Proceedings of the 11th World Congress in Mechanism and Machine Science
August 18-21, 2003, Tianjin, China
China Machinery Press, edited by Tian Huang

Improved Design of Sprocket Pump

Ricardo Chicurel
Instituto de Ingeniería, Universidad Nacional Autónoma de México, Coyoacán 04510, México, D.F., México
e-mail:rcu@pumas.iingen.unam.mx

José León
DYFIM,S.A., Periférico Ote. 4873, Tlalpan 14610, México, D.F., México

Abstract: The sprocket pump, patented in 1988, is used in applications such as fuel pumping in boilers, transfer of fluids, water pumping in treatment plants, and oil pumping in lubrication systems. Design modifications to improve its performance are described. These include: (1) changing the cross section of the pins that produce the pumping action, from circular to a combination of circular arcs and straight segments, in order to achieve surface rather than line sealing; (2) introduction of an extension to the sprocket wheel resulting in a hydrostatically balanced condition; and (3) providing pressure zones on the back face of the drive disk to partially balance the force due to the pressure distribution on the front face.

Keywords: Positive displacement pumps, Sprocket pumps, Rotary machines

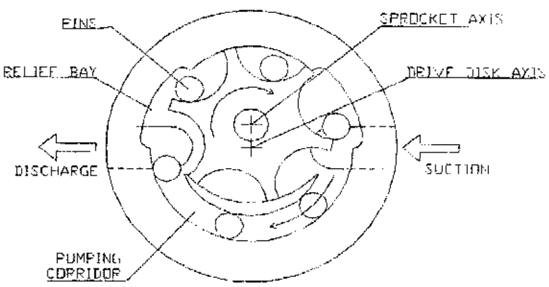

Figure 1. Schematic diagram of a sprocket pump

1 Introduction

The sprocket pump was patented by the first author in 1988[1]. Although simpler, it bears a certain resemblance to a concept patented by Foster [2] in 1938. The pump has been manufactured in limited quantities in Mexico and is used in various industries in applications which include fuel pumping in boilers, transfer of fluids with viscosities up to 500,000 cps, untreated water pumping in treatment plants, and oil pumping in lubrication systems. The sprocket pump, shown schematically in Figure 1, competes mainly with internal gear pumps, resembling these in its configuration. Instead of gears, there is a drive disk having a number of axially oriented cantilevered pins which engage and drive a sprocket wheel whose axis is offset from that of the drive disk. As the pins come out of engagement with the sprocket, they enter a pumping corridor between a crescent shaped barrier and the boundary of a cylindrical cavity. All sealing surfaces are flat or circular cylinder segments. The geometric proportions embodied in the design were derived from an optimization study[3]. Other aspects of the design, construction, performance and applications have been discussed in previous papers[4-6]

It is expected that certain design improvements, to be discussed in this paper, will expand the applications of the sprocket pump. These include: 1) a modification of the pin cross section geometry to achieve better sealing, 2) a hydrostatically balanced sprocket, and 3) reduction of the axial hydrostatic force on the drive disk.

2 Pin Cross-section Modification

2.1 Geometry

In the sprocket pump standard configuration depicted in Figure 1, line seals exist between the pins and the walls of the pumping corridor. After the pins are press fitted onto the drive disk, it is possible to mount the assembly on a lathe and reduce the diameter of the circle circumscribing the pins to achieve a surface seal on one side of the pumping corridor[5]; however, this entails reducing the width of the corridor and a sacrifice in pump displacement. The possibility of enlarging the inscribed pin circle is discarded because this would reduce, or even eliminate, the necessary overlap in the pins' driving action on the sprocket. Another option, which has been tried quite successfully, consists in modifying the pins´ cross-section as illustrated in Figure 2. That figure is a composite diagram showing a portion of a sprocket wheel with standard recesses, labeled "A", and modified ones, labeled "B", together with the corresponding pins. This arrangement is merely a graphical means of comparing the two designs. These would not be combined in an actual pump. In Figure 2, the axes of the drive disk and sprocket are labeled O_1 and O_2 respectively.

In the modified design, the pin cross section changes from a circle to a shape bounded by four circular arcs and two straight line segments. Two of the circular arcs are concentric with the sprocket axis and provide surface, rather than line seals against the walls of the pumping corridor. The other two circular arcs are surfaces used for driving the sprocket, whose recesses must be modified accordingly. Instead of having a

$$M_a = 1.244a^3 p \tag{3}$$

Then, from equations (1), (3), in order to achieve a balance of moments, the distance between the lines of action of the forces F_r, must be:

$$s = 0.622\frac{a^2}{h}$$

An experimental pump satisfying this condition and with $h = 0.557a$ is being built.

