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Seminario de diseño de Procesos
UOP

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HEAT EXCHANGERS

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INTRODUCTION

SCOPE AND OBJECTIVES

We do not expect to make a heat exchanger expert out of you; this takes years of specialized training and experience.

What we want to do is to give you a better understanding of some of the process considerations that go into the specification and design of heat exchangers used in petroleum refineries and to supplement your knowledge of heat transfer, that you have acquired in school and on the job, so that you will be better able to:

- 1) review quotations for heat exchanger and make selections
- 2) predict what will happen to the performance of a heat exchanger when certain process conditions are changed
- 3) make minor revamps involving heat exchangers
- 4) trouble-shoot heat exchanger problems.

OUTLINE OF PRESENTATION

I. Basic Heat Transfer Theory

II. Process Considerations

- A. Purpose
- B. Temperature Approaches
- C. Temperature of Products Going to Storage
- D. Fluid Routing
- E. Allowable Pressure Drops
- F. Various Types of Reboilers

III. Mechanical Considerations

- A. General
- B. Shell and Tube Exchangers
- C. Hairpin Type Exchangers
- D. Air Cooled Exchangers

IV. Appendix A (Exhibits)

Appendix B (Problems)

Appendix C (Excerpts From TEMA)

Appendix D (Supplement)

HEAT EXCHANGERS IN PETROLEUM REFINERIES

I. Basic Heat Transfer Theory

A. The fundamental equation for convection heat transfer is:

$$Q = UA \Delta T_m$$

where Q = Heat transferred, BTU/Hr.

U = Overall heat transfer coefficient,
BTU/(Hr.)(°F)(Sq.Ft. outside surface)

A = Outside heat transfer surface, Sq.Ft.

ΔT_m = Corrected log mean temperature difference,
°F

B. Q , the heat transferred, or duty of the exchanger, can be

1. All sensible heat, determined by the equation:

$$Q = W C_p \Delta T$$

where W = fluid flow rate, Lb/Hr.

C_p = fluid specific heat, BTU/(Lb)(°F)

ΔT = fluid temperature change, °F

2. All latent heat, determined by the equation:

$$Q = W \lambda$$

where W = fluid flow rate, Lb/Hr

λ = latent heat, BTU/Lb.

3. Or, for mixtures of two or more components, a combination of sensible and latent heat.

C. U , the overall heat transfer coefficient, = $\frac{1}{R}$;

$$1. R = r_s^f + r_s^d + r_m + r_t^d \left(\frac{A_o}{A_i} \right) + r_t^f \left(\frac{A_o}{A_i} \right)$$

where r_s^f = film resistance of shell side fluid,
(Hr)(°F)(Sq.Ft. of outside surface)/BTU

r_s^d = fouling resistance on outside of tubes,
(Hr)(°F)(Sq.Ft. of outside surface)/BTU

r_m = resistance of tube wall referred to outside
surface of the tube, including extended
surface if present,
(Hr)(°F)(Sq.Ft. of outside surface)/BTU

r_t^d = fouling resistance on inside of tubes,
(Hr)(°F)(Sq.Ft. of inside surface)/BTU

r_t^f = film resistance of tubeside fluid,
(Hr)(°F)(Sq.Ft. of inside surface)/BTU

$\frac{A_o}{A_i}$ = Ratio of outside to inside surface of
tubing.

2. The fluid film resistances, r_s^f and r_t^f , are determined from correlations involving the tube diameters, the fluid flow rate and the viscosity, thermal conductivity, and specific heat of the fluid. (See Exhibit 1, Appendix A, for some typical film resistances.)
3. Values for the fouling resistances, r_s^d and r_t^d may be known from previous operating experience or estimated from tables of typical fouling resistances found on pages 140-142 of the Standards of Tubular Exchanger Manufacturers Association, which is usually referred to as TEMA. (SEE APPENDIX 'C')
4. The resistance of the tube wall, r_m , for a bare tube, is calculated by the following equation:

$$r_m = \frac{t_w D_o}{12 k_m (D_o - t_w)}$$

where D_o = tube O.D., in.

t_w = tube wall thickness, in.

k_m = tube-metal thermal conductivity,
BTU/(Hr)(Sq.Ft.)/(°F/Ft.)

- D. ΔT_m , the logarithmic mean temperature difference is calculated by this equation:

$$\Delta T_m = \frac{GTDD - LTDD}{\ln \frac{GTDD}{LTDD}}$$

where GTDD = the greater terminal temperature
difference

LTDD = the lesser terminal temperature
difference

1. This equation gives the log mean temperature difference for either true counterflow or cocurrent flow exchangers. Only hairpin type exchangers and exchangers with one pass on each side or with two passes on each side fall into this category.

2. For exchangers with two or more tube passes and one shell pass, true counterflow is not experienced, and therefore a correction factor must be applied to the calculated counterflow MTD.

Charts for determining MTD correction factors for several types of exchangers are given as excerpts from TEMA. (SEE APPENDIX C)
The assumptions on which these charts are based are:

- a. The over-all heat transfer coefficient is constant
- b. Each pass has the same amount of heat transfer surface
- c. The mass flow rate of each fluid is constant
- d. The specific heat of each fluid is constant
- e. There are no phase changes in any part of the exchanger
- f. Heat losses are negligible
- g. Each fluid stream may be characterized by a single temperature at any cross section in the exchanger.

It is not advisable to design an exchanger with a MTD correction factor less than 0.8, because in practice some of these assumptions are no longer valid at the lower correction factors. Also, it has been shown that it is not economically feasible to design an exchanger with a correction factor less than 0.8.

3. Straight line temperature-enthalpy relationships of both fluids is assumed when calculating a MTD as shown previously. This is not always true, especially where there is condensation or vaporization involved. In this case a weighted MTD must be calculated.

II. Process Considerations

A. Purpose - why use heat exchangers?

1. To cool process streams

- a. Example: gasoline product going to storage is cooled to reduce losses because of its vapor pressure.
- b. Example: Hydrar unit intercoolers remove the heat of reaction between reactors.
- c. Example: Absorber intercoolers on gas concentration units remove the heat of absorption and thereby increase the efficiency of the absorber.
- d. Example: Fractionator condensers condense the overhead, part of which may be the product and the other part of which may be reflux that is returned to the column to help effect a separation.

2. To heat process streams
 - a. Example: Fractionator reboilers are used to add heat to fractionation column which effects a separation.
 - b. Example: Reactor charge heaters are used to heat the charge up to the reaction temperature.
3. To exchange heat between hot and cold process streams
 - a. Example: Platforming combined feed exchanger which is used to heat the reactor charge by exchanging heat with the reactor effluent.
 - b. Example: Fractionator feed-bottoms exchanger which is used to heat the feed by exchanging heat with the bottoms.

B. Temperature Approaches

Definition: For a process stream being cooled, it is the difference between the hot fluid's outlet temperature and the cooling medium's inlet temperature. For process stream being heated, it is the difference between the process fluid's outlet temperature and the heating medium's inlet temperature. When both streams are process streams, the temperature approach is the lesser terminal temperature difference.

1. Example: Water coolers and condensers can be designed for approach temperatures as low as 10 to 15°F.
2. Example: Air coolers and condensers are normally designed for a minimum approach temperature of about 20°F, although most are in the 30 to 40°F range.
3. Example: Steam heated reboilers can be designed for an approach of 25°F for very small process unit. Minimum exchanger surface is reached at approach temperatures above 100°F.
4. Example: Hot oil heated reboilers and exchangers are designed for approach temperatures of about 50°F or higher.
5. Example: Platforming combined feed-reactor effluent exchangers are designed for MTD of about 75 to 85°F. The economics of incremental surface are compared with fuel and cooling costs to arrive at the right approach temperature.
6. Example: The crude column overhead cooling system is an example of optimizing a vapor exchanger, an air cooler, and a water trim cooler.

C. Temperature of Product Streams Going to Storage

1. Gasolines and lighter hydrocarbons should always be cooled to 100°F or less.
2. Kerosene and middle distillates are usually cooled to 125°F to stay below a normal flash point of 150°F.
3. Vacuum gas oils are normally cooled to 150°F.
4. Reduced crude or residuum are cooled to 175°F or 200°F so as to be below the boiling point of water, which may accumulate in the storage tanks, and which could lead to an explosion if the product is not sufficiently cooled.

D. Fluid Routing in Shell and Tube Exchangers

1. Fouling fluids should be put in the tube side, if other factors do not prohibit it. Chemical cleaning methods that are now available make it unnecessary to follow this practice as strictly as it had been in the past.
2. Fluids having a viscosity greater than about 2 centistokes should be put on the shellside to reduce the amount of heat transfer surface required. A higher heat transfer coefficient is obtained on the shellside, because of turbulence brought about by flowing across the tubes & baffles.
3. Fluids containing suspended solids should be put on the tubeside.
4. It is usually more economical to put the fluid with the higher pressure on the tubeside. If the high pressure fluid is routed through the shell, the tube bundle and shell must be designed for the high pressure, however, if the high pressure fluid is through the tubes only the channel, channel cover, and tube bundle must be designed for high pressure. Another reason why it is usually more economical to route the high pressure fluid through the tubes is that the internal working pressure of a given tube is about twice the external working pressure.
5. Likewise, it is usually more economical to put the higher temperature fluid on the tube side.
6. Cooling water is usually put in the tube side for ease of cleaning and because of the possible presence of solids.
7. If one stream is corrosive and requires alloy materials, it is usually more economical to put it on the tube side since less alloy material will be required.

E. Allowable Pressure Drops

1. The allowable pressure drop usually specified on the water side of coolers and condensers is 10 psi per exchanger (shell) in series. Experience has shown that this is approximately the pressure drop that is required to meet our 5 ft/sec minimum cooling water velocity limitation. (This velocity limitation in turn is specified to minimize cooling water fouling.) A lower drop may be required where the system pressure is very low or on exchangers at high elevations. In this case the water side fouling factor should be increased, accordingly.
2. The normal pressure drop allowed for fluid exchangers ranges from about 5 to 25 psi., depending on the availability of pressure and the characteristics of the system. Usually a minimum of 5 psi per exchanger (shell) in series is allowed on each side.
3. A minimum of 2 psi pressure drop per exchanger (shell) on each side, is usually allowed for combined feed exchangers with multipass tubes, on low pressure units. For high pressure units this is increased to about 5 psi per exchanger on each side.

For vertical Platforming combined feed exchangers with single pass tubes the allowable pressure drop varies from 10 psi total, both sides, for exchangers with 40 ft. tubes to 12 psi total, both sides, for exchangers with 60 ft. tubes.

4. Condensers in atmospheric pressure service or above, will normally have allowable pressure drops of about 1 to 5 psi on the process side.
5. Reboilers will normally have between 0.5 and 2 psi pressure drop. Most thermosyphon reboilers are specified for 0.5 to 0.75 psi with the higher pressure columns, such as depropanizers, requiring the lowest drops. Once-through and vertical reboilers can usually stand 2 psi pressure drop because of the high static head available.
6. Condensers in vacuum service will usually be specified for 10 to 25 mm Hg. pressure drop.

F. Various Types of Reboilers (See Exhibits 3A-F, Appendix A)

1. Kettle Reboiler

- a. Operation - Because their operation is not critically dependent on the circulation of large quantities of two-phase fluid through external piping such as in the case of thermosyphon and once-through reboilers, Kettle reboilers are relatively insensitive to large changes in process operating conditions.

b. Advantages

- (1) Weir holds constant level above tubes
- (2) Equivalent to at least one theoretical tray
- (3) Temperature of product kept at minimum

c. Disadvantages

- (1) Surge volume for liquid is low
- (2) Have greater tendency to foul than other reboiler types, since heavy residual material accumulates continuously behind weir. Therefore, they are not recommended for potentially fouling service especially when bottoms draw-off is less than about 20 percent of the required boil-up

2. Internal (Stabbed-In) Reboilers

a. Operation - Essentially the same as for kettle reboilers.

b. Advantages

- (1) Saves cost of external piping and shell
- (2) Equivalent to at least one theoretical tray
- (3) Weir holds constant level above tubes
- (4) Full use of the bottom portion of the column can be taken for surge volume

c. Disadvantages

- (1) Their use in small columns can be impractical
- (2) Has same fouling tendency as kettle type
- (3) Bundle diameter may have to be large because tube length is limited by column diameter.

3. Vertical Thermosyphon Reboilers

a. Operation - Stable operation depends on a balance between the net static head developed by the liquid in the column and the friction and acceleration losses in the system. Therefore, sudden changes in composition or temperature can upset this balance often causing severe oscillations in the system which could lead to poor column control and a loss in heat transfer efficiency.

b. Advantages

- (1) Relatively inexpensive fixed tube sheet construction may be used
- (2) No pockets to trap dirt; high circulation rate should keep the reboiler clean
- (3) Tube side readily accessible for mechanical cleaning
- (4) Piping is short
- (5) Plot plan requirements are small

c. Disadvantages

- (1) Is not equivalent to one theoretical tray
- (2) For wide boiling material, the outlet temperature is high
- (3) Column must be elevated to provide head
- (4) Level control must be restricted to narrow range

4. Horizontal Thermosiphon Reboilers

a. Operation - Some what less sensitive to large changes in process condition than vertical thermosiphons

b. Advantages

- (1) No pockets to trap dirt; high circulation rate should keep the reboiler clean
- (2) Since the exchanger is of horizontal design, longer tubes may be used than in the vertical reboilers and therefore more surface can be placed in a single shell.

c. Disadvantages

- (1) Is not equivalent to one theoretical tray
- (2) For wide boiling material, the outlet temperature is high
- (3) Column must be elevated to provide head
- (4) Level control must be restricted to narrow range

5. Vertical Once-through Reboilers

a. Operation - Similar to vertical thermosiphon reboilers

b. Advantages

- (1) All advantages listed for vertical thermosiphon reboiler, plus
- (2) Temperature of product is kept at minimum
- (3) Is equivalent to one theoretical tray
- (4) Full use of the bottom portion of the column can be taken for surge volume

c. Disadvantages

- (1) Piping design is critical
- (2) Column must be elevated to provide head
- (3) Can bypass bottom tray liquid, if trap is not functioning

6. Horizontal Once-through Reboilers

a. Operation - Similar to horizontal thermosiphon reboilers

b. Advantages

- (1) All advantages listed for horizontal thermosiphon reboiler, plus
- (2) Temperature of products is kept at minimum
- (3) Is equivalent to one theoretical tray
- (4) Full use of the bottom portion of the column can be taken for surge volume

c. Disadvantages

- (1) Piping design is critical
- (2) Column must be elevated to provide head
- (3) Can bypass bottom tray liquid, if trap is not functioning

III. Mechanical Considerations

A. General

1. Code Requirements and Standards - UOP specifications require that shell and tube heat exchangers, hair pin type heat exchangers, and air cooler tube bundles shall conform to the requirements of the Sect VIII of the ASME Boiler and Pressure Vessel Code except that those exchangers in steam generator service shall conform to Sect. I of the Code on the steam side.

In the case of shell and tube heat exchangers, the ASME Code is supplemented by TEMA and UOP Specification 4-11-0. For hair pin type exchangers, UOP Spec. 4-12-0 supplements the Code. Air coolers are covered by UOP Spec. 4-13-0.

In addition, numerous other codes and standards may be referenced and certain local codes, contractor's specification, and licensee's specifications may be involved.

2. Design Pressure

- a. Normally, the design pressure of exchangers is set at a minimum of 25 psi or 10% above the maximum operating pressure, whichever is greater.
- b. In cases of exchangers operating above 1000 psig, the design pressure is sometimes taken as 100 psi or 5% above the maximum operating pressure, whichever is greater.

- c. The design pressure for the tube bundle can be reduced for certain types of exchangers, such as "combined feed vs. effluent" type, where there is assurance that both sides will always be subjected to the operating pressure together. In these cases, the tube bundle is designed for the maximum differential pressure expected, plus a safety factor of 10% of the operating pressure or 50 psi, whichever is greater. If the tube bundle is designed for a differential pressure, the exchanger nameplate should indicate this so that it can be taken into account during hydrostatic testing.

3. Design Temperature

- a. The design temperatures of exchangers should be high enough to account for failure or fouling in the preceding exchangers. Normally, about 50°F above the maximum operating temperature will be sufficient.
- b. The water side of coolers and condensers is usually designed for 250°F. However, the process fluid inlet temperature should not exceed 400°F where the cooling medium is fresh water or 300°F where the cooling medium is sea water, in order to avoid excessive scaling on the water side.
- c. When setting design temperatures of exchangers, consideration is given to the required flange rating, since a difference of a few degrees in some cases can lower the flange ratings, at a considerable savings. (ANSI-B16.5 LATEST EDITION FOR FLANGE RATING)

- 4. Materials of Construction - Normally, heat exchangers are made of carbon steel because it is relatively inexpensive, has good strength characteristics at moderate temperatures, and is readily available. However, in some cases, steel alloy containing chromium and molybdenum, or even stainless steels must be used, because of high temperatures and the presence of hydrogen, hydrogen sulfide, or sulfur in the process fluid.

Some refiners use admiralty tubes and naval brass tube sheets for all cooling water service, while others use carbon steel, but only when the water is properly treated with chemicals such as chromates, to reduce the corrosiveness to a reasonable level.

For sea water service, non-ferrous materials such as aluminum brass or copper-nickel are used for the tubes and tube sheets. Galvanic corrosion can occur between the non-ferrous tubes and tube sheets, and the carbon steel channel. Therefore, the channel is usually protected with a phenolic or epoxy type coating along with cathodic protection employing sacrificial anodes placed inside the channel. The anodes are usually zinc, magnesium or aluminum.

5. Corrosion Allowance - TEMA Standards for Class "R" heat exchangers, requires a minimum corrosion allowance of 1/8" for all carbon steel pressure parts except tubes. (TEMA defines carbon steel as any steel or low alloy falling within the scope of Part UCS of ASME Code.)

Since practically no corrosion allowance can be incorporated in the tubes, alloy is used whenever any appreciable corrosion is expected.

B. Shell and Tube Exchangers

1. Nomenclature - TEMA, pages 4, 5 and 6
2. TEMA Type Designation - TEMA, page 3
3. Thermal Design involves the determination of the amount of heat transfer surface required, the tube diameter, the tube length, the tube pitch, the number of tube passes, the baffle cut and the baffle pitch.

} SEE APPENDIX 'C'

In general, the individual film resistances decrease with an increase in mass velocity, specific heat, and thermal conductivity and increase with increases in tube diameter and viscosity. On the other hand the fluid pressure drop decreases with an increase in density and tube diameter, but increases with an increase in mass velocity and viscosity. The result is an increase in pressure drop for an increase in the overall heat transfer coefficient, and therefore less required heat transfer surface, or putting it another way its a balance of operating costs (pressure drop) versus capital cost (heat transfer surface).

4. Special Considerations

- a. Slurry Handling - To avoid plugging or excessive erosion in exchangers handling slurry oil on FCC units, certain features must be incorporated in their design:
 - (1) Routing the slurry oil through the tubes
 - (2) Limiting the minimum, as well as the maximum, slurry oil velocity
 - (3) Using tubes no smaller than 1" O.D. and no thinner than 12 BWG.
 - (4) Using straight tube construction
 - (5) Designing horizontal exchangers with the slurry in horizontal flow or downflow through the bundle
 - (6) Providing a replaceable deflector or impact baffle at the inlet to divert the slurry flow from the pass partition plate

- b. 'HF' Acid Service - Exchangers handling HF acid must include the following features in their design:
 - (1) To minimize chances of acid leakage to the process area, exchanger design shall provide as few joint closures as is practical, e.g. U-tubes, removable TEMA "C" stationary head, etc.
 - (2) ~~Strength~~ welding tube to tube sheet joints, where the steam pressure is higher than the process stream containing 'HF' acid.
 - (3) Provision that process streams containing 'HF' acid flow downward to assure good drainage
 - (4) No threaded surface are permitted in the presence of 'HF' acid to avoid stress cracking
- c. Integral Low Fin Tubing per ASTM A 498 (ferrous) and B 359 (non-ferrous)
 - (1) UOP specifications permit the use of this tubing when the fluid film coefficients corrected for fouling, show that the shell side is controlling by a ratio of 2:1 or more (EX: STEAM IN TUBES; HCBN IN SHELL)
 - (2) In revamps, the capacity of an exchanger may be increased by retubing the bundle with integral low fin tubing

5. Revamping

Exchangers in sensible heat transfer service, that is no phase change on either side, may be revamped for new duties by using the relationships shown on Fig. 9-10 of the Natural Gas Processors Suppliers Association's Engineering Data Book, 10th Edition, 1987 (See Exhibit 4, Appendix A)

C. Hairpin Type Exchangers

1. Description

- a. Double pipe - one U-tube element inside U-shell. Element may be finned or bare depending on application. (See Exhibit 5, Appendix D)
- b. Multi-tube - one U-shaped bundle composed of U-tube elements inside a U-shell. Elements may be finned or bare depending on application. (See Exhibit 6, Appendix D)
- c. Series - Parallel Hookups - Frequently, one heat exchanger section does not have sufficient capacity for a given duty. In these cases, several exchanger sections will be hooked up in series and/or in parallel.

2. Advantages

- a. For small duties, hair pin type exchangers are usually less expensive than shell and tube exchangers. This is because hair pin type exchangers are of standard design, while shell and tube exchangers must be custom built for each application. Also, the cost of shell and tube heat exchangers on a dollars per sq. ft. basis, increases as the exchanger size decreases.
- b. Because of the modular like construction, hair pin exchangers can frequently be revamped by rearranging the hookup or by adding similar sections.

3. Disadvantages

- a. Manifolds for parallel hook ups can be expensive.

4. Applications

- a. Double pipe - usually where the duty is less than one million BTU per hour.
- b. Multitube - usually where the duty is greater than one million BTU per hour and less than five to ten million BTU per hour.

D. Air Cooled Exchangers

1. Description - Major Parts

- a. The tube bundle or coil which is composed of the headers, the tubes, and the frame
- b. The structure which includes the supporting columns, bracing, the plenum chamber, the fan deck, and the fan ring
- c. The mechanical equipment which typically consists of the fan and the drive assembly.

2. Nomenclature (See Exhibits 7A-G, Appendix D)

3. Drives

- a. Most air cooler fans are driven by means of electric motors. For small fans, say up to about 5 ft diameters, the fans are directly driven. For fans with diameters greater than about 5 ft., right angle gear drives are required, unless the electric motors are rated below 30 hp, in which case V-belt drives may be used.
- b. A few air cooler fans in refinery services are driven with steam turbines, in which case right angle gear drives are required.

4. Headers - There are four types of air cooler header designs.

- a. By far, most air coolers in refinery service are the plug header or welded box design. They can be designed for pressures up to about 3000 psig. These headers are usually made by joining the four side plates with corner welds and then welding on the end plates. Another fabrication approach is to join two U-shaped channels together with double-butt welds and then welding on the end plates. The latter approach is considered to be of superior construction, especially for high pressure units.

The main distinguishing feature of this type of header is, of course, a plug opposite the end of each tube. Removal of the plug allows access to the tube for re-rolling or cleaning the tube.

- b. For pressures above 3000 psig, the manifold type header is used. In this design the tubes are welded into pipe manifolds on the inlet and outlet, with U-bends on the floating ends. There are no plugs to provide access to the tubes for cleaning or inspection.
- c. Another type of header is the removable-cover-plate header. It is similar in design to the plug header except the front (cover) plate is bolted to the rest of the header, thus providing access to the tubes. It is usually used for services where heavy fouling is expected - where it is felt that greater access to the tubes, than is provided by removable plugs, is required for cleaning. The design pressure of these headers is limited to about 300 psig, due to difficulties encountered in designing the long flanged joints.
- d. A variation of the removable-cover-plate header is the removable-bonnet header, which essentially has the same features as the removable-cover-plate header, except that the tube sheet is flanged to the header rather than the cover plate. This means that the piping must be disconnected to gain access to the tubes, in the case of the removable-bonnet header. This type of header construction does have the advantage, that the header, exclusive of the tube sheet can be replaced independent of the tube bundle.

5. Finned Tubing - the following describes several types of commonly used finned tubing (See Exhibit 8, Appendix D):

- a. Aluminum fins wrapped under tension and mechanically embedded in a groove (0.010 in. $\pm$ 0.002 in. deep) spirally cut into the outside surface of the tube.

- b. Inner tube mechanically bonded to an aluminum outer tube from which fins have been formed by extrusion.
- c. Aluminum footed fins wrapped under tension over the outside surface of a tube with the fin ends secured to prevent loosening or unraveling of the fins.
- d. Aluminum fins edge-wound and wrapped under tension over the outside surface of the tube, with the fin ends secured to prevent loosening or unraveling of the fins.
- e. Tubes to which fins are bonded to the outside surface by hot dip galvanizing, soldering, brazing, welding or some other method of attachment.

Since most air cooler tubes used in petroleum refining are steel, temperature limitations are put on the use of finned tubes of the "a" through "d" types to minimize the gap resistance caused by the aluminum fin expanding away from the steel tube. The usual allowable maximum metal temperatures are:

Type 'a'	700°F (embedded)
Type 'b'	500°F (extruded)
Type 'c'	400°F (Footed)
Type 'd'	250°F (wrapped-on)

6. Forced Draft vs. Induced Draft

- a. Definition: forced draft - the fan forces or pushes the air across the tube bundle; induced draft - the fan pulls or sucks the air across the bundle.
- b. Comparison
 - (1) Forced draft requires less horsepower than induced draft because the fan is moving unheated air
 - (2) On a forced draft unit the fan and drive assembly is located in the unheated air
 - (3) There is less air recirculation with an induced draft unit because the air is discharged higher above the section and at a greater velocity than in a forced draft cooler
 - (4) The bundle is protected from the weather in an induced draft unit by the plenum and the fan

7. The design of the manifolding for multiple bundle units is important for good distribution, especially where two-phase flow is involved.

8. Thermal Design

- a. Tubeside - the same correlations that are used on the tubeside of shell at tube exchangers are used for the tubeside of air coolers

- b. Finned surface - an allowance for the efficiency of the extended surface must be made; it is usually close to about 0.95 for air coolers, however. Fin efficiency is defined as the ratio of the difference between the average temperature of the fin and the air temperature and the difference between the fin base temperature and the air temperature. (There has to be a temperature gradient between the tip of the fin and the base of the fin to conduct heat along the fin.)
 - c. Heat transfer and pressure drop correlations for the air side are generally of a proprietary nature, and have been developed for the particular geometry of a particular manufacturer's tube bundle design.
 - d. The air flow is a variable that is resolved by an economic balance between fan horsepower and surface.
9. Design Ambient Air Temperature - the usual basis for selection is similar to that shown on Exhibit 9, Appendix A. For example, for the United States, it is the temperature which will not be exceeded more than 5% of the 12 hours during the middle of the day in June to September inclusive, of a normal summer. It is felt that use of the maximum recorded temperature would result in air coolers that are uneconomically large. On the other hand, some refiners will add a safety factor of a few degrees, to the temperature determined in the above manner, to allow for some additional heating of the ambient air in a refinery, because of the heat given off by the various pieces of equipment.
10. Control - there are several commonly used schemes for controlling the process fluid outlet temperatures from air coolers:
- a. Two-speed motors
 - b. Multiple fans
 - c. Variable pitch fans
 - d. Louvers

Control may be required for process reasons or to prevent freezing of the process stream.

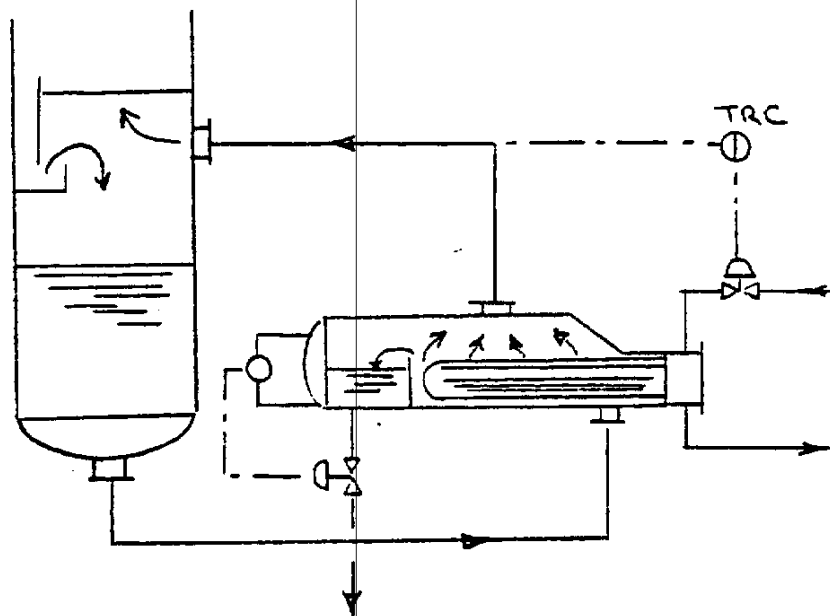
APPENDIX 'A'

(EXHIBITS)

EXHIBIT 1, TYPICAL FILM RESISTANCES

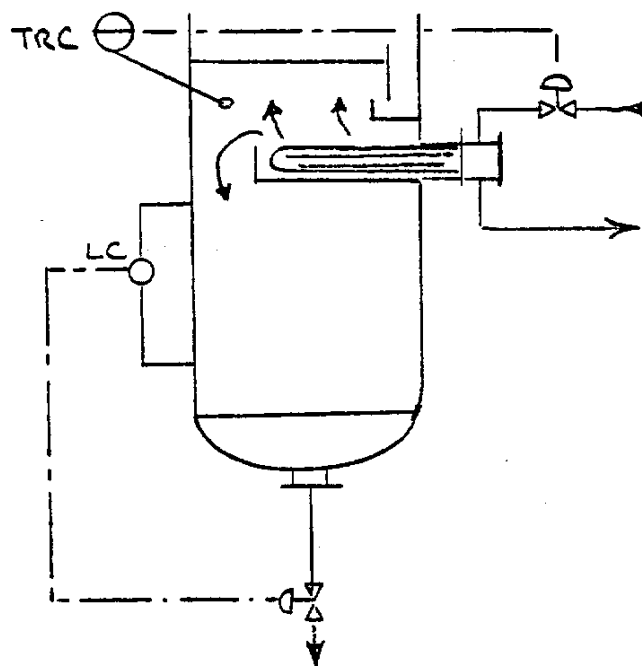
	(ENGLISH) hr-°F-ft ² /BTU	(METRIC) hr-°C-M ² /Kcal
Liquid Water:	0.001	0.00020
Boiling Water:	0.0008	0.00016
Condensing Steam:	0.0005	0.0001
Liquid Light Hydrocarbon (Propane-Pentane):	0.003	0.0006
Boiling Light Hydrocarbon (Propane-Pentane):	0.004	0.0008
Condensing Light Hydrocarbon (Propane-Pentane):	0.006	0.0012
Liquid Kerosene:	0.005	0.0010
Boiling Kerosene:	0.008	0.0016
Condensing Kerosene:	0.007	0.0014
Liquid Crude Oil:	0.01	0.002
Liquid Bunker 'C' Fuel Oil:	0.02	0.004
Natural Gas at 50 psig:	0.018	0.0036
Natural Gas at 100 psig:	0.013	0.0026
Natural Gas at 1000 psig:	0.005	0.001
Low Pressure Platformer Combined Feed or Reactor Products:	0.007	0.0014
Moderate Pressure Unibon Combined Feed or Reactor Products:	.004	0.0008
High Pressure Unibon Combined Feed or Reactor Products:	.003	0.0006

EXHIBIT 3A



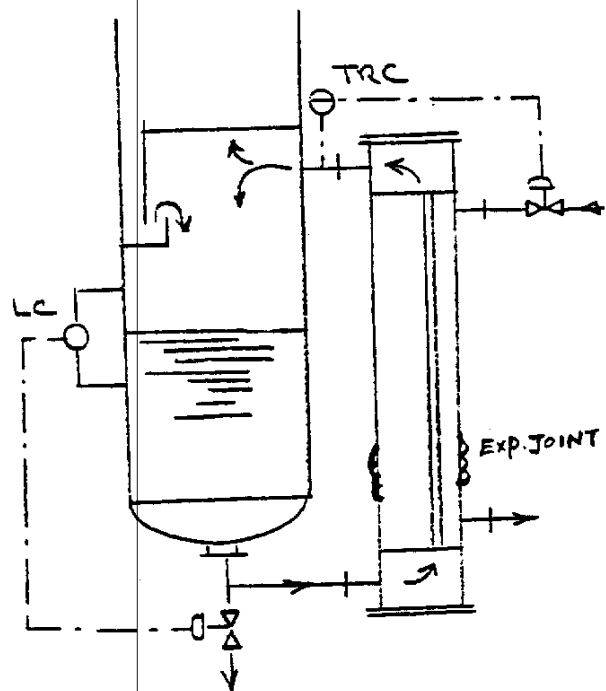
KETTLE REBOILER

EXHIBIT 3B



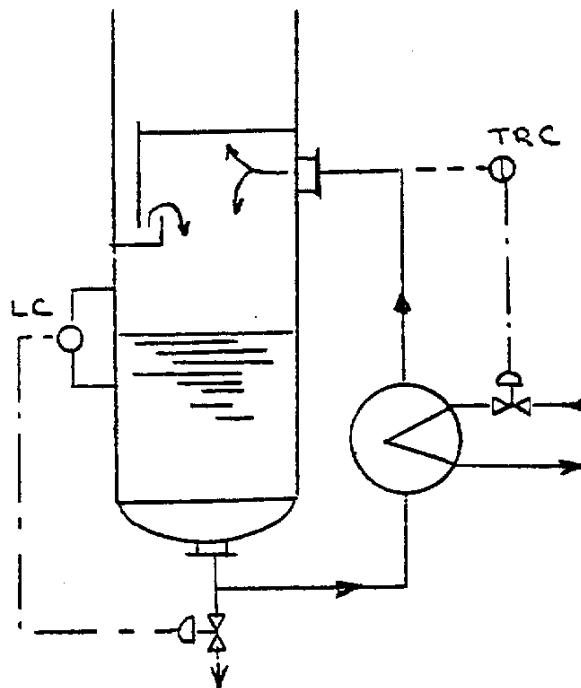
INTERNAL (STABBED-IN) REBOILER

EXHIBIT 3C



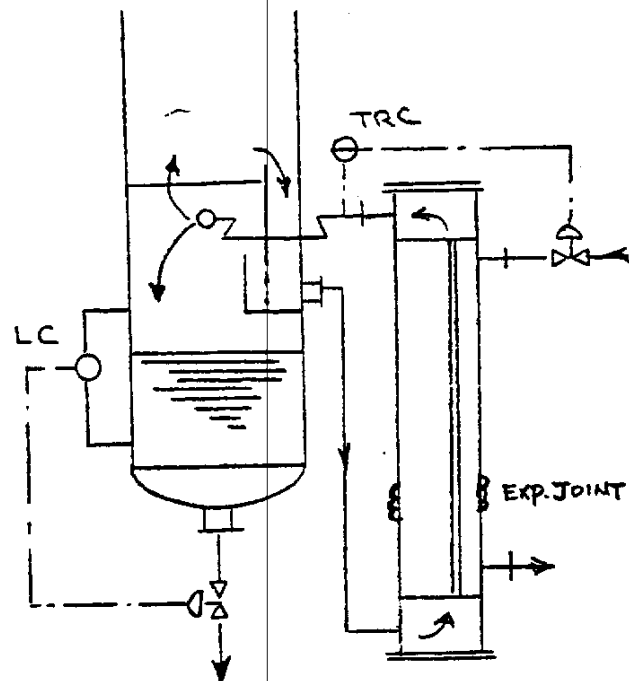
VERTICAL THERMOSYPHON REBOILER

EXHIBIT 3D



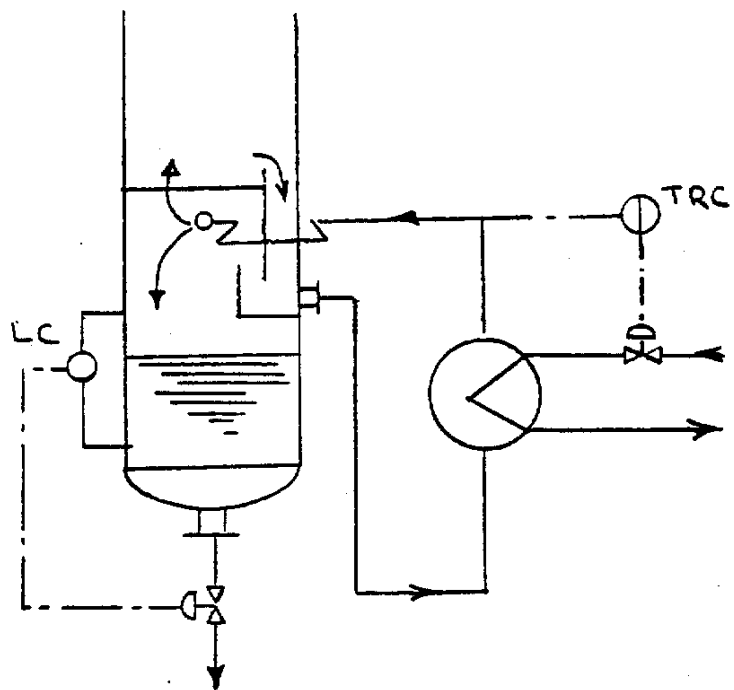
HORIZONTAL THERMOSYPHON REBOILER

EXHIBIT 3E



VERTICAL ONCE-THROUGH REBOILER

EXHIBIT 3F



HORIZONTAL ONCE-THROUGH REBOILER

FIG. 9-10
Variables in Exchanger Performance

Variable*	Flow Regime	$f_2 = (f) \cdot (r_1) \uparrow$		$\Delta P_2 = (f) \cdot (\Delta P_1) \uparrow$	
		Shell (f)	Tube (f)	Shell (f)	Tube (f)
Viscosity	Turbulent	$(\mu_2/\mu_1)^{0.27}$	$(\mu_2/\mu_1)^{0.47}$	$(\mu_2/\mu_1)^{0.35}$	$(\mu_2/\mu_1)^{0.2}$
Viscosity - bulk to wall correction	Streamline		$\left[\frac{\mu_1 \cdot \mu_{w2}}{\mu_2 \cdot \mu_{w1}} \right]^{0.14}$		(μ_2/μ_1)
Thermal conductivity	Turb. or Streamline	$(k_1/k_2)^{0.67}$	$(k_1/k_2)^{0.67}$		
Sp. heat capacity	Turb. or Streamline	$(C_{p1}/C_{p2})^{0.33}$	$(C_{p1}/C_{p2})^{0.33}$		
Mass velocity	Turbulent	$(G_1/G_2)^{0.6}$	$(G_1/G_2)^{0.8}$	$(G_2/G_1)^{1.85}$	$(G_2/G_1)^{1.8}$
(or mass flowrate)	Streamline		$(G_1/G_2)^{0.33}$		(G_2/G_1)
Density	Turb. or Streamline			(ρ_1/ρ_2)	ρ_1/ρ_2
Tube diameter	Turbulent	$(D_{o2}/D_{o1})^{0.4}$	$(D_{i2}/D_{i1})^{0.2}$	$(D_{o1}/D_{o2})^{0.15}$	$(D_{i1}/D_{i2})^{1.2}$
Tube diameter	Streamline		$(D_{i2}/D_{i1})^{0.33}$		$(D_{i1}/D_{i2})^2$
Tube length	Streamline		$(L_2/L_1)^{0.33}$		
Tube passes	Turb. or Streamline				(N_{p2}/N_{p1})
No. baffle spaces	Turb. or Streamline			SP_2/SP_1	
No. tube rows crossed ‡				RC_2/RC_1	

* Use consistent units for any one variable in both cases.

