Aug. 7, 1962 E. P. FRANZEN ET AL 3,048,117

PUMP WITH FLOW-RESTRICTIVE ORIFICE

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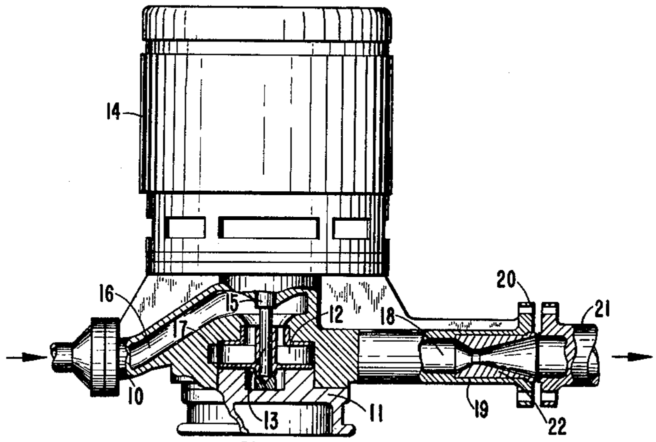

FIG. 1

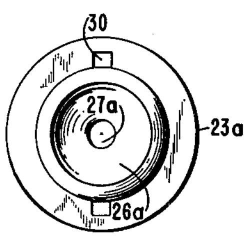

FIG. 4

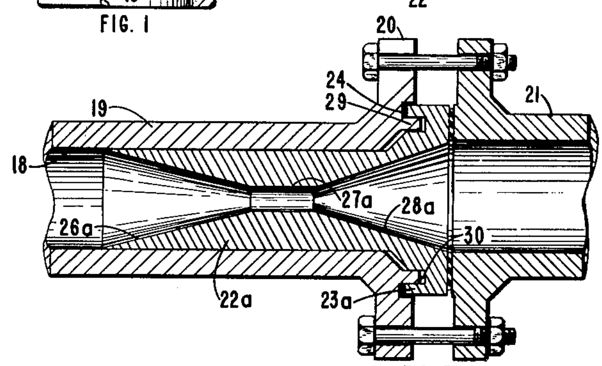

FIG. 3

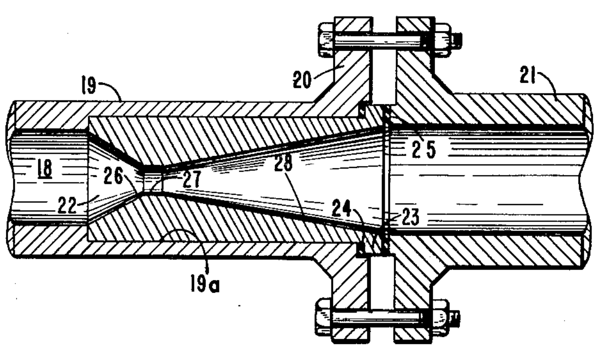

FIG. 2

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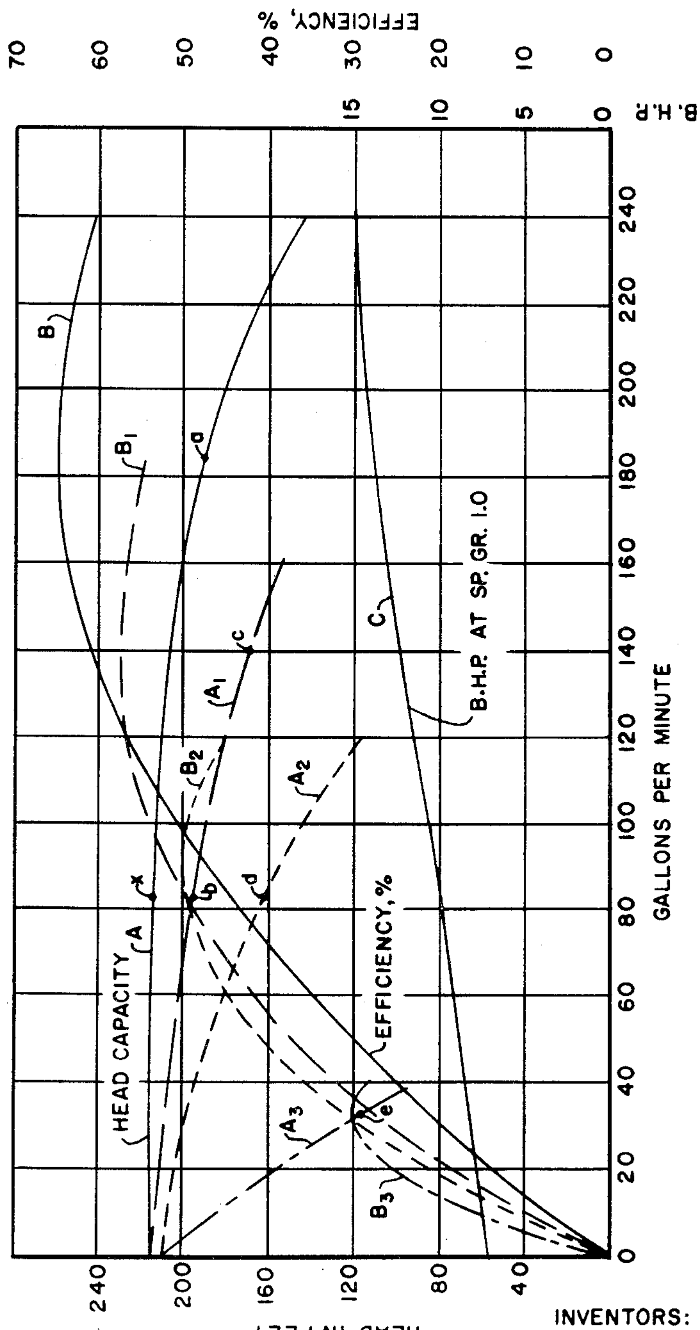

FIG. 5

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PUMP WITH FLOW-RESTRICTIVE ORIFICE

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This is a continuation-in-part of our application Serial Number 688,307 filed October 4, 1957, now abandoned.

The invention relates to centrifugal pumps and is particularly concerned with broadening the range of applicability of such pumps to meet different specific pump performance requirements and to protect the driving motor against being damaged by overloading.

A large variety of pumps, designed to meet a variety of specific pump characteristics as required in different units of a plant, are commonly employed in chemical, petroleum and other industries. By pump performance characteristics is meant the relation of the rate of liquid delivered by the pump to the back-pressure, usually expressed as head. Despite a recognition of the desirability of standardizing centrifugal pumps, the trend has been to install a multiplicity of different pumps, each conforming more or less to the required characteristic. This has entailed high capital costs due to the need to design, procure and install each pump as a special item, high costs in stocking spare parts for maintenance replacement of a large variety of pumps and their components, and high maintenance costs.

