Report CE-865

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Metallurgical Project

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TECHNOLOGICAL RESEARCH - ENGINEERING DEVELOPMENT P-9

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THE RESULTS OF A TRIP TO HEAT EXCHANGER MANUFACTURERS

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August 14, 1943

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I. INTRODUCTION

The Andale Company, the Schutte and Koerting Company and the American Heat Reclaiming Corp. were approached with the problem of constructing a leak-tight stainless steel liquid-liquid heat exchanger having as small a liquid hold-up as possible. The operating conditions considered were:

Shell side - soft water entering at 59° F (15° C).

Tube side - a liquid with physical properties very much the same as water, specific gravity about 1.35, and containing solid particles 10-100 microns in diameter, entering at 230° F (110° C) and leaving at 122° F (50° C).

Velocities - as high as possible to increase the rate of heat transfer.

Working Pressure - 50-75 Lbs/sq.in.

This memorandum is a summary of the facts and opinions obtained from the three manufacturers in regard to constructing and operating an exchanger that would meet the given conditions.

SECRET

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II. SUMMARY

A single pass tubular heat exchanger with straight 1/4 inch (18 B.W.G.) tubes rolled and expanded into tube sheets can be constructed and operated under the conditions predicated by a homogeneous pile. The following dimensional and operating limitations are recommended by the manufacturers:

Tube spacing	3/8 inch centers, triangular array
Tube sheets	one fixed and one floating
Baffles	segmental
Maximum tube length	10 feet
Maximum velocity through tubes	10 ft./sec.
Maximum velocity through shell	6-7 ft./sec.

The manufacturers' opinion varied considerably regarding the allowable unsupported tube length; one recommended 8 inches, the other 18 inches.

The use of single grooved tube sheets with expanded tubes is the most satisfactory method of preventing P-9 leakage.

Preliminary calculations indicate that the amount of cooling water required will result in excessive shell side velocities; double water exits and entrances may alleviate this condition.

The best overall heat transfer coefficient expected for a tubular heat exchanger is 800 ± 50 B.T.U./(hr)(sq.ft)(°F.)

Stainless steel spiral heat exchangers can be made with a 3/8 inch annulus and 54 inches long. Preliminary calculations indicate allowable velocities are limited by the high pressure drop inherent in this type of exchanger.

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III. DISCUSSION

Part I - Tubular Heat Exchangers

A. Tube Size and Spacing

Both the Andale Company and the Schutte & Koerting Company agreed that:

1.) 1/4-inch stainless steel tubes can be satisfactorily put into tube sheets on 0.375 inch centers.

2.) 3/16-inch tubes are below the practical limit for shop fabrication for tubular exchangers, although they can be used in lengths less than 5 feet long and spaced not less than 0.375 inches apart.

3.) The maximum allowable length of 1/4-inch stainless steel tubes is 10 ft. This limit is predicated by the bowing and twisting of long tubes.

4.) For protection against erosion, 18 B.W.G. is the lower limit for tube thickness - 14 B.W.G. being preferred. (Erosion at the ends where cold working has occurred during the expanding process will be a critical factor in determining the life of the exchanger.)

5.) Further annealing of the stainless steel tube ends is not recommended because of the distortion and loss of strength likely to result.

B. Tube Sheets and Tube Expanding

Both Schutte & Koerting and Andale agree that:

1.) Because of the high stresses developed at the tube ends with the given temperature differences, fixed tube sheets are not suitable. However, this type sheet could be used with bowed tubes or an expanding shell. (See Figure LZ-D70 - Details C and E).

2.) One fixed and one floating tube sheet will be satisfactory.

3.) 1/4 inch tubes "drifted" and expanded into grooved holes in a single rather than a double tube sheet will give tight connections.

4.) The tubes and tube sheets should be made of the same material.

5.) Using extended tube sheets independently bolted to the head and shell (see Fig. G-4229) facilitates the removal of the tube bundle, should this operation be necessary.

The Andale Company recommends the following stainless steel tube sheet thicknesses for 125 lbs. working pressure:

Shell Diameter, in.	Tube Sheet Thickness, in.	
	Fixed Sheet	Floating Sheet
8 1/4	1	1 1/4
10 1/8	1 1/8	1 1/2
15 5/8	1 1/4	1 1/2
20	1 3/8	1 1/2

The above figures include factors of safety for corrosion, erosion and strength. The S. & K. Co. follow the specifications given in Tubular Exchanger Manufacturers Association. The floating tube sheet is thicker than the fixed tube sheet because the former contains the stuffing box packing gland. Preliminary calculations indicate that decreasing the tube sheet thickness to reduce hold-up is impractical since a 1/4 inch reduction in thickness decreases the hold-up in 1400 tubes 8 feet long by only 0.3%.

The Andale Company prefers the use of forged steel tube sheets to cold rolled ones for the following reasons:

1.) The forged steel tube sheets have greater tensile strength.

2.) Tubes can be worked without working the tube sheet.

3.) No cracking of the ligaments is likely to occur.

The Schutte & Koerting Company prefers the use of cold rolled tube sheets.

Welding the tubes to the tube sheet is impossible with small tubes on close centers because there is neither enough space between tubes nor enough metal in the tube walls.

The Andale Company recommends the use of a false or blank tube sheet at the entrance of the tubes to help decrease erosion at the tube ends. (See Detail D, Figure IZ-D70). This sheet would not completely prevent erosion but would lessen it considerably.

C. Shell

The Andale Company recommends the use of cast iron shells when using water. The shell should be bored and machined on the inside for close baffle to shell fits and minimum tube side pressure drop. A 1/32 inch baffle to shell clearance is recommended. The use of impingement plates inside the shell at the inlet water nozzle is not required unless high velocities and dirty water are used. The use of a continuous shell venting system is essential. (See Figure IZ-D70 - Detail A).

The Schutte & Koerting Company have no recommendations as to the material of construction for the shell.

D. Baffles - Baffle Spacing

The Andale Company recommends the use of segmental baffles instead of orifice baffles due to the low velocities obtained with the latter. To prevent bi-metal couple action between baffles and tubes, both should be made of the same material. It is possible to cut baffle-tube clearances down to a few thousanths of an inch. The limiting factors are allowable cross leakage, tube diameter irregulatiries, and distortions produced in tubes when expanding into tube sheets.

The Schutte & Koerting Company recommends a 1/64 inch baffle-tube clearance in accordance with T.E.M.A. * standards.

The Andale Company prefers a maximum baffle spacing of 4 inches and an unsupported tube length of 9 inches, whereas the Schutte & Koerting Company believes as much as 18 inches of unsupported tube can be tolerated. Minimum baffle spacing depends entirely upon allowable pressure drops. Excessive baffle spacing not only allows tube "bowing" but creates "shaded" or dead spaces of low liquid velocities.

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*Tubular Exchanger Manufacturers Association

Baffle thickness depends upon corrosion rates, strength and shell diameter. The Schutte & Koerting Company prefers T.E.M.A. standards, whereas the Andale Company recommends the following for stainless steel baffles:

Shell Diameter - in.	Baffle Thickness - in.
8	1/16
15	3/32
20	1/8

Andale often recommends Phenolite composition baffles and for each of the above diameters, a 1/4 inch thickness is desirable. Phenolite is a phenol-formaldehyde resin that is light, inexpensive, has a low coefficient of expansion, is corrosion resistant and does not swell appreciably.

