



APV Heat Transfer Handbook

A History Of Excellence



For more than 75 years, APV has provided customers worldwide with the latest advancements in heat exchanger technology. Today, we continue to lead the industry with our world renowned state-of-the-art technology, unsurpassed process knowledge and an unwavering commitment to our customers.

APV has evolved and grown over the years to better meet the changing needs of our customers and their industries. The first commercially successful plate-and-frame heat exchanger was introduced in 1923 by the Aluminum and Vessel Company Ltd., which became known as APV. The first Paraflow Plate Heat Exchanger, constructed of cast gunmetal plates and enclosed within a crude frame, set the standard for today's computer-designed thin metal plates.

Our vision for the future is rooted in a long standing tradition of excellence and commitment to progress. We strive to offer customers the highest quality products and services today, tomorrow and beyond.



*Stainless Steel
Tie Bar Frame*



*Carbon Steel
Tie Bar Frame*

This handbook describes the paraflow and its operating principle, compares it with conventional tubular exchangers, and outlines the many types of thermal duties for which it may be used.

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THE PARAFLOW AND ITS PRINCIPLE

The paraflow is the original plate type heat exchanger designed by APV to provide maximum efficiency and cost effectiveness in handling thermal duties while minimizing maintenance downtime and floor space requirements.

THE FRAME

The paraflow plate heat exchanger, as shown in Figure 1, consists of a stationary head and end support connected by a top carrying bar and bottom guide rail. These form a rigid frame, which supports the plates and moveable follower. In most units, plates are securely compressed between the head and follower by means of tie bars on either side of the exchanger. In a few models, central tightening spindles working against a reinforced end support are used for compression. When Paraflows are opened, the follower moves easily along the top bar with the aid of a bearing-supported roller, to allow full access to each individual plate. With the exception of some sanitary models, which are clad with stainless steel, paraflow frames are fabricated of carbon steel and are finished in chemical resistant epoxy paint. Frame ports accept bushings of stainless steel or alternative metals, which with various types of flanged or sanitary connections, form the inlet and outlet nozzles. By using intermediate connector plates as shown in Figure 2, units can be divided into separate sections to accommodate multiple duties within a single frame.

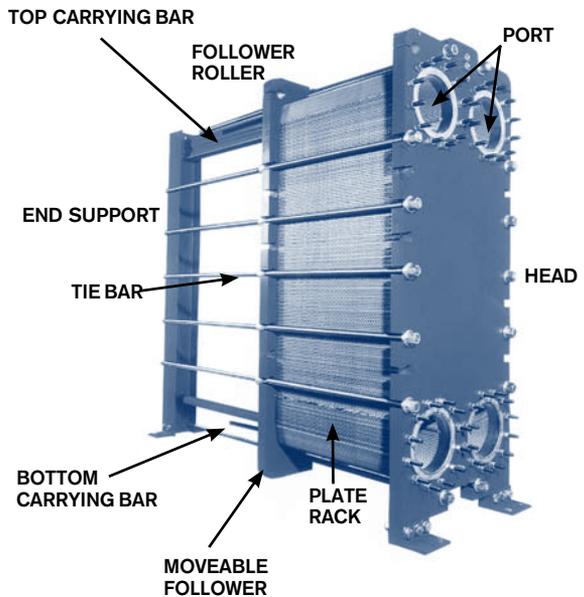


Figure 1

THE PLATES

The closely spaced metal heat transfer plates have troughs or corrugations, which induce turbulence to the liquids flowing as a thin stream between the plates (Figure 3). The plates have corner ports, which in the complete plate pack form a manifold for even fluid distribution to the individual plate passages (Figure 4).

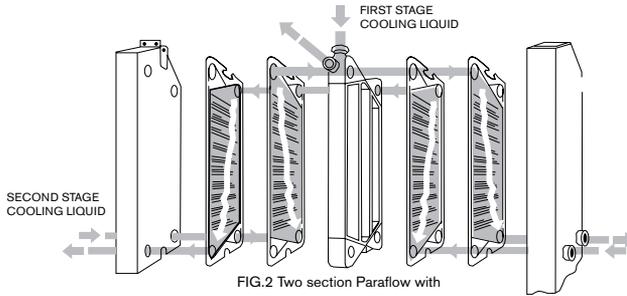


Figure 2. Two section paraflow with connector plate.

THE GASKETS

The seal between the plates is established by a peripheral gasket which also separates the thruport and flow areas with a double barrier. The interspace is vented to atmosphere to prevent cross-contamination in the rare event of leakage.

UNIQUE INTERLOCKING GASKETS

As an exclusive feature, paraflow heat exchanger plates have interlocking gaskets in which upstanding lugs and scallops are sited intermittently around the outside edges. These scallops ensure that there are no unsupported portions of the gaskets, and in combination with the patented form of pressed groove, provide mechanical plate-to-plate support for the sealing system. The upstanding lugs maintain plate alignment in the paraflow during pack closure and operation. The groove form provides 100% peripheral support of the gasket, leaving none of the material exposed to the outside. In addition, the gasket/groove design minimizes gasket exposure to the process liquid.

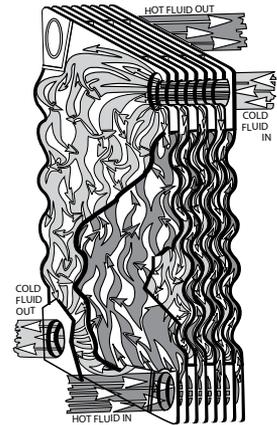


Figure 3. Cutaway of paraflow plate shows turbulence during passage of product and service liquids.

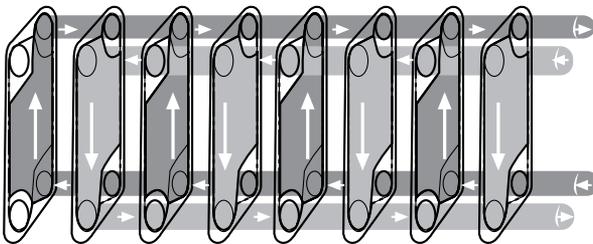


Figure 4. Single pass counter-current flow.

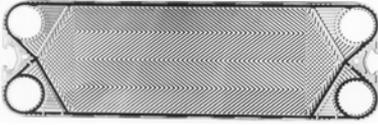


Figure 5.

PLATE ARRANGEMENT

Comparison of paraflow plate arrangement to the tube and shell side arrangement in a shell and tube exchanger is charted in Figure 7. Essentially, the number of passes on the tube side of a tubular unit can be compared with the number of passes on a plate heat exchanger. The number of tubes per pass also can be equated with the number of passages per pass for the paraflow. However, the comparison with the shell side usually is more difficult since with a paraflow, the total number of passages available for the flow of one fluid must equal those available for the other fluid to within ± 1 . The number of cross passes on a shell, however, can be related to the number of plate passes. Since the number of passages/pass for a plate is an indication of the flow area, this can be equated to the shell diameter. This is not a perfect comparison but it does show the relative parameters for each exchanger.

With regard to flow patterns, the paraflow advantage over shell and tube designs is the ability to have equal passes on each side in full counter-current flow, thus obtaining maximum utilization of the temperature difference between the two fluids. This feature is particularly important in heat recovery processes with close temperature approaches and even in cases with temperature crossovers.

Whenever the thermal duty permits, it is desirable to use single pass, counter-current flow for an extremely efficient performance. Since the flow is pure counter-flow, correction factors required on the LMTD approach unity. Furthermore, with all connections located at the head, the follower is easily moved and plates are more readily accessible.

PLATE CONSTRUCTION

Depending upon type, some plates employ diagonal flow while others are designed for vertical flow (Figure 8). Plates are pressed in thicknesses between 0.020 and 0.036 inches (0.5 to 0.9 mm), and the degree of mechanical loading is important. The most severe case occurs when one process liquid is operating at the highest working

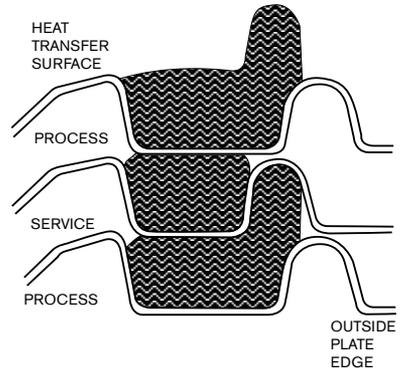


Figure 6. Exclusive interlocking gasket.

	Shell and tube	Plate equivalent	
Tube side	No. of passes	No. of passes	Side one
	No. of tubes/pass	No. of passages/pass	
Shell side	No. of cross passes (No. of baffles + 1)	No. of passes	Side two
	Shell diameter	No. of passages/pass	

Figure 7. Pass arrangement comparison: plate vs. tubular.

pressure, and the other is at zero pressure. The maximum pressure differential is applied across the plate and results in a considerable unbalanced load that tends to close the typical 0.1 to 0.2 inch gap.

It's essential, therefore, that some form of interplate support is provided to maintain the gap and two different plate forms do this.

One method is to press pipes into a plate with deep washboard corrugations to provide contact points for about every 1 to 3 square inch of heat transfer surface (Figure 9). Another is the chevron plate of relatively shallow corrugations with support maintained by peak/peak contact (Figure 10). Alternate plates are arranged so that corrugations cross to provide a contact point for every 0.2 to 1 square inch of area. The plate then can handle a large differential pressure and the cross pattern forms a tortuous path that promotes substantial liquid turbulence, and thus, a very high heat transfer coefficient.

MIXING AND VARIABLE LENGTH PLATES

To obtain optimum thermal and pressure drop performance while using a minimum number of heat exchanger plates, mixing and variable length plates are available for several APV paraflow plate heat exchanger models. These plates are manufactured to the standard widths specified for the particular heat exchanger involved but are offered in different corrugation patterns and plate lengths.

Since each type of plate has its own predictable performance characteristics, it is possible to calculate heat transfer surface, which more precisely matches the required thermal duty without oversizing the exchanger. This results in the use of fewer plates and a smaller, less expensive, exchanger frame.

To achieve mixing, plates – which have been pressed with different corrugation angles – are combined within a single heat exchanger frame. This results in flow passages that differ significantly in their flow characteristics, and thus, heat transfer capability from passages created by using plates that have the same corrugation pattern.

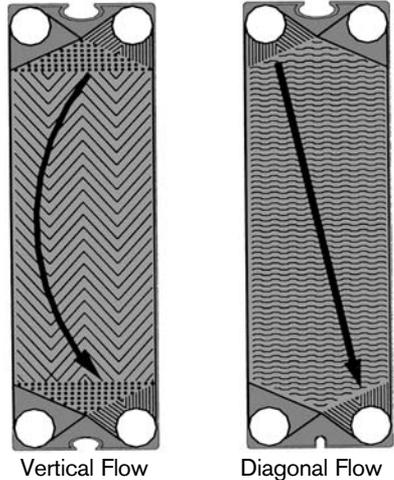


Figure 8.

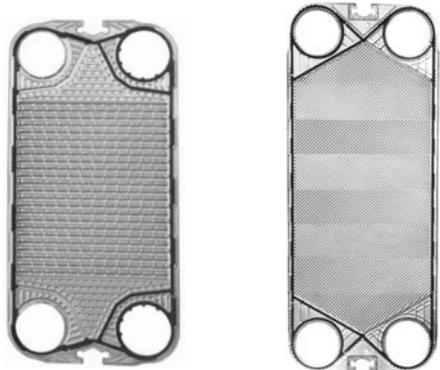


Figure 9. Corrugations pressed into plates are perpendicular to the liquid flow.

Figure 10. Troughs are formed at opposite angles to the center line in adjacent plates.



Figure 11. Low HTU passage.

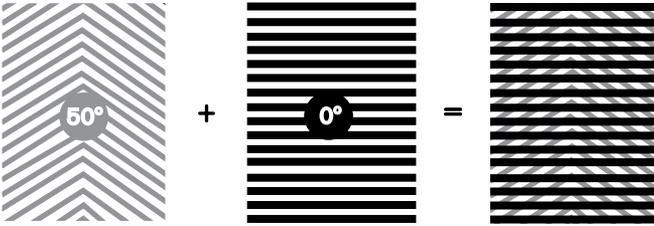


Figure 12. High HTU passage.

For example, a plate pack (Figure 11) of standard plates, which have a typical 50° corrugation angle (to horizontal), develops a fixed level of thermal performance (HTU) per unit length. As plates of 0° angle (Figure 12) are substituted into the plate pack up to a maximum of 50% of the total number of plates, the thermal performance progressively increases to a level that typically is twice that of a pack containing only 50° angle plates.

Thus, it is possible for a given plate length to fine tune the paraflow design in a single or even multiple pass arrangement, exactly to the thermal and pressure drop requirements of the application.

A more recent development is plates of fixed width with variable lengths, which extend the range of heat transfer performance in terms of HTU.

This is proportional to the effective length of the plate and typically provides a range of 3 to 1 from the longest to the shortest plate in the series. As shown in Figure 13, mixing also is available in plates of varied lengths and further increases the performance range of the variable length plate by a factor of approximately two.

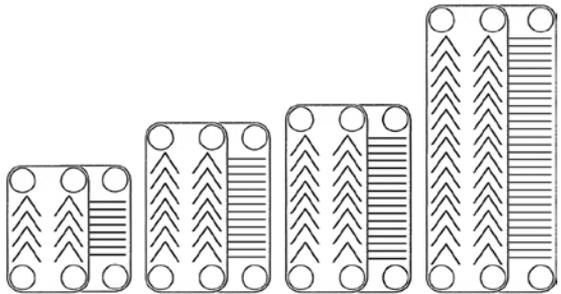


Figure 13.

This extreme flexibility of combining mixing and variable length plates allows more duties to be handled by a single pass design, maintaining all connections on the stationary head of the exchanger to simplify piping and unit maintenance.

PLATE SIZE AND FRAME CAPACITY

Paraflow plates are available with an effective heat transfer area from 0.65 ft² to 50 ft² and up to 600 of any one size can be contained in a single standard frame. The largest paraflow can provide in excess of 30,000 ft² of surface area. Flow ports are sized in proportion to the plate area and control the maximum permissible liquid throughput (Figure 14). Flow capacity of the individual Paraflow, based on a maximum port velocity of 20 fps, ranges from 25 GPM in the TR1 to 11,000 GPM in the Model SR23AO.

This velocity is at first sight somewhat high compared to conventional pipework practice. However, the high fluid velocity is very localized in the exchanger and progressively is reduced as distribution into the flow passages occurs from the port manifold. If pipe runs are long, it is not uncommon to see reducers fitted in the piping at the inlet and exit connections of high throughput machines.

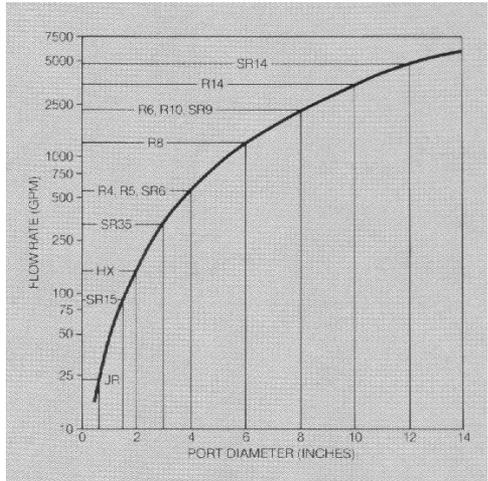


Figure 14. Throughput vs. port diameter at 14 ft/sec.

PLATE MATERIALS

Paraflow plates may be pressed from 304 or 316 stainless steels, Avesta 254SM0 or 904L, nickel 200, Hastelloy B-2, C2000, C-276 or G-3, Incoloy 825, Inconel 625, Monel 400, titanium or titanium-palladium as required to provide suitable corrosion resistance to the streams being handled.

GASKET MATERIALS

As detailed in Figure 15, various gasket materials are available as standard, which has chemical and temperature resistance, coupled with excellent sealing properties. These qualities are achieved by specifically compounding and molding the elastomers for long term performance in the APV paraflow.

Since the temperatures shown are not absolute, gasket material selection must take into consideration the chemical composition of the streams involved, as well as the operating cycles.

Gasket material	Approximate maximum operating temperature		Application
Nitrile	300°F	148°C	General aqueous service, aliphatic hydrocarbons
EPDM	320°F	160°C	High temperature resistance for a wide range of chemicals and steam
Paraprene (Polychloroprene/chloroprene)	194°F	90°C	Refrigerants: Ammonia (R717) Freon (13, 22, R134a)
Paradur (Fluoroelastomer)	400°F	205°C	Mineral oil, fuels, vegetable and animal oils
Parafloor (Fluoroelastomer)	400°F	205°C	Steam, sulfuric acids
Paramine	400°F	205°C	Rich amine service, but amine with welded pairs, acids, alkalis, sour gas

Figure 15. Parafloor gasket guide.

