Advantages of Brazed Heat Exchangers in the Gas Processing Industry

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ABSTRACT

Brazed aluminum heat exchangers have superior heat transfer capabilities and can be cost effective for non-corrosive gases and liquids as compared with traditional shell-and-tube exchangers. Even so, brazed aluminum exchangers are often not considered because of complicated design equations and complex stacking arrangements. The simpler yet less efficient shell-and-tube exchangers or networks of shell-and-tubes are employed instead. Recently, the design equations for multistream brazed aluminum heat exchangers for both single and multiphase flow have been added to the Heat Exchanger Rating package of the process simulator PROSIM® . This paper presents guidelines for designing a brazed exchanger, and the brazed exchanger is compared with traditional shell-and-tube exchangers and networks of exchangers in several examples.

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INTRODUCTION

The performance and profitability of gas process operations depends on efficient and economical heat transfer equipment. Several recent papers assist the engineer in selecting the proper heat exchanger type[1,2,3]. These papers compare applications using the common shell-and-tube exchangers with more specialized plate-frame and spiral exchangers. None of these recent articles, however, mention the brazed exchanger as a viable alternative. Several of these publications use the term "compact exchangers" in referring to many different types of exchangers including plate-frame and spiral. In this paper "compact exchangers" refers exclusively to plate-fin exchangers primarily constructed from aluminum using a brazing process.

There are two main reasons for the lack of exposure for brazed exchangers in the trade magazines.

1. Design equations for compact exchangers are not readily available in the literature, and
2. The design equations for compact exchangers tend to be complex and not suitable for hand calculation.

As a result, engineers are handicapped in the area of heat exchanger selection. In fact, C.R. Giovanni[4] candidly states that industry all too often is reluctant to implement new technology and this may influence biased decisions concerning exchanger selection.

Historically, shell-and-tube exchangers have dominated the market. Since design equations for shell-and-tube exchangers are widely available in the literature[5,6], a large number of commercial software analyze these exchangers with good accuracy. In addition, numerous college level textbooks and handbooks describe how to design shell-and-tube exchangers from simple hand calculations.

By comparison, design data and handbooks for compact exchangers are limited. One reason for this is manufacturer proprietary development. However, sources in the open literature have recently become available. For example, Kays and London[7] publish an extensive set of data containing heat transfer film coefficients and friction factors for compact exchangers. Mike Taylor[8] reports design equations and methodologies for simple compact exchangers. Recently, the brazed exchanger manufacturers produced a set of guidelines for exchanger design[9] similar to that of the Tubular Exchanger Manufacturers Association.

The complexity of compact exchanger design equations results from the exchangers unique ability to transfer heat between multiple process streams and the wide array of possible flow configurations. These complexities make hand calculations tedious and simple correlations inapplicable. However, computer programs and process simulators allow engineers to more easily rate complex brazed aluminum exchangers. The technical development staff at Bryan Research and Engineering has recently incorporated the brazed exchanger design equations into the Heat Exchanger Rating package of its process simulator PROSIM.

Obviously, compact exchangers are not suitable for all applications. Many applications should not be considered simply because the process streams are unclean, or corrosive, or operate at greater than 400°F. Even with these restrictions, there are many opportunities in the gas processing industry to exploit compact exchangers.

MECHANICAL CONSTRUCTION

Compact exchangers were initially developed for the aerospace industry during the 1940s. Typical applications required exchangers which provided a large amount of surface area for heat transfer, were lightweight, and occupied a relatively small volume. Aluminum was the material of choice since it is easily machined, relatively lightweight, and has a high thermal conductivity.

For the process industry, the compact exchanger's large heat transfer surface area to weight and volume ratio was not as important as some other features. Aluminum has superior mechanical properties at cryogenic temperatures. (Specialized materials of construction superior to aluminum exist but at tremendous costs.) Furthermore, brazed exchanger construction produces nearly ideal countercurrent flow among the process streams for optimum heat transfer. With increased interest from the process industry, manufacturers have improved brazing technology enabling them to build larger, more complex exchangers.

Even with relatively clean process fluids, brazed exchangers are constrained by operating pressure and temperature limitations. Above ambient temperatures, aluminum rapidly loses its mechanical strength although operating temperatures to 400°F are possible. Stainless steels are usually used in process applications to 1000°F. Operating pressure affects exchanger volume and cost. Exchangers with operation pressures that exceed 1440psi are uncommon. Exchanger size or volume is limited exclusively by the vendor's brazing furnace; however, manufacturers can increase the exchanger size by welding cores together.

