

July 13, 1937.

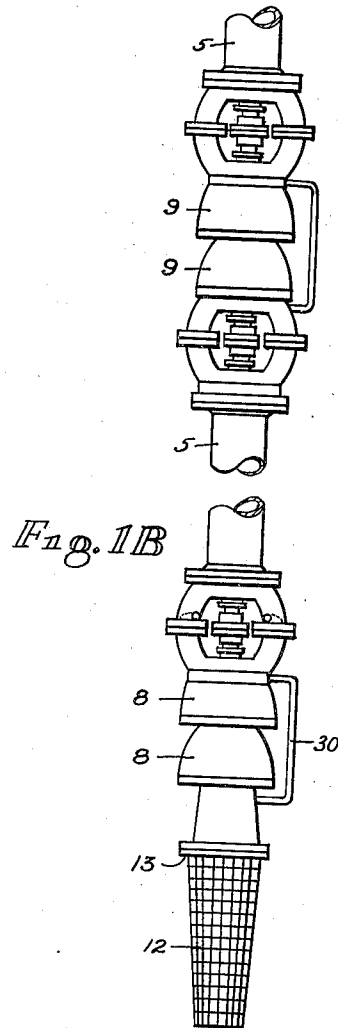
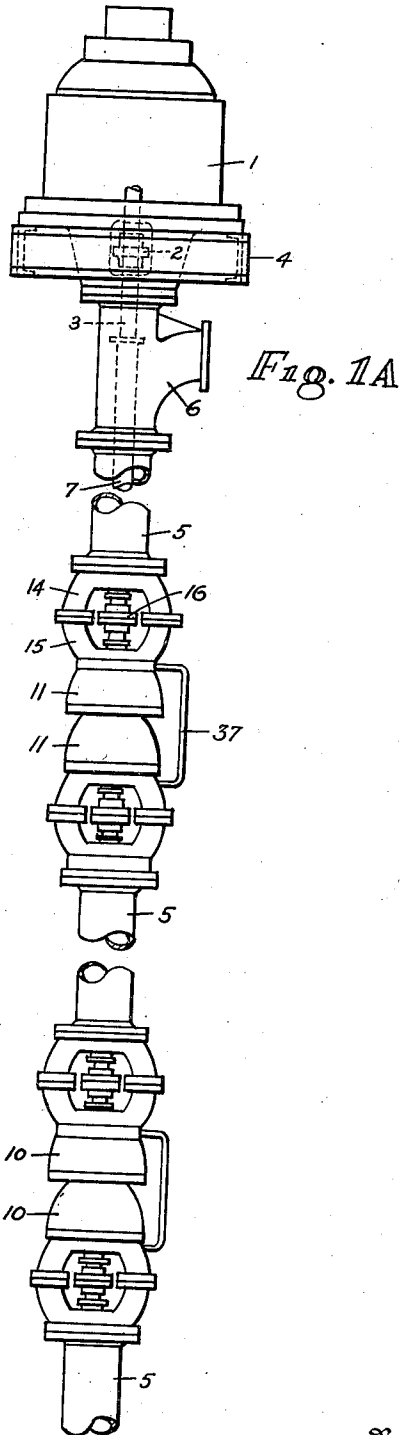
A. HOLLANDER