THERMAL PERFORMANCE

The parafloor plate heat exchanger is used most extensively in liquid-liquid duties under turbulent flow conditions. In addition, it is particularly effective for laminar flow heat transfer and is used in condensing, gas cooling and evaporating applications. For plate heat transfer in turbulent flow, thermal performance can best be exemplified by a Dittus Boelter type equation:

$$NU = (C) (Re)^n (Pr)^m (\mu/\mu_w)^x \text{ where}$$

Nu	Nusselt number	hDe/k
Re	Reynolds number	vDe/μ
Pr	Prandtl number	$C_p\mu/k$
De	Equivalent diameter, (2x average plate gap)	

(μ/μ_w) Sieder Tate correction Factor

and reported values of the constant and exponents are

$$C = 0.15 \text{ to } 0.40 \quad m = 0.30 \text{ to } 0.45$$

$$n = 0.65 \text{ to } 0.85 \quad x = 0.05 \text{ to } 0.20$$

Typical velocities in plate heat exchangers for water-like fluids in turbulent flow are 1 to 3 ft/s (0.3 to 0.9 m/s), but true velocities in certain regions will be higher by a factor of up to four due to the effect of the corrugations. All heat transfer and pressure drop relationships are, however, based on either a velocity calculated from the average plate gap or on the flow rate per passage.

Figure 16 illustrates the effect of velocity for water at 60°F on heat transfer coefficients. This graph also plots pressure drop against velocity under the same conditions. The film coefficients are very high and can be obtained for a moderate pressure drop.

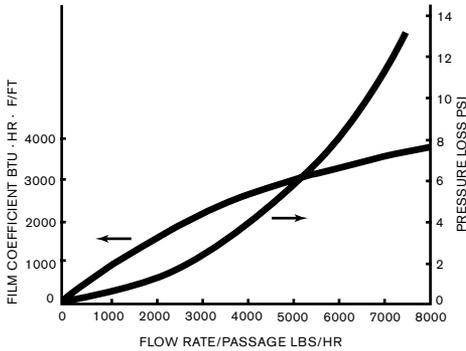


Figure 16.

One particularly important feature of the paraflow is that the turbulence induced by the troughs reduces the Reynolds number at which the flow becomes laminar. If the characteristic length dimension in the Reynolds number is taken at twice the average gap between plates, the Re number at which the flow becomes laminar varies from about 100 to 400 according to the type of plate.

To achieve these high coefficients, it is necessary to expend energy. With the plate unit, the friction factors normally encountered are in the range of ten

to 400 times those inside a tube for the same Reynolds number. However, nominal velocities are low and plate lengths do not exceed 7.5 ft so that the term $(V^2)L/(2g)$ in the pressure drop equation is much smaller than one normally would encounter in tubulars. In addition, single pass operation will achieve many duties so that the pressure drop is efficiently used and not wasted on losses due to flow direction changes.

The friction factor is correlated with the equations:

$$f = B/(Re)^y \quad \Delta p = f \cdot LrV^2/2g \cdot d$$

where y varies from 0.1 to 0.4 according to the plate and B is a constant characteristic of the plate.

If the overall heat transfer equation $Q = U \cdot A \cdot \Delta T$ is used to calculate the heat duty, it is necessary to know the overall coefficient U , the surface area A and the mean temperature difference ΔT .

The overall coefficient U can be calculated from

$$1/U = r_{fh} + r_{fc} + r_{dh} + r_{dc}$$

The values of r_{fh} and r_{fc} (the film resistances for the hot and cold fluids, respectively) can be calculated from the Dittus Boelter equations previously described, and the wall metal resistance (r_w) can be calculated from the average metal thickness and thermal conductivity. The fouling resistances of the hot and cold fluids (r_{dh} and r_{dc}) often are based on experience.

The value taken for A is the developed area after pressing. That is the total area available for heat transfer, and due to the corrugations, will be greater than the projected area of the plate, i.e., 1.81 ft² vs. 1.45 ft² for an HX plate.

The value of ΔT is calculated from the logarithmic mean temperature difference multiplied by a correction factor. With single pass operation, this factor is about 1, except for plate packs of less than 20 when the end effect has a significant bearing on the calculation. This is due to the fact that the passage at either end of the plate pack only transfers heat from one side, and therefore, the heat load is reduced.

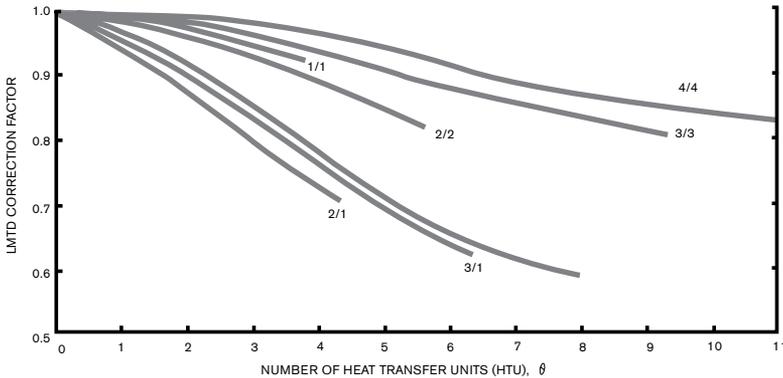


Figure 17. LMTD correction factor.

When the plate unit is arranged for multiple pass use, a further correction factor must be applied. Even when two passes are countercurrent to two other passes, at least one of them must experience co-current flow. This correction factor is shown in Figure 17 against a number of heat transfer units (HTU = temperature rise of the process fluid divided by the mean temperature difference). As indicated, whenever unequal passes are used, the correction factor calls for a considerable increase in area. This is particularly important when unequal flow conditions are handled. If high and low flow rates are to be handled, the necessary velocities must be maintained with the low fluid flow rate by using an increased number of passes. Although the plate unit is most efficient when the flow ratio between two fluids is in the range of 0.7 to 1.4, other ratios can be handled with unequal passes. This is done, however, at the expense of the LMTD factor.

LESS PLATE FOULING

The question of how to specify a fouling resistance for a plate heat exchanger is difficult to resolve. Manufacturers generally specify 5% excess HTU for low fouling duties, 10% for moderate fouling, and 15 to 20% excess for high fouling. The allowed excess surface almost always is sufficient even though in many cases it represents a low absolute value of fouling in terms of $(\text{Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})^{-1}$. If a high fouling resistance is specified, extra plates have to be added, usually in parallel. This results in lower velocities, more extreme temperatures during startup, and the probability of higher fouling rates. Because of the narrow plate gaps, and in particular because of the small entrance and exit flow areas, fouling in a plate usually causes more problems by the increase in pressure drop and/or the lowering of flow rates than by causing large reductions in heat transfer performance. Increasing the number of plates does not increase the gap or throat area, and a plate unit sometimes can foul more quickly when over surfaced.

Unless the customer has considerable experience with both his/her process and the plate heat exchanger, he/she should allow the equipment manufacturer to advise on fouling and the minimum velocity at which the exchanger should operate.

LAMINAR FLOW

The other area suitable for the plate heat exchanger is that of laminar flow heat transfer. It has been pointed out already that the paraflow can save surface by handling fairly viscous fluids in turbulent flow because the critical Reynolds number is low. Once the viscosity exceeds 20 to 50 cP, however, most plate heat exchanger designs fall into the viscous flow range. Considering only Newtonian fluids, since most chemical duties fall into this category, in laminar ducted flow the flow can be said to be one of three types: 1. fully developed velocity and temperature profiles (i.e., the limiting Nusselt case), 2. fully developed velocity profile with developing temperature profile (i.e., the thermal entrance region), or 3. the simultaneous development of the velocity and temperature profiles.

The first type is of interest only when considering fluids of low Prandtl number, and this does not usually exist with normal plate heat exchanger applications. The third is relevant only for fluids such as gases which have a Prandtl number of about one. Therefore, consider type two.

As a rough guide for plate exchangers, the rate of the hydrodynamic entrance length to the corresponding thermal entrance length is given by

$$L_{th}/L_{hyd} = 1.7 Pr$$

Plate heat transfer for laminar flow follows the Dittus Boelter equation in this form.

$$Nu = c (Re \cdot Pr \cdot De/L)^{1/3} (\mu/\mu_w)^x$$

where L = nominal plate length

c = constant for each plate (usually in the range 1.86 to 4.50)

x = exponent varying from 0.1 to 0.2 depending upon plate type

For pressure loss, the friction factor can be taken as $f = a/Re$ where "a" is a constant characteristic of the plate.

It can be seen that for heat transfer, the plate heat exchanger is ideal because the value of "d" is small and the film coefficients are proportional to $d^{-2/3}$. Unfortunately, however, the pressure loss is proportional to $(d)^{-4}$ and the pressure drop is sacrificed to achieve the heat transfer.

From these correlations, it is possible to calculate the film heat transfer coefficient and the pressure loss for laminar flow. This coefficient combined with the metal coefficient and the calculated coefficient for the service fluid, together with the fouling resistance, are then used to produce the overall coefficient. As with turbulent flow, an allowance has to be made to use the LMTD to allow for either end effect correction for small plate packs and/or concurrency caused by having concurrent flow in some passages. This is particularly important for laminar flow since these exchangers usually have more than one pass.

BEYOND LIQUID/LIQUID

Over many years, APV has built up considerable experience in the design and use of paraflow plate heat exchangers for process applications that fall outside the normal turbulent liquid flow that is common in chemical operations. The paraflow, for example, can be used in laminar flow duties, for the evaporation of fluids with relatively high

viscosities, for cooling various gases, and for condensing applications where pressure drop parameters are not overly restrictive.

CONDENSING

One of the most important heat transfer processes within the CPI is the condensation of vapors – a duty which often is carried out on the shell side of a tubular exchanger but is entirely feasible in the plate type unit. Generally speaking, the determining factor is pressure drop.

For those condensing duties where permissible pressure loss is less than one PSI, there is no doubt but that the tubular unit is most efficient. Under such pressure drop conditions, only a portion of the length of a paraflow plate would be used and substantial surface area would be wasted. However, when less restrictive pressure drops are available the plate heat exchanger becomes an excellent condenser since very high heat transfer coefficients are obtained, and the condensation can be carried out in a single pass across the plate.

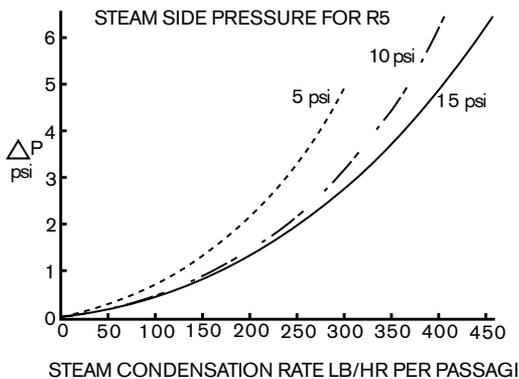


Figure 18. Steam side pressure drop for R5.

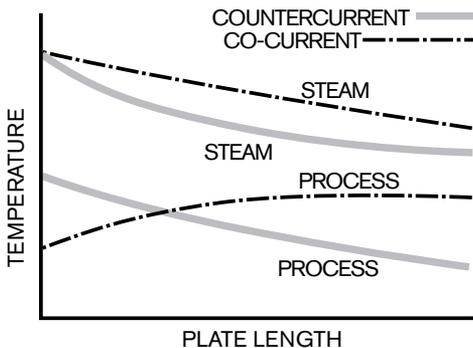


Figure 19. Temperature profile during condensation of steam.

PRESSURE DROP OF CONDENSING VAPORS

The pressure drop of condensing steam in the passages of plate heat exchangers has been experimentally investigated for a series of different paraflow plates. As indicated in Figure 18 which provides data for a typical unit, the drop obtained is plotted against steam flow rate per passage for a number of inlet steam pressures.

It is interesting to note that for a set steam flow rate and a given duty, the steam pressure drop is higher when the liquid and steam are in countercurrent rather than co-current flow. This is due to differences in temperature profile.

Figure 19 shows that for equal duties and flows, the temperature difference for countercurrent flow is lower at the steam inlet than at the outlet with most of the steam condensation taking place in the lower half of the plate. The reverse holds true for co-current flow. In this case, most of the steam condenses in the top half of the plate,

Water flow rate 16,000 lbs/hr. Inlet water temperature 216°F. Inlet steam temperature 250°F. Total number of plates -7.				
Available pressure loss PSI	1	2	3	4
Total duty BTU/hr	207,000	256,000	320,000	333,000
Fraction of plate flooded	40	30	4	0
Effective overall heat transfer coefficient, clean	445	520	725	770
Pressure loss PSI	1	2	4	4.5

Figure 20. Steam heating in an R4 Paraflow.

the mean vapor velocity is lower, and a reduction in pressure drop of between 10% to 40% occurs. This difference in pressure drop becomes lower for duties where the final approach temperature between the steam and process fluid becomes larger.

The pressure drop of condensing steam therefore is a function of steam flow rate, pressure and temperature difference. Since the steam pressure drop affects the saturation temperature of the steam, the mean temperature difference, in turn, becomes a function of steam pressure drop. This is particularly important when vacuum steam is being used since small changes in steam pressure can cause significant changes in the temperature.

By using an APV computer program and a Martinelli Lockhart approach to the problem, it has been possible to correlate the pressure loss to a high degree of accuracy. Figure 20 cites a typical performance of a steam heated Series R4 paraflow. From this experimental run during which the exchanger was equipped with only a small number of plates, it can be seen that for a 4 to 5 PSI pressure drop and above, the plate is completely used. Below that figure, however, there is insufficient pressure drop available to fully use the entire plate and part of the surface, therefore, is flooded to reduce the pressure loss. At a 1 PSI allowable pressure drop, only 60% of the plate is used for heat transfer, which is not particularly economic.

This example, however, well illustrates the application of a plate heat exchanger to condensing duties. If sufficient pressure loss is available, then the plate type unit is a good condenser. The overall coefficient of 770 Btu/hr • ft² • °F for 4.5 PSI pressure loss is much higher than a coefficient of 450 to 500 Btu/hr • ft² • °F, which could be expected in a tubular exchanger for this type of duty. However, the tubular design for shell side condensation would be less dependent on available pressure loss and for a 1 PSI drop, a 450 to 500 Btu overall coefficient still could be obtained. With the plate, the calculated coefficient at this pressure is 746 Btu, but the effective coefficient based on total area is only 60% of that figure or 445 Btu/hr • ft² • °F.

GAS COOLING

Plate heat exchangers also are used for gas cooling with units in service for cooling moist air, hydrogen, and chlorine. The problems are similar to those of steam heating since the gas velocity changes along the length of the plate due either to condensation or pressure fluctuations. Designs usually are restricted by pressure drop, so machines with low pressure drop plates are recommended. A typical, allowable, pressure loss would be 0.5 PSI with rather low gas velocities giving overall heat transfer coefficients in the region of 50 Btu/hr • ft² • °F.

EVAPORATING

The plate heat exchanger also can be used for evaporation of highly viscous fluids when as a paravap the evaporation occurs in the plate or as a paraflash the liquid flashes after leaving the plate. Applications generally have been restricted to the soap and food industries. The advantage of these units is their ability to concentrate viscous fluids of up to 5,000 centipoise.

CONCLUSION

It has been shown, therefore, that the plate heat exchanger is a relatively simple machine on which to carry out a thermal design. Unlike the shell side of a tubular exchanger where predicting performance depends on baffle/shell leakage, baffle/tube leakage and leakage around the bundle, it is not possible to have bypass streams on a paraflow. The only major problem is that the pressure loss through the ports can cause unequal distribution in the plate pack. This is overcome by limiting the port velocity and by using a port pressure loss correlation in the design to allow for the effect of unequal distribution.

The flow in a plate also is far more uniform than on the shell side. Furthermore, there is no problem over calculation of heat transfer in the window, across the bundle or of allowing for dead spots (as is the case with tubular exchangers). As a result, the prediction of performance is simple and very reliable once the initial correlations have been established.

THE PROBLEM OF FOULING

In view of its complexity, variability and the need to carry out experimental work on a long-term basis under actual operating conditions, fouling remains a somewhat neglected issue among the technical aspects of heat transfer. Still, the importance of carefully predicting fouling resistance in both tubular and plate heat exchanger calculations cannot be overstressed. This is well illustrated by the following examples. Overall coefficient 1,000 Btu/hr • ft² • °F.

A typical water/water-tubular design			B R405 paraflow, water/water duty		
Clean overall coefficient 500 Btu/hr • ft ² • °F			Overall coefficient 1,000 Btu/hr • ft ² • °F Single pass – pressure loss 9 PSIG		
Fouling resistance hr • ft ² • °F/Btu	Dirty coefficient Btu/hr • ft ² • °F	% Extra surface required	Fouling resistance hr • ft ² • °F/Btu	Dirty coefficient Btu/hr • ft ² • °F	% Extra surface required
.0002	455	10	.0002	833	20
.0005	400	25	.0005	666	50
.001	333	50	.001	500	100
.002	250	100	.002	333	200

Note that for a typical water/water duty in a plate heat exchanger, it would be necessary to double the size of the unit if a fouling factor of 0.0005 was used on each side of the plate (i.e. a total fouling of 0.001).

