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PAPER A1

APPLICATIONS AND PERSPECTIVES OF A NOVEL SPROCKET TYPE PUMP

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SYNOPSIS

A novel positive displacement pump is now being manufactured and used in various industrial applications involving transfering, metering and circulation of a diversity of fluids. Units ranging in capacity from 2.5 l/min to 2000 l/min have been built. The maximum discharge pressures are about $3.5\times10^6 Nm^{-2}$. A pump capable of delivering hydraulic oil at a pressure of $10^7 Nm^{-2}$ is under development. Design aspects and performance are discussed.

NOMENCLATURE

a sprocket radius, cm
b crescent thickness, cm
e eccentricity between drive-disk and sprocket axes, cm
L pin length, cm
n revolutions per minute
N number of pins in drive-disk
p discharge pressure, Nm^{-2}
P power to drive pump, W
Q theoretical volume rate of flow, m^3s^{-1}
r pin radius, cm
R radius of circle circumscribing pins, cm
t sprocket arm minimum thickness, cm
μ dynamic viscosity, cP

INTRODUCTION

A rotary pump of novel design developed in Mexico, has been made available to industry on a commercial basis starting in 1981. Early work on this unit has been described in previous publications [1,2,3]. The first applications of the pump were for the transfer and circulation of non-corrosive fluids. One example is the pumping of hot oil (200°C) heated by solar energy in an experimental installation at the National Autonomous University of Mexico. Another example is the dispensing of diesel fuel from tank trucks. More recently, metering pumps have been built. Also, experience has been accumulated in handling a variety of fluids, such as acids, alkalis, resins, alcohol, cosmetic creams and shampoo, water, milk, and chocolate paste. The latter substance has a viscosity of the order of 60,000cP.

Production units range in capacity from 2.5 l/min to 2000 l/min. It is estimated that the practical limit is about 5000 l/min. With present technology, the maximum working pressure with a typical fluid such as light machine oil, is about $3.5\text{x}10^6 Nm^{-2}$. Currently, research is being conducted to develop a pump for use in hydraulic power systems that will provide oil at a pressure of $10^7 Nm^{-2}$

PUMP MECHANISM

The operating principle of the sprocket pump may be understood by means of the diagram of Fig 1. The drive-disk pins engage the eccentrically mounted sprocket causing it to rotate. As the pins leave the sprocket recesses they enter a pumping corridor separated from the sprocket by a crescent shaped barrier and displace fluid causing suction on the downstream end and elevating the pressure on the upstream end. The relief bays serve the purpose of avoiding trapping of fluid in a variable volume chamber bounded by a pin, the sprocket, and the housing wall.

Sometimes, in order to reduce internal leakage, the external pin circle is reduced by machining in a lathe (resulting in pins which are no longer circular in cross-section) and specifying a narrower pumping corridor. In this manner, surface contact, rather than line contact is achieved on one side of such corridor. The photograph of Fig 2 shows this construction.

DISTINGUISHING CHARACTERISTICS

The sprocket pump produces a discharge free of fluctuations. Another interesting characteristic is that there is no hydrostatic torque acting on the sprocket and, therefore, the forces transmitted to it by the drive-disk pins are very low. However, a resultant radial hydrostatic force is present, which must be resisted by the bearing on which the sprocket is mounted. Also, the drive-disk and shaft are subjected to a resultant axial hydrostatic load which, moreover, acts eccentrically. This point will be discussed later.

It will be noticed that no normal forces are transmitted at the surfaces which provide a seal between the high and low pressure sides of the pump. Consequently, wear at those points is minimal.

DESIGN CONSIDERATIONS

The most important considerations which have guided the design of sprocket pumps will now be commented upon.

Geometric parameters

The size of a sprocket pump is characterized by

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the radius of the circle circumscribing the drive-disk pins, R, and the length of the pins, L. The other parameters needed to determine the basic geometry of the pump are: the pin radius, r; the eccentricity between drive-disk and sprocket axes, e; the sprocket outside radius to the tip of the arms, a; and the number of pins, N. These cannot be prescribed arbitrarily since a number of restrictions must be complied with. These are: (1) the crescent thickness, b, must be sufficient to insure adequate strength, (2) the sprocket arms minimum thickness, t, must likewise be adequately sized for strength, (3) the actions of the pins driving the sprocket must overlap, (4) there must be at all times at least one sprocket arm sealing against the crescent and against the body on the opposite side of the crescent, and, (5) there must be at all times at least one pin in the pumping corridor. An optimization scheme by means of which the volumetric displacement is maximized, is presented in reference [3]. From it, the following values were determined: r/R=0.139, e/R=0.237, a/R=0.823, N=6. This result was based on a specification of lower limits for b/R and t/R of 0.137 and 0.111 respectively, which in turn were obtained from rough stress calculations assuming L/R=0.28 and a pressure rise from suction to discharge of $3\times10^{6}Nm^{-2}$. Most of the pumps built so far have been designed on the basis of the above values of r/R, e/R, a/R, and N, even though L/R has been increased to values as high as 0.56 in some cases. It is desirable to increase the pump displacement by increasing L rather than R, since the viscous drag losses, wear and probability of cavitation all increase with R. Thus, a very important objective of future research is considered to be the precise determination of stresses in the crescent that will permit the use of the largest possible values of L/R. A modified design is being contemplated wherein L/R could be made as high as 1.4. An additional support for the crescent would be provided, eliminating the present cantilevered condition.

