May 5, 1970 S. H. S. RAUB 3,510,230

INTERNAL SEALS FOR PUMPS WITH ENCLOSED IMPELLERS

Filed April 3, 1968 2 Sheets-Sheet 1

Fig. 1.

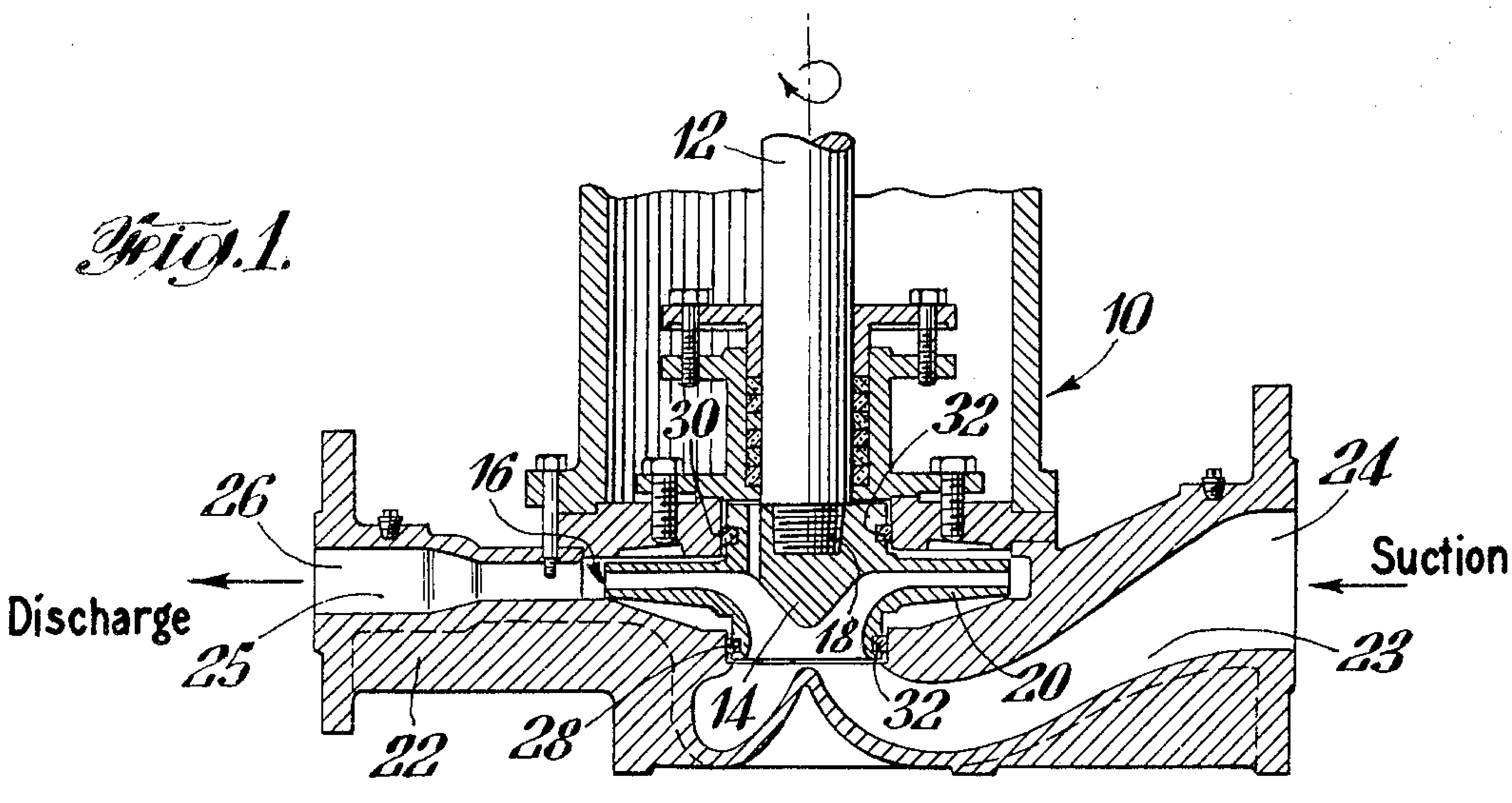


Fig. 2.

