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Studies on the Volute Pump Handling Viscous Fluids

(1st Report, Representation Method of Oil-Pumping Characteristics)*

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Recently there has been a heavy increase in the demand for volute pumps handling viscous fluids for chemical industries. It is suggested that the properties of viscous fluid influence the hydraulic performance of a centrifugal pump as compared with action of water, and already a few scholars have reported on this influence. It seems, however, that there remain many problems unsettled as to the pumping behavior of a volute pump handling viscous fluids.

The authors measured the hydraulic performance characteristics of a 75 mm volute pump under various pumping conditions in order to analyze their characteristics. In this paper they describe the method of representing all the properties under various conditions and show the performance characteristic diagrams indicating the influence of viscosity.

Nomenclature

H : head produced by pump
 Q : discharge
 L : input horsepower
 N : r. p. m. of impeller
 ω : angular velocity in radians per sec.
 D_1 : diameter of entrance section of impeller
 D_2 : diameter of exit section of impeller
 b_1 : width of entrance section of impeller
 b_2 : width of exit section of impeller
 z : number of vanes
 β_2 : outlet angle
 u_2 : tangential velocity at impeller exit
 v_{m2} : meridian velocity at impeller exit
 ϕ : discharge coefficient v_{m2}/u_2
 ψ : pressure coefficient, $H/(u_2^2/g)$
 μ : input power coefficient, $L/(\gamma A_2 u_2^3/g)$
 R_e : Reynolds number, $\omega D_2^2/\nu$
 γ : specific weight of fluid
 ν : kinematic viscosity

1. Introduction

The influence of viscosity on the performance of volute pumps has been studied already by some pump engineers, and many trials⁽¹⁾ for the prac-

tical treatment have been reported. One of the most noteworthy reports, up to this time, is the paper written by A.T. Ippen⁽²⁾, who tested systematically many types of volute pumps and showed the viscosity correction diagrams for centrifugal pumps in maximum efficiency point based on the experiments. It seems, however, that the phenomena accompanied with the flow of viscous fluids through impeller and volute chamber are so complicated that any attempt to treat it analytically or theoretically is seldom successful in spite of the efforts.

In recognition of the need for analytical and systematical investigation of this problem, the writers, as a first step, measured the hydraulic performance of a 75 mm volute pump with various viscosities up to 1500 centi-stokes and speeds from 1500 r.p.m. to 600 r.p.m. In this paper they describe the method of representing all the properties including maximum efficiency points of a pump under various pumping conditions; that is, the non-dimensionized performance of a pump is represented as function of Reynolds number $R_e = \omega D_2^2/\nu$ only, if the discharge coefficient ϕ is used as a parameter. Up to this time, the representation method or the viscosity correction diagrams are for the maximum efficiency points only, but the proposed method is not restricted to them. In practice any other point as well as the maximum efficiency point are so important

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that the method is considered to be very useful not only for practical use, but for theoretical treatment.

The writers show practically the performance characteristic diagrams for the tested pump and that the influence of viscosity or pumping conditions on the pump performance is represented considerably well.

2. Representation method of the performance characteristics

When a volute pump handles high viscous fluids, the so-called similarity law of pump no longer holds. The performances of the volute pump under various pumping conditions cannot be represented as simply as in the case of water. However when the non-dimensionized performance is plotted against Reynolds number using the discharge coefficient as a parameter, all the properties may be represented fairly well in a diagram. This illustrated as follows.

Assuming that the pump head is given by the following equation

$$H = \frac{1}{1+p} \frac{u_2^2}{g} (1 - \phi \cot \beta_2) - h_f - h_d - h_s \quad (1)$$

where h_f , h_d , h_s are the hydraulic pressure losses due to hydraulic friction, bend or expansion, and shock of flow respectively, these pressure losses may be written as

$$h_f + h_d = k_f \frac{v_{m2}^2}{2g} + k_d \frac{v_{m2}^2}{2g}$$

$$h_s = k_s \frac{(v_{m2} - c_s u_2)^2}{2g}$$

where $c_s = v_{ms2}/u_2$, and k_f , k_d , and k_s are certain coefficients which are considered to be almost constant when the pump handles water. In a general case these are considered to be functions of Reynolds number.