Vendors construct brazed exchangers from alternating layers of corrugated sheets and flat parting sheets. Heat is exchanged between fluids through both sheets. The stacked arrangement is then brazed, yielding the exchanger core as shown in Figure 1. Headers and nozzles are attached to route the fluid in and out of the core.

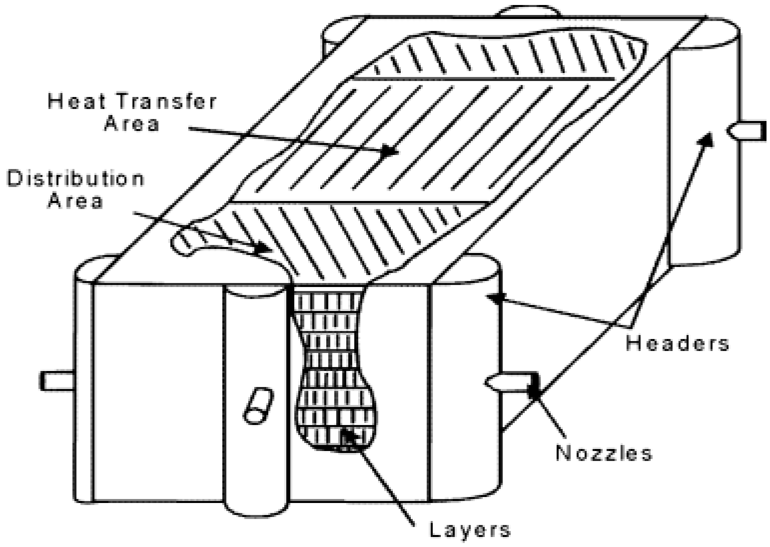


Figure 1. Brazed Exchanger Core.

The corrugations, or fins, not only serve as additional area for heat transfer but also provide the mechanical support for the core. Operating pressures and pressure differentials establish fin and parting sheet thickness. Depending on the service, fins may either be left unaltered (plain) or enhanced as shown in Figure 2. Manufacturers can modify the corrugations in a variety of ways. The most common modification is serrating or lancing, which produces offset fins to promote turbulence. Perforated fins are used to assist in laminar boundary layer break up of a stream. Fin height and density (fins per inch) are a function of both the process fluid characteristics and the operating pressure.

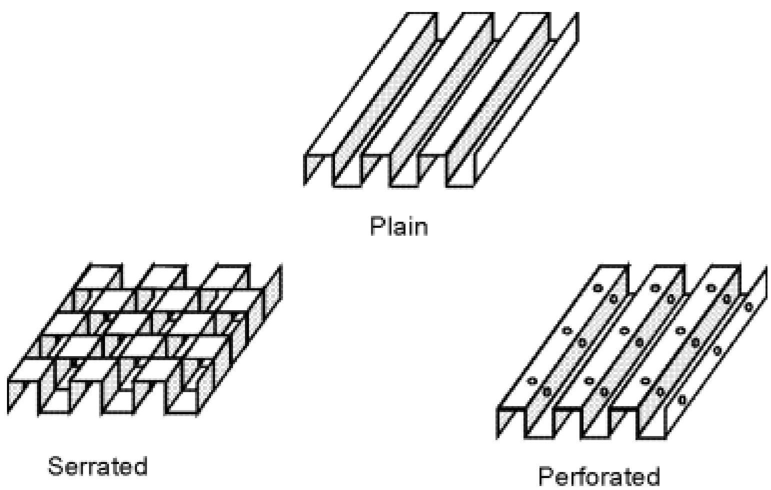


Figure 2. Types of Fins for Brazed Exchangers.

The brazed exchanger layer is divided between distribution and heat transfer areas as shown in Figure 3. Distribution areas are constructed of plain fins which direct the fluid from the nozzles to the heat transfer area. Distribution areas are usually designed to account for less than 25 % of the pressure drop for a stream through the core. Larger pressure drops in the distribution area tend to cause maldistribution and adversely affect exchanger performance. Because of pressure drop and maldistribution problems, nozzle and header sizes are critical. Unfortunately, securing large nozzle and headers to the core increases the exchanger cost disproportionately. Since quantifying and modeling heat transfer in this area is extremely difficult and since the distribution fins are typically much less efficient at heat transfer, distribution area is not included in the area available for heat transfer. This additional area does serve to ensure a conservative design.

