HELIXCHANGER[®] HEAT EXCHANGER - FIELD EXPERIENCE AND ONGOING DEVELOPMENT

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ABSTRACT

Case studies spanning 20 years of experience with Lummus Technology's HELIXCHANGER®1 heat exchanger at a number of refineries provide a body of evidence that this technology can be used to increase run length and reduce total operating expense (OPEX) associated with fouled heat exchange surfaces [1,2]. Laboratory testing of HELIXCHANGER bundles provides insights into the mechanisms involved, and a basis for future predictive fouling models for this type of equipment. The methods developed in this study were used to evaluate the effectiveness of bypass sealing devices (e.g. sealing strips) which are sometimes used to reduce bypass flow around the outside of a tube bundle. Axial seal strips as recommended for segmental baffles were found to provide no discernible performance improvement when applied to HELIXCHANGER helical baffles, and research is ongoing to develop a bypass sealing device that is more effective for a helical flow pattern. Some promising initial results are presented.

INTRODUCTION

This paper describes the use of HELIXCHANGER heat exchangers in refinery fouling services with an emphasis on crude oil fouling applications. In a HELIXCHANGER heat exchanger, the conventional segmental baffle plates are replaced by quadrant shaped baffles positioned at an angle to the tube axis (Fig. 1). For more information on HELIXCHANGER construction, design, and applications the reader is referred to the numerous publications on this well-known technology [1-5] and to US Patent 6827138 [6].



Fig. 1. HELIXCHANGER heat exchanger bundle during fabrication

Referring to Fig. 2, some key features of the HELIXCHANGER baffle design are:

- a) *Quadrant* shaped baffles are positioned at an angle to the tube axis along a *central pipe*. The baffles in this configuration act as guide vanes rather than flow barriers.
- b) Baffles are cut from an *elliptical plate*. The elliptical baffle shape ensures that the clearance between the baffle and the shell is consistent.
- c) The quadrants are constructed with an *overlap* which enables a balance between the rotational component in the outer region and the higher velocity vortical flow in the center of the bundle. Correct selection of the *baffle angle*, and *overlap* will yield a uniform temperature profile and minimize axial dispersion.



Fig. 2. Helical baffle layout

In the first section, a number of case studies are reviewed that demonstrate lower or reduced fouling rates with HELIXCHANGER technology in a variety of fouling services.

¹ HELIXCHANGER[®] heat exchanger is a proprietary shell and tube heat exchanger technology licensed by McDermott's Lummus Heat Transfer business

In the second section a brief review of threshold type fouling models applied to shell-side flow is provided, and the approaches used to characterize shear stress on the tube exterior surfaces are discussed. An alternative approach that is well suited to shell-side flow in general and helical shell-side flow in particular is suggested. The theory behind the optimal application of sealing strips to prevent bypass flow is also reviewed.

In the final section, experimental data is presented which allows a direct comparison of different sealing arrangements and their effect on the HELIXCHANGER performance.

SECTION I – HELIXCHANGER CASE STUDIES (FOULING MONITORING)

Several case studies have been provided by Master et al. [2] covering refinery applications in crude preheat and feed-effluent services. In one example, an increase in run length from 1 year between cleanings to over 2 years in crude preheat service is reported. Since that time the use of fouling monitoring systems has become more widespread and more data has become available.

In this section we present additional case studies using data from plant data historian and fouling monitoring systems collected by HELIXCHANGER users over several years.

5 year Study by US Refiner

HELIXCHANGER bundles were used to replace conventional segmental bundles in one of the most fouling services in the preheat train (vacuum residue/desalted crude). The higher fouling vacuum residue fluid was switched to the shell-side to minimize piping changes, although previous experience was that TEMA² fouling resistance (0.0026 m² °C/W) would likely be exceeded after 3-6 months and cleaning was required every 2 years.

