COMPACT HEAT EXCHANGER

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ABSTRACT

The heat transfer efficiency of a heat exchanger is increased by making the entire flow pattern of the exchanger a straight-through flow pattern, thereby providing a compact heat exchanger which is substantially 100 percent effective and compact in size. Thus, the required cooling effect is accomplished with an exchanger which is smaller in size than a prior exchanger and has a reduced cost.

10 Claims, 6 Drawing Figures
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COMPACT HEAT EXCHANGER

BACKGROUND OF THE INVENTION

This invention relates, generally, to heat exchangers and, more particularly, to a heat exchanger suitable for use inside the compact casing of a centrifugal pump of the type described in a pending application Ser. No. 123,039 filed Mar. 10, 1971, now U.S. Pat. No. 3,652,179 which is a continuation of a now abandoned application Ser. No. 810,312, filed Mar. 25, 1969, by Oskar Hagen and assigned to the Westinghouse Electric Corporation.

A prior heat exchanger, or cooler, provided for a prior centrifugal pump had two features, which were essential for its operation. First, a separator plate was required mid-way in the exchanger to divert the fluid flow. Secondly, a labyrinth seal was required on the separator plate at the shaft of the pump. The cooling operation of the prior exchanger was considerably less than 100 percent effective due to the flow characteristics of the fluid being cooled. Therefore, the prior exchanger could not be compact and still provide the required cooling effect.

SUMMARY OF THE INVENTION

In accordance with one embodiment of the invention, a centrifugal pump having a shaft rotatably mounted in a generally cylindrical casing is provided with an annular heat exchanger which surrounds the shaft between a first labyrinth seal and a second labyrinth seal located on the shaft where it passes through a first wall and a second wall of the exchanger. The fluid to be cooled enters water channels or passageways located at the first labyrinth seal. From the water channels it collects in a first header one wall of which is a disc-like distribution plate having a series of holes therein. The area of the holes is increased as the distance of the holes to the center line of the shaft increases. The fluid passes from the first header through the holes in the plate into the heat exchanger compartment at a uniform flow rate and flows through the compartment with a straight-through flow pattern. After passing over the heat exchanger coils the fluid is collected in a second header constructed in a manner similar to the first header. The cooled fluid flows from the second header through channels located at the second labyrinth seal.

BRIEF DESCRIPTION OF THE DRAWINGS

For a better understanding of the nature of the invention, reference may be had to the following detailed description, taken in conjunction with the accompanying drawings, in which:

FIG. 1 is a view, partly in section and partly in elevation, of a centrifugal pump having a heat exchanger embodying principal features of the invention;

FIG. 2 is an enlarged view, in plan, of one of the disc-like distribution plates utilized in the heat exchanger;

FIG. 3 is an enlarged view, in plan, of a modified disc-like distribution plate;

FIG. 4 is an enlarged view, in section, of a portion of a heat exchanger constructed in accordance with the prior art;

FIG. 5 is an enlarged view, in section, of a portion of the present heat exchanger; and

FIG. 6 is an enlarged view in plan, of one of the cooling tubes or coils utilized in the heat exchanger.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to the drawings, particularly to FIG. 1, the centrifugal pump 10 shown therein comprises a generally cylindrical casing 11 having a radial discharge nozzle 12 near the upper end of the casing and a suction nozzle 13 at the lower end of the casing, a main annular flange 14 having a generally cylindrical neck 15 extending downwardly inside the casing 11, a rotatable shaft 16 extending through the flange 14 and the neck 15, and an impeller 17 attached to the lower end of the shaft 16 by means of an impeller nut 18 and a bolt 19. The shaft 16 may be driven by an electric motor (not shown) mounted on a motor support stand 21 attached to the top of the main flange 14 by bolts 22. The shaft of the motor is connected to the shaft 16 by means of a coupling 23. The upper end of the casing 11 is attached to the main flange 14 by means of bolts 24 which extend through the flange 14 and are threaded into the casing 11.