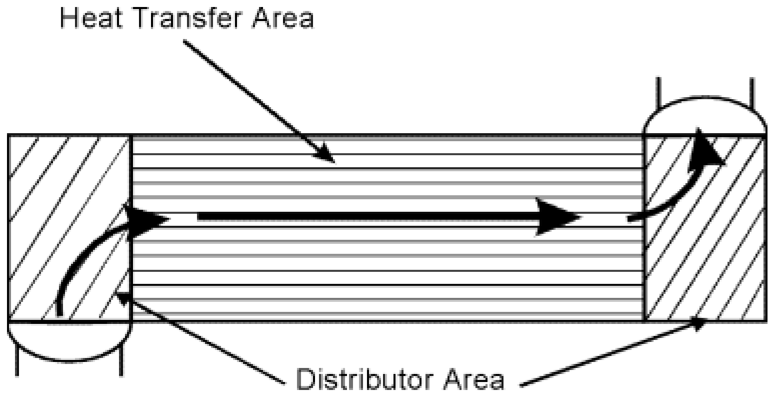


Figure 3. Heat Transfer and Distribution Areas in Brazed Exchangers.

The process fluid flow for the heating and cooling fluids is usually countercurrent in the heat transfer area. Compact exchangers achieve closer temperature approaches than shell-and-tube exchangers with baffles since baffled exchangers always have some degree of cross flow. The fins in the heat transfer area are usually serrated, perforated, or some combination depending on the fluid conditions.

Designers can enhance or diminish heat transfer with a stacking arrangement. For optimum heat transfer, heating and cooling streams should be placed in adjacent layers. Sometimes, two heating or cooling streams must be adjacent to alleviate excessive pressure drop.

Stacking arrangements can be even more sophisticated if multiple fluids are routed on the same layers. This arrangement is also referred to as "multiple zones". Figure 4 compares a layer with multiple zones (streams) with a layer containing a single fluid. Even with multiple fluids on a layer, the layer is still divided between distribution and heat transfer areas. Manufacturers separate and route the fluid with a series of transfer bars and distribution areas. Routing multiple streams onto a common layer in separate zones is normally done if the proper temperature profile is possible. The exiting temperature of one stream should be close to the entering temperature of the adjoining stream.

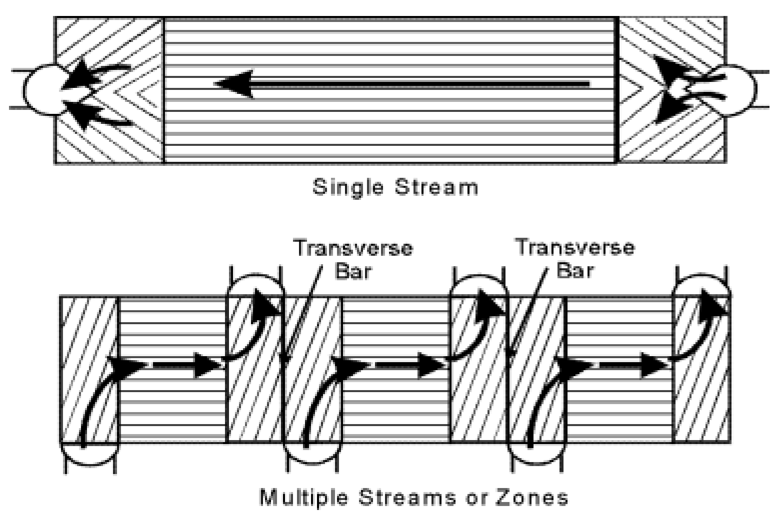


Figure 4. Comparison between Single Streams and Multiple Streams on One Layer.

The stacking arrangement is usually expressed as a sequence of repeating patterns. For example, consider a two-stream exchanger. The first stream is being cooled and has approximately twice the volumetric flow rate of the second stream. The first and second streams are denoted "A" and "B", respectively. To account for the differences in the flow rates a stacking arrangement such as:

10[ABA] or ABAABAABAABAABAABA...

may be specified. This is not necessarily the optimum stacking arrangement for heat transfer, and other factors may significantly affect the exchanger design.

EXCHANGER COSTS

Exchanger costs are usually quoted proportional to the exchanger heat transfer area. A direct comparison between shell-and-tube and brazed exchangers is difficult because they have different definitions for heat transfer area. Shell-and-tube exchangers (two fluids only) report a single area which is usually the outside surface area of the tubes. Compact exchanger design correlations report surface area for each process stream. The number of layers and fin characteristics required for different fluids may result in dramatically different heat transfer areas. The total heat transfer surface area for compact exchangers is the sum of the areas for all of the process streams.

Using these definitions, Purohit[10,11] provides costs estimations for shell-and-tube exchangers made from carbon steel at approximately $20/$ft^2$. Exchangers made of stainless steel can be as high as $100/$ft^2$. For small cores (<10,000 ft^2), brazed exchangers made from aluminum cost between $6-15/$ft^2$. Larger cores (>10,000 ft^2) cost between $3-8/$ft^2$. Obviously, this is an overly simplified cost comparison. Brazed exchanger costs vary depending on the number of streams, design pressures, types of connections and special features and testing.