Although fouling is of great importance, there is relatively little accurate data available and the rather conservative figures quoted in Kern (Process Heat Transfer) are used all too frequently. It also may be said that many of the high fouling resistances quoted have been obtained from poorly operated plants. If a clean exchanger, for example, is started and run at the designed inlet water temperature, it will exceed its duty. To overcome this, plant personnel tends to turn down the cooling water flow rate and thereby reduce turbulence in the exchanger. This encourages fouling and even though the water flow rate eventually is turned up to design, the damage will have been done. It is probable that if the design flow rate had been maintained from the onset, the ultimate fouling resistance would have been lower. A similar effect can happen if the cooling water inlet temperature falls below the design figure and the flow rate is again turned down.

SIX TYPES OF FOULING

Generally speaking, the types of fouling experienced in most CPI operations can be divided into six fairly distinct categories. First is crystallization – the most common type of fouling which occurs in many process streams, particularly cooling tower water. Frequently superimposed with crystallization is sedimentation which usually is caused by deposits of particulate matter such as clay, sand or rust. From chemical reaction and polymerization often comes a build-up of organic products and polymers. The surface temperature and presence of reactants, particularly oxygen, can have a very significant effect.

Coking occurs on high temperature surfaces and is the result of hydrocarbon deposits. Organic material growth usually is superimposed with crystallization and sedimentation, and is common to sea water systems. Corrosion of the heat transfer surface produces an added thermal resistance, as well as a surface roughness.

In the design of the plate heat exchanger, fouling due to coking is of no significance since the unit cannot be used at such high temperatures. Corrosion also is irrelevant since the metals used in these units are noncorrosive. The other four types of fouling, however, are most important. With certain fluids such as cooling tower water, fouling can result from a combination of crystallization, sedimentation and organic material growth.

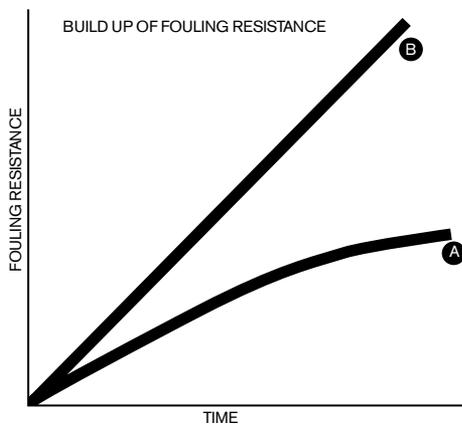


Figure 21.

A FUNCTION OF TIME

From Figure 21, it is apparent that the fouling process is time dependent with zero fouling initially. The fouling then builds up quite rapidly and in most cases, levels off at a certain time to an asymptotic value as represented by curve A. At this point, the rate of deposition is equal to that of removal. Not all fouling levels off, however, and curve B shows that at a certain time the exchanger would have to be taken off line for cleaning. It should be noted that a paraflow is a particularly useful exchanger for this type of duty because of the ease of access to the plates and the simplicity of cleaning.

In the case of crystallization and suspended solid fouling, the process usually is of the type A. However, when the fouling is of the crystallization type with a pure compound crystallizing out, the fouling approaches type B and the equipment must be cleaned at frequent intervals. In one particularly severe fouling application, three Series HMB paraflows are on a 4 1/2 hour cycle and the units are cleaned in place for 1 1/2 hours in each cycle.

Biological growth can present a potentially hazardous fouling since it can provide a more sticky type surface with which to bond other foulants. In many cases, however, treatment of the fluid can reduce the amount of biological growth. The use of germicides or poisons to kill bacteria can help.

LOWER RESISTANCE

It generally is considered that resistance due to fouling is lower with paraflow plate heat exchangers than with tubular units. This is the result of five paraflow advantages:

1. There is a high degree of turbulence which increases the rate of foulant removal and results in a lower asymptotic value of fouling resistance.
2. The velocity profile across a plate is good. There are no zones of low velocity compared with certain areas on the shell side of tubular exchangers.
3. Corrosion is maintained at an absolute minimum.
4. A smooth heat transfer surface can be obtained.
5. In certain cooling duties using water to cool organics, the very high waterfilm coefficient maintains a moderately low metal surface temperature which helps prevent crystallization growth of the inversely soluble compounds in the cooling water.

The most important of these is turbulence HTRI (Heat Transfer Research Incorporated). It has shown that for tubular heat exchangers fouling is a function of flow velocity and friction. Although flow velocities are low with the plate heat exchanger, friction factors are very high and this results in lower fouling resistance. The effect of velocity and turbulence is plotted in Figure 22.

Marriot of Alfa Laval has produced a table showing values of fouling for a number of plate heat exchanger duties (see right).

These probably represent about one half to one fifth of the figures used for tubulars as quoted in Kern but it must be noted that the Kern figures probably are conservative, even for tubular exchangers.

APV, meanwhile, has carried out test work which tends to confirm that fouling varies for different plates

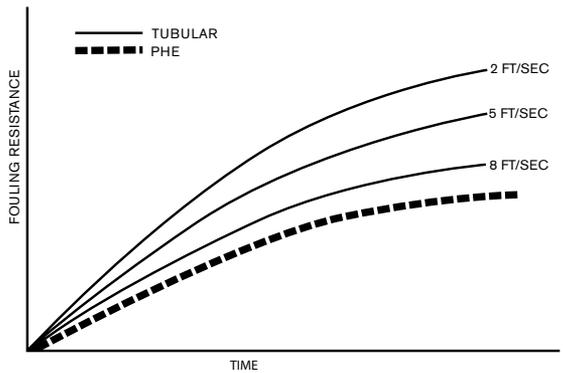


Figure 22.

Fluid	Fouling resistance (hr · ft ² · °F/Btu)
Water demineralized or distilled	0.00005
Soft	0.00010
Hard	0.00025
Cooling tower (treated)	0.00020
Sea (coastal) or estuary	0.00025
Sea (ocean)	0.00015
River, canal	0.00025
Engine jacket	0.00030
Oils, lubricating	0.00010 to 0.00025
Oils, vegetable	0.00010 to 0.00030
Solvents, organic	0.00005 to 0.00015
Steam	0.00005
Process fluid, general	0.00005 to 0.00030

with the more turbulent type of plate providing the lower fouling resistances. In testing an R405 heating a multi-component aqueous solution containing inverse solubility salts (i.e., salts whose solubility in water decreases with increasing temperature), it was learned that the rate of fouling in the paraflow was substantially less than that inside the tubes of a tubular exchanger. The tubular unit had to be cleaned every three or four days while the paraflow required cleaning about once a month.

Additional tests on cooling water fouling were sponsored by APV at Heat Transfer Research, Inc. (HTRI) with the test fluid being a typical treated cooling tower water. Results of these experiments are described in the following section.

OBJECTIVES

In many streams fouling is an unavoidable byproduct of the heat transfer process. Fouling deposits can assume numerous types such as crystallization, sedimentation, corrosion, and polymerization. Systematic research on fouling is relatively recent and extremely limited (Refs. 1, 2, 3) and almost exclusively concentrated on water — the most common fluid with fouling tendencies. The few research data which exist usually are proprietary or obtained from qualitative observations in plants. However, it generally is recognized that the prime variables affecting fouling buildup are flow velocity, surface temperature and surface material. In the case of water, the water quality and treatment must also be considered.

The importance of fouling on the design of heat exchangers can be seen from the rate equation

$$1/U = 1/h_1 + 1/h^2 + R_f$$

where

U = overall heat transfer coefficient (Btu/hr • ft² • °F)

h_1, h^2 = film coefficients of the two heat transferring fluids (Btu/hr • ft² • °F)

R_f = fouling resistance (hr • ft² • °F/Btu)

It is obvious from the equation that the higher the film coefficients, the greater effect the fouling resistance will have on the overall coefficient, and therefore, on the size of the exchanger.

In tubular heat exchangers, waterside heat transfer coefficients in the order of magnitude of 1,000 Btu/hr • ft² • °F are quite common. In plate exchangers, the coefficients are substantially higher, typically around 2,000 Btu/hr • ft² • °F. Assuming that both types of equipment operate with water-water systems, overall clean coefficients of 500 and 1,000 Btu/hr • ft² • °F, respectively, are obtained. Using a typical fouling resistance of 0.001 hr • ft² • °F/Btu (equal to a coefficient of 1,000 Btu/hr • ft² • °F), the inclusion of the fouling will cause the tubular exchanger size to increase by a factor of four while for the plate exchanger, the corresponding factor is seven.

This example clearly demonstrates the crucial importance of fouling, especially in plate exchangers. Yet, fouling resistances, which are unrealistically high, often are specified and invariably have been based on experiences derived from tubular equipment.

The common source of water fouling resistances is TEMA4 which recommends values of R_f spanning a tremendous range between 0.0015 and 0.005 hr • ft² • °F/Btu.

Flow velocity, as mentioned earlier, is a crucial operating parameter which influences the fouling behavior. For flow inside of tubes, the definition of flow velocity and the velocity profile is straightforward. But in plate exchangers, flow velocity is characterized by constant fluctuations as the fluid passes over the corrugations. It is postulated that this induces turbulence which is superimposed on the flow velocity as a factor which diminishes fouling tendencies. This has been observed qualitatively in practical applications and confirmed by unpublished APV research.

TEST APPARATUS AND CONDITIONS

The plate heat exchanger (PHE) tested was an APV Model 405 using APV Type R40 stainless steel plates, 45 inches high and 18 inches wide. The plate heat transfer area was 4 sq ft with a nominal gap between plates of 0.12 inch. Due to the plate corrugations, the maximum gap was 0.24 inch. Seven plates were used creating three countercurrent passages each of the cooling water and the heating medium. A schematic diagram of the installation is shown in Figure 23.

The PHE was mounted on the HTRI Shellside Fouling Research Unit (SFRU) together with two small stainless steel shell and tube exchangers. The PHE was heated using hot steam condensate and the fouling was determined from the degradation of the overall heat transfer coefficient. Simultaneously, tests also were run on an HTRI Portable Fouling Research Unit (PFRU) which uses electrically heated rods with the cooling water flowing in an annulus. The SFRU and PFRU are described in a paper by Fischer et al (reference 5).

The fouling tests were conducted at a major petrochemical plant in the Houston, Texas area. The test units were installed near the cooling tower basin of the plant. The cooling tower's operating characteristics are summarized in Table 1. The 140,000 GPM system had two 1,600 GPM sidestream filters. Filter backwash accounted for most of the blowdown. The makeup water for the cooling water system came from several sources and was clarified with alum outside the plant. The water treatment used is summarized below:

- Chromate-zinc based inhibitor for corrosion control (20.25 ppm chromate)
- Organic phosphonate and polymer combination as dispersant (2 ppm organic phosphonate)
- Polyphosphate as anodic passivator (5 to 6 ppm total inorganic phosphate)
- Chlorine for biological control (continuous feeding of 386 lbs/day)
- Biocide for biological control (15 ppm biocide once/week)
- Sulfuric acid for pH control (pH of 6 to 6.5)

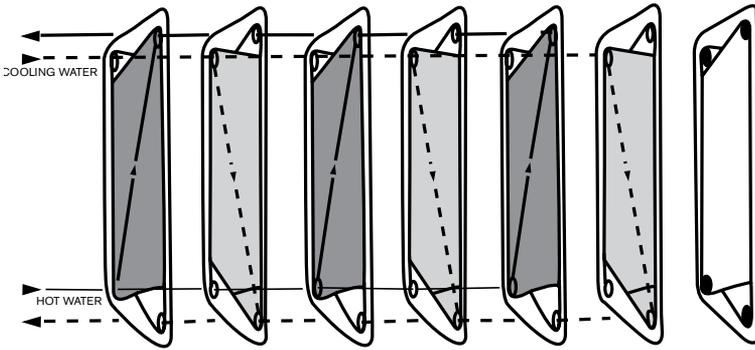


Figure 23. The APV plate heat exchanger with three parallel countercurrent parallel passages for each of the cooling water and hot water.

Tower and circulation system		Description	
Circulation rate		140,000 GPM	
Temperature difference across tower		23.4°F	
Number of sidestream filters		2	
Corrosion rate from carbon steel coupon tests		< 1 mil/year	
Water description		Cooling tower water	Makeup water
Treatment		10 Cycle concentration	Unclearified San Jacinto River 1,000 GPM
		Blowdown from coolers and sidestream filters	Clarified San Jacinto River 2,400 GPM Lissie Sand Well 475 GPM
Typical composition			
Total hardness as CaCO ₃ , ppm		520	48
Calcium as CaCO ₃ , ppm		420	40
Magnesium as CaCO ₃ , ppm		120	8
Methyl Orange Alkalinity as CaCO ₃ , ppm		20	34
Sulfate as SO ₄ , ppm		1,600	34
Chloride as Cl, ppm		800	48
Silica as SiO ₂ , ppm		150	18
Total inorganic phosphate as PO ₄ , ppm		10	-
Orthophosphate as PO ₄ , ppm		7	-
pH		6	7.5
Specific conductance, micromhos, 18C		5,000	300
Chromate as CrO ₄ , ppm		25	-
Chromium as Cr, ppm		0.5	-
Soluble zinc as Zn, ppm		2.5	-
Total iron as Fe, ppm		-	0.8
Suspended solids, ppm		100	10

Table 1. Cooling water system characteristics.

TEST PROCEDURE

The velocity for the PHE is defined by $V = w/rAc$

where V = velocity
 w = mass flow rate
 r = fluid density
 Ac = cross-sectional flow area

The cross-sectional flow area is based on the compressed gasket spacing. In other words, the flow area is computed as if there was no corrugation but smooth plates instead. The surface temperature is defined as the fluid-deposit interface temperature. Since the PHE was heated by constant temperature steam condensate, the metal temperature remained constant and the surface temperature decreased as fouling built up. In addition, due to the counterflow nature of the PHE, the surface temperature was not constant from inlet to outlet. The surface temperature reported here is the initial surface temperature at the midpoint of the plate.

Conditions of operation were selected so that all the SFRU test exchangers operated roughly at the same nominal flow velocity and surface temperature. During the operation, weekly water samples were taken for chemical analysis. At the end of each test, deposits were photographed and sampled for chemical analysis. The exchanger plates then were cleaned and the unit reassembled and new test conditions established.

TEST AND RESULT DESCRIPTION

Five test series were performed testing several velocities and surface temperatures. Typical fouling time histories are shown in Figure 24 along with the operating conditions. Notice that the PHE fouling resistances establish a stable asymptotic value after about 600 to 900 hrs. of operation. The results of the other tests are shown in Figure 25 only as the values of the asymptotic fouling resistances, as these are the data required for design.

The chemical analysis of the fouling deposit was made after each test and the results are summarized in Table 2. The primary elements found were phosphorus, zinc, and chrome from the water treatment, and silicon from suspended solids in the water.

The picture of the plates of the PHE shown in Figure 23 at the termination of Test 2 indicates that fouling only occurred in the upper third of the plates. This is the region near the hot water inlet and cold water outlet, a region of high surface temperature. Figure 24 shows a surface temperature profile for Test 2 conditions of 2.8 ft/sec and an inlet bulk temperature of 90°F. From the water chemistry parameters, the saturation temperature for calcium phosphate above which precipitation is expected was calculated to be 163°F. The shaded region on Figure 26 indicates the expected precipitation region and corresponds closely to the fouled region seen in Figure 23.

The surface temperatures during some of the tests were higher than those experienced in normal PHE operation. This is the result of the setup condition criteria that the three exchangers on the SFRU operate at the same midpoint surface temperature.

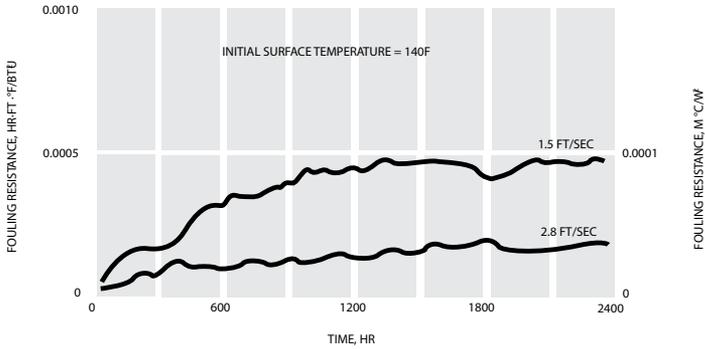


Figure 24. Typical PHE fouling curves.

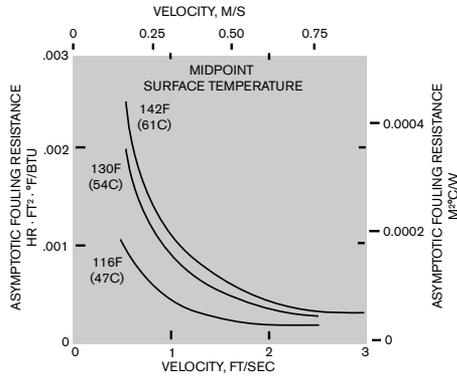


Figure 25. Asymptotic fouling resistance versus velocity with surface temperature as a parameter.