Sprocket

The sprocket bearing poses certain problems that will now be discussed. Depending on the application, the following types of bearings have been used: plain, ball, roller, and needle. The bearing is mounted on a pin protruding from the pumps's body. In order that the radial load be centered on the bearing, the latter cannot be longer than the sprocket thickness. Also, the maximum possible diameter of the hole in the sprocket to accomodate the bearing, is about half of the sprocket's outside diameter. Thus, the pressure on the bearing based on projected area on a diameter is at least twice the pressure developed by the pump. This condition rules out plain bearings for discharge pressures exceeding about $10^{6}Nm^{-2}$. In general, only semifluid lubrication is possible in this case. Bearings with rolling elements permit higher discharge pressures but are not suitable when corrosive fluids are pumped, unless they are made out of stainless steel. Needle bearings offer the highest load capacity, and indeed such type is being used in an experimental pump intended for hydraulic power systems and designed to deliver 45 l/min of hydraulic oil at a pressure of $10^{7}Nm^{-2}$. This pressure constitutes the upper limit dictated by a permissible load on the sprocket bearing. Some thought has been given to the possibility of using an altogether different method of supporting the sprocket by means of a shaft rigidly attached to it, which would penetrate the pump's body and would be exposed to hydrostatic forces at certain points along its length, created by suitable pockets fed by high pressure fluid. These forces would balance the force acting directly on the sprocket. Fluid film lubrication could then be sustained.

Drive-disk

A factor which must be taken into account when considering the forces acting on the drive-disk is that leakage of high pressure fluid to the shaft side of the disk causes a build-up of pressure in the space within the pump's cover. This pressure results in a thrust force on the disk which exceeds an opposite force arising from the discharge pressure acting on a portion of the pin side of the disk. Such condition is undesirable because it is difficult to provide axial support against an inward thrust on the drive-disk, and also because of the increased pressure differential on the shaft seal. An expedient solution taken is to provide a passage from the suction side of the pump to the space on the shaft side of the drive-disk.

The drive-disk shaft is mounted on two bearings. Generally, the one nearest the disk is a ball bearing and the other is a tapered roller bearing. This arrangement does not comply with the usual practice of installing tapered roller bearings in pairs with a certain amount of pre-load. The reason it was adopted is that, while the pump operates, there is always a thrust force acting on the bearing. In this manner, a simple method of adjusting the axial position of the drive-disk is possible by means of a threaded bushing against which the roller bearing outer race bears.

When corrosive fluids are pumped, the shaft seal is placed between the drive disk and the bearings, which results in some sacrifice of shaft rigidity.

Materials

The body and cover are, in general, gray cast iron parts. For heavy duty service, cast carbon steel is more appropiate.

In selecting a material for the sprocket, consideration is given to the fact that it is less costly than the drive-disk and should, if possible, be first to wear out. A satisfactory combination is high carbon steel for the drive-disk, including the pins, and gray cast iron for the sprocket. When the sprocket itself acts as a bearing, a suitable bronze may be used for this part.

For pumping highly corrosive fluids, such as sulphuric acid, or for sanitary applications, an austenitic stainless steel is used for the body, cover, and drive-disk, in combination with a reinforced teflon sprocket, provided the pressure is relatively low.

PERFORMANCE

Following is a discussion of the present state-of-the-art performance of sprocket pumps. Because these pumps are at an early stage of evolution, significant improvements are envisioned in the near future.

Volumetric efficiency

An important index of performance is volumetric efficiency. Typical curves are shown in Fig 3. The solid line refers to the standard B-40 model (42.6 cm^3 displacement) and the broken line represents preliminary results obtained with the experimental high pressure pump (25.5 cm^3 displacement) mentioned earlier. The improvement in volumetric efficiency is due in part to a more rigid input shaft which reduces the out of plane angular displacement of the drive-disk. This displacement results in an increased clearance in the annular sealing area between drive-disk and pump body in the high pressure zone. The shaft deflection is caused by the eccentric action of the resultant axial hydrostatic force on the drive-disk. It would, of course, be preferable to avoid this condition by hydrostatically balancing such force. This is being pursued by introducing a properly located pressure zone on the shaft side of the drive-disk. The objective is not only to avoid an eccentric load but to reduce its magnitude.

The experimental high pressure pump also differs from the standard design in that longer leakage paths have been provided, for example, by increasing the sprocket arms' thickness

Power requirements

In order to estimate the power needed to drive a sprocket pump, the following equation has been found to give reasonably good results:

$$P=(9.12\times10^{-6}+5.66\times10^{-8}\mu)n^{1.8}\,R^3+pQ$$

The first term on the right hand side was obtained empirically from test data. The equation is applicable to pumps with the geometrical proportions given under DESIGN CONSIDERATIONS, and with any value of L/R in the range 0.28-0.56.

