Increasing Heat Exchanger Performance

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ABSTRACT

Engineers are continually being asked to improve processes and increase efficiency. These requests may arise as a result of the need to increase process throughput, increase profitability, or accommodate capital limitations. Processes which use heat transfer equipment must frequently be improved for these reasons. This paper provides some methods for increasing shell-and-tube exchanger performance. The methods consider whether the exchanger is performing correctly to begin with, excess pressure drop capacity in existing exchangers, the re-evaluation of fouling factors and their effect on exchanger calculations, and the use of augmented surfaces and enhanced heat transfer. Three examples are provided to show how commercial process simulation programs and shell-and-tube exchanger rating programs may be used to evaluate these exchanger performance issues. The last example shows how novel heat transfer enhancement can be evaluated using basic shell-and-tube exchanger rating calculations along with vendor supplied enhancement factors.

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INTRODUCTION

Increasing heat exchanger performance usually means transferring more duty or operating the exchanger at a closer temperature approach. This can be accomplished without a dramatic increase in surface area. This constraint directly translates to increasing the overall heat transfer coefficient, U. The overall heat transfer coefficient is related to the surface area, A, duty, Q, and driving force, ΔT. This equation is found in nearly all heat exchanger design references[1-3].

$$Q = UA\delta T \qquad (1)$$

As stated in this form, U can be calculated from thermodynamic considerations alone. This calculation results in the required U such that the heat is transferred at the stated driving force and area. Independent of this required U based on thermodynamics, an available U can be determined from transport considerations. For this calculation, U is a function of the heat transfer film coefficients, h, the metal thermal conductivity, k, and any fouling considerations, f. An exchanger usually operates correctly if the value of U available exceeds the U required.

For basic shell-and-tube exchangers, there are a number of literature sources that describe how to estimate heat transfer film coefficients based on the flow regime and the type of phase change: boiling or condensing[1-4]. As a point of reference, Table 1 shows some typical values for the different film coefficients.

Table 1
Examples of Heat Transfer Film Coefficients

Description	h (W/m^2 °C)	h (Btu/hr ft^2 °F)
Forced Convection		
Liquid, Water	10,500	2,000
Vapor, Air	85	15
Condensation		
Steam, film of horizontal tubes	9,000-25,000	1,600-4,400
Steam, drop wise	60,000-120,000	11,000-21,000
Boiling		
Water, pool boiling	3,000-35,000	530-6,200
Water, film boiling	300	50

The precise calculation of *U* from the transport relationships accounts for all of the resistances to heat transfer. These resistances include the film coefficients, the metal thermal conductivity, and fouling considerations. The calculation of *U* is based upon an area. For shell-and-tube exchangers, the area is usually the outside surface of the tubes.

$$U = f(h, k, f, A) \tag{2}$$

Table 2 shows design overall heat transfer coefficients for some common shell-and-tube exchanger conditions[3]. These coefficients do not necessarily represent final designs but are sufficient for estimating purposes. The overall heat transfer coefficient can also be calculated by equation 3, provided the inside and outside film coefficients, h_i and h_o, and the fouling resistance, *f*, are known.

$$1/U = 1/hi + 1/ho + f \tag{3}$$

Table 2. Examples of overall heat transfer coefficients

Shell-and-tube exchangers	U (W/m^2 °C)	U (Btu/hr ft^2 °F)
Single phase		
Gas-Gas (Low Pressure, 1 bar)	5-35	1-6
Gas-Gas (High Pressure, 250 bar)	150-500	25-90
Gas-Liquid (Low Pressure)	15-70	3-15
Gas-Liquid (High Pressure)	200-400	35-70
Liquid-Liquid	150-1200	25-210
Liquid-Condensing	300-1200	50-210
Condensation		
Water	1,500-4,000	100-300
Organics	300-1200	50-160
Boiling		
Water	600-1,700	250-700
Organics	300-900	50-210

U can be calculated from the following simplified equation, provided the fouling resistance, and the metal thermal conductivity are not significant compared to the convective film coefficients. Also, the inside tube area must be approximately the same as the outside tube area.

$$1/U = 1/ho + 1/hi \tag{4}$$

Note that even with no fouling considerations, the overall heat transfer coefficient is always less than one-half of the highest film coefficient (h_i or h_o) and usually in the neighborhood the of the lowest film coefficient. More detailed methods to calculate an overall film coefficient are provided in the references[1-4].