	Fouling deposit analysis, percent				
	Series 2	Series 3	Series 4	Series 5	Series 6
Loss on ignition	–	25	25	17	21
Phosphorus	19	24	13	12	22
Aluminum	2	2	8	6	4
Silicon	1	4	18	16	13
Calcium	4	8	3	15	9
Chrome	5	13	15	15	9
Iron	2	7	4	5	13
Zinc	2	20	12	13	7
Sodium	–	1	1	–	1
Magnesium	–	–	1	–	1

Table 2. Deposit analysis.

Operating velocity (both sides)	Overall heat transfer Coefficient		
	Clean	Fouling coefficient	
		TEMA 0.002 hr • ft ² • °F/Btu (0.00035 • m ² • °C/W)	Present study 0.0005 hr • ft ² • °F/Btu (0.00009 • m ² • °C/W)
1.5 ft/sec	1078 Btu/hr • ft ² • °F	341 Btu/hr • ft ² • °F	700 Btu/hr • ft ² • °F
0.45 m/s	6121 • W/m ² • °C	1936 • W/m ² • °C	3975 • W/m ² • °C

Table 3. Effects of fouling resistance on PHE performance.

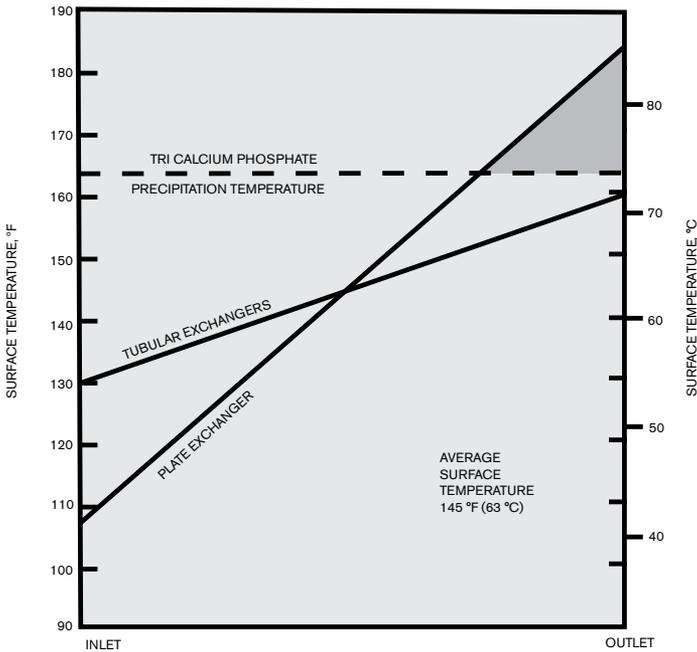


Figure 26. Surface temperature profile with indicated calcium phosphate precipitation.

Since the ratio of flow rate-to-surface area for the shell side of the shell and tube exchangers is twice that for the PHE, the cooling water had a longer residence time in the PHE than in the tubular exchanger. As a result, the lower flow rate-to-surface area ratio yields a larger temperature rise of the cooling water. The result is the steeper surface temperature profile shown in Figure 26. Although co-current operation was investigated as a correction for this problem, the required higher hot water flow rate was not available. Consequently, the PHE data in some of these tests was slightly penalized due to part of the surface being above the critical temperature for tricalcium phosphate precipitation, a condition which normally would not be encountered in industrial operations.

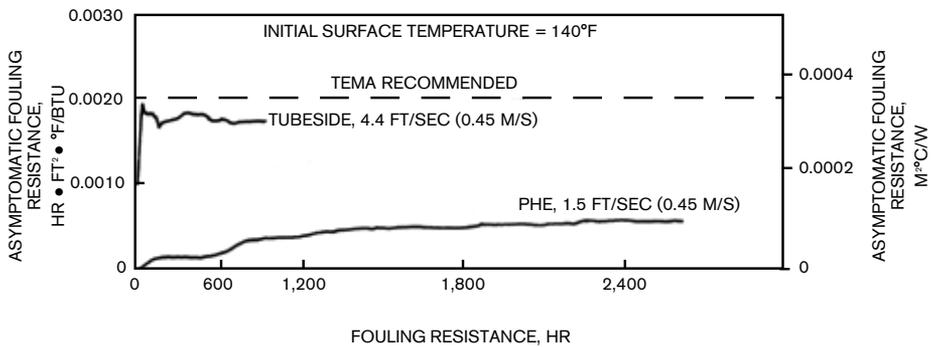


Figure 27. Comparison of tubeside and PHE fouling.

FOULING DEPOSIT ANALYSIS, PERCENT		
	Plate	Shellside
Loss on ignition	17	24
Phosphorus	12	9
Aluminum	6	7
Silicon	16	33
Calcium	15	7
Chrome	15	9
Iron	5	4
Zinc	13	8

Table 4. Comparison of deposit analyses from plate exchanger and shellside test exchanger for Series 5.

It is difficult to make meaningful comparisons between the fouling tendencies of different types of heat exchangers. Unless carefully done, the results are misleading. This is particularly true when attempting to compare the fouling experience in the PHE to that of tubeside operation as in the PFRU. The geometries, surface area, and operational characteristics are very different even though the cooling water may be the same. Such comparisons often are attempted using TEMA4-recommended fouling resistances.

TEMA recommends a fouling resistance of 0.002 hr • ft² • °F/Btu for the water system used. However, the maximum fouling resistance measured in the PHE was less than 0.0005 hr • ft² • °F/Btu – only 25% of the TEMA recommendation. This confirms the earlier assumption that applying TEMA-recommended fouling resistances for a shell and tube exchanger to the PHE seriously handicaps the performance ratings. Because of the inherently high heat transfer coefficients, the effects of fouling resistances are more pronounced. Consider a PHE operating at 1.5 ft/sec. From Table 3, the fouled overall coefficient using a TEMA fouling resistance of 0.002 hr • ft² • °F/Btu is about one half of the overall coefficient using the measured fouling resistance of this investigation. This example illustrates the need for caution in using TEMA fouling recommendations for equipment other than shell and tube exchangers.

Figure 27 compares the performance of the PHE with typical tubeside data and with the TEMA-recommended fouling resistance.

A direct comparison of the PHE and the shellside exchangers can be made through the deposit analyses. A sample analysis is given in Table 4. The amounts of phosphates, calcium, chrome, and zinc are somewhat higher for the PHE. The operation of the PHE at a higher surface temperature than the shellside units resulted in more crystallization fouling. However, the amount of silicon for the shellside unit is much higher than for the PHE. The high silicon concentration is due to sedimentation in the baffle-shell corners of the bundle, while the high turbulence promoted by the corrugated plates in the PHE minimizes sedimentation fouling. In heavily sediment-laden waters, the PHE would be especially superior.

CONCLUSIONS

Although the tests were performed on one water system only, experience with tubular data indicates that the overall trends generally are valid, i.e., plate exchangers should be designed to substantially lower values of fouling than would be used on tubular equipment for the same stream. However, the comparison is not always that straight forward, as flow velocity itself is not a valid criterion. The wall shear stress in a PHE operating at 2.8 ft/sec. is equivalent to that in a tubeside exchanger operating at 8.2 ft/sec. On the basis of typical industrial velocities, it was found that the PHE at 1.5 ft/sec. fouls about one half as much as a tubeside exchanger operating at 5.9 ft/sec. Furthermore, because of the high heat transfer coefficients typical to the PHE, surface temperature may differ from those in tubular equipment and should be carefully watched.

The turbulence inducing corrugation pattern prevalent in heat exchanger plates produces very high local velocities. This results in high friction factors, and therefore, high shearing. This high shear, in turn, results in less fouling in plate exchangers than in tubular units. For cooling water duties, Heat Transfer Research Incorporated (HTRI) has shown that the plate exchanger fouls at a much lower rate than either the tubeside or shellside of tubulars. TEMA recommendations for fouling, therefore, should never be used for plate units since they probably are five times the value found in practice. In particular, the plate exchanger is far less susceptible to fouling with heavy sedimented waters. This is contrary to popular belief. It must be noted, however, that all particulate matter must be significantly smaller than the plate gap and generally particles over 0.1 inch in diameter cannot be handled in any plate heat exchanger.

It has been shown that quoting a high fouling resistance can negate a plate heat exchanger design by adding large amounts of surface and thereby overriding the benefits of the high coefficients. Therefore, fouling design resistance should be chosen with care, keeping in mind that with a paraflow unit it always is possible to add or subtract surface to meet exact fouling conditions.

As a general policy, HTRI as the project operator disclaims responsibility for any calculation or design resulting from the use of the data presented.

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COMPARING PLATE AND TUBULAR EXCHANGERS

In forming a comparison between plate and tubular heat exchangers, there are a number of guidelines which will generally assist in the selection of the optimum exchanger for any application. In summary, these are:

1. For liquid/liquid duties, the plate heat exchanger usually has a higher overall heat transfer coefficient and often the required pressure loss will be no higher.
2. The effective mean temperature difference will usually be higher with the plate heat exchanger.
3. Although the tube is the best shape of flow conduit for withstanding pressure, it is entirely the wrong shape for optimum heat transfer performance since it has the smallest surface area per unit of cross sectional flow area.
4. Because of the restrictions in the flow area of the ports on plate units, it is usually difficult, unless a moderate pressure loss is available, to produce economic designs when it is necessary to handle large quantities of low-density fluids such as vapors and gases.
5. A plate heat exchanger will usually occupy considerably less floor space than a tubular for the same duty.
6. From a mechanical viewpoint, the plate passage is not the optimum, and gasketed plate units are not made for operating pressures much in excess of 300 PSIG.
7. For most materials of construction, sheet metal for plates is less expensive per unit area than tube of the same thickness.
8. When materials other than mild steel are required, the plate will usually be more economical than the tube for the application.
9. When mild steel construction is acceptable and when a close temperature approach is not required, the tubular heat exchanger will often be the most economic solution since the plate heat exchanger is rarely made in mild steel.
10. Plate heat exchangers are limited by the necessity that the gasket be elastomeric. Even compressed asbestos fiber gaskets contain about 6% rubber. The maximum operating temperature therefore is usually limited to 500°F.

HEAT TRANSFER COEFFICIENTS

Higher overall heat transfer coefficients are obtained with the plate heat exchanger compared with a tubular for a similar loss of pressure because the shellside of a tubular exchanger is basically a poor design from a thermal point of view. Considerable pressure drop is used without much benefit in heat transfer due to the turbulence in the separated region at the rear of the tube. Additionally, large areas of tubes even in a well designed tubular unit are partially bypassed by liquid. Thus, low heat transfer areas are created.

Bypassing in a plate exchanger is less of a problem and more use is made of the flow

separation, which occurs over the plate troughs, since the reattachment point on the plate gives rise to an area of very high heat transfer.

For most duties, the fluids have to make fewer passes across the plates than would be required through tubes or in passes across the shell. Since a plate unit can carry out the duty with one pass for both fluids in many cases, the reduction in the number of required passes means less pressure lost due to entrance and exit losses, and consequently, more effective use of the pressure.

	Tubular	Plate
Size	3 shells in series: 20' tubes, 1 shellside pass, 4 tube passes, 2238 tubes, 12,100 sq ft area	R10 Paraflow with 313 plates arranged for 3 process passes and 3 service passes, 3630 sq ft of area
Overall coefficient	270 Btu/hr • ft ² • °F	700 Btu/hr • ft ² • °F
MTD	7.72°F	9.0°F
Pressure loss	10 PSI/19 PSI	10.8 PSI/8.8 PSI

Figure 28. DUTY: Demineralized water/sea water. To cool 864,000 lb/hr of water from 107°F to 82.4°F using 773,000 lbs/hr of 72°F sea water.

MEAN TEMPERATURE DIFFERENCE

A further advantage of the plate heat exchanger is that the effective mean temperature difference is usually higher than with the tubular unit. Since the tubular is always a mixture of cross and contraflow in multi-pass arrangements, substantial correction factors have to be applied to the log mean temperature difference. In the plate heat exchanger where both fluids take the same number of passes through the unit, the LMTD correction factor is usually in excess of 0.95. As is illustrated in Figure 28, this factor is particularly important when a close or relatively close temperature approach is required.

In practice, it is probable that the sea water flow rate would have been increased to reduce the number of shells in series if a tubular had to be designed for this duty. While this would reduce the cost of the tubular unit, it would result in increased operating costs.

Case study	Fluid A lbs/hr	Fluid B lbs/hr	Duty °F	Tubular design pressure loss			Plate design pressure loss			Type
				A	B	ft ²	A	B	ft ²	
A	13,380 hydrocarbon	53,383 water	320 A 120 120 B 92	.42	3.05	301	.3	2.2	92	HX
B	864,000 water	77,300 sea water	107 A 82.4 99.1 B 71.6	19	19	12,100	10.8	8.8	3,630	R10
C	17,500 solvent	194,000 water	140 A 104 95 B 79	2.9	4.5	1,830	3.0	3.7	445	R5
D	148,650 desalter effluent	472,500 salt water	222 A 100 106 B 68	10.8	8.0	1,500	4.5	10	380	R4

Figure 29. Case studies of plate heat exchangers and tubular designs — all tubular designs were carried out with the aid of the HTRI program ST3.

DESIGN CASE STUDIES

Figure 29 covers a number of case studies on plate vs. tubular design. It demonstrates the remarkable heat transfer performance that can be obtained from paraflow units. Even for low to moderate available pressure loss, the plate heat exchanger usually will be smaller than a corresponding tubular.

Because of the high heat transfer rates, the controlling resistance usually is fouling, so an allowance of between 20% to 50% extra surface has been made based on APV experience.

One limitation of the plate heat exchanger is that it is rarely made in mild steel; the most inexpensive material of construction is stainless. Therefore, even when the plate surface requirement is much lower than the tubular on some heat transfer duties, the tubular will be less expensive when mild steel construction is acceptable. If a close temperature approach is required, however, the plate unit always will cost less. Where stainless steel or more exotic materials are required for process reasons, the plate unit usually will cost less.

PHYSICAL SIZE

One important advantage of the plate over a tubular unit is that for a particular duty, the plate heat exchanger will be physically smaller and require far less floor space. This is shown most graphically in Figure 30 where a Series R405 paraflow is being installed next to tubular units that need twice the amount of space for the identical duty. It is further illustrated in the volumetric comparisons of Figure 31.

If the tightened plate pack of a Paraflow is regarded as a rectangular box, each cubic foot contains from 50 ft² to 100 ft² of heat transfer area according to the type of plate used. Allowing for metal thickness, the contained liquid is some 80% of this volume. Thus, the total of both heating and cooling media is about five gallons. Expressed in another way, the liquid hold up per square foot for each stream varies according to plate type from about 0.06 gallons down to 0.03 gallons.



Figure 30. Picture of Series R405 Paraflow being installed.

Plate			Tube			
Plate pitch	Heat transfer area per cubic foot of exchanger	Liquid contained per sq ft of heat transfer area	Tube dia.	Ratio tube pitch tube dia.	Heat transfer area per cubic foot of exchanger	Liquid containment per sq ft of heat transfer area (average of both sides)
1/4"	50 ft ²	0.06 gal	2"	1.5	10 ft ²	0.32 gal
1/8"	100 ft ²	0.03 gal	1/2"	1.5	40 ft ²	0.085 gal

Figure 31. Volumetric comparison, plate vs. tube.

By comparison, one cubic foot of tubular exchanger of equilateral triangular pitch with a tube pitch/tube diameter ratio of 1.5 has a surface area of 10 ft² for 2" in OD tubes or 40 ft² for 1/2" GD tubes. The average contained liquid is proportionately 0.27 gal/ft² down to 0.07 gal/ft² of heat exchanger area with no allowance for the headers. If the heat transfer coefficient ratio between plate/tubular is conservatively taken as two, the plate exchanger volume to meet a given duty varies from 1/10th to 1/5th of that of the tubular. For a lower tube pitch/tube diameter ratio of 1.25, the comparison becomes 1/7th to 1/4th. These facts demonstrate why the paraflow plate heat exchanger is referred to as "compact."

THERMAL LIMITATIONS

While it would appear offhand that the plate heat exchanger always provides a better performance at usually a lower price than the tubular exchanger, consideration must be given to the thermal as well as mechanical limitations of the plate type machine. These usually are based on allowable pressure loss.

For single phase liquid/liquid duties, the plate heat exchanger can be designed for moderately low pressure loss. However, if the pressure loss across any plate passage which has liquid flowing downward is lower than the available liquid static head, the plate will not run full and performance therefore will be reduced. This is termed low plate rate. Use of a plate below the minimum plate rate is inadvisable since it causes a wastage of surface area and results in unreliable operation. It is, however, possible to function below the minimum plate rate in a single pass arrangement by making sure that the low plate rate is operated with a climbing liquid flow.

These problems are not quite so severe with a tubular exchanger, and therefore, operation at a moderately lower available pressure loss is possible.

CONCLUSION

To summarize, the gasketed plate heat exchanger generally will be the most economical heat exchanger for liquid/liquid duties providing the material of construction is not mild steel and providing the operating temperatures and pressures are below 500°F and 300 PSIG respectively. For other types of duties such as gas cooling, condensation or boiling, the plate heat exchanger can be a very economical type of unit if the pressure loss allowed is sufficient to utilize the very high heat transfer performance characteristics of the plate.

