exchanger with the additional benefit that the additional pressure drop is converted into flow turbulence with higher tube side heat transfer in this region.

In the paper mentioned earlier, Hammerschmidt [28] also investigated flow instabilities. Under plain empty tube vacuum conditions the water flow collapsed when operating with liquid heads below 80 % of the tube length, whereas with hiTRAN a stable recirculation was maintained with even lower liquid heads. When operating with the more viscous water-glycerol mixtures under vacuum conditions the changes in flow stability were even more pronounced. In the plain empty tube experiments for liquid levels of 100 % tube length and below severe fluctuations were measured. In contrast, the tube with hiTRAN installed over the whole tube length showed a higher recirculation flow with only minor fluctuations for all liquid levels investigated. As reported before this has also beneficial impact of the measured heat transfer in such scenarios.

3.2.3 hiTRAN in Dry Wall Condition Boiling

Dry wall conditions can be found at any vapor content. When encountered at low vapor quality or even under sub cooled boiling conditions then the wall superheat and heat flux is high enough to form a continuous vapor blanket between tube wall and liquid. The flow pattern under these conditions can be described as inverted annular flow, with the liquid in the center core and the vapor flowing near to the wall. This can also be referred to as forced convection film boiling.

At high vapor qualities and low vapor velocities dry out occurs when all the liquid at the wall evaporates leaving the entrained droplets in the vapor to still be evaporated, whereas at high vapor velocities last remaining liquid at the wall is sheared off from the wall and redistributed in the vapor flow as droplets (D-E in Fig. 15). In vertical tubes this effect happens gradually over the whole inner tube circumference, whereas in horizontal tubes the dry zones occur first on the top and then gradually over the whole circumference of the tube. In general, in literature the regime is referred to as dispersed or mist flow boiling.

Dry wall heat transfer rates can be as low as 1/30th of the wet wall heat transfer coefficient [32]. The extent of the so called "transition region" from wet wall conditions to dry wall conditions where the heated surface is wetted intermittently, is often difficult to predict and can cause uncertainty. Therefore, it is important to identify those conditions and to account for them in the design of the vaporizer. They are target areas to apply heat transfer enhancement technology such as hiTRAN wire matrix elements in order to improve the poor heat transfer coefficient or to prevent the onset of dry wall conditions.

In order to encounter dry wall conditions at low vapor qualities or even in a sub cooled regime, the critical convective boiling heat flux (CHF) has to be reached. The first encounter of dry wall patches is often referred to as "departure from nucleate boiling" (DNB), since it only happens in regions where nucleate boiling is the dominant heat transfer mechanism. Under those conditions a thin annular film of vapor and a central liquid core can be observed. In order to understand how hiTRAN heat transfer enhancement can prevent dry wall conditions or shift the DNB to higher heat fluxes, the controlling influences on two-phase boiling heat transfer in tubes have to be discussed.

To describe tube side two-phase flow boiling heat transfer, models consider normally the contribution of nucleate and convective boiling. In this assumption convective boiling refers to the convective-only process between tube wall and liquid phase, whereas nucleate boiling accounts for the heat transfer induced by the bubble formation at the tube wall. Nucleate boiling is typically a strong function of heat flux and not effected very much by mass velocities, the conditions for convective boiling are reversed with a strong influence of vapor and liquid flow velocities and weaker influence of the temperature field at the wall. This means under plain empty tube conditions nucleate boiling tends to be dominant at low vapor qualities and high heat fluxes and convective boiling at high vapor qualities and mass velocities and low heat fluxes. If both heat transfer mechanisms exist at the same time, superposition models in a form first proposed by Chen [33] can be applied. Here, the two-phase heat transfer is calculated by adding the convective and nucleation heat transfer. In those correlations it is assumed that forced flow partially suppresses nucleation of boiling sites (suppression term *S*). On the other hand, generated vapor increased the liquid velocity and hence the convective coefficient (enhancement term *F*). The correlations are written as:

$$\alpha_{\mathrm{tp}} = \alpha_{\mathrm{nb}} S + \alpha_{\mathrm{cb}} F \tag{4}$$

Experiments with enhancement devices such as twisted tapes and mesh inserts have shown increased critical heat flux values when compared with plain empty tubes at similar process conditions. Increased CHF values up to several hundred percent are reported [34–37]. Those experiments indicated an increased influence of the second term in Eq. (4) on the overall two-phase heat transfer. Since the two-phase boiling heat transfer can be expressed by:

$$q_{\mathrm{tp}} = \alpha_{\mathrm{tp}} (T_{\mathrm{W}} - T_{\mathrm{sat}}) \tag{5}$$

It is obvious that as the convective component in Eq. (4) increases by the use of enhancement devices, the wall superheat is reduced in case of heat flux boundary conditions such as fossil fuel boilers. In case of fluid heat exchangers with temperature boundary conditions, the increase in convective boiling can be used to operate the device with a lower wall temperature to prevent CHF.