Cavitation

A major concern in the development of this pump has been to establish a criterion for the prediction of cavitation. This has been done on the basis of calculating the pressure drop that must take place on the suction side as the fluid is accelerated to the linear velocity of the drive-disk pins in the pumping corridor. In general, a limiting value of 12 m/s for that velocity has been used as a design rule.

FUTURE PERSPECTIVES AND CONCLUSIONS

The sprocket pump has proven itself in industrial applications with a variety of fluids where moderate pressures are required. It is competing with gear and vane pumps in such cases. Other possibilities for its utilization which are being considered will now be discussed briefly.

High pressure pump

As already mentioned, a prototype of a pump for delivering up to $10^7 Nm^{-2}$ of pressure is in an advanced stage of development. The intended use is in hydraulic power systems. In order to attain even higher pressures, a major design effort, not contemplated at this time, would be required.

Hydraulic motor

Obviously, the sprocket pump could also be used as a motor, and some preliminary testing as such has been conducted. An attractive feature is the absence of speed and torque fluctuations.

Large capacity pumps

When flows in excess of 2,000 l/min are required, a modified design with a much increased L/R ratio will be needed to achieve an acceptable power efficiency. Even in the case of smaller pumps, the gain in efficiency justifies investigating this possibility. It is considered that this approach would permit competing with screw pumps, which are extensively employed for pumping large volumes of viscous fluids.

Air motor

A version of the sprocket pump for use as an air motor has been designed and two units have been built. Preliminary tests show an excellent mechanical efficiency, flat torque characteristics, and capability of high speed operation (6,000 rpm or more). The drawback is that air consumption is quite high.

ACKNOWLEDGEMENTS

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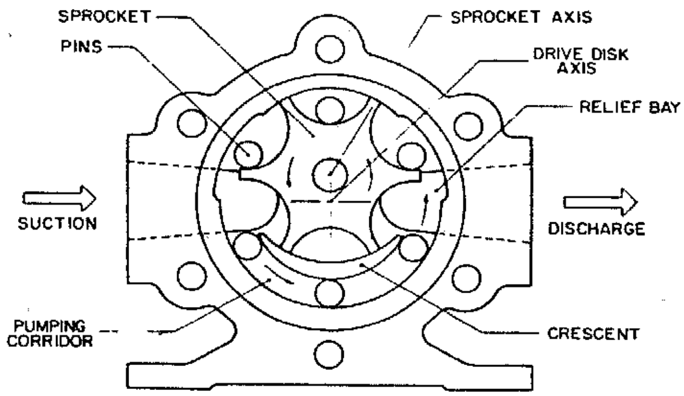
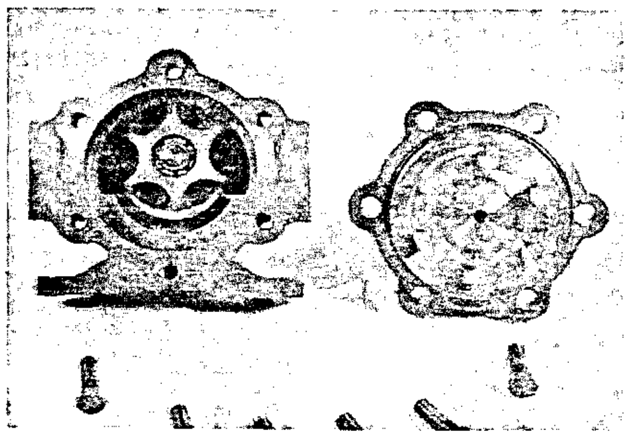
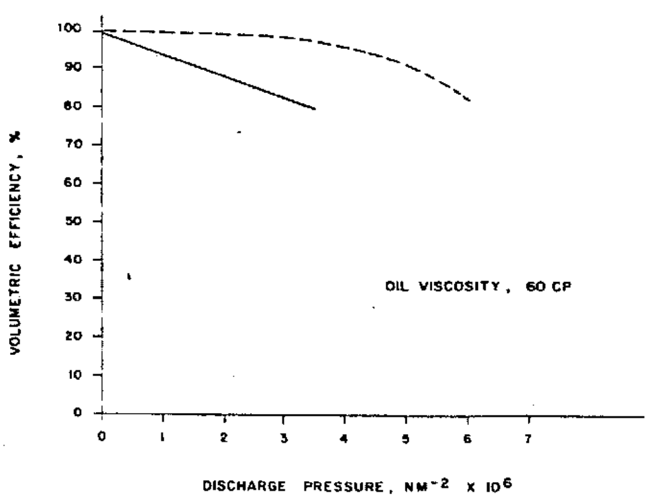


1 Schematic diagram of sprocket pump



2 Partially disassembled sprocket pump



3 Volumetric efficiency of B-40 model pump (solid line) and of experimental high pressure pump (broken line)