In selecting centrifugal pumps engineers have heretofore laid primary emphasis on operating efficiencies and have, therefore, specified a wide variety of pumps to meet different operating requirements. This practice has overlooked other considerations which often produce greater costs than marginal economies in power consumption, among which are capital costs and pump maintenance. The present invention, while in part concerned with efficiencies, departs from such prior practice by effecting significant economies through standardization of pumps throughout a plant.

It is, of course, possible to employ an oversized pump and to place a throttling valve in the pump discharge to reduce the flow; this is, however, not usually a satisfactory solution for the reasons that it is difficult to adjust such valves to the correct point and that it is in practice necessary to use a large and, hence, costly motor to drive the pump at full capacity when throttle valve is open. The latter necessity arises from the danger of damaging a motor, such as an electric motor, of smaller size if the throttling valve is by some oversight opened too far, with the consequence that the pump runs faster and consumes power at a rate in excess of the rated power of the motor.

It is known to construct centrifugal pumps with restricted discharge passages to attain specific pump performance characteristics. Each such pump is, however, specifically designed for a particular pump performance characteristic and hence presents the objectionable consequences previously noted.

It is, therefore, the broad object of this invention to provide a centrifugal pump that can be easily altered to meet a specific pump performance characteristic, so that one or a small number of different pumps can serve a large variety of requirements in a plant having different pump requirements.

A further primary object is to protect the electric motor against damage when it has a size insufficient to operate its centrifugal pump at full capacity, the protection being provided by an orifice device in the pump discharge which prevents operation of the pump at an output rate which would damage the motor.