The use of passive enhancement such as hiTRAN does also have a further suppression effect on the generation of nucleation sites. At high convection boiling heat transfer the wall is cooled below the superheat needed in order to sustain nucleation. The heat flux needed in order to initiate nucleate boiling can be expressed as follows [32]:

$$q_{\mathrm{ONB}} = \frac{2\sigma T_{\mathrm{sat}} \alpha_{\mathrm{lp}}}{r_0 \rho_{\mathrm{V}} \Delta h_{\mathrm{V}}} \tag{6}$$

This shows that an increase in liquid phase convection heat transfer (α_{lp}), as achieved with hiTRAN, shifts the heat flux required for the onset of nucleate boiling to higher values. This is related to the wall cooling effect with higher convective heat transfer. This in turn reduces the amount of vapor generated directly at the tube wall for a given heat flux. For those reasons hiTRAN has been applied successful to suppress film boiling in operating conditions susceptible to convective film boiling.

It has to be noted that nucleate boiling is a very effective heat transfer mechanism. As outlined before, passive enhancement such as hiTRAN can suppress bubble formation and therefore reduce the effectiveness of this process. It is therefore only recommended to use this technology in case of operating conditions near the CHF to suppress convective film boiling.

In cases where film boiling is sustained even in the presence of hiTRAN, it can be expected that the heat transfer rates in the vapor blanket are increased. In general, the vapor film thickness increases along the flow path and can be as small as 10 – 4 mm in cases where film boiling starts in the sub cooled region. According to Groeneveld [38], in this zone heat is transferred through the vapor by conduction only. Along the flow path the film becomes thicker and more agitated. In this region heat transfer through the vapor film can be treated as an analogy to film wise condensation [32]. It can be anticipated that the loops disturbing this vapor blanket do increase the heat transfer.

Feedback from industrial applications does confirm this assumption. This is illustrated in a real case study, a BEU 2 pass TEMA type shell-and-tube heat exchanger (702 tubes, 4 m length) with ethylene evaporating on the tube side. The exchanger performed below specification. Evaluating the process conditions, convective film boiling over large sections of the exchanger was suspected. After retrofitting the exchanger with hiTRAN elements, the increase in performance indicated suppression of film boiling with much higher heat transfer. In Tab. 1 the conditions before and after retrofit are shown.

Table 1. Results after retrofit of ethylene evaporator with hiTRAN.

	Plain empty	hiTRAN (after retrofit)
Flow rate [$\mathrm{kg\,s^{-1}}$]	14.5	21.1
Temp. in/out [°C]	−100/−1 (sat.)	−100/30 (superheated)
Pressure in/out [bar]	40/39.93	40/39.74
Heat transfer [$\mathrm{W\,m^{-2}K^{-1}}$]	613	2390
Heat duty [kW]	261	618

3.2.4 hiTRAN in Dispersed Flow Boiling

One of the typical characteristics of the post dry-out regime (Fig. 15, D, E) in mist flow is the departure from thermodynamic equilibrium between the vapor temperature and liquid temperatures of the droplets. Groeneveld [38] states that the extent of non-equilibrium encountered is typically a function of pressures and mass flow rates. Large degrees of vapor superheat have been measured in this region. Since conditions are very complex to determine, the real vapor temperature is unknown and can be found somewhere between pure vapor superheat temperature without the presence of droplets and the droplet saturation temperature.

The vaporization of droplets is slow because the droplets move at about the same velocity as the vapor. The saturation temperature of the droplet increases with decreasing radius and makes it therefore more difficult to evaporate small droplets. The two main heat transfer mechanisms in such a system are convective heat transfer from the wall to the vapor and evaporation of the entrained droplets by superheated vapor. Whereas the first mechanism is single-phase heat transfer, the heat transfer to the droplets is more difficult to describe. To do this, at any cross section of the tube the droplet size distribution has to be known. Nevertheless, it is possible to estimate the heat transfer from the vapor to a single droplet with:

$$\mathrm{Nu_D} = 2 + 0.6\,\mathrm{Re_D^{1/2}}\,\mathrm{Pr_V^{1/3}} \tag{7}$$

The term, including the droplet Reynolds number, accounts for the convective contribution to heat transfer, whereas the constant "2" just describes the conductive term. In this equation $\mathrm{Re_D}$ is determined by the characteristic droplet diameter and the difference in velocity between vapor and droplet.

$$\mathrm{Re_D} = \frac{\rho_{\mathrm{V}}\ d_{\mathrm{D}}\,(u_{\mathrm{V}} - u_{\mathrm{D}})}{\mu_{\mathrm{V}}} \tag{8}$$

This indicates that the heat transfer from the vapor to the droplets becomes very low in case of small droplets with velocities equal to the vapor flow. In those cases, the heat transfer is dominated by conduction from the vapor to the droplet interface.