RECOVERY OF PROCESS HEAT

Since it is quite clear that never again will energy be as inexpensive as in the past, it therefore is necessary to conserve this natural resource – to recover more of the process heat that currently is dissipated to waterways and the atmosphere. Some of this heat can be recovered with the aid of high efficiency heat exchangers, which can economically operate with a close temperature approach at relatively low pumping powers. One type of unit that is particularly suited for this duty is the APV paraflow plate heat exchanger. For many applications this equipment can transfer heat with almost true countercurrent flow, coupled with high coefficients, to provide efficient and inexpensive heat transfer.

Unfortunately, the plate heat exchanger has been considered by many chemical engineers to be suitable only for hygienic heat transfer duties. This, of course, is not so. Nowadays, many more plate-type units are sold for chemical and industrial use than are sold for hygienic applications. A further mistake is the claim that the plate heat exchanger can be used only for duties when the volumetric flows of the two fluids are similar. Again, this is not true, although it must be stated that the plate heat exchanger is at its most efficient when flows are similar.

PLATE OR TUBULAR?

Since plate and tubular heat exchangers are the most widely used types of heat transfer equipment, it is helpful to draw a brief comparison of their respective heat recovery capabilities for the energy-conscious plant manager.

While the plate heat exchanger does have mechanical limitations with regards to withstanding high operating pressures above 300 PSIG, it is thermally more efficient than shell and tube units, especially for liquid/liquid duties. In many waste heat recovery applications, however, both pressure and temperature generally are moderate. The plate type unit is an excellent choice since its thermal performance advantage becomes very significant for low temperature approach duties. Higher, overall, heat transfer coefficients are obtained with the plate unit for a similar loss of pressure because the shellside of a tubular basically is a poor design from a thermal point of view. A certain pressure drop is used without much benefit in heat transfer on the shellside due to the flow reversing direction after each cross pass. In addition, even in a well designed tubular heat exchanger, large areas of tubes are partially bypassed by liquid and areas of low heat transfer are thus created. Conversely, bypassing of the heat transfer area is far less of a problem in a plate unit. The pressure loss is used more efficiently in producing heat transfer since the fluid flows at low velocity, but with high turbulence, in thin streams between the plates.

For most duties, the fluids also have to make fewer passes across the plates than would be required either through tubes or in passes across the shell. In many cases, the plate heat exchanger can carry out the duty with one pass for both fluids. Since there are fewer passes, less pressure is lost due to entrance and exit losses, and the pressure is used more effectively.

A further advantage of the plate heat exchanger is that the effective mean temperature difference usually is higher than with the tubular. Since in multi-pass arrangements the tubular is always a mixture of cross- and contra-flow, substantial correction factors have to be applied to the log mean temperature difference. In the plate unit for applications where both fluids take the same number of passes through the exchanger, the LMTD correction factor approaches unity. This is particularly important when a close or even relatively close temperature approach is required.

THERMAL PERFORMANCE DATA

Although the plate heat exchanger now is widely used throughout industry, precise thermal performance characteristics are proprietary, and thus, unavailable. It is possible, however, to size a unit approximately for turbulent flow liquid/liquid duties by use of generalized correlations which apply to a typical plate heat exchanger.

The basis of this method is to calculate the heat exchanger area required for a given duty by assuming that all the available pressure loss is consumed and that any size unit is available to provide this surface area.

For a typical plate heat exchanger, the heat transfer can be predicted in turbulent flow by the following equation (1):

$$(1) \quad hDe/k = 0.28(GDe/\mu)^{0.65} (C\mu/k)^{0.4}$$

$$(2) \quad f = 2.5(GDe/\mu)^{-0.3}$$

$$(3) \quad \Delta P = 2fG^2L/grDe$$

The pressure loss can be predicted from equations (2) and (3). Obviously, equation (1) cannot accurately represent the performance of the many different types of plate heat exchangers that are manufactured. However, plates that have higher or lower heat transfer performance than given in equation (1) usually will give correspondingly higher or lower friction factors in equation (2). Experience indicates that the relationship for pressure loss and heat transfer is reasonably consistent for well designed plates. In the Appendix, equations (A1) — (A3) are further developed to show that it is possible for a given duty and allowable pressure loss to predict the required surface area. This technique has been used for a number of years to provide an approximate starting point for design purposes and has given answers to $\pm 20\%$. For accurate designs, however, it is necessary to consult the manufacturer.

HEAT RECOVERY DUTIES

In any heat recovery application, it always is necessary to consider the savings in the cost of heat against the cost of the heat exchanger and the pumping of fluids. Each case must be treated individually since costs for heat, electricity, pumps, etc. will vary from location to location.

One point is obvious. Any increase in heat recovery, and thus heat load, results in a decrease in LMTD and considering a constant heat transfer coefficient, subsequently in the cost of the exchanger. This effect is tabulated in Figure 32. Because the cost of an exchanger increases considerably for relatively small gains in recovered heat above the 90% level, such applications even with the plate heat exchanger must be closely studied to verify economic gain. The economic break-even point is far lower for a tubular exchanger. Situations where it is advantageous to go above 90% recovery usually involve duties where higher heat recovery reduces subsequent heating or cooling of the process stream. High steam or refrigeration costs, therefore, justify these higher heat recoveries.

Heat recovered (Btu/h)	Temp. (°F)	12 lb/in ² Pressure loss						25 lb/in ² Pressure loss				
		LMTD	Factor	Actual MTD	Area (ft ²)	Price	LMTD	Factor	Actual MTD	Area (ft ²)	Price	
3,600,000	200→140	40	0.985	39.4	85	5,500	40	0.985	39.4	74	5,400	
	160←100											
4,500,000	200→125	25	0.975	24.4	197	6,450	25	0.980	24.5	172	6,250	
	175←100											
5,100,000	200→115	15	0.965	14.4	425	8,400	15	0.965	14.4	362	7,850	
	185←100											
5,400,000	200→110	10	0.95	9.5	739	11,000	10	0.92	9.2	629	10,100	
	190←100											
5,700,000	200→105	5					5	0.83	4.16	1580	18,100	
	195←100											

hot liquid 60,000 lb/h of water at 200°F
cold liquid 60,000 lb/h of water at 60°F

Figure 32. Effect of percentage heat recovery on PHE cost.

As shown in Figure 33, the cost of increasing heat recovery from 85% to 90% at a constant pressure loss of 12 lbs/in² is \$2600. From a practical standpoint, going from 90% to 95% requires a significantly higher pressure loss and nearly doubles the exchanger cost. However, even with this 95% heat recovery and assuming steam costs at \$6.00/1,000 lbs, payback on the plate heat exchanger would take 530 hours.

Thus, the plate heat exchanger provides a most economic solution for recovering heat. This degree of heat recovery cannot be economically achieved in a tubular exchanger since the presence of crossflow and multipass on the tube side causes the LMTD correction factors to become very small, or alternately, requires more than one shell in series. This is shown in the Figure 34 comparison.

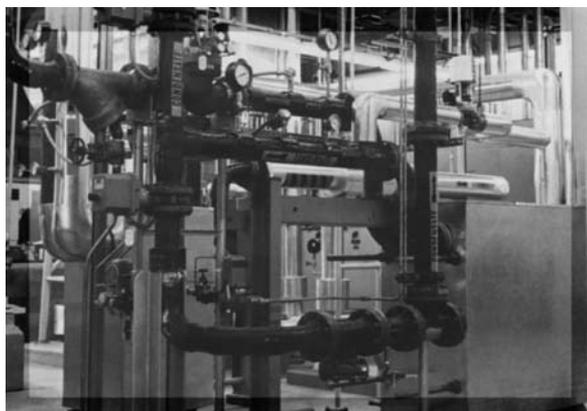


Figure 33. Two shrouded parafflow units provide an optimum return on investment with energy and cost saving regeneration of 88% and 81%.

Duty: To heat 1,300,000 lb/h of water from 73°F to 97°F using 1,300,000 lb/h of water at 107°F. Available pressure loss 15 lb/in² for both streams. Heat recovery = 70.5%. Number of heat transfer units (HTU) = 2.4.

	Plate	Tubular	
Heat transfer area	4,820	13,100	ft ²
Heat transfer coefficient (<i>clean</i>)	734	386	Btu/hr • ft ² • °F
Heat transfer coefficient (<i>dirty</i>)	641	271	Btu/hr • ft ² • °F
Effective mean temperature difference	10.0	8.7	°F
Pressure drop: <i>hot fluid/cold fluid</i>	4.6/4.7	15/15	lb/in ²
Fouling resistance: <i>hot fluid/cold fluid</i>	0.0001/0.0001	0.0005/0.0005	(Btu/hr • ft ² • °F) ⁻¹
Pass arrangement	1/1	4 tube side 3 shells in series	baffled
Approximate price	\$95,000 Stainless Steel	\$135,000 Mild Steel	

This example demonstrates that quite high heat transfer coefficients can be obtained from a PHE with only a moderate pressure loss.

Figure 34. Heat recovery — comparison between plate and tubular heat exchangers.

As detailed, this example illustrates that the plate heat exchanger has considerable thermal efficiency, and therefore, a price advantage over the tubular exchanger for a heat recovery of 70%. Since the overall heat transfer coefficient and the effective mean temperature difference both are much higher for the plate unit, reduced surface area is needed. Furthermore, because of crossflow temperature difference problems in the tubular, three shells in series were needed to handle the duty within the surface area quoted. Using only two shells would have resulted in a further 40% increase in surface area.

The small size of the plate heat exchanger also results in a saving of space and a lower liquid hold up. For this type of heat recovery duty, a stainless steel plate heat exchanger almost always will be less expensive than a mild steel tubular unit. Although the tubular

exchanger physically will be capable of withstanding higher temperatures and pressures, there is a considerable (and for the most part unnecessary) penalty to pay for these features both in price and size. For heat recovery duties in excess of 70%, the plate heat exchanger will become increasingly more economic than the tubular.

TYPICAL APPLICATIONS

One of the more common uses of regeneration is found in many of the nation's breweries where it is necessary to cool huge amounts of hot wort before it is discharged to fermentation tanks. Typical in scope is an operation where 850 barrels per hour of wort (220,000 lbs/hr) are cooled from 200°F to 50°F by means of 242,000 lbs/hr (R = 1.1:1) of water entering at 35°F and being heated to 165°F. The result: approximately 33,000,000 Btu/hr are saved, there is an excellent water balance for use throughout the brewery, and no water is discharged to the sewer.

For a dairy, regeneration usually involves the transfer of heat from pasteurized milk to cooler raw milk entering the system. After initially heating 100,000 lbs/hr of milk from 40°F to 170°F by high temperature, short time pasteurization, 90% regeneration permits cooling of the milk back down to 40°F with a savings of 11,700,000 Btu/hr. Only 10% of the total heat or cooling must be supplied and no cooling medium such as city water is used and discarded.

Chemically, there are many and varied regenerative applications. For desalination, APV has supplied a number of paraflows which are virtually perpetual motion machines. These units achieve 95% regeneration in heating 87,000 lbs/hr (175 GPM) from 70°F to 197°F while cooling a secondary stream flowing at 78,000 lbs/hr from 214°F to 73°F. Savings in the Btu load in this case are 11,050,000 per hour.

While flow rates in hot oil applications are quite low in comparison, temperatures are very high. It is possible to cool 400 GPH of vegetable oil from 446°F to 266°F while heating 400 GPH of oil from 200°F to 400°F with 80% regeneration.

And in the production of caustic soda where very corrosive product streams are encountered, paraflows with nickel plates are being used to cool 10,000 lbs/hr of 72% NaOH from 292°F to 210°F while heating 14,000 lbs/hr of 50% NaOH from 120°F to 169°F.

DOLLAR SAVINGS

To examine a hypothetical case of Paraflow regeneration from the viewpoint of efficiency and dollar savings, consider the following process duty:

Duty

Heat 100 GPM of fluid #1 from 40°F to 190°F while cooling 100 GPM of fluid #2 from 190°F to 40°F.

Under ordinary conditions, steam required to heat fluid #1 would be in the nature of 7,500 lbs/hr with an equivalent cooling requirement for the second stream of 625 tons of refrigeration. Using 90% regeneration, however, the energy needs are drastically reduced.

At this point in the process, fluid #2 in a conventional system has been cooled to only 90°F by means of 85°F city water and will require supplemental refrigeration for final cooling to 40°F. At the same time, fluid #2 in an APV regenerative system has been cooled to 55°F by means of 90% regeneration and must be cooled further to 40°F.

Fluid #1	Conventional	90% Regeneration
	Heat 100 GPM 40°F → 190°F	Heat 100 GPM 40°F → 175°F by cooling fluid #2 from 190°F → 55°F
Steam required	$50,000 \times 150 = 7,500$ lbs/hr 1,000 to heat to 190°F	$50,000 \times 15 = 750$ lbs/hr 1,000 supplemental heat to raise temp to 190°F
Assuming average steam cost:	\$8.00 per 1,000 lbs.	
steam cost/hr	$7.5 \times \$8.00 = \60.00 /hr	$0.75 \times \$8.00 = \6.00 /hr
annual water cost	$\$60 \times 24 \times 365 = \$525,600$	$\$6 \times 24 \times 365 = \$52,560$

Fluid #2	Conventional	90% Regeneration
	Cool 100 GPM 190°F → 40°F	Cool 100 GPM 190°F → 55°F by heating fluid #1 from 40°F → 175°F
Equivalent refrigeration required	$7,5000,000 = 625$ tons 12,000 to cool to 40°F	$750,000 \times 15 = 62.5$ tons 12,000 supplemental cooling to lower temp to 40°F
Using available 85°F city water	200 GPM to cool 190°F → 90°F $200 \times 60 = 12,000$ GPH	None None
Assuming average water cost:	\$0.07 per 1,000 gal	None
water cost/hr	$12 \times 0.07 = \$0.84$ /hr	None
annual water cost	$\$0.84 \times 24 \times 365 = \$7,360$	None

Duty	Cool from 90°F to 40°F	Cool from 55°F to 40°F
Supplemental refrigeration required	$50,000 \times 50 = 208.3$ tons 12,000	$50,000 \times 15 = 62.5$ tons 12,000
Power required	$208.3 \times 1.2 \times 0.746 = 186$ KWH	$62.5 \times 1.2 \times 0.746 = 56$ KWH
Assuming average electrical cost		0.08/KWH
Annual supplemental refrigeration cost	$0.08 \times 186 \times 24 \times 365 = \$130,350$	$0.08 \times 56 \times 24 \times 365 = \$39,245$

	Original Cost	Regenerative System Cost	Regenerative Savings
Steam	\$525,600	\$52,560	\$473,040
Cooling Water	\$7,360	None	\$7,360
Refrigeration	\$130,350	\$39,245	\$91,105
	\$663,310	\$91,805	\$564,145 annual savings

Note: Capital cost of a Paraflo plate heat exchanger for the above duty would be approximately \$30,000.

DEFINITION

The number of heat transfer units (HTU) is defined as the temperature rise of fluid one divided by the mean temperature difference. A further term for the HTU is the temperature ratio (TR).

i.e. $HTU = t_1 - t_2 / MTD$

APPENDIX

For heat transfer in a typical PHE with turbulent flow one can write the heat transfer performance in terms of a dimensionless Dittus Boelter type equation:

$$(A1) \quad hDe/k = 0.28(GDe/\mu)^{0.65} (Cr\mu/k)^{0.4} (\mu/\mu_w)^{0.14}$$

For applications in turbulent flow it is usually sufficiently accurate to omit the Sieder Tate viscosity ratio and therefore the equation reduces to:

$$(A2) \quad hDe/k = 0.28(GDe/\mu)^{0.65} (Cr\mu/k)^{0.4}$$

The pressure drop can be predicted in a similar exchanger by equations (A3) and (A4).

$$(A3) \quad F = 2.5(GDe/\mu)^{-0.3}$$

$$(A4) \quad \Delta EP = 2fG^2L/grDe$$

To solve the above equations for a particular duty it is necessary to know G, L, De and it is shown below how these can be eliminated to produce a general equation.

For any plate:

$$(A5) \quad G = m/Af$$

where m is the total mass flowrate and Af the total flow area.

De is defined by APV as four times the flow area in a plate divided by the wetted perimeter. Since the plate gap is small compared with the width then:

$$(A6) \quad De = 4 \bullet Af/\text{wetted perimeter}$$

where h1 and h2 are calculated from an empirical modification of the constants in equation (A8). The powers in equation (A8) are not modified.

Since As = L x wetted perimeter where As is the total surface area:

$$(A7) \quad L = As/4 \bullet Af De$$

and it is possible to eliminate L from equation (A4).