A specific object is to provide a centrifugal pump hava removable, fixed flow-restrictive device for adapting the pump to a desired pump performance characteristic which device is easily installed in the pump. Ancillary thereto, it is an object to provide an arrangement for mounting the flow-restrictive device in such a manner that the pump cannot be coupled to the discharge pipe in a leak-proof manner without interposition of the said device, whereby the danger of operating the pump at a load in excess of the capacity of the motor through inadvertent omission of the flow-restrictive device is obviated.

A further specific object is to improve the efficiency of the combined pump and flow-restrictive device relative to the efficiency to be expected when conventional restrictive devices are employed.

In summary, according to the invention the pump discharge passage is fitted with a removable orifice device so that the pump can be given a desired performance characteristic, such device being preferably a sleeve having a length in excess of twice the minimum internal diameter thereof and advantageously having a convergent entrance section. In the preferred embodiment the sleeve is shaped to provide a venturi-shaped passage, which includes both convergent and divergent sections, for improving the pump efficiency.

The principle involved in the invention will become apparent from the following detailed description, taken in connection with the accompanying drawings which form a part of this specification and show two illustrative specific embodiments, wherein:

FIGURE 1 is an elevation view of a pump installation according to the invention, parts being shown in section;

FIGURE 2 is an enlarged, fragmentary longitudinal view of a part of FIGURE 1;

FIGURE 3 is a fragmentary longitudinal view of an alternative embodiment of the flow-restrictive device;

FIGURE 4 is an end elevation of the removable orifice device shown in FIGURE 3; and

FIGURE 5 is a diagram showing certain pump characteristics and the effects thereon of the flow-restrictive devices according to the invention.

Referring to FIGURES 1 and 2, the centrifugal pump includes a casing having an outer section **10** and a bottom closure **11** and containing an impeller **12** mounted on a vertical shaft **13** which is the output shaft of an electric motor **14** and is journalled in the motor. The motor is, in the illustrative embodiment, mounted directly on the pump casing and the impeller is keyed to the shaft, which carries a seal sleeve **15**. The pump and motor can be mounted on a support, not shown, which may be a pedestal acting through the closure **11** or through a hanger. The pump is shown schematically, and certain elements, such as wear rings and additional mechanical seals, which are conventional and which would be included in practice, are not shown. The impeller has a central inlet or suction eye to which liquid is supplied from an inlet passage **16** and discharges the liquid at the periphery into a volute chamber **17** which leads into a discharge passage **18** formed in a nozzle **19**, which is advantageously integral with the casing section **10**. The nozzle includes a bolting flange **20** by which it is coupled to a discharge pipe **21** which advantageously has a diameter at least as great as that of the discharge passage **18**, as shown. The outer end of the passage within the nozzle is enlarged or counter-bored at **19*a*** to receive a flow-restrictive orifice device **22**.

The device **22** is formed as a sleeve and has a flange **23** at the outer end thereof which is pressed against a gasket **24** fitted in an annular recess in the flange face; the outer face of the flange **23** seats against a gasket **25** which seals the flange to the discharge pipe **21**. The sleeve contains a restricted passage which is, in accordance with the preferred arrangement, shaped as a venturi and includes a convergent section 26, a throat 27 and a divergent section 28 which widens gradually and has an included angle less than about 20°, e.g., about 14° as shown. To minimize shock and improve the efficiency of the pump it is desirable that the entrance and discharge sections of the venturi passage merge smoothly with the discharge passage 18 and the bore of the discharge pipe 21, respectively.

It is advantageous to position the sleeve 22 so that the inner end thereof is spaced downstream from the impeller, that is, displaced from the cut water or volute throat where the passage 18 emerges from the impeller chamber. This facilitates an accurate design of the pump casing with a minimum clearance between the running edge of the impeller and the cut water or volute throat to give the maximum head, capacity and efficiency, with minimum hydraulic loss. If the sleeve 22 were extended to or close to the impeller it would be impracticable to achieve such a small clearance, particularly when a sleeve or circular cross section is used. It is advantageous to displace the sleeve from the impeller by a distance at least as great as the internal diameter of the passage 18.