Substituting these into Eq. (1) and non-dimensionizing it, we get as pressure coefficient

$$\psi = \frac{1}{1+p} (1 - \phi \cot \beta_2) - \frac{1}{2} k_f \phi^2 - \frac{1}{2} k_d \phi^2$$

$$- \frac{1}{2} k_s (\phi - c_s)^2 \quad (2)$$

Therefore the pressure coefficient ψ of a pump is the function of k_f , k_d , k_s and discharge coefficient ϕ . When water is handled, ψ is the function of ϕ only, but of ϕ and Reynolds number when viscous fluids are handled. Then ψ is represented as a function of Reynolds number only, if $\phi = \text{const.}$

The shaft horsepower is given by the equation

$$L = \gamma Q H + L_h + L_f + L_v \quad (3)$$

where L_h , L_f and L_v are the powers dissipated by pressure loss, disc friction and leakage flow

respectively. L_h and L_f are represented as follows

$$L_h = \gamma A_2 v_{m2} (h_f + h_d + h_s)$$

$$L_f = C_f \frac{\gamma}{g} \omega^3 \left(\frac{D_2}{2} \right)^5$$

C_f is the so-called disc friction coefficient.

The leakage flow q through wearing ring may be put as

$$q = c_l \frac{\gamma}{g} u_2$$

c_l is the function of Reynolds number and discharge coefficient. Substituting these into Eq. (3) and non-dimensionizing the latter, we get

$$\mu = \frac{\phi + \Delta\phi}{1+p} (1 - \phi \cot \beta_2) + C_f \frac{D_2}{4b_2} \quad (4)$$

where $\Delta\phi = q/A_2 u_2$. Therefore, the power coefficient μ also is the function of discharge coefficient only when water is handled, but is the function of Reynolds number and discharge coefficient when viscous fluids are handled.

Similarly the efficiency η also is represented as the function of Reynolds number and discharge coefficient.

Accordingly it is considered that all the dimensionless performances of a volute pump under various pumping conditions are arranged as a function of Reynolds number only, using discharge coefficient ϕ as a parameter.

In this report we use $\omega D_2^2/\nu$ as Reynolds number; however, this is different from $4mv/\nu$ which is related to the coefficients k_f , k_d , k_s etc. There is simple relation between these, as is mentioned below.

$$R_e' = \frac{4mA_2}{D_2 A} \phi R_e \quad (5)$$

where $R_e = \omega D_2^2/\nu$, $R_e' = 4mv/\nu$. In the same pump it becomes

$$R_e' = c\phi R_e$$

where $c = \text{const.}$

3. Experimental equipment

The schematic arrangement of experimental equipments is shown in Fig. 1. Tested pump is a single suction volute pump with the main dimen-

Fig. 1 Schematic illustration of the experimental equipments

sions as listed in Table 1. The liquids employed for experiments are water and two kinds of oils and the essential data on the viscosities of these oils are given in Fig. 3. Testing was done for nine different viscosities up to 1500 c.st. Any viscosity for these tests is obtained with the temperature of the two oils varied from 50°C to 10°C by the heater and cooler equipped to the oil tank and the temperature deviation was held within 0.5°C during the test. All the pump heads were measured by the mercury-manometer and the discharge was metered by the orifice meter, which was calibrated by weighing method. The flow coefficient of orifice is not constant usually when it is used in viscosious fluids other than water. The calibrated results are shown in Fig. 2, where the flow coefficient $k=Q/(\pi/4)d^2\sqrt{2gh}$, Reynolds number $R=\sqrt{2gh} \cdot d/\nu$.

The pump was driven by a 7.5 horsepower direct current motor, which was able to change the speed from 1800 r.p.m. to 600 r.p.m. by controlling the field current and the source voltage. During testing the speed was always kept constant by controlling the field current manually and the deviation from the specified test speed was less than 1 or at least 2 r.p.m. The speeds were metered by a special electronic tachometer which was devised for this experiment. It was built so that small deviation of speed might be watched on a Brown tube, and the speed was so adjusted that the deviation might be zero. The certainty of this meter is considered to be higher than 10^{-5} .