To prevent by-passing the water around the tube bundle into shell-bundle free space, the Andale Company and Schutte & Koerting recommend the use of circular tie rods rather than strip segments. Andale recommends Phenolite rods. Rods are more readily attached to the bundles and facilitate removal of the bundle. Figure G-4229 shows the detail and arrangement of this type of connection.

The Andale Company recommends a baffle design such that the water velocity through the cutdown is 75-80% of the velocity between the baffles. This design appreciably decreases the shellside pressure drop without noticeably affecting the shell side coefficient. Preliminary calculations by Andale engineers show that the shell side pressure drop is quite sensitive to baffle design.

E. Header Design

Preliminary calculations indicate that 75-80% of the liquid hold-up will occur in the headers. Consequently considerable time was given to header design. There are no definite conclusions, as numerous calculations must be made before a header design with minimum hold-up can be presented. It was agreed by all manufacturers that the limiting factor is the distribution of the fluid through the tubes. The lower the velocity of the fluid in the head just before entering the tubes, the better the distribution will be.

The header design is independent of the type of tube sheet; the inside header diameter is slightly larger than the diameter of the outer tube circle. (See Figure G-4229). Both Schutte & Koerting and Andale believe it possible to bring the inside wall of the header to within 3/4 inches of the tube circle at the fixed tube sheet. At the floating head end, enough room must be left for the tongue and groove joint (See Figure LZD-72)which insures against P-9 leakage.

The Andale Company recommends a side entering nozzle (see G-4229) with a flat cover plate on the header. They believe that curved surface headers, venturi nozzles at the top of the header, and impingement plates are not practical. Thus the header depth becomes a function of the nozzle diameter, the welding procedures used, and the thickness of the header flanges. Regardless of design, the headers must be stainless steel and the header flanges designed according to American Standards Association (125 lb. rating) standards.

The Schutte & Koerting Company believes nozzles on the side of curved headers can be used. They also believe a nozzle entering at the top and a perforated impingement plate can be used. In either case, a minimum header depth of 5 inches is necessary.

The expansion and contraction losses constitute 95% of the total loss of pressure in the headers.

F. <u>Design Coefficients - Velocities</u>

The Andale Company and the Schutte & Koerting Company agree on the following points:

1.) The maximum allowable cooling water velocity across 1/4 inch tube bundles is 6-7 ft./sec. This figure is not based upon pressure drop limitations but upon the fact that considerable vibration occurs in the bundle when higher velocities are encountered. Preliminary calculations indicate a water velocity of 15-20 ft./sec. for a 59-167° F (15-75° C) cooling water rise. It may be necessary to do one of three things to decrease this velocity:

1. Use more exchangers.
2. Use a much higher outlet water temperature.
3. Use a double water-cooled shell as shown in Figure LZ-D 71. This would divide the bundle into two parts with two cold water inlets and in effect be two small exchangers.

2.) The maximum tube velocity for this particular application is essentially a function of erosion rates. A maximum velocity of 10 ft./sec. is recommended. However, Schutte & Koerting believed a higher velocity might be realized if testing showed that severe erosion at the tube entrance did not occur and excessive pressure drops were not encountered in 1/4 inch tubes.

3.) Only with extreme operating conditions, i.e., high velocities, no scaling, etc., will it be possible to realize overall coefficients of 800-900 B.T.U./(hr.)(sq.ft.)(°F.)

The Andale Company gives the following table of overall coefficients for 1/4 inch tubes based on many shop tests with water:

Velocity (inside) ft./sec.	(cm/sec)	Velocity (outside) ft./sec.	(cm/sec)	U_o (based on outside area) B.T.U./(hr.)(sq.ft.)(°F.)
6	(180)	7	(200)	680
6	(180)	6	(180)	640
10	(300)	7	(200)	790
10	(300)	6	(180)	740

The following shell side coefficients are given:

ft./sec. (mean vel.)	B.T.U./(hr.)(sq.ft.)(°F.)
15 (360 cm/sec)	2700
7 (200)	1550 (max.)
6 (180)	1200
3.4 (50)	970

G. Multi-Pass and Bent Tube Exchangers

The use of two-pass exchangers and bent tube exchangers was not recommended by either Andale or Schutte & Koerting for the following reasons:

1.) The tubes contract and expand unevenly.

2.) The tube size and spacing required for this case are too small for shop practice.

3.) High velocities in bent tubes make them pull away from tube sheets.

4.) Small stainless steel tubes are difficult to bend without developing cracks, high wall stresses and distortion.

5.) Tube bends are more readily eroded.

6.) No appreciable saving in hold-up can be realized in the headers.

H. Pumps in Headers

The possibility of incorporating a pump in the exchanger headers to decrease hold-up was briefly discussed with Andale and Schutte & Koerting engineers. They believed that it was quite possible for the header to serve as the pump impeller casing but pointed out that it would require the combined efforts of pump and exchanger manufacturers to present a satisfactory design.

The Andale Company felt reluctant to share the responsibility of the combined unit operation and indicated that considerable development would be required.

Representatives of both companies were of the opinion that the problem of water distribution in the tubes would again be the controlling factor in the design.

DISCUSSION

Part II - Spiral Exchangers

Discussion with Mr. E. Creutz, of the American Heat Reclaiming Corporation indicates that the use of spiral exchangers may be satisfactory. Preliminary calculations, although quite incomplete, indicate that a much higher ratio of square feet of surface to cubic feet of liquid hold-up may be realized with this type exchanger than with conventional types.

Spiral exchangers can be made of any material, and many stainless steel exchangers are in service.

Present manufacturing methods prevent making exchangers more than 54 inches long. (See Figures LZA 69, and LZA 68, and American Heat Reclaiming Bulletin in Appendix.)

The problem of preventing P-9 loss can be satisfactorily handled by the use of neoprene gaskets and welded sealing rods as shown in Figure LZA 69, and Detail 5.

The maximum overall transfer coefficients that can be realized without developing excessive pressure drops with this type of exchanger is of the order of 750 B.T.U./(hr.)(sq.ft.)(°F.)

The practical lower limit to annular spacing is 3/8 inches. A typical calculation for an exchanger with a 3/8 inch annular spacing, a mean temperature difference of 65° F., and 1190 sq. ft. of heat transfer surface shows:

A. Velocity inside = 9.55 ft./sec. = 725 gal./min.

Pressure drop inside = 40 Lbs/sq.in.

B. Velocity outside = 13.0 ft/sec. = 815 gal./min.

Pressure drop outside = 55-60 Lbs./sq.in.

C. Exchanger diameter = 47 inches

D. Tube side heat transfer coefficient = 2220 B.T.U/(hr.)(sq.ft.)(°F.)

Shell side heat transfer coefficient = 1800 " " "

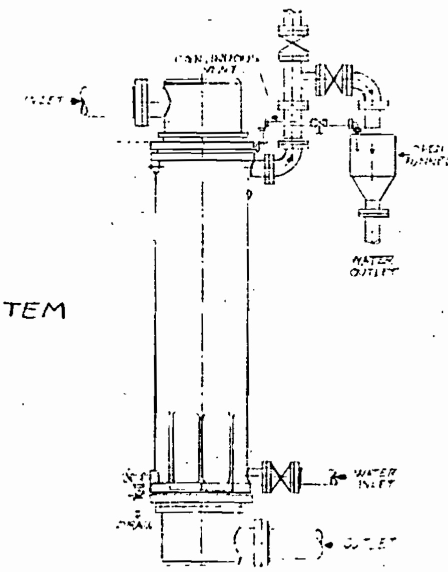
E. Overall heat transfer coefficient = 600 " " "

F. Metal wall thickness = 12 U.S.G. - - - - 0.1072 inches

G. Cubic feet hold-up in exchangers including inlet and outlet ports and headers = 23.0 cu. ft.

The use of 1/4 inch spacing cuts the hold-up to 12.4 cu. ft. in the spirals and 2.5 cu.ft. in the headers and ports.