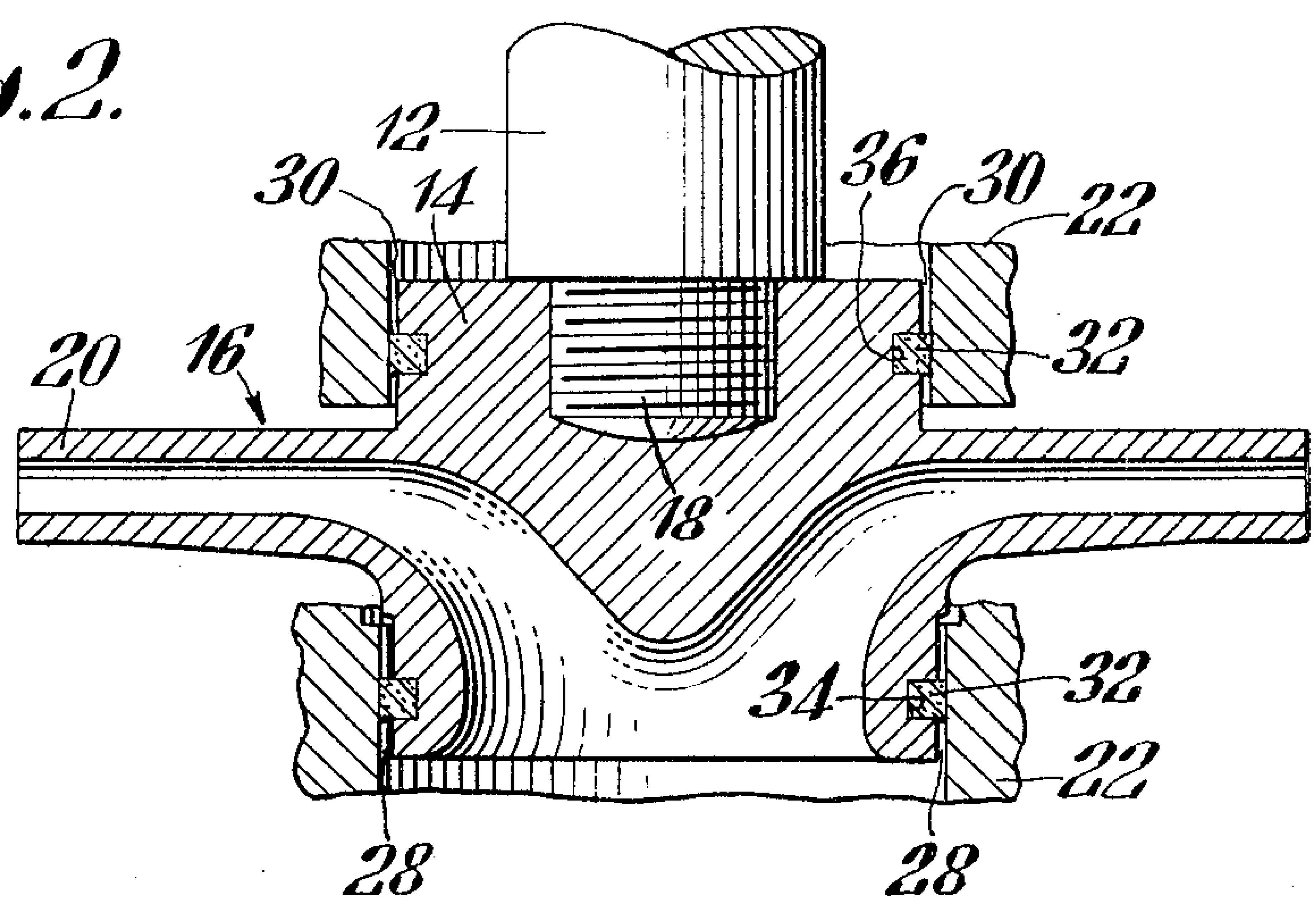