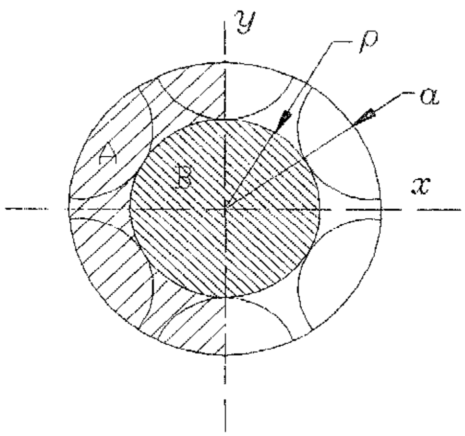

Fig. 5. Areas considered for determining the axial forces acting on the sprocket.

4 Reduction of Axial Force on Drive Disk

The front face of the drive disk, that is, the face from which the pins protrude, is subjected to fluid pressure varying from a maximum equal to the discharge pressure to a minimum equal to the suction pressure. Thus, there is a resultant eccentric axial force which gives rise to fluctuating bending stresses on the drive shaft as well as deflections of the shaft and disk leading to increased internal leakage. In order to reduce this effect, the idea of inducing an appropriate pressure distribution on the back face of the drive disk to approximately balance that on the front face, was put forth. An attempt to achieve this condition in a unit having a 91.5 mm diameter drive disk and 42 cm^3 displacement per revolution, was made in the following manner:

A very small clearance was left between the back face of the drive disk and a confining surface having two symmetrical depressions, or pockets, 2.5 mm deep, one on the high pressure side and the other on the low pressure side. Each of these had an area approximately equal to 25% of the area of the disk face. Six small holes were drilled through the thickness of the disk at the same radial distance as the pins, midway between them. The holes transmit the pressure to which they are exposed on the front face of the disk to the back side. Thus, as they traverse the suction or the discharge zone, the corresponding pocket in back of the disk is pressurized accordingly.

Tests were carried out at 450 rpm. Pressure readings were taken at the discharge port and at each of the pockets. Table 1 summarizes the results.

Diameter of drive disk holes, mm	Discharge pressure, MPa	Pocket pressure, discharge side, MPa	Pocket pressure, suction side, MPa
1.6	0.3	0.19	0.10
2.4	0.3	0.25	0.08
2.4	0.4	0.35	0.06
Without holes	0.3	0.17	0.10

Table 1. Test results for pump with pockets for balancing the axial force on the drive disk

5. Discussion of Results and Conclusions

Three design modifications of the sprocket pump, aimed at improving its performance and reducing hydrostatic imbalances, have been described. The first of these refers to a change in pin geometry by means of which is achieved a surface seal between the pins and the side walls of the pumping corridor. Test results showed a very substantial improvement in volumetric efficiency justifying the adoption of this modification in applications demanding relatively high pressures of the order of 5 Mpa.

With reference to the hydrostatically balanced sprocket, a prototype has been designed, but is not yet available for testing.

Reduction of the axial force on the drive disk has been shown to be feasible by means of small holes drilled on the disk to partially equalize the pressure at corresponding points on its front and back faces. The tests carried out so far are preliminary. Further work is required to determine the effect of different hole arrays.

References

1. Chicurel, R., Bomba rotatoria de desplazamiento positivo, patent no. 155,565, Mexico, March 24, 1988
2. Foster, F.G., Engine, pump, meter and the like, U.S. patent no. 2,124,140, July19, 1938.
3. Chicurel,R., Resendiz, R., Optimized design of a new positive displacement pump, *paper no. 82-DE-18*, ASME, 1982
4. Chicurel, R., Resendiz, R., Bomba rotatoria de desplazamiento positivo de concepción novedosa, *Proc. 7th Congress National Academy of Engineering (Mexico)*, Oaxaca, Mexico:333-336,1981
5. Chicurel, R., Leon, J., Resendiz, R., Applications and perspectives of a novel sprocket type pump, *Proceedings International Conf. on Positive Displacement Pumps*, Chester, England:1-4,1986
6. Chicurel, R., Leon, J., Modelo para el calculo de la eficiencia volumetrica de las bombas de estrella, *Proc. 3rd Congress*, Sociedad Mexicana de Ingenieria Mecanica, Morelia, Mexico:132-135, 1997.