Manufacturers of brazed exchangers sometimes assess a brazing furnace charge, depending on the brazing furnace demand. Unfortunately, the demand on the brazing furnace depends on market conditions and can be somewhat unpredictable. For the following comparisons, we assume a $20,000 brazing furnace charge.

EXAMPLES

Four examples illustrate the differences in design and cost between compact and shell-and-tube exchangers. The first two examples each contain two process streams; therefore, the comparison between a single compact exchanger and a shell-and-tube is fairly straightforward. The third example contains three process streams and compares one compact with two shell-and-tube exchangers. The fourth example has four process streams comparing one compact versus three shell-and-tube exchangers. In all of these examples, the pressure drop for each stream was different. However, both the shell-and-tube and brazed exchangers yielded pressure drops within the allowable limit for each stream.

Case 1: Vapor Ethane/Liquid Ethane Exchanger

Table I.

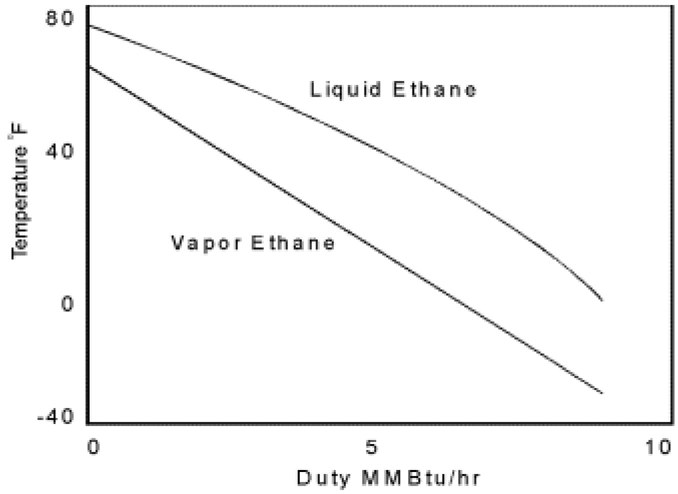
Process Information for Case 1

	Side A Ethane Vapor	Side B Ethane Liquid
Flow rate (lb_m/hr)	140000	231000
Pressure (psia)	805	132
T_{in}/T_{out} (^oF)	75/-1	-29/65

Figure 5.

Duty versus Temperature for Case 1.

Duty (MMBtu/hr)	-9.5	9.5



Pure ethane vapor is available to cool a pure liquid ethane stream from 75 to -1°F. The conditions of the process streams are provided in Table I. After specifying the stream flow rates and temperatures, PROSIM calculated the duty and the temperature of the ethane vapor. A plot of duty versus temperature shown in Figure 5 indicates that the exchanger has no internal pinch points or crosses. Both process streams are transferring heat by sensible heating/cooling so the lines are fairly straight. The curvature is due solely to the small pressure drop through the exchanger. This exchanger has a fairly large mean temperature difference of 20.3°F.

After completing the process simulation and confirming that the two process streams can accomplish the heat transfer from a thermodynamic standpoint, our next step is to design a configuration to achieve this transfer. Table II lists the parameters for both a compact and a shell-and-tube exchanger, each providing sufficient area for heat transfer. The table also reports exchanger area, volume, and cost for both types. The area required to transfer the heat in the compact exchanger is greater than the shell-and-tube exchanger because heat transfer area definitions are different for the two exchanger types. The compact exchanger occupies 1/5 the volume and costs about 1/2 as much as the three shell-in-tube exchangers in series.

Table II. Exchanger Information for Case 1.

		Side A Ethane Liquid	Side B Ethane Vapor	Area ft^2	Volume ft^3	Cost US$
Compact	Fins:					
	Type	Serrated	Serrated			
	Height (in.)	0.28	0.38			
	Thickness (in.)	0.016	0.010			
	Density (1/in.)	17	14.5			
	96in. long, 35in. wide, 32in. tall					
	Stacking: 24[BAB]			15200	60.6	$80,000
Shell-and-Tube		Shell	Tubes			
		40in. ID	0.75in. OD			
		20% Baffle Cut	12ft. long			
		13 Crosspasses	1.25 Pitch ratio			
		3 Shells in Series	Triangular			
			Carbon Steel	10250	314	$200,000