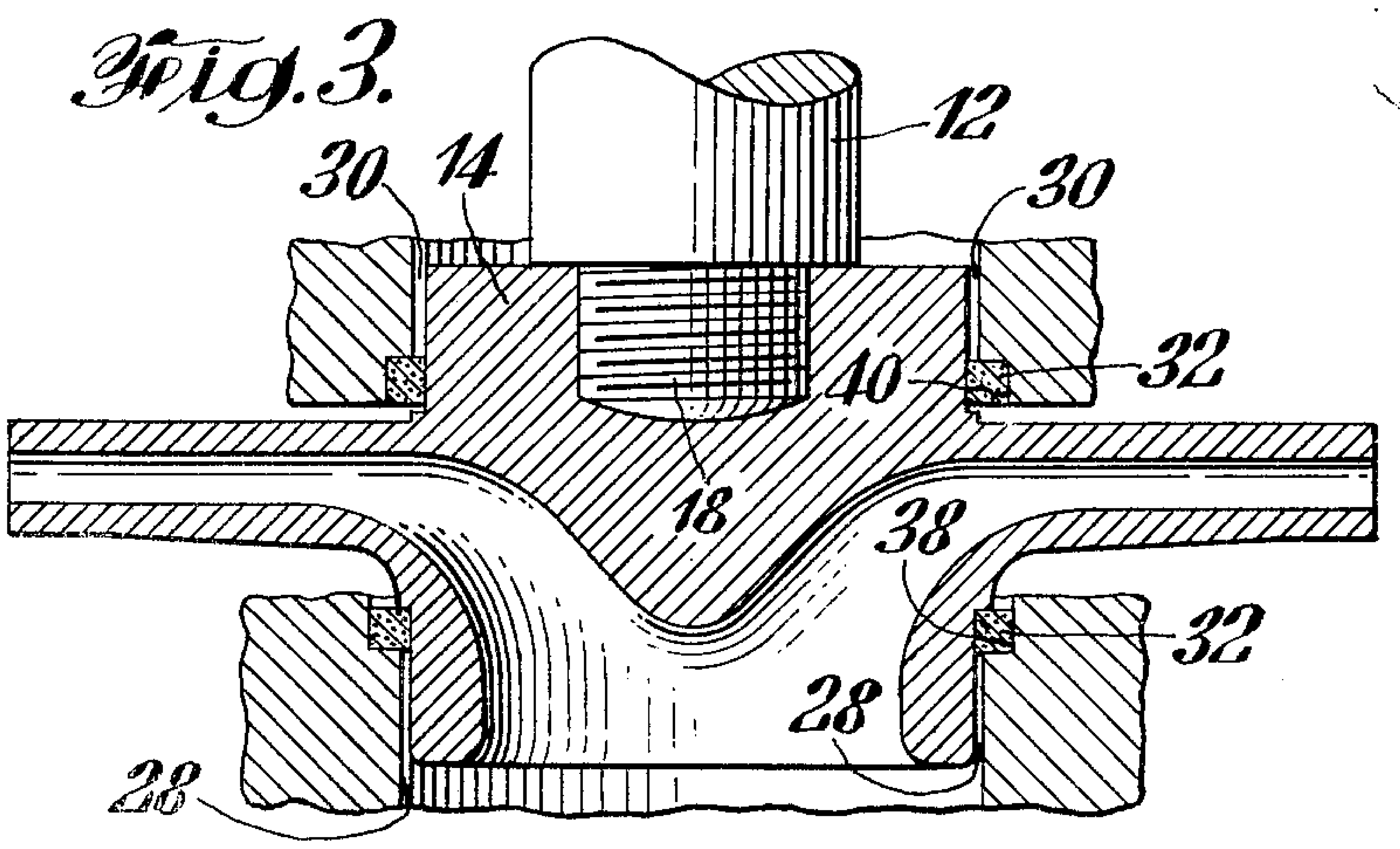