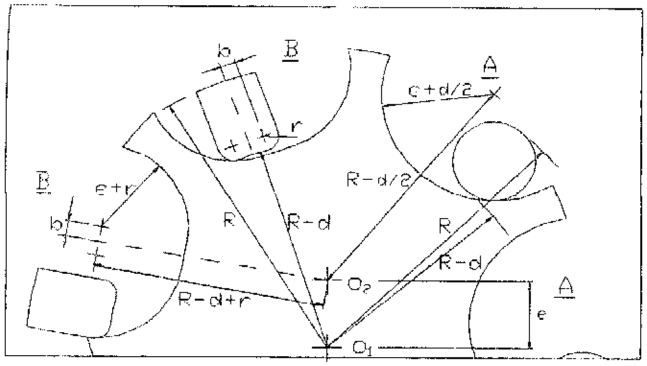

Figure 2. Composite diagram of a portion of a sprocket wheel with standard recesses and pins labeled "A" and modified ones labeled "B"

contour consisting of one circular arc, the modified recesses have a contour made up of two circular arcs connected by a straight line.

In order to visualize the action of the pins on the sprocket, it is convenient to consider the relative motion imagining that the sprocket is fixed. The pins then execute a circular translatory motion of radius "e". Then, in the case of the conventional design, the envelope of a pin's locus is a circle of radius $e+d/2$, a portion of which constitutes the recess in the sprocket. In the case of the modified design, each arc of radius "r" of the pin's cross-section (see figure 2) is part of a circle which revolves relative to the sprocket within an enveloping circle of radius $e+r$. The centers of the two enveloping circles are a distance $2b$ apart. The sprocket recess contour in this case is comprised of arcs of these circles joined by a straight tangent line.

2.2 Test results

An experimental pump prototype incorporating the aforementioned modifications was built. In order to further reduce internal leakage, the sprocket used had recesses open on one side only, as shown in figure 3. This required a depression in the pump's body to accomodate the added sprocket thickness. The radii "r" of the corner arcs of the pin's cross section (see figure 2) were chosen equal to zero to facilitate machining. The sharp edges which resulted were blunted by hand filing. This was deemed acceptable for two reasons: (1) the prototype was intended for short duration tests, and (2) the contact forces between the pins and the sprocket needed to drive the latter are very small by virtue of the fact that there is no hydrostatic torque acting on the sprocket[5]. The pump displacement was 26.8 cm^3/rev. It was directly coupled to a 15 hp, four-pole, induction motor with a nominal speed of 1,750 rpm. Testing was carried out by pumping a light oil with a viscosity of 60 cps from a tank at atmospheric pressure and discharging it at a pressure controlled by means of a throttling valve. A volumetric efficiency of 90% was obtained at a discharge pressure of 5.0 MPa, dropping down to 81% at the maximum test pressure of 6.0 MPa. By comparison, the 90% volumetric efficiency point of an unmodified pump having the same displacement occurs at a discharge pressure of only 1.6 MPa.

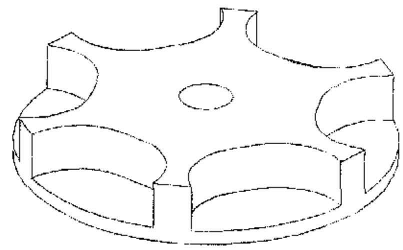

Figure 3. Sprocket with recesses open on one side only

3 Hydrostatically Balanced Sprocket

A considerable hydrostatic radial load acting on the sprocket poses a significant problem with regard to the design of an appropriate bearing support. A possible solution being investigated consists in adding an extension to the sprocket to which an equal and opposite hydrostatic load is imposed. The resulting couple is balanced by the pair of non-collinear hydrostatic axial forces acting on the faces of the sprocket. The concept is illustrated in the diagram of figure 4. The extension's lower end is a mirror image of the upper end which does not have any function related to pumping. The suction and discharge pressures are made to act on the lower end on opposite sides as on the upper end by means of suitable passages in the body of the pump. The arcs providing a seal on the periphery of the sprocket are also reproduced on the lower end.

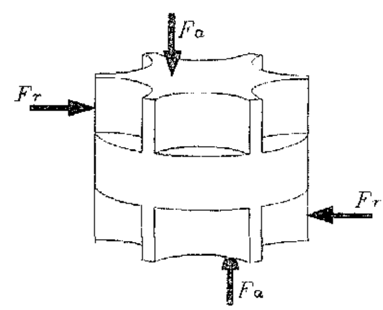

Figure 4. Balanced sprocket

The radial forces, F_r (figure 4), may be expressed as:

$$F_r = 2ahp \tag{1}$$

where a is the sprocket radius, h the depth of the sprocket recesses and p the differential pressure.

The axial forces, F_a , and their lines of action (figure 4), are determined by assuming a uniform pressure, p, acting on the area "A" shown in figure 5 and a pressure decreasing linearly with x from a maximum value of p at $x=-\rho$ to 0 at $x=+\rho$ on the area "B". On this basis, the moment resulting from the forces F_a is found to be:

$$M_a = \frac{4}{3}\left(a^3 - \frac{\rho^3}{3}\right)p \tag{2}$$

In the adopted pump design, $\rho = 0.586\ a$, so that