† f is the ratio of the new value to the old value for a given variable. The overall f is the product of the individual fs.

‡ Number of rows of tubes exposed to cross flow (as opposed to parallel flow). This number is determined by baffle and bundle geometry.

FIG. 9-11
Base Values for Use with Fig. 9-10⁽¹⁾

Fluid	Flow Regime	Local r	Tubeside (One Pass)						
			k	Cp	p	ΔP/ft	μ(2)	G _i	D _i
Water	Turbulent	0.0009	0.358	1.0	62.1	0.068	0.764	265	0.62
HC Oil	Turbulent	0.0038	0.0788	0.5	46.9	0.061	0.726	185	0.495
Methane	Turbulent	0.0058	0.02	0.54	0.27	0.137	0.0113	31.2	0.62
HC Oil	Streamline	0.049	0.0716	0.526	51.3	(4)	(3)	42.3	0.834
			Shellside						
			k	Cp	p	ΔP(6)	μ(5)	G(7)	D _o
Water	Turbulent	0.0005	0.395	1.007	59.8	0.23	0.282	156.7	0.625
HC Oil	Turbulent	0.0028	0.0762	0.557	46.8	0.25	0.549	132.4	0.625
Methane	Turbulent	0.0038	0.037	0.655	0.23	0.09	0.0182	6.19	0.625

(1) Symbols and units are defined in Fig. 9-1

(2) Bulk average viscosity

(3) 6.62 and Wall viscosity is 27.75

(4) 0.14 psi for a 17 ft tube

(5) Average film viscosity

(6) Crossflow ΔP/baffle space/10 tuberos crossed between centroids of cut openings

(7) Average crossflow mass velocity (see crossflow area calculation in Fig. 9-13)

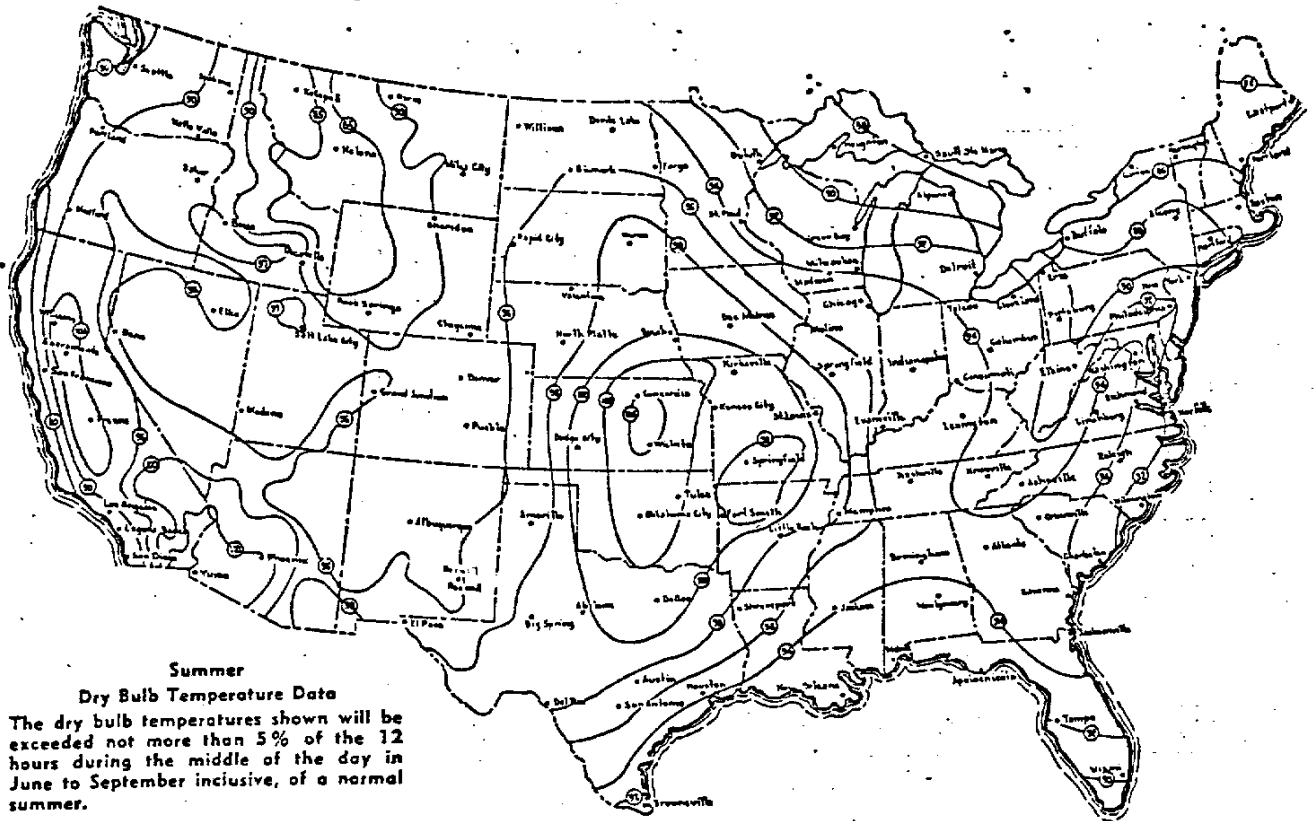


EXHIBIT [9]

Reference: Natural Gas Processors Suppliers Association's Engineering Data Book, Ninth Edition, 1972.
Page 12-4.

APPENDIX 'B'

(PROBLEMS)

PROBLEMS

PROBLEM NO. 1

Calculate the tubeside film resistance in an exchanger with two tubeside passes and 128 - 1 in. OD - 12 BWG tubes. The tubeside liquid is 57°API Gasoline with a "K" of 11.8 and a viscosity of 0.47 centipoise at 100°F, (its average temperature in the exchanger).

The tubeside flow rate is 125,000 lbs. per hr..

Use the figures on pages attached to obtain the specific heat (C_p) and thermal conductivity (K_b) values.

Use the following equation to obtain the tubeside heat transfer coefficient h_t , which, of course, equals $1/r_t^f$

$$\frac{h_t D}{K_b} = .023 \left(\frac{DG}{\mu_b} \right)^{0.8} \left(\frac{C_p \mu}{K} \right)_b^{0.4}$$

Where, h_t = the tubeside heat transfer coefficient
BTU/(hr)(sq. ft.) (°F)

D = inside diameter of the tube, ft.

K_b = the bulk (average) thermal conductivity of fluid,
BTU/(hr)(sq. ft.)(°F per ft.)

μ_b = the bulk viscosity of the fluid, lb/(hr)(ft);
($\mu = 2.42 \times \mu'$, where μ' is expressed in centipoises)

G = mass velocity, lb/(hr)(sq. ft. of cross section)

C_p = the bulk specific heat of the fluid, BTU/(lb)(°F)

(for 1" O.D. 12 BWG Tubes; ID = 0.782"; 0.4803 in² cross-sectional area)

(FOR PROB. 1)

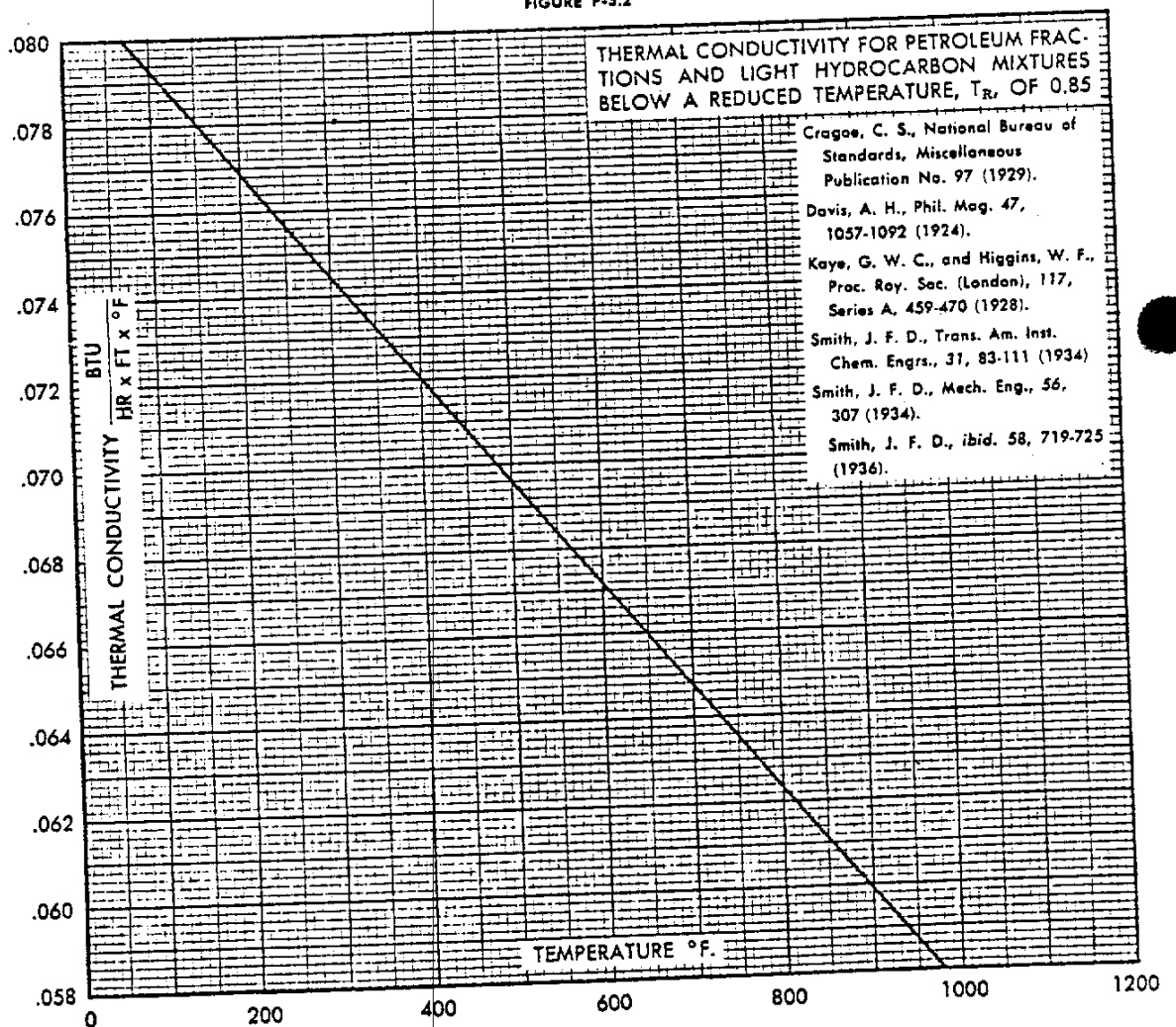
SECTION 10
Physical Properties of Fluids

TABLE P-5.1

THERMAL CONDUCTIVITY CONVERSION FACTORS

BTU hr-ft ² x °F/ft	g-cal sec-cm ² x °C/cm	watts cm ² x °C/cm	kg-cal hr-m ² x °C/m	BTU hr-ft ² x °F/in
1	.004134	.01731	1.488	12
241.8	1	4.183	360	2901
57.8	.239	1	86.1	694
.672	.002778	.01162	1	8.06
.0833	.0003447	.001441	.1241	1

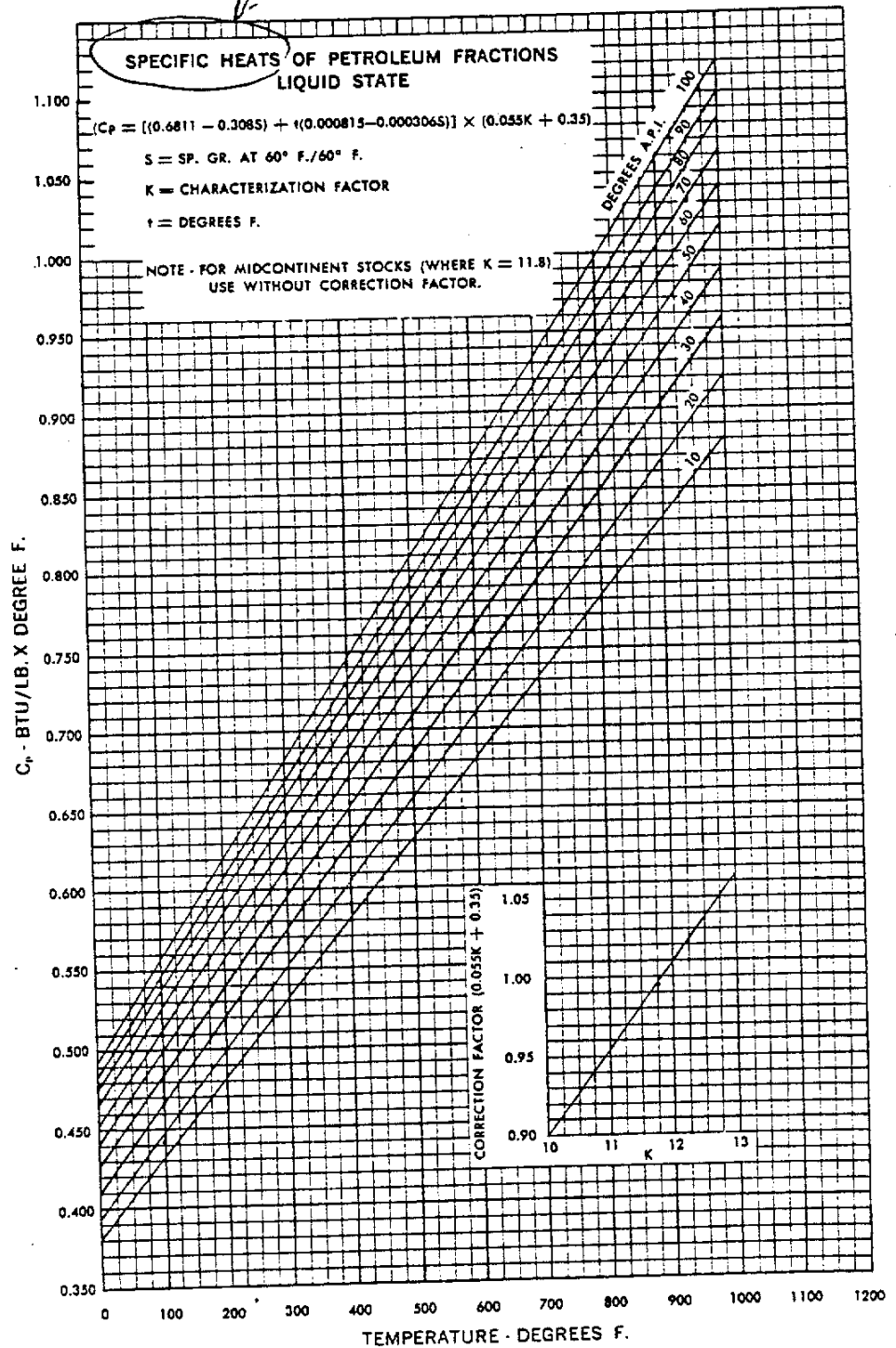
FIGURE P-5.2



(FOR PROB. 1)

SECTION
Physical Properties of Fluids

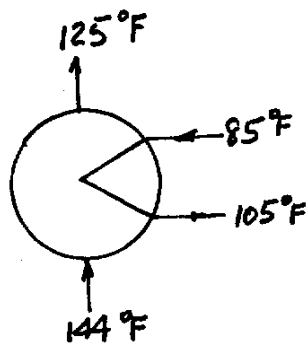
FIGURE P-2.1



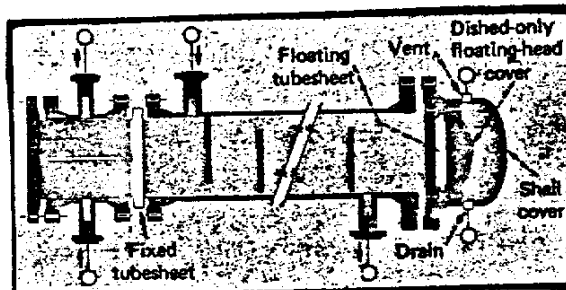
STANDARDS OF TUBULAR EXCHANGER MANUFACTURERS ASSOCIATION

PROBLEM NO. 2

Calculate the MTD correction factor for a TEMA Type 'AES' exchanger with 4 pass tubes. The tube side fluid enters at 85°F and leaves at 105°F ; the shellside fluid enters at 144°F and leaves at 125°F .



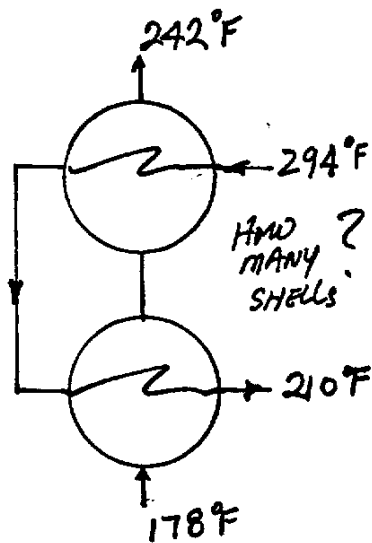
(SEE SKETCH OF
'AES' TYPE EXCHANGER
ON NEXT SHEET FOR
PROB. NO. 3)



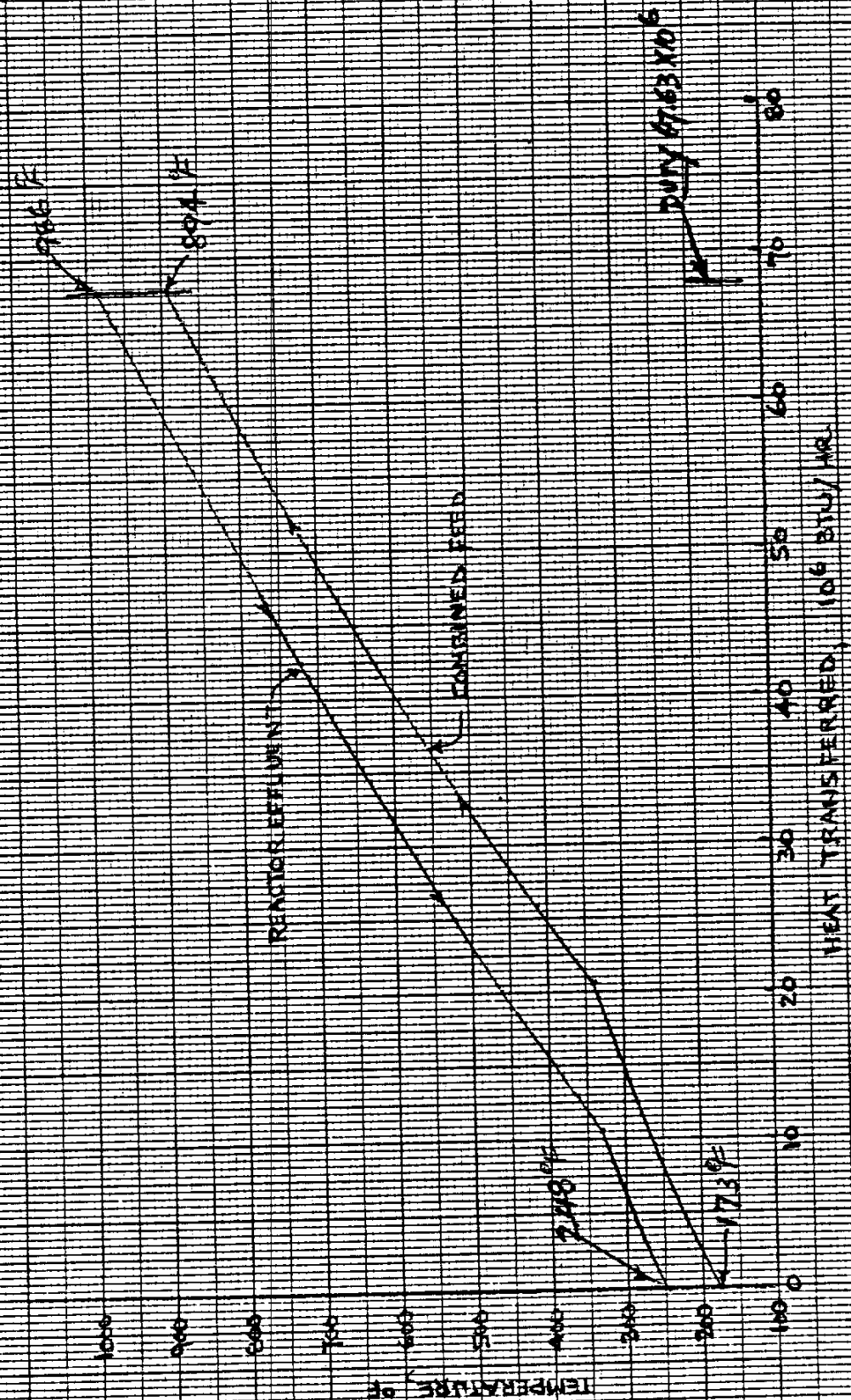
TEMA Type AES split and floating head, two tube passes

PROBLEM NO. 3

How many TEMA Type 'AES' exchangers (shells) are required in series for an exchanger unit in which the tubeside fluid enters at 294°F and exits at 210°F , and the shellside fluid enters at 178°F and exits at 242°F ? There are 2 tubeside passes per exchanger (shell).



PROBLEM NO. 1A: DETERMINE THE WEIGHTED MID FOR AN EXCHANGE WITH THE INDICATED HEAT PROFILES.



PROBLEM NO. 5

Using the relationships shown on Exhibit 4 (Appendix A), estimate the new overall heat transfer coefficient for a Platforming Combined Feed Exchanger, if the flow rates through both sides are reduced to 50% of the original flow rates, assuming there is no change in the fluid properties. The original overall heat transfer coefficient was 50, the shell-side fouling factor is 0.001, the tubeside fouling factor is 0.002 and the resistance of the tube is 0.0004. (Assume 3/4 O.D. 14 BWG tubes)

$$\begin{array}{l} \text{GIVEN: } U = 50 \\ r_s^d = 0.001 \\ r_t^d = 0.002 \\ r_m = 0.0004 \\ \text{Assume } r_s^f = r_t^f \end{array}$$

PROBLEM NO. 6

Estimate the tubeside pressure drop for the new condition for Problem No. 5, if it was 3 psi for the original condition.

REVAMP

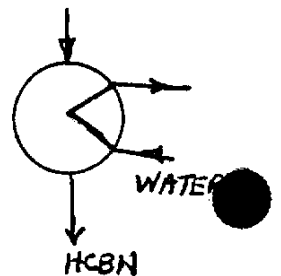
[SAMPLE PROBLEM]

EXAMPLE : PRODUCT COOLER

TUBE SIDE $\Delta P = 10 \text{ PSI}$

WHAT IS NEW DELTA-P IF FLOW
IS INCREASED 50% ON TUBE SIDE ?

$$\begin{aligned} \text{(I)} \quad \Delta P_2 &= \Delta P_1 \left(\frac{G_2}{G_1} \right)^{1.8} \\ &= 10 \left(\frac{1.5}{1.0} \right)^{1.8} \\ &= 10 (2.075) = 20.75 \text{ PSI} \end{aligned}$$



REVAMP

[SAMPLE PROBLEM]

II FOR SAME PRODUCT COOLER :

GIVEN: $\frac{3}{4}$ " O.D. 14 BWG TUBES $\frac{A_o}{A_i} = 1.284$
 $h_i = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ (FOR H_2O)

OVERALL $U = 90$

$$r_m = 0.0003$$

$$r_t^d = 0.0020$$

$$r_s^d = 0.0015$$

→ WHAT IS THE SHELL SIDE FILM RESISTANCE r_s^f ?

→ WHAT IS THE ESTD NEW OVERALL COEFF IF SHELL SIDE FLOW IS INCREASED 33% ?

$$U = 90 \left(R = \frac{1}{U} = 0.0111 \right)$$

$$r_m = 0.00030$$

$$r_t^d \left(\frac{A_o}{A_i} \right) = 0.00260$$

$$r_s^d = 0.00150$$

$$r_t^f \left(\frac{A_o}{A_i} \right) = 0.00128$$

$$R - r_s^f = 0.00568$$

$$\therefore r_s^f = R - 0.00568$$

$$r_s^f = \left\{ \frac{0.01110}{0.00568} \right\} = 0.00543$$

FROM TABLE

$$r_{s_2}^f = r_{s_1}^f \left(\frac{G_1}{G_2} \right)^{0.6}$$
$$= 0.00543 \left(\frac{1}{1.33} \right)^{0.6}$$
$$= 0.00458$$

$$R_2 = r_{s_2}^f + 0.00568$$
$$= 0.00458 + 0.00568$$
$$= 0.01026$$

$$U_2 = \frac{1}{R_2} = 97.5$$

NEW 'U' VALUE

AIR COOLED EXCHANGERSSAMPLE PROBLEM

Find approximate size and horsepower required for Platforming Reactor

Product Condenser:

Duty:	39.18×10^6 BTU/hr
Flow:	210,284 lbs/hr (HCBN + 74.8 mol % H ₂)
Process Temp.:	270°F to 113°F
Design Dry Bulb Temp.:	86°F
	(Say U 3.6 to 3.8 BTU/hr-ft ² -°F)

1st Try, 6 layer, surface

$$2 \times 75,800 \text{ ft}^2 = 151,600 \text{ ft}^2$$

$$\Delta T_{\text{air}} = 39.18 \times 3/2 = 58.8$$

$$(\text{Outlet air} = 86 + 58.8 = 144.8^\circ\text{F})$$

$$\begin{array}{r} \text{MTD} \quad 270 \rightarrow 113^\circ\text{F} \\ \quad \quad 144.8 \leftarrow 86 \\ \hline \quad \quad 125.2 \quad 27 \\ \quad \quad \quad \swarrow \quad \searrow \\ \quad \quad \quad 64 \end{array}$$

$$\text{So, } \text{MTD}_C = 64 \times 0.92 = 58.9^\circ\text{F}$$

Corr. Fac. to LMTD

$$R = \frac{58.8}{157} = 0.37$$

$$X = \frac{157}{184} = 0.85$$

From Chart for air cross flow
F = 0.92

$$U_{\text{req'd}} = \frac{Q}{A \Delta T_m} = \frac{39.18 \times 10^6}{(58.9)(151600)} = 4.39 \text{ Too High! (Needs more surface)}$$

2nd Try, 6 layer, 2 x 87700 = 175,400 ft.²

$$\Delta T_{\text{Air}} = 39.18 \times 2.7/2 = 52.9 \text{ (Air out} = 86 + 52.9 = 138.9^\circ\text{F)}$$

$$\begin{array}{r} \text{MTD} \quad 270 \rightarrow 113^\circ\text{F} \\ \quad \quad 138.9 \leftarrow 86 \\ \hline \quad \quad 131.1 \quad 27 \\ \quad \quad \quad \swarrow \quad \searrow \\ \quad \quad \quad 65.9 \end{array}$$

$$U_{\text{req'd}} = \frac{39.18 \times 10^6}{(65.9)(0.92)(175400)}$$

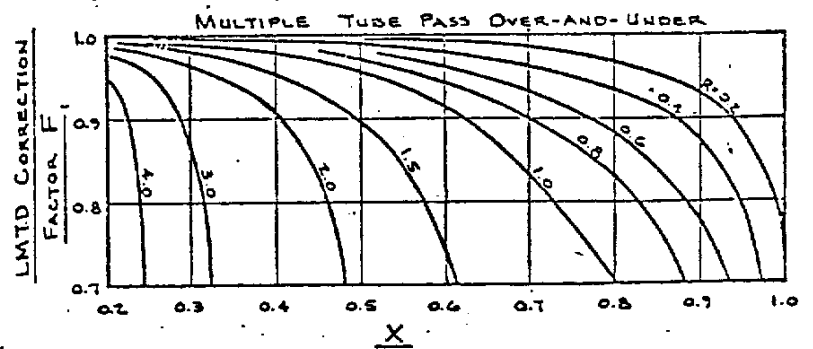
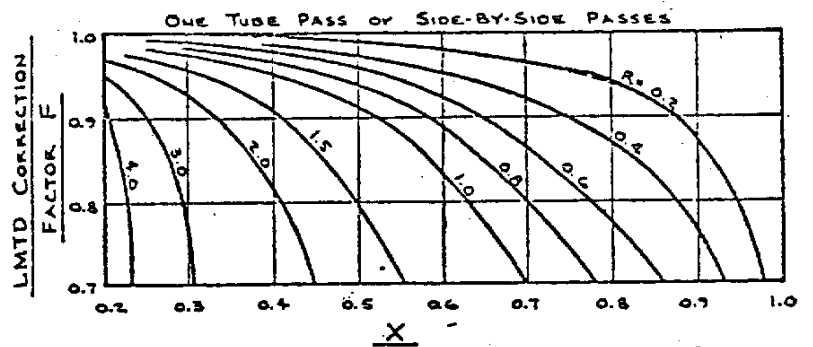
$$= 3.68 \text{ (ACCEPTABLE)}$$

∴ Use 175,400 ft² (finned), 6 layer, size ≥ 38' x 30' (Plot)
HP = 130.

Illustration 1—

OIL & GAS JOURNAL JAN 13, 1949

Correction factors for LMTD air crossflow



$$R = \frac{T_2 - T_1}{T_3 - T_4}$$

T = °F.

T₁ = AMBIENT AIR

T₂ = AVG. HOT AIR

T₃ = TUBESIDE INLET

T₄ = TUBESIDE OUTLET

$$X = \frac{T_3 - T_4}{T_3 - T_1}$$

CORRECTED LMTD = F × LMTD

Short Cut Estimating Method for PERFEX AIRFINS

APPENDIX 'B'

SAMPLE PROBLEM
AIR COOLED EXCHGR

The following estimating system has been developed to quickly approximate the size, horsepower and price of a Perfex Airfin exchanger.

Example

Cool 300,000 lbs/hr of 45° API hydrocarbon liquid from 350°F to 150°F, with ambient air at 80°F. Total heat load is 34,000,000 Btu/hr.

Step 1 — Select Heat Transfer Rate —

From Table 1 the transfer rate is selected within the range given for a light gas oil. Factors affecting the choice include allowable pressure drop, cooling range and viscosity. Pick rate of 3.5 Btu/(hr) (sq ft) (°F).

Step 2 — Figure Log Mean Temp Difference —

Assume an air outlet temperature based on a temperature rise between 20°F and 90°F. For this example, with a long cooling range, assume a 70°F air rise. Adding this to the 80°F ambient temperature gives an air outlet of 150°F.

Liquid Cooling	350	→	150
Air Heating	150	←	80
	200		70
	GTTD		LTTD

From Table 2 read LMTD as 124°F

Step 3 — Calculate Transfer Surface —

$$\text{Required Surface} = \frac{\text{Heat Load}}{\text{Rate} \times \text{LMTD}} = \text{sq ft} \left(A = \frac{Q}{U \Delta T_{lm}} \right)$$

$$\text{Required Surface} = \frac{34,000,000}{3.5 \times 124} = 78,500 \text{ sq ft}$$

Step 4 — Select Airfin Exchanger —

Refer to Table 1 and select 6-tube layers because of long cooling range.

Refer to Table 3, and select Airfin unit with six-tube layers having approximately 78,500 sq ft of transfer surface. The closest unit is 19 ft wide by 30 ft long and has 87,700 sq ft of surface.

Now check the air temperature rise to verify the estimated air outlet temperature of 150°F. The selection shows a 2.7°F rise per million Btu heat load. Therefore, the total temperature rise will be $2.7 \times 34,000,000$ Btu/hr heat load, which equals 92°F. The corrected outlet temperature is 172°F (80°F ambient + 92°F temp rise).

Refigure the LMTD:

Liquid Cooling	350	→	150
	172	←	80
	178		70
	GTTD		LTTD

From Table 2 read LMTD as 116°F.

Step 5 — Recheck Transfer Surface —

$$\text{Required Surface} = \frac{34,000,000}{3.5 \times 116} = 83,700 \text{ sq ft}$$

Referring to Table 3, closest surface available is 87,700 sq ft, in one bay having six layers. The plot area is 19 ft x 30 ft. Given horsepower is 65 and tube length is 30 ft.

$$\text{So, } U = \frac{3.5 \left(\frac{1172}{87700} \right)}{1.0} = 3.34$$

Price Estimating

Refer to Table 4 showing estimating prices based on dollars per square foot of extended surface:

$$87,700 \text{ sq ft} \times \$0.87/\text{sq ft} = \$77,000, \text{ fob shipping points}$$

$$U_{\text{Cor}} = \frac{3.34}{0.94} = 3.55$$

TABLE 1—Transfer Rates for Airfin Exchangers

COOLING SERVICE	*TRANSFER RATE	**NO. OF TUBE LAYERS	CONDENSING SERVICE	*TRANSFER RATE	**NO. OF TUBE LAYERS
Engine Jacket Water	6-7	4	Steam	7-8	4
Light Hydrocarbons	4-5	4 or 6	Light Hydrocarbon	4-5	4 or 6
Light Gas Oil	3-4	4 or 5	Reactor Effluent	3.5-4.5	6
Heavy Gas Oil	2.5-3	4 or 6	Still Overhead	2.75-3.5	4 or 6
Lube Oil	1-2	4 or 6			
Bottoms	.75-1.5	6 or more			
Flue Gas @ 100 psig & 50° PD	2-2.5	4			
Nat. Gas — 50#	1-2	4			
Nat. Gas — 100#	2-3	4			
Nat. Gas — 1000#	3.5-4.5	4			

*Transfer rate, Btu/(hr)(sq ft)(°F), based on outside fin tube surface.

**The number of tube layers cannot be accurately predicted for all services. In general, coolers having a cooling range up to

80°F and condensers having a condensing range up to 50°F are selected with 4 tube layers. Cooling and condensing ranges with ranges exceeding these values are generally figured with 6 tube layers.

TABLE 2—Chart for Solving
MTD Formula

$$MTD = \frac{GTDD-LTTD}{\text{Log} \frac{GTDD}{LTTD}}$$

NOTES: For points not included on this sheet multiply Greater Terminal Temperature Difference and Lesser Terminal Temperature Difference by any multiple of 10 and divide resulting value of curved lines by same multiple.