Fig. 3.



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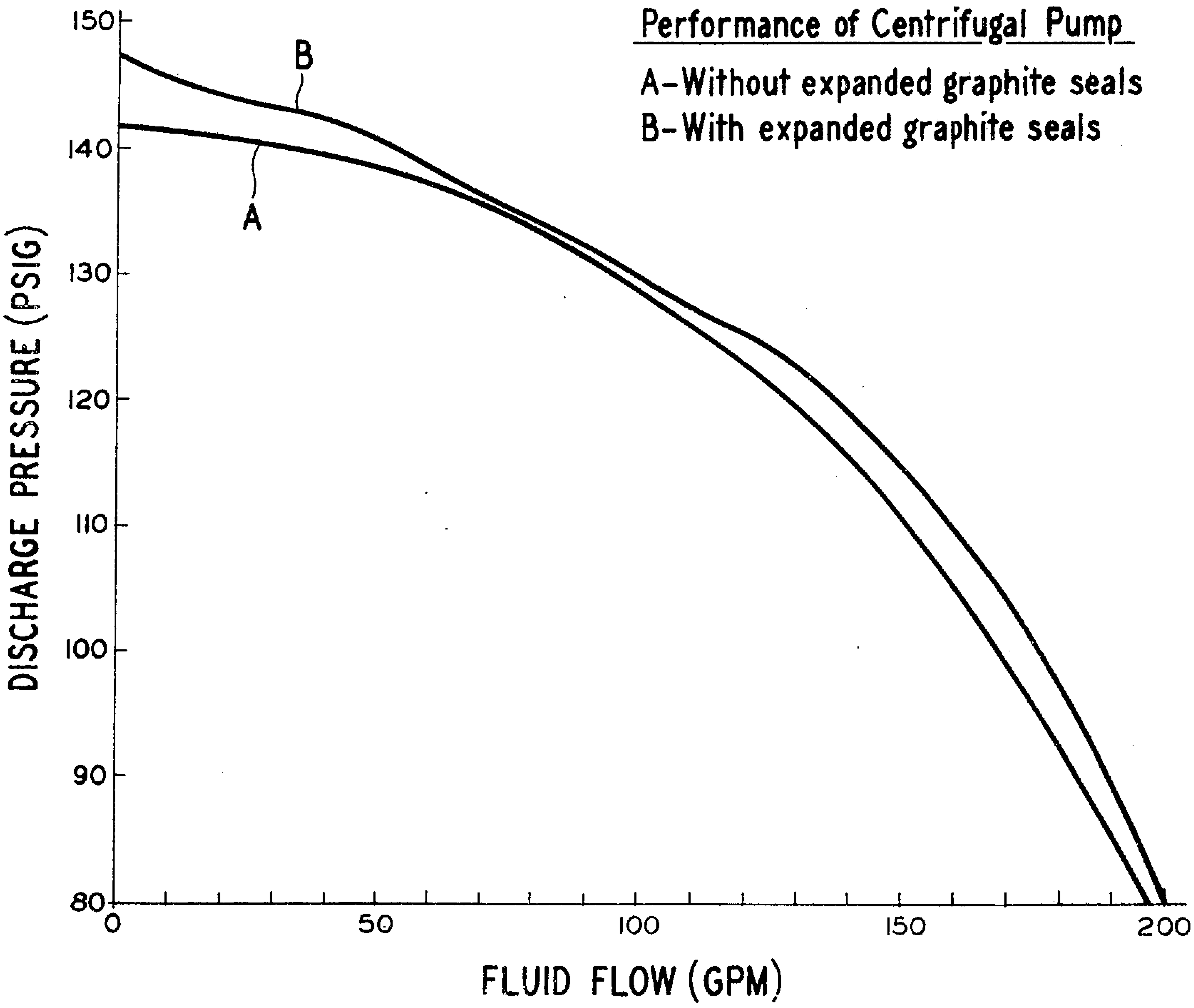

Fig. 4.

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INTERNAL SEALS FOR PUMPS WITH ENCLOSED IMPELLERS

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U.S. Cl. 415—173 7 Claims

ABSTRACT OF THE DISCLOSURE

The fluid leakage in a centrifugal pump is reduced by filling substantially all of the clearance space between the hub segment of the rotating impeller and the pump casing with expanded graphite. The expanded graphite may be positioned on the hub and rotated with the impeller or placed on the stationary casing at the wearing surfaces of the pump. The efficiency of the pump is increased due to the increase in discharge head caused by the reduction of fluid leakage.

FIELD OF INVENTION

This invention relates to an improved seal for preventing internal leakage loss in closed impeller pumps.

DESCRIPTION OF PRIOR ART

A centrifugal pump can be broadly defined as consisting of a vane-carrying wheel or impeller and a stationary casing. The impeller imparts kinetic energy to the fluid which is being pumped while the volute of the casing converts kinetic energy to pressure and guides the fluid away from the impeller. There are several other essential components in this type of pump. These include a shaft which the impeller is secured to and driven by, and a stuffing box or shaft seal which prevents leakage along the shaft in the area where the shaft passes through the casing. In addition, most if not all such pumps are characterized by what are commonly called "wearing surfaces," that is, those surfaces where the rotating and stationary parts are machined close together to prevent fluid leakage from the high pressure side to the low pressure side of the impeller. A centrifugal pump which is typical of the type herein described is manufactured by Union Carbide Corporation in Hastelloy[1] alloys.