This discussion is limited to the shell-and-tube type exchangers. These exchangers are the most common in the process industry and can be easily modified in most cases. Furthermore, there are many sources available to estimate the shell-and-tube heat exchanger performance. Other types of exchangers such as air coolers may also be applicable with respect to cleaning and the use of tube inserts. Most of the more exotic heat exchangers such as plate-fin type exchangers, are not easily modified or enhanced to increase performance and are not considered here. However, during an investigation to increase performance, some of the exotic exchangers may be a viable alternative if all of the other options have been exhausted.

Sometimes increasing heat exchanger performance may not result from increases in throughput or higher duties. These issues may arise simply because the exchanger is not working correctly at the present capacity. Gulley[5] describes the pertinent information to diagnose the problems and possible solutions for shell-and-tube heat exchangers that are not working. Solving these problems is usually the first step.

A plan for increasing heat exchanger performance for shell and tube exchangers should consider the following steps.

1) Determine that the exchanger is operating correctly as designed. Correcting flaws in construction and piping that may have a detrimental effect on heat transfer and pressure drop may be the solution.

2) Estimate how much pressure drop is available. For single phase heat transfer coefficients, higher fluid velocity increases heat transfer coefficients and pressure drop.

3) Estimate fouling factors that are not overstated. Excessive fouling factors at the design state result in oversized exchangers with low velocities. These low velocities may exacerbate the fouling problem. More liberal fouling factors and periodic cleaning may increase the heat exchanger's performance.

4) Consider using a basic shell-and-tube exchanger with enhancement or intensification such as finning, tube inserts, modified tubes, or modified baffles.

One simple and obvious solution for increasing shell-and-tube heat exchanger performance might be to switch the shell-and-tube fluids. The placement of the process fluids on the tube or shell side is usually not dependent on the most efficient heat transfer area. A primary concern is pressure. High-pressure fluids tend to be placed in the tubes rather than the shell, resulting in less construction material and a less expensive exchanger. Handling phase changes may dictate where fluids are placed. Switching the tube-and-shell side process streams may only be valid if the process streams have no phase change and are approximately the same pressure.

For the first three steps, engineers can use operating data and commercial software with shell-and-tube exchanger rating packages to perform the calculations and predict the resulting changes. For the fourth criteria, engineers can use software programs for the base calculation but must obtain additional information to account for the increases in film coefficients for a particular type of enhancement.

Enhanced surfaces

Since there are so many different types of heat exchanger enhancements, it is highly unlikely that a commercial simulator could support them all. Furthermore, some propriety data from the manufacturers of the heat transfer enhancement might never be released. However, that does not mean that process and project engineers can not perform some of the preliminary calculations for new technologies.

The following provides background information on many different types of heat exchanger enhancements. Heat exchanger enhancement must always satisfy the primary goal of providing a cost advantage relative to the use of a conventional heat exchanger[6]. Other factors that should be addressed include fouling potential, reliability and safety.

Heat exchanger enhancement can be divided into both passive and active methods. Passive methods include extended surfaces, inserts, coiled or twisted tubes, surface treatments, and additives. Active techniques include surface vibration, electrostatic fields, injection, and suction. Hewitt[3] provides numerous examples of the different enhancements. The majority of the current discussion is related to the passive methods involving mechanical modifications to the tubes and baffles. Figure 1 shows several different schematics of enhancements to heat exchanger tubes including finning, inserts, and twisting.