Because an understanding of the pump characteristics is fundamental to an understanding of the principles of the invention they will be summarized in connection with FIGURE 5, which shows characteristics of a centrifugal pump of the type shown having a $7\frac{9}{16}''$ diameter impeller, a 3″ diameter suction inlet and a 2″ diameter discharge passage, operated at a constant speed of 3550 r.p.m. with a liquid having a specific gravity of unity. The discharge rates, in gallons per minute, are plotted as abscissae against three separate ordinate scales representing head, efficiency and brake horsepower.

The three solid lines show the pump performance at the highest capacity, i.e., when fitted with the largest of a series of flow-restrictive devices. This largest device had a throat or orifice diameter of $1\frac{1}{4}''$. The curve A gives the relation of the head, measured at the end of the discharge nozzle, to the discharge rate; the curve B indicates the efficiency for each discharge rate; and the curve C the brake horsepower applied to the impeller for each discharge rate. It is evident that the brake horsepower increases with the discharge rate being, for example, 14 H.P. at a discharge rate of 184 g.p.m. when the pump works against a head of 190 feet, at an efficiency of about 65%. This operating point is indicated by the point *a*. This pump is obviously large enough to pump liquid at smaller heads and/or discharge rates, such as those shown at points *b*, *c*, *d* and *e*, which lie below the curve A; but it is necessary to alter the pump performance characteristics to attain these head-rate relations. For example, it is not possible, without throttling the discharge, to deliver 83 g.p.m. against heads less than those of 212 feet (denoted at *x* on curve A), such as against heads denoted by the points *b* and *d*.

Considering the latter example in greater detail, suppose that the pump is to deliver 83 g.p.m. against a head of 162 ft., denoted by the point *d*. When the pump is operated against this head it would accelerate until the head-rate relation conforms to the curve A; thus, if the head remained constant the pump would deliver about 202 g.p.m., and draw almost 15 H.P. from the electric motor. The delivery of 83 g.p.m. against 162 feet would require only 10 H.P.; however, if a 10 H.P. electric motor were installed and the pump discharge were left unthrottled, the motor would be damaged because the pump would accelerate, overloading the motor until failure. While the pump delivery rate can be reduced by a throttling valve, whereby the head at the pump discharge is increased to 212 ft., this is not practicable because of difficulty in adjusting such a valve and the danger that a workman may inadvertently leave the valve open, leading to overrunning of the pump, consumption of power in excess of the capacity of the motor, and failure of the latter.

Now, in accordance with the invention, this danger is avoided while the pump performance characteristics are modified to meet any desired requirement by placing into the pump discharge passage a sleeve 22 having an appropriate orifice size. FIGURE 5 shows curves A_1, A_2 and A_3, corresponding to the curve A but for the same pump having orifice devices according to the invention, in the form of sleeves with throats respectively of 1.0″, 0.875″ and 0.50″ diameters. Curves B_1, B_2 and B_3 show the corresponding efficiencies, determined on the basis of the head immediately downstream of the venturi constriction. It will be seen that the curve A_2 passes through the point operating *d*, which corresponds to one desired operating point.

When operating at the point *d*, at a rate of 83 g.p.m., against a head of 162 feet, the efficiency is about 49%, so that there is a waste of some power when compared to a peak efficiency of 65% for operating point *a*. This waste is, however, small in terms of absolute power expenditure because the total power involved is small, and a part of this waste can be reduced as explained below; moreover, it represents an expenditure which is small in relation to the savings in installation and maintenance. It may be noted that while this approach involves the use of a pump which has a larger casing and impeller than would be required for operating at the operating point *d*, such a pump in practice costs the same or less than a smaller pump due to the savings incident to standardization. However, electric motors are commercially available in different sizes with commensurate savings, and it becomes feasible with this invention to employ a motor of the size required for any desired delivery rate in accordance with the curve C, that is, considerably smaller than 14 H.P., because the danger of overloading the motor is obviated.

Three additional features of the invention should be noted. The first concerns the use of a flow-restrictive device which is tubular and has a length at least twice as long as the minimum internal diameter, especially one having a convergent entrance section. The convergent section reduces random turbulence and shock while the length minimizes changes in flow characteristics with use. This permits the use of a cheaper metal, having a lower hardness, than would be the case if a simple orifice plate were used.