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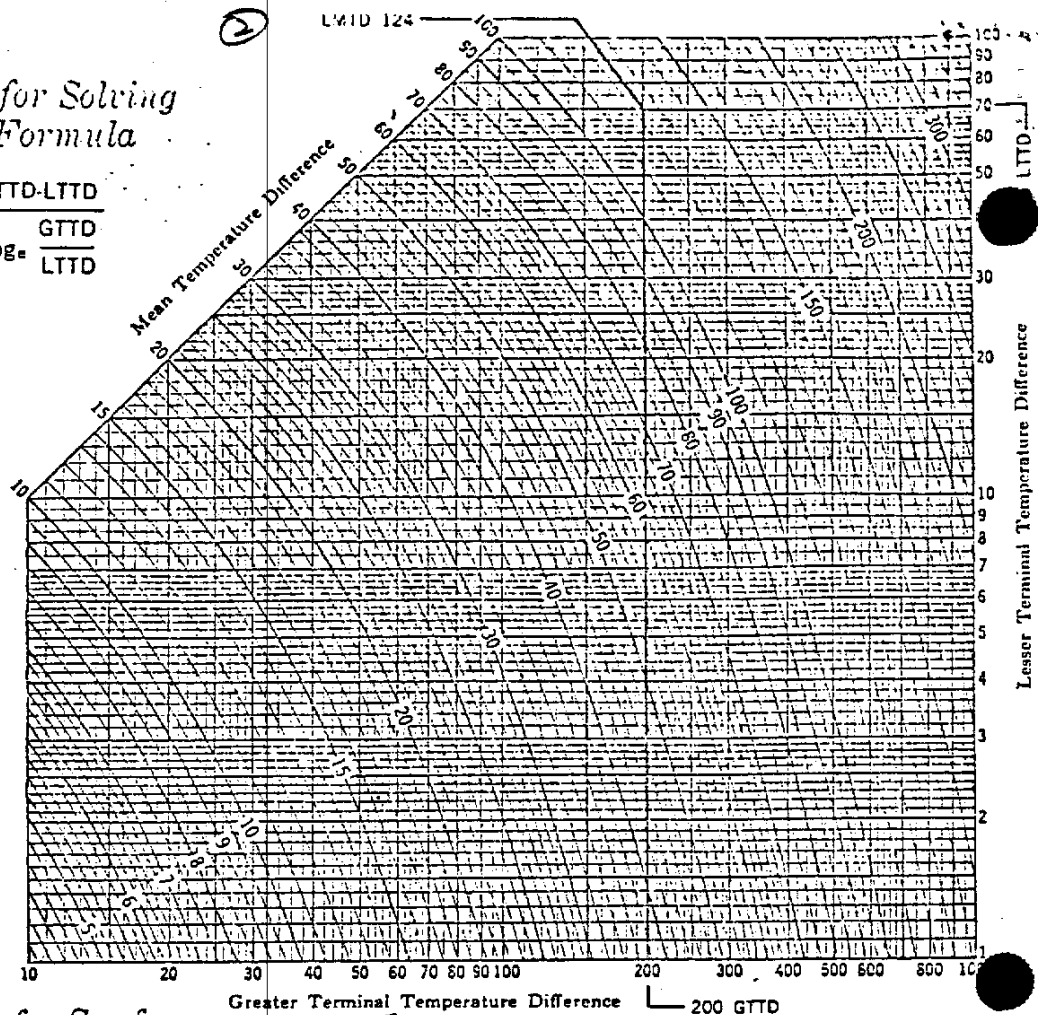


TABLE 3—Transfer Surface

APPROX. TOTAL SURFACE SQ. FT.	TUBE LENGTH	APPROX. DIM. EACH BAY WIDTH X LGTH.	*AIR TEMP. RISE °F. PER MILLION BTU	*APPROX. TOTAL FAN HP
BAYS WITH FOUR LAYER SECTIONS				
8,200	15'	5' x 15'	14.5	12
12,500	15'	8' x 15'	9.7	18
16,800	20'	8' x 20'	7.5	24
20,200	24'	8' x 24'	6.3	28
25,300	30'	8' x 30'	5.0	36
26,600	24'	11' x 24'	4.6	37
33,200	30'	11' x 30'	3.7	47
40,400	24'	16' x 24'	3.1	56
50,600	30'	16' x 30'	2.5	70
58,500	30'	19' x 30'	2.2	83
66,400	30'	21' x 30'	1.8	95
75,900	30'	24' x 30'	1.7	100
83,000	30'	27' x 30'	1.5	118
BAYS WITH SIX LAYER SECTIONS				
18,800	15'	8' x 15'	11.6	14
25,200	20'	8' x 20'	9.0	18
30,300	24'	8' x 24'	7.5	22
37,900	30'	8' x 30'	6.0	28
39,800	24'	11' x 24'	5.8	29
49,800	30'	11' x 30'	4.6	37
60,500	24'	16' x 24'	3.8	44
75,200	30'	16' x 30'	3.0	56
87,700	30'	19' x 30'	2.7	65
99,600	30'	21' x 30'	2.3	74
113,700	30'	24' x 30'	2.0	84
124,500	30'	27' x 30'	1.9	92

*Temp rise per bay. When 2 or more bays are required, divide temp rise by no. of bays and multiply hp by no. of bays.

TABLE 4—Airfin Pricing

NOM. TUBE LENGTH	**STEEL TUBES AND WELDED STEEL BOX HEADERS	**STEEL TUBES AND WELDED STEEL COVER PLATE HDR.
BAYS WITH FOUR LAYER SECTIONS		
15'	\$.43	\$.46
20'	.50	.43
24'	.37	.40
30'	.34	.37
BAYS WITH SIX LAYER SECTIONS		
15'	\$.35	\$.39
20'	.33	.36
24'	.30	.33
30'	.27	.30

**Prices based on steel headers, 300 lb. design, 1/4" corrosion allowance, 1" OD x 12 BWG welded steel tubes and Perfex Type SG grooved aluminum fins.

PRICES LISTED

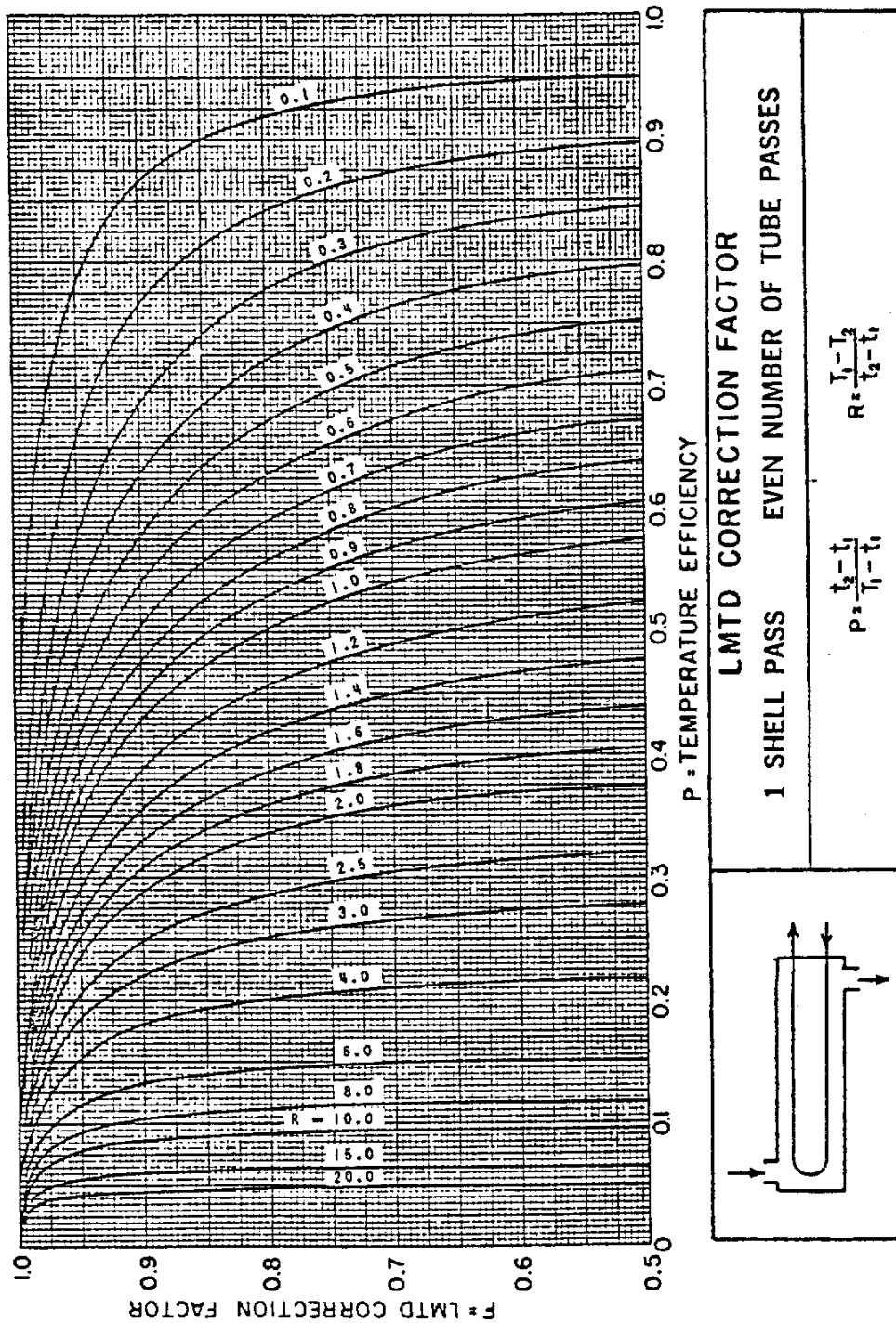
Prices listed are for Airfin Exchangers, f.o.b. shipping points, complete with fin sections, supporting structure, fan and gear drive assembly with TEFC motors.

APPENDIX 'C'

(EXCERPTS FROM TEMA)

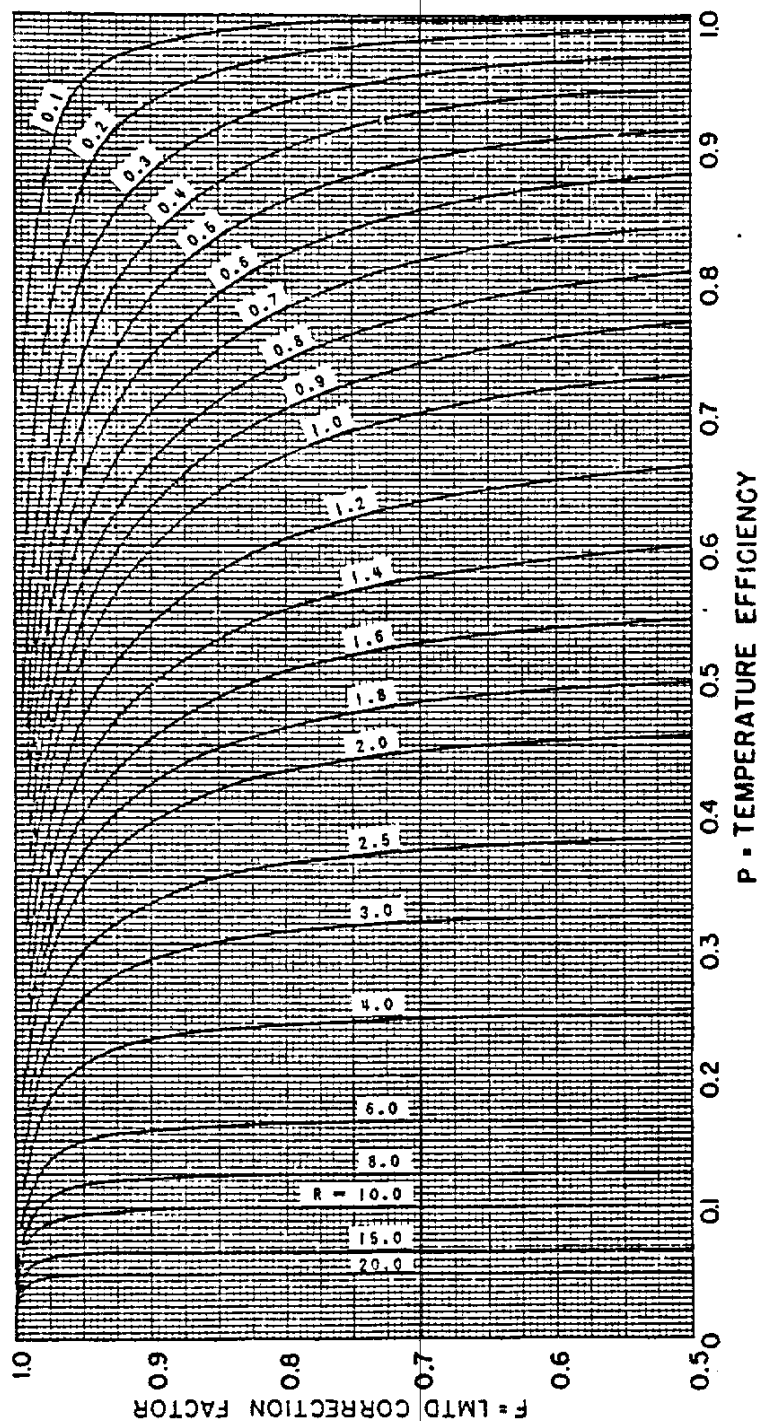
SECTION 9
Thermal Standards

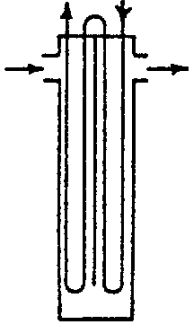
FIGURE T-3.2A



SECTION 9
Thermal Standards

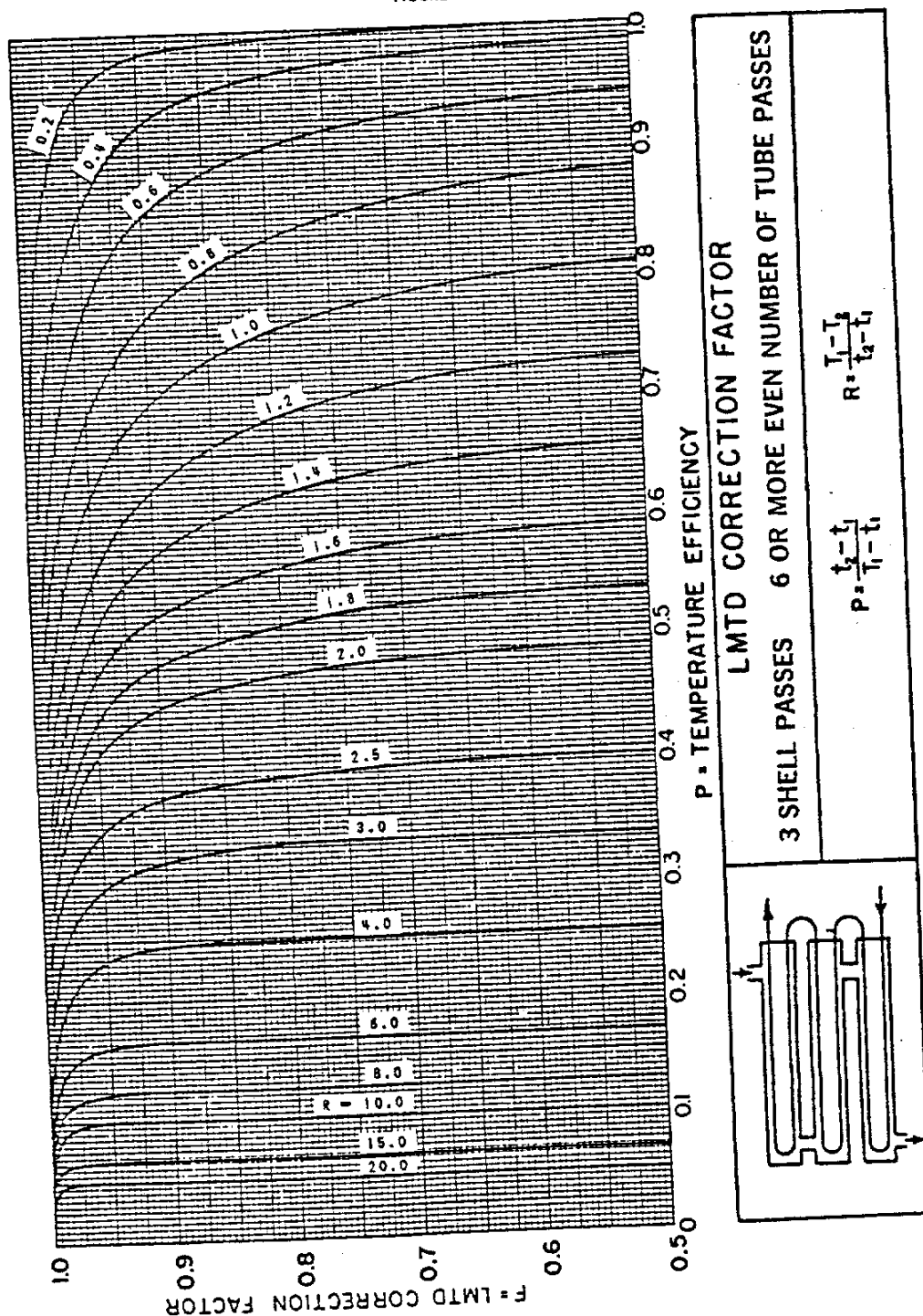
FIGURE T-3.28



LMTD CORRECTION FACTOR 2 SHELL PASSES 4 OR MULTIPLE OF 4 TUBE PASSES	$R = \frac{T_1 - T_2}{t_2 - t_1}$
	

SECTION 9
Thermal Standards

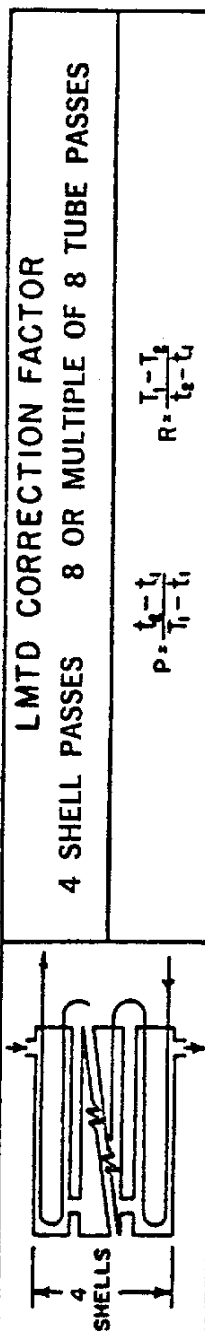
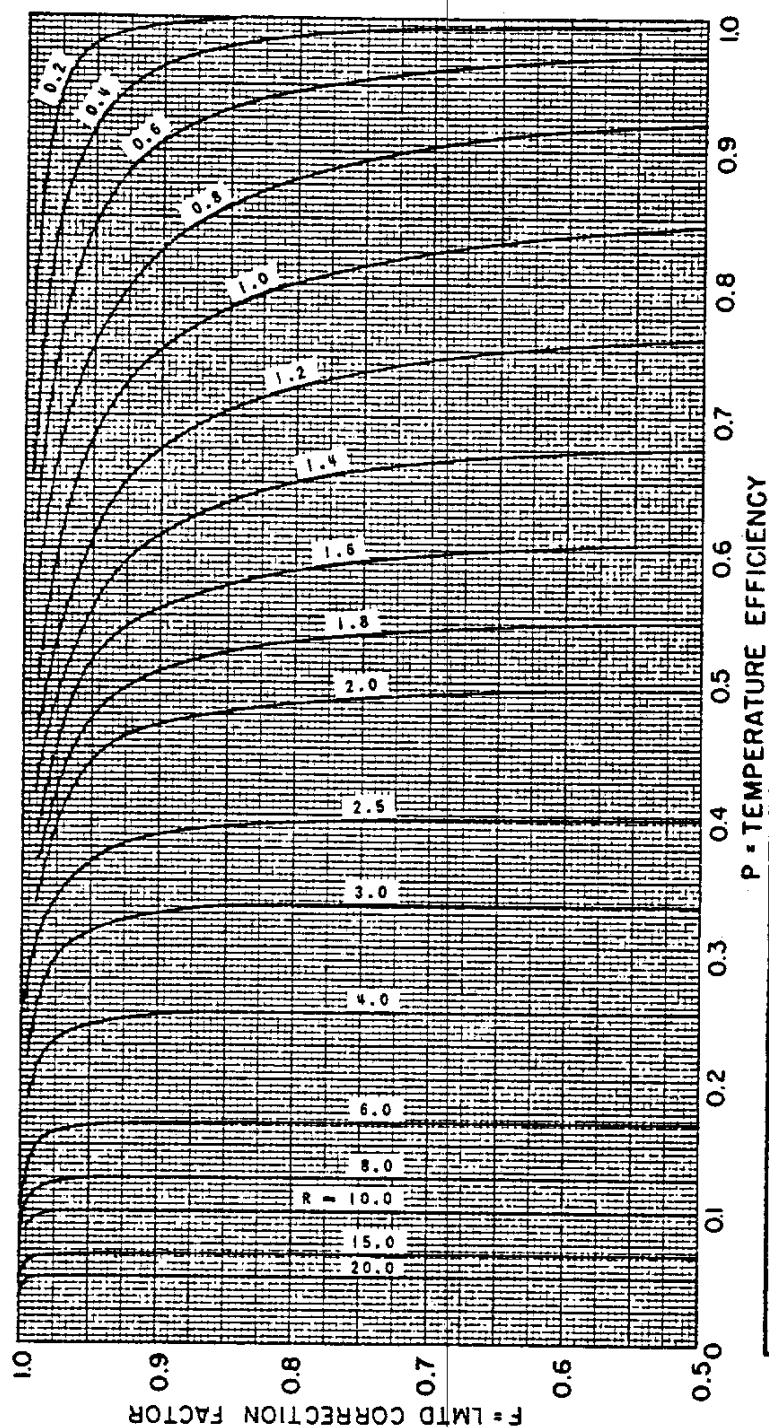
FIGURE T-3.2C



SECTION 9

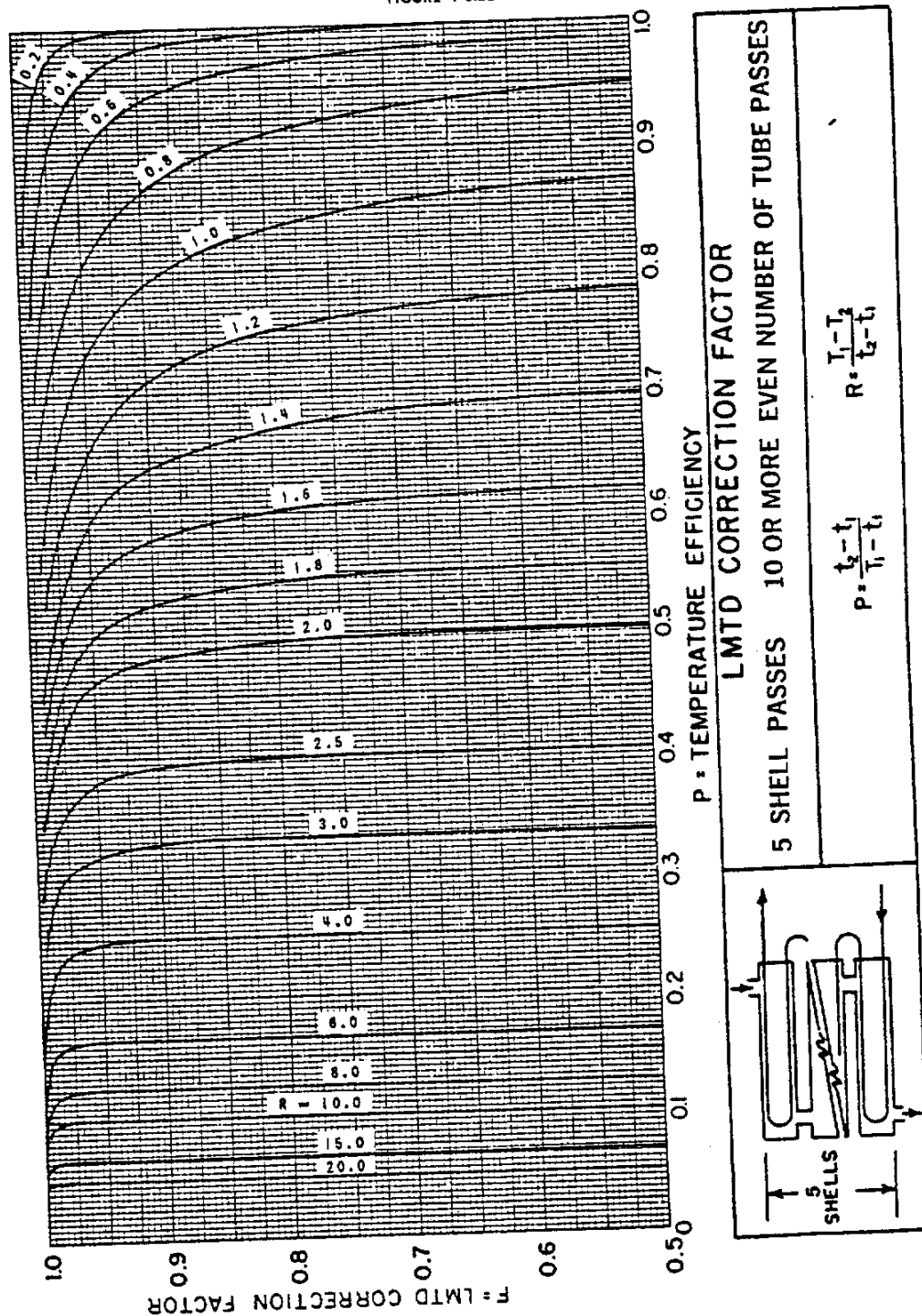
Thermal Standards

FIGURE T-3.2D



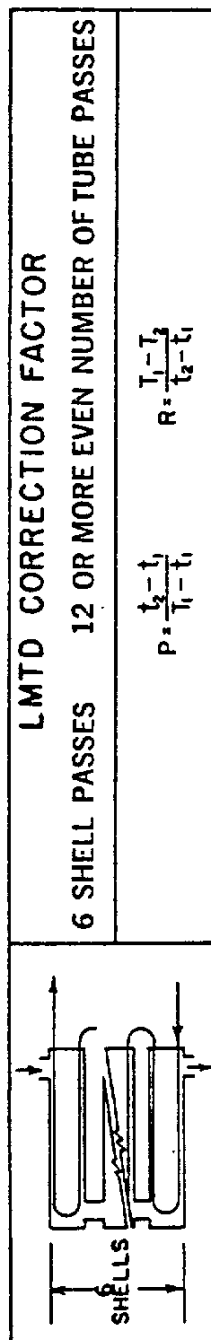
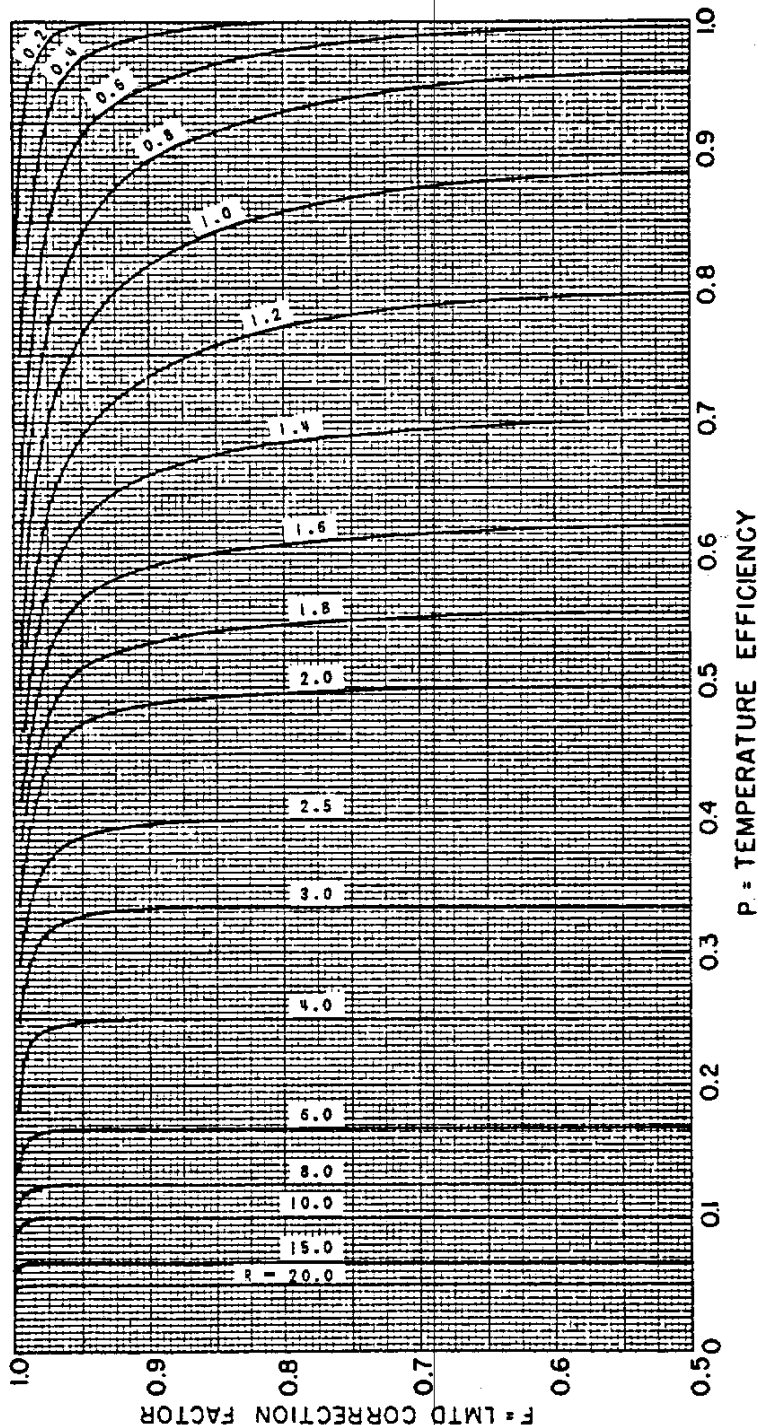
SECTION 9 Thermal Standards

FIGURE T-3.2E



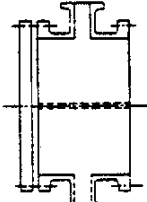
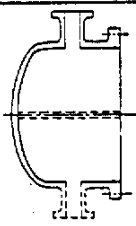
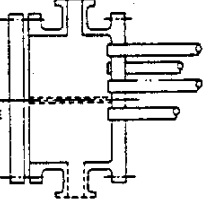
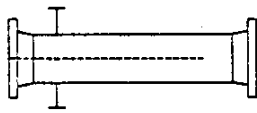
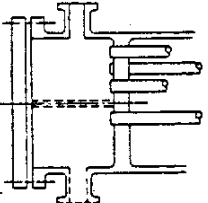
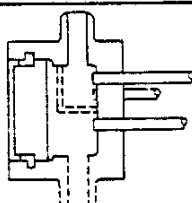
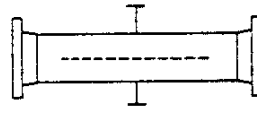
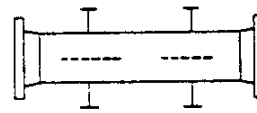
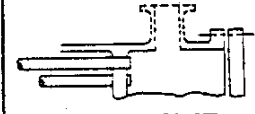
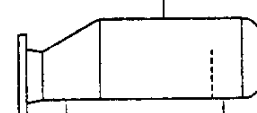
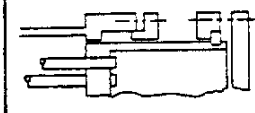
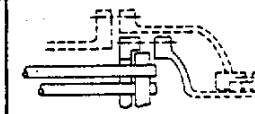
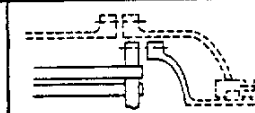
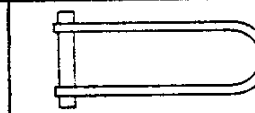
SECTION 9
Thermal Standards

FIGURE T-3.2F



Heat Exchanger Nomenclature

FIGURE N-1.2

FRONT END STATIONARY HEAD TYPES		SHELL TYPES		REAR END HEAD TYPES	
A		E		L	
	CHANNEL AND REMOVABLE COVER				
					
	BONNET (INTEGRAL COVER)				
					
	REMOVABLE TUBE BUNDLE ONLY	F		M	
B	CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER				
					
	CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER				
					
	SPECIAL HIGH PRESSURE CLOSURE				
C		G		N	
	DOUBLE SPLIT FLOW				
					
	DIVIDED FLOW				
					
D	KETTLE TYPE REBOILER	K		P	
	CROSS FLOW				
					
	FLOATING HEAD WITH BACKING DEVICE				
					
E	PULL THROUGH FLOATING HEAD				
					
	U-TUBE BUNDLE				
					
	EXTERNALLY SEALED FLOATING TUBESHEET				

SECTION 1

Heat Exchanger Nomenclature

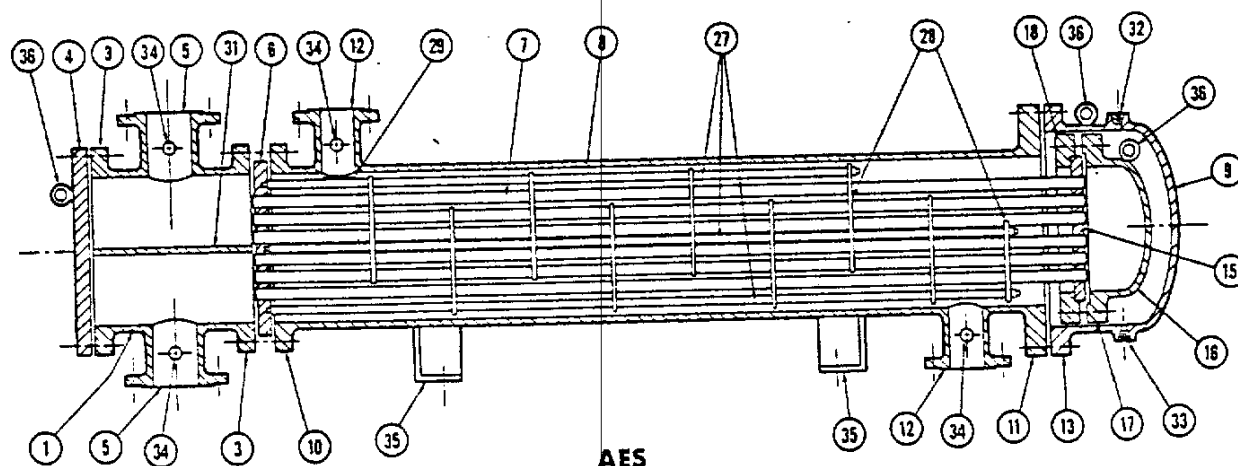
N-2 NOMENCLATURE OF HEAT EXCHANGER COMPONENTS

For the purpose of establishing standard terminology, Figure N-2 illustrates various types of heat exchangers. Typical parts and connections, for illustrative purposes only, are numbered for identification in Table N-2.