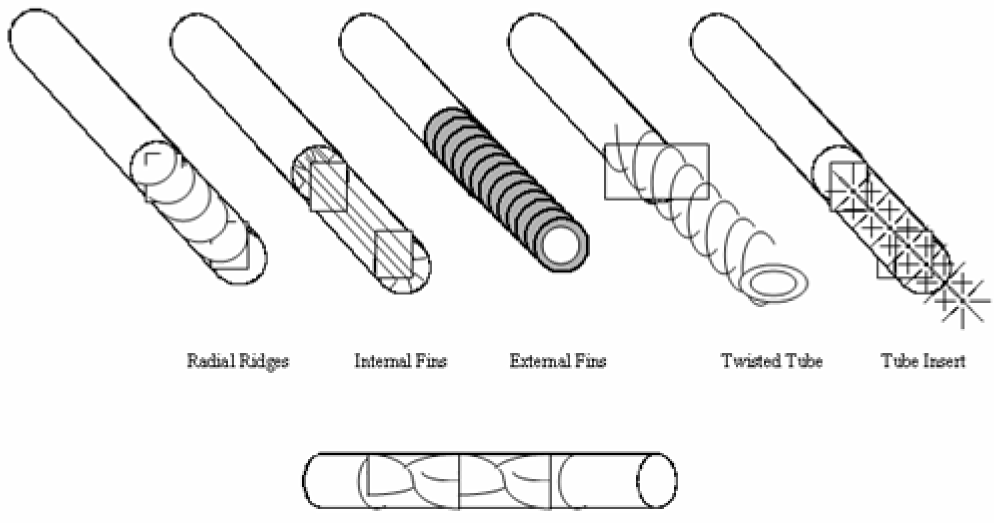

Figure 1. Examples of tubes with heat transfer enhancement.

Finning

Tubes can be finned on both the interior and exterior. This is probably the oldest form of heat transfer enhancement. Finning is usually desirable when the fluid has a relatively low heat transfer film coefficient as does a gas. The fin not only increases the film coefficient with added turbulence but also increases the heat transfer surface area. This added performance results in higher pressure drop. However, as with any additional surface area, the fin area must be adjusted by an efficiency. This fin efficiency leads to an optimum fin height with respect to heat transfer. Most of the heat transfer and film coefficients for finned tubes are available in the open literature and supported in most commercial heat exchanger rating packages. Recent papers also describe predicting finned tube performance[10]. Data for the performance of low finned tubes as compared to generalized correlations are also available in the literature[11].

Tube Inserts

Inserts, turbulators, or static mixers are inserted into the tube to promote turbulence. These devices are most effective with high viscosity fluids in a laminar flow regime[9,12-15]. Increases in the heat transfer film coefficients can be as high as five times. Inserts are used most often with liquid heat transfer and to promote boiling. Inserts are not usually effective for condensing in the tube and almost always increase pressure drop. Because of the complex relationships between the geometry of the insert and the resulting increase in heat transfer and pressure drop, there are no general correlations to predict enhancements. However, through the modification of the number of passes, a resulting heat transfer coefficient gain can be achieved at lower pressure drop in some situations[9].

Tube Deformation

Many vendors have developed proprietary surface configures by deforming the tubes. The resulting deformation appears corrugated, twisted, or spirally fluted. Marto *et al.*[16] compares the performance of 11 different commercially available tubes for single tube performance. The surface condenses steam on the outside and heats water on the inside. The author reports a 400 % increase in the inside heat transfer film coefficient; however, pressure drops were 20 times higher relative to the unaltered tube at the same maximum inside diameter.

Recently, Shilling[12] describes some of the benefits of a new twisted tube technology including the fact that tube vibration can be minimized. Furthermore the author describes how baffles may be eliminated completely. Similar to the tube inserts, these twisted tubes promote turbulence and enhance boiling. Unfortunately, no quantitative results are provided to show the increase in film coefficients for both the shell and tube fluids.

Baffles

Baffles are designed to direct the shell side fluid across the tube bundle as efficiently as possible. Forcing the fluid across the tube bundle ultimately results in a pressure loss. The most common type of baffle is the single segmental baffle which changes the direction of the shell side fluid to achieve cross flow. Deficiencies of the segmented baffle include the potential for dead spots in the exchanger and excessive tube vibration.

Baffle enhancements have attempted to alleviate the problems associated with leakage and dead areas in the conventional segmental baffles. The most notable improvement has resulted in a helical baffle as shown in Figure 2. Van der Ploeg and Master[17] describe how this baffle is most effective for high viscosity fluids and provide several refinery applications. The author further describes how the baffles promote nearly plug flow across the tube bundle. The baffles may result in shell reductions of approximately 10-20%.