2,086,806

DEEP WELL CENTRIFUGAL PUMP

Filed Nov. 4, 1935

2 Sheets-Sheet 1



Aladar Hollander
Inventor

By J. D. Chesnut

Attorney

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A. HOLLANDER

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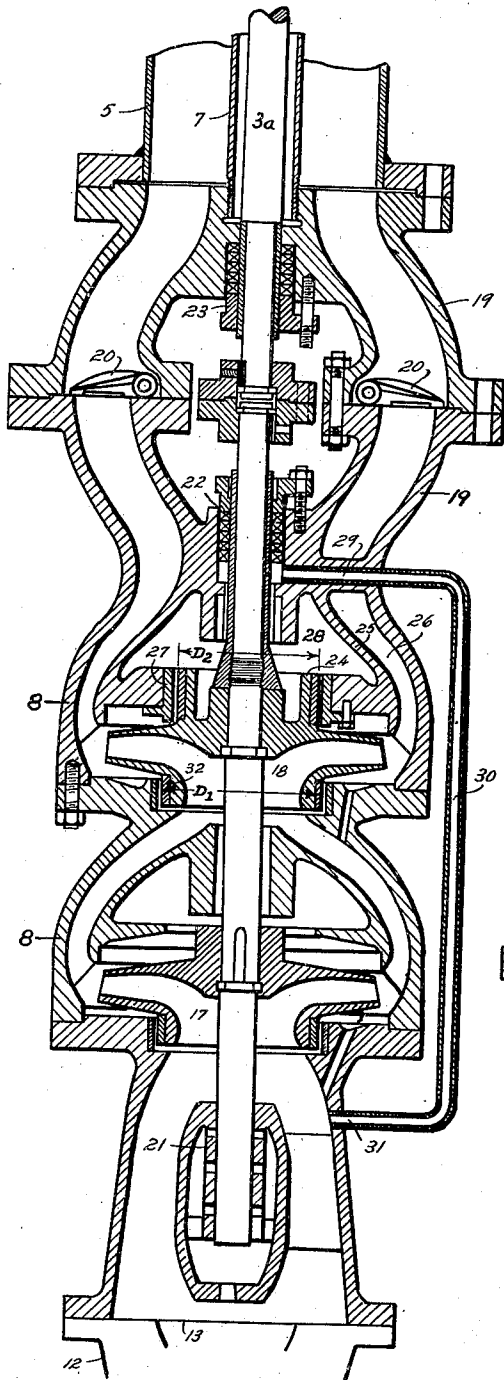


Fig. 2

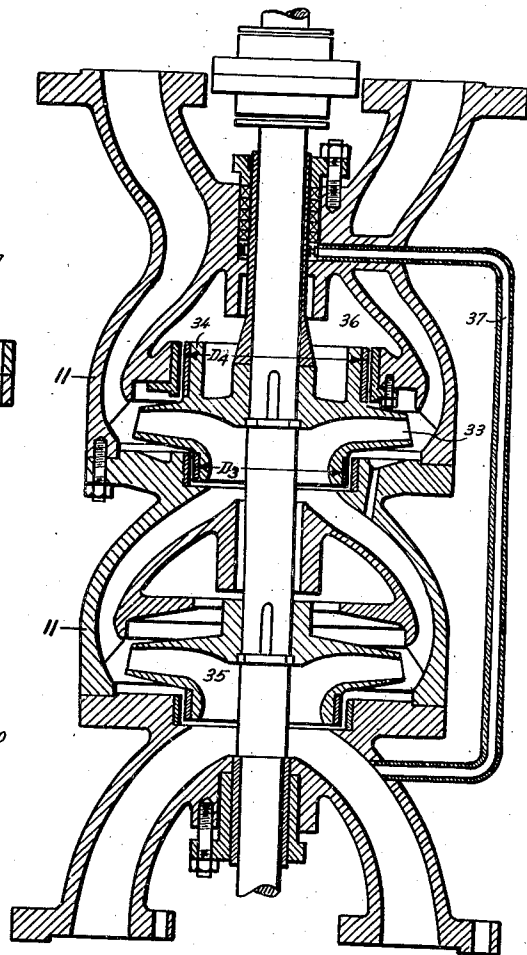


Fig. 3

Aladar Hollander

Inventor

By J. D. Chesnut

Attorney

UNITED STATES PATENT OFFICE

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DEEP WELL CENTRIFUGAL PUMP

Aladar Hollander, Berkeley, Calif., assignor to
Byron Jackson Co., Huntington Park, Calif., a
corporation of Delaware

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3 Claims. (Cl. 103—102)

This invention relates to centrifugal pumps of the multiple stage type such as are used for raising fluid from relatively deep wells, wherein the pump is used in the well and is driven by a long shaft extending from the pump to a motor at the top of the well.

The usual construction is to place the pump at the bottom of the well and connect it by means of a long shaft to a motor at the top of the well. The objection to this construction for pumps in very deep wells is the great weight of the shaft connecting the pump to the motor. Not only are such shafts very expensive, but a great deal of power is required to rotate them. The cost is further increased by the large size of the radial bearings of the column and in the large expensive axial thrust bearing in the pump head. In an effort to overcome these disadvantages, it has heretofore been proposed to divide the multi-stage pump into a series of pumps of less stages, each series being spaced a considerable distance above the next lower series, and progressively reducing the shaft diameter between each pump series, progressing downwardly. While this expedient has been of some advantage in reducing the dead weight of the shaft it has left much to be desired.

It is the object of the present invention to provide a series pump construction such that the tension on the shaft is still further reduced, and the size and weight of the shaft can therefore be reduced.

Another object is to provide balancing means between the series of pumps whereby the thrust created by a balancing means will assist in carrying the load on the shaft, thereby permitting a reduction in the size of and power consumed by the thrust bearing.

Other objects and advantages will appear from the following description and drawings wherein:
Fig. 1A is an elevation of the upper portion of the pump and column as assembled in a well.