Operating data was collected from the initial installation in 2008 over a 5-year period. A conventional exchanger employing segmental baffles in a similar service but at a different location was monitored over the same timeframe for comparison. The HELIXCHANGER fouling factor was between 0.0001 m² °C/W and 0.0021 m² °C/W over the full 5 year performance evaluation. During that time a conventional exchanger would be expected to exceed the TEMA fouling resistance. This is based on data from a similar refinery application where the fouling resistance increased from 0.0011 m² °C/W to 0.0043 m² °C/W over a 2 year operating period between shutdowns.

Fig. 3 is a plot of measured fouling resistance data from the plant data historian. For reference the performance of the conventional segmental baffled heat exchanger in similar service at another location is also shown.



Fig. 3. Shell-side fouling resistance from plant measurements at separate locations for vacuum residue/crude (2008-2012).

Indian Refiner – Delayed Coker

HELIXCHANGER technology has been used extensively in delayed cokers. In this study an Indian refinery monitored two HELIXCHANGER units at a new facility which were used to heat vacuum residue on the shell-side from 200°C to 293°C against HCGO. Design heat duty was 11.6 MW with a specified fouling factor of 0.0026 m² °C/W based on TEMA recommendations. Shellside velocity was in the range 1-1.5 m/s.

² Standards of the Tubular Exchanger Manufacturers

Association - Section 10



Fig. 4. Shell-side fouling resistance from two HELIXCHANGER units at Indian refinery

Operating data over a 5 month period is shown in Fig. 4. The measured fouling resistance remained below the design value over the entire monitoring period. Inlet temperature had been reduced to 166 °C for operational reasons, however as a result of the lower fouling resistance, the HELIXCHANGER units were still able to achieve the design outlet temperature of 293°C. The operating duty was 40% higher than the design duty.

North American Refinery

HELIXCHANGER technology was used to replace existing segmental bundles for several services in a North American refinery.



Fig. 5. Shell-side fouling resistance for E-08 before and after HELIXCHANGER bundle installation (Nov 2003)

A fouling monitoring system was used to record fouling resistance before and after the replacement, allowing a direct comparison of fouling rates. The observed fouling rate shown in Fig. 5 was taken from a fouling monitoring system. A screenshot from the system is also shown. For Exchanger E-08, fouling rate was approximately 3.6 times slower after the conventional bundles had been replaced with the HELIXCHANGER bundles.

SECTION II – HELIXCHANGER FOULING MODELS (FOULING PREDICTION)

HELIXCHANGER technology has been successfully applied to reduce fouling rates for over 20 years. From the preceding section it is clear that a large amount of data has been generated over this time. However, despite a large body of experience and data, there is relatively little guidance regarding the application of threshold fouling models (such as first proposed by Ebert and Panchal [7]) with helical baffles to facilitate prediction of fouling rates. Whilst there have been some attempts to model shell-side fouling for segmental baffles, these are not directly transferrable to HELIXCHANGER geometry. In addition, the available shell-side methods do not consider sealing devices which are often necessary to prevent bypassing of the heat transfer surface, but can also contribute to the form drag component of pressure drop. In this section we review shellside fouling models and suggest a new approach that could be applied for shell-side fouling in general and HELIXCHANGER technology in particular.

Shell-side Fouling Models

To understand how the helical flow pattern employed in HELIXCHANGER heat exchangers can be incorporated into fouling prediction methods, it is first helpful to review prior work on shell-side fouling models.