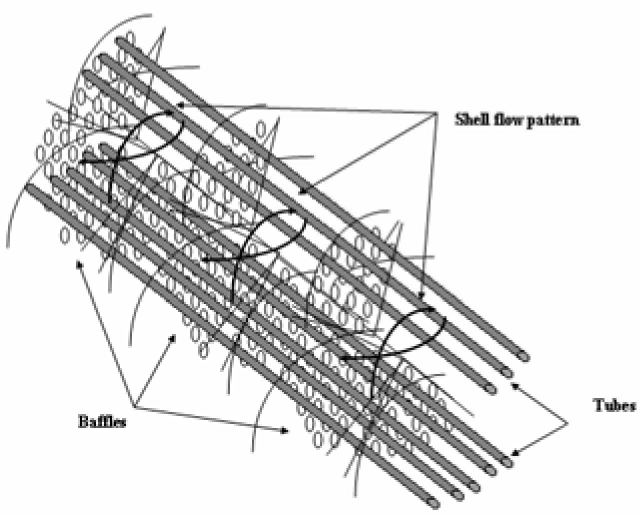

Figure 2. Schematic of helical baffles.

Combined Enhancement

Several reports have discussed the use of combined enhancement including both the effects of passive and active methods. The combination of passive methods are somewhat limited, with the most common being both internal and external finned tubes.

Other combinations may be nearly impossible because of the manufacturing techniques used in modifying the tube geometry.

One recent article is of particular interest describing the use of both helical baffles and tube inserts[18]. This exchanger was used in a crude preheat train and provides some quantitative comparisons for both the tube and shell side film coefficients along with some qualitative pressure drop information.

Enhancement Effects on Fouling

Heat exchanger enhancement may also decrease the effects of fouling, as described by Gibbard[7]. The author describes the different methods by which fouling occurs and the ability of heat exchanger enhancement to abate some of that fouling. The author also strongly cautions that the standard fouling factors reported by TEMA might not be applicable when analyzing and designing an exchanger with heat transfer enhancement. Mukherjee[8] and Polley and Gibbard[9] describe the use of tube inserts for dirty hydrocarbon services in crude oil refining. The inserts tend to promote radial flow from the center to the wall. This churning motion minimizes the material deposits on the tube wall.

Examples and Illustrative Calculations

In these examples, the process simulation programs PROSIM® and TSWEET® were used to perform the thermodynamic calculations[19]. The accompanying heat exchanger rating package was used to estimate the heat transfer coefficients and pressure drop for the basic shell-and-tube exchangers. Engineers could perform these same calculations with comparable process simulation programs and heat exchanger rating packages. Data for the increased film coefficients due to the enhanced surfaces were gathered from the listed references. Further proprietary data for new enhanced surfaces will probably have to be obtained from the vendors.

Example 1 - Amine Lean/Rich Exchanger.

The first example is a lean/rich exchanger for a simple amine plant as shown in Figure 3. Kohl and Nielsen[20] describe the basic operation of the amine facility. The objective of the lean/rich exchanger is energy conservation. Energy available from the lean amine stream is transferred to the rich amine prior to introducing the rich amine to the stripper. This energy transfer results in a decreased energy requirement for the stripper. The lean/rich exchanger has liquid streams on both sides. Furthermore, these liquids have roughly the same physical and thermal properties and the same flow rates. This results in linear heat release curves.