TABLE N-2

- | | |
|---|--|
| 1. Stationary Head—Channel | 20. Slip-on Backing Flange |
| 2. Stationary Head—Bonnet | 21. Floating Head Cover—External |
| 3. Stationary Head Flange—Channel or Bonnet | 22. Floating Tubesheet Skirt |
| 4. Channel Cover | 23. Packing Box |
| 5. Stationary Head Nozzle | 24. Packing |
| 6. Stationary Tubesheet | 25. Packing Gland |
| 7. Tubes | 26. Lantern Ring |
| 8. Shell | 27. Tierods and Spacers |
| 9. Shell Cover | 28. Transverse Baffles or Support Plates |
| 10. Shell Flange—Stationary Head End | 29. Impingement Plate |
| 11. Shell Flange—Rear Head End | 30. Longitudinal Baffle |
| 12. Shell Nozzle | 31. Pass Partition |
| 13. Shell Cover Flange | 32. Vent Connection |
| 14. Expansion Joint | 33. Drain Connection |
| 15. Floating Tubesheet | 34. Instrument Connection |
| 16. Floating Head Cover | 35. Support Saddle |
| 17. Floating Head Flange | 36. Lifting Lug |
| 18. Floating Head Backing Device | 37. Support Bracket |
| 19. Split Shear Ring | 38. Weir |
| | 39. Liquid Level Connection |

FIGURE N-2

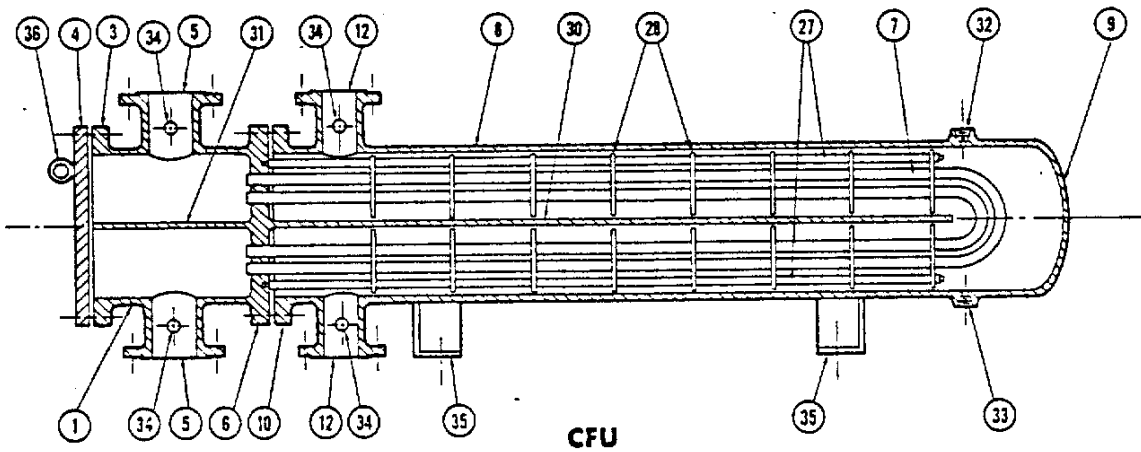
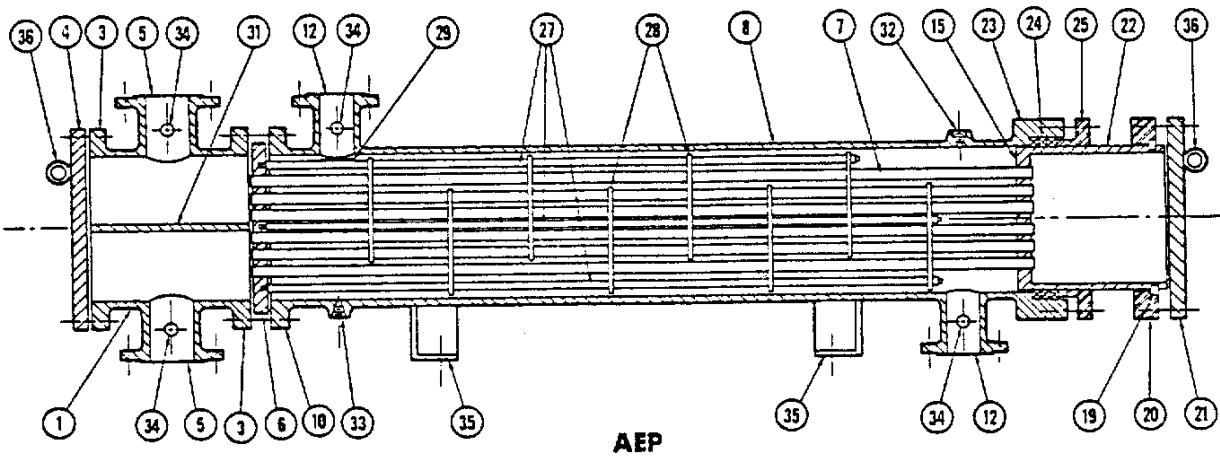
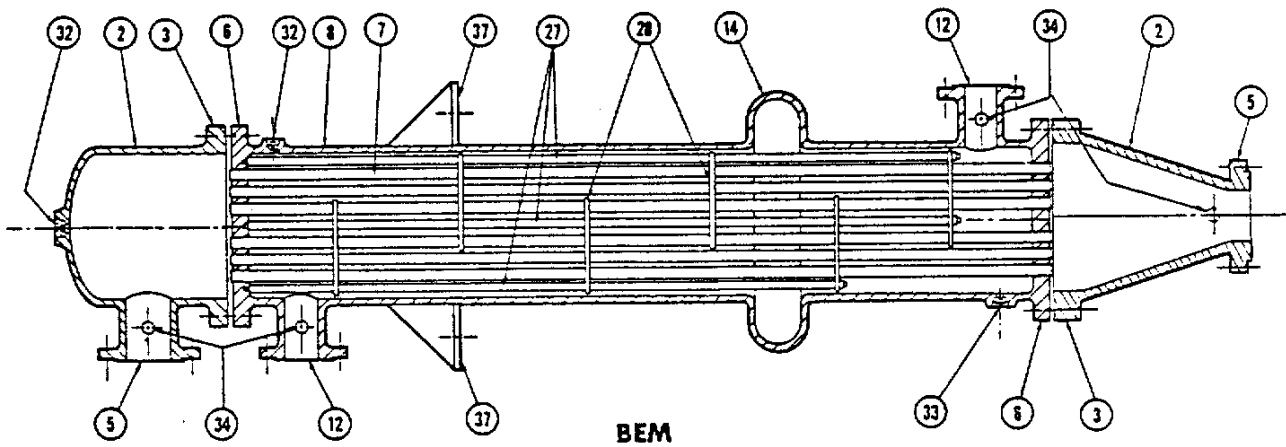


APP 'c' (9)

SECTION I

Heat Exchanger Nomenclature

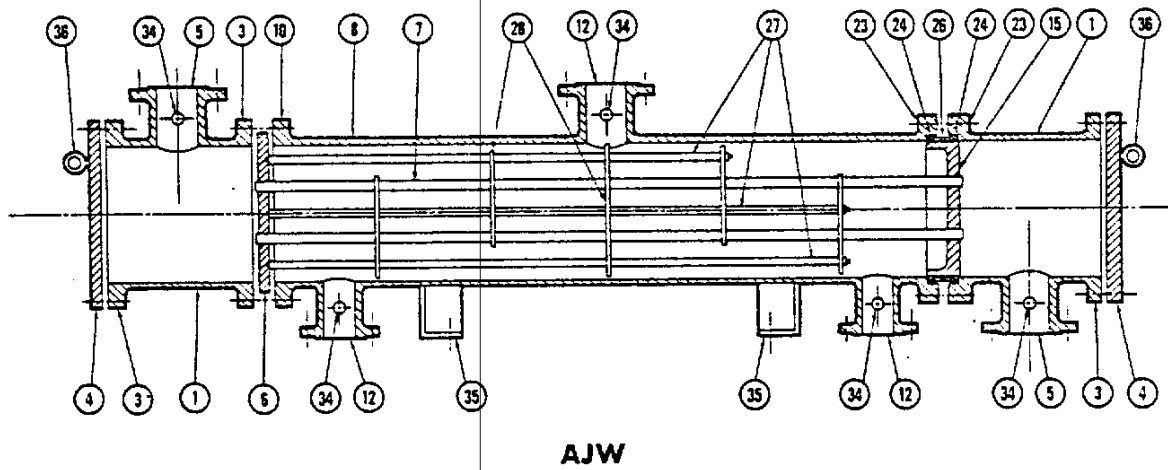
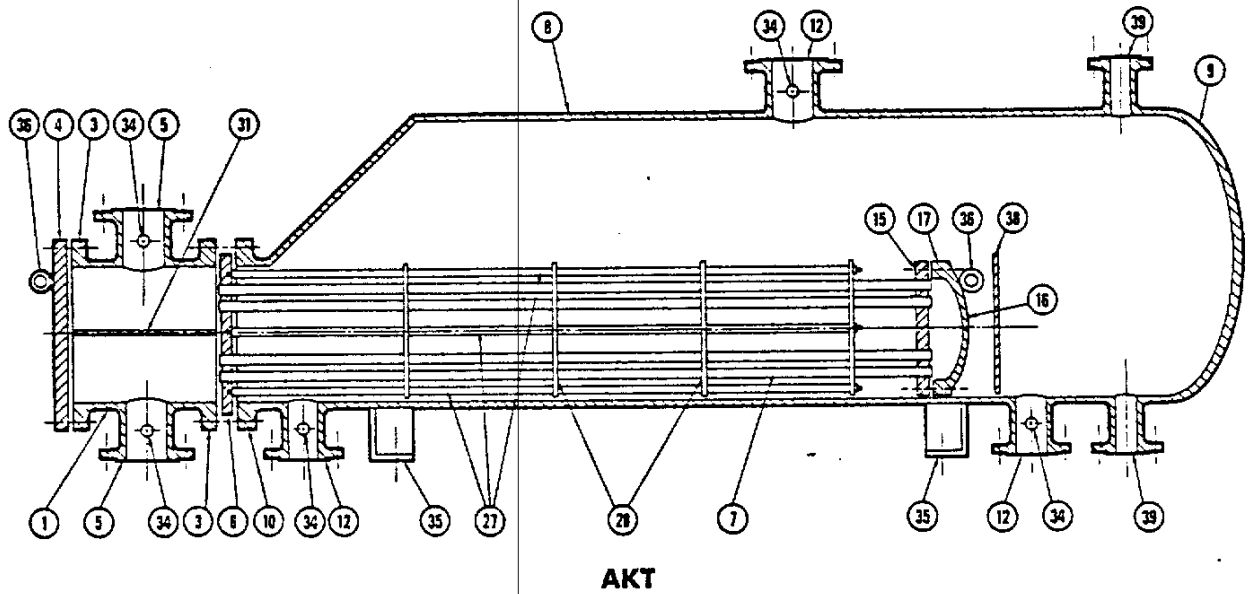
FIGURE N-2 (CONTINUED)



SECTION 1
Heat Exchanger Nomenclature

App'c' (10)

FIGURE N-2 (CONTINUED)



SECTION 9 Thermal Standards

App 'c' (11)

T-2 FOULING RESISTANCES—(Continued)

T-2.41 FOULING RESISTANCES FOR WATER

Temperature of Heating Medium	Up to 240° F.		240° - 400° F.*	
Temperature of Water	125° F. or Less		Over 125° F.	
Types of Water	Water Velocity Ft./Sec.		Water Velocity Ft./Sec.	
	3 and Less	Over 3	3 and Less	Over 3
Sea Water	.0005	.0005	.001	.001
Brackish Water	.002	.001	.003	.002
Cooling Tower and Artificial Spray Pond:				
Treated Makeup	.001	.001	.002	.002
Untreated	.003	.003	.005	.004
City or Well Water (Such as Great Lakes)	.001	.001	.002	.002
River Water:				
Minimum	.002	.001	.003	.002
Average	.003	.002	.004	.003
Muddy or Silty	.003	.002	.004	.003
Hard (Over 15 grains/gal.)	.003	.003	.005	.005
Engine Jacket	.001	.001	.001	.001
Distilled or Closed Cycle Condensate	.0005	.0005	.0005	.0005
Treated Boiler Feedwater	.001	.0005	.001	.001
Boiler Blowdown	.002	.002	.002	.002

*Ratings in columns 3 and 4 are based on a temperature of the heating medium of 240°-400° F. If the heating medium temperature is over 400° F. and the cooling medium is known to scale, these ratings should be modified accordingly.

T-2.42 FOULING RESISTANCES FOR INDUSTRIAL FLUIDS

(HR² FT²/BTU)

T-2.421 OILS

Fuel Oil	.005
Transformer Oil	.001
Engine Lube Oil	.001
Quench Oil	.004

T-2.422 GASES AND VAPORS

Manufactured Gas	.01
Engine Exhaust Gas	.01
Steam (non-oil bearing)	.0005
Exhaust Steam (oil bearing)	.001
Refrigerant Vapors (oil bearing)	.002
Compressed Air	.002
Industrial Organic Heat Transfer Media	.001

T-2.423 LIQUIDS

Refrigerant Liquids	.001
Hydraulic Fluid	.001
Industrial Organic Heat Transfer Media	.001
Molten Heat Transfer Salts	.0005

T-2.43 FOULING RESISTANCES FOR CHEMICAL PROCESSING STREAMS

T-2.431 GASES AND VAPORS

Acid Gas	.001
Solvent Vapors	.001
Stable Overhead Products	.001

NOTE: (ENGLISH) $\text{HR}^2 \text{FT}^2 / \text{BTU}$ $\times 0.2 =$ (METRIC) $\text{HR}^2 \text{CM}^2 / \text{KCAL}$
 EXAMPLE: $0.001 \times 0.2 = 0.0002$

SECTION 9
Thermal Standards

T-2 FOULING RESISTANCES—(Continued)

T-2.432 LIQUIDS

MEA & DEA Solutions	.002
DEG & TEG Solutions	.002
Stable Side Draw and Bottom Product	.001
Caustic Solutions	.002
Vegetable Oils	.003

T-2.44 FOULING RESISTANCES FOR NATURAL GAS-GASOLINE PROCESSING STREAMS

T-2.441 GASES AND VAPORS

Natural Gas	.001
Overhead Products	.001

T-2.442 LIQUIDS

Lean Oil	.002
Rich Oil	.001
Natural Gasoline & Liquefied Petroleum Gases	.001

T-2.45 FOULING RESISTANCES FOR OIL REFINERY STREAMS

T-2.451 CRUDE & VACUUM UNIT GASES AND VAPORS

Atmospheric Tower Overhead Vapors	.001
Light Naphthas	.001
Vacuum Overhead Vapors	.002

T-2.452 CRUDE & VACUUM LIQUIDS

Crude Oil

	0-199°F.			200°-299°F.		
	Velocity Ft./Sec.			Velocity Ft./Sec.		
	Under 2	2-4	4 and Over	Under 2	2-4	4 and Over
Dry	.003	.002	.002	.003	.002	.002
Salt†	.003	.002	.002	.005	.004	.004

	300°-499°F.			500°F. and Over		
	Velocity Ft./Sec.			Velocity Ft./Sec.		
	Under 2	2-4	4 and Over	Under 2	2-4	4 and Over
Dry	.004	.003	.002	.005	.004	.003
Salt†	.006	.005	.004	.007	.006	.005

† Normally desalted below this temperature range. († to apply to 200-299°F., 300-499°F., 500°F. and over.)

Gasoline	.001
Naphtha & Light Distillates	.001
Kerosene	.001
Light Gas Oil	.002
Heavy Gas Oil	.003
Heavy Fuel Oils	.005
Asphalt & Residuum	.010

(HR² FT²/BTU)

APP 'C' (13)

SECTION 9 Thermal Standards

T-2 FOULING RESISTANCES—(Continued)

HR FT²/BTU

T-2.453 CRACKING & COKING UNIT STREAMS

Overhead Vapors	.002
Light Cycle Oil	.002
Heavy Cycle Oil	.003
Light Coker Gas Oil	.003
Heavy Coker Gas Oil	.004
Bottoms Slurry Oil (4½ ft./sec. minimum)	.003
Light Liquid Products	.002

T-2.454 CATALYTIC REFORMING, HYDROCRACKING, & HYDRODESULFURIZATION STREAMS

Reformer Charge	.002
Reformer Effluent	.001
Hydrocracker Charge & Effluent**	.002
Recycle Gas	.001
Hydrodesulfurization Charge & Effluent**	.002
Overhead Vapors	.001
Liquid Product over 50° A.P.I.	.001
Liquid Product 30°-50° A.P.I.	.002

**Depending on charge characteristics and storage history, charge resistance may be many times this value.

T-2.455 LIGHT ENDS PROCESSING STREAMS

Overhead Vapors & Gases	.001
Liquid Products	.001
Absorption Oils	.002
Alkylation Trace Acid Streams	.002
Reboiler Streams	.003

T-2.456 LUBE OIL PROCESSING STREAMS

Feed Stock	.002
Solvent Feed Mix	.002
Solvent	.001
Extract†	.003
Raffinate	.001
Asphalt	.005
Wax Slurriest	.003
Refined Lube Oil	.001

†Precautions must be taken to prevent wax deposition on cold tube walls.

SECTION 1
General InformationTABLE D-7
CHARACTERISTICS OF TUBING

Tube O.D. Inches	B.W.G. Gage	Thickness Inches	Internal Area Sq. Inch	Sq. Ft. External Surface Per Foot Length	Sq. Ft. Internal Surface Per Foot Length	Weight Per Foot Length Steel Lbs.*	Tube I.D. Inches	Moment of Inertia Inches ⁴	Section Modulus Inches ³	Radius of Gyration Inches	Constant C**	O.D. I.D.	Transverse Metal Area Sq. Inch
1/4	22	.028	.0295	.0655	.0508	.066	.194	.00012	.00098	.0792	46	1.289	.0195
1/4	24	.022	.0333	.0655	.0539	.054	.206	.00011	.00083	.0810	52	1.214	.0159
1/4	26	.018	.0360	.0655	.0560	.045	.214	.00009	.00071	.0824	56	1.168	.0131
1/4	27	.016	.0373	.0655	.0570	.040	.218	.00008	.00064	.0829	58	1.146	.0117
1/2	18	.049	.0603	.0982	.0775	.171	.277	.00068	.0036	.1164	94	1.354	.0502
1/2	20	.035	.0731	.0982	.0798	.127	.305	.00055	.0029	.1213	114	1.233	.0374
1/2	22	.028	.0799	.0982	.0835	.104	.319	.00046	.0025	.1227	125	1.176	.0305
1/2	24	.022	.0860	.0982	.0867	.083	.331	.00038	.0020	.1248	134	1.133	.0244
3/4	16	.065	.1075	.1309	.0969	.302	.370	.0022	.0085	.1556	168	1.351	.0888
3/4	18	.049	.1269	.1309	.1052	.236	.402	.0018	.0072	.1606	198	1.244	.0694
3/4	20	.035	.1452	.1309	.1126	.174	.430	.0014	.0056	.1649	227	1.163	.0511
3/4	22	.028	.1548	.1309	.1162	.141	.444	.0012	.0046	.1671	241	1.126	.0415
1	12	.109	.1301	.1636	.1066	.602	.407	.0061	.0197	.1864	203	1.536	.177
1	13	.095	.1486	.1636	.1139	.537	.435	.0057	.0183	.1903	232	1.437	.158
1	14	.083	.1655	.1636	.1202	.479	.459	.0053	.0170	.1938	258	1.362	.141
1	15	.072	.1817	.1636	.1259	.425	.481	.0049	.0156	.1971	283	1.299	.125
1	16	.065	.1924	.1636	.1296	.388	.495	.0045	.0145	.1993	300	1.263	.114
1	17	.058	.2035	.1636	.1333	.350	.509	.0042	.0134	.2016	317	1.228	.103
1	18	.049	.2181	.1636	.1380	.303	.527	.0037	.0118	.2043	340	1.186	.089
1	19	.042	.2298	.1636	.1416	.262	.541	.0033	.0105	.2068	358	1.155	.077
1	20	.035	.2419	.1636	.1453	.221	.555	.0028	.0091	.2089	377	1.126	.065
1 1/4	10	.134	.1825	.1963	.1262	.884	.482	.0129	.0344	.2229	285	1.556	.260
1 1/4	11	.120	.2043	.1963	.1338	.809	.510	.0122	.0326	.2257	319	1.471	.238
1 1/4	12	.109	.2223	.1963	.1393	.748	.532	.0116	.0309	.2299	347	1.410	.220
1 1/4	13	.095	.2463	.1963	.1466	.666	.560	.0107	.0285	.2340	384	1.339	.196
1 1/4	14	.083	.2679	.1963	.1529	.592	.584	.0098	.0262	.2376	418	1.284	.174
1 1/4	15	.072	.2884	.1963	.1587	.520	.606	.0089	.0238	.2410	450	1.238	.153
1 1/4	16	.065	.3019	.1963	.1623	.476	.620	.0083	.0221	.2433	471	1.210	.140
1 1/4	17	.058	.3157	.1963	.1660	.428	.634	.0076	.0203	.2455	492	1.183	.126
1 1/4	18	.049	.3339	.1963	.1707	.367	.652	.0067	.0178	.2484	521	1.150	.108
1 1/4	20	.035	.3632	.1963	.1780	.269	.680	.0050	.0134	.2532	567	1.103	.079
1 1/2	8	.165	.3526	.2618	.1754	1.462	.670	.0392	.0784	.3009	350	1.493	.430
1 1/2	10	.134	.4208	.2618	.1916	1.237	.732	.0350	.0700	.3098	656	1.366	.366
1 1/2	11	.120	.4536	.2618	.1990	1.129	.760	.0327	.0654	.3140	708	1.316	.316
1 1/2	12	.109	.4803	.2618	.2047	1.037	.782	.0307	.0615	.3174	749	1.279	.279
1 1/2	13	.095	.5153	.2618	.2121	.918	.810	.0280	.0559	.3217	804	1.235	.235
1 1/2	14	.083	.5463	.2618	.2183	.813	.834	.0253	.0507	.3255	852	1.199	.210
1 1/2	15	.072	.5755	.2618	.2241	.714	.856	.0227	.0455	.3291	898	1.167	.181
1 1/2	16	.065	.5945	.2618	.2278	.649	.870	.0210	.0419	.3314	927	1.149	.164
1 1/2	18	.049	.6390	.2618	.2361	.496	.902	.0166	.0332	.3366	997	1.109	.146
1 1/2	20	.035	.6793	.2618	.2435	.350	.930	.0124	.0247	.3414	1060	1.075	.106
1 3/4	7	.180	.6221	.3272	.2330	2.057	.890	.0890	.1425	.3836	970	1.404	.605
1 3/4	8	.165	.6648	.3272	.2409	1.921	.920	.0847	.1355	.3880	1037	1.359	.565
1 3/4	10	.134	.7574	.3272	.2571	1.598	.982	.0741	.1186	.3974	1182	1.273	.470
1 3/4	11	.120	.8012	.3272	.2644	1.448	1.010	.0688	.1100	.4018	1250	1.238	.426
1 3/4	12	.109	.8365	.3272	.2702	1.329	1.032	.0642	.1027	.4052	1305	1.211	.391
1 3/4	13	.095	.8825	.3272	.2775	1.173	1.060	.0579	.0926	.4097	1377	1.179	.345
1 3/4	14	.083	.9279	.3272	.2838	1.033	1.084	.0521	.0833	.4136	1440	1.153	.304
1 3/4	16	.065	.9852	.3272	.2932	.823	1.120	.0426	.0682	.4196	1537	1.116	.242
1 3/4	18	.049	1.042	.3272	.3016	.629	1.152	.0334	.0534	.4250	1626	1.085	.185
1 3/4	20	.035	1.094	.3272	.3089	.456	1.180	.0247	.0395	.4297	1707	1.059	.134
2	10	.134	1.192	.3927	.3225	1.955	1.232	.1354	.1806	.4853	1860	1.218	.575
2	12	.109	1.291	.3927	.3356	1.618	1.252	.1159	.1546	.4933	2014	1.170	.476
2	14	.083	1.398	.3927	.3492	1.258	1.334	.0931	.1241	.5018	2181	1.124	.370
2	16	.065	1.474	.3927	.3587	.996	1.370	.0756	.1008	.5079	2299	1.095	.293
2 1/2	11	.120	2.433	.5236	.4608	2.410	1.760	.3144	.3144	.6660	3795	1.136	.709
2 1/2	12	.109	2.494	.5236	.4655	2.201	1.782	.2904	.2904	.6697	3890	1.122	.647
2 1/2	13	.095	2.573	.5236	.4739	1.934	1.810	.2586	.2586	.6744	4014	1.105	.569
2 1/2	14	.083	2.642	.5236	.4801	1.699	1.834	.2300	.2300	.6784	4121	1.090	.500

* Weights are based on low carbon steel with a density of 0.2833 lbs./cu. in. For other metals multiply by the following factors:

Aluminum.....	0.35	Nickel-Chrome-Iron.....	1.07
Titanium.....	0.58	Admiralty.....	1.09
A.I.S.I. 400 Series Stainless Steels.....	0.99	Nickel and Nickel-Copper.....	1.13
A.I.S.I. 300 Series Stainless Steels.....	1.02	Copper and Cupro-Nickels.....	1.14
Aluminum Bronze.....	1.04		
Aluminum Brass.....	1.06		

** Liquid Velocity = $\frac{\text{Lbs. Per Tube Per Hour}}{C \times \text{Sp. Gr. of Liquid}}$ in feet per sec. (Sp. Gr. of Water at 60° F. = 1.0)

BWG
(BIRMINGHAM WIRE GAGE)

— USED TO SPECIFY
TUBE WALL THICKNESS

FOR EXAMPLE:

$$\frac{3}{4}" \text{ O.D. } \times \underline{14 \text{ BWG}} \Rightarrow \overset{\text{TUBE WALL THICKNESS}}{(2.1^{\text{mm}}) 0.083"}$$

$$1" \text{ O.D. } \times \underline{12 \text{ BWG}} \Rightarrow (2.77^{\text{mm}}) 0.109"$$

LOWER "GAGE NO." MEANS
THICKER TUBE WALL

APPENDIX 'D'

(SUPPLEMENT)

I BASIC HEAT TRANSFER THEORY.

$$Q = UA \Delta T_m$$

Q = HT. TRANSFERRED (Btu/hr)

U = OVERALL HT. TRANSFER
COEFF. (Btu/hr °F ft²)

A = SURFACE
(OUTSIDE) (ft²)

ΔT_m = CORR'D LOG. MEAN TEMP.
DIFF. (°F)

(a) FOR SENSIBLE HEAT:

$$Q = W C_p \Delta T$$

W = FLUID FLOW RATE (lb/hr)

C_p = FLUID SPECIFIC HEAT
(Btu/lb.°F)

ΔT = FLUID TEMP. CHANGE
(°F)

FOR LATENT HEAT:

(b) $Q = W \lambda$

W = FLUID FLOW RATE (lb/hr)

λ = LATENT HEAT (Btu/lb.)

Q , HEAT TRANSFERRED (DUTY) CAN BE

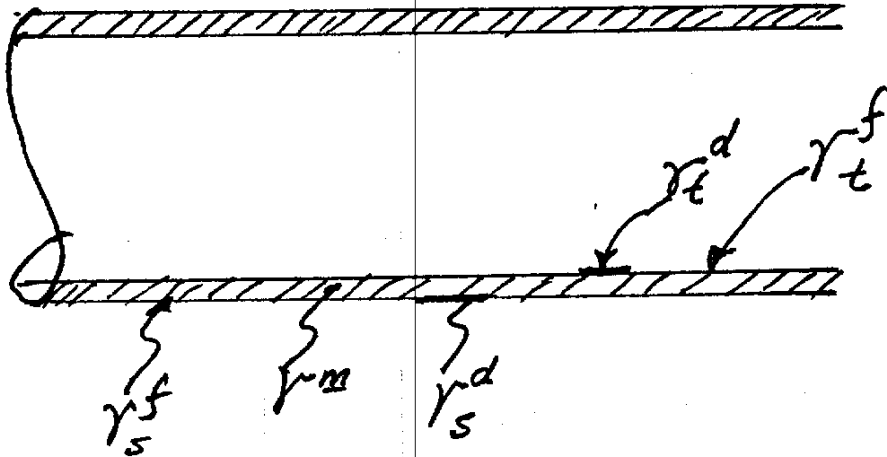
(a), (b) OR A MIXTURE OF BOTH.

OVERALL HT. TRANSFER COEFF.

$$U = \frac{1}{R} \quad (\text{Btu/hr. ft}^2)$$

R = SUM OF 5 RESISTANCES.

$$R = r_{\text{shell}}^{\text{film}} + r_{\text{shell}}^{\text{dirt}} + r_{\text{metal}} + r_{\text{tube}}^{\text{dirt}} \left(\frac{A_o}{A_i} \right) + r_{\text{tube}}^{\text{film}} \left(\frac{A_o}{A_i} \right)$$



$$\frac{A_o}{A_i} = \text{RATIO OF OUTSIDE TO INSIDE SURFACE}$$

$$= \frac{\pi D_o}{\pi D_i} = \frac{D_o}{D_i}$$

METAL RESISTANCE

FOR BARE TUBE;

$$(1) \quad V_m = \frac{t_w D_o}{12(K_m)(D_o - t_w)}$$

D_o = TUBE O.D.
 t_w = WALL THICKNESS
 K_m = METAL
 THER.
 CONDUCTIVITY

FOR EX:

$\frac{3}{4}$ " O.D. CARBON STEEL TUBE 14 BWG (0.083" TK)
 $K_m \approx 30 \text{ BTU/HR FT}^2/\text{°F(FT)}$

$$\begin{aligned} V_m &= \frac{0.083(0.75)}{12(30)(0.75 - 0.083)} \\ &= 0.000259 \text{ (HR } \text{°F FT}^2/\text{BTU)} \end{aligned}$$

FOULING RESISTANCES.

$\gamma_{\text{dirt SHELL}}$ or $\gamma_{\text{dirt TUBE}}$

- OPER. EXPERIENCE
- EST'D FROM TABLES
SUCH AS IN "TEMA"

FLUID FILM RESISTANCES $r_{\text{SHELL}}^f, r_{\text{TUBE}}^f$
ARE DETERMINED FROM:

CORRELATIONS INVOLVING

- TUBE DIAM.
- FLUID FLOW
- VISCOSITY
- THER. CONDUCTIVITY
- SPECIFIC H.T.

FOR EXAMPLE: TUBE SIDE TRANSFER,
COEFF. CAN BE CALCD
FROM:

DITTUS-BOELTER
EQ: $\left. \begin{array}{l} \frac{h_b D}{K_b} \\ N_{Nu} \end{array} \right\} = 0.023 \left(\frac{DG}{\mu_b} \right)^{0.8} \left(\frac{c_p \mu_b}{K_b} \right)^{0.4}$

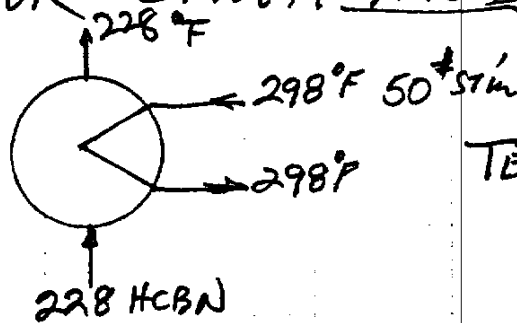
$\swarrow$ N_{Re} $\searrow$ N_{Pr}

LOG MEAN TEMP. DIFFERENCE

$$Q = UA \Delta T_m$$

$$\Delta T_m = \frac{GTTD - LTTD}{\ln \frac{GTTD}{LTTD}}$$

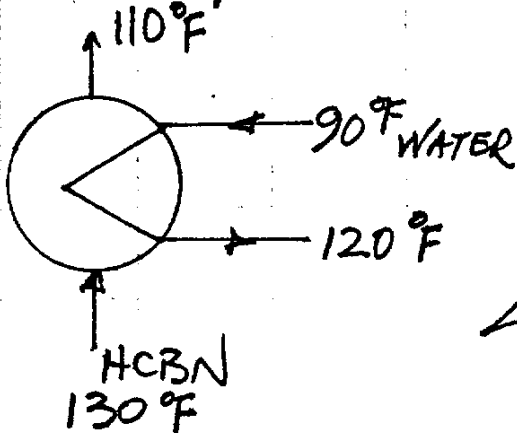
FOR STIM HEATED REBOILER



TEMP. DIFF $\left\{ \begin{array}{l} 298^\circ\text{F} \\ - 228 \\ \hline 70^\circ\text{F} \end{array} \right.$

IS LIMITED

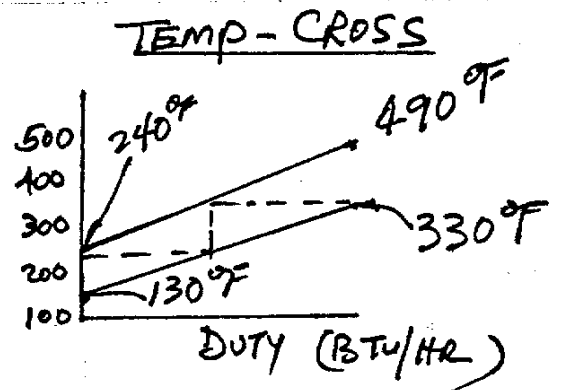
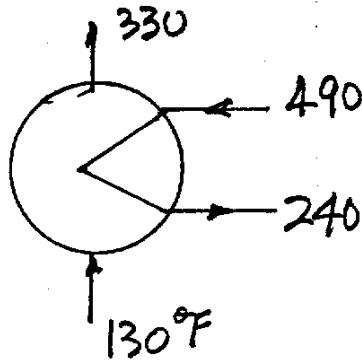
NORMALLY:



$$\begin{array}{rcl} 130 & \rightarrow & 110^\circ\text{F} \leftarrow \text{HCBN} \\ 120 & \leftarrow & 90 \leftarrow \text{WATER} \\ \hline 10 & & 20 \end{array}$$

$$\Delta T_m = \frac{20 - 10}{\ln \frac{20}{10}} = \underline{14.4^\circ\text{F}}$$

HOW TO CALCULATE CORRECTED LMTD



$$\begin{array}{rcl}
 490 \rightarrow 240 & = & \Delta T \\
 330 \leftarrow 130 & = & 200 \\
 \hline
 160 & & 110 \quad 360
 \end{array}
 \quad
 \begin{array}{l}
 \frac{250}{200} = 1.25 = R \\
 \frac{200}{360} = 0.555 = P
 \end{array}$$

$$\begin{aligned}
 \Delta T_m &= \frac{160 - 110}{\ln \frac{160}{110}} \\
 &= \frac{50}{0.375} \\
 &= 133.4
 \end{aligned}$$

FROM TEMA CHARTS
(2 SHELL PASSES)
 $F = 0.87$

$$\begin{aligned}
 \text{OR } R' &= \frac{1}{R} = \frac{200}{250} = 0.8 \\
 P' &= PR = \frac{250}{360} = 0.694 \\
 F' &= 0.87 = F
 \end{aligned}$$

$$\Delta T_c = \Delta T_m \times F = 133.4 \times 0.87 = \underline{\underline{116^\circ \text{F}}}$$

EXCHANGERS

NOT TRUE COUNTERFLOW

- ① WHEN TUBE PASSES NOT EQUAL
TO SHELL PASSES

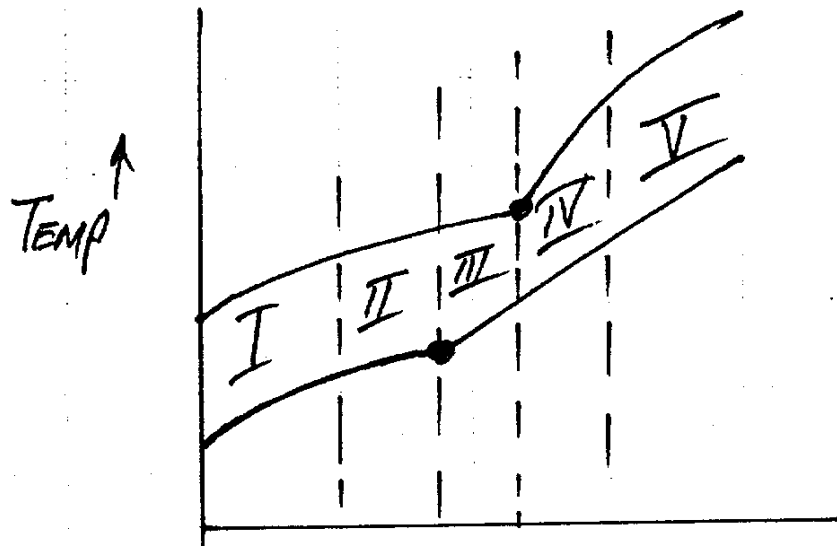
FOR EX: (1 SHELL PASS WITH 2 OR MORE
TUBE PASSES)

- ② THEN, USE LMTD CORRECTION
FACTOR OF 0.8 OR HIGHER

HOW TO CALCULATE WTD, MTD

- REQUIRED WHEN PROFILE GRAPH IS NON-LINEAR
- DIVIDE GRAPH INTO CONVENIENT ZONES (SECTIONS)
- BE SURE TO GO THROUGH DEFLECTION PTS.

FOR EXAMPLE: DIVIDE AS FOLL:



SEE
EXAMPLE (EXHIBIT 2B)
CALC:

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PROJECT SPECIFICATION

NUMBER

REVISION

— 402 —

SHEET

OF

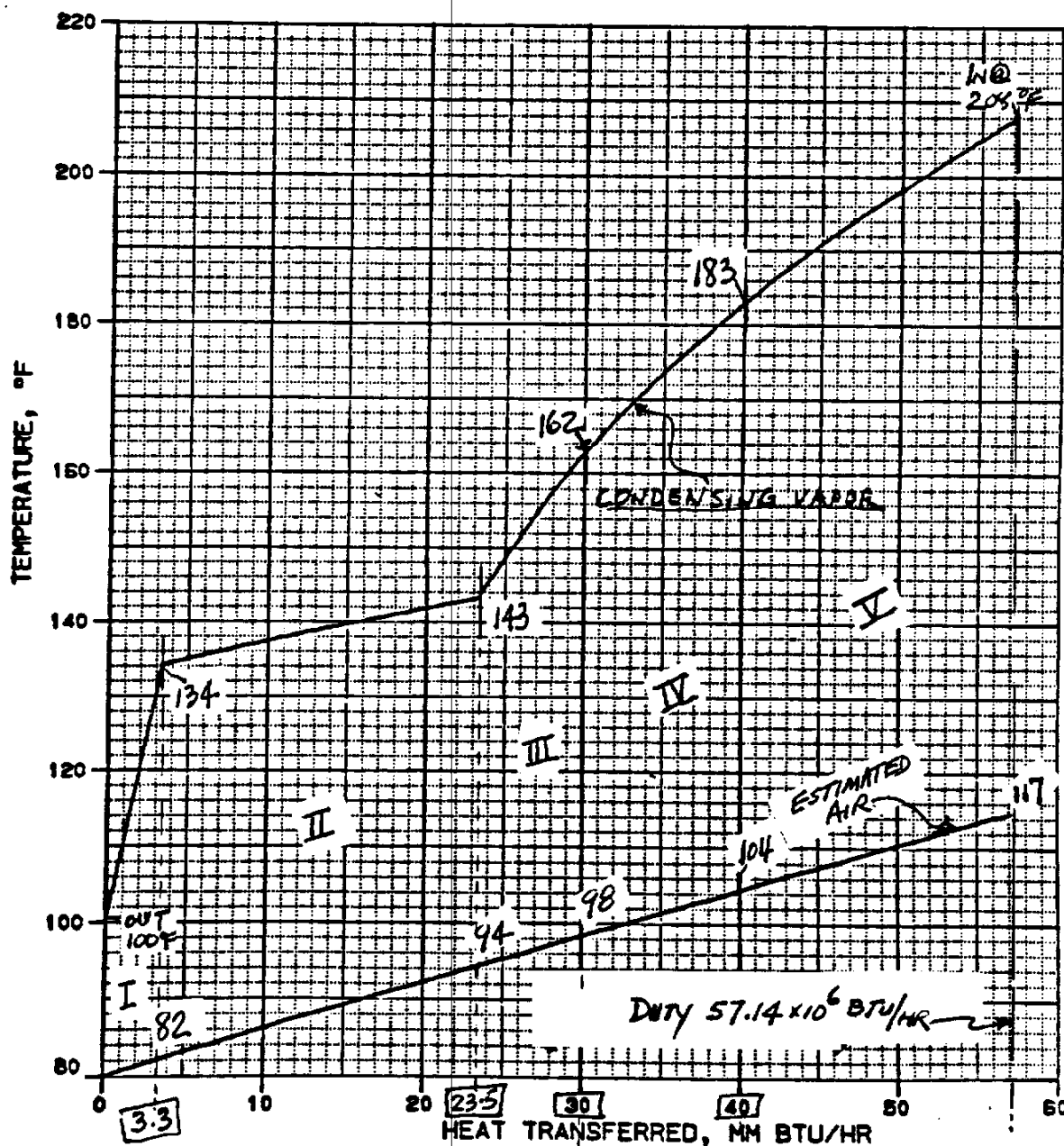
BY

APP'D

DATE

AIR COOLED EXCHANGERSEXHIBIT 2AHEAT RELEASE CURVES(RECOVERY COLUMN CONDENSER)

→ SAMPLE; WTD, MTD.