Crude oil fouling research has often focused on predictive models that describe fouling behavior in heat exchanger tubes using fitted parameters established from laboratory data or increasingly from field data derived from dedicated fouling monitoring software. Where fouling occurs on the shell-side, prediction is more difficult because of the complex geometries involved. However, it is generally accepted that fouling will be reduced when the shear stress is maximized and average tube skin temperature is reduced, which in turn requires that the skin friction drag on the heat transfer surface is a high fraction of the total pressure drop. This is the principle behind the semi-empirical threshold fouling model introduced by Ebert and Panchal [7], Equation 1,

$$\frac{dR_{f}}{dt} = deposition - suppression$$

$$= a_{1}Re^{-b_{1}}exp\left[\frac{-E_{a}}{RT_{f}}\right] - c_{1}\tau_{w}$$
(1)

Fouling models for shell-side flow have generally followed similar approaches to those developed for tube-side fouling. Diaz-Bejarano and Coletti [8] developed a model for simultaneous shell-side and tube-side fouling based on the threshold fouling model of Panchal [7]. Their analysis also considered occlusion of clearances due to fouling deposits. Although the resulting model was successfully fitted to operating data, it was found that thermal performance might be predicted equally well by considering either shellside or tube-side fouling in isolation, and that pressure drop data was required to decouple the problem and allow estimation of the model parameters.

Brignone et al. [9] have provided a comparison of shell-side fouling rates with longitudinal flow grid type baffles (EMBaffle[®]) and segmental baffles used to heat crude oil. They used fouling data from a plant monitor to determine dynamic fouling behavior and applied the model to predict cleaning frequency. It was also noted that pressure drop data were required to improve the predictions of the model.

Both of the preceding studies acknowledged that the actual shear stress on the tube outer surface (required to determine the suppression term in Equation 1) is unknown, and used an equivalent shell-side shear stress which included both skin friction and form drag for subsequent analysis and data reduction. Other efforts to apply the threshold fouling approach to geometries other than flow in tubes have been well summarized by Wilson et al. [10]. In particular, for turbulent flow across a tube boundary layer separation and eddy regions dissipate energy away from the heat exchange surfaces which contributes to pressure drop without acting directly on any deposit. A higher fraction of total pressure drop that is due to form drag means that a smaller fraction of the total pressure drop contributes to the suppression term in Equation 1.

Bennett and Hohmann [11] have summarized a number of methods to estimate the average shear stress for various heat exchange surfaces. For the crossflow sections of a single segmental baffled heat exchanger they decoupled skin friction and form drag for the crossflow regions using a skin friction multiplication factor (m), where

$$a_x = \frac{mf_x \rho v_B^2}{8} \tag{2}$$

The factor m in Equation 2 represents the fraction of total pressure drop which is due to skin friction. A correlation was provided for *m* as a function of the crossflow Reynolds number based on CFD simulations performed by HTRI for a single segmental baffle with 23.4% baffle cut. The value of *m* varies non-linearly from 1.0 at crossflow Reynolds number of 10 to around 0.3 at a crossflow Reynolds number of 10,000. To account for bypass and leakage effects, m is determined only for the B-Stream portion of the flow (V_B) and therefore this approach is most useful in combination with a simulation package such as Xist[®] from HTRI that uses stream analysis to determine the various flow fractions as defined by Tinker [12] and later refined by Palen and Taborek [13].

Pressure drop conversion factor

An alternative method for determining the effectiveness of a shell-side flow configuration is the so called "pressure drop conversion factor" (C_{pd}) defined by Palen and Taborek [14], which is derived based on the Colburn analogy. This method is documented in HTRI reports and so is not reproduced here.

The parameter C_{pd} provides a measure of the efficiency with which pressure drop is converted to heat transfer. It is reasonable to assume that a high conversion of pressure drop to heat transfer correlates somewhat with a higher shear stress acting on a fouling deposit. Hence, when measured under clean conditions C_{pd} can potentially be used in a similar way to the parameter *m* defined by Bennett and Hohmann. However, it has the important property that it also accounts for the bypass and leakage effects inherent to shell-side flow without prior knowledge or estimation of the B-stream fraction. If the leakage/bypass is relatively large there will be a lower pressure drop but correspondingly also a lower apparent heat transfer coefficient. Moreover, C_{pd} can be determined directly from pressure drop measurements given a knowledge of the exchanger geometry. The values for C_{pd} may be determined from experimental or field measurements. Bouharie [15] has used the pressure drop conversion factor C_{pd} to compare various seal strip configurations for segmental baffles, and again this information is available from HTRI.