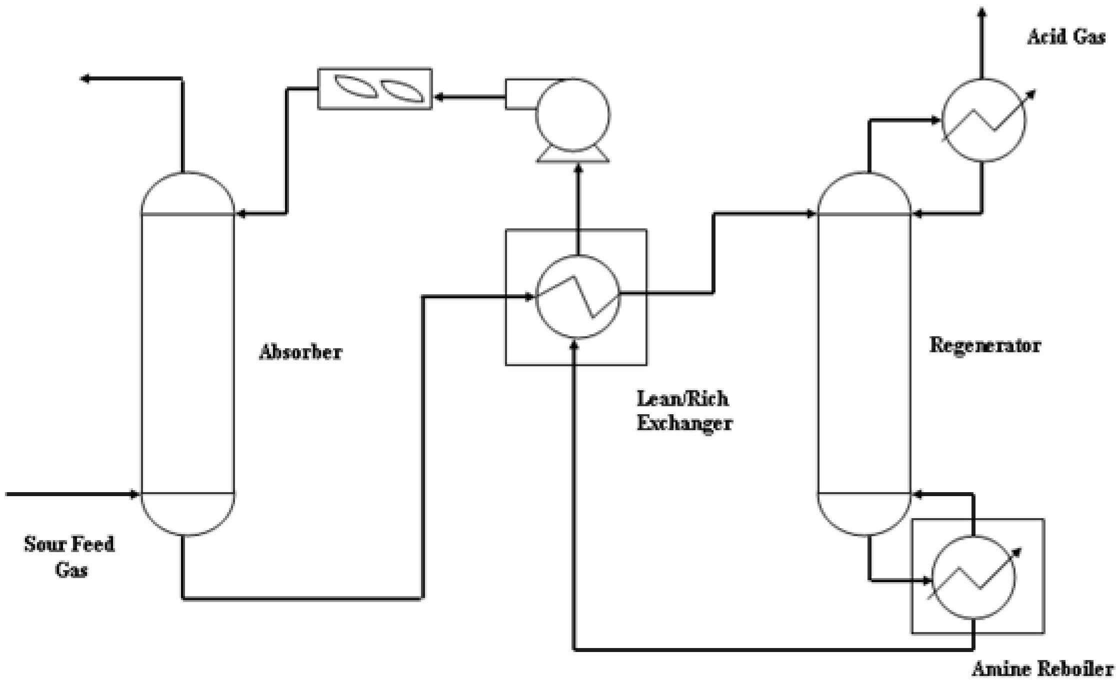

Figure 3. Schematic of a simple amine unit.

The exchanger configuration and process calculations based on a flow rate of 100 gpm is shown in Tables 3 and 4. Fouling resistances are from TEMA[21]. Table 5 shows that at the flow rate of 100 gpm, the pressure drop is only about 2/3 of the allowable pressure drop. Due to increases in capacity the amine flow rate is to be increased by 120 gpm. The outlet temperature of the rich amine should be maintained at 212 ºF.

Table 3
Process information for lean/rich exchanger

Process	m lb/hr	T_{in} ºF	T_{out} ºF	Q MMBtu/hr	DP psi
Rich amine	52,000	121	212	4.33	3
Lean amine	52,000	257	168	-4.33	3

Table 4
Lean/rich exchanger information

Shell	Tubes
Lean Amine	Rich Amine
AEN	0.75 inch OD
20 in ID	24 ft Long
8 in Baffle Spacing	4 Passes
0.002 hr ft^2 ºF/Btu	0.002 hr ft^2 ºF/Btu

Area 1394 ft^2

Table 5
Lean/rich exchanger base case calculations.

	h	DP	Q	DT	Required U	Available U
	Btu/hr ft^2 °F	psi	MMBtu/hr	°F	Btu/hr ft^2 °F	Btu/hr ft^2 °F
Rich amine	186	1	4.33	46.4	66.9	69.0
Lean amine	291	1.9				

This suggests that the exchanger could handle additional flow rate since the pressure drop is not the limiting criteria. Since both the streams are liquids, an increase in the velocity increases both the heat transfer film coefficient and the resulting pressure drop. The process simulation program and heat exchanger rating can be used to confirm that the increase in film coefficients is enough to compensate for the 20% increase in duty within the allowable pressure drop. Table 6 gives the heat exchanger calculations for the increased flow rate and shows that the exchanger appears to be viable at the new conditions.

Table 6
Lean/rich exchanger with 20 % increased circulation

	h	DP	Q	DT	Required U	Available U
	Btu/hr ft^2 °F	psi	MMBtu/hr	°F	Btu/hr ft^2 °F	Btu/hr ft^2 °F
Rich amine	230	1.3	5.18	46.4	79.6	78.2
Lean amine	323	2.7				

Example 2 - Amine Reboiler.