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EXHIBIT 2B

SAMPLE CALCULATION OF WEIGHTED LMTD

(1) DIVIDE PROFILE INTO CONVENIENT ZONES:

	V	IV	III	II	I	
$\Delta Q, \text{BTU/HR}, 10^6$	7.14	10	6.5	20.2	3.3	$\Sigma Q = 57.14 \times 10^6$
	208 → 183	183 → 162	162 → 143	143 → 134	134 → 100	°F HC
	117 ← 104	104 ← 98	98 ← 94	94 ← 82	82 ← 80	°F AIR
	91	79	64	49	52	20
MTD =	84.9	71.2	56.2	50.5	33.5	

(2) DIVIDE EACH ZONE BY MTD FOR THAT ZONE

$$\left(\frac{Q}{T}\right)_V + \left(\frac{Q}{T}\right)_{IV} + \left(\frac{Q}{T}\right)_{III} + \left(\frac{Q}{T}\right)_{II} + \left(\frac{Q}{T}\right)_I$$

$$10^6, \quad 0.2019 + 0.1404 + 0.1156 + 0.40 + 0.098 = \Sigma \left(\frac{Q}{T}\right)$$

(3) WTD MTD (UNCORRECTED) = $\frac{Q_{TOTAL}}{\left(\frac{Q}{T}\right)_I + \left(\frac{Q}{T}\right)_{II} + \dots}$

$$\frac{57.14 \times 10^6}{0.9564 \times 10^6} = \underline{59.7^\circ\text{F}}$$

(4) IF STRAIGHT LINE MTD (UNCORRECTED): $\frac{208 \rightarrow 100}{117 \leftarrow 80}$

91 20

(5) % SURFACE SAVED:

$$\frac{(59.7 - 46.9)}{46.9} \times 100 = \underline{27.3\%}$$

46.9°F

(EXHIBIT 2C)

HEAT RELEASE CURVES

HOT	SIDE	DRY	DEW	POINT	195°F	PRESS	5 PSIG
HOT	SIDE	BUBBLE	POINT	111°F	PRESS	-1 PSIG	
HOT	SIDE	H ₂ O	DEW	POINT	162°F	PRESS	4 PSIG

(STRIPPER OVERHEAD CONDENSER)

HEAT TRANSFERRED 19.82 MM BTU/HR

ESTIMATED WEIGHTED COUNTERFLOW MTD 45°F ←

NOTE: IF STRAIGHT LINE MTD:

228 → 105

121 ← 80

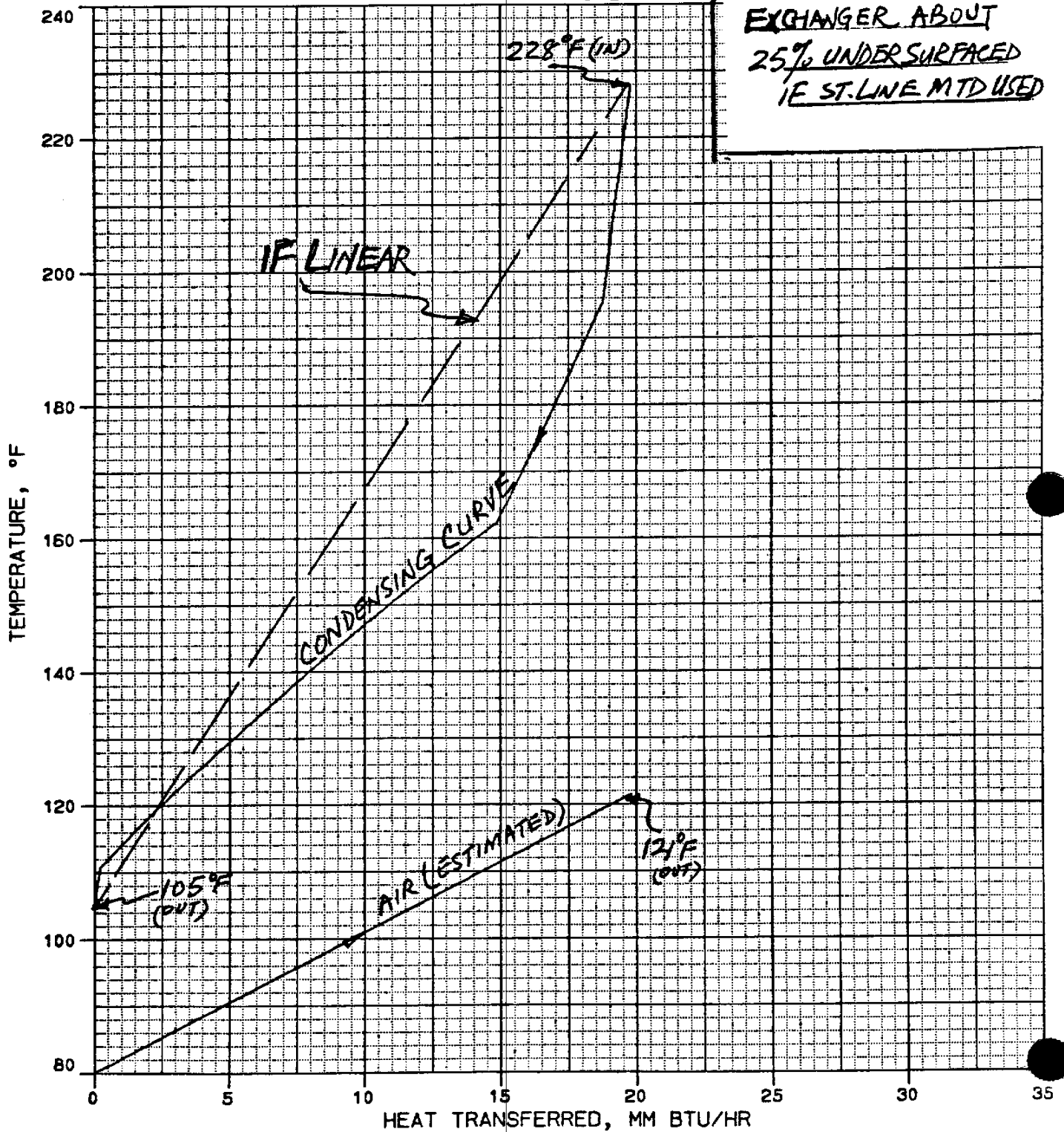
107 25

56.4°F UNCORRECTED

EXCHANGER ABOUT

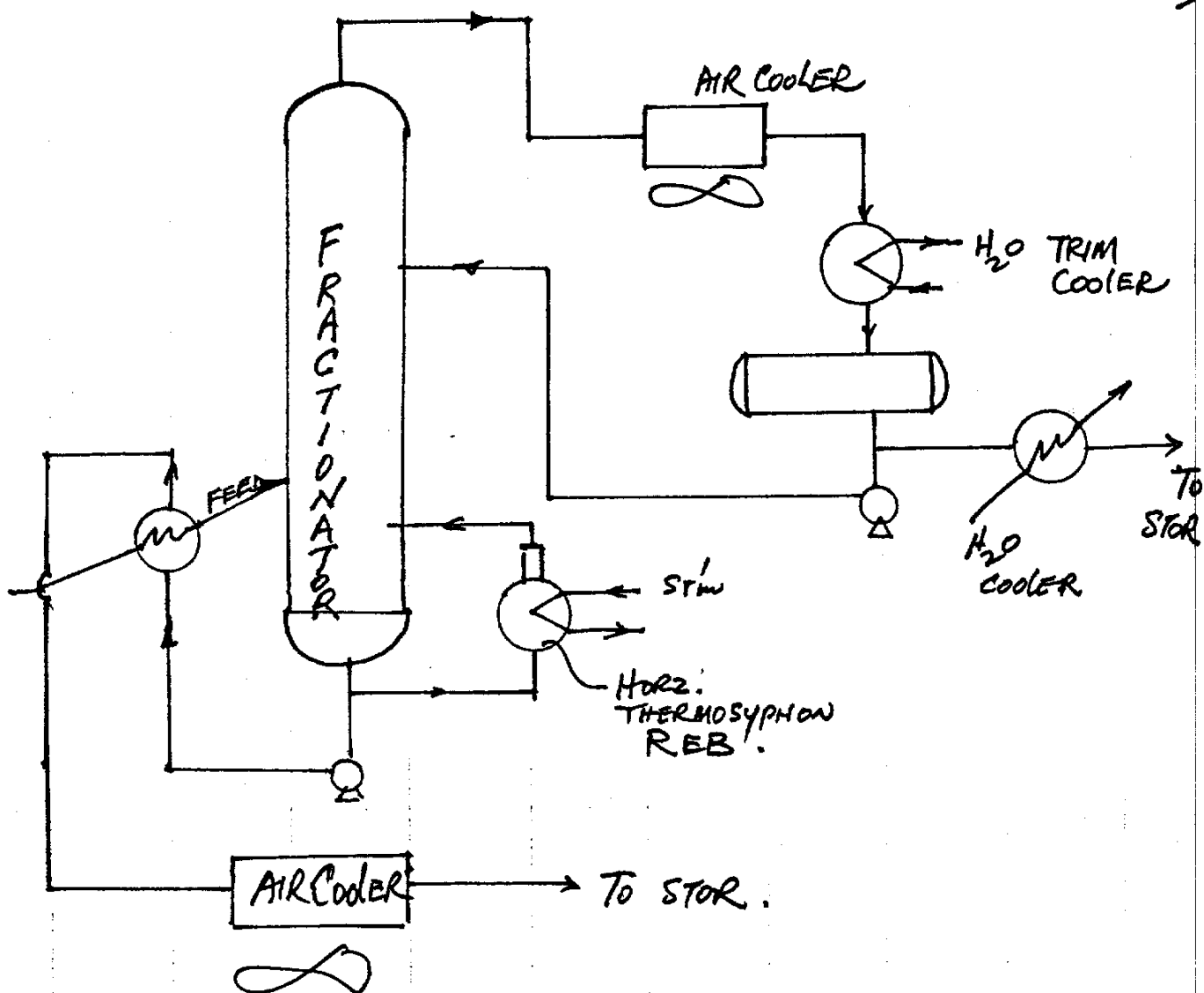
25% UNDERSURFACED

IF ST. LINE MTD USED



TYPICAL USE OF EXCHANGERS

(PROCESS CONSIDERATIONS)



TEMP APPROACHES

I PROCESS STREAM BEING COOLED

$$\begin{array}{rcl}
 130 & \longrightarrow & 110 \text{ HCBN} \\
 120 & \longleftarrow & 90 \text{ WATER} \\
 \hline
 10 & & 20
 \end{array}
 \begin{array}{l}
 \\
 \\
 \longleftarrow 20^\circ\text{F COLD END} \\
 \text{APPROACH}(\Delta T)
 \end{array}$$

II PROCESS STREAM BEING HEATED

$$\begin{array}{rcl}
 & 560 & \longrightarrow 410^\circ\text{F HOT OIL} \\
 & 460 & \longleftarrow 350 \text{ HCBN} \\
 & \hline
 & 100 & 60
 \end{array}
 \begin{array}{l}
 \\
 \\
 \longrightarrow 100^\circ\text{F} \\
 \text{HOT END APPROACH}(\Delta T)
 \end{array}$$

III IF BOTH STREAMS ARE PROCESS

$\Delta T(\text{APPROACH})$ IS THE LESSER TERMINAL ΔT .

$$\begin{array}{rcl}
 985 & \longrightarrow & 250^\circ\text{F} \\
 885 & \longleftarrow & 185 \\
 \hline
 100 & & 65
 \end{array}$$

TEMPERATURE OF PRODUCT TO STORAGE

IN GENERAL, LIGHTER HYDROCARBON
GOES TO STORAGE COLDER.

EXAMPLES:

- GASOLINE, BUTANES, $\approx 100^{\circ}\text{F}$
- KEROSENE; MID-DISTILLATES $\approx 125^{\circ}\text{F}$
- VACUUM GAS OIL $\approx 150^{\circ}\text{F}$
- REDUCED CRUDE ≈ 175 TO 200°F

FLUID ROUTING

IN GENERAL

- FOULING FLUIDS
- HIGH PRESSURE SIDE
- SLURRY (FCC UNIT)
- COOLING WATER
- STEAM
- HIGH TEMP. FLUID
- CORROSIVE FLUID
(Ex: HF. ACID)

IN TUBES

- HIGH VISCOSITY FLUIDS
(≥ 5 CENTIPOISE)
- LOW FLOW

ON SHELL
SIDE

TUBE PATTERN

IN GENERAL

- SQ. PITCH FOR REBOILERS
(IF SHELL SIDE BOILING)
- SQUARE OR ROTATED SQ. PITCH
WHEN SHELL SIDE FLUID IS FOULING
($\geq 0.002 \text{ HR}^\circ\text{FT}^2/\text{BTU}$)
THIS ALLOWS MECHANICAL CLEANING
- TRIANGULAR PITCH CAN BE
USED FOR CLEANER FLUIDS
IN SHELL ($\leq 0.0015 \text{ HR}^\circ\text{FT}^2/\text{BTU}$)
- ALSO, 15 TO 30% MORE TUBES
FOR SAME SIZE SHELL

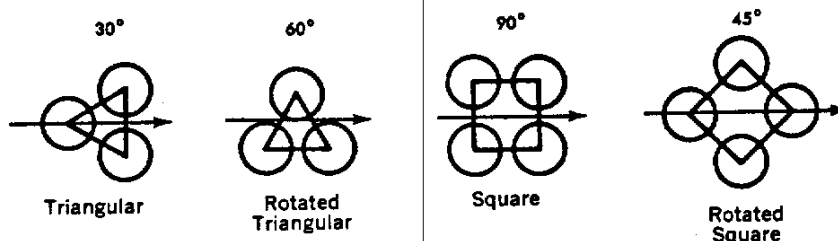
SECTION 5 Mechanical Standards TEMA Class "R" Heat Exchangers

R-2 TUBES—(Continued)

R-2.4 TUBE PATTERN

Standard tube patterns are shown in Figure R-2.4.

FIGURE R-2.4



Note: Flow arrows are normal to the baffle cut edge.

R-2.41 SQUARE PATTERN

In removable bundle units, when the tubes are laid out on a square or a rotated square pattern, tube lanes should be continuous throughout the bundle.

R-2.42 TRIANGULAR PATTERN

Triangular or rotated triangular pattern should not be used when the shell side is to be cleaned mechanically.

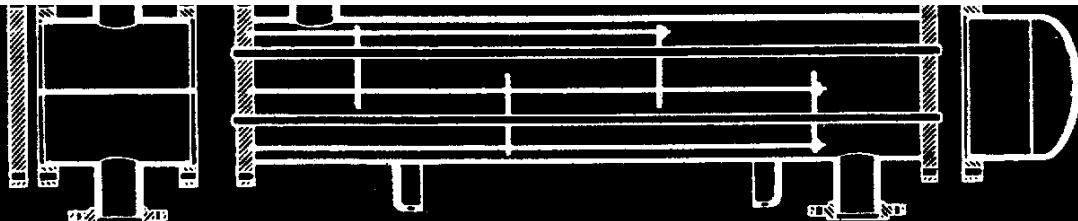
R-2.5 TUBE PITCH

Tubes shall be spaced with a minimum center-to-center distance of 1.25 times the outside diameter of the tube. When tubes are on a square pattern, a minimum cleaning lane of $\frac{1}{4}$ " shall be provided. Tolerances shown in Paragraph R-7.2 will apply.

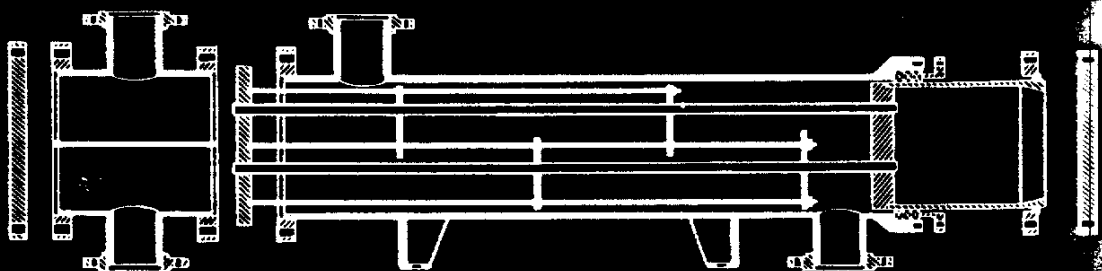
R-2.6 TUBE LAYOUT

The tube layout should minimize by-passing around the tube bundle. See Paragraph R-4.8.

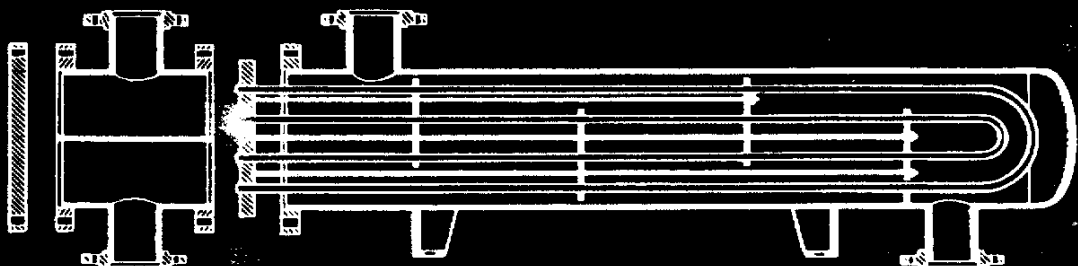
**FIXED
TUBE SHEET**



**EXTERNALLY
PACKED
FLOATING
HEAD**



**U"-TUBE
REMOVABLE
BUNDLE**



TUBE COUNT TABLES

The Tube Count Tables on these pages may be effectively applied with the type of heat exchangers depicted above.

The number of tubes shown is the optimum number of tubes that can be installed in the size and type of heat exchangers listed. These tube counts are based on TEMA nominal baffle and tube-to-shell clearances and TEMA bundle entrance and exit area requirements. The shell side nozzle sizes are approximately $1/5$ of the shell diameter with an impingement plate beneath the entrance nozzle and a ρV^2 value of 1500 in the nozzles.

3/4" 15/16" PITCH

SHELL I. D. (INCHES)	FIXED TUBE SHEET				OUTSIDE PACKED FLOATING HEAD				U-TUBE		
	NO. OF PASSES				NO. OF PASSES				NO. OF PASSES		
	1	2	4	8	1	2	4	8	2	4	8
17.25	254	236	224	212	229	216	200	188	101	98	92
19.25	330	322	300	272	301	290	268	244	141	130	120
21.25	387	370	348	336	358	346	324	316	163	160	152
23.25	483	472	448	412	450	436	412	384	213	202	190
25.00	565	544	512	480	522	502	472	432	246	238	216
27.00	658	634	608	560	612	598	568	520	294	280	260
29.00	763	740	704	660	719	698	656	616	343	326	308
31.00	875	848	808	756	827	800	768	716	394	382	356
33.00	993	964	932	872	942	918	880	824	450	438	408
35.00	1118	1086	1044	1020	1064	1034	988	956	510	494	484
37.00	1285	1248	1208	1144	1221	1184	1144	1088	587	570	542
39.00	1426	1396	1344	1272	1351	1328	1280	1220	660	640	608
42.00	1641	1602	1548	1488	1569	1538	1480	1420	759	740	706
45.00	1906	1864	1808	1736	1837	1792	1732	1664	887	868	834
48.00	2188	2142	2088	2004	2102	2062	2000	1912	1026	1002	966
51.00	2480	2440	2376	2292	2396	2356	2292	2204	1171	1140	1104
54.00	2769	2716	2652	2568	2672	2626	2564	2480	1304	1282	1226
60.00	3451	3396	3320	3220	3345	3288	3208	3116	1642	1608	1562
64.00	3921	3860	3792	3676	3798	3744	3668	3560	1869	1840	1784
68.00	4432	4366	4284	4160	4305	4244	4148	4032	2114	2086	2020
72.00	5023	4956	4868	4744	4878	4816	4732	4600	2406	2372	2306
76.00	5578	5506	5412	5276	5449	5374	5284	5156	2680	2640	2574
80.00	6179	6100	5996	5868	6031	5964	5868	5720	2968	2930	2862
84.00	6868	6788	6688	6544	6717	6644	6532	6384	3313	3272	3196
90.00	7873	7798	7680	7528	7724	7634	7520	7376	3809	3762	3686
96.00	9019	8926	8812	8632	8849	8762	8644	8472	4368	4320	4238
108.00	11377	11272	11132	10928	11196	11096	10968	10788	5524	5470	5382
120.00	14186	14068	13924	13720	13982	13888	13732	13516	6913	6846	6748

D-15 B

TUBE COUNT TABLES

CHARTS INDICATE
NUMBER OF TUBES

3/4" O.D. TUBES 1" 

SHELL I. D. (INCHES)	FIXED TUBE SHEET				OUTSIDE PACKED FLOATING HEAD				U-TUBE		
	NO. OF PASSES				NO. OF PASSES				NO. OF PASSES		
	1	2	4	8	1	2	4	8	2	4	8
17.25	229	220	208	184	204	192	176	164	94	90	80
19.25	283	272	260	248	260	250	228	212	120	114	110
21.25	350	334	324	300	322	310	292	268	149	142	132
23.25	426	406	392	364	394	382	364	336	183	176	162
25.00	491	476	452	420	450	440	412	384	217	208	192
27.00	576	562	532	500	537	524	492	456	256	248	228
29.00	674	650	620	600	633	610	580	552	299	290	280
31.00	771	746	712	672	727	704	676	628	347	336	314
33.00	875	852	820	768	827	812	776	728	398	386	360
35.00	985	960	928	896	934	914	876	848	448	438	428
37.00	1109	1074	1044	1016	1052	1026	988	952	521	492	484
39.00	1234	1236	1192	1144	1179	1172	1124	1080	583	568	542
42.00	1457	1418	1376	1316	1391	1358	1316	1252	670	656	628
45.00	1657	1650	1604	1540	1589	1588	1540	1476	787	768	742
48.00	1910	1872	1824	1780	1845	1800	1752	1716	910	876	858
51.00	2176	2142	2088	2016	2094	2062	2004	1940	1026	1006	968
54.00	2427	2378	2332	2284	2341	2308	2248	2200	1162	1122	1106
60.00	2985	2996	2928	2844	2903	2894	2828	2744	1441	1420	1376
64.00	3467	3408	3340	3252	3357	3308	3240	3140	1646	1622	1576
68.00	3913	3858	3792	3680	3795	3738	3672	3572	1865	1842	1792
72.00	4383	4330	4256	4148	4268	4208	4144	4024	2104	2072	2014
76.00	4878	4876	4792	4664	4771	4756	4676	4576	2371	2338	2282
80.00	5419	5352	5260	5204	5289	5228	5152	5092	2633	2574	2538
84.00	6031	5972	5884	5760	5902	5832	5748	5620	2909	2876	2808
90.00	6920	6860	6760	6632	6801	6726	6636	6492	3351	3312	3242
96.00	7897	7810	7712	7620	7736	7658	7568	7492	3818	3780	3746
108.00	9991	9888	9776	9704	9822	9740	9628	9548	4884	4804	4762
120.00	12413	12310	12180	12068	12246	12142	12016	11932	6090	5998	5942

3/4" O.D. TUBES 1" 

SHELL I. D. (INCHES)	FIXED TUBE SHEET				OUTSIDE PACKED FLOATING HEAD				U-TUBE		
	NO. OF PASSES				NO. OF PASSES				NO. OF PASSES		
	1	2	4	8	1	2	4	8	2	4	8
17.25	203	192	180	176	180	168	160	148	80	82	76
19.25	251	240	228	220	226	216	204	204	103	104	100
21.25	300	294	288	256	276	272	264	228	130	128	114
23.25	368	360	356	324	340	336	324	304	162	160	146
25.00	426	400	392	380	398	376	360	344	190	180	172
27.00	508	472	468	456	468	442	432	420	228	214	212
29.00	592	560	540	532	550	520	508	492	268	256	246
31.00	670	648	628	584	627	610	592	556	296	298	278
33.00	767	740	720	680	719	700	680	648	342	342	322
35.00	870	840	824	776	816	802	784	736	392	390	368
37.00	972	944	928	912	927	900	884	868	444	438	432
39.00	1086	1064	1036	1020	1036	1014	992	968	498	498	484
42.00	1257	1220	1212	1184	1204	1176	1148	1120	579	574	562
45.00	1452	1436	1412	1352	1396	1380	1360	1308	683	678	652
48.00	1666	1634	1608	1576	1602	1574	1540	1516	778	772	762
51.00	1880	1868	1848	1784	1814	1800	1768	1716	890	888	862
54.00	2125	2096	2064	2000	2051	2020	1988	1928	998	988	962
60.00	2608	2596	2568	2496	2532	2512	2468	2412	1246	1244	1208
64.00	2996	2982	2936	2868	2892	2884	2840	2776	1436	1424	1390
68.00	3375	3328	3288	3260	3270	3232	3188	3164	1605	1598	1588
72.00	3826	3752	3720	3676	3714	3648	3608	3560	1842	1808	1790
76.00	4270	4196	4152	4112	4170	4088	4052	4008	2060	2030	2004
80.00	4715	4664	4624	4528	4614	4560	4524	4424	2266	2260	2212
84.00	5223	5168	5120	5028	5109	5056	5000	4916	2512	2498	2454
90.00	5992	5964	5912	5816	5870	5844	5796	5680	2908	2900	2844
96.00	6852	6812	6780	6648	6720	6688	6636	6508	3331	3318	3256
108.00	8682	8618	8552	8412	8543	8476	8412	8288	4214	4198	4134
120.00	10782	10704	10636	10488	10634	10560	10492	10344	5252	5242	5164

1" O.D. TUBES 1-1/4" 

SHELL I. D. (INCHES)	FIXED TUBE SHEET				OUTSIDE PACKED FLOATING HEAD				U-TUBE		
	NO. OF PASSES				NO. OF PASSES				NO. OF PASSES		
	1	2	4	8	1	2	4	8	2	4	8
17.25	143	130	124	104	130	118	112	96	53	50	44
19.25	177	170	160	152	159	152	144	132	69	66	62
21.25	208	216	200	180	192	196	184	168	92	86	76
23.25	254	262	244	220	241	238	224	212	112	104	96
25.00	304	290	276	264	285	272	252	240	126	120	114
27.00	359	350	324	316	341	322	304	292	154	144	138
29.00	422	404	384	372	396	382	364	348	181	174	168
31.00	483	472	452	424	455	444	428	396	213	206	190
33.00	565	544	524	484	526	512	484	456	246	238	220
35.00	611	614	588	556	588	582	564	520	277	268	254
37.00	695	698	676	628	658	664	632	600	321	310	290
39.00	783	784	748	716	748	744	716	672	349	346	326
42.00	905	906	880	832	863	868	832	792	420	408	386
45.00	1069	1038	1008	956	1020	998	964	924	484	472	452
48.00	1219	1182	1152	1124	1168	1136	1104	1076	553	540	530
51.00	1391	1370	1328	1264	1343	1312	1276	1232	644	626	596
54.00	1556	1538	1484	1428	1510	1478	1436	1388	721	704	676
60.00	1914	1876	1832	1788	1849	1812	1772	1744	891	876	856
64.00	2200	2158	2108	2064	2126	2090	2044	2008	1030	1008	992
68.00	2494	2456	2400	2336	2420	2384	2336	2260	1175	1152	1112
72.00	2777	2778	2724	2644	2702	2690	2648	2552	1331	1304	1270
76.00	3117	3062	3008	2960	3038	2996	2936	2888	1471	1450	1428
80.00	3471	3418	3364	3276	3388	3346	3284	3200	1646	1620	1582
84.00	3846	3792	3736	3632	3751	3710	3644	3540	1829	1802	1754
90.00	4391	4342	4276	4232	4309	4252	4188	4124	2096	2070	2046
96.00	5047	4984	4912	4804	4942	4892	4812	4700	2416	2382	2328
108.00	6386	6310	6232	6104	6285	6224	6148	6032	3070	3038	2982
120.00	7905	7900	7800	7660	7784	7788	7684	7556	3849	3808	3744

1" O.D. TUBES 1-1/4" 

SHELL I. D. (INCHES)	FIXED TUBE SHEET				OUTSIDE PACKED FLOATING HEAD				U-TUBE		
	NO. OF PASSES				NO. OF PASSES				NO. OF PASSES		
	1	2	4	8	1	2	4	8	2	4	8
17.25	124	110	104	96	109	100	92	92	49	44	44
19.25	151	144	136	132	136	132	124	104	59	56	52
21.25	186	186	176	160	174	168	164	132	78	74	66
23.25	228	228	220	204	216	208	204	188	96	98	88
25.00	263	252	240	228	242	232	228	216	108	106	106
27.00	319	302	292	288	296	280	276	256	132	132	126
29.00	373	360	348	320	345	336	324	300	158	160	146
31.00	416	400	392	380	390	376	368	356	176	180	170
33.00	486	468	452	440	459	442	424	420	208	208	202
35.00	546	532	520	488	518	508	492	468	241	236	224
37.00	614	606	592	560	584	576	556	532	276	272	260
39.00	688	676	672	640	652	648	636	600	313	308	294
42.00	806	768	752	740	766	732	712	708	353	352	346

PROBLEM NO. 6 SOLUTION

$$\Delta P_2 = (G_2/G_1)^{1.8} (\Delta P_1)$$

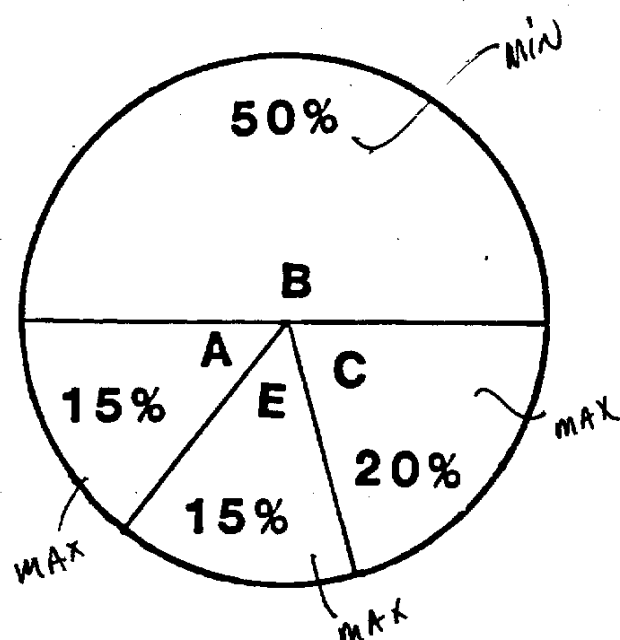
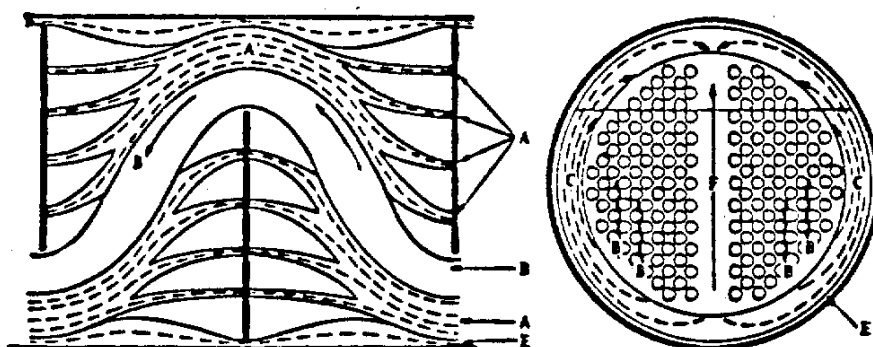
$$G_2 = G_1/2$$

$$\Delta P_2 = (G_1/2G_1)^{1.8} (3)$$

$$= (.5)^{1.8} (3)$$

$$\Delta P_2 = \underline{\underline{0.86}} \text{ psi}$$

Shellside Flow Distribution



ALLOWABLE PRESSURE DROP

D-17

- COOLING WATER IN TUBES; 8 TO 10 PSI PER EXCHANGER
(VELOCITY: 3 TO 5 FT/SEC)
- MULTIPLE SHELLS IN SERIES SUCH AS
HYDROTREATING COMBINED/FEED EXCHANGERS
IF 6 IN SERIES; ALLOW 8 PSI PER EXCHGR
OR 48 PSI TOTAL DELTA-P.

⇒ IN GENERAL

TUBE SIDE : ALLOW 5 TO 10 PSI PER EXCHGR

SHELL SIDE : 2 TO 5 PSI PER EXCHGR

SPECIAL CASES SUCH AS (HIGH VISCIOUS
MATERIAL) VACUUM BOTTOMS ETC
NEED MORE DELTA-P (15 PSI/SHELL)

REBOILERS : SHELL SIDE BOILING 0.5 TO 1.0 PSI
TUBE SIDE BOILING 2 TO 3.5 PSI
(VERT. REB)

CONDENSERS : LOW PRESSURE OPERATION OR
VACUUM SERVICE 0.5 TO 1.0 PSI

SPECIAL VERT. (SINGLE PASS) EXCHANGERS

10 PSI DELTA-P (SIZE APPROX: 60' x 65')
{ 2 PSI TUBE SIDE }
{ 8 PSI SHELL SIDE }

REBOILERS

IN GENERAL:

LIMIT TO ABOUT 33% VAPORIZATION

→ EXCEPT KETTLE & STAB-IN TYPE
(UP TO 80% VAPORIZATION)

→ FOR KETTLE & STAB-IN

LIMIT FLUX $\frac{Q}{A}$ $\left(\frac{\text{BTU}}{\text{HR. FT}^2} \right)$ TO

12,000 FOR HOT OIL } AS HEATING
15,000 FOR STEAM } MEDIUM.

SUMMARY

PROCESS CONSIDERATIONS

II

- A PURPOSE (WHY USE HT. EXCH'GS)
- B TEMP. APPROACHES.
- C PRODS. TO STORAGE.
- D FLUID ROUTING (TUBE/SHELL)
- E ALLOWABLE PRESSURE DROP
- F VARIOUS TYPES OF REBOILERS

MATERIAL OF CONSTRUCTION

→ DEPENDS ON PROCESS FLUIDS:

- HYDROGEN
- WET H_2S
- SEA WATER **
- TEMP / PRESSURE / SULFUR LEVEL ▲▲
- HF ACID ▲
- SLURRY (SUCH AS FCC. UNITS)

** SEA WATER REQUIRES NON-FERROUS MATERIALS
SUCH AS 70/30 Cu/Ni; AL-BRASS; TITANIUM; MONEL

▲ HF ACID CAN USE: KILLED C. STEEL (COLD Temp.)
MONEL (High Temp.)

▲▲ Temp / SULFUR LEVEL CAN REQUIRE UPGRADE

FROM CARBON STEEL → 1/4 CR - 1/2 MO → SEA - 1/2 MO → TP400/TP430

↓

TP 304 SS

MECHANICAL CONSIDERATION

GENERAL: TREAT EXCHANGER SHELL AS VESSEL
PER ASME SECT VIII PRESSURE
& VESSEL CODE

— USE TEMA CLASS "R"

DESIGN PRESSURE: MAX. OPERATING PLUS 25 PSI (MIN).

OR OPERATING PLUS 50 PSI (CHECK FLANGE RATING)

OR OPERATING PRESSURE + 10%

SPECIAL CASE: DIFFERENTIAL PRESSURE DESIGN.

THIS ALLOWS THINNER TUBE SHEET.