Fig. 1B is an elevation of the lower portion of the pump and column shown in Fig. 1A.

Fig. 2 is a cross-sectional elevation taken through the two lowermost stages of the pump shown in Fig. 1B.

Fig. 3 is a cross-sectional elevation taken through the two uppermost stages of the pump shown in Fig. 1A.

Referring to Fig. 1 of the drawings, there is shown an electric motor 1 direct connected by a coupling 2 to a pump shaft 3. The motor is placed at the top of the well (not shown) on a suitable base 4, which supports the pump column

5, the upper end of which is provided with a discharge piece 6. The column shaft may be of the open water-lubricated type, or it may be of the oil lubricated type, in which latter event it is preferably enclosed in a tubular sleeve 7 in the conventional manner. The pump column is made up of a number of tubular sections 5. The pump comprises a series of stages spaced apart between the column sections 5. While there may be any number of stages arranged in any suitable grouping, I have shown eight stages arranged in pairs as shown at 8, 9, 10 and 11. A strainer 12 protects the suction inlet 13. Although the column sections 5 may be connected together in any of the usual methods, I prefer to divide the column into two parts by means of Y fittings 14 and 15. This construction makes the shaft couplings 16 accessible. The shaft sections 3 between the pairs of pump stages are progressively reduced in diameter toward the lower end of the column.

Refer now to Fig. 2 which shows the lowermost pair of pump stages, the first stage being shown at 17 and the second stage at 18. The Y coupling 19 above the second stage is provided with check valves 20 to prevent the return of fluid down the column 5. The shaft section 3a which extends from the lowermost pair of pump stages to the pump series next above is of smaller diameter than the shaft section between the next series of pumps above. In the present instance, the shaft 3a is $3\frac{3}{4}$ inches in diameter. The shaft increases in diameter in an upward direction to $5\frac{1}{4}$ inches. The pump is designed to pump 8,000 G. P. M. against an 800 ft. head, utilizing a 2500 H. P. electric motor running at 720 R. P. M. The diameter of the pump bowls is 36 inches. The lower end of the shaft 3a is enclosed in a conventional radial thrust bearing 21. Stuffing boxes 22—23 are provided on each side of the Y coupling.

If no means were employed for providing a pressure balance each of the pumping units would exert a downward thrust, due partly to the weight of the shaft and impellers and more largely to the downward hydraulic thrust of each impeller.

In the present invention a pressure balancing system is employed to give the upper series of pumping units an upward thrust which is larger than the hydraulic thrust of these units but smaller than the total of hydraulic thrust plus the weight of the rotating parts of the unit plus the column shaft below, thus leaving a slight downward force on the column shaft next above.

The first or lowermost group of impellers is hydraulically balanced to leave a small downward thrust on the lowermost shaft section to keep it in tension. In the present instance, this balancing is accomplished by placing a balancing ring on the upper side of the second stage impeller of each series and venting the pressure on the balancing ring back to the suction side of the next lower stage impeller. In the lowermost series of stages (Fig. 2) the upper impeller 18 is provided on its upper side with a balancing ring 24. The bowl which houses the impeller 18 is formed in the usual manner with a central cone-shaped portion 25 defining the inner wall of the diffusion space 26. The lower end of cone-shaped portion 25 is provided with an inwardly extending flange 27 adapted to make a close running fit with the balancing ring 24. The cone 25, together with the back of the impeller 18, forms a pressure chamber 28. Chamber 28 communicates at its lower end with the space between the balancing ring 24 and the flange 27 and is therefore exposed to the discharge pressure of the second stage impeller 18. The upper end of chamber 28 is provided with an outlet passage 29 which communicates through conduit 30 with the suction entrance to the first stage impeller 17 as shown at 31. If the diameter D_2 of the balancing ring were equal to D_1 and the piping connecting the suction were large enough so that its throttling effect is negligible, then the downward hydraulic thrust on the two stages would be zero. I prefer, however, to make the diameter D_2 of the balancing ring 24 somewhat less than the diameter D_1 of the impeller ring 32. This causes a slight downward thrust which can be reduced to any desired figure by choosing the proper diameter D_2 of the balancing ring.