As shown in FIG. 1, a radial bearing assembly 25 for the shaft 16 is attached to the lower end of the neck 15 by means of bolts 26. The bearing assembly 25 may be of any suitable type. As shown, the assembly includes a sleeve 27, pressed on the shaft 16, suitable bearing material 28 and a housing 29 enclosing the bearing material 28.

As also shown in FIG. 1, breakdown bushings 31 surround the shaft 16 inside the neck 15 above the bearing assembly 25. The bushings 31 may be of a type well known in the art. Shaft seals 30 of a controlled leakage type are disposed in the main flange 14 above the breakdown bushings. The seals 31 are retained in position by a housing 32 attached to the top of the flange 14 by bolts 33. A vapor seal housing 34 is attached to the seal housing 32 by means of bolts 35. A cover 36 for the housing 34 is attached to the housing by means of bolts 37. The shaft seals within the housings 32 and 34 may be of a type well known in the art, such as, for example, the seal arrangement described in U.S. Pat. No. 3,347,552, issued Oct. 17, 1967, to Erling Frisch. A suitable leak-off connection may be made to the housing 32 by means of a coupling 38 and a pipe 39 attached to the housing 32. Likewise, a leak-off connection may be made to the housing 34 by means of a coupling 41 and a pipe 40 attached to the housing 34. The leak-off liquid is returned to the circulatory system in a manner well known in the art.

The present pump is suitable for circulating a liquid in a circulatory system at a relatively high temperature and a high pressure. Suitable connections may be made to the suction nozzle 13 and to the discharge nozzle 12 to enable the pump to circulate the liquid in the system.

In addition to the impeller 17, the pump includes a main impeller seal ring 42, a diffuser 43 extending upwardly from the impeller 17 inside the casing 11 and a guide elbow 44 connecting the diffuser 43 with the radial discharge nozzle 12. Thus, the liquid drawn into the pump through the suction nozzle 13 by the impeller 17 is directed to the discharge nozzle 12 by means of the diffuser 43 and the elbow 44.
In order to protect the bearing assembly 25 and the shaft seals in housings 32 and 34 from the heat of the high temperature liquid circulated by the pump, a cylindrical thermal barrier 45 surrounds the bearing assembly 25, and an annular heat exchanger or cooler 46 is disposed between the impeller 17 and the bearing assembly 25. The cylindrical barrier 45 includes an inner wall 47 and an outer wall 48 spaced from the inner wall 47. The upper ends of the walls 47 and 48 are secured to a thermal barrier flange 49, as by welding. The flange 49 is clamped between the main flange 14 and the casing 11 by means of the flange bolts 24. The lower end of the space between the walls 47 and 48 is closed by an annular bottom wall 51. A flange ring 53 is provided on the outer wall 48. A diffuser 43 is attached to the flange ring 53, as by welding. The guide elbow 44 has upwardly extending support members 54 and 55 which are attached to the flange 49 by means of bolts 56. A cylindrical member 52 extends between the flange 53 and the support members 55.

As shown more clearly in FIG. 5, the heat exchanger 46 comprises an annular compartment 57 defined by a first or lower wall 58 surrounding the shaft 16, a second or upper wall 59 surrounding the shaft and spaced along the shaft from the first wall, an inner sleeve 61, and an outer sleeve 62 radially spaced from the inner sleeve. The sleeves 61 and 62 extend parallel to the shaft between the first wall 58 and the second wall 59. A plurality of cooling coils or tubes 63 are disposed within the compartment 57.

In order to obtain a uniform flow rate of fluid, such as water, into the compartment 57, a first header 64 in the first wall 58 is provided with a disc-like distribution plate 65 having a series of holes 66 therein. Likewise, a second header 67 in the second wall 59 is provided with a disc-like distribution plate 65 similar to the plate in the first header to permit a uniform flow of fluid out of the compartment 57.