EXAMPLE: NAPHTHA HYDROTREATER:

→ [COMBINED FEED/EFFLUENT EXCHANGER]

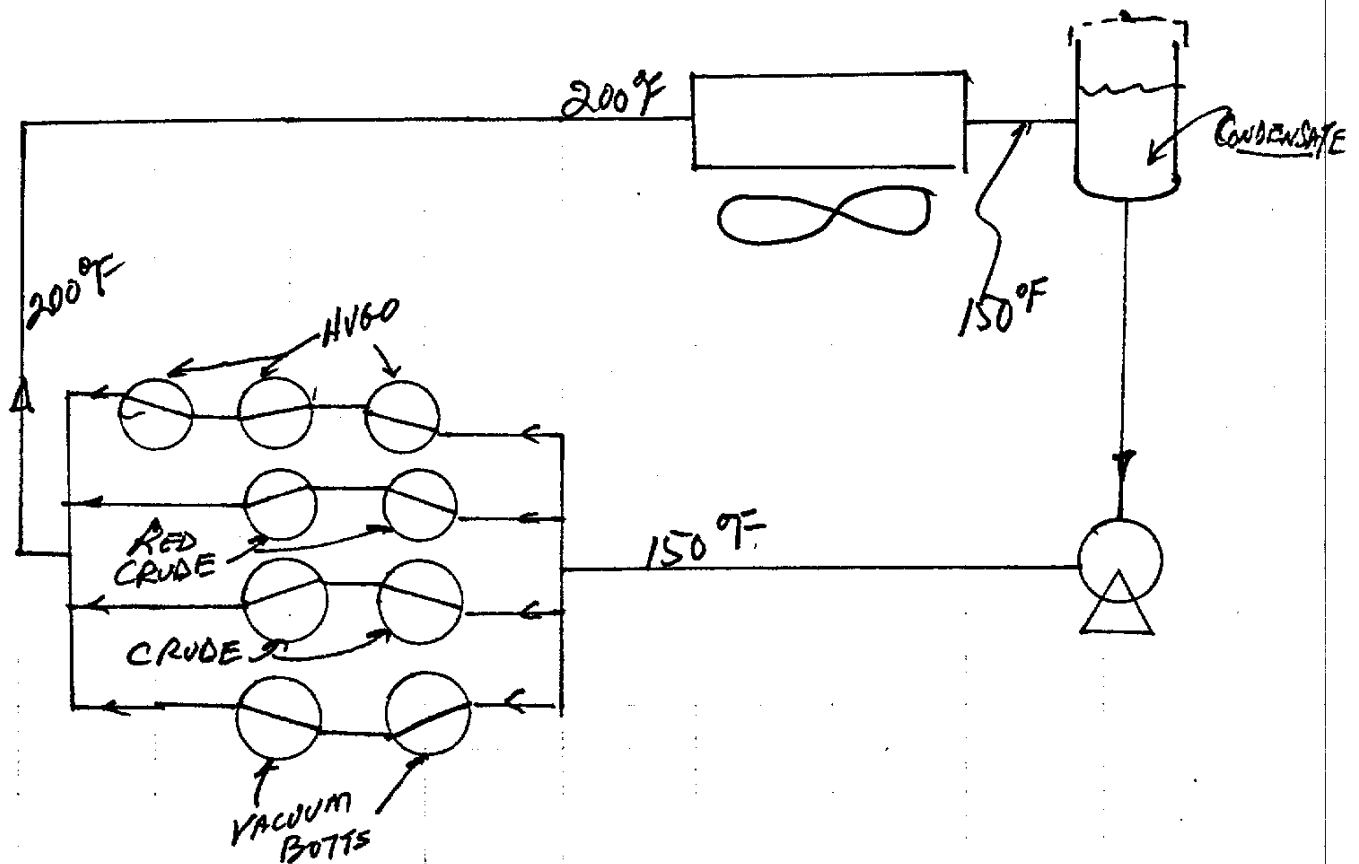
DESIGN TEMPERATURE

→ USUALLY OPERATING TEMP. PLUS 50°F

→ WATER SIDE; USUALLY 250°F

TYPICAL TEMPERED WATER SYSTEM

→ USED FOR HIGH POUR PT MATERIAL
≠ HIGH VISCOSITY "



REVAMP

- TO INCREASE THROUGH-PUT WITH EXISTING EQUIPMENT
- MAY HAVE TO PARALLEL/SERIES WITH EXISTING
- CHECK PRESSURE / TEMPERATURE DESIGN LIMITS
- CAN RATIO FLOWS, DENSITY ETC;

SEE 'NGPSA' TABLE FIG. P-7

APPENDIX 'A' (EXHIBIT 4).

HAIRPIN TYPE EXCHANGERS

IN GENERAL:

→ FOR SMALL DUTY : APPROXIMATELY ↓

DOUBLE PIPE TYPE : $\leq 1.0 \times 10^6 \text{ BTU/HR}$
 $(0.25 \times 10^6 \text{ Kcal/HR})$

→ MULTITUBE TYPE : 2 TO $5 \times 10^6 \text{ BTU/HR}$
 $(0.5 \text{ TO } 1.26 \text{ Kcal/HR})$

→ TEMPERATURE CROSS CAN BE ALLOWED
 FOR TRUE COUNTER CURRENT FLOW

EXAMPLE :

130 → 110 °F

120 ← 90

→ A SINGLE SECTION COULD BE USED
 IF SURFACE SUPPLIED IS ENOUGH.

→ ARRANGEMENT IS FLEXIBLE

CAN BE PARALLEL; SERIES,

OR

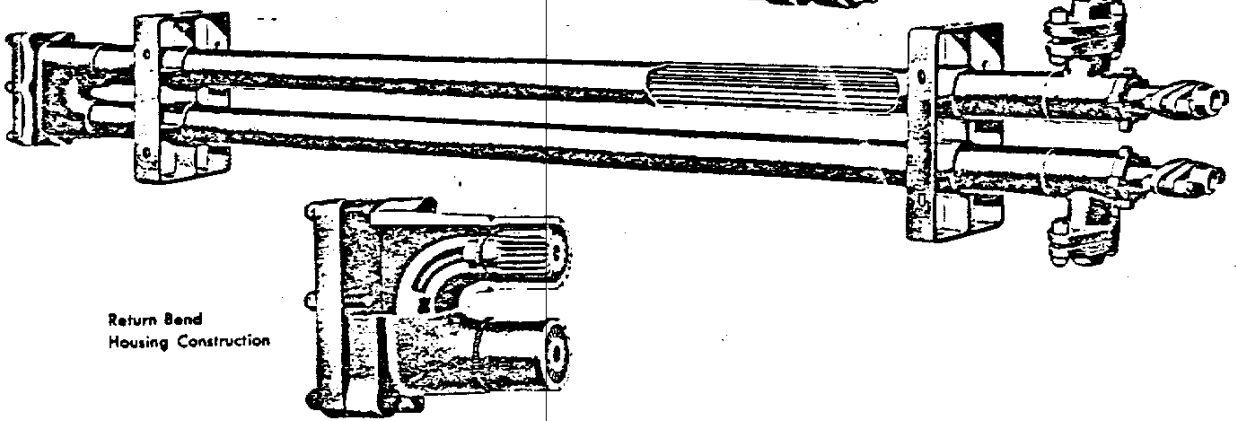
COMBINATION: SERIES/PARALLEL

Brown Fintube

TYPE "O"—3" LOW

EXHIBIT 5

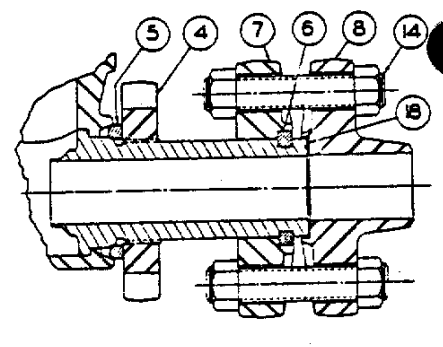
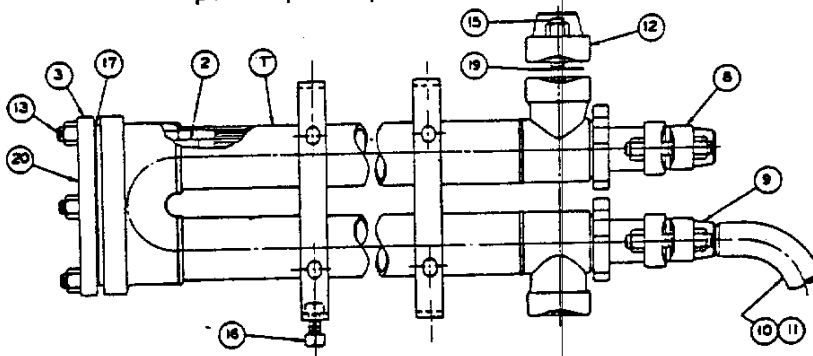
Shell to Tube Closure
and Tubeside Joint



Return Bend
Housing Construction

PARTS LIST

When ordering parts, furnish serial number as shown on the nameplate and designate parts required by number and name as shown in the following tables.



1. Shell Assembly
2. Tube Assembly
3. Cover Plate
4. Compression Flange
5. Sealing Ring
6. Split Ring
7. Fintube Fitting Flange
8. Fintube Stub End Flange
9. } Tube Return Bend Connector
10. }
11. }
12. Shell Nozzle Companion Flange
13. Cover Plate Bolting
14. Tubeside Bolting

15. Shell Nozzle Bolting
16. Bracket Bolts
17. Cover Gasket
18. Fintube Gasket
19. Shell Nozzle Gasket
20. Nameplate

RECOMMENDED SPARE PARTS INVENTORY PER SECTION

Two Cover Gaskets (17) Four Sealing Rings (5)
Four Fintube Gaskets (18) Two Split Rings (6)

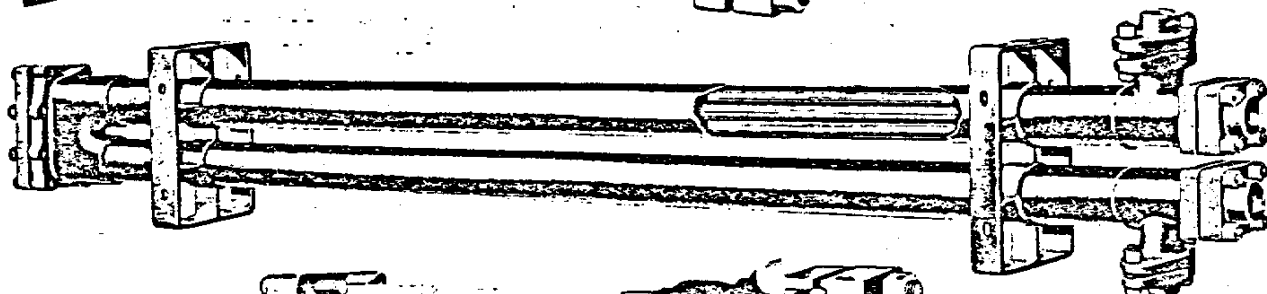
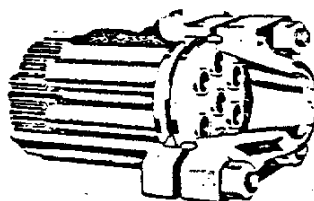
FOR ADDITIONAL SUGGESTIONS ON INSTALLATION

Brown Fintube

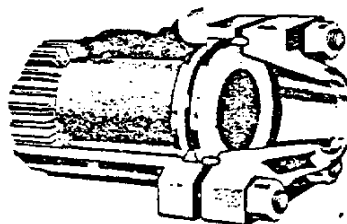
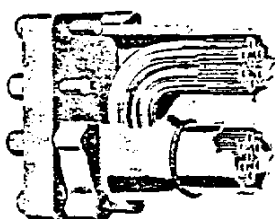
TYPES 2, 4, 6 and 8—4" LOW

EXHIBIT 6

Shell to Tube Closure and
Tubeside Joint of Multi-Tube Section



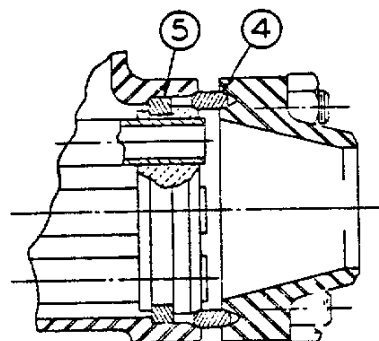
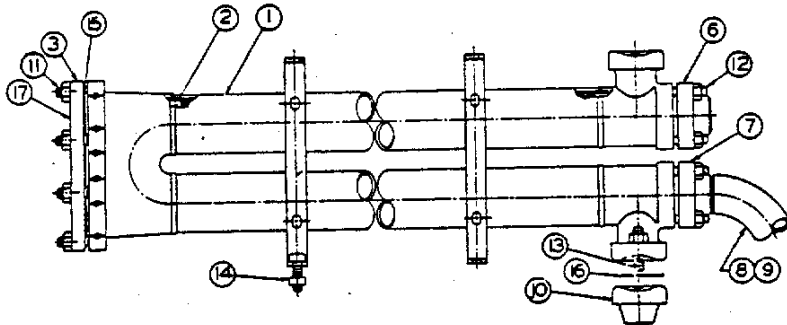
Return Band Housing Construction
of Multi-Tube Section



Shell to Tube Closure and
Tubeside Joint of Single Tube Section

PARTS LIST

When ordering parts, furnish serial number as shown on the nameplate and designate parts required by number and name as shown in the following tables.



1. Shell Assembly
2. Tube Assembly
3. Cover Plate
4. Sealing Ring
5. Split Ring
6. Fintube Stub End Flange
7. } Tube Return Bend Connector
8. }
9. }
10. Shell Nozzle Companion Flange
11. Cover Plate Bolting
12. Tubeside Bolting
13. Shell Nozzle Bolting

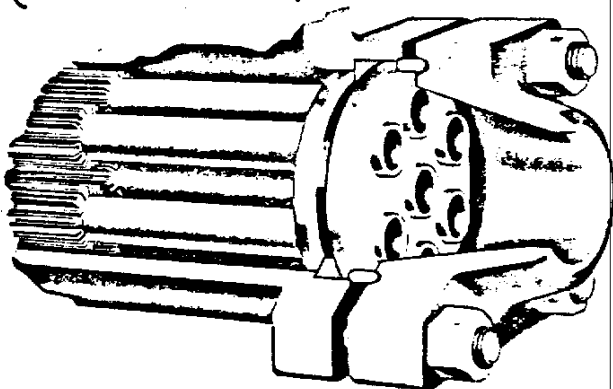
14. Bracket Bolts
15. Cover Gasket
16. Nozzle Gasket
17. Nameplate

RECOMMENDED SPARE PARTS INVENTORY PER SECTION

Two Cover Gaskets (15) Four Sealing Rings (4)
Two Split Rings (5)

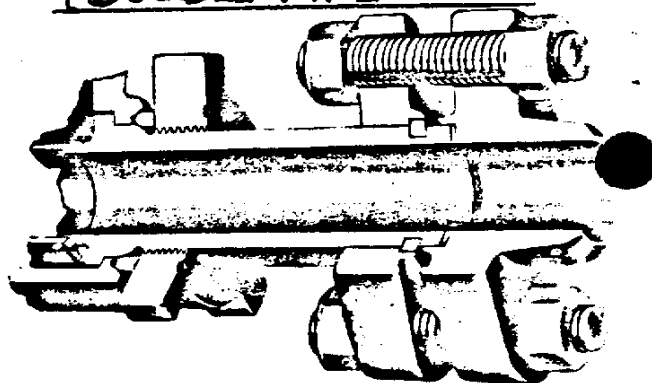
FOR ADDITIONAL SUGGESTIONS ON INSTALLATION

(MULTI-TUBE)

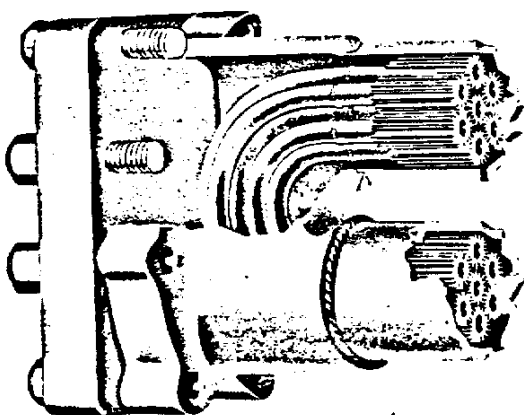


Shell to Tube Closure and Tubeside Joint of 4" Multi-tube Finned Low Pressure Section

(DOUBLE PIPE)

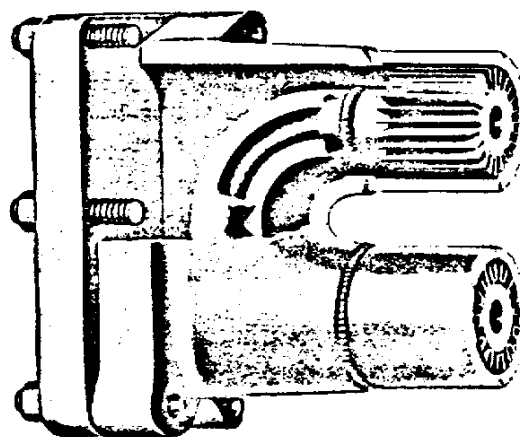


Shell to Tube Closure and Tubeside Joint of 3" Low Pressure Section.



"BROWN-FIN"

Cut-Away Sectional View Showing 4" Low Pressure (Return Bend Housing)

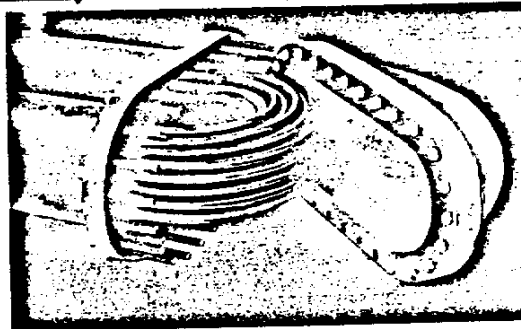
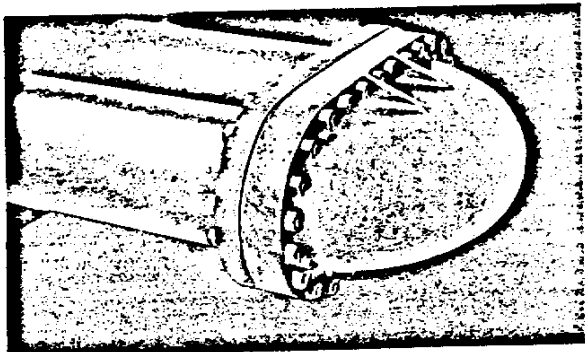


Cut-Away Sectional View Showing Low Pressure (Return Bend Housing Construction.)

↓ BASTEX CORP ↓

The full floating U-tube bundle compensates for the differential shell-tube expansion.

Only one casting is used in a Bas-Tex section and it is of simple design.



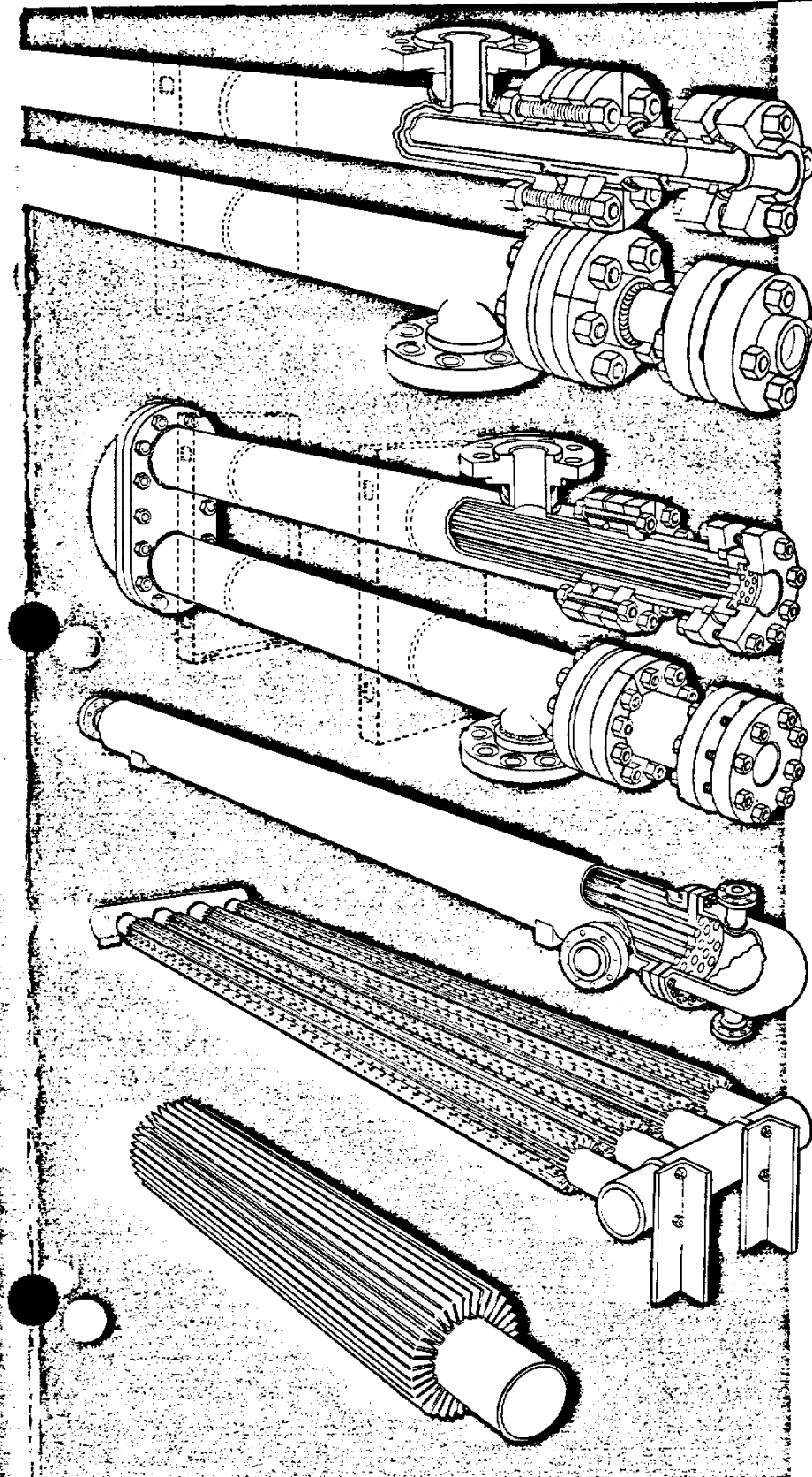
Return End Bonnet Closure is a conservative through bolted design. When the bonnet is removed the U-bends of the tubes are completely exposed to permit full field inspection.

BAS-TEX CORPORATION

**Quality heat exchangers
for petroleum, petrochemical,
power and general
industry applications**

EXHIBIT 6B

BULLETIN 60



Double pipe fintube heat exchangers

Bas-Tex double pipe exchangers feature the patented Bas-Tex closure — a combination of external split flange and compression ring flange — for easy removal and positive alignment. Through-bolt design cuts maintenance.

Multi-tube hairpin heat exchangers

These exchangers also feature the Bas-Tex patented split flange closure for easy removal of tube bundle. Quality construction assures dependability and long life in high and low pressure services. Full range of sizes for most applications.

Line and Suction heaters

Efficient for all viscous fluids. Features of Bas-Tex heaters include: heavy wall-12 BWG longitudinal fin tube surface, tapped and through bolted tube sheets which permit removal of bonnets without disturbing the shell seal, rugged full penetration welding, and ASME Code and TEMA Construction. A wide range of temperature-pressure ratings is available.

Horizontal and vertical fintube tank heaters

Bas-Tex design provides better steam dispersion, greater heating surface. Fewer welds mean lower maintenance and longer trouble-free operation. Compact design saves space, displaces less product. Both horizontal and vertical models available for a wide variety of applications.

Fintubes and custom equipment

Bas-Tex method of resistance welding assures welded fins in one piece with the tube, with maximum resistance to thermal shock. Available in carbon steel, stainless, monel, inconel, nickel and other weldable metals in a full range of sizes.

AIR COOLERS

- OBVIOUSLY USED WHERE WATER IS SCARCE.
- OFTEN USED IN TANDEM WITH WATER TRIM COOLER
- CAN BE ECONOMICAL, ESPECIALLY WHERE ONLY SEA WATER IS AVAILABLE
- TWO TYPES (BASICALLY)
- ① FORCED DRAFT (WITH FAN BELOW TUBE BUNDLE)
AIR IS FORCED OR PUSHED ACROSS TUBES



- ② INDUCED DRAFT (WITH FAN ABOVE TUBE BUNDLE)
AIR IS PULLED ACROSS TUBES

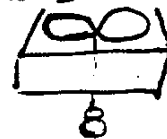


EXHIBIT 7A

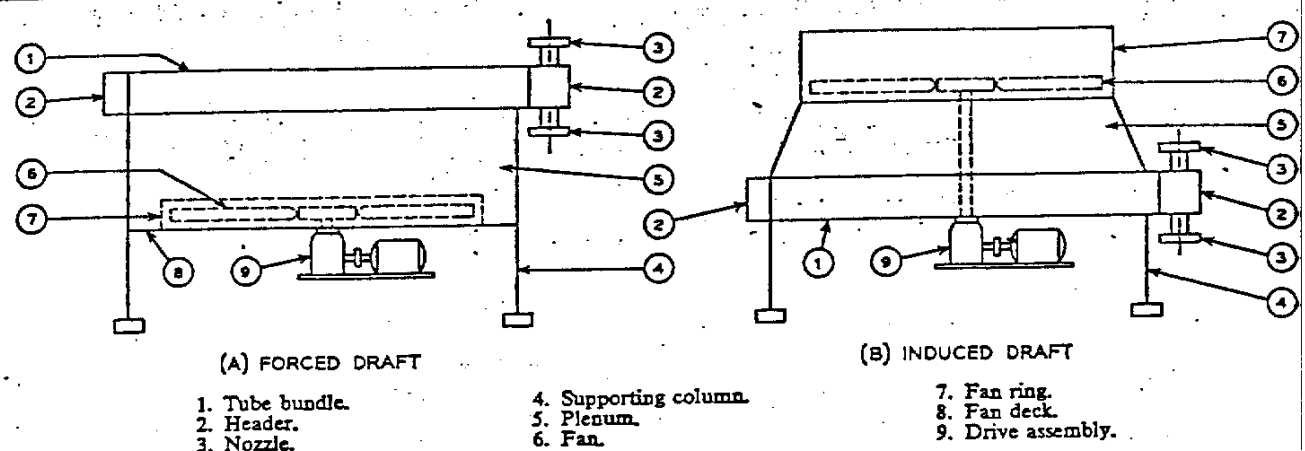


FIG. 1—Typical Components of an Air-Cooled Heat Exchanger.

EXHIBIT 7B

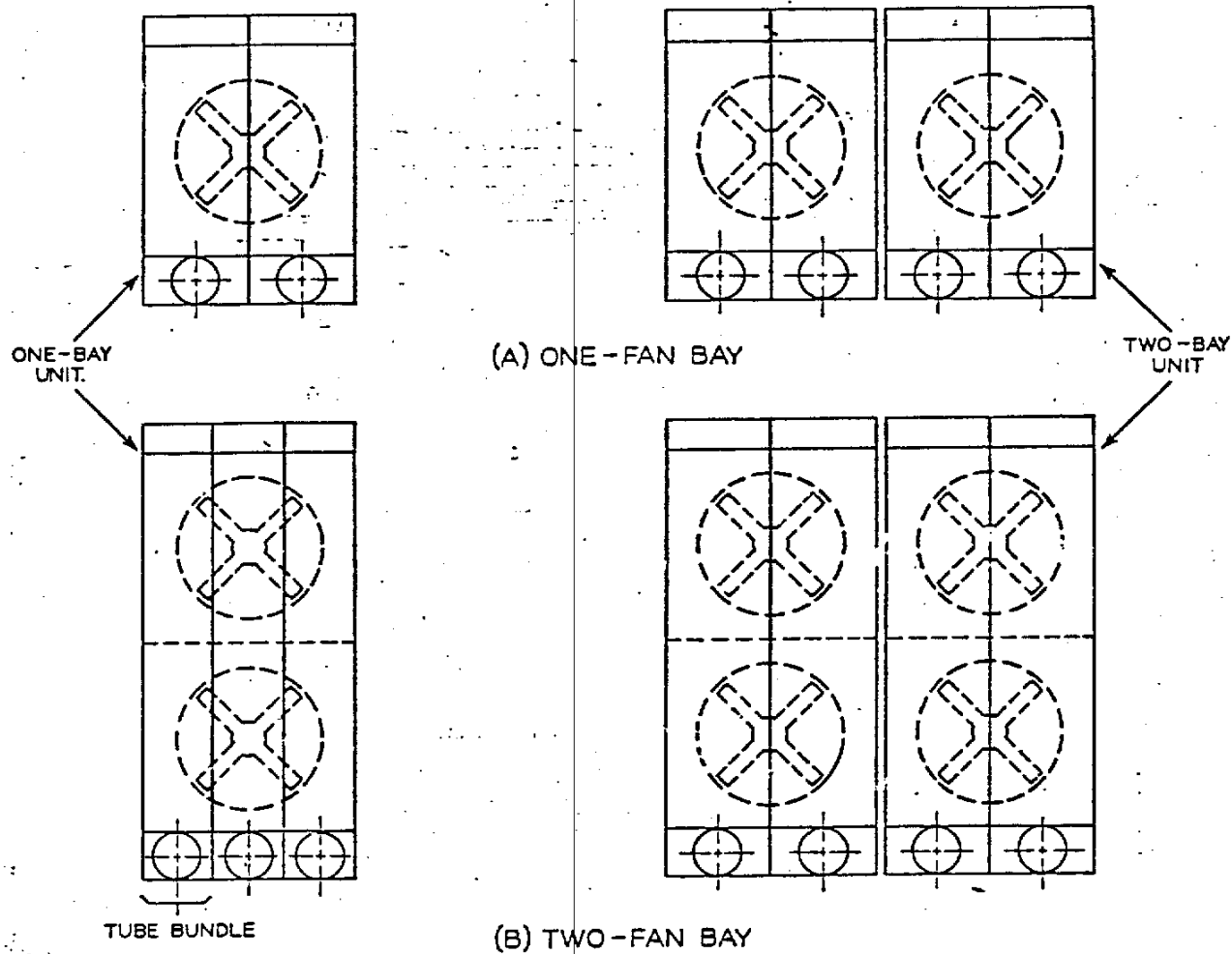


FIG. 2—Typical Bay Arrangements.

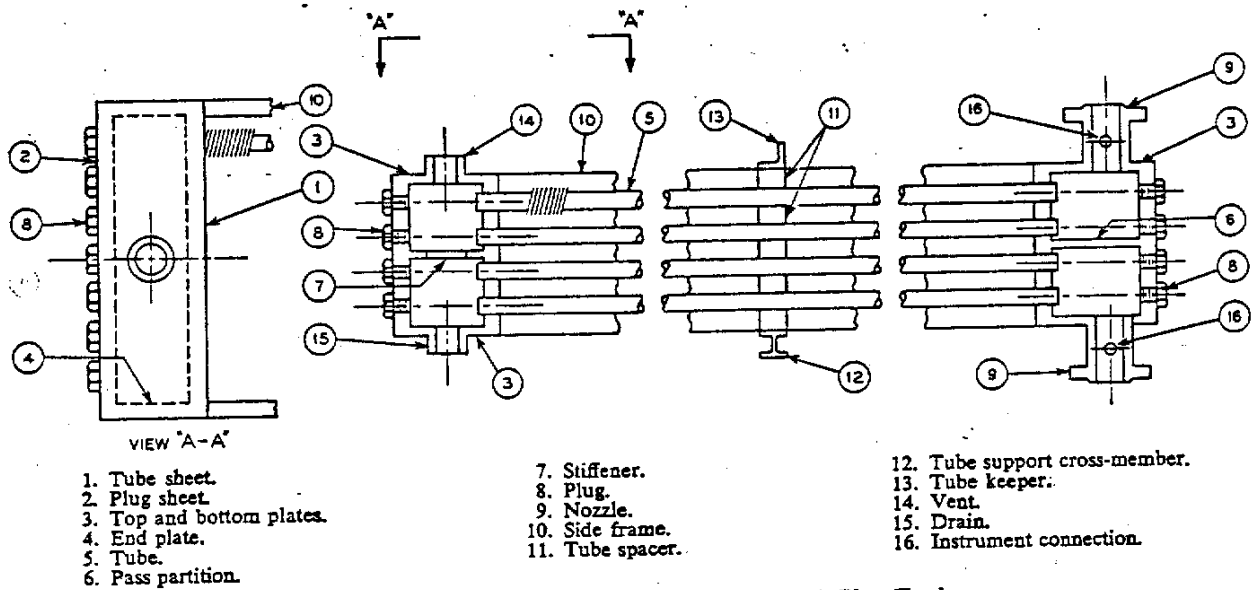


FIG. 5—Typical Construction of a Tube Bundle with Plug Headers.

EXHIBIT 7C

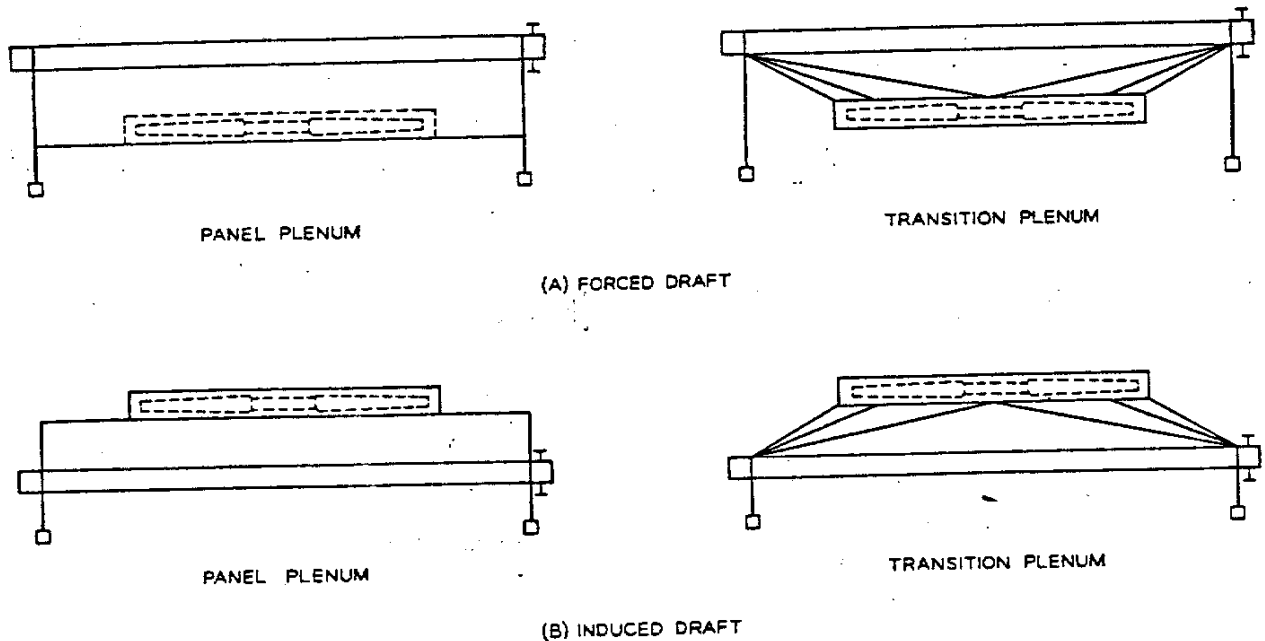


FIG. 6—Typical Plenum Arrangements.