The use of hiTRAN wire matrix technology offers several advantages when operating under those dispersed flow conditions:

- Higher heat transfer rate between wall and vapor: In order to evaporate the remaining small droplets, a superheated vapor environment is required. Since hiTRAN improves the single-phase heat transfer several times in high Reynolds number vapor flows, the achieved superheat can be much greater compared to a plain empty tube, therefore, enabling higher heat flux from the vapor to the droplet.
- Droplet breakup: Because of the staggered loop arrangement the loop wires represent an obstacle in the flow path of the droplet. In case the droplet hits the loop wire it can break up, this depends on the impact velocity, droplet size and droplet fluid properties. CFD simulations show that the velocity field near the wires does show steep gradients at high vapor velocities. A droplet in that region is therefore exposed to different shear forces on the interface, which makes the droplet surface unstable and can also lead to break up of the droplet. The stability of a single drop and its maximum size can be determined by calculating the corresponding Weber number:

$$\mathrm{We_D} = \frac{\rho_\mathrm{V}\ (u_\mathrm{V} - u_\mathrm{D})^2\ d}{\sigma} \tag{9}$$

Main parameter is the relative velocity between vapor and droplet. For vibrational droplet break up, critical weber numbers from 7 to 12 are reported in literature [39, 40].

- Heat transfer rate between vapor and droplet: The heat transfer from the vapor to the droplet depends on the relative velocity as shown with Eq. (7). Since the relative velocity is changed considerably when hitting an insert loop wire, it can be assumed that the heat transfer rate is higher compared to the empty tube flow. CFD simulations show that the loop wires cause local velocity fluctuations near to the loops, due to the inertia of the droplets there is a lack in response to such fluctuations, causing increased relative velocities with increased vapor droplet heat transfer. Another effect is the deflection of droplets perpendicular to the main flow caused by velocity field variations near to the loop wire; this makes it more likely that droplets hit the hot tube wall with high heat transfer rates.

Over many years, hiTRAN has been used in various industrial applications to improve the thermal performance. Under dispersed flow conditions the reported performance was in general better that the plain empty tube performance.

3.2.5 Summary of the Use of hiTRAN Technology in Two-Phase Flow Applications

As a result of observations outlined in Sect. 3, the benefits of using hiTRAN in two-phase flow applications can be summarized as follows:

- Improved film heat transfer in laminar and laminar wavy-flow conditions for condensing and falling film applications;
- Higher mass transfer rates between vapor-liquid interfaces in film flow applications where the rate is limited due to low interfacial shear;
- Increased sensible vapor heat transfer coefficients, benefiting applications with vapor cooling duties, e.g., condensation of organic mixtures;
- Reduction of pre heat length in reboiler applications for viscous flow conditions or in vacuum applications;
- In reboilers flow instabilities can be reduced by additional pressure drop in the single phase entrance region;
- By suppressing nucleate boiling the critical heat flux can be shifted to higher values;
- Increased heat transfer coefficients when operating under dry out wall conditions;
- Reduction of mist flow and droplet carryover in vaporizers.

Peter Drögemüller studied production engineering at Bremen University, where he also received his Ph.D. on the subject of evaporation of viscous liquids. In 1998 he joined Cal Gavin Ltd (based in the UK) where he is now the Research & Development manager. His main research areas are the development of insert technologies to enhance single and two-phase heat transfer and the use of inserts for process intensification. He is also involved in the programming of software to design tubular heat exchangers with this technology.

Symbols used

d	[m]	diameter
F	[–]	convection enhancement term
Gr	[–]	Grashof number
Nu	[–]	Nusselt number
Pr	[–]	Prandtl number
q	[W m^{-2}]	heat flux
r_0	[m]	critical nucleation radius
Re	[–]	Reynolds number
S	[–]	nucleation suppression term
T	[K]	temperature
u	[m s^{-1}]	velocity
We	[–]	Weber number
y	[m]	coordinate

Greek Symbols

α	[W m^{-1}K^{-1}]	heat transfer coefficient
Δh_V	[J kg^{-1}]	latent heat of vaporization
Γ	[kg s^{-1}m^{-1}]	flow rate per wetted perimeter
σ	[N m^{-1}]	surface tension
ρ	[kg m^{-3}]	density
T	[N m^{-1}]	shear stress
μ	[N m^{-1}s^{-1}]	dynamic viscosity

Abbreviations

CFD	computational fluid dynamic
CHF	critical (convective) heat flux
DNB	departure from nucleate boiling
HTFS	Heat Transfer and Fluid Flow Services (now part of Aspentech)
LDV	laser doppler velocimetry
PIV	particle image velocimetry
UHF	uniform heat flux
UWT	uniform wall temperature

Subscripts

b	bulk fluid value
c	constant fluid property value
D	droplet
cb	convective boiling
F	film
i	inside
lp	liquid phase
m	mean value
n	experimental derived exponent
nb	nucleate boiling
onb	onset of nucleate boiling
sat	saturation
tp	two-phase
tr	transition from laminar to turbulent
V	vapor
w	wall value

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