For a HELIXCHANGER, C_{pd} is defined based on a single helical lead (one single rotation), so the relevant parameters are calculated according to this definition. According to Palen and Taborek, the theoretical maximum value of C_{pd} for pure crossflow with no leakage or bypass effects is ~62.5% (for turbulent flow) and ~25% (for laminar flow). By comparing the measured value of C_{pd} to the theoretical maximum it is possible to determine the effectiveness of a given bundle configuration. Additionally, it can be used to determine the relative performance of different sealing arrangements in converting pressure drop to heat transfer, and by analogy indicate shear stress at the tube surface. It can also be used for comparative studies of different shell-side configurations as shown in Section III.

Axial dispersion

Roetzel and Das [16] have described how shell-side performance can also be analyzed using an axial dispersion model. The Peclet (*Pe*) number as defined by Roetzel and Balzereit [17] represents the ratio of dispersive to convective components during the heat transfer process, and can be used to provide an experimentally derived aggregate measurement of the combined effects of leakage, maldistribution, backmixing, and bypassing. High values of *Pe* indicate plug flow, whilst low values indicate a large deviation from an ideal 1-D temperature profile. For a 1-2 pass heat exchanger (*NTU* ~2) the effectiveness begins to reduce measurably below *Pe* =10.

Both C_{pd} and Pe have proved to be very useful when comparing experimental data for different seal strip configurations as described in the following sections. The calculation of Pe is independent of any of the measurements used to calculate C_{pd} . Hence, by measuring both factors simultaneously it is possible to verify that a high value of C_{pd} does indeed correspond to a high value of *Pe* and that the factor C_{pd} can be useful to screen shell-side configurations that will have a higher shear stress allowing the HELIXCHANGER geometry to be optimized for this important parameter. In a heat exchanger network, units with a low C_{pd} value under clean conditions can be identified as these will be candidates for retrofit or upgrade to HELIXCHANGER technology.

Sealing devices to prevent bypass

One area that has been minimally addressed in the preceding studies is the influence of the sealing devices on the fouling potential. Large bypass streams are clearly undesirable since they will result in uneven temperature distribution and lower shear stress on the tube surfaces. However, the seal strip design can have a significant impact on the temperature profile and distribution of shear stress over the tube surfaces.

Sealing arrangements for segmental baffles are generally prescribed by design heuristics, such as those provided in API 660. The pre-eminent study of seal strips in tubular heat exchangers was performed by Taylor and Currie [18] who used laser-Doppler anemometry to determine the local velocity field, from which a normalized heat transfer coefficient could be derived. The relative performance of different sealing arrangements under turbulent flow conditions was compared based on heat transfer enhancement and pressure drop. Taylor and Currie noted that whilst zero gap did provide the highest heat transfer rate, the pressure drop penalty was considerable. It was concluded that the optimal sealing strip shape was rectangular and that heat transfer could be close to the zero-gap value by maintaining a gap equal to the difference between the tube pitch and the tube diameter. This was a satisfying result because the clearance required for practical construction coincided with the optimal performance. Bouharie [15] has provided a very detailed review and analysis of seal strip design choices, which is available to HTRI members.

Considerations specific to HELIXCHANGER

HELIXCHANGER bundles are often used in retrofits where available pressure drop is limited and bundle diameter is constrained by an existing shell. The HELIXCHANGER construction provides a shorter unsupported span which in turn reduces the risk of flow induced vibration at higher velocities. Where vibration concerns or available shell-side pressure drop are limiting HELIXCHANGER's unique design allows baffle angle, spacing and external sealing to be varied in order to maximize shell-side velocities.