As shown more clearly in FIG. 2, the holes 66 in each plate 65 are located in a plurality, for example, three circular rows spaced radially outwardly from the center of the shaft. The number of holes in successive rows increases as the radial distance from the center of the shaft increases. In the example shown, the inner row contains 16 holes, the middle row contains 20 holes and the outer row contains 24 holes, all of the same diameter. Thus, the area of the holes in the plates increases as the distance to the center line of the shaft increases, thereby obtaining a uniform flow rate of fluid into and out of the compartment 57 since the area of the holes in the distribution plates increases as the pressure of the fluid decreases.

In the modified plate 65a shown in FIG. 3, the number of holes 66a in each row is the same, but the diameter of the holes in successive rows increases as the radial distance from the center line of the shaft increases. Accordingly, the area of the holes in the modified plates increases as the distance to the center line of the shaft increases.

As shown in FIG. 5, a first labyrinth seal 68 is provided on the shaft 16 where the first wall 58 encircles the shaft. Likewise, a second labyrinth seal 69 is provided on the shaft where the second wall 59 encircles the shaft. The first wall 58 has a plurality of passageways 71 therein extending from the first header 64 to the shaft 16 at the first seal means 68. The second wall 59 has a plurality of passageways 72 therein extending from the second header 67 to the shaft at the second seal means 69.

As shown in FIG. 1, relatively small openings 73 and 74 are provided in the impeller 17 to admit liquid into a chamber 75 above the impeller. In this manner, a relatively small amount of the liquid being circulated by the impeller is caused to flow through a portion of the first labyrinth seal 68 and the first passageways 71 into the first header 64. As shown by the arrows in FIG. 5, the liquid from the header 64 through the holes 66 in the first distribution plate 65 into the heat exchanger compartment 57 at a uniform flow rate over the area of the compartment. The liquid flows through the compartment over the cooling tubes 63 with a straight-through flow pattern. The cooled liquid is permitted to flow out of the compartment through the holes 66 in the second distribution plate 65 into the second header 67, thence through the second passageways 72 and a portion of the second labyrinth seal 69 along the shaft 16 into the area around the bearing assembly 25. The first labyrinth seal 68 directs the liquid into the first passageways 71 and the second labyrinth seal 69 directs the liquid along the shaft into the area around the bearing assembly. Thus, the bearing assembly and the shaft seals are protected from the heat of the liquid being circulated through the pump by the impeller 17.

It is essential that the entrances to the first passageways 71 and the exits from the second passageways 72 be located on the same diameter with reference to the center line of the shaft. Otherwise, there would be an unbalance of pressure which would cause liquid to circulate back along the shaft 16.

Provision is made for circulating a cooling liquid, such as water, through the cooling tubes 63. As shown in FIG. 1, passageways 76 and 77 in the flange 49 communicate with pipes 78 and 79, respectively. These pipes pass through the space between the neck 15 and a cylindrical wall 81 which extends between the flange 49 and the second or upper wall 59 of the heat exchanger. The pipe 78 communicates with a semi-cylindrical header 82 which extends along the heat exchanger 46 outside of the outer sleeve 62 of the exchanger. One side of the header 82 is closed by a wall 83 which extends between the first wall 58 and the second wall 59 of the exchanger. The bottom of the header is closed by a wall or plate 84.

Likewise, the pipe 79 communicates with a header 85 (FIG. 6) constructed in a manner similar to the header 82. The passageway 76 may be connected to a suitable supply source by means of a pipe 86. A discharge pipe 87 is connected to the passageway 77. As shown in FIG. 6, the coils or tubes 63 communicate with the headers 82 and 85, thereby causing the cooling liquid to circulate through the tubes as it flows from the header 82 into the header 85 from which it is discharged through the pipes 79 and 87. The coils or tubes 63 are of a pancake type with spacers 88 disposed between the turns of each coil.

Some of the liquid being circulated by the main impeller 17 is admitted from the pump casing 11 into the thermal barrier 45 through openings 89 in the outer wall 48 and the member 52. Thus, the thermal barrier is filled with liquid which is cooled somewhat by the
liquid flowing through the pipes 78, 79 the headers 82, 85 and by the liquid in the area around bearing assembly 25. 