The effect of overestimating fouling factors is discussed for an amine reboiler. The objective of the reboiler is to regenerate the amine before contacting with the sour gas. The exchanger description and calculations based upon a 100 gpm flow rate are provided in Tables 7 and 8. These calculations are based on TEMA specified fouling resistances. Unlike the lean/rich exchanger with heat transfer in the liquid phase, the amine reboiler transfers heat to a boiling fluid usually from a condensing fluid such as steam.

Table 7
Process information for amine reboiler

Process	m	T_{in}	T_{out}	Q	DP
	lb/hr	°F	°F	MMBtu/hr	psi
Reboiler	56720	255.7	257.6	5.53	0.1
Stream	5953	274.5	274.5	-5.53	0.5

Table 8
Amine reboiler exchanger information

Shell	Tubes
Reboiler Bottoms	Steam

BKU	1 inch OD
40 inch ID	24 ft long
	2 Passes
0.002 hr ft^2 °F/Btu	0.001 hr ft^2 °F/Btu
Area 1714 ft^2	

The process simulation program and heat exchanger rating calculations for the increased amine circulation rate are provided in Table 9. The boiling and condensing film coefficients are nearly independent of the fluid velocity. As a result there is no increase in the overall heat transfer coefficient to compensate for the 20% increase in duty. Based on the assumptions, the exchanger does not have enough area to transfer the heat.

Table 9
Amine reboiler exchanger base calculations

	h	DP	Q	DT	Required U	Available U
	Btu/hr ft^2 °F	psi	MMBtu/hr	°F	Btu/hr ft^2 °F	Btu/hr ft^2 °F
Bottoms	985	0.1	5.53	17.6	183.3	198.2
Steam	3201	0.2				

The prediction of the insufficient exchanger might result from the conservative fouling factors. The condensing and boiling coefficients are relatively high compared to other heat transfer regimes. Fouling factors have a dramatic effect on processes with high heat transfer coefficients.

To investigate the influence of the fouling on the exchanger predictions, the fouling factor for the reboiled amine is decreased from 0.002 to 0.001 Btu/hr ft^2 F. The heat exchanger calculations before and after decreasing the fouling factors are shown in Table 10. It appears that the fouling factors have a large influence on the prediction of the exchanger surface area. This also suggests that taking steps to maintain lower fouling conditions may be less expensive than purchasing additional surface area.

Table 10
Amine reboiler exchanger with 20% increase in circulation

	h	DP	Q	DT	Required U	Available U
	Btu/hr ft^2 °F	psi	MMBtu/hr	°F	Btu/hr ft^2 °F	Btu/hr ft^2 °F
Bottoms	1115	0.1	6.64	17.6	220.1	202.1
Steam	3139	0.2				
With decrease in fouling factor from 0.002 to 0.001 hr ft^2 °F/Btu						
					220.1	254.5

The final aspect of increasing heat transfer performance is through the use of enhancement or intensification. The objective of enhancement is to increase the heat transfer film coefficient, supply the exchanger with secondary heat transfer surface area, and abate the fouling tendency. Heat exchanger enhancement is easily divided into several categories: Internal or external finned tubes, fluted or twisted tubes, tube inserts, and modified baffle arrangements.

Example 3 - Crude Oil Preheater.

The following example was taken from Storey and Van der Ploeg[18]. This example contains both process information and exchanger geometry for a conventional shell-and-tube and an enhanced exchanger in a crude oil preheat train. Figure 4 shows a schematic of the crude preheat train. Details of crude preheat trains and heavy oil processing are described by Nelson[22]. The authors provide the complete exchanger geometry, although some of the process information was omitted. The enhanced exchanger has both tube inserts and helical baffles.

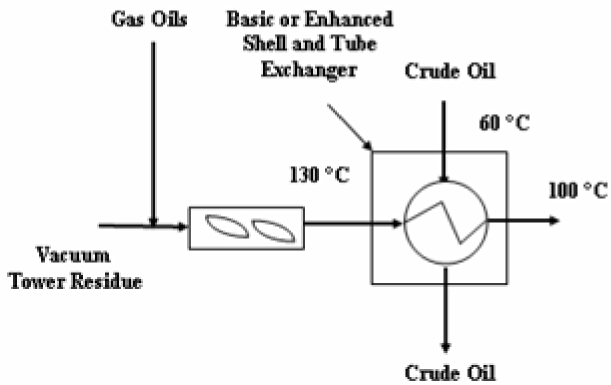

Figure 4. Schematic of a Crude Preheat unit.