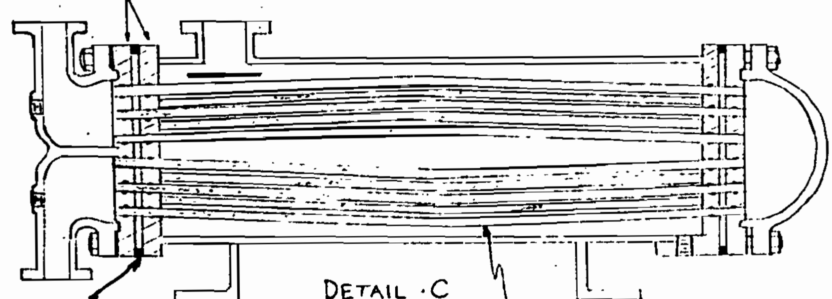
Another type of spiral heat exchanger called the Votator is made by the Girdler Corporation. Inquiries are being made of them by mail.



SHELL VENTING SYSTEM

DETAIL A

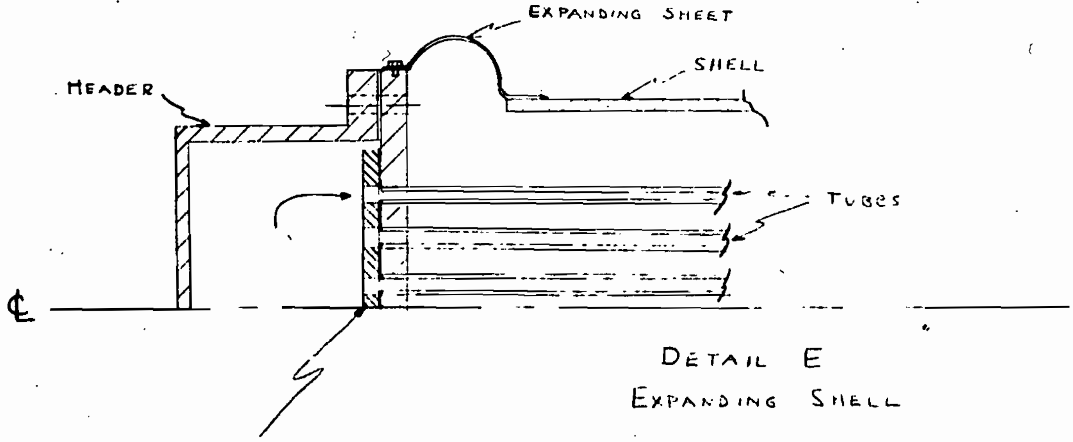
DOUBLE TUBE SHEET
DETAIL B



LEAKAGE DRAIN

DETAIL C

BOWED TUBES
FIXED TUBE SHEETS



DETAIL E
EXPANDING SHELL

BLANK TUBE SHEET
DETAIL D

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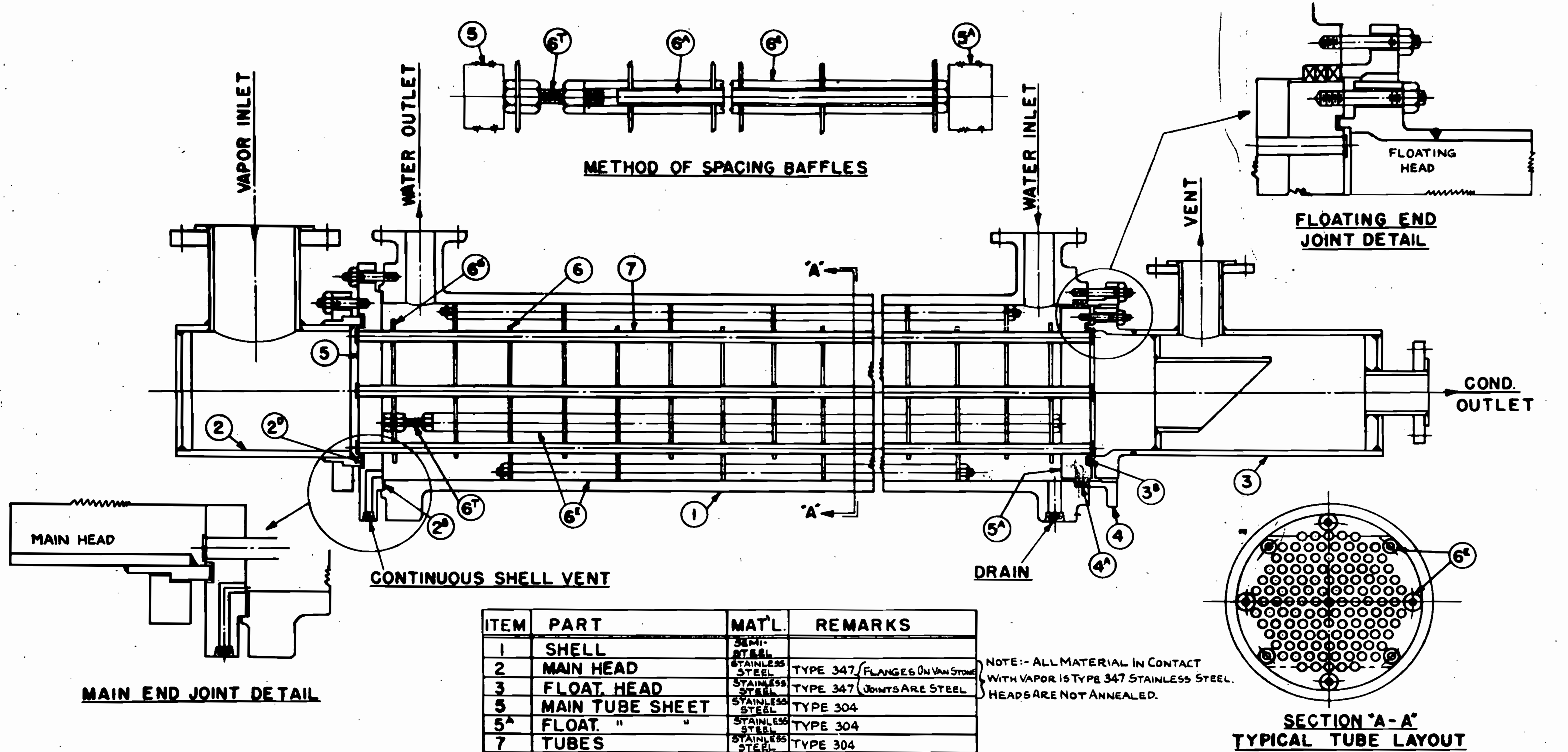
			LZ-D70



ITEM	PART	MAT'L.	REMARKS
1	SHELL	SEMI-STEEL	
2	MAIN HEAD	STAINLESS STEEL	TYPE 347 { FLANGES ON VAN STONE
3	FLOAT. HEAD	STAINLESS STEEL	TYPE 347 { JOINTS ARE STEEL
5	MAIN TUBE SHEET	STAINLESS STEEL	TYPE 304
5^A	FLOAT. " "	STAINLESS STEEL	TYPE 304
7	TUBES	STAINLESS STEEL	TYPE 304
6	BAFFLES	STAINLESS STEEL	TYPE 303
6^G	SUPPORT BAFFLES	STAINLESS STEEL	TYPE 303
6^E	BAFFLE SPACERS	PHENOLITE	
6^A	SPACER RODS	STAINLESS STEEL	TYPE 303
6^T	TIGHTENER	STAINLESS STEEL	TYPE 303
4	GLAND	BRZ.	
2^B	GASKET	J.M.	
2^D	DB'L. SHRD'D GASKET	STAINLESS STEEL	TYPE 304 SHROUDED ASBESTOS
3^B	DB'L. SHRD'D. GASKET	STAINLESS STEEL	TYPE 304 SHROUDED ASBESTOS
4^A	PACKING	NEOPRENE	
	STUDS	STAINLESS STEEL	TYPE 410 A.S.T.M. 193-39T-B6
	NUTS	STAINLESS STEEL	TYPE 416

NOTE:- ALL MATERIAL IN CONTACT WITH VAPOR IS TYPE 347 STAINLESS STEEL. HEADS ARE NOT ANNEALED.