Similarly, by substituting G using equation (A5) in equations (A2), (A3) and (A4) — and then by rearranging the equations to eliminate Af — it is possible to arrive at equation (A8) for the film heat transfer coefficient.

$$(A8) \quad H = J2/J10.241 (m/As)^{0.0241} De^{-0.28}$$

$$\text{where } J1 = 0.25\mu^{0.3}/2gr\Delta EP$$

$$\text{and } J2 = 0.28(Cr\mu/k)^{0.4} k\mu^{-0.65}$$

That is, the film heat transfer coefficient is expressed only in terms of the surface area and equivalent diameter. A computer program solves equation (A8) using a constant value of De and using an empirical factor Z to account for port pressure loss and other deviations from equations (A2) to (A4). The equation is solved using the HTU approach where:

$$HTU = t1 - t2 /MTD = UAs/m1c1$$

That is, the number of heat transfer units (HTU) is the temperature rise of fluid divided by the mean temperature difference.

U is defined as:

$$1/U = 1/h1 + 1/h2 + \text{metal resistance}$$

Note that h1 and h2 are calculated from an empirical modification of the constants in equation (A8). The powers in equation (A8) are not modified.

PARAFLOWS IN SEA WATER COOLING SYSTEMS

The use of sea water as a cooling medium is far from new. It has been common practice with powered ships for many decades. It is a result of technical experiences from on board ships that many of the materials used in the handling of sea water and the practices related to its use have been developed. On shore, the extensive use of sea water is a somewhat later development, although it now is common practice at coastal sites and becoming even more so with the increasing shortage of good quality industrial cooling water. The rapid development of the global offshore industry and of amphibious chemical, steel and power plants will lead to an even more intensive use of this medium.

BASIC ESSENTIALS OF A SEA WATER COOLER

Sea water is a very complex liquid. It can vary in cleanliness and salinity. In addition, it is an excellent nutrient medium supporting many forms of animal and plant life. Regardless of variations, two common potential problems can arise in all cooling operations. Corrosion and fouling, particularly biological fouling, affect both downtime and maintenance costs. Therefore, it is essential that cooling equipment, which is intended to employ sea water, should embody a number of basic features. Coolers should be constructed from corrosion-resistant materials. They should minimize “dead” areas, i.e., regions of low velocity which could permit organisms to settle and develop, or silt to deposit. They should also be designed to discourage surface fouling.

The traditional materials for handling sea water are copper alloys, such as aluminum brass, and the cupronickels specifically developed for marine use. Even with marine applications, there are disadvantages to these materials, principally that of being erosion-prone. This limits the upper level of velocities that can be used in the design of the cooling equipment. Shipboard applications of sea water are relatively simple in that the medium to be cooled usually is non-corrosive, being either a lubricating oil or clean water used in the jackets and pistons of the ship’s prime mover.

For other industries, particularly chemical, the cooling problem is more complex. It frequently is found that a more sophisticated material has to be employed, not only to combat the corrosivity of the sea water but also that of the medium which is being cooled. The latter may be any of the wide range of process streams handled by a modern chemical plant. Typical of these materials are titanium, titanium/palladium alloy, Incoloy 825 and the various Hastelloys. Of these, the one which appears to be gaining the widest prominence is commercially pure titanium – although this is by no means a panacea and other materials often are needed. Unfortunately, metals or alloys which can successfully cope with this dual corrosion situation generally tend to be expensive. The list of attributes required of cooling equipment using sea water should include a “high rate of heat transfer” in order to minimize the extent of usage of these high-cost alloys.

One class of heat exchanger, the plate and frame type, in large measure possesses these requirements since it is designed to avoid pockets and crevices. This automatically fulfills one of the criteria for its use with sea water and other biomedica.

PLATE HEAT EXCHANGER

In operation, the plate heat exchanger pack is tightly clamped between the head and follower, but the unit is easily dismantled for cleaning or maintenance. It also has the further advantage of flexibility because a heat transfer surface can either be added to or deleted from the original configuration.

When the plate pack is assembled the geometry of the flow channel on both sides of the plates is identical. Thermal and pressure drop predictions can be made much more accurately, than for instance, in the case of shell and tube (S and T) heat exchangers. In tubular exchangers the shellside geometry, in particular, is so complex that large margins of safety have to be allowed for the film coefficient on this side of the unit. Furthermore, geometrical similarity of the plate heat exchanger passages and equal access to both sides of the plates removes the tubular exchanger dilemma of which shall be the shellside fluid and which the tube side.

Case	Flow rate - lbs/hr		Temperature profile °F	Surface area (ft ²) and P (PSI)					
	Fluid A	Fluid B		S&T			PHE		
				ft ²	PSI A	PSI B	ft ²	PSI A	PSI B
1	13,404 hydrocarbon	53,495 water	320 A 120 120 B 92	301	0.4	3	93	0.3	2.2
2	865,810 water	774,594 sea water	107 A 82 99 B 72	12,110	19	19	3,633	11	8.7
3	17,537 solvent	19,445 water	140 A 104 95 B 79	1,830	2.9	44	446	2.9	3.6
4	198,966 desalter effluent	473,498 sea water	222 A 100 106 B 68	1,502	10.9	8	380	4.4	10.2

Table 5. Comparison of areas and pressure drops (S and T versus PHE).

Without doubt, the most important feature of plate units in the context of this paper is that the overall heat transfer coefficient achieved on liquid-to-liquid heat transfer duties is of two to three times that which can be achieved even in a well-designed tubular exchanger. The criterion of economy in the use of costly constructional materials is readily met. Typical overall design coefficients for a sweet water/sea water PHE would be in the range of 600 to 800 Btu/hr • ft² • °F. Table 5 compares areas and pressure drops for S and T, and PHE designs for typical industrial duties – two of which are for a sea water cooling function. The S and T designs were prepared with the assistance of the HTRI (Heat Transfer Research Inc.) S.T.3 program.

PROBLEMS WITH DIRECT SEA WATER COOLING

Plate heat exchangers are, of course, extensively used in many process industries for direct sea water cooling, i.e., the employment of sea water on one side of the cooling surface and product liquid on the other. The main advantage is that the full difference between the product temperatures and those of the sea water can be used, resulting in a smaller heat transfer area for a given thermal duty. It is rare, that any technological system possesses only advantages. Direct cooling, while widely practiced, is no exception to this rule. A typical chemical industry example, the cooling of high strength sulphuric acid by sea water, illustrates this point. Over certain temperature ranges product corrosiveness is such that a relatively unsophisticated and moderately priced material, such as 316 stainless steel, could be quite readily used. At these temperatures, 316 stainless steel will corrode in sea water so a more sophisticated material is needed. High nickel alloys (like Incoloy 825) which probably represent the next step up in sophistication and price, usually are avoided. While they are well suited to contact with the sulphuric acid, there still is some risk on the sea water side. Titanium, next upward in price, would be ideal for sea water but does not sufficiently resist acid corrosion. Therefore, the final choice (Figure 35) often has to be with an alloy such as Hastelloy "C". This not only is the most expensive of those mentioned but also has a comparatively low thermal conductivity. In addition to a higher cost per pound, it sometimes is necessary with Hastelloy "C" to provide a greater area to do the same work. Differences in metal thermal conductivity on high efficiency heat exchangers can be quite significant. Stainless steel and titanium have conductivities of the same order of magnitude – about $112 \text{ Btu/hr} \cdot \text{ft} \cdot ^\circ\text{F}$, but Hastelloy "C" is about half this at $62.4 \text{ Btu/hr} \cdot \text{ft} \cdot ^\circ\text{F}$. For normal plate thicknesses and typical PHE film coefficients of about $2025 \text{ Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$ for water and $670 \text{ Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$ for strong acid, the additional resistance of Hastelloy "C" would result in an increase in area of 7% to 10% for this acid cooler.

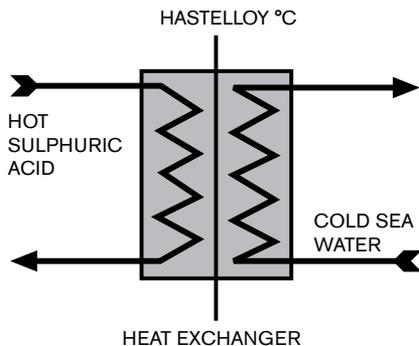


Figure 35. Direct cooling.

ALTERNATE SOLUTION – INDIRECT COOLING

This corrosion dilemma can be solved by indirect cooling, i.e., by interposing a closed circuit carrying clean and non-corrosive water between the acid and the sea water (Figure 36). The closed loop water initially flows through the sulphuric acid coolers, which can be of any material that suits the corrosive conditions, then rejects the heat gained to other coolers – which generally are fabricated of titanium to resist sea water.

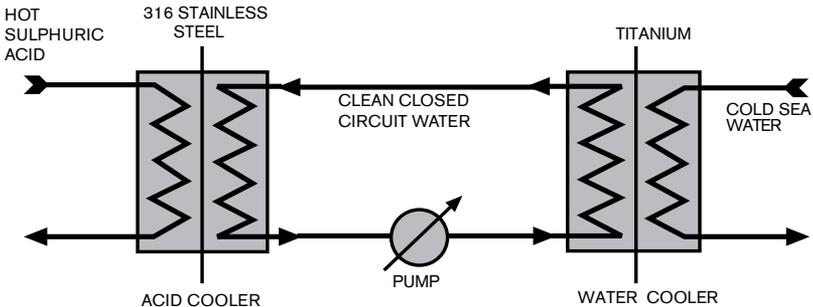


Figure 36. Indirect cooling.

As a bonus, the acid coolers are not subject to cooling water fouling and can remain on stream longer without cleaning.

The disadvantages are that this system requires a pump for the closed loop water, and that the temperature differences between acid and sea water have to be apportioned between the process cooler and the clean water/sea water cooler. There is some compensation for this loss of temperature difference since less fouling allowance must be built into the acid cooler design, and in this particular example, by being able to revert to a material with a higher thermal conductivity. For both acid and clean water/sea water heat exchangers, it is essential that high efficiency liquid-to-liquid units be chosen. This is where PHEs are finding widespread application.

THE HUNTERSON “B” POWER STATION

The acid cooling example illustrates the philosophy behind the increasing adoption of indirect cooling systems for local problem areas within many different types of industries. This applies to nuclear or fossil-fired power stations, chemical or fertilizer plants, offshore oil platforms, or pulp and paper plants that require substantial quantities of cooling water.

The choice of closed loop cooling for the reactor and auxiliary systems for Hunterston “B” nuclear power station of the South of Scotland Electricity Board is a typical case. Located at a coastal site, the natural choice of cooling medium is sea water. Corrosive liquids, however, cannot be allowed into sensitive areas. All reactor cooling circuits must be of extremely high integrity and must utilize water with bland corrosion and fouling characteristics. It is common nuclear plant practice for this to be demineralized water which flows in a closed circuit and rejects the heat to sea water via heat exchangers. Upon completing all tests, the station was equipped with twelve full-scale units fitted with titanium plates. The exchangers were on intermittent duty for two years during commissioning, and went on to full load early in 1976. Since then, they have been in continuous operation.

FUNDAMENTAL PRECAUTIONS

Correct sea water treatment and management is essential in any such system. Blockage by marine debris must be avoided and bio-fouling reduced to acceptable levels. This means adequate straining of the sea water and chemical dosing.

Trash catchers with 2 in. mesh usually are the first stage of straining at a water inlet, followed by finer levels of straining nearer to the point of usage. At Hunterston, this takes the form of backwashed rotary strainers, followed by smaller duplex-in-line units. Finally, at the exchangers there are the specially designed strainers referred to earlier. These employ expanded titanium mesh with the usual diamond-shaped hole axes that are approximately 0.125 in • 0.25 in (3.2 mm • 6.4 mm).

Chlorine injection, which is the normal method of chemical dosing, may be intermittent (shock dosing), continuous, or a combination of both dependent upon the type of bio-fouling encountered. Concentration levels are determined by sea water temperatures and the presence of neutralizing effluents in the sea water.

The general intention is to have a residual of 0.1 to 0.2 ppm beyond the last point requiring protection. For continuous dosing, the injection level usually is 1 to 2 ppm above natural water demand in temperate climates and 3 to 4 ppm in tropical areas. For shock dosing, the corresponding levels would be about 5 to 6 ppm and 8 to 10 ppm, both maintained for about 15 minutes every six to eight hours. The selection of chlorine as a control medium causes no problem when titanium is used but could cause considerable trouble with many other materials. It is common practice for chlorination to be effected by a chlorine generator to avoid the potential risks associated with storage of substantial quantities of liquid chlorine.

FOULING

Biological fouling, of course, is not the only type to be expected in heat exchangers. Taborek et al² and Fischer et al³ classify fouling into six groups:

- Crystallization
- Silting (particulate sedimentation)
- Chemical reaction and polymerization
- Coking
- Organic material growth (i.e., bio-fouling)
- Corrosion

Depending upon the type of duty, these can occur singly or in almost any combination. For sea water plate heat exchangers, however, it is possible to eliminate at least three. The choice of titanium removes the problem of corrosion, and at the same time, neither polymerization nor coking is a feature of this type of cooling duty. Cooper and Sutor⁴ observe that the plate heat exchanger fouls less than the shell and tube sides of tubular exchangers under typical operating conditions and also is much less sensitive to silt fouling. Other than biological fouling which already has been discussed, the remaining

type is crystallization.

Liquid hold up volumes of plate heat exchangers are very small. Even in the largest units, the particle dwell time for this type of duty is only three to five seconds, so that crystallization, which requires a substantial time to develop – is reduced. In the general chemical industry, there are many examples of plate heat exchangers being deliberately selected for conditions where crystallization would be known to occur in any other class of equipment. Thus, although the plate type unit is not totally immune to crystalline fouling, the rate of formation is much slower. In most cases of mixed fouling, the plate unit reaches an asymptotic fouling level after a relatively short operating time.

CLOSED CIRCUIT COOLING

Using closed circuit cooling systems (Figure 37) to serve complete installations has gained widespread acceptance and is of particular value at the plant design stage. It gives greater freedom of choice for construction materials and permits more economical process equipment design due to the virtual elimination of cooling water fouling allowances. A further advantage of closed circuit cooling sometimes is overlooked but nevertheless is important. It is common practice to design the central cooler station for the “worst case,” i.e. summer temperatures, so that cooler sizes are fixed by this criterion. By simply recycling some of the heated sea water, it is possible to maintain “summer” conditions throughout the year, thereby avoiding seasonal fluctuations in the temperature of the closed circuit water. This is important in some installations such as closed circuit water used for jacket water cooling of large diesel power sets.

It cannot be denied that handling sea water is troublesome in general, not only in process plants but also in the sea water pipelines. From the viewpoint of subsequent maintenance, there is a distinct advantage in keeping these pipe runs as small in diameter and as short as possible. The high, overall, heat transfer coefficients of plate heat exchangers – coupled with the fact that most are designed for almost 100% true countercurrent flow – results in less sea water being required in the central coolers than if any other type of heat exchanger was used. Sea water piping and pumps, therefore,

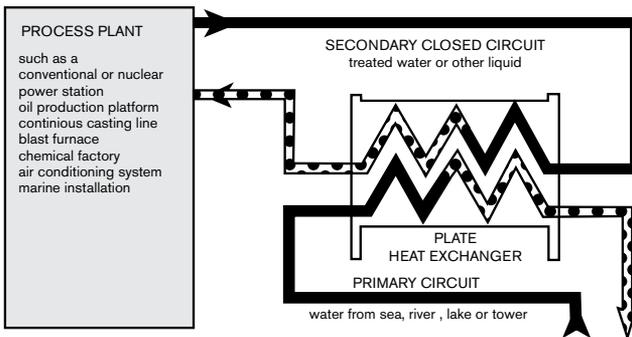


Figure 37. Closed circuit cooling system.

may be smaller.

There are a number of plants currently under consideration where the process installation will be about two to three miles inland from the coast. If direct sea water cooling is used, the problems of four to six miles of flow and return sea water culvert are obvious. Instead, these plants are being designed with the central cooling station at the shore. Sea water pipelines are short and of relatively small diameter. Since the pipelines between shore and the plant will carry only clean non-corrosive water, a relatively trouble-free installation will result.

Figure 38 shows a plate heat exchanger installation in the Middle East during the construction of a new ammonia/urea fertilizer plant. The 13 exchangers act as central coolers for the whole of the process plant so that sea water piping extends only from the intake through the treatment plant, then to the coolers and back to the sea. The



Figure 38. Ammonia/urea fertilizer plant under construction with 13 plate heat exchangers as central coolers.

position of the underground manifolds for the sea water and closed circuit water flow and return can be located easily in the photograph, the stub pipes for subsequent connection to the heat exchangers being plainly visible.

These coolers provide almost 40,000 GPM of recirculating cooling water for the main plant process items. As in the case of the sulphuric acid cooler, isolation of sea water from the process plant enabled the designers to select materials to suit corrosion characteristics of process streams being handled at any point, and generally resulted in lower cost materials being used. With no corrosion problems at the central coolers, any fouling which does occur can be due only to the fouling characteristics, if any, of the process streams.

COSTS

Reference has been made to the plate heat exchanger's economic use of the more expensive metals and alloys. Figure 39 shows relative costs per unit area of stainless steel PHEs against three different material combinations for tubular units. On this basis, PHE surface is always less expensive than any tubular unit partially or wholly

constructed of stainless steel. The graph indicates that the unit area cost of a mild steel shell and tube exchanger is lower than that of a plate heat exchanger with stainless steel surfaces. However, since smaller areas are needed for plate type exchangers, it can happen on a duty-for-duty basis that the plate unit sometimes is less expensive than an all mild steel tubular materials.