As described in the aforesaid copending application, the effectiveness of the thermal barrier 45 may be increased by providing a plurality of spaced concentric sleeves 91 between the inner wall 47 and the outer wall 48. The sleeves 91 separate the liquid inside the thermal barrier into vertical layers, thereby increasing the effectiveness of the thermal barrier.

The advantages of the present heat exchanger over prior exchangers may be seen by referring to FIGS. 4 and 5. In the prior exchanger shown in FIG. 4, a separator plate 92 is provided midway in the exchanger to divert the fluid flow. As indicated by the arrows, a liquid flowing through the exchanger channels in a manner to decrease the effective heat transfer area which is shown between dot-dash lines. Thus, the prior exchanger is considerably less than 100 percent effective due to the flow characteristics and, therefore cannot be compact. Furthermore, a labyrinth seal 93 is required on the separator plate 92 where it encircles the shaft, thereby increasing the cost of the exchanger.

As indicated by the arrows in FIG. 5, the liquid enters the exchanger compartment 57 at a uniform flow rate over the area of the compartment and flows through the compartment with a straight-through flow pattern. Thus, the effective heat transfer area, which encompasses all turns of the coils or tubes 63, is greatly increased as compared with the exchanger in FIG. 4. Therefore, the present exchanger can be more compact and requires less space for installation inside the casing of a centrifugal pump.

From the foregoing description, it is apparent that the invention provides a heat exchanger which is highly effective in its cooling operation. Therefore, it is compact in structure and is particularly suitable for installation inside a compact shaft seal casing for a centrifugal pump.

We claim:

1. An annular heat exchanger for use with a centrifugal pump having an impeller driven by a shaft rotatably mounted in a generally cylindrical casing, comprising first and second seal means spaced along the shaft inside the casing, an annular compartment defined by a first wall surrounding the shaft, a second wall surrounding the shaft and spaced along the shaft from the first wall, at least one cooling tube disposed within the compartment, a first header in the first wall with a disc-like distribution plate having a series of holes therein, the area of the holes increasing as the distance to the center line of the shaft increases, a second header in the second wall similar to the first header, said first wall having first passageways therein extending from the first header to the shaft at said first seal means, said second wall having second passageways therein extending from the second header to the shaft at said second seal means, means causing part of the fluid circulated by the impeller to flow into the exchanger compartment through said first passageways and said first header at a uniform flow rate, and said second header permitting the fluid to flow out of the compartment through said second passageways at a uniform flow rate.

2. The annular heat exchanger defined in claim 1, including radially spaced inner and outer sleeves extending parallel to the shaft between said first and second walls.

3. The annular heat exchanger defined in claim 1, wherein the first seal means directs the fluid into said first passageways, and the second seal means directs the fluid out of the second passageways along the shaft.

4. The annular heat exchanger defined in claim 3, wherein the seal means are of the labyrinth type with the passageways being intermediate the extremities of the seals.

5. The annular heat exchanger defined in claim 1, wherein the entrances to the first passageways and the exits from the second passageways are located on the same diameter with reference to the center line of the shaft.

6. The annular heat exchanger defined in claim 1, wherein the fluid flows through the heat exchanger compartment with a straight-through flow pattern.

7. The annular heat exchanger defined in claim 1, wherein the holes in each disc-like distribution plate are located at different radial distances from the center of the shaft and the number of holes increases as the radial distance from the center line of the shaft increases.

8. The annular heat exchanger defined in claim 1, wherein the holes in each disc-like distribution plate are located at different radial distances from the center of the shaft and the number of holes in successive rows increases as the radial distance from the center line of the shaft increases.

9. The annular heat exchanger defined in claim 1, wherein the holes in each disc-like distribution plate are located at different radial distances from the center of the shaft and the diameter of each hole increases as its radial distance from the center line of the shaft increases.

10. The annular heat exchanger defined in claim 1, wherein the holes in each disc-like distribution plate are located in a plurality of circular rows spaced radially outwardly from the center of the shaft and the diameter of the holes in successive rows increases as the radial distance from the center line of the shaft increases.