18-8 STAINLESS STEEL

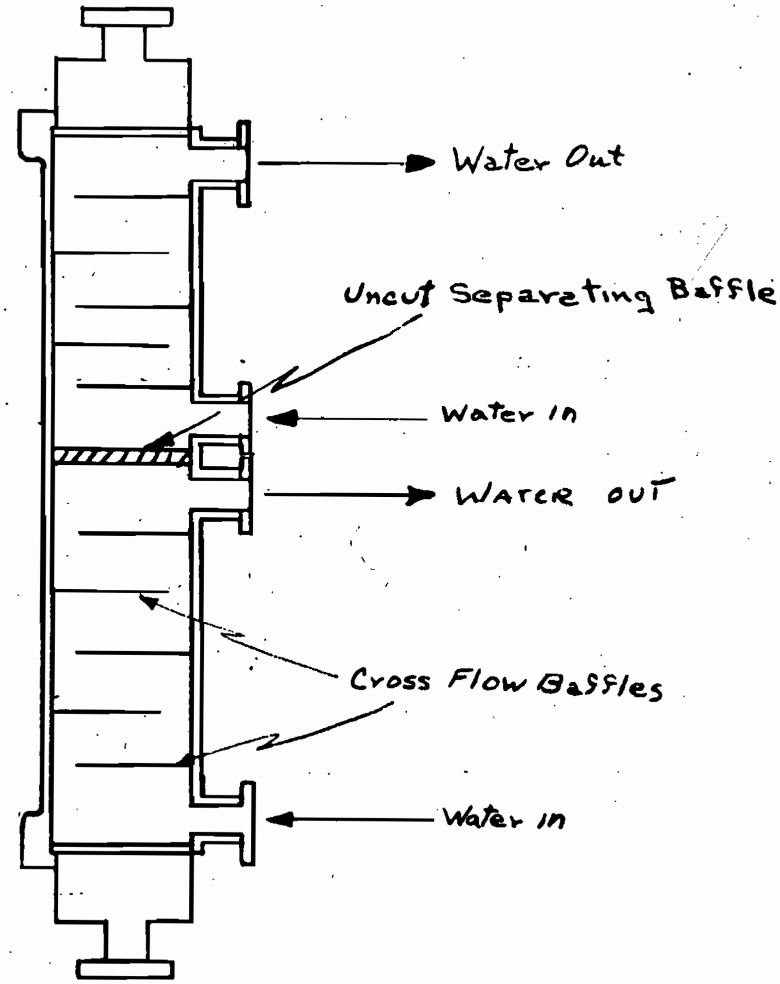
SECTIONAL ASSEMBLY & MAT'L. LIST
ANDALE VAPOR CONDENSER
TYPE 206S

ANDALE CO.
PHILA. PA.