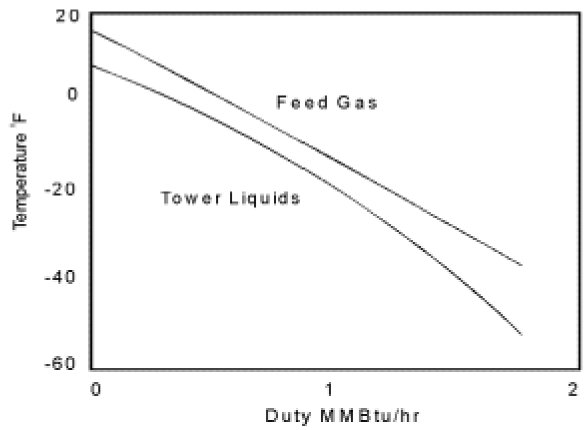
The dramatic difference in exchanger volume may be critical for existing processes that might need to be upgraded to increase capacity. Space might not be available for more shell-and-tube exchangers and interconnecting insulated piping in the existing facility.

Case 2: Inlet Feed/Side Reboiler

Table III.
Process Information for Case 2

	Side A Feed Gas Vapor	Side B Tower Liquids
Flow rate (lb_m/hr)	34000	33000
Pressure (psia)	975	298
T_{in}/T_{out} (^oF)	17/-33	-42/11
Duty (MMBtu/hr)	-1.9	1.9

Figure 6.
Duty versus Temperature for Case 2.



This example uses a portion of the refrigerated inlet feed gas to drive a side reboiler on a demethanizer. The feed is a mixture of light hydrocarbons with some impurities, and the tower liquids is predominantly light hydrocarbons. Table III presents the process conditions. With the outlet temperature of the side reboiler specified, PROSIM calculated the duty and outlet temperature of the feed gas. Figure 6 shows the duty versus temperature curve for this exchanger. Even though the reboiler liquid is being vaporized, the mixture has such a wide boiling range that the duty versus temperature curve does not display the plateau associated with phase changes. The mean temperature difference for this exchanger is about 4°F. Figure 6 also shows that the temperature differences in the middle of the exchanger are less than at either end. The endpoint log mean temperature difference yields a driving force for heat transfer that is greater than the actual driving force. Using the endpoint log mean temperature difference for design would yield an exchanger with insufficient heat transfer area. Although the temperature differences between the two streams decrease in the exchanger, the demand duty never exceeds supply duty. This suggests that the two streams can transfer the heat.

For the exchanger design, Table IV reports the exchanger parameters, area, volume and cost for both compact and shell-and-tube exchangers. The operating temperature of the exchanger is below the design temperature limit for carbon steel so stainless tubes ($30/$ft^2$) were specified. As a result, the cost for the shell-and-tube exchanger in this case is dramatically higher. The compact exchanger occupies half the volume of the shell-and-tube exchanger and costs about 1/3 as much.

Table IV. Exchanger Information for Case 2.

		Side A Feed Gas	Side B Tower Liquid	Area ft^2	Volume ft^3	Cost US$
Compact	Fins:					
	Type	Perforated	Serrated			
	Height (in.)	0.15	0.25			
	Thickness (in.)	0.024	0.012			
	Density (1/in.)	18	17			
	175in. long, 35in. wide, 32in. tall					
	Stacking: 41[AB]A			12000	41.7	$60,000

Shell-and-Tube	Shell 40in. ID 20% Baffle Cut 15 Crosspasses 2 Shells in Parallel	Tubes 0.75in. OD 12ft. long 1.25 Pitch ratio Triangular Stainless	6900	314	$200,000

Case 3: Gas/Gas/Gas Exchanger

Table V.
Process Information for Case 3

	Side A Feed Gas	Side B Residue Gas	Side C Recycle Gas
Flow rate (lb_m/hr)	40650	31320	14780
Pressure (psia)	810	205	285
T_{in}/T_{out} (^oF)	120/-54	-106/113	-106/113
Duty (MMBtu/hr)	-5.5	3.7	1.8

Figure 7. Duty versus Temperature for Case 3 with Brazed Exchanger.