Where more sophisticated materials are mandatory for both classes of exchangers, the price differential rapidly widens in favor of the PHE – by a factor that can at times exceed 4:1, as was the case for the ammonia/urea fertilizer plant previously mentioned. World Bank financing was involved and international bidding was by two plate and five tubular exchanger manufacturers. Both plate heat exchanger bids were for titanium construction. The lowest priced tubular offer, which used mild steel shell and aluminum brass tubes, was 2.2 times the PHE prices. The lowest priced titanium tubed units were 4.1 times the PHE prices.

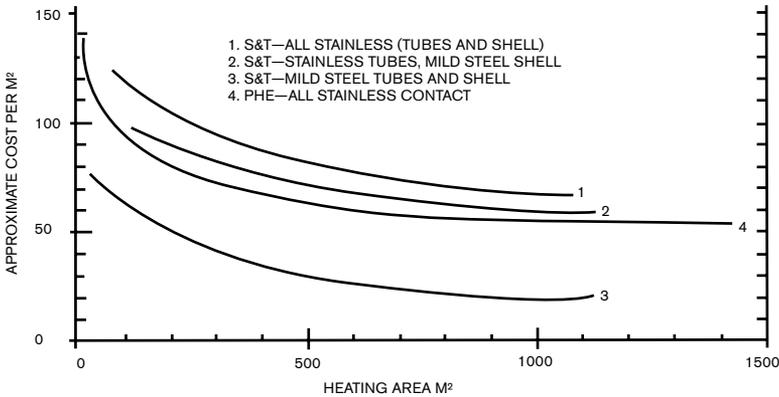


Figure 39. Relative costs: plate vs. tubular exchangers.

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- 3 Fischer, P.; Suito, J. W.; Ritter, R.B. – “Fouling Measurement Techniques,” Chemical Engineering Progress, pages 66-72, July 1975.
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CORROSION AND HEAT TRANSFER

The chemical industry and certain segments of food processing have to deal with widely ranging corrosive processes. Engineers must be particularly conscious of the need for detection and replacement of corroded sections of equipment. The ease with which this is achieved with the plate heat exchanger must, in no small part, account for the widespread acceptance of this method of heat transfer. However, with careful attention to detail at the design stage and observation and implementation of the correct operating and maintenance schedules, any need for plate replacement because of corrosion should be minimized.

As described in the initial section of the APV Heat Transfer Handbook, "The Paraflow and its Principles," methods of construction and operating characteristics of the plate heat exchanger are fundamentally different from the more traditional forms of heat transfer equipment, such as the shell and tube heat exchanger. Since the configuration and geometry of the plate heat transfer surfaces are unique, new material specification philosophy has to frequently be adopted when considering materials of construction if a plate exchanger will be operated on a corrosive process or service stream.

CATERING FOR GENERAL CORROSION

Since the flow plates of a plate heat exchanger are quite thin (0.024 in./0.036 in.) in contrast to other heat transfer surfaces (0.048 in. and greater), corrosion allowances normally quoted for chemical plants in standard reference books tend to be meaningless. Such works will give a grade-A rank to a corrosion rate of 0.005 in./year or less – a rating which is realistic when considering material ten times this thickness. However, a corrosion of 0.005 in./year with metal of 0.024 in. nominal thickness represents over 50% reduction in thickness in just three years of use. To most chemical engineers, this would be unacceptable. As a general guideline, the maximum permissible corrosion rate for a plate heat exchanger is 0.002 in./year, and at this level the concept of general corrosion in plate units is brought into line with normally accepted levels for other pieces of process equipment. In a specific corrosive environment, the implications are that a change from tubular exchangers to plate units may necessitate an upgrading of the alloy used for the heat transfer surface. In spite of this occasional need for a more expensive material of construction, the advantage of using thin-gauge material and the inherently high heat transfer coefficients of the plate heat exchangers frequently means that the actual cost is less. The added attraction of using a highly alloyed material is the extra corrosion protection built into the equipment. This permits excursions from normal design conditions into the more corrosive regimes which frequently are encountered in the startup and shutdown of a modern chemical plant.

MATERIALS FOR HEAT EXCHANGER PLATES

Before describing the various types of corrosion that may occur, a summary of the composition and corrosion characteristics of the metals and alloys from which heat exchanger plates are produced is appropriate.

Alloy	Cr	Ni	Mo	Cu	Major alloying elements					
					Fe	Ti	Pd	Ta	W	Others
Stainless steels										
304	18	10	0	0	Bal.					
316	18	10	2.5	0	Bal.					
AJJ 904L	20	25	0	0	Bal.					
Avesta 254SLX	20	25	4.5	2	Bal.					
Avesta 254SMO	20	18	6	0.75	Bal.					
Nickel alloys										
Nickel 200		99								
Hastelloy B-2	<1	Bal.	28		2					Co 2.5
Hastelloy C-276	15.5	Bal.	16		5				3.5	Co 2.5
AJJ Hastelloy C-2000		22.5	Bal.	17.5	1.6					
Hastelloy G-30	22.5	Bal.	6.5	2	20			2	1	Co 2.5
Inconel 625	18.5	Bal.	9		3					Cb 4
Incoloy 825	21.5	42	3	2	Bal.					
Monel 400		Bal.		30						
Others										
Titanium						>99				
Titanium-Palladium						>99	0.15			

Cb = Columbium
 Co = Cobalt
 Cr = Chromium
 Cu = Copper
 Fe = Iron
 Mo = Molybdenum
 Ni = Nickel
 Pd = Palladium
 Ta = Tantalum
 Ti = Titanium
 W = Tungsten

Table 6: Alloy compositions (nominal).

STAINLESS STEELS

304: The most inexpensive grade of the austenitic stainless steels. While offering a general corrosion resistance to a range of organic and inorganic products, it exhibits a poor resistance to sulphuric and hydrochloric acids. It is particularly susceptible to crevice corrosion induced by cooling waters containing chloride. The marginal cost savings which can be achieved by its use as an alternative to 316 stainless steel is rarely justified.

Wall temp	25°C	50°C	75°C	100°C
Cl — ppm (mg/l) #	100	75	40	<20

For unaerated, approximately neutral pH (7), flowing aqueous solutions, these figures offer a guide to maximum content of chloride.

316: A general purpose stainless steel suitable for use in a wide range of environments, and with limited applications, in sulphuric acid. The presence of molybdenum in the alloy confers a level of resistance to chloride-induced crevice corrosion, which will permit its successful application in most cooling water duties.

Wall temp	25°C	50°C	75°C	100°C
Cl — ppm (mg/l) #	400	180	120	50

For unaerated, approximately neutral pH (7), flowing aqueous solutions, these figures offer a guide to maximum content of chloride.

ALLOY 904L: This alloy is of particular value for duties involving sulphuric acid and especially phosphoric acid where there is contamination with chloride/fluoride ions. The high nickel content imparts a good degree of resistance to stress corrosion cracking,

a failure mode frequently associated with pitting/crevice attack in AISI 316 stainless steels. This is an alloy which offers an excellent compromise between price and corrosion resistance for a wide range of acidic and chloride containing environments.

Wall temp	25°C	50°C	75°C	100°C
Cl — ppm (mg/l) #	1,000	500	250	130

For unaerated, approximately neutral pH (7), flowing aqueous solutions, these figures offer a guide to maximum content of chloride.

AVESTA 254SM0: An improved grade of 316 resulting from the increased level of molybdenum which imparts better resistance to chloride-induced pitting/crevice corrosion. It is for use in brackish waters where the corrosion resistance of 316 is inadequate. The performance in mineral acids is significantly better than 316. In general terms, the corrosion resistance improves with increasing molybdenum content.

Wall temp	25°C	50°C	75°C	100°C
Cl — ppm (mg/l) #	5,000	1,800	750	400

For unaerated, approximately neutral pH (7), flowing aqueous solutions, these figures offer a guide to maximum content of chloride.

NICKEL 200: Use of this material tends to be confined to high strength (>50 %) caustic soda at high temperatures (up to boiling point), although it is susceptible to chloride-induced crevice corrosion by brackish cooling waters.

HASTELLOY B-2: An expensive alloy with a useful resistance to hydrochloric acid and many sulphuric acid concentrations. Its disadvantage lies in its inability to withstand oxidizing environments and contamination of acid streams with ions such as ferric and cupric. If the acid is aerated, it can have a catastrophic effect on corrosion resistance. It is particularly good for handling the products of Friedel Craft reactions.

HASTELLOY C-276: An expensive alloy that is almost immune to attack by chloride ions at low pH values. It has an extremely good resistance to a wide range of sulphuric acid concentrations and can be used with certain strengths of hydrochloric acid. It is particularly resistant to wet process phosphoric acid. One of the few alloys suitable for use in hot, concentrated sulphuric acid.

HASTELLOY G-30: Possessing a high nickel and molybdenum content, the alloy is suitable for dilute and middle ranges of sulphuric acid concentrations. It is good where contamination with chloride and fluoride ions occurs (i.e., gas scrubbing duties), and exhibits an improved resistance in such environments over Incoloy 825 but at an increased price.

INCONEL 625: Suitable for many environments including chloride ions and acids. It provides a compromise solution where conditions demand something better than Incoloy 825 but do not necessitate the extremely good corrosion resistance (and high price) of Hastelloy C-276.

INCOLOY 825: A reasonably priced alloy with a good resistance to high strength caustic soda and virtual immunity to stress corrosion cracking but is susceptible to chloride induced crevice corrosion.

MONEL 400: Particularly good for acid-chloride environments, and under certain circumstances, brackish and saline waters. It has excellent resistance to high strength caustic soda (30% to 50%) at high temperatures, but it is susceptible to attack by mercury that is occasionally present as a contaminant. In such cases, Incoloy 825 is preferred.

OTHER METALS

TITANIUM: The material par excellence for chloride solutions. It is virtually immune to attack by sea water up to 248°F (120°C) and to other chloride solutions such as calcium chloride. It is not immune to attack by some concentrated chloride solutions at temperatures over 212°F (100°C) .

GENERAL OR UNIFORM CORROSION

This type of corrosion commonly is encountered with carbon and low-alloy steels (rusting), and as the name suggests, is a form of attack which results in a more or less uniform deterioration and thinning of the metal. It is, however, rarely encountered with stainless steels and more highly alloyed metals. When it does occur, it is almost exclusively the result of attack by certain mineral acids or certain particularly corrosive organic acids (e.g. hot concentrated oxalic acid solutions). Insofar as general corrosion rates for most metals and alloys are well documented, this type of attack is relatively easy to predict and the selection of suitable materials, bearing in mind the criterion of tolerable metal loss previously mentioned, presents few problems. Because of its predictability, this could well account for the fact that it is so rarely encountered. Metal which has been the subject of general corrosion assumes an etched appearance and the original metal surface morphology usually is lost. Figure 40 shows a typical example of this form of corrosion.

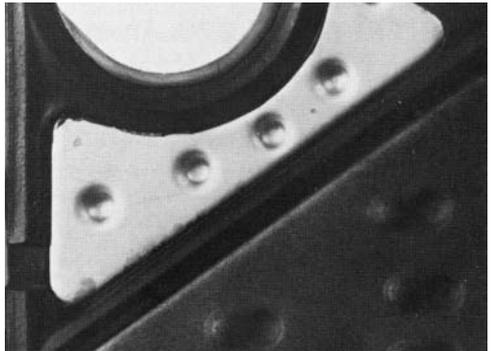


Figure 40. General corrosion of heat transfer surface. (Note pristine condition of area outside product contact).

General corrosion always is a temperature-related phenomenon. The higher the temperature, the greater is the corrosion. In any heat transfer situation, it is not sufficient to assume that the maximum bulk liquid temperature represents the most corrosive condition. The temperature of the heating/cooling medium always will dictate the actual skin metal temperature. In a cooling duty, the bulk liquid represents the most corrosive condition since metal around the inlet ports will be at this temperature even though the heat transfer surface may be cooler. On the other hand, the most corrosive conditions in a heating duty exist where the heating medium first contacts the product via the heat transfer surface. The actual maximum skin metal temperature, therefore, must be determined before selection of the correct material can be effected.

PITTING CORROSION

One common cause of equipment-corrosion failure is pitting corrosion. This attack characteristically is caused by chloride or chlorine bearing fluids. Consequently, a large number of environments exist within the process industries which potentially are corrosive and for which correct material selection is critical.

Pitting corrosion damage is highly localized and tends to be deep and undercut in nature. Figure 41 shows pitting upon a stainless steel pipe, and as can be seen, the surface damage is little more than a series of randomly sited small holes with the adjacent metal being entirely unaffected and retaining its original polished appearance. Viewed in cross-section, the insidious nature of pitting corrosion perhaps is easier to appreciate. Underneath the surface, the metal has been corroded away in an undercut fashion and complete perforation of the metal is imminent. This is due largely to the product from the corrosion reaction being more corrosive than the initiating process stream. Additionally, these products also tend to be more dense, and it is common to see pitting grow under the influence of gravity.

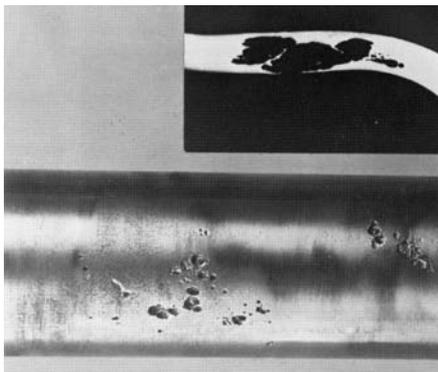


Figure 41. Pitting corrosion. Inset shows a micrograph of a section through a corrosion pit.

From all the foregoing, the actual loss of metal is negligible when compared with the bulk. This highlights the inadequacies of corrosion tables which quote weight loss or "mean" loss of metal thickness.

The austenitic stainless steels are susceptible to this type of attack and in very broad terms, their susceptibility is an inverse function of the molybdenum content of the alloy. For example, 304 stainless steel (no molybdenum) is particularly prone to pitting corrosion by neutral/acidic solutions containing as little as 50 ppm of chloride. This, therefore, limits the use of this grade of stainless steel to all but the purest of cooling waters and seriously restricts the chemical environments in which it can be successfully employed. On the other hand, 316 stainless steel (21/2% molybdenum) offers a reasonably good resistance to pitting attack and as a very rough guide, 150 ppm chloride concentration is taken as a safe maximum level. Thus, this grade of steel is suitable for use in a wide range of industrial waters, even cooling tower waters, assuming that the water quality circulating in such a system is maintained and the chemistry of it is controlled. It must be emphasized that all the foregoing is for typical waters. Other factors such as pH or scaling tendency can have a marked influence on the performance of the various materials. Specific recommendation, therefore, can be made only in the light of all the available information on the particular stream.

For more aggressive waters or process streams containing higher levels of chloride, alloys such as 904L (41/2% molybdenum) or Avesta 254SM0 (6% molybdenum) may be required. For sea water cooling applications (21,000 ppm chloride) the 6% molybdenum alloys can be used for lower temperatures, but at high temperatures alloys such as Inconel 625 may be needed, or even titanium, which is immune to this form of damage at temperatures up to 248°F (120°C).

Chlorination of water supplies is a common practice and although the 300 series stainless steels are suitable in potable waters containing up to 2 ppm residual chlorine, more highly alloyed materials may be necessary where there are high levels. As with chloride solutions, titanium offers immunity even to saturated chlorine solutions such as may be encountered in chlorine production units.

Brines based on sodium or calcium chloride commonly are used in coolant circuits. These, too, can initiate pitting corrosion of stainless steels. For their successful use in equipment constructed of 300 series stainless, strict pH control and deoxygenation is essential. During shutdowns, draining and rinsing of brine residuals is essential to avoid intense pitting attack. While the precautionary requirements for successful use of 316 can be enforced in food processing plants, it is uncommon for them to be fulfilled in an industrial chemical plant. Under these circumstances, the use of the 300 series steels is inadvisable. As a material able to withstand chemical "abuse," titanium ideally should be employed.

CREVICE CORROSION

Whenever pitting corrosion is encountered, it is common for crevice corrosion to be found as well. Crevices inherently occur within modern process plants at a number of places, i.e., pipeline joints, valves, and within plate heat exchangers. Figure 42 shows crevice corrosion on a heat exchanger plate at a plate-to-plate contact point. Crevice conditions also can be created on heat transfer surfaces by the presence of scale or product debris.

Crevice corrosion is, in fact, essentially similar to pitting corrosion except that by definition can occur only under crevice conditions where several added features can exist which are capable of adversely influencing the pitting corrosion process. It is caused by exactly the same range chloride and chlorine bearing environments. This similarity with pitting is clearly evident from the cross-sectional view.