DRW. G.W.F. DATE 3-3-41

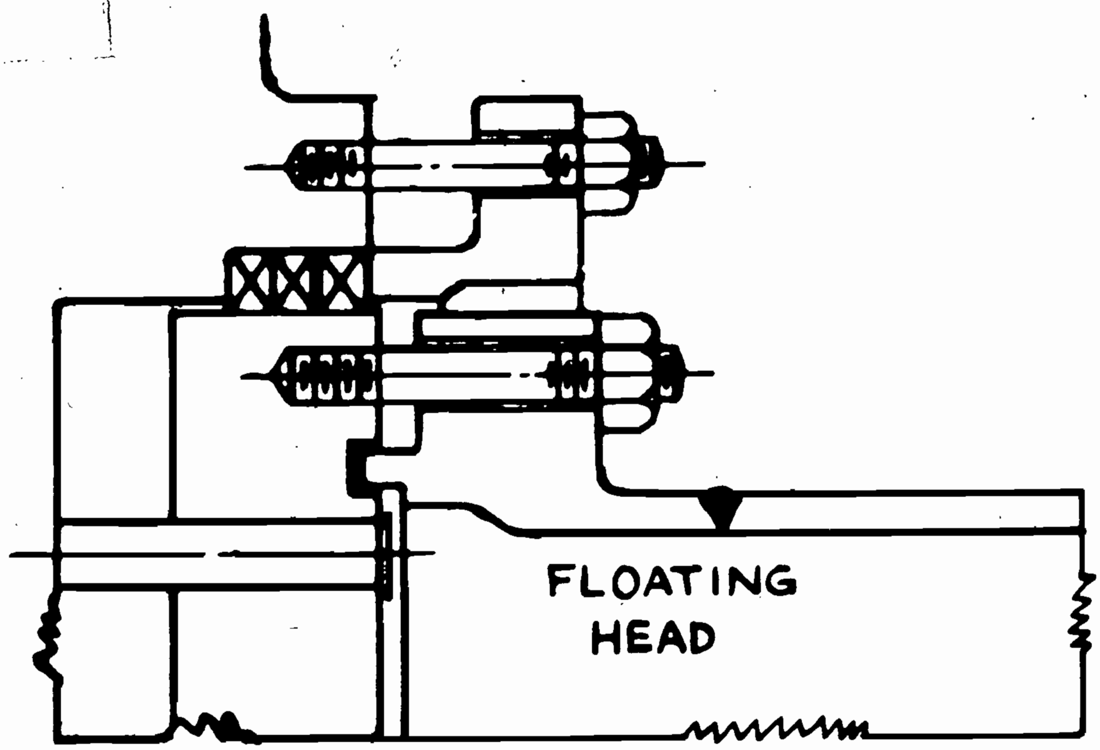
G4229

REF. ANDALE MAT'L. SPEC. No. E-4

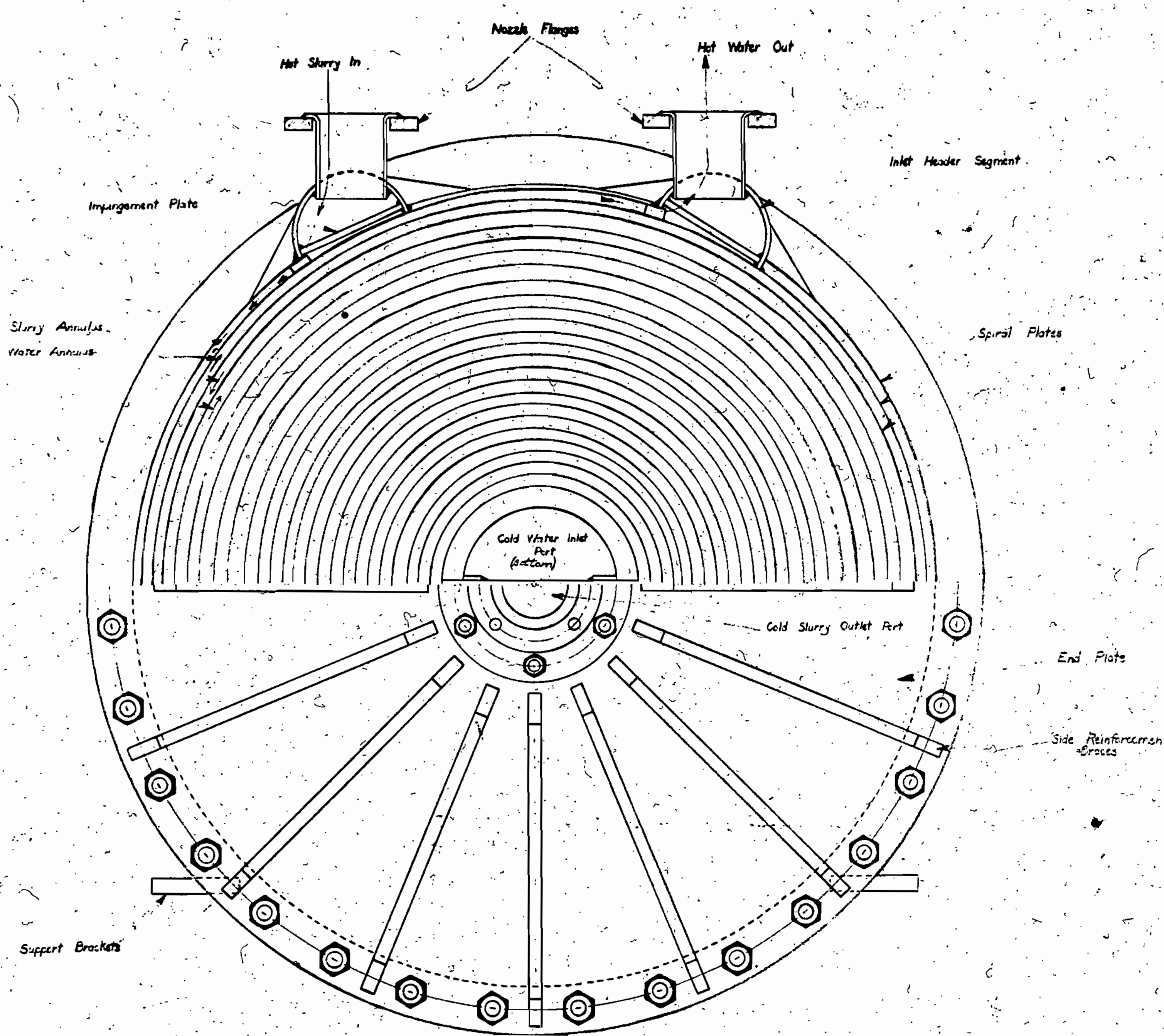


Double Water-Cooled Shells

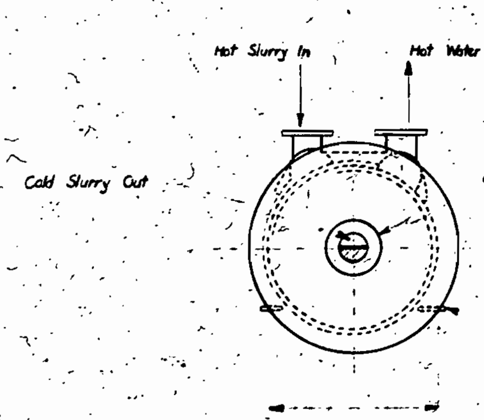
LZ-D71



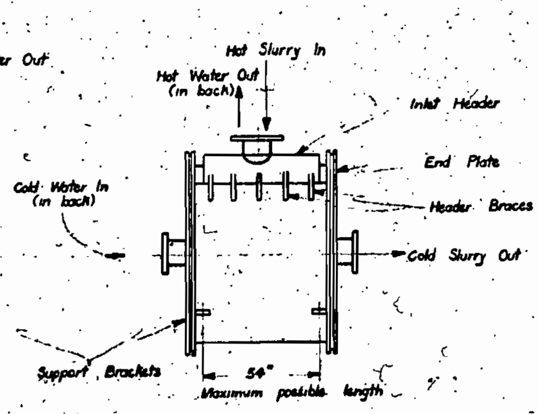
FLOATING END JOINT DETAIL



END ELEVATION and PARTIAL SECTION



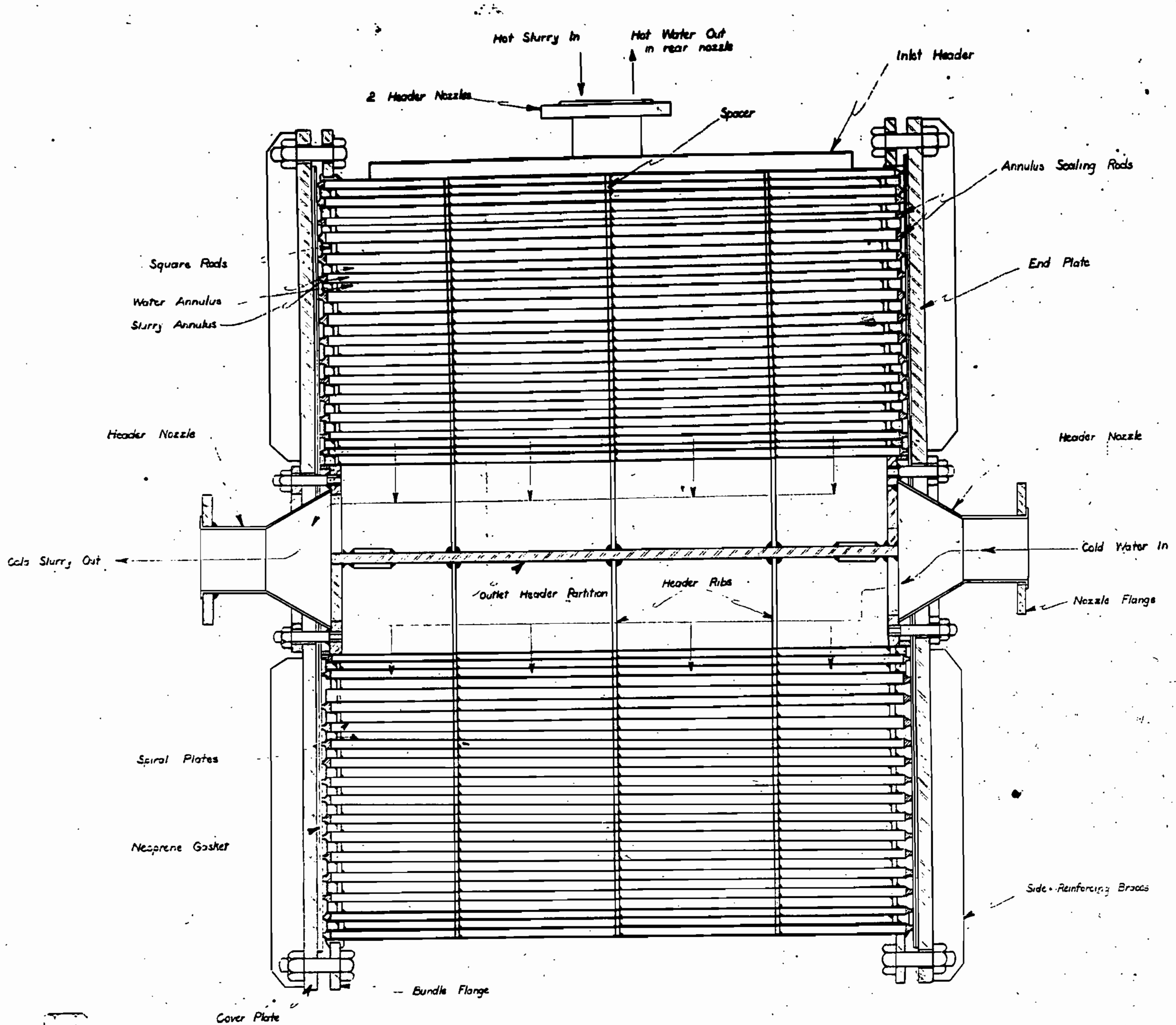
End Elevation



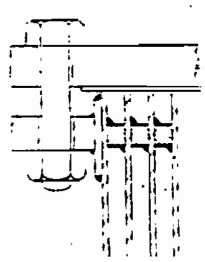
Side Elevation

SPIRAL HEAT EXCHANGER

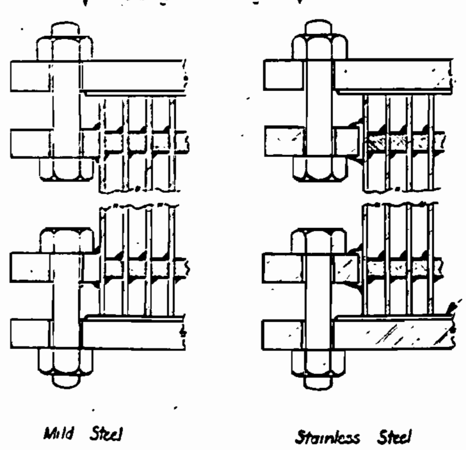
END ELEVATION & PARTIAL SECTION



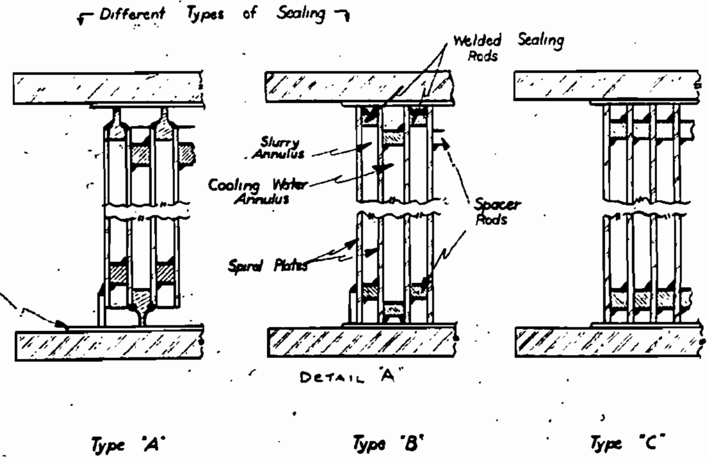
SECTIONAL ELEVATION



Welding of Flanges



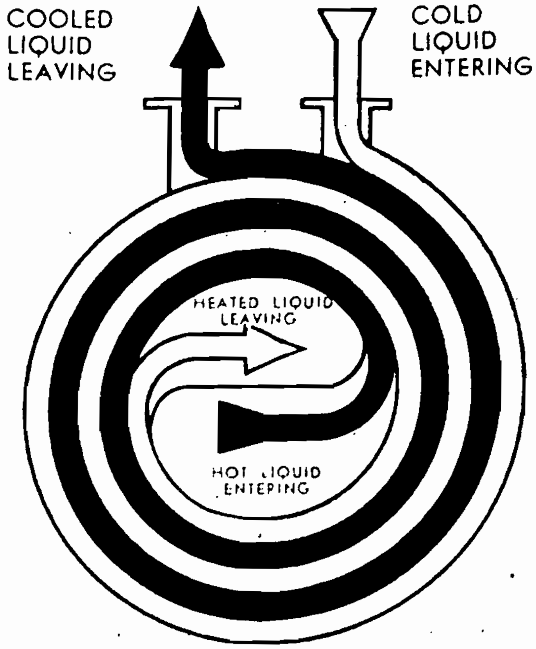
Different Types of Sealing



SPIRAL HEAT EXCHANGER

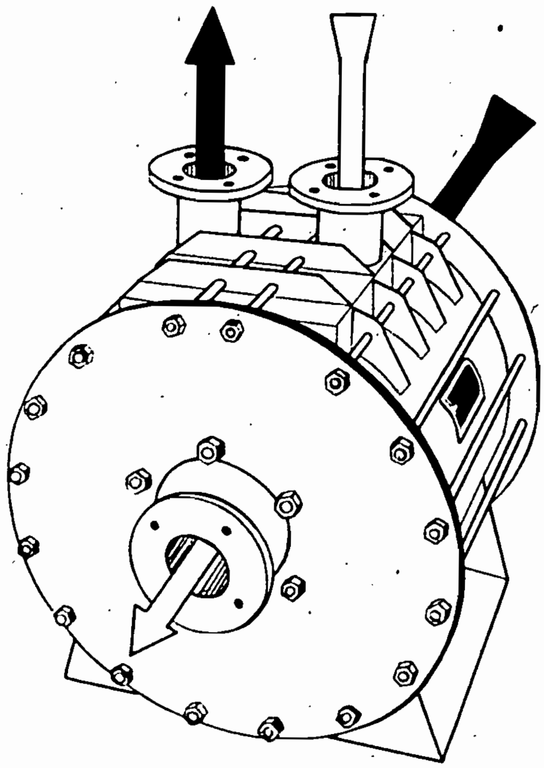