The leakage loss at the wearing surfaces inversely affects the efficiency of the pump. Thus, it is important to reduce this to a minimum if maximum efficiency is to be accomplished. A wide variety of structural devices and modifications have been employed for this purpose but the improvements achieved have not provided maximum efficiency nor have they been simple in design. If the clearance between casing and impeller hub is minimal and rubbing occurs, abrasion, galling, and/or corrosion due to the presence of a corrosive fluid will quickly increase the clearance space thereby negating any temporary improvement in efficiency. In closed impeller type pumps reduction in leakage loss has been partially corrected by limiting the leakage path rather than by controlling the clearance area. Structural devices which have been used for controlling internal leakage are a labyrinth engagement, circular or spiral grooves, and the like. None of these devices provide either an interference or a constant rubbing relationship between relatively moving parts, even though the metal inserts into which these devices are placed are called "wear rings."

Notwithstanding many attempts to solve the leakage problem in rotary pumps, no universally satisfactory method of sealing the moving surfaces or compensating for wear has been developed.

[1] Hastelloy—A registered trademark of Union Carbide Corporation.

DESCRIPTION OF THE INVENTION

It is the primary object of this invention therefore, to reduce the leakage loss of this type to a minimum and to increase the efficiency of operation of a centrifugal pump.

Broadly, the invention comprises the controlled positioning of a flexible, wear resistant, thermally conductive resilient expanded graphite material in the normal clearance space between certain moving and non-moving surfaces in a centrifugal pump so that there is virtually no clearance space for internal leakage during the operation of the pump. Because of the inherent properties of the expanded graphite, wear and corrosion will be effectively resisted and the seal will remain intact for long durations of time. If the graphite is placed on the rotating hub, the centrifugal force developed is exerted on it during rotation and presses it against the casing to provide an excellent seal. The resilience and natural lubricity of expanded or fluffy graphite permits it to be flexible enough while being so pressed to withstand abrasion. Equally important improvements in efficiency are experienced if the graphite is placed on the casing such that it extends across the clearance space to the impeller hub. In the preferred embodiment of the invention the expanded graphite is mounted on the hub.

Graphites are characterized in their internal structure by layer planes of hexagonal networks of carbon atoms. These layer planes are substantially flat and are generally parallel to and equidistant from each other. They are bonded together by weak van der Waals forces which as it has been discovered, can be attacked with certain materials to further weaken the bond between layers. The result of such an attack is that the spacing between the superposed carbon layers can be increased so as to effect a marked expansion in the direction perpendicular to the layers, that is in the "C" direction, thereby forming expanded graphite particles. The particles may then be subsequently treated to form a useable product.

In U.S. Pats. 1,137,373 and 1,191,383 natural graphite particles are expanded by first subjecting the graphite particles for a suitable period of time to an oxidizing environment at a suitable temperature. Upon completion of the oxidizing treatment, the soggy particles are washed with water and then heated to a temperature of between about 350° C. and 600° C. to expand the particles in the "C" direction. The particles are thus expanded up to 25 times their original "C" direction dimension and are then combined with a phenolic resin and molded into desirable shapes.

It has recently been discovered that particles which have been expanded at least 80 times and preferably 200 times in the "C" direction can be compressed together in the absence of a binder to form a cohesive sheet, paper strip, foam or the like. The product formed can be made to have a density of from 5 pounds per cubic foot or less to 137 pounds per cubic foot by varying the pressure during the forming process. This unique material has excellent flexibility, good strength and an appreciable degree of anisotropy. A full disclosure of this material and the method of making it is set forth in U.S. Pat. 3,404,061 issued to James H. Shane et al. In addition, this material is commercially sold under the trade name "Grafoil," a registered trademark of Union Carbide Corporation.