Bennett and Nesta [19] have provided a set of guidelines for a "No-foul" heat exchanger design and operation with segmental baffles. Essentially the authors recommend maximizing velocity (and therefore shear stress on heat exchanger surfaces), minimizing wall temperature and avoiding large excess area inherent to the use of fixed fouling factors. Bott [20] has further explained how the threshold fouling concept can be applied to the design of shell and tube heat exchangers to define a design envelope.

Barletta and Chunungad [21] have pointed out that HELIXCHANGER technology can be applied to crude pre-heat trains to increase both shell-side *and* tube side velocity by enabling a smaller shell diameter than can be achieved with segmental baffles within the given constraints. In other words, the use of HELIXCHANGER technology can provide a design envelope that allows for higher velocities on both shell-side *and* tube-side. They suggest that fouling rates can be reduced by maintaining shell-side velocities of 1.2 - 2.4 m/s and tube side velocities of 2.4 m/s or higher which will offset initially high pumping costs by reducing pressure drop associated with fouled units. Many of the design considerations for helical baffles in fouling service are similar to those applied for segmental baffles, however there are some notable differences:

- (1) Three main flow paths in a
- HELIXCHANGER bundle can be defined as shown in Fig. 6 (Outer helical, main helical and core) which are continuous and in parallel, unlike the alternating pattern of window and crossflow in the conventional segmental baffles. Whilst the bulk of the flow (analogous to B-Stream) follows the main helical path around the baffles, the outer helical (analogous to C-Stream) and core flow are also important contributors to the overall heat transfer rate.
- (2) A segmental baffled heat exchanger requires correct selection of baffle cut and spacing to approach an ideal crossflow. Referring to Fig. 6, the HELIXCHANGER equivalent of an ideal helical flow pattern requires that the temperature profiles for fluid in each of the three flow paths should be similar.
- (3) The parallel flow paths are interdependent. So called "seal strips" actually balance rather than block flow in the outer helical region. Indiscriminately blocking the outer helical path can actually result in both higher pressure drop and disruption of the main helical flow which in fact reduces the heat transfer.
- (4) The flow in a HELIXCHANGER always has a longitudinal component. The flow in the core region generally has a higher longitudinal component than the main helical and outer helical regions
- (5) For a HELIXCHANGER, there is no distinction between 45 and 90° layouts or between 30 and 60° layouts.



Fig. 6. HELIXCHANGER flow regions.

SECTION III - EXPERIMENTAL STUDY OF EFFECT OF SEALING ON HELIXCHANGER BUNDLES

An experimental facility was built to gather data on HELIXCHANGER performance under a range of flow conditions, including the effect of different sealing arrangements. Two HELIXCHANGER bundles were constructed for the study as shown in Fig. 7. The first (A) was constructed as a TEMA type BES with two tubeside passes. This bundle had relatively large bundle to shell gap and a smaller baffle angle of 7°. The second (B) was constructed as a TEMA type BEU with smaller bundle to shell gap and a baffle angle of 12°. The bundle to shell clearance for Bundle B was small enough that it would not normally necessitate the use of seal strips.



Fig. 7 – (a) Bundle A (TEMA type BES) and (b) Bundle B (TEMA type BEU)

Both test bundles were used to investigate the effects of various sealing arrangements on the heat transfer, pressure drop and flow distribution over a wide range of flow conditions. The test fluids included water, and various fluids of low, medium and high viscosity including Newtonian and non-Newtonian (shear thinning) fluids. The cooling fluid on the tube-side was chilled water.

The two bundles were tested with one of three sealing configurations:

- 1. "No Seal" no sealing device at all
- 2. **"4 strips"** TEMA seal strips at 4 locations
- 3. "New Seal" Patent pending seal strip design for helical flow.

To date the "new seal" (3) has only been tested for Bundle A.