Table 1 Pump dimensions

D_1	D_2	b_1	b_2	z	N_s for water
94	270	19	8	6	157

Fig. 2 Fluid-meter calibrations

4. Performance curves for water

The experiments began by determining the performance of the pump for water with the speeds varied from 1800 r.p.m. to 600 r.p.m. The experimental results are shown in Fig. 4, where (c), (d) and (e) are the non-dimensionized performance curves as mentioned before. In spite of the so wide range of speeds, the performance curves are represented almost by the only one curve and it is ascertained that the so-called similarity law of the pump holds fairly well. This indicates moreover that the coefficients k_f , k_d and k_s etc. as mentioned in Chapter 2 can be regarded as constant.

5. Experimental results

The test was done for nine different viscosities with varying temperature of oils and the speed from 1800 r.p.m. to 600 r.p.m. with the exception of high viscosity case, where the test was not done at high speed, since the torque exceeded the driving motor capacity.

Some examples of the performance curves are given in Figs. 5~7, where (c), (d) and (f) are the non-dimensionized performance curves. There is difference between these curves for each speed, and this indicates that the similarity law of pump no longer holds. However, when the viscosity is not so high, the similarity law of pump head only holds nearly. But all these performances are not represented simply in general.

As stated already, all these performances are

Fig. 3 Viscosity of oils

represented fairly well if we arrange the non-dimensional performances against Reynolds number using the discharge coefficient ϕ as a parameter. In Figs. 8~10 the performance characteristic diagrams which are obtained by these methods for the test pump are given. Other experimental results also which are obtained by

other investigators may be arranged similarly. From these diagrams the influence of viscosity and running speed on the pump performance can be known considerably well. From the head characteristic diagram it is seen that the pressure coefficient ψ is nearly constant while Reynolds number is larger than 2×10^5 and that the simila-

Fig. 4 Performance curves for water

Fig. 5 Examples of performance curves for oil ($\nu=66.64$ c.st.)

rity law about head holds for the test pump and the increase of pressure loss due to the increase of viscosity is hardly noticeable for larger R_e than 2×10^5 . On the other hand these areas for the input-power and the efficiency diagrams are limited almost to the area of water. Since the leakage loss decreases with the increase of viscosity, it is

considered that the increase of input-power or the decrease of efficiency is caused almost by the horsepower dissipated by disc friction at least in the area of larger R_e than 2×10^5 .

In this experiment a very unstable state was observed when viscosity was 565 c. st. and speeds over 1 200 r. p. m. This state is indicated in head

Fig. 6 Examples of performance curves for oil ($\nu = 245$ c.st.)

Fig. 7 Examples of performance curves for oil ($\nu = 915$ c.st.)

characteristic diagram as discontinuous points corresponding to $Re = 1.9 \times 10^4$. It is hard to understand precisely why the unstable state arises.

The performance characteristic curves for the maximum efficiency point also are plotted in the diagrams with dotted line and then the pumping characteristics under various conditions are shown better.

Ratio of the performances to the water performance also is shown in Figs. 11~13.

6. Conclusion

The writers measured the hydraulic performances of a 75 mm volute pump handling various viscous fluids and showed that the whole dimensionless performances of the pump under various pumping conditions were represented as a function of Reynolds number only if the discharge coefficient was used as a parameter. The pumping behavior of a pump handling viscous fluids can be known easily from the performance characteristic diagrams. For the test pump the similarity law

Fig. 8 Head characteristic diagram

Fig. 9 Power characteristic diagram

Fig. 10 Efficiency characteristic diagram

Fig. 11 Ratio of the head for oil H_o to the head for water H_w Fig. 12 Ratio of the power for oil L_o to the power for water L_w Fig. 13 Ratio of the efficiency for oil η_o to the efficiency for water η_w

about the pump head holds when the Reynolds number R_e is larger than 2×10^5 . On the other hand these areas about input-power or efficiency are limited almost to the area handling water. It is reasoned that the increase of input-power or the decrease of efficiency is caused by the increase of disc frictional loss alone for $R_e > 2 \times 10^5$.

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