For the purposes of this invention, therefore, expanded graphite is intended to encompass within its meaning graphite which has been formed by expanding a graphite starting material which may be natural graphite, pyrolytic graphite, Kish flake graphite or any other type, and then recompressing the expanded material. The preferred expanded graphite contains no binder as taught in the aforementioned copending application but may contain a binder if the amount of binder content is less than ten percent by weight of the graphite product. Greater quantities binder impart a rigidity to the graphite which lessens its ability to withstand abrasion and reduces its flexibility. It is understood that other materials may be added to the expanded graphite either before or after it is molded to size provided that the important properties of resilience and lubricity are not seriously adversely affected. Metal powders or filaments, fibrous reinforcing materials such as fiber glass, clay and the like may well be included to reinforce or strengthen the compressed product or to improve the thermal conductivity thereof.

DESCRIPTION OF THE DRAWINGS

The invention will be further explained and described in conjunction with the drawings, wherein:

FIG. 1 is a plan view in cross section of a typical centrifugal pump embodying the principles of this invention;

FIG. 2 is a plan view in cross section of a segment of the pump shown in FIG. 1 including one embodiment of the seal structure of the invention;

FIG. 3 is a plan view in cross section of a segment of the pump shown in FIG. 1 including another embodiment of the seal structure of the invention; and

FIG. 4 is a graph comparing the performance of a pump with and without the seals of the invention.

Referring to FIG. 1 there is illustrated a centrifugal pump designated generally by the numeral **10**. The pump includes a shaft **12** which is secured to the hub or center part **14** of an impeller **16** by connecting means such as the threaded section **18**. The impeller **16** includes vanes such as at **20** and the entire assembly is enclosed within a casing **22**. The casing **22** is provided with an inlet opening **24** and a discharge nozzle **26** which are in direct contact with channels **23** and **25** respectively. Wearing surfaces, that is, those diametrically opposite areas where the casing **22** is machined close to the hub **14** are shown at **28, 30**. Immediately adjacent each of these wearing surfaces, expanded graphite **32** is positioned so as to seal virtually all of the space between the members.

In operation, a fluid enters through the opening **24** and channel **23** to contact the impeller vanes which are being rotated through shaft **12** by motor means (not shown). The vanes draw in the fluid through suction nozzle **24** and channel **23** and discharge through channel **25** and nozzle **26**. The fluid is prevented by the expanded graphite seals **32** from recirculating past the wearing surfaces and thus substantially all of the incoming fluid is forced out of discharge nozzle **26**.

FIG. 2 illustrates quite clearly one means of securing the expanded graphite material to the hub **14**. As there shown, circular grooves **34, 36** are notched into the hub at the wearing surfaces **28, 30** and the expanded graphite is pressed therein. In this embodiment, the graphite rotates with the impeller and rubs against the casing **22**. In this position the graphite should have a density of 60 pounds per cubic foot or more for maximum life and sealing ability while rotating. Furthermore, the grooves are preferably slightly rounded at the edges to minimize the possibility of damage to the graphite.

In FIG. 3, the graphite seals **32** are positioned in the stationary casing **22** in grooves **38, 40** and are contacted by the hub **14** at the wearing surfaces **28, 30** as the impeller rotates. The graphite can have a density of less than 50 pounds per cubic foot if positioned on the casing since it is to be made to undergo compression as a packing during rotation of the impeller rather than expansion as it would if secured to the impeller itself.

In order to test the effectiveness of the invention a series of experiments were run with a centrifugal pump similar to that shown in FIG. 1. The impeller measured 8½ inches in diameter and was fully enclosed. The inlet opening was 3 inches in diameter and the discharge orifice

was 1½ inches in diameter. The clearance space at the hub wearing surfaces was .025 inch without inserting expanded graphite. In the first test expanded graphite seals were used and water was passed through the pump at various velocities, and the discharge pressure, suction pressure and input power were recorded for each velocity. Table 1 indicates the data recorded.

TABLE 1

Fluid flow, gallons per minute	Discharge pressure, p.s.i.g.	Suction pressure, inches Hg	Power input, kilowatts
0	143	−.5	7.5
20	142	−.6	8.125
40	141.5	−.8	9
60	138.5	−1.25	9.087
80	134	−1.85	10.75
100	129	−2.55	11.75
120	123	−3.45	12.52
140	115	−4.45	13.5
160	105.5	−5.65	14.5
180	95	−6.8	15.25
200	80	−8.3	16.05

The same test was carried out with the same pump but expanded graphite laminated rings 3½ inches outside diameter by ¼ inch thick having no binder contained therein and having a density of approximately 65 pounds per cubic foot were included. The rings were split and placed in ¼ inch by ¼ inch grooves on the hub of the impeller as shown in FIG. 2 and completely sealed the clearance spaces between hub and casing. Data was again recorded for various velocities and is shown in Table 2.