Measurements of overall heat transfer rate and pressure drop were supplemented with intermediate temperature and pressure measurements and a residence time measurement using an optical technique. The latter was used to determine the experimental value of *Pe*. Fig. 8 shows the location of intermediate temperature measurements which were used to determine the main helical, outer helical and core temperature profiles based on the definitions shown in Fig. 6.



Fig. 8. Locations of temperature measurements

The Peclet number and dispersion coefficient were calculated after Roetzel and Balzereit [17] using a method presented by the American Institute of Chemical Engineers, Dudukovic [22]. First, the mean residence time and variance for the inlet and outlet response curves were determined. The mean residence time and variance of the HELIXCHANGER were calculated by taking the difference of the inlet and outlet residence time and variance (Equations 2 and 3). The Peclet number was calculated as shown in Equation 4.

$$\mu = \mu_{out} - \mu_{in} \tag{2}$$

$$\sigma^{2} = \sigma_{out}^{2} - \sigma_{in}^{2}$$
(3)
$$Pe = \begin{cases} \frac{2\mu^{2}}{\sigma^{2}} & \frac{2\mu^{2}}{\sigma^{2}} > 10\\ \frac{\mu^{2} + \sqrt{\mu^{2}(\mu^{2} + 8\sigma^{2})}}{\sigma^{2}} & \frac{2\mu^{2}}{\sigma^{2}} < 10 \end{cases}$$

Where, in this case, μ is the mean residence time (not viscosity), and σ^2 is the variance.

It should also be noted that the Reynolds number calculated by Roetzel and Balzeriet [17] is not the same as the helical Reynolds number $(Re_{helical})$ used in later analysis. The former is calculated using a method from *VDI-Wärmeatlas*, whilst the latter is calculated based on tube outside diameter, with a proprietary method to determine the fluid velocity for the helical fluid path.

Roetzel and Xuan [23] and later Roetzel and Lee [24] have shown experimentally that *Pe* for a shell and tube heat exchanger will depend strongly on the shell to baffle clearance (STB) and can vary from 5 (at 2.5 mm STB clearance) to over 50 (for zero STB clearance). For reference the current test units were fabricated using (nominal) TEMA STB clearances of 1.6 mm, although later measurements revealed that the as-built value was somewhat larger (around 2.3 mm).

Results and Discussion

Measurements from each of the two test bundles were recorded with one of the three sealing configurations over a range of flow and viscosity corresponding to helical Reynolds number ($Re_{helical}$) in the range 60 to 40,000 (Bundle A) and 30 to 20,000 (Bundle B). A representative temperature profile under turbulent flow conditions for each bundle is shown in Fig. 9. Experimental values of C_{pd} and Pe as a function of $Re_{helical}$ are shown in Fig. 10.

Water (7,000< Rehelical<40,000)

Bundle A exhibited similar temperature profiles with no seal strips (\blacktriangle) and with 4 continuous seal strips (■). Referring to Fig. 9(a), in both cases the core and main helical flow temperatures were closely matched, whilst the outer helical temperature was higher. The relatively large outer helical flow results in a temperature pinch in the main helical flow in the last four helical leads. The heat transfer coefficient with 4 strips was marginally increased (~10%) but this was associated with a pressure drop increase of around 30% compared to the "no seal" case. Referring to Figure 10(a) for both configurations in the Reynolds number range 4000 to 40,000 Pe was between 10 and 12 and C_{pd} was relatively constant at $\sim 40\%$, indicating that the addition of seal strips had no significant benefit.