EXHIBIT 7D

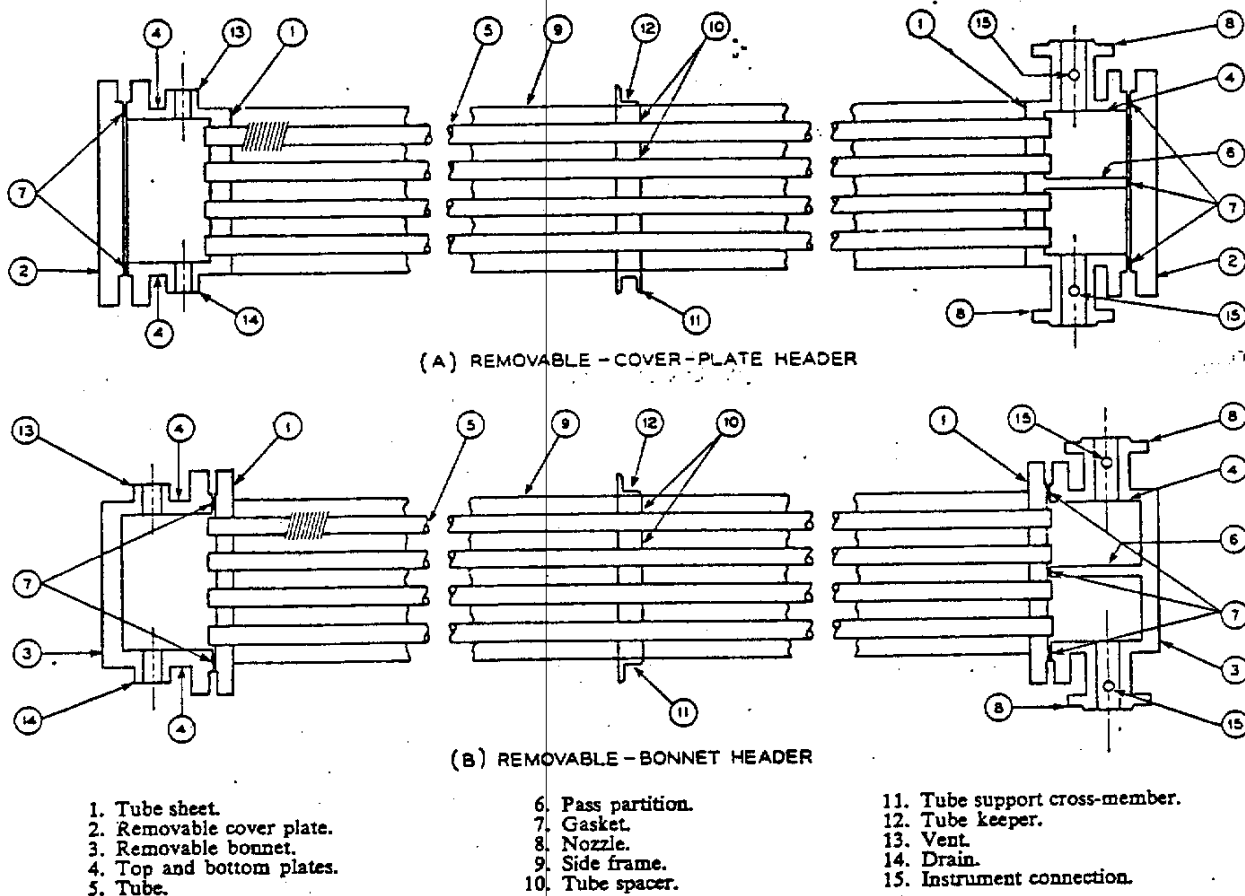


FIG. 3—Typical Construction of Tube Bundles with Removable-Cover-Plate and Removable-Bonnet Headers.

EXHIBIT 7E

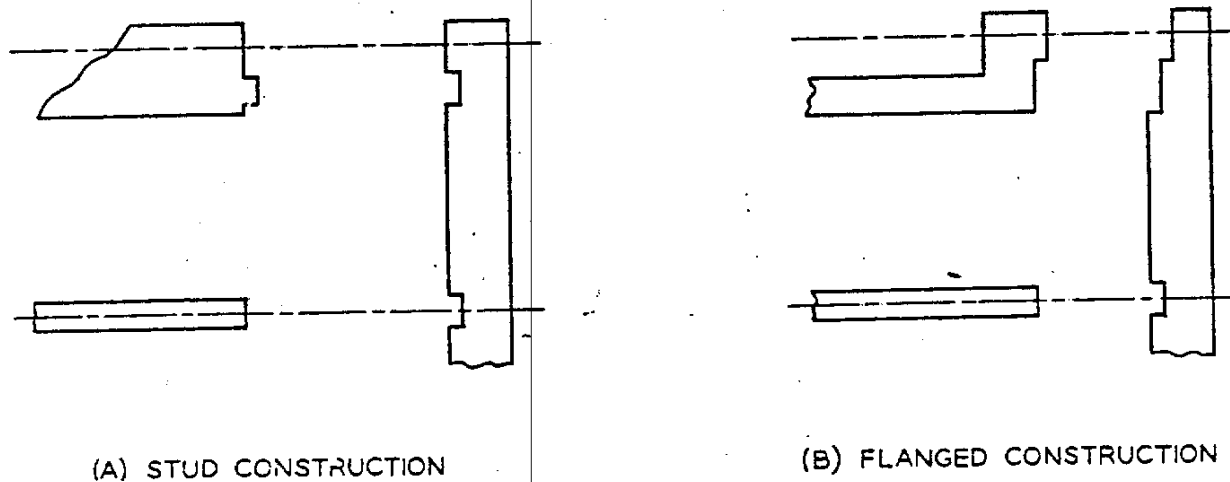
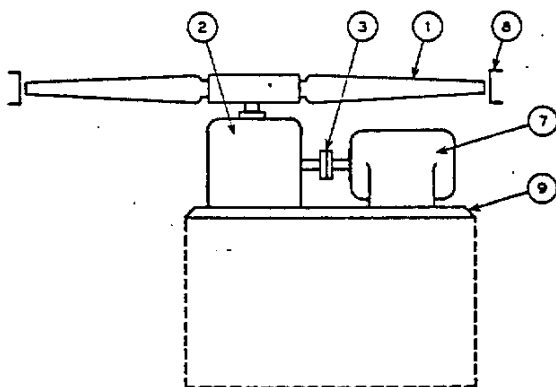
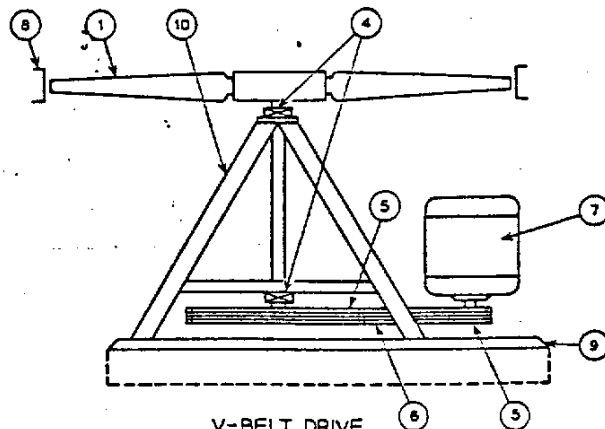


FIG. 4—Typical Confined Gasket Joint Details.

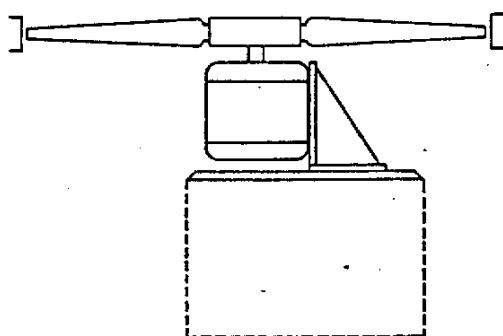
EXHIBIT 7F



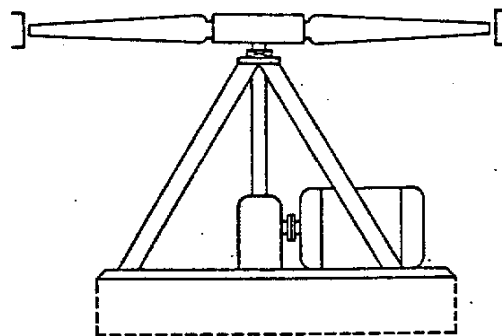
DIRECT RIGHT-ANGLE GEAR DRIVE



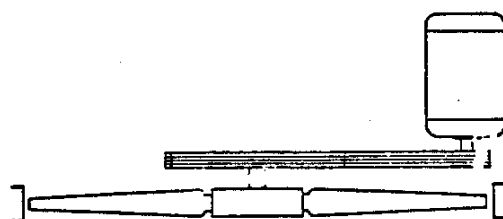
V-BELT DRIVE



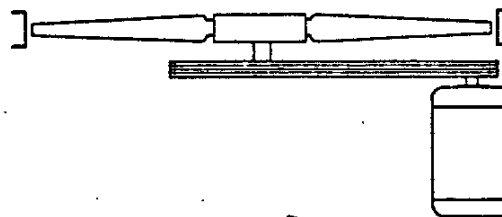
DIRECT MOTOR DRIVE



RIGHT-ANGLE GEAR DRIVE WITH FAN SUPPORT



TOP-SUSPENDED V-BELT DRIVE



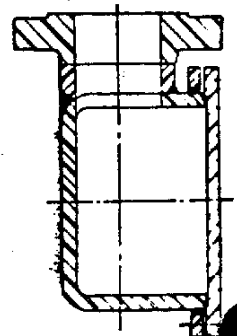
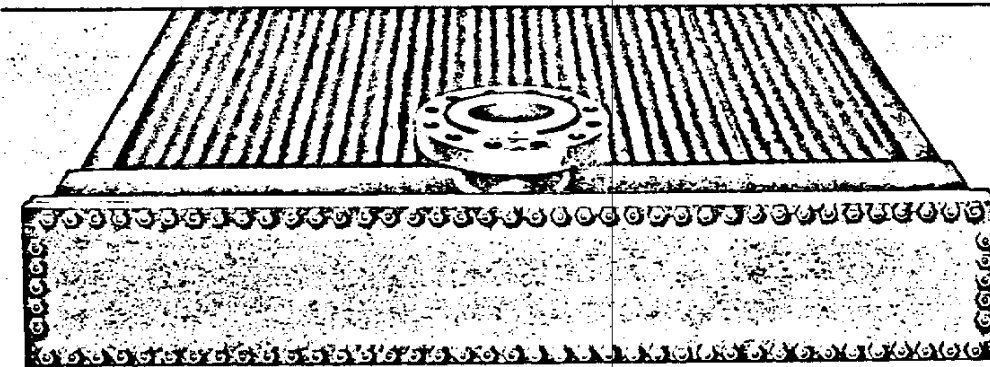
BOTTOM-SUSPENDED V-BELT DRIVE

1. Fan.
2. Gear.
3. Coupling.
4. Bearing.
5. Sheave.

6. V-belts.
7. Motor.
8. Fan ring.
9. Base plate.
10. Fan support.

FIG. 7—Typical Drive Arrangements.

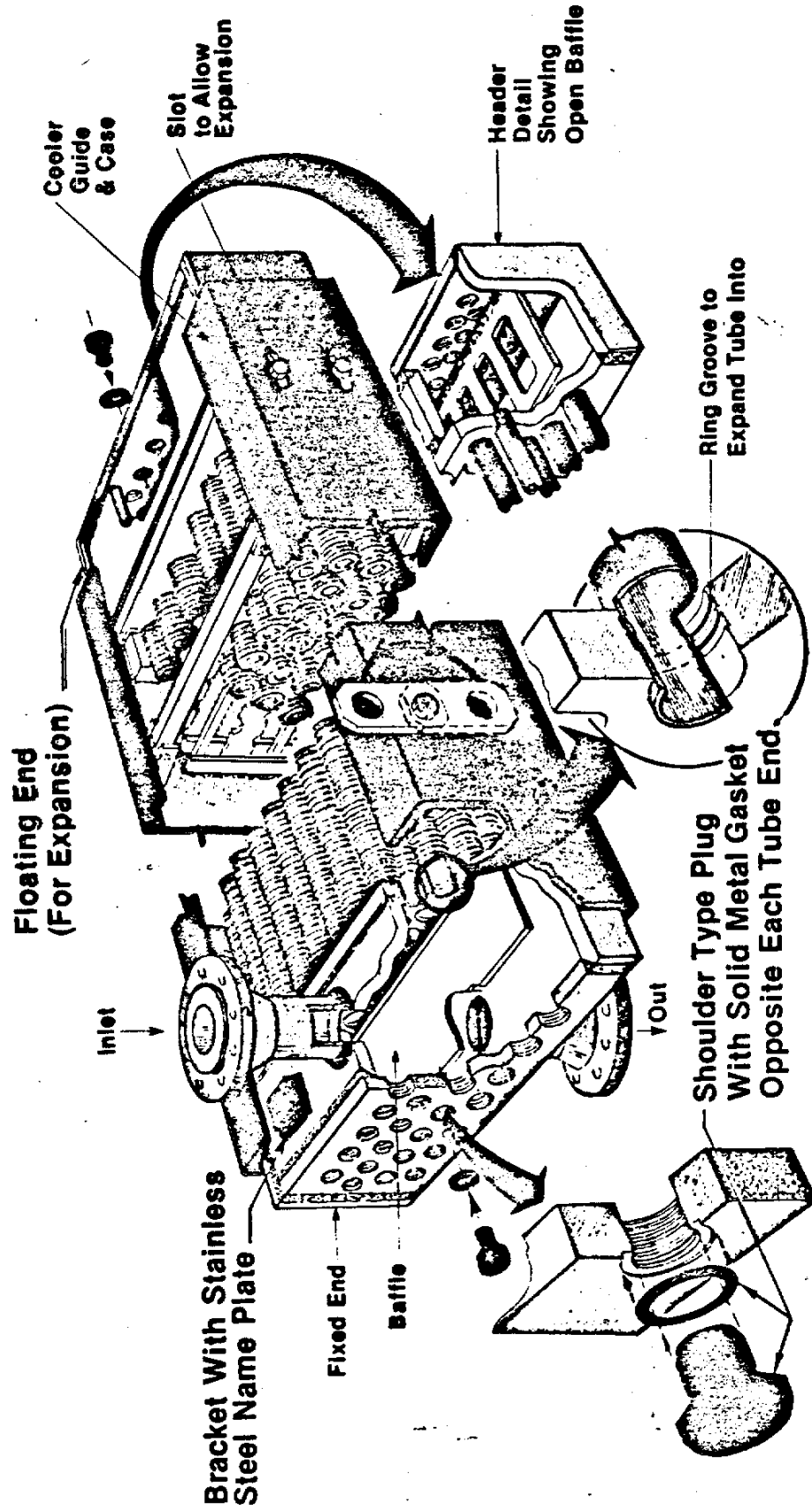
EXHIBIT 7G



Coverplate Header

The Coverplate Header design finds its greatest use in high fouling conditions where more complete access to the tubes and headers is required. This design is applicable to installations of moderate pressures. It is available in single or multiple pass designs, with drains for each pass.

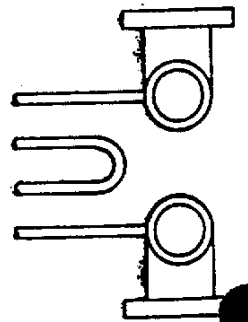
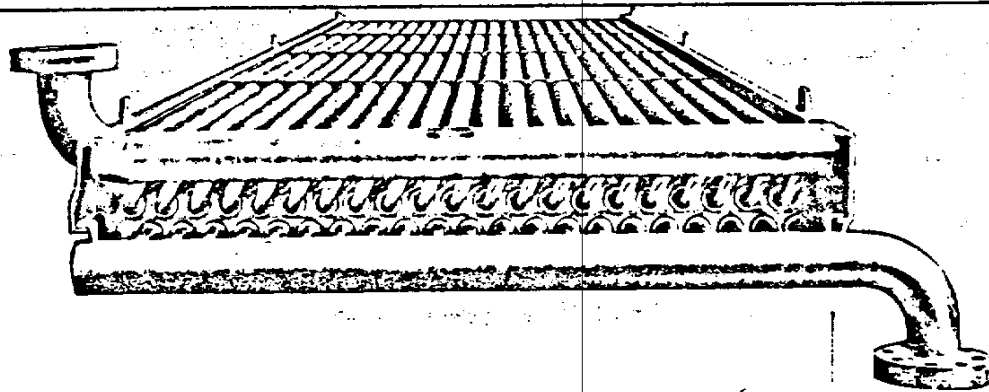
FIN FAN COOLERS



Tubes are expanded into tube sheet

32)

72

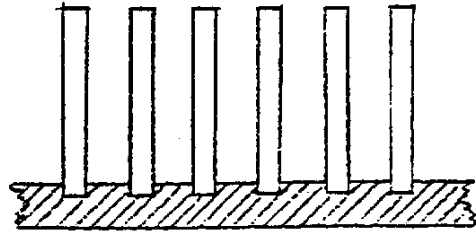


Manifold Header

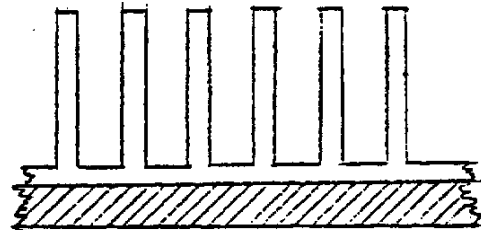
The Manifold Header is an all-welded design and is particularly adaptable to high pressure service. It is available in single or multiple pass designs that can be furnished in self draining and self venting types. The unit is available with welded or flanged connections.

EXHIBIT BSEVERAL TYPES OF FINNED TUBING USED IN AIR COOLED
HEAT EXCHANGERS

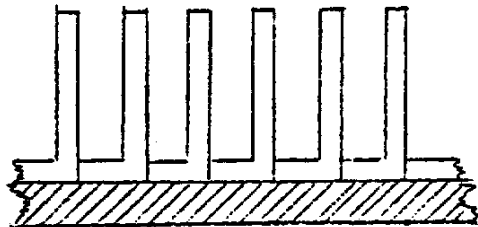
a) EMBEDDED

MAX. METAL TEMP ($\approx 700^{\circ}\text{F}$)

b) EXTRUDED

($\approx 500^{\circ}\text{F}$)

c) FOOTED

($\approx 400^{\circ}\text{F}$)

d) WRAPPED-ON

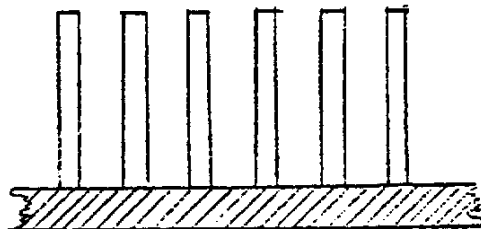
($\approx 250^{\circ}\text{F}$)

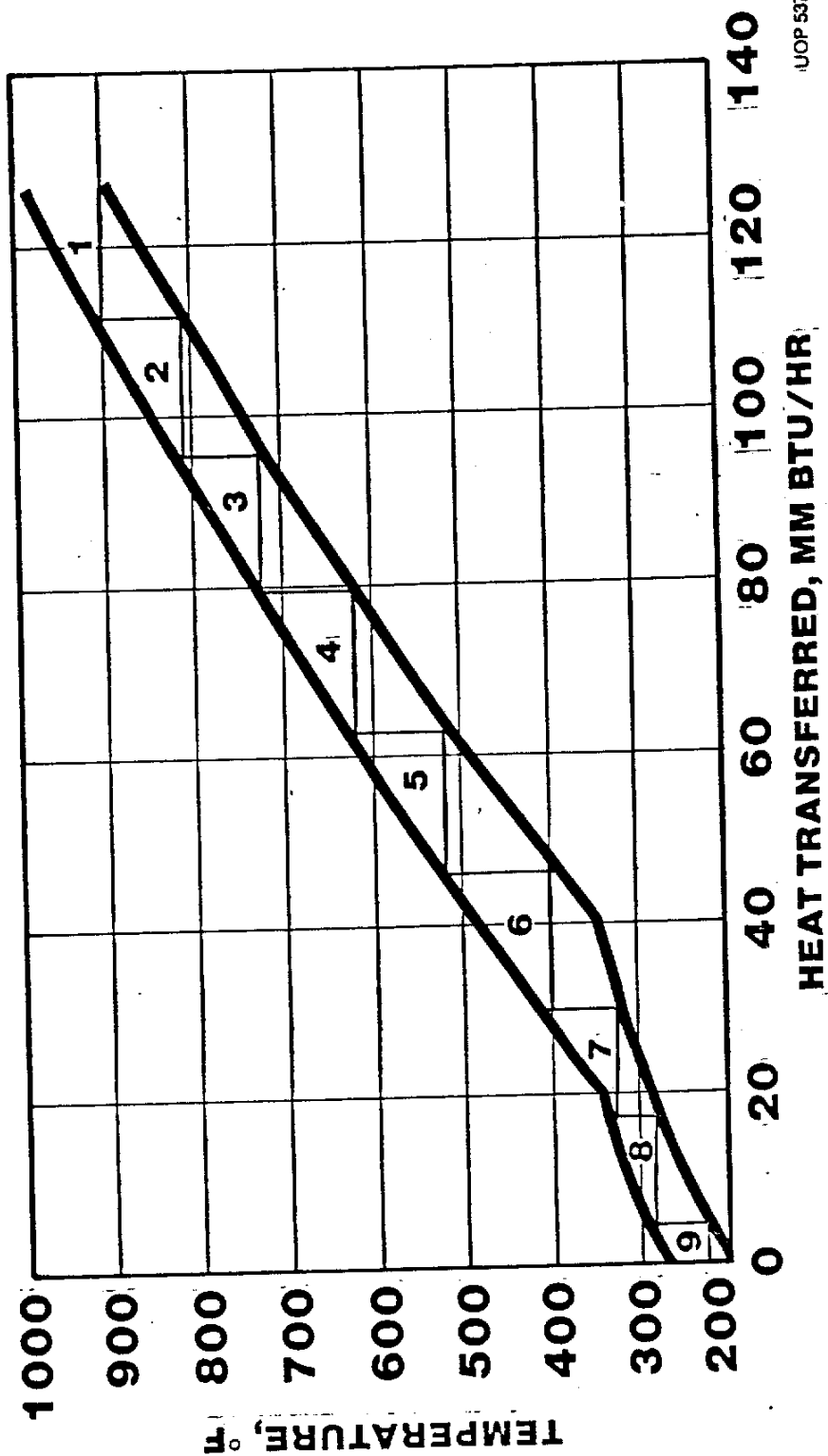
TABLE 1
EARLY UNITS

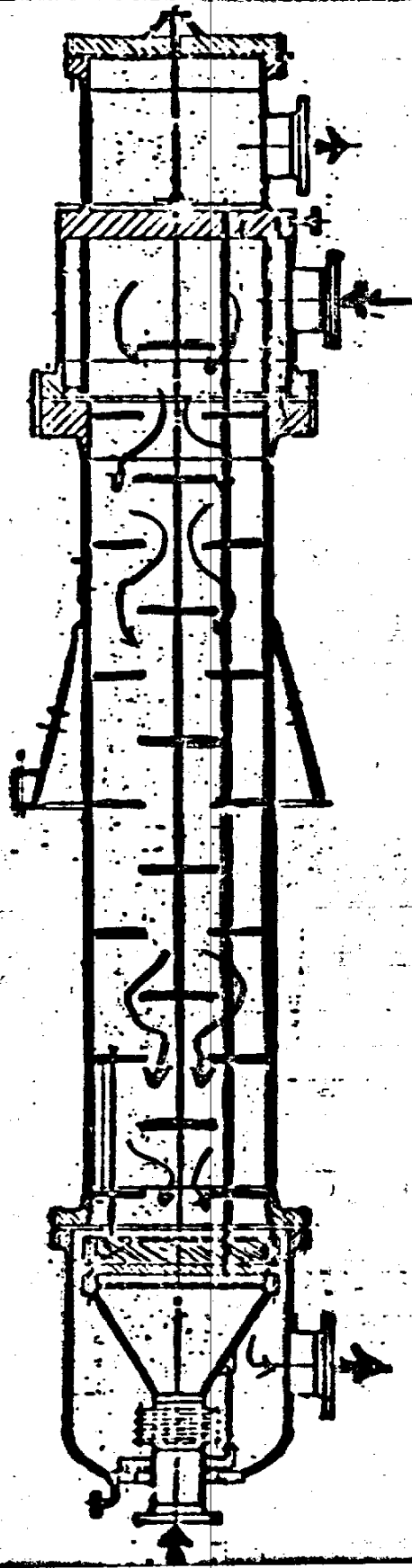
<u>DWG DATE</u>	<u>USER</u>	<u>SIZE</u>	<u>SURFACE</u>	<u>MFGR</u>
FEB 1966	SHELL OIL (WOODRIVER)	2 PAR 51" X 60'	21000 FT2 EA	IND FAB CO
MAY '69	NCRA (KANSAS)	40" X 40'	6017 FT2	WESTERN SUPPLY
OCT '69	COASTAL STATES (TEX)	2 PAR 56" X 20'	8500 FT2 EA	IND FAB CO

UOP 537B-4

D-28

HEAT RELEASE CURVES

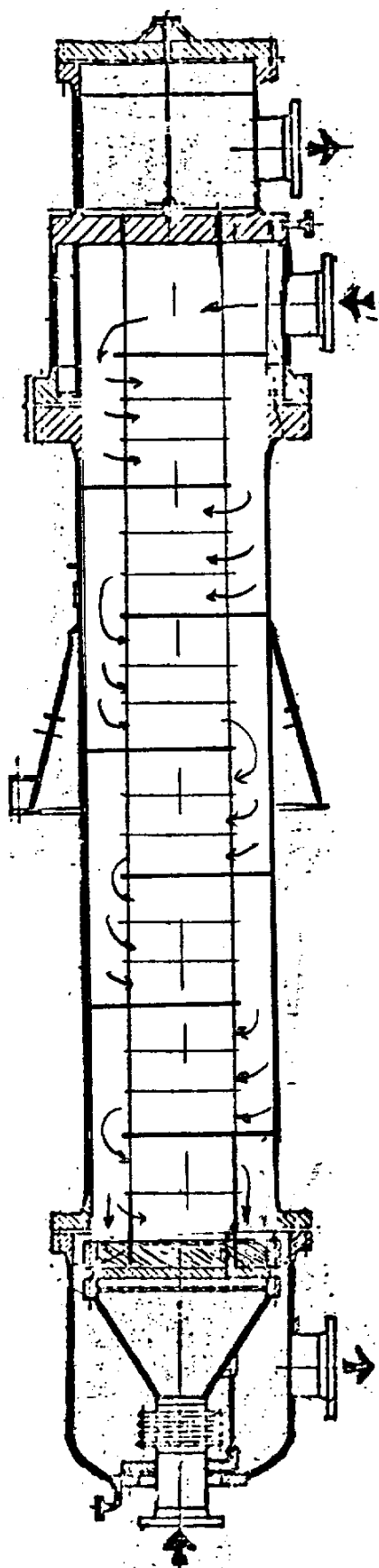


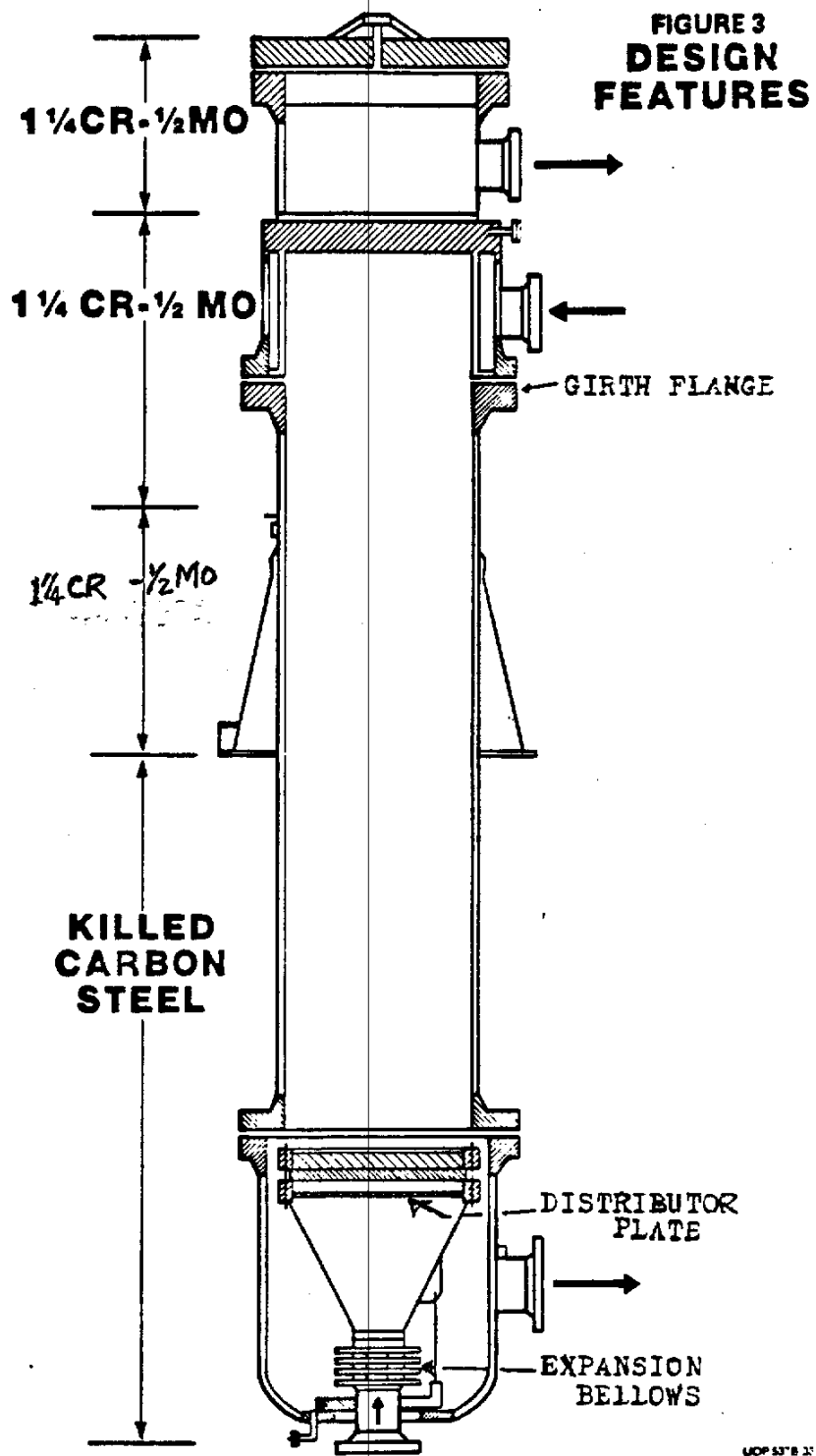


DESIGN
FEATURES

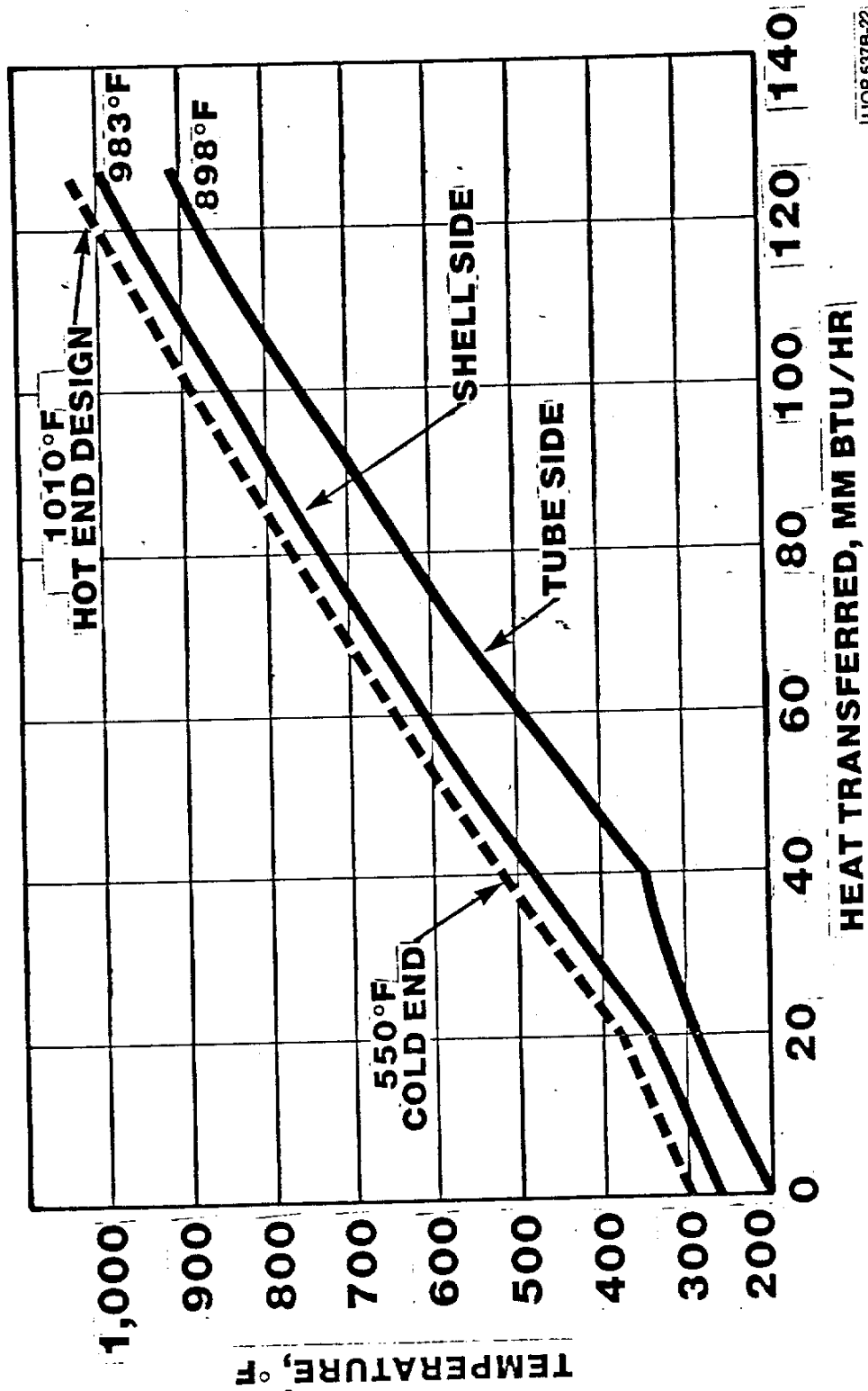
DOUBLE
SEGMENTAL

NTIW
DESIGN





OPERATING & MECH. DESIGN TEMPS



[UOP 537B-22]

GRADED METALS

MAX. OPER. TEMPS.

LATEST (FOR PARTIAL PRESSURE H₂ ≤ 200 PSI)

ABOVE 525°F

1 1/4 CR-1/2 MO OR 1 CR-1/2 MO

≤ 525°F

KILLED CARBON STEEL

UOP 537B-5

SUMMARY

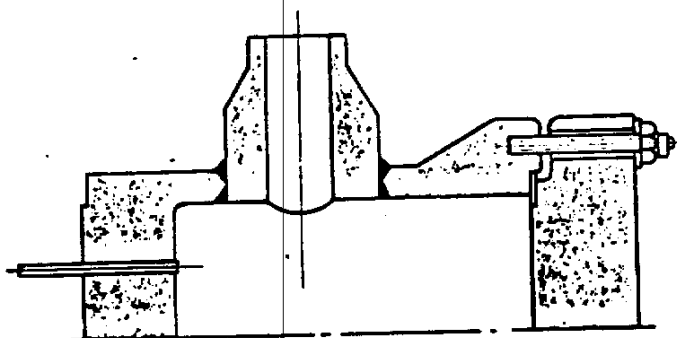
DESIGN FEATURES

- CONICAL INLET WITH DISTRIBUTOR PLATE
- INTERNAL EXPANSION BELLOW
- SHELL GIRTH FLANGE
- GRADED METALS IN SHELL
- TEMP. BREAK PT. FOR METALS
- DOUBLE SEGMENTAL OR NTIW DESIGN
- DERESONATING (LONG) BAFFLE
- CLOSE TOLERANCES FOR TUBE HOLES IN BAFFLES

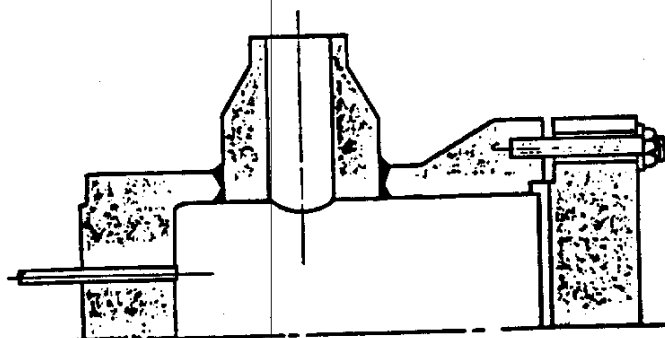
UOP 537B-7

Closures for High Pressure Service

FLAT GASKET CLOSURE



SEAL WELDED DIAPHRAGM CLOSURE



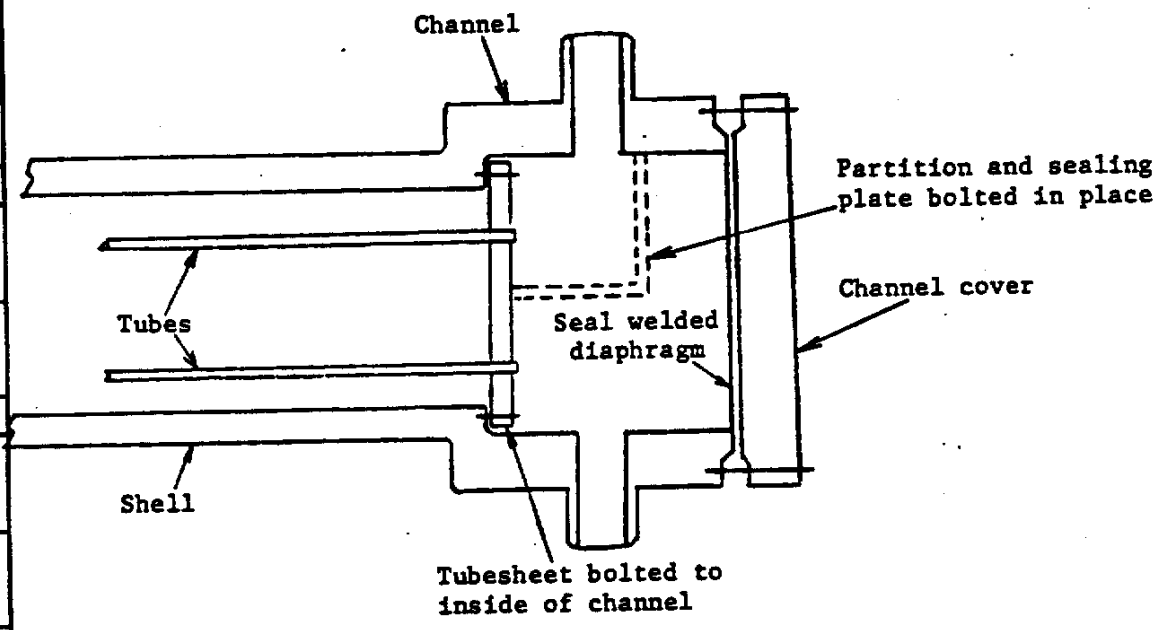
UOP
Process Division • UOP Inc.
20 UOP Plaza — Algonquin & Mt. Prospect Roads • Des Plaines, Illinois 60016 • U.S.A.