Sectional Elevation

LZ-A69



The Spiral Heat Exchanger

PATENTED IN MOST INDUSTRIAL COUNTRIES



AMERICAN HEAT RECLAIMING CORP.

1270 SIXTH AVENUE (RKO Building) New York City

Representative in Canada

PAPER MILL EQUIPMENT LTD., 921 SUN LIFE BUILDING, MONTREAL, P. Q.

FIG 1

OPEN CHANNELS

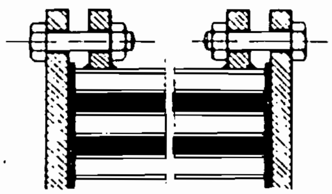


FIG. 1

WELDED CHANNELS

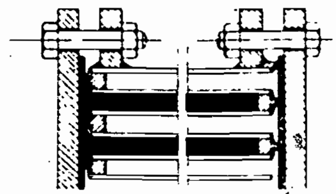


FIG. 2

SPIRAL COVER

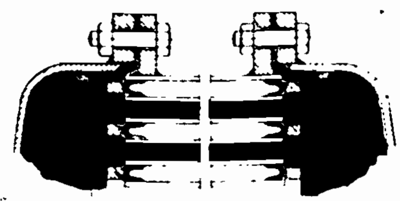


FIG. 3

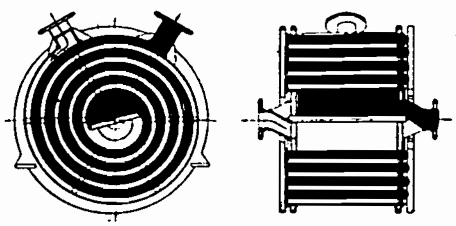


FIG. 4

Design and Operation

The principle of waste heat recovery from liquids and gases is not new, but the Spiral Heat Exchanger offers a completely new design, which has shown economies hitherto unobtainable with other types of heat exchange equipment.