TABLE 2

Fluid flow, gallons per minute	Discharge pressure, p.s.i.g.	Suction pressure, inches Hg	Power input, kilowatts
0	148	−.6	7.00
20	146	−.6	7.38
40	144	−.8	7.75
60	138	−1.2	8.00
80	135	−1.8	10.00
100	130	−2.6	10.75
120	126	−3.4	12.00
140	118	−4.4	13.00
160	110	−5.8	14.00
180	99	−7.0	14.62
200	84	−8.6	15.88

A comparison of the two tables indicates clearly that the use of expanded graphite in the manner disclosed herein imparts a greater operating efficiency to the pump. The greater efficiency results from a combination of the lower power input requirements and higher head (discharge pressure) at identical fluid flow rates.

FIG. 4 graphically represents the marked improvement which is set forth in the foregoing tables by means of a plot of the discharge pressure (ordinate) versus the fluid flow rate (abscissa) for the pump with and without the use of expanded graphite. Curve A is a plot of the data set forth in Table 1 (expanded graphite not employed) and Curve B is a plot of the data set forth in Table 2 (expanded graphite employed). As can be observed, Curve B shows an increase in discharge pressure over substantially all of the fluid flow range. The increase in discharge head at zero fluid flow measured outside the pump clearly indicate the effective leakage loss which is greatly reduced within the pump when graphite seals are used.

Since the horsepower output of a pump is directly related to the product of discharge pressure and the fluid flow of the pump, any increase in the discharge head, only, will result in an increase in output horsepower. Thus the use of expanded graphite seals provides an increase in output horsepower as well as discharge pressure. As shown in Table 2, the input power is actually decreased at given rates of flow when expanded graphite seals are employed. The increase in output horsepower coupled with a smaller input power indicates the significant increase in efficiency which can be expected when expanded graphite seals are used. An increase of as much as 10%

in efficiency was achieved in the aforementioned test, for example.

A variety of modifications of the invention can be made to facilitate the incorporation of the graphite into a centrifugal pump. For example, when the expanded graphite is inserted into the casing, or hub, it can be provided with facing rings of a more impervious material which can be employed as a retaining ring for the graphite. Furthermore, expanded graphite can be inserted into the impeller hub or the casing in one piece of continuous axial length or can be broken into segments of shorter length with spacers positioned between segments. Other modifications, such as spring loading the facings rings, if employed, will be obvious to those skilled in the art.

It will be appreciated that the present invention satisfies a much needed demand for a sealing means which effectively reduces the clearance space between rotating parts in a pump. The success of this invention is particularly important in view of the very high velocities achieved by the rotating impeller during the operation of the pump.

What is claimed is:

1. In a centrifugal pump having at least one impeller enclosed within a casing, said impeller having a hub segment, the improvement which comprises an expanded resilient graphite material positioned between the hub segment of said impeller and said casing such that during the rotation of said impeller the clearance space between said hub segment and said casing is substantially completely sealed with said expanded resilient graphite material, said material comprising a mass of expanded resilient graphite particles compressed together in the absence of a binder, said expanded resilient graphite particles prior to compression having a *c* direction dimension which is at least 80 times that of the graphite particles from which said expanded resilient graphite particles are formed.

2. The pump of claim **1** wherein said expanded graphite is secured to said impeller hub and rotates therewith.

3. The pump of claim **2** wherein said hub is provided with at least one groove and said graphite is placed therein.

4. The pump of claim **3** wherein said expanded graphite has a density of greater than 60 pounds per cubic foot and contains less than ten percent by weight of a binder material.

5. The pump of claim **1** wherein said expanded graphite is secured to said casing and is positioned such that during rotation of said impeller, said hub is in direct contact with said graphite.

6. The pump of claim **5** wherein said casing is provided with at least one groove at the wearing surface of said pump and said graphite is pressed into said groove.

7. The pump of claim **6** wherein said graphite has a density of less than 50 pounds per cubic foot and contains less than ten percent by weight of a binder material.

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HENRY F. RADUAZO, Primary Examiner

U.S. Cl. X.R.

103—103; 277—237