There is a divergence of opinion as to the reaction mechanism, which

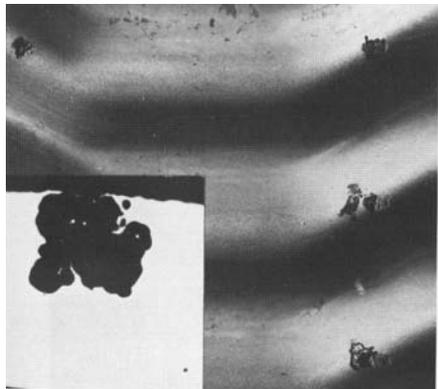


Figure 42. Crevice corrosion at the contact points of a heat exchanger plate. Inset shows a corrosion site.

it
of

will initiate crevice corrosion. It originally was considered that it resulted from an electrochemically induced breakdown of the protective oxide film on the metal surface caused by a difference in concentration of an ionic species or oxygen within the crevice and in the bulk liquid surrounding the crevice. Modern views tend to favor a mechanism, whereby due to the restricted irrigation of the liquid trapped in the crevice, restricted replacement of oxygen in the crevice fluid causes chloride or other corrosive ions to migrate into it and create very high concentrations. This, in turn, causes accelerated attack.

The preventive measures for crevice corrosion are, as one might expect, the same as those for pitting corrosion. While certain steps such as keeping plant surfaces clean will alleviate the risk, such measures rarely are practical on equipment located in a continuously operated industrial complex. For such equipment, correct material choice is the important factor. It is worth restating that 304 stainless steel is totally unsuited for coping with crevice corrosion. Although 316 offers some degree of resistance, for many situations the highly alloyed stainless steels or titanium may be necessary.

As with pitting corrosion, values quoted in corrosion tables for loss in weight and "mean" metal penetration rate are meaningless.

STRESS CORROSION CRACKING (S.C.C.)

Yet another form of corrosion damage which frequently is encountered and which can cause extremely rapid equipment failure is stress corrosion cracking. This form of damage occurs when a metal under the influence of stress comes into contact with certain highly specific environmental conditions. While these vary depending upon the metal, in the case of the chromium/nickel alloys it is most commonly chloride ions. It also is a feature of stainless steels that S.C.C. only occurs in environmental conditions when the temperature is in excess of 140°F (60°C).

Figure 43 shows the appearance of a typical stress corrosion failure, the cracking being a network of highly branched cracks. It will be noted that the cracking is transgranular, i.e., the cracks pass through the metal grains rather than following the grain boundaries. From a point of view of diagnosing plant failures, this transgranular route of the crack path is of vital significance in helping to classify the corrosive agency as chloride ions.

The stress component of the failure mechanism may be present either as an inbuilt stress resulting from welding and fabrication processes, or alternatively, may be due to process pressure conditions. In either case, it is clear that in a modern process plant the presence of stressed metal is unavoidable. It also

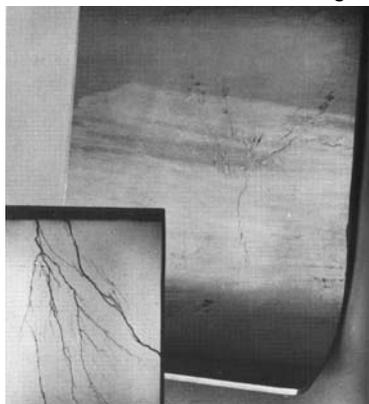


Figure 43. Typical stress corrosion cracking of a 304 stainless steel storage tank. Inset shows a section through a crack.

must be pointed out that other failure processes can play a role in stress corrosion by acting as stress raisers, complementing the existing stress levels. For example, it is common to find stress corrosion cracks initiating from within pits formed by crevice or pitting corrosion (Figure 44).

The susceptibility of an austenitic stainless steel to S.C.C. is a function of the nickel content of the alloy. It is an unfortunate fact that the greatest susceptibility is exhibited in stainless steels containing ca. 10% nickel, which is the alloying level in both 304 and 316. Where an environmental situation exists when there is a serious risk of S.C.C., the remedial action necessary is to select an alloy with a higher or lower nickel content. The so-called duplex structure stainless steels with a nominal 5% nickel and the ferritic stainless steels with zero nickel exhibit extremely good resistance to S.C.C. Unfortunately, their mechanical properties and ductility are such that it is not possible to form anything other than the (geometrically) simplest heat exchange plate form, although for the fabrication of other items of process equipment, they offer many advantages over the 300 series stainless steels. Increasing the nickel content to 25% (904L) imparts a very high degree of resistance to chloride induced S.C.C. With an alloy containing 40% nickel (Incoloy 825), it is claimed that the material exhibits immunity.

Before closing this section on S.C.C., a brief mention must be made of a second, specific, environmental condition which can cause this mode of failure – namely, caustic alkalis. However, the conditions required for such attack are quite severe. At 230°F (110°C), for example, caustic concentrations of some 30% to 60% are necessary. The unique feature of this type of S.C.C., which differentiates it from chloride induced S.C.C., is that the cracking is intergranular (following the grain boundaries) as shown in Figure 45.

As previously mentioned, the remedial action is to employ a material of higher nickel content. Although Monel 400 can be used for high temperature duties involving caustic soda in the concentration range of 30% to 50%, for higher strengths and temperatures nickel is the ultimate choice. However, both these materials have disadvantages, which have been covered in a previous section.



Figure 44. Stress corrosion cracking associated with crevice and pitting corrosion.



Figure 45. Photomicrograph of intergranular stress corrosion cracking.

FATIGUE

Fatigue is theoretically an in-vacuo phenomenon. More correctly, examples of fatigue in plate heat exchangers, the incidence of which is very low, should be defined as corrosion fatigue.

Fatigue is caused by the metal being allowed to flex under the influence of cyclic or pulsating load conditions. The end result is a straight, unbranched crack as shown in Figure 46.

It may be argued that insofar as fatigue is the result of a purely mechanical action, it does not come into the field of corrosion. In many if not most instances, however, there is a small promotional corrosion influence which has an adverse influence on the "time to failure." In some cases, the participation of the corrosion element in the cracking process is evident. For example, stress corrosion cracks propagating from the main fatigue crack (Figure 47) sometimes is observed. Although there is a corrosive element invariably associated with fatigue, it generally is considered that in the absence of a fatiguing condition the corrosive element is insufficient to initiate failure from this source alone.

Numerous factors, both mechanical and environmental, can influence the fatigue hazard. In terms of avoiding failures in plate heat exchangers, there are only two satisfactory preventive measures and these stem from the original definition of a fatigue failure – the metal being flexed under the influence of a pulsating load. The most common source of pulsating load originates from positive displacement pumps, and thus wherever possible, the use of centrifugal pumps is to be advised since these produce a smooth pressure output. In correctly assembled and maintained process plants, however, the equipment is less able to flex under any prevailing load. Thus, more severe conditions can be withstood before failure starts to occur. This highlights the need to correctly tighten plate packs within a plate heat exchanger frame.



Figure 46. Fatigue crack.



Figure 47. Fatigue crack with stress corrosion cracking emanating from the fracture face.

For a plate heat exchanger, the necessary plate support comes from the plate-to-plate contact of support points which are pressed into the plates at calculated sites. If these contact points become distorted or flattened either by adverse pressure conditions existing within the plant or by operators overtightening the plate pack during assembly, this good support will be lost.

Figure 48 shows fatigue cracking on a heat exchanger plate near support pipes that have been flattened by overtightening the machine. This adverse situation occurred in an effort to effect a seal with aged and compressed gaskets that should have been replaced. Although plates eventually were regasketed and the plate pack easily compressed to the correct dimensions, fatigue cracking still resulted because of the flattened contact points and loss of plate support.

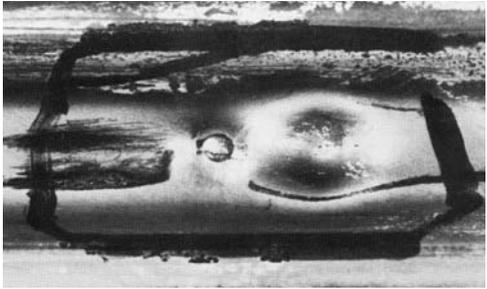


Figure 48. Fatigue cracking adjacent to a mechanically damaged support pipe.

EROSION CORROSION

Impingement attack or erosion corrosion is the increase in the rate of deterioration on a metal attributable to the movement of a liquid over the surface. It usually is characterized by the appearance of troughs or gullies on the surface. This type of damage generally does not prove to be a problem with stainless steels but more usually is associated with the softer copper based alloys. Figure 49, however, shows erosion damage to stainless steel at an inlet orifice where the velocity of a mildly corrosive process stream was extremely high. In other areas where the velocity was greatly reduced, no attack was noted.

For a given plant material of construction-product combination, flow velocity and turbulence are critical factors in determining the erosion risk. For this reason, a flow velocity maxima often is imposed where alloys sensitive to erosive conditions are encountered.

When solutions contain abrasive particulate solids, even the stainless steels may be subject to erosion – although, because of their excellent resistance to this form of deterioration, limiting the velocity usually is an adequate precaution.

CONCLUSION

It has been possible in this paper to cover the performance of materials of construction for plate heat exchangers only in the broadest terms. There are many forms of corrosion of which no mention has been made (hydrogen embrittlement, caustic embrittlement, intergranular attack, galvanic corrosion) since these forms of damage either are peculiar to materials not used in plate heat exchangers or are the result of bad manufacturing techniques which also are not used in making plate type units.

Corrosion engineering is a complex subject and a blind interpretation of corrosion data presented in many reference works can lead to problems. Only by taking all the factors into account at the specification stage can materials offering optimum performance be selected. For such a choice to be made, it is imperative that all the facts are stated and the full and exact composition and maximum temperature of the process streams given. For example, as little as 50 ppm of fluoride ions as an impurity in an acidic process stream can cause the catastrophic failure of titanium. On the other hand, for a duty involving sulphuric acid (which normally would demand an expensive plate material), the presence of oxidizing species such as ferric or cupric ions will permit the use of a much less expensive alloy.

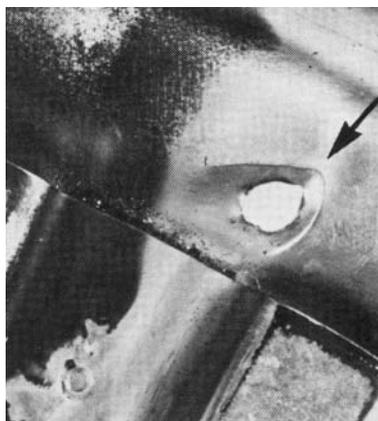


Figure 49. An example of erosion corrosion.

THE DESIGN OF PLATE HEAT EXCHANGERS FOR REFRIGERANTS

CONDENSATION

Refrigerant gas from the compressor delivery enters a PHE superheated. The sensible or desuperheating duty is usually small and the cooling medium temperature low enough to initiate and maintain wet wall desuperheating, which is completed in a small fraction of the plate length. In rare cases where wet wall conditions cannot be assumed a separate desuperheating section a unit may be employed. Subcooling of the condensate is best achieved using a secondary, small PHE with dedicated control on the cooling media flow rate.

EVAPORATION

In dry expansion systems, the refrigerant normally flows counter currently to the product being cooled to enable a control gas superheat to be most easily generated at the PHE outlet. In flooded systems both fluids are arranged in concurrent upward rising channels to provide the maximum evaporating side wall temperature. Hence, vapor generation occurs as early in the boiling process as possible. Concurrent flow also ensures that the refrigerant temperature coincident with the product outlet is at a minimum.

In most practical applications high, single phase, heat transfer combined with micro-convective boiling ensures that the sub-cooled zone occupies an insignificant part of the total heat transfer area.

Although a vast number of papers have been published on boiling heat transfer, Chen's correlations⁶ have been one of the most universally successful – in form – used to model two-phase boiling. In summary,

$$h_{tp} = h_c + h_m$$

The two-phase film heat transfer coefficient is taken as the sum of the macro-convection (h_c) and micro-convection (h_m) forms of convection.

Here, h_m is the product of the nucleate boiling coefficient h_{nb} and a suppression factor S arising from departure from pool boiling conditions. The suppression factor S is a function of the overall heat transfer coefficient, wall temperature profile, surface properties and reduces rapidly with increasing turbulence. The term h_c is the enhanced forced convection film coefficient, the product of a single-phase film coefficient h_f and an enhancement factor F that is always greater than unity. The enhancement factor accounts for added turbulence and convection resulting from gas-liquid interaction in addition to the artificially increased turbulence that results from the corrugated plate form. Single-phase film HTC's were first presented for PHE by Bond⁷ of APV.

Experimental work by Shaw⁸ of APV has resulted in local correlations for S and F , enabling evaporative heat transfer coefficients for the primary refrigerants to be calculated. These indicate that the average enhancement is between 3.4 and 6.3.

Parallel work by Bojesen⁴ of APV on freon 22 and ammonia concluded similar values for enhancement F. Both APV workers confirmed the following:

- a) Enhancement factor F is a strong function of corrugation angle to the flow direction.
- b) Enhancement factor F can be correlated as a function of the Lockhart-Martinelli factor X_{tt} .

Passage average Refrigerant film coefficients in both studies typically ranged from:

R12 and R22	1.3 to 4.7
R717 ammonia	3.4 to 12.9 kW/m ² /K

These values are averaged values for an overall passage and large variations apply between inlet and outlet conditions.

For most channel configurations fully wetted conditions apply at least up to 70% exit gas quality and above due to the tortuous flow path in most PHEs which acts as an effective droplet separator keeping the surface wetted.

In summary, the boiling coefficients for upward flow of refrigerants in plate channels are high and these are combined with high single-phase coefficients for the process side to allow for rapid heat transfer. A corollary of high HTC's plus near ideal parallel temperature profiles is that small temperature approaches can be obtained between refrigerant and the process outlet temperatures usually in a single pass.

PRESSURE DROPS

Pressure losses for evaporation are much larger than those in condensation due to large momentum increases caused by vapor generation. Care must be taken to ensure that losses are not excessive since temperature differences can be adversely affected. For most routine applications losses are such that a natural circulation or thermosiphon system can be designed. Shaw³ reports values of 0.02 to 0.44 BAR depending on geometry, refrigerant and operating conditions. Bojesen⁴ reports 0.011 to 0.25 BAR for various conditions.

FOULING

A fouling resistance can exist on both sides of a PHE. On the refrigerant side, a small amount of compressor oil is always present. Depending on oil/refrigerant type and operating conditions, this oil layer will offer some thermal resistance. Typically, oil resistance ranges between 0.05 and 0.15 m²/K/kW. In the event that oil accumulates, a fall off in heat transfer will occur, as is the case for any form of tubular unit.

In general, the designer must account for the liquid process side fouling which may be more significant. In the case of condensers – high, superheat, gas temperatures above 70°C can result in some water scaling and high local plate fouling. On average, water fouling is not usually a serious consideration.

COMPARISONS WITH SHELL AND TUBE

1. Shell and tube units can be designed for very high pressures whereas PHEs are limited to approximately 30 BAR Design pressures exceeding 20 BAR are rarely called for in refrigerant applications.
2. Shell and tube units can be designed for lower temperatures. PHEs contain elastomeric polymers limited to approximately -45°C . Small, fully brazed PHE units can operate to very low temperatures.
3. Heat transfer coefficients in PHEs are much higher on the liquid side. In most cases, they are higher for refrigerant evaporating and condensing sides, too.
4. The artificially induced turbulence in PHE channels is responsible for much lower fouling rates.
5. PHEs have modular construction, so they can be easily reconfigured to accommodate different duties.
6. Gaskets and welded-pair PHEs are resistant to failure induced by product freezing and flow-induced vibration.
7. Because of narrow flow channels and high HTCs, PHEs are compact – typically requiring a quarter to half the foot print of horizontally installed shell and tube units. Refrigerant charge is similarly reduced.
8. Plate heat exchangers are made from stainless steel, titanium and other corrosion resistant alloys in all product wetted components. Capital costs are much lower if the comparable shell and tube unit is compelled to employ similar materials. If a mild steel shell and tube unit is acceptable, then a PHE may be more expensive.

CONCLUSIONS

Plate heat exchangers are well suited to evaporation and condensation of primary refrigerants, and within limits of pressure and temperature, offer several important advantages over traditional alternatives.

NOMENCLATURE

(Physical property group defined in equation (I)
g	Gravitational acceleration (m/s ²)
h	Film heat transfer coefficient (W/sq m K)
kL	Liquid thermal conductivity (W/m K)
m	Condensation rate per passage (kg/s)
PrL	Liquid Prandtl number
ReLT	Condensation Reynolds number
w	Channel width, m
mL	Liquid Viscosity (Pa's)
Pr	Liquid density (kg/m ³)
PrL	Vapor density (kg/m ³)
GL	Condensation rate/unit wetted perimeter (kg/m)
htp	Two phase film heat transfer coefficient (HTC) (kW/m ² /K)
hc	Macro convection film HTC component (kW/m ² /K)
hm	Micro convection film HTC component (kW/m ² /K)
hnb	Pool boiling film HTC (kW/m ² /K)
S	Suppression factor where S ²⁰ (-)
hf	Single phase film heat transfer coefficient (kW/m ² /K)
F	Two phase enhancement factor where F ³¹ (-)
Xtt	Lockhart-Martinelli factor for turbulent flows (-)

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