While the majority of heat exchangers are of the tubular type, the Spiral type employs specially rolled plates, so placed, one within the other, that two separate passages are formed, in which the media flow at the most economical speed, either contra or cross current in the manners illustrated in figures 4, 5, and 6.

When calculating a Spiral Heat Exchanger for a certain purpose, the length and the cross section of the two passages are so chosen that both media obtain the most ideal flow conditions.

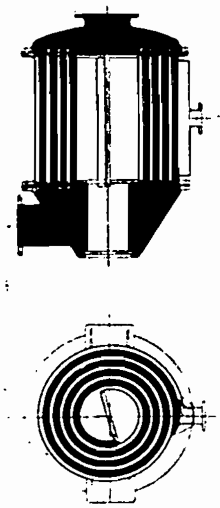


FIG. 5

Sealing

The ends of the spiral are kept tight by means of gaskets of suitable material, pressed against the apparatus by means of flat steel covers. These covers are bolted to flanges welded to the heat exchanger. See figure 1.

If operating conditions require, one of the two passages is welded tight at one side and the other passage at the opposite side. This design leaves one passage on each side for inspection and cleaning if required. The sealing is completed with gaskets and flat steel covers as described above. See figure 2.

In the surface condenser, the liquid passage is made tight by means of U-shaped rubber packing. This packing is kept in place by means of a "Spiral Cover". See figure 3.

Types of Spiral Heat Exchangers

There are three general types of Spiral Heat Exchangers adequate for practically all industrial process requirements. Briefly outlined and illustrated herein, they are as follows:

<u>TYPE I - Figure 4</u>

Liquid to liquid heat exchanger. The cooling medium enters the outer passage at the periphery, absorbs the heat, and leaves the apparatus at the center. The heating medium enters the heater at the center, flows through the inner channel and leaves at the periphery.

The contra flow principle is perfectly utilized in this case.

<u>TYPE II - Figure 5</u>

Surface condenser. For condensing [illegible] and cooling of gases operated on the cross flow principle.

<u>TYPE III - Figure 6</u>

Surface condenser, contra flow type. [illegible] steam enters at the top of the heat exchanger. This apparatus [illegible] used for heating liquids, utilizing either fresh steam, [illegible] mixing, etc. The steam or steam and gas mixing [illegible] the heat exchanger. The intake is so arranged that constant velocity is maintained through the passage.

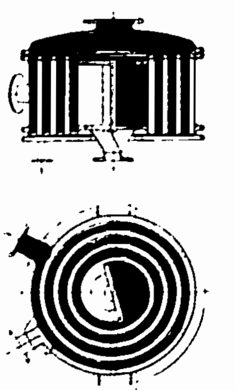


FIG. 6

Fig. 2

Process Industries Application

Food Processing

Heating, cooling and pasteurization of tomato pulp (catsup), fruit juices, syrups, oils and vinegar, etc.

Figure 7. Spiral Heat Exchanger for cooling of tomato pulp by means of city water in a packing company.

Design see figure 4.

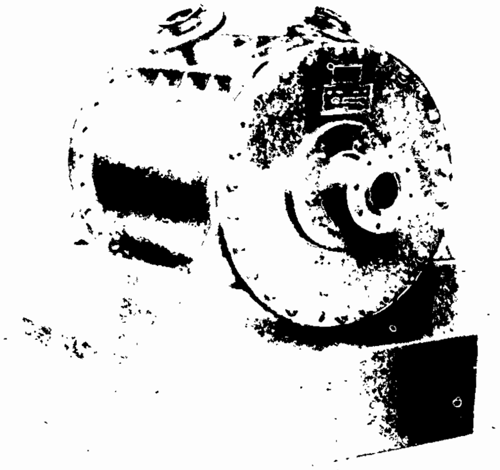


FIG. 7

Pulp Manufacture

Acid Heaters, Relief Coolers, Condensate Coolers, Preheaters for black liquor.

Figure 8. Spiral Condenser for cooling of SO_2 gas. Material: Acid resisting steel.

Design see figure 5.

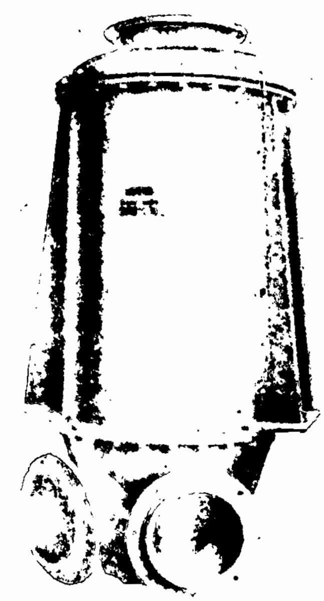


FIG. 8

Sugar Industry

Juice Heaters, Heat Exchangers for Steffen waste, Liquor Heaters, etc. Figure 9. Spiral Heat Exchanger for heating thin sugar liquor by means of vapor from the multiple effect evaporation plant.

Design see figure 6.

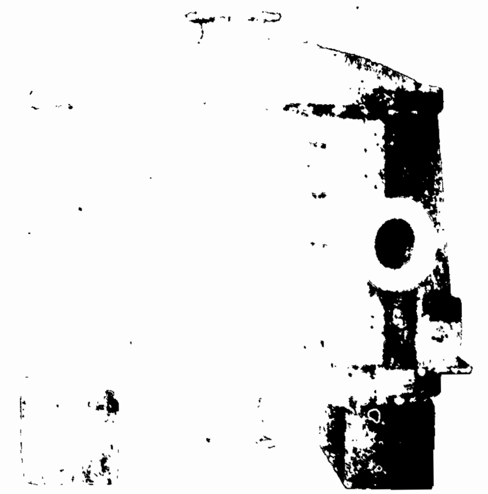


FIG. 9

General

Breweries, Distilleries, Dye Works, Chemical Plants, Oil Coolers. Coolers for oleum, Sulphuric acid.

Materials of Construction

We manufacture Spiral Heat Exchangers from all the usual materials, and also from a wide range of special alloys to resist corrosion or to prevent contamination of the products handled by the heat exchanger.

FIG. 3

OPERATING ADVANTAGES

1. **HIGHER HEAT TRANSFER.** The design permits the spiral passages to be so dimensioned that the most economical flow of the heating and cooling media can be obtained. The added effect of perfect contra-flow conditions contribute to the high performance of the Spiral Heat Exchanger.

2. **LOW PRESSURE DROP.** Absence of sharp bends and other obstructions such as baffles, results in low operating costs for the same high performances. The pressure drop is effectively utilized for heat transfer.

3. **COMPACTNESS AND LOW WEIGHT.** Because of the compact design of the Spiral Heat Exchanger, great savings in space and weight are possible.

4. **GREATER WALL THICKNESS.** Particularly advantageous for corrosive conditions, the Spiral Heat Exchanger incorporates a wall thickness greater than that which is generally used in the conventional types of exchangers.

5. **ACCESSIBILITY.** The removal of the end covers exposes the heating surface to inspection. Cleaning can easily be accomplished by washing with chemical solutions, or mechanically if required.

6. **NO MIXING OF MEDIA.** Because the seals are effective and easy to keep tight, no mixing of the media passing through the spirals can take place.

7. **EASY TO KEEP CLEAN.** Heating surface can be cleaned by pumping hot water, possibly containing some suitable chemical, through the heat exchanger. The high and even velocity of the media flowing through the Spiral Heat Exchanger, minimizes the risk of scale formation and accumulation of impurities on the heating surface. When handling severely scale forming liquids where scaling is unavoidable, the exchanger is designed with comparatively large spacing within narrow width to facilitate cleaning by means of mechanical tools.