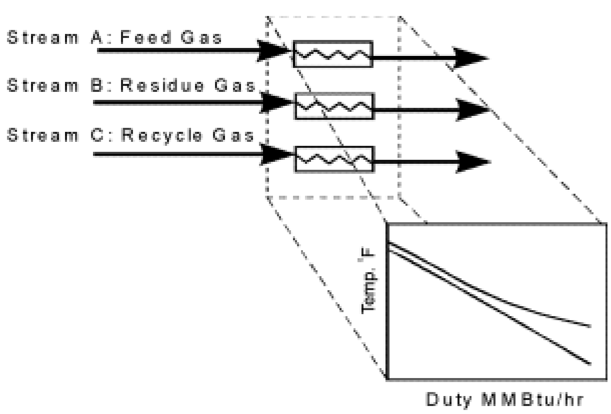
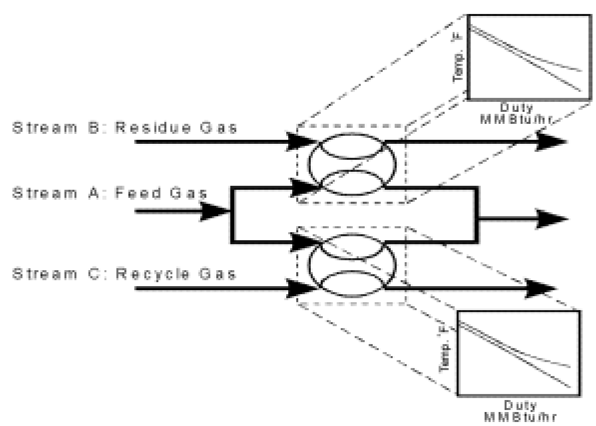


Figure 8. Duty versus Temperature for Case 3 with Shell-and-tube Network.



This example uses residue and recycle gas streams to cool a feed stream prior to processing. Unlike the previous two examples, this example contains three process streams. Table V lists the process conditions for the streams. The composition of all streams is light hydrocarbons with some nitrogen and carbon dioxide. Residue and recycle gas flow rates and inlet and outlet temperatures were specified along with the feed gas inlet and outlet temperatures. PROSIM calculated the amount of feed gas that could be processed and the required duty. Figure 7 shows the flowsheet and the duty versus temperature curve for the three process streams. Polasek *et al*.[12] describe in detail the generation of duty versus temperature curves for multisided exchangers. The effective mean temperature difference is 10°F. Since the curves do not intersect, at least thermodynamically, the heat can be transferred. The parameters for the compact exchanger are given in Table VI. Notice that this exchanger has a fairly sophisticated stacking arrangement to accommodate the three process streams.

Table VI. Exchanger Information for Case 3.

		Side A Feed Gas	Side B Tower Liquid	Side C Recycle	Area ft^2	Volume ft^3	Cost US$
Compact	Fins:						
	Type	Serrated	Serrated	Serrated			
	Height (in.)	0.28	0.28	0.28			
	Thickness (in.)	0.016	0.016	0.016			
	Density (1/in.)	17	17	17			
	180in. long, 25in. wide, 24in. tall						
	Stacking: 3[BABCABBACBABCABBACBAB]				17100	60	$80,000
Shell-and-Tube		Shell 18in. ID 15% Baffle Cut 8in. Baffle Sp 4 Shells in Series	Tubes 0.5in. OD 24ft. long 1.25 Pitch ratio Triangular Stainless				
		Shell 18in. ID 15% Baffle Cut 8in. Baffle Sp 3 Shells in Series		Tubes 0.75in. OD 24ft. long 1.25 Pitch ratio Triangular Stainless	13610	297	$400,000

We cannot use the same flowsheet for a comparison with the shell-and-tube exchangers as in the two previous examples. We must instead devise a process that accomplishes the same objectives as the previous figure using only two-sided exchangers. The shell-and-tube network that we selected is described below. This is not the only configuration that could be selected.

This shell-and-tube network was set up so the feed gas was split proportionally by the duty available in the residue and recycle streams. This split yields similar duty versus temperature curves for both exchangers without impossible temperature crosses as shown in Figure 8. This is because both the residue and recycle streams are exchanging heat by sensible heating. The inlet gas is condensing some of the components which accounts for the curvature. (Dividing the stream proportionally between the duty sometimes causes impossible temperature crosses depending on the shape of the duty versus temperature curve.)

Table VI lists the parameters for the feed gas/residue exchanger and feed gas/recycle exchanger. For simplicity, we elected to use the same basic shell-and-tube exchanger and simply use multiple shells in series to achieve the required heat transfer area. However, for the shell-and-tube exchangers, stainless tubes were used due to cryogenic temperatures. Comparing the cost and volume of the compact and shell-and-tube exchangers, the compact occupies 1/5 the volume at roughly 1/4 the cost.

Case 4: LPG Recovery with Propane Refrigeration

Table VII.
Process Information for Case 4

	Side A Ethane Vapor	Side B Ethane Liquid	Side C Liquid	Side D Propane
Flow rate (lb_m/hr)	25450	18200	7250	6140
Pressure (psia)	833	833	833	32
	113/-5	-5/105	-5/40	-12/-9

T_{in}/T_{out} (^oF)				
Duty (MMBtu/hr)	-2.5	1.2	0.2	1.1

Figure 9. Duty versus Temperature for Case 4 with Brazed Exchanger.