PROJECT SPECIFICATION	
NUMBER	REVISION
— 401 —	
SHEET	OF
BY	APP'D
DATE	

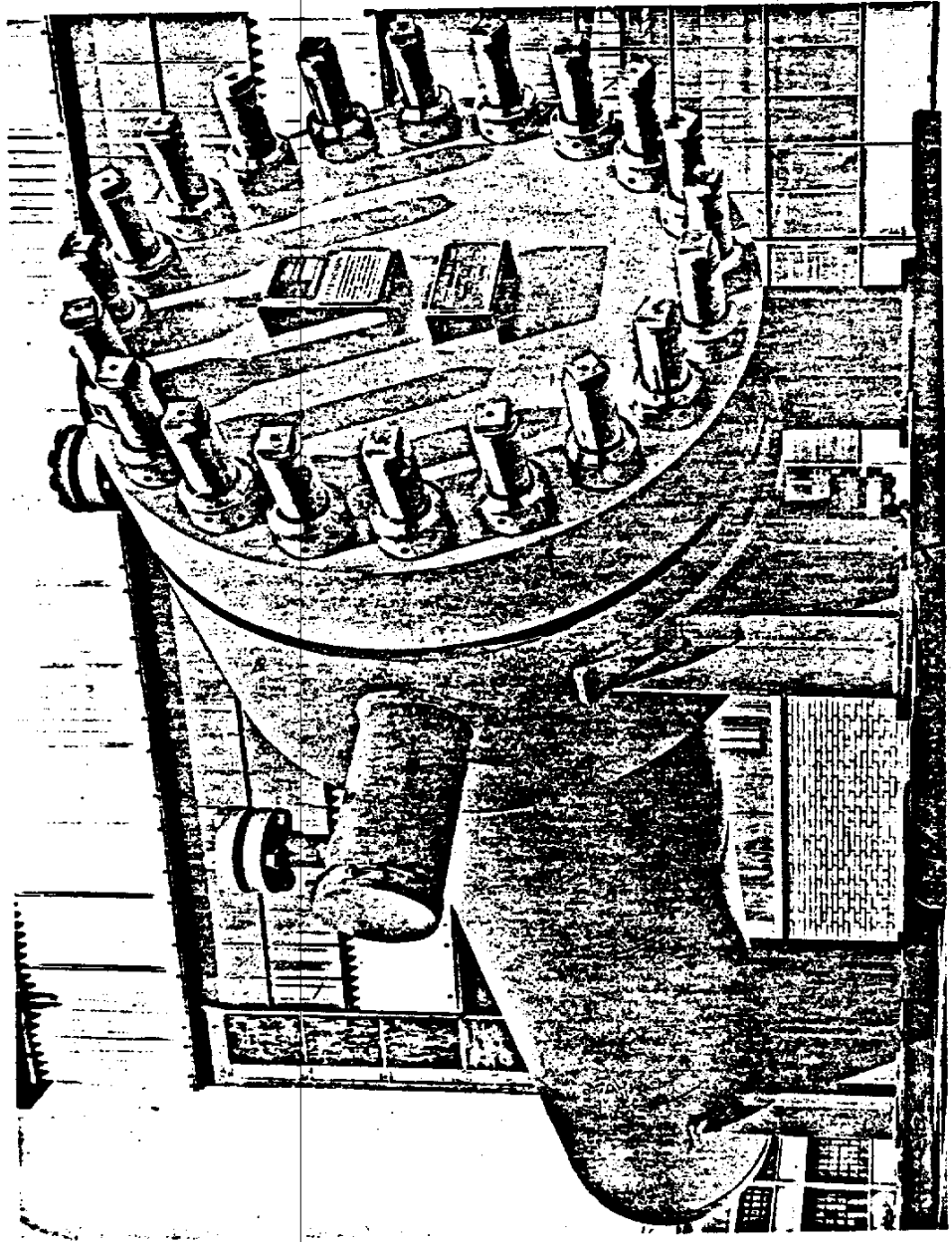
TUBULAR EXCHANGERS

SUGGESTED DESIGN: TYPE "MODIFIED DEU" EXCHANGERS

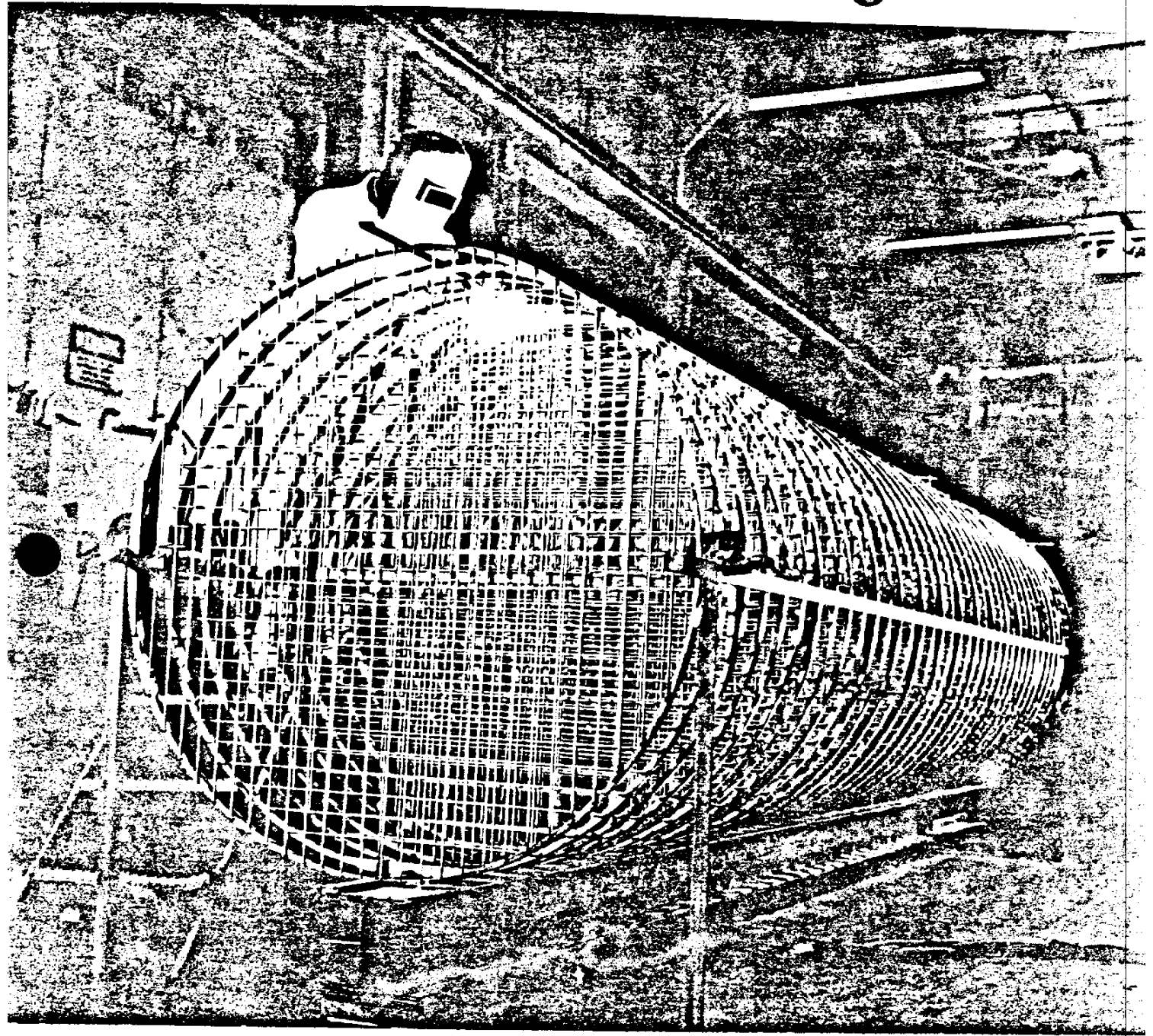
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high pressure and high temperature heat exchangers



RODbaffle Heat Exchangers



UOP Inc.

Algonquin & Mt. Prospect Roads • Box 5017 • Des Plaines, Illinois 60017-5017 • U.S.A.

WELDED PLATE TYPE EXCHANGERS "PACKINOX"

PROJECT SPECIFICATION

NUMBER

REVISION

- 416 -

SHEET

OF

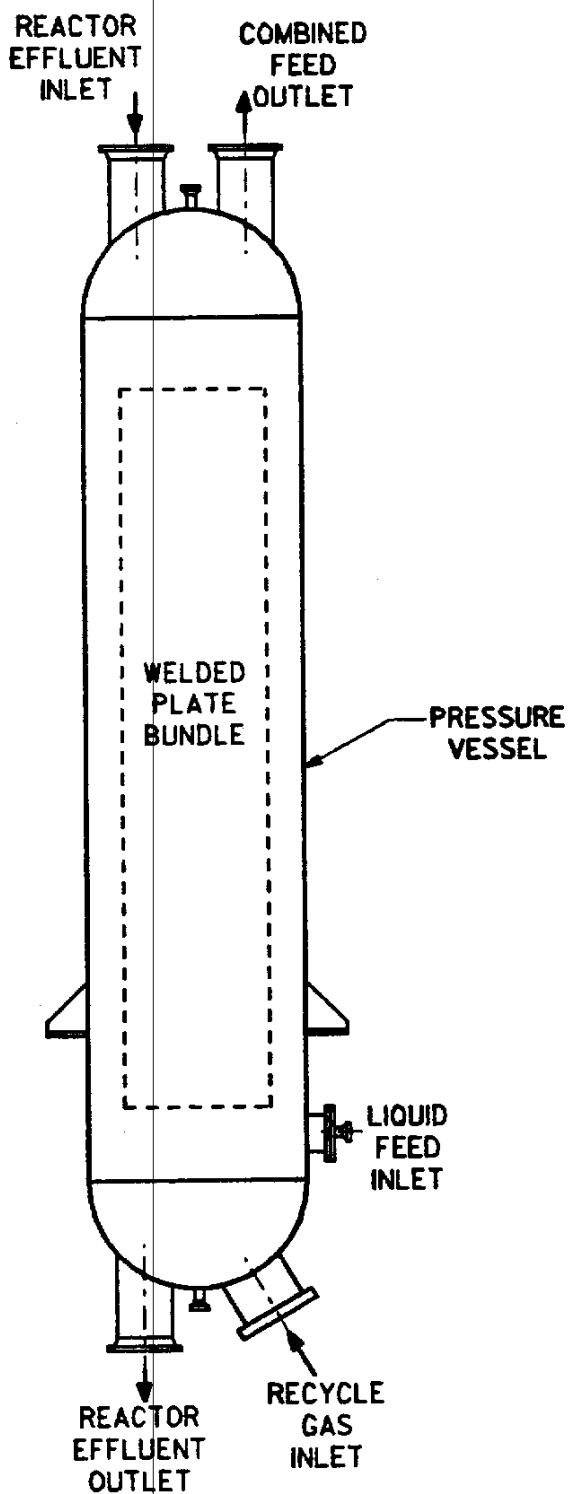
340

BY

APP'D

DATE

TYPICAL ARRANGEMENT FOR WELDED PLATE EXCHANGERS ITEM NO. _____

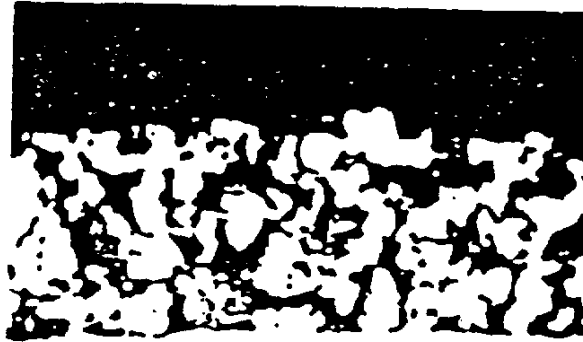


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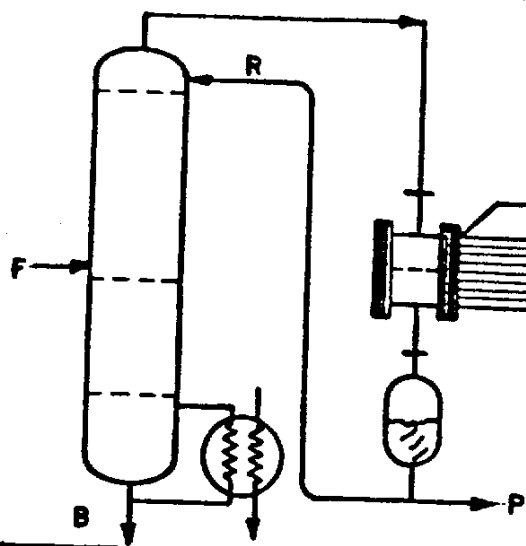
HIGH FLUX TUBING

- POROUS METAL FILM APPLIED TO I.D. OR O.D. OF TUBES.



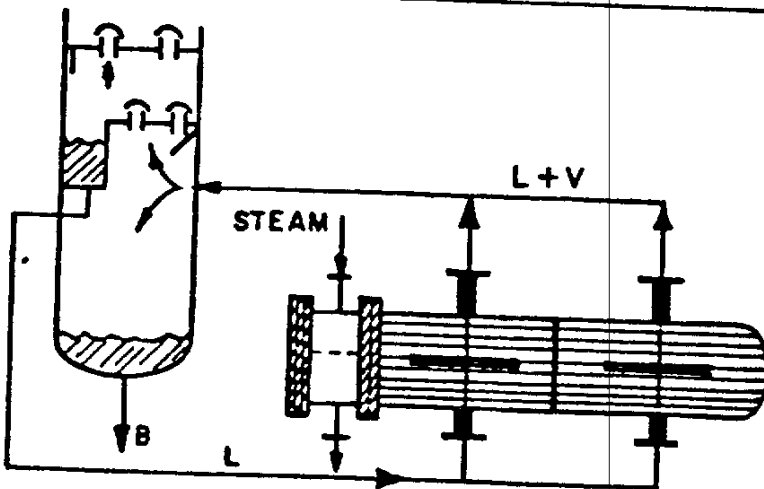
CROSS SECTION OF
HIGH FLUX SURFACE
(100 X)

- IMPROVES BOILING COEFFICIENT 10 X OR MORE. USED ONLY IN BOILING SERVICES.
- IMPROVES OVERALL TRANSFER RATE 2 - 4 TIMES
- AVAILABLE IN MOST ALLOYS. MAY BE COMBINED WITH FLUTED CONDENSING ENHANCEMENT.
- WIDELY APPLIED IN PETROCHEMICALS, REFINERY AND REFRIGERATION INDUSTRIES.
- RESISTANT TO FOULING - OVER 1,700 UNITS IN SERVICE.
- REDUCES CAPITAL OR OPERATING COST OF EXCHANGERS.



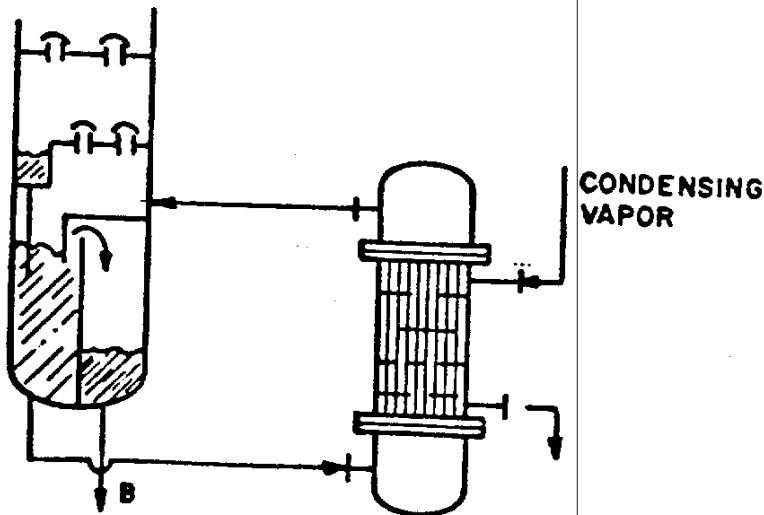
KETTLE TYPE
CHILLER-CONDENSER

O.D. COATED



ONCE THROUGH
HORIZONTAL H-TYPE
THERMOSIPHON REBOILER

O.D. COATED



CIRCULATING TYPE
VERTICAL THERMO-
SIPHON REBOILER

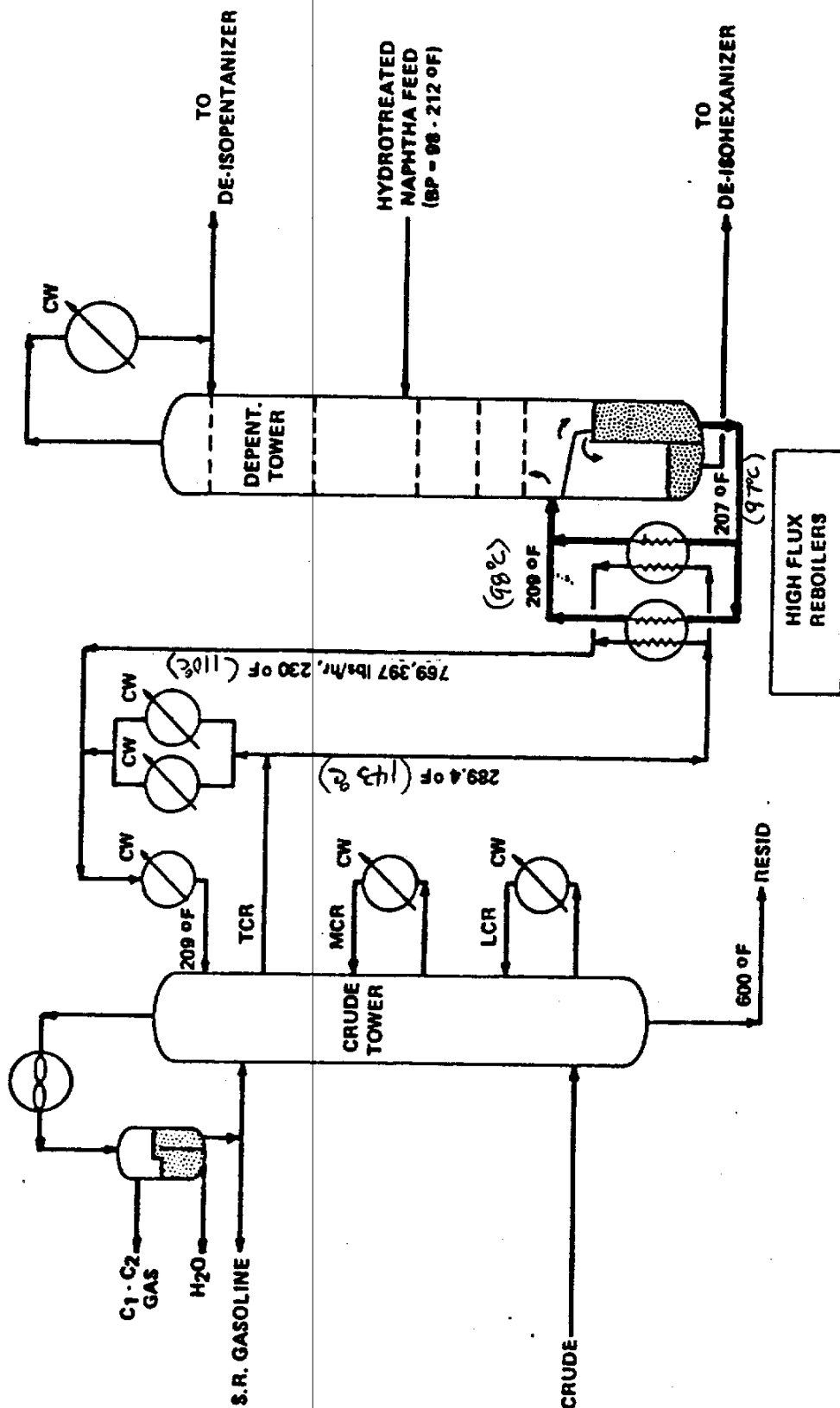
*I.D. COATED
O.D. FLUTED*

FIGURE 1 COMMON 'HIGH FLUX' HEAT EXCHANGER LAYOUTS

CASE HISTORY
DEPENTANIZER REBOILER RETROFIT

- REVAMP OF PETROLEUM REFINERY REBOILERS TO IMPROVE HEAT RECOVERY.
- REBOILERS DRIVEN BY TOP CIRCULATING REFLUX (TCR) STREAM COMING FROM CRUDE TOWER.
- ORIGINAL TCR FLOW TO REBOILERS - 500,000 KG/HR.
- CHANGES IN REFINERY CRUDE & OTHER MODIFICATIONS REDUCED TCR FLOW TO REBOILERS TO 350,000 KG/HR.
- HEAT DUTY OF REBOILERS DROPPED.
- DEBOTTLENECK OF REBOILERS ACHIEVED BY RETUBING EXISTING REBOILERS WITH HIGH FLUX TUBING USING SAME SHELL, HEADS AND PIPING.

FIGURE 2
SIMPLIFIED FLOWSHEET
CRUDE UNIT/DEPENTANIZER HEAT INTEGRATION



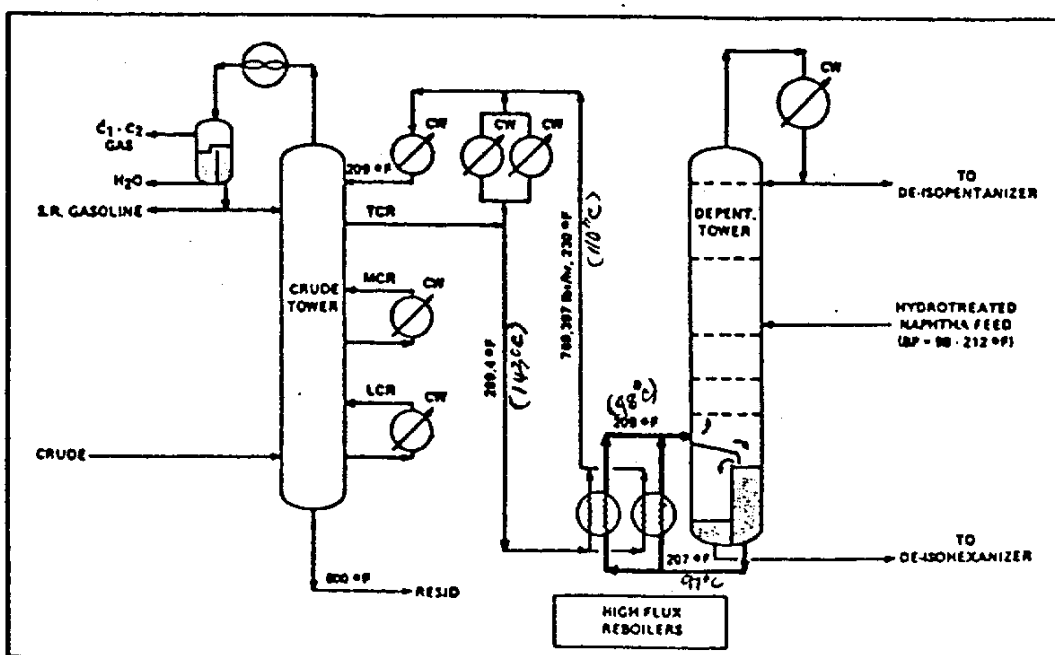
HIGH FLUX REPLACEMENT BUNDLES IN DEPENTANIZER REBOILERS ALLOWED A 41% INCREASE IN HEAT DUTY

TABLE 1: COMPARISON OF ORIGINAL AND REVAMPED DEPENTANIZER REBOILER OPERATION

ITEM	BARE TUBE REBOILERS		HIGH FLUX REBOILERS	
	E-702 A/B			
	As Designed (1969)	Actual Operation (1980)	As Designed (1981)	Max. Expected
Heating TCR Flow Kg/Hr	488,800	349,250	349,250	349,250
TCR Inlet Temp °C	149	143	143	143
TCR Outlet (°C)	126	120	110	106
Depent. Bottoms °C	103	97	97.3	97.3
Overall ΔT °C	33.1	33.2	25.8	22.3
Reboiler Surface, M ²	484	484	484	484
Overall U, K cal/Hr M ² C	<u>429</u>	<u>305</u>	<u>552</u>	<u>713</u>
Reboiler Duty MM K cal/Hr	6.875	<u>4.908</u> *	<u>6.875</u> *	7.701

**SIMPLIFIED
FLOW SHEET,
CD6 UNIT,**

*(40% INCREASE
IN DUTY
WITH
HI-FLUX)



HIGH FLUX REBOILER APPLICATION IN UOP PARA-XYLENE (PAREX) EXTRACTION SYSTEM

ITEM	UNIT A	UNIT B
LOCATION	FRANCE	USA
REBOILER SIZE	* 1-65"x240"	2-53"x240"
BOILING	XYLENES + EB	XYLENES + EB
CONDENSING	TOLUENE	TOLUENE
HEAT DUTY MM BTU/HR	45.0	71.8
EXCHANGER ORIENTATION	<u>VERTICAL</u>	<u>HORIZONTAL</u>
OVERALL U BTU/HR FT ² °F	210	185
TUBE TYPE	* < OD FLUTED (1 1/4") ID COATED	UD COATED
BENEFIT	FUEL SAVINGS	FUEL SAVINGS
STARTUP	2/84	1982

* IF BARE 1" OD TUBES
 U-VALUE APPROX: 85 BTU/HR FT²
 REQUIRE 2 EXCHANGERS EACH APPROX: 65"x240"
OR 4 " " " 48"x240"

SOLUTION TO PROBLEMS

PROBLEM NO. 1 SOLUTION

$$D = 0.782/12 = 0.0652 \text{ Ft.}$$

$$G = \frac{125,000 \times 2 \times 144}{128 \times 0.4803} = 585,572 \text{ lbs/hr-ft}^2$$

$$N_{re} = \left(\frac{DG}{\mu} \right)^{0.8} = \left(\frac{0.0652 \times 585,572}{2.42 \times 0.47} \right)^{0.8} = 4176$$

$$N_{pr} = \left(\frac{C_p \mu}{K} \right)^{0.4} = \left(\frac{0.51 \times 242 \times 0.47}{0.0793} \right)^{0.4} = 2.22$$

$$N_{nu} = \frac{h_t \times 0.0652}{0.0793} = 0.023 \times 4,176 \times 2.22$$

$$\begin{aligned} h_t &= 0.023 \times 4,176 \times 2.22 \left(\frac{0.0793}{0.0652} \right) \\ &= 259 \text{ (TUBE SIDE TRANSFER COEFFICIENT)} \end{aligned}$$

$$r_t^f = 1/h_t = 1/259 = 0.00386 \text{ (HR)}^2 \text{ FT}^2/\text{BTU}$$

(TUBE SIDE FILM RESISTANCE)

PROBLEM NO. 2 SOLUTION

From figure on page 146 of TEMA (Appendix 'C')

$$t_1 = 85$$

$$t_2 = 105$$

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$

$$= \frac{105 - 85}{144 - 85}$$

$$= 0.34$$

$$T_1 = 144$$

$$T_2 = 125$$

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$= \frac{144 - 125}{105 - 85}$$

$$= 0.95$$

OR

$$\begin{array}{l} \Delta T \\ 144 \rightarrow 125 = 19 = R \\ 105 \leftarrow 85 = \frac{20}{59} = P \end{array}$$

$$R = 0.95$$

$$P = 0.34$$

$$\therefore F = 0.955$$

From figure, $F = \underline{0.955}$

(MTD Correction factor)

PROBLEM NO. 3 SOLUTION

(From TEMA Charts: Appendix 'C')

$$t_1 = 294$$

$$t_2 = 210$$

$$T_1 = 178$$

$$T_2 = 242$$

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$P = \frac{210 - 294}{178 - 294}$$

$$R = \frac{178 - 242}{210 - 294}$$

$$= 0.72$$

$$= 0.76$$

F for one shell = < 0.5; no good.

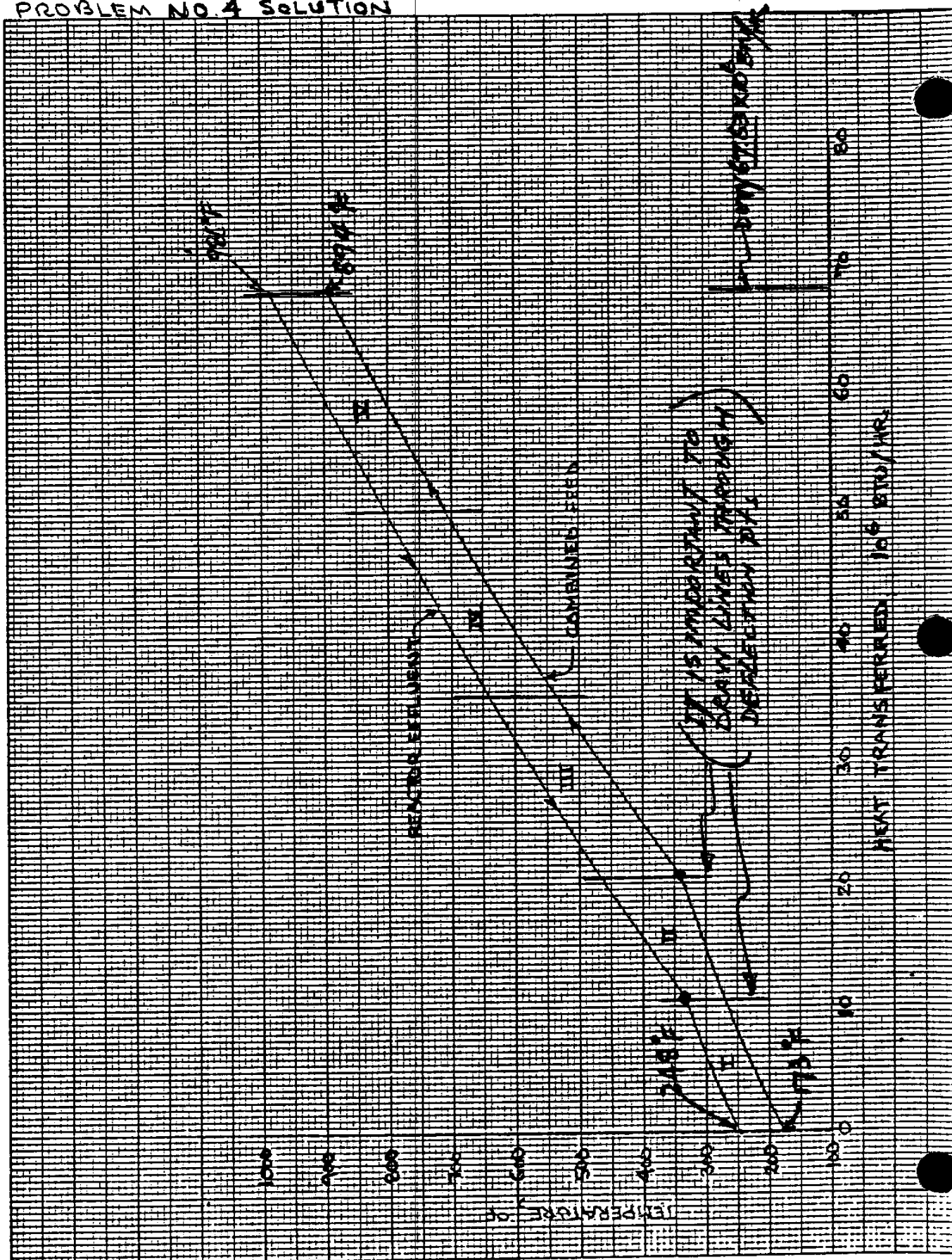
F for two shells in series = 0.87

Therefore, two shells in series are required.

$$\text{OR : } \begin{array}{l} 294 \xrightarrow{\Delta T} 210 = \frac{84}{116} = 1.31 = R \\ 242 \xleftarrow{\quad} 178 = \frac{64}{116} = 0.55 = P \end{array}$$

$F \approx 0.865$
(2 in SERIES)
REQUIRED

PROBLEM NO. 4 SOLUTION



PROBLEM NO. 4 SOLUTION

$$\begin{array}{lcl} \text{ZONE I} & \begin{array}{r} 330 \rightarrow 248 \\ 270 \leftarrow 173 \\ \hline 60 \quad 75 \end{array} & = 67.2; \end{array}$$

$$(Q/MTD)_I = \frac{10,500,000}{67.2}$$

$$= 156,205$$

$$\begin{array}{lcl} \text{ZONE II} & \begin{array}{r} 465 \rightarrow 330 \\ 340 \leftarrow 270 \\ \hline 125 \quad 60 \end{array} & = 88.6; \end{array}$$

$$(Q/MTD)_{II} = \frac{10,000,000}{88.6}$$

$$= 112,920$$

$$\begin{array}{lcl} \text{ZONE III} & \begin{array}{r} 640 \rightarrow 465 \\ 535 \leftarrow 340 \\ \hline 105 \quad 125 \end{array} & = 114.7; \end{array}$$

$$(Q/MTD)_{III} = \frac{14,500,000}{114.7}$$

$$= 126,411$$

$$\begin{array}{lcl} \text{ZONE IV} & \begin{array}{r} 804 \rightarrow 640 \\ 710 \leftarrow 535 \\ \hline 94 \quad 105 \end{array} & = 99.4; \end{array}$$

$$(Q/MTD)_{IV} = \frac{15,000,000}{99.4}$$

$$= 150,910$$

$$\begin{array}{lcl} \text{ZONE V} & \begin{array}{r} 986 \rightarrow 804 \\ 894 \leftarrow 710 \\ \hline 92 \quad 94 \end{array} & = 93.0; \end{array}$$

$$(Q/MTD)_V = \frac{17,630,000}{93}$$

$$= 189,570$$

$$\Sigma(Q/MTD) = 736,020$$

$$\text{Uncorrected Weighted MTD} = \frac{67,630,000}{736,020}$$

$$= 91.9^\circ\text{F}$$

$$= \frac{Q_{\text{TOTAL}}}{\Sigma \frac{Q}{\Delta T_m}}$$

Uncorrected Straight Line MTD:

$$\begin{array}{lcl} & \begin{array}{r} 986 \rightarrow 248 \\ 894 \leftarrow 173 \\ \hline 92 \quad 75 \end{array} & = 83.2 \text{ (Versus } 91.9^\circ\text{F for WTD, MTD)} \end{array}$$

PROBLEM NO. 5 SOLUTION

$$\begin{array}{lcl} R_1 & = & 1/U_1 \\ & = & 1/50 \\ R_1 & = & 0.02 \end{array} \quad \left| \quad \begin{array}{l} R_1 = r_{s1}^f + r_s^d + r_m + r_t^d + r_{t1}^f \\ 0.02 = r_{s1}^f + .001 + .0004 + .002(1.284) + r_{t1}^f \\ r_{s1}^f + r_{t1}^f = 0.016 \end{array} \right.$$

If we assume $r_{s1}^f = r_{t1}^f = .008$

and $G_2 = G_1/2$

then $r_{s2}^f = \left(\frac{2G_1}{G_1} \right)^{0.6} r_{s1}^f$

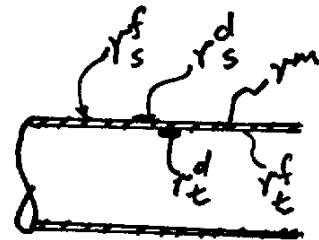
$$= (2)^{0.6} (.008)$$

$$= 0.0122 \text{ --- (A)}$$

and $r_{t2}^f = \left(\frac{2G_1}{G_1} \right)^{0.8} r_{s1}^f$

$$= (2)^{.8} (0.008)$$

$$= 0.0139 \text{ --- (B)}$$



Summing up Resistances:

$$r_{s2}^f = 0.0122 \text{ --- (A)}$$

$$r_s^d = 0.0010$$

$$r_m = 0.0004$$

$$r_t^d = 0.0026$$

} SAME AS BEFORE

$$r_{t2}^f = 0.0139 \text{ --- (B)}$$

$$R_2 = 0.0301$$

$$U_2 = 1/0.0301$$

THEREFORE: $U_2 = \underline{\underline{33}}$