8. **MINIMUM RADIATION LOSSES.** In the Spiral Heat Exchanger, the cooling medium enters at the periphery and the hot medium at the center. Therefore, the radiation losses are negligible and the apparatus is easier to handle.

9. **NO SCALING.** The tendency to scale is substantially reduced by comparatively high velocity of the liquid and intense turbulence created by the spiral flow, which in the Spiral Heat Exchanger remains constant at every point and during the whole process of flow.

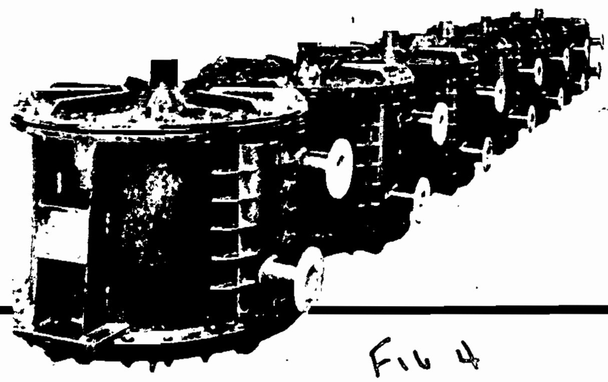
10. **RAPID TEMPERATURE CHANGE.** The design of the Spiral Heat Exchanger permits a high and even flow velocity. This prevents the liquid handled from "burning" and permits an exceptionally fast temperature rise.

    The above point is essential, for instance, in the food and other industries, where close temperature control is vitally essential.

The compactness of the Spiral Heat Exchanger makes it possible to install the same in places where the conventional type of heat exchanger cannot be installed.

The efficiency of the Spiral Heat Exchanger makes it possible to recover waste heat where heretofore it has been considered uneconomical to use heat recovery apparatus.

The Engineering Department of the American Heat Reclaiming Corporation will be glad to help solve your heat transfer problems.



7 Spiral Heat Exchangers ready for shipment to a sulphuric acid plant for cooling oleum.

Fig 4

VAPOR TIGHTNESS.

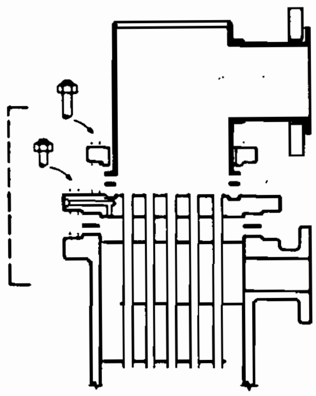
A condenser must be completely tight on the gas side. There is no such rating as "percentage of tightness" in dealing with gases and vapors. In most stiuations, at any time when a condenser is not tight on the gas side it must be out of action.

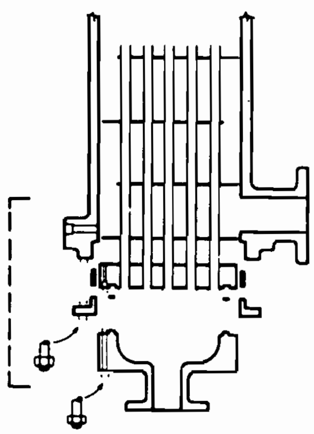
In Andale rules of construction, note the following:—Gas joints and water joints are entirely separate,—bolts, glands and packings. Gaskets on gases, vapor and condensate are always of the confined pressure type; tongue and groove. All seals are accessible. No gasket is buried.

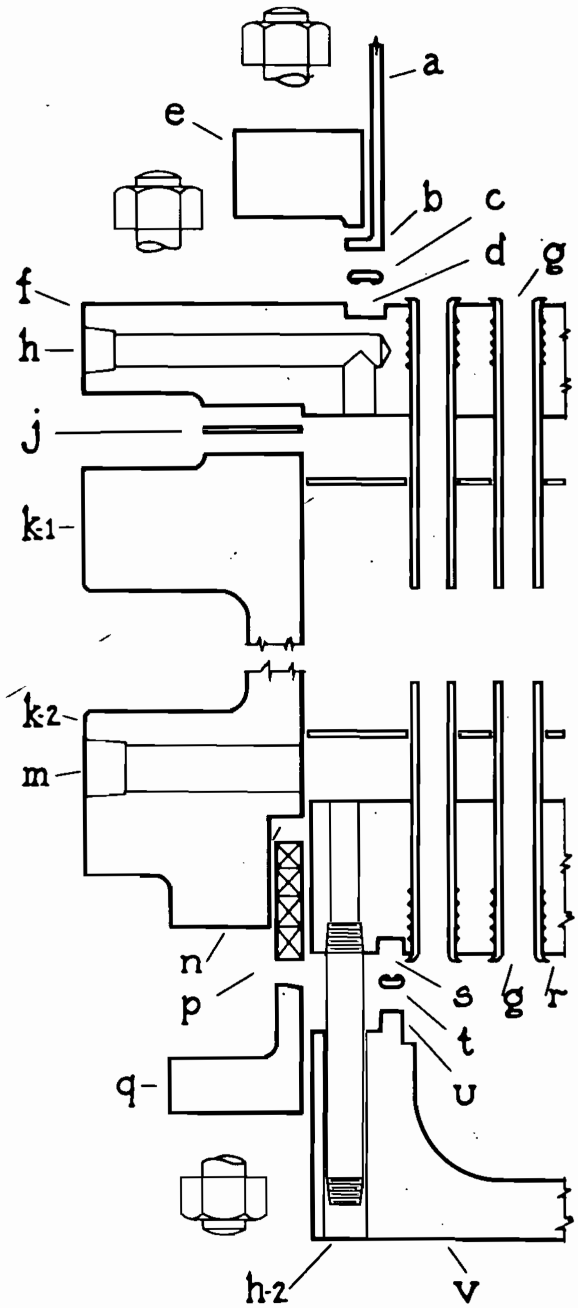
If any leak occurs it is located more easily, tightened easily, and its tightness can be checked and proved easily.

All Andale vapor condensers are vented from the underside of the main tube sheet, (see h), to prevent accumulation there of entrained air, released from the water,—which otherwise would favor corrosion, and would have an insulative effect.

The illustrations are not to scale; they are diagrammatic.







a. part of the high main head.
b. main head vanstone.
c. gasket on the gases or vapor mixture; confined type; tongue and groove type.
d. gas gasket groove, in main tube sheet.
e. main head flange; rugged.
f. main tube sheet.
g. tubes; expanded into tube sheet; metal to metal joint; belled.
h. vent, through main tube sheet from water side; to prevent accumulation of occluded gases released from the water; (to be piped and always open).
h-2. drain, through lower tube sheet; piped through a free hole in head flange. For complete drainage of shell. Open only when draining.
j. gasket on the water.
k-1. upper end of shell.
k-2. lower end of shell.
m. drain hole from water path.
n. packing box, on water only; to side of floating tube sheet.
p. packing, on water only.
q. gland; rugged.
r. floating tube sheet.
s. gas gasket groove, in floating tube sheet.
t. gasket, on the discharge vapors and condensate; confined type; tongue and groove type. Any leak shows to atmosphere.
u. floating head tongue.
v. floating head (one type), on floating end of tube bundle; on discharge vapors and condensate.
—. studs and bolts, separate sets on water and on gas, are not shown completely. Their positions are indicated clearly.

FIGURE 5