For Bundle B the addition of seal strips increased the heat transfer coefficient by 15% with a pressure drop penalty of 18%. Bundle B had a much smaller bundle to shell clearance than Bundle A such that it would not normally necessitate the use of seal strips. Nevertheless, it was found that for higher Reynolds numbers, the measured temperature profile with seal strips was somewhat improved since the main helical, core and outer helical temperatures were very similar (Fig. 9(b)). Referring to Fig. 10(b) Pe was 14 - 16 for both sealed and unsealed runs at Reynolds numbers between 7000 and 11,000. The higher value of Pe for Bundle B was expected since the bundle to shell clearance was smaller and the baffle angle was larger relative to Bundle A, resulting in reduced bypass flow. This is analogous to the optimization of baffle cut for segmental baffles, and in general confirmed the expectation that a larger baffle angle

with smaller clearances would perform better at higher velocity. The value of C_{pd} was 45 -50% and was actually slightly improved by the addition of sealing strips.



Fig. 9 - Temperature profiles for turbulent flow (a) Bundle A (BES) and (b) Bundle B (BEU)

High viscosity fluid tests (30 < Re_{helical} < 1,000)

For shell-side Reynolds numbers around 1000 Bundle A showed a 13% increase in heat transfer for the sealed case (4 strips) with a pressure drop penalty of 20%. Measurements of *Pe* were in the range 10 to 12 and were not affected by the addition of seal strips. At Reynolds number from 1000 down to 60 both C_{pd} and *Pe* decreased from about 38% to 15% and from 12 to 8 respectively, as the effects of flow bypass and leakage became more prominent. Again, the addition of seal strips had little or no discernible effect on either parameter.

For very low shell-side Reynolds numbers in the range 50 - 150, large temperature gradients were observed in all cases. For Bundle A the conventional seal increased the heat transfer coefficient by ~14% at a pressure drop penalty of 36%. However, C_{pd} for the sealed and unsealed cases were comparable in the range 15 – 24%.



Fig. 10 C_{pd} and Pe as a function of $Re_{helical}$ for (a) Bundle A (BES) and (b) Bundle B (BEU)

A reduction in both C_{pd} and Pe at lower Reynolds numbers was also observed for Bundle B. For Reynolds numbers from 100 to 1000 a slightly lower value of C_{pd} was observed for the runs with seal strips than those without seal strips. Measurements in this range support the existing design practice which would not require seal strips for the BEU bundle.

New seal strip design

From the tests performed with no seal (\blacktriangle) and with conventional seal strips (\blacksquare), there were two observations:

- 1. The pressure drop penalty of adding seal strips for Bundle A was significant compared to the relatively modest increase in heat transfer.
- 2. Shell-side performance as determined through Pe and C_{pd} was not improved by the addition of seal strips.

As introduced in Section II, whilst the conventional sealing arrangement does indeed block the flow in the region between the tubes and the shell, the resulting form drag also disrupts the main helical flow path. Heat transfer enhancement resulting from turbulence generated by the sealing strip is restricted to a local region around the strip. The net result may be that any local heat transfer improvement is offset by a reduced heat transfer rate in the main helical flow.

A number of alternative sealing designs were considered in an attempt to reduce the pressure drop whilst maintaining the heat transfer performance. A new sealing arrangement was developed to reduce the flow of the outer helical stream whilst accounting for its interaction with the main helical flow. The most promising alternative is labelled as "New Seal" in Fig. 11 and Table 1. Fig. 10 is a plot of the performance of the new seal on top of the data from Fig. 10. The new seal improved the shell-side performance over the entire range of test conditions. At higher Reynolds numbers (1,000 to 40,000) the new seal exhibited a pressure drop close to the "No seal" case whilst maintaining a higher heat transfer coefficient close to the conventional seal value.



Fig. 11. C_{pd} and Pe as a function of $Re_{helical}$ for Bundle A (BES) with (\blacktriangle) no seal, (\blacksquare) 4 strips and (\bullet) new seal.

The prior tests had shown that for higher Reynolds numbers (above 4000) the value of C_{pd} was relatively constant (~40%) for both the sealed and unsealed cases. However, with the new seal installed, C_{pd} was observed to increase from 40% to over 50% with an apparent peak at a Reynolds number of 5000. This observation seems to indicate that further optimization is possible.