The second additional feature relates to the use of a venturi which includes both a convergent and a gradually divergent section. This was unexpectedly found to improve the pump efficiency. Thus, considering the pump equipped with an orifice device 22 having a 0.875″ diameter to attain the operating point *d*, it would be expected that the head immediately upstream of the constriction would be as indicated by the point *x*, and that the power input to the impeller would remain at 10, indicated by the curve C. However, it was determined in actual tests that the power input when operating at the point *d* was actually somewhat lower than 10 H.P., resulting in a saving in power requirements. In other words, the venturi constriction reduced the discharge rate not merely by dissipating a fraction of the head but by also reducing the power input to the impeller to a level below that predicted by the curve C by a physical effect which is not understood but which may involve some diffusion phenomenon. In one series of tests the power input reduction was of the order of 2–5%.

Thirdly, it should be noted that by using the orifice restrictive device and a pump casing and impeller larger than otherwise necessary for a given pump characteristic, it is easier to construct the pump casing with minimum clearance between the running edge of the impeller and the cut water or volute throat. In the majority of pumps used in chemical plants of the horsepower ranges from 3 to 25, it is too costly to procure specially casted and machined volutes having the minimum clearance required for maximum efficiency. It is at the cut water point that the conversion of velocity energy into head energy occurs and that the greatest hydraulic loss occurs. The improved efficiency, resulting from the reduced clearance (made feasible according to the invention due to the increased size and the standardization to a fewer number of sizes) in practice further compensates for the efficiency loss suggested by the curves B_1, B_2 and B_3.

Referring to FIGURES 3 and 4, there is shown a modified construction wherein the orifice device **22**_a_ is a tubular sleeve having a convergent entrance section **26**_a_ and an elongated passage **27**_a_ of uniform diameter, the length being at least twice its diameter. The sleeve may, optionally, also include a divergent section **28**_a_. These views further show an optional feature to guard against inadvertent omission of the sleeve, including lugs **29** on the face of the flange **20**_a_, disposed radially inside of the gasket **24**, which enter recesses **30** in the flange **23**_a_ of the sleeve. The discharge pipe **21** has no such recesses, making it impossible to couple it to the pump discharge nozzle in fluid-tight relation without interposition of the sleeve.

We claim as our invention:

1. A pump installation including: a centrifugal pump including a volute casing with intake and discharge passages, and an impeller within said casing, said discharge passage having a section of substantially uniform diameter extending outward from said impeller; an electric motor drivingly connected to said impeller, said motor having a power rating less than that necessary to operate the pump at full capacity against a preselected head when said discharge passage is unrestricted; and means for protecting said motor against overload, said means including a removable, fixed flow-restrictive device situated in said discharge passage at the downstream end of said section of uniform diameter and displaced outwardly from the pump impeller by a distance at least as great as the diameter of said section for producing a desired pump performance characteristic, said removable, fixed flow-restricted device reducing the discharge capacity of the pump whereby said means protects said motor against overload at said preselected head.

2. A pump installation according to claim 1 wherein said flow-restrictive device is a sleeve which provides a restricted flow passage having a length which is in excess of twice the internal diameter thereof.

3. A pump installation according to claim 1 wherein said flow-restrictive device is a sleeve containing a flow-passage shaped as a venturi, said flow-passage including a convergent section which merges smoothly with said uniform-diameter section of the discharge passage of the pump casing and a divergent section, the latter having an included angle less than 20°.

4. A pump installation according to claim 1 wherein said flow-restrictive device is a sleeve containing a flow-passage which converges outwardly from said uniform diameter section of the discharge section of the pump casing.

5. A centrifugal pump including a volute casing containing an impeller, said casing having an intake passage and a discharge nozzle providing a discharge passage having a uniform diameter for a substantial distance, said nozzle having coupling means at the discharge end thereof; a removable flow-restrictive orifice device in the form of a sleeve fitted within said discharge passage in engagement with the nozzle wall, said sleeve having a central flow passage which is substantially restricted in relation to the said nozzle discharge passage for producing a desired pump performance characteristic against a preselected head and limiting the power consumption of said impeller and the innermost part of said sleeve being displaced outwardly from the impeller by said uniform-diameter passage for a distance at least as great as the internal diameter of said discharge passage; and a discharge pipe secured directly to said coupling means and mounted in engaging relation to said sleeve to urge the sleeve into the nozzle passage, said nozzle and sleeve having aligned abutments which limit the inward movement of the sleeve.

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