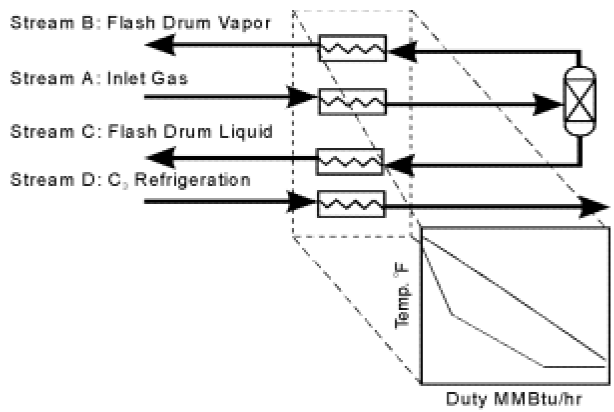
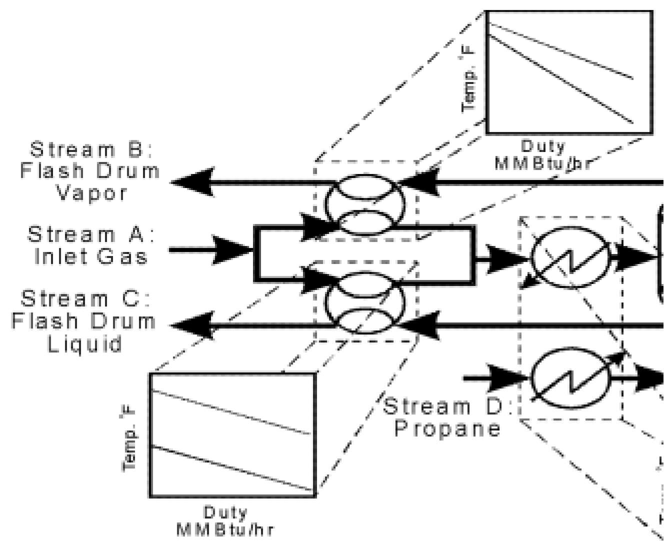


Figure 11. Duty versus Temperature for Case 4 with Shell-and-t



This example is a case study described by Polasek *et al*.[12] Inlet gas is cooled and condensed by the vapor and liquid portions leaving a flash tank. Additional cooling is provided by propane refrigeration. The original case was modified to better represent an actual design. The inlet gas flow rate is increased by an order of magnitude and the propane refrigeration is heated to 1 degree of superheat rather than to 110°F.

Table VII gives the process stream data. We specified the inlet gas mass flow rate, the outlet temperatures for the inlet gas, vapor from the flash drum, and liquid from the flash drum. With the inlet temperature and pressure of the propane specified, PROSIM calculated the propane flow rate and the duty. Figure 9 shows the flowsheet and the complex duty versus temperature curve.

The complex group as shown in Polasek *et al*.[12] was modified as follows. The original configuration assumes that the vapor and liquid streams from the flash tank are on the same layers. With the large vapor flow rate, this configuration either has a very large pressure drop for the vapor portion or very low heat transfer coefficients for the liquid. This configuration also assumes that the feed gas is over 40°F at the point in the exchanger where the liquid stream from the flash tank exits the exchanger. Since this stream enters the demethanizer tower, temperature fluctuations in this stream could cause column upsets. An alternate configuration eliminates the two problems as shown in Figure 10. The feed gas travels the entire length of the exchanger. The propane liquid is split such that a portion is directed to the layers with flash drum vapor and the balance routed to the layers with the flash drum liquid. The flow length for both the vapor and liquid process streams from the flash drum is the same. The propane is introduced at the opposite end of the exchanger from the feed gas and is forced to exit part way down by a series of transverse bars. The vapor and liquid streams from the cold separator are introduced on the other side of the transverse bar and exit the exchanger at the feed gas entrance. In this modified configuration, the number of layers for the vapor and liquid streams from the cold separator are independent. The exit temperature of the flash drum liquid could be controlled by adjusting the flow rate of the flash drum vapor stream through a bypass valve. Table VIII gives the dimensions of the compact exchanger.

Table VIII. Exchanger Information for Case 4.