Referring now to Figure 3 which shows the two uppermost stages of the pump series, 33 and 35, the last stage impeller 33 is provided with a balancing ring 34 similar to the balancing ring 24 on impeller 18 except that the diameter D_4 of the balancing ring is greater than the diameter D_3 of the impeller ring so that there is a considerable upward hydraulic thrust imparted to the impeller shaft. The uppermost stage 33 is provided with a pressure balancing chamber 36 which communicates with the suction to stage 35 through conduit 37. The result of this construction is, that in addition to the hydraulic downward thrust of the upper stages 11, the weight of the pump shaft between the uppermost pair of stages 11 and the next lower pair 10 is compensated and taken up by the hydraulic balance, so that it does not have to be taken up by the thrust bearing. By properly proportioning the areas of the balancing ring 34 to the impeller ring the upward hydraulic thrust can be made so great that the resultant downward thrust on the axial thrust bearing in the motor 1 can be reduced to a very small force with a consequent material reduction in the size of the thrust bearing.

Each of the intermediate pump series 9 and 10 is balanced in a manner similar to the uppermost series 11 so as to provide an upward hydraulic thrust sufficient to greatly diminish the load created by the downward thrust of the next series above together with the load created by the weight of the pump shaft. Although it would be possible, by the balancing method herein described, to give each series of impellers an upward hydraulic thrust great enough to place the shaft section next above under compression, this would

be undesirable because it would tend to buckle the shafts. Instead, it is preferable to provide only sufficient upward hydraulic thrust to leave a slight downward total thrust on each shaft section, thus keeping the shaft sections in tension.

In the particular example described above for a pump to handle 8000 G. P. M. against an 800 foot head, the reduction in load due to dividing the pumping unit into a series of separate units and reducing the shaft diameter between series results in a reduction of the axial load on the thrust bearing from about 75,000 lbs. to about 55,000 lbs., a saving of about 10 tons.

The unbalanced hydraulic load created by a pump of this size amounts to about 85,000 lbs. By dividing the pump into a series of units and balancing the hydraulic thrust in each series, I can reduce the unbalanced hydraulic load to any desired amount, which in the present instance would be about 3,000 lbs.

Thus the total unbalance load on the thrust bearing in the pump described above would be about 58,000 lbs., whereas in a pump of the usual construction (having the pump at the bottom of the shaft) the unbalanced load would be 160,000 lbs. The reduction in total load due to the use of the construction described above would be about 102,000 lbs., a very material saving both in first cost and in power requirements.

It is to be understood that while I have described one method of hydraulically balancing the various units of the pump that other well known devices and constructions for accomplishing the same result may be used, and my invention is not to be considered as limited to the particular balancing system described above.

While I have shown the pump driving means as an electric motor directly connected to the top of the pump shaft, it is to be understood that other driving means may be employed, such as a conventional belted head or right angle gear drive, the essential requirements being a driving means provided with an axial thrust bearing for supporting the shaft load.

The balancing device described herein is one example of that class of balances known as cylindrical throttle balances. Such balances are independent of and not affected by the axial adjustment of the pump shaft and impellers.

I claim:

1. In a multiple-stage deepwell centrifugal pump having a rigid shaft, driving means connected to the upper end of said shaft, an axial thrust bearing for supporting the shaft load, a series of centrifugal pumping units spaced apart along said shaft, the shaft diameter being progressively reduced between each of the series of pumping units, and a cylindrical throttle balance ring on a higher stage of each of said pumping units, vent means from said ring back to the inlet of a lower stage of said pumping unit adapted to compensate for slightly less than the weight of the rotating parts of each pumping unit plus the normal downward hydraulic thrust of said unit whereby all portions of said shaft are subjected to a relatively small amount of tension and the load on the thrust bearing is reduced to a predetermined relatively small amount when the pump is in operation.

2. In a multiple-stage deepwell centrifugal pump having a rigid shaft, driving means connected to the upper end of said shaft, an axial thrust bearing for supporting the load on said shaft, a series of multi-stage centrifugal pumping units spaced apart along said shaft, a cylin-

drical throttle balance ring on a higher stage of each pumping unit, vent means from said ring back to the inlet of a lower stage of said pumping unit, said balancing means being adapted to compensate for slightly less than the weight of the rotating parts of said pumping unit plus the normal downward hydraulic thrust of said unit whereby all portions of said shaft are subjected to a relatively small amount of tension and the load on the thrust bearing is reduced to a predetermined relatively small amount.

3. In a multiple-stage deepwell centrifugal pump having a rigid shaft, driving means con-

nected to the upper end of said shaft, an axial thrust bearing for supporting the load on said shaft, a series of two-stage centrifugal pumping units spaced apart along said shaft, a cylindrical throttle balance ring on the higher stage of each of said pumping units, vent means from said ring back to the inlet of the lower stage of said pumping unit adapted to compensate for slightly less than the total normal downward thrust of each pumping unit when the pump is in operation.

ALADAR HOLLANDER.