This example uses the first exchanger to determine the crude oil and vacuum tower bottoms flow rate. Then based on these flow rates, the enhanced exchanger can be calculated from the basic shell-and-tube exchanger along with the increased film coefficients resulting from the enhancement.

Table 11 shows the specified shell-and-tube exchanger. A representative crude oil analysis was used in the simulation. The result of the fractionation of the crude in both the atmospheric and vacuum towers was also simulated. The amount of gas oils blended with the vacuum residue was such that it yielded a kinematic viscosity of 80 cSt at 100 ºC. The flow rate was increased such that the corresponding duty and pressure drop agreed with the values in the paper for the conventional shell-and-tube. Table 12 shows the corresponding flow rates, duty, and temperature differences.

Table 11
Crude preheat exchanger information

Shell	Tubes
Vacuum Tower Bottoms	Crude Oil
AES	2.5 cm OD
105 cm ID	4.88 m long
15 cm Baffle Spacing	4 Passes
2 in Parallel	
0.00035 m^2 ºC/W	0.00035 m^2 ºC/W
Area 504 m^2	

Table 12
Process information for crude preheat

Process	m	T_{in}	T_{out}	Q	DP
	kg/hr	ºC	ºC	MMkJ/hr	kPa
Vacuum	217,520	130	100	14.2	50
Crude Oil	700,000	68	78	-14.2	40

Table 13 gives the heat exchanger specifications for the enhanced exchanger. Based on the flow rates for the conventional exchanger, the heat exchanger rating program was used to estimate film coefficients for the base exchanger with segmental baffles and no tube inserts. The film coefficients are reported in Table 14. With these film coefficients the heat exchanger does not have enough surface area, as expected. However, the paper reports the expected enhancement in the film coefficients based on the inserts and the helical baffles. The enhanced film coefficients are shown in Table 15 along with the corresponding available *U* calculated from equation 4. With these modified film coefficients the new available *U* shows that the exchanger has sufficient area to transfer the heat.

Table 13
Crude/Vacuum gas oils enhanced exchanger information

Shell	Tubes
Crude Oil	Vacuum Tower Bottoms
AES	2.54 cm
99.1 cm	4.88 m long
Helical Baffles	2 passes
	Tube inserts
Area 240 m^2	

Table 14
Vacuum/crude oil exchanger base case calculations.

	h	DP	Q	DT	Required U	Available U
	W/m^2 ºC	kPa	MMkJ/hr	ºC	W/m^2 ºC	W/m^2 ºC
Vacuum	125	20	14.2	41.4	397	100
Crude Oil	875	40				

Table 15
Vacuum/crude oil exchanger enhancement calculations

	h	DP	Q	DT	Required U	Available U
	W/m^2 ºC	kPa	MMkJ/hr	ºC	W/m^2 ºC	W/m^2 ºC
Vacuum	125.2 × 6	20	14.2	41.4	397	422
Crude Oil	875.3 × 1.1	40				

It is important to note that this type of analysis for the enhanced exchanger can not be done *a priori*. Specific data from the manufacturers for the specific type of enhancement may be required. However, with this knowledge, engineers can analyze exchangers with moderate process changes, provided the physical properties and flow rates do not the enhancement factors.

CONCLUSION

Engineers can evaluate increasing heat exchanger performance through a logical series of steps. The first step considers if the exchanger is initially operating correctly. The second step considers increasing pressure drop if available in exchangers with single-phase heat transfer. Increased velocity results in higher heat transfer coefficients, which may be sufficient to improve performance. Next, a critical evaluation of the estimated fouling factors should be considered. Heat exchanger performance can be increased with periodic cleaning and less conservative fouling factors. Finally, for certain conditions, it may be feasible to consider enhanced heat transfer through the use of finned tubes, inserts, twisted tubes, or modified baffles. Most of these proprietary technologies can not be predicted *a priori*. However, combined with the enhancement information obtained from the vendors for specific cases along with estimations of heat transfer film coefficients, engineers can perform preliminary evaluations using these new technologies to increase shell-and-tube heat exchanger performance.

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