At lower Reynolds numbers, $Re_{helical} \sim 150$ the new seal was particularly effective. The new seal design increased the heat transfer over the conventional seal by 7% with a pressure drop reduction of 38%. Whereas the conventional seal had little impact on the shell-side performance, the new seal actually exhibited a slightly lower pressure drop than the no seal case, and a slightly higher heat transfer coefficient than the conventional seal. This somewhat surprising result can be explained by the fact that the ratio of tube wall to bulk viscosity ratio was relatively high (~6), and by reducing cold bypass flow the new seal enabled a lower average viscosity. At the lowest Reynolds number tested, C_{pd} was increased from under 20% to over 30% by the addition of the new seal, whilst the *Pe* value was increased from 8 to 10.

Table 1. Shell-side ΔP , Nu, Pe and C_{pd} with various sealing designs for Bundle A. (a) Water - Reynolds number ~ 20,000 (b) High viscosity fluid - Reynolds number ~150

(a) Water - Rehelical ~ 20,000				
	ΔP	Nu	Pe	C_{pd}
	kPa	-	-	-
No seal	6.8	76	9	36%
4 strips	9.0	86	10	36%
New Seal	6.9	83	12	42%
(b) Viscous fluid - Rehelical ~ 150				
	ΔP	Nu	Pe	C_{pd}
	kPa	-	-	-
No seal	13	12	9	27%
4 strips	18	14	8	28%
New Seal	11	15	10	35%

CONCLUSION

Case studies from fouling measurements at a number of refineries clearly demonstrate that HELIXCHANGER technology can be beneficially applied to fouling services on the shell-side. However, despite this success, there remains a need to adapt the threshold fouling model for shell-side flow and to account for the unique features of helical baffles if such models are to be used to make predictions of fouling rates for HELIXCHANGER bundles.

Experimentally derived measurements of the Peclet number (*Pe*) and pressure drop conversion factor (C_{pd}) were determined directly from heat transfer, pressure drop and residence time measurements for two test bundles. It is proposed that these methods can be adapted to estimate the shell-side shear stress to facilitate fouling prediction methods for shell-side flow in general and HELIXCHANGER bundles in particular.

Conventional wisdom for sealing segmental baffles when applied to helical baffles may not result in an optimum flow pattern. The conventional continuous sealing strip arrangement tested was found to increase the pressure drop with little apparent heat transfer benefit, and minimal effect on axial dispersion. It is suggested that the conventional seal strip arrangement may disrupt the flow in the main helical region (analogous to the B-Stream fraction for segmental crossflow).

A new patent pending sealing device was developed for which the heat transfer performance matched or exceeded the heat transfer performance of conventional (continuous) sealing strip with a 15-45% reduction in pressure drop. The design of seal strips specific to HELIXCHANGER bundles will further improve the effectiveness of this technology in fouling service by reducing unnecessary form drag and maintaining a higher velocity throughout the bundle.

Work is ongoing to further improve the shellside performance by optimizing the new seal strip configurations identified in this study. The experimental data is being used to validate Computational Fluid Dynamics models of the HELIXCHANGER test bundles which can then be extended to larger industrial scale units.

NOMENCLATURE

- a_1 Parameter in fouling model, m²K/J
- b_1 Parameter in fouling model, dimensionless
- c1 Parameter in fouling model, dimensionless
- C_{pd} Pressure drop conversion factor, dimensionless
- *D* Tube outside diameter, m
- *E_a* Activation energy in fouling model, J/mol
- f Friction factor, dimensionless
- *m* Skin friction factor, dimensionless
- Pe Peclet number, dimensionless
- Re Reynolds number, dimensionless
- t time, s
- T Temperature, °C
- V Velocity, m/s
- μ mean residence time, s
- τ Shear stress, N/m²
- σ^2 Variance

Subscript

- B B-Stream
- F Film
- helical Helical flow
- PL Longitudinal flow
- w Wall

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