		Side A Feed Gas	Side B Vapor	Side C Liquid	Side D Propane	Area ft^2	Volume ft^3	Cost US$
Compact	Fins:							
	Type	Serrated	Serrated	Serrated	Perforated			
	Height (in.)	0.25	0.25	0.25	0.25			
	Thickness (in.)	0.016	0.016	0.016	0.010			
	Density (1/in.)	17	17	19	14			
	Length (in.)	60	20	20	40			
	72in. long, 18in. wide, 22in. tall							
	Stacking: 10[A(BD)A(CD)A(BD)A]					4410	16.6	$70,000
Shell-and-Tube		Tube 0.75in. OD 16ft. long 1.25 Pitch ratio Triangular Carbon Steel 2 passes	Shell 24in. ID 25% Baffle Cut 8in. Baffle Sp					
		Tube 0.75in. OD 8ft. long 1.25 Pitch ratio Triangular Hair-pin		Shell 4in. ID				
		Tube 0.75in. OD 8ft. long 1.25 Pitch ratio Triangular Carbon Steel 4 passes			Shell 36in. ID 24in. Bundle Dia. Kettle	2520	112	$50,000

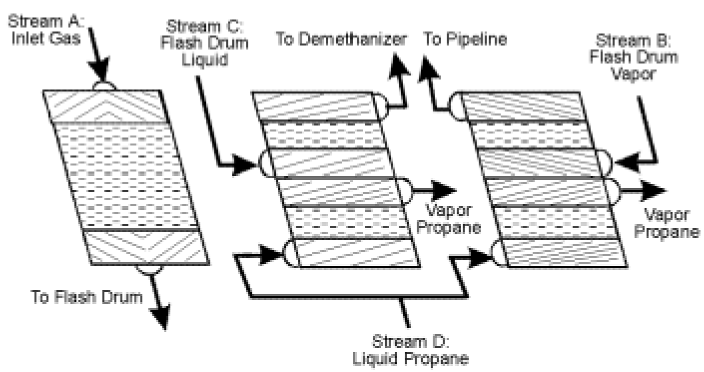


Figure 10. Stacking Arrangement and Fluid Routing for Case 4.

For the shell-and-tube network, Figure 11 shows the combination of parallel and series exchangers. In this configuration, the feed gas is divided in direct proportion to the duties of the flash drum vapor and liquid streams. Additional cooling is provided by propane refrigeration. Figure 11 also gives the duty versus temperature curves for all the exchangers. There are no impossible temperature crosses; therefore, the network is thermodynamically feasible. Table VIII also provides the dimensions for the shell-and-tube exchangers.

In this case, the compact exchanger has about 1/6 the volume of the shell-and-tube network. However, the shell-and-tube network costs less than the compact exchanger. The non cryogenic operating temperatures combined with large mean temperature difference of 26°F in the exchangers makes the shell-and-tube a more economical alternative.

Table IX. Comparison among the Four Cases

			Brazed			Shell-and-Tube		
Case	MTD °F	Duty MMBTU/hr	Area ft^2	Volume ft^3	Cost US$	Area ft^2	Volume ft^3	Cost US$
1	20	9.5	15200	60.6	$80,000	10250	314	$200,000
2	4	1.9	12000	41.7	$60,000	6900	105	$205,000
3	10	5.5	17100	60	$80,000	13610	297	$400,000
4	26	2.4	4420	16.6	$70,000	2520	112	$50,000

For all four cases, Table IX reports the mean temperature, duty, initial cost, area, and volume for both the brazed and corresponding shell-and-tube exchangers. This table suggests that brazed exchangers are more economical for small mean temperature differences and large duties. However, for large mean temperature differences and relatively small duties, the shell-and-tube networks are the more attractive option.

CONCLUSIONS

Historically, engineers may have not fully utilized the brazed exchanger technology. This is partially because of the lack of design equations in the open literature and the extremely complex nature of the design equations. Even though these exchangers should only be used with relatively clean process streams, the advantages of close temperature approaches, true countercurrent flow, and a unique ability to exchange heat with multiple streams make them viable alternatives to traditional shell-and-tube exchangers. As process simulators, such as PROSIM, incorporate brazed exchanger design equations into their utilities, engineers can more readily make comparisons between the different exchangers.

This paper shows that brazed exchangers are more economical from an initial capital cost standpoint, especially when the temperature approach for the process streams is less than 10°F. A temperature approach greater than 10°F may favor shell-and-tube exchangers. For process streams approaching cryogenic conditions, the brazed aluminum exchanger is less expensive than shell-and-tube exchangers due to the superior mechanical properties of aluminum. In addition, shell-and-tube exchangers for cryogenic operations require special alloys which increase initial costs. For the four examples considered the brazed exchangers occupy significantly less volume than the corresponding shell-and-tube exchangers or networks. Given the power to make these exchanger comparisons relatively easily, engineers can make informed decisions about heat exchanger equipment which should result in increased performance and profits.

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