I. Green

M. Mainland

The George W. Woodruff School of Mechanical Engineering,
Georgia Institute of Technology,
Atlanta, GA 30332-0405

I. Etsion

Department of Mechanical Engineering,
Technion, Haifa 32000, Israel

Experimental Investigation of a New Concept Viscous Pump

A new concept viscous pump that was described earlier in the literature was built and tested to evaluate its performance. It was found that the new concept is feasible and has promising features for use in various applications where high level of cleanliness is required. The pressure-flow characteristic is linear with an easy and accurate flow control which can be attractive in metering pumps.

Introduction

A new concept viscous pump was described and analyzed by Etsion and Yaier (1988), who also cited other relevant publications. The main features of the new pump are: (*a*) simple geometry which provides pumping without contact between surfaces in relative motion, (*b*) elimination of dynamic seals or valves, (*c*) nonpulsating flow and discharge pressure, and (*d*) a linear pressure-flow characteristic with accurate control of small flow rates at relative low pressures.

The noncontact feature eliminates contamination of the pumped fluid by wear debris. This may be important in cases where extreme levels of cleanliness are required, e.g., pharmaceutical application and in the food industry. The elimination of dynamic seals and valves as well as the nonpulsating feature may be of interest in medical applications for pumping physiological fluids into the human body. It may even be possible to use the new concept for pumping blood without the damage caused to red cells in peristaltic devices. This application, however, requires extensive testing with blood to investigate the effect of shear stress on this fluid.

The principle of operation of the new pump was explained by Etsion and Yaier (1988) and is briefly repeated here for clarity. Figure 1(*a*) presents a view of the pump stator. It consists of a circular disk and a number of circular lobes pretruding from the disk surface. These raised lobes separate the central cavity of the stator from its outer circumference. Each lobe is stepped in its middle so that when facing a flat rotor disk parallel to the stator, it forms a small gap c and a large gap C (the difference C-c being the step height s). The rotating disk facing the lobes shears fluid circumferentially as demonstrated by the streamlines in Fig. 1(*b*). Since the shear induced flow rate along any streamline is directly proportional to the gap size, more fluid is sheared from B to D across the gap C than from A to B across the gap c. Hence, a net flow Q_0 is obtained from the outer circumference of the stator towards its center cavity enclosed by the lobes. If the pressure in the center cavity is raised above the pressure level in the outer circumference by an amount P, then a pressure induced flow from the center cavity outward will result. This will reduce the net flow rate to a value of $Q < Q_0$. Increasing the pressure differential to a certain value P^* may completely offset the shear induced flow thereby reducing the net flow of the pump to zero.

The net flow rate Q of the pump at a given discharge pressure P was found as (Etsion and Yaier, 1988)

$$Q = Q_0(1 - P/P^*) \tag{1}$$

where Q_0, the flow-rate at zero pressure, is

$$Q_0 = \frac{n\omega}{4} s R_0^2 \left[1 - \left(\frac{R_i}{R_0} + \frac{\Delta r}{R_0}\right)^2\right] \tag{2}$$

and P^*, the discharge pressure at zero flow, is

$$P^* = 3\mu\omega \frac{sR_0^2}{C^3 + c^3} \frac{1}{\alpha} \left[\left(\frac{R_i}{R_0} + \frac{\Delta r}{R_0}\right)^2 - 1\right] \ln\left(1 - \frac{\Delta r}{R_0 \tan\beta}\right) \tag{3}$$

where μ is the fluid viscosity and the other symbols are according to Fig. 1.

Based on the optimal number of lobes that resulted from the analysis by Etsion and Yaier (1988) a three lobe pump was built and tested to experimentally study its performance and feasibility.

Pump and Test Apparatus

The 110 mm diameter stator consisted of three lobes as shown in Fig. 2. Each lobe was 22.4 mm wide and had 60° arc length. The average step height (C-c) was 43 μm. Figure 3 shows a cross section of the pump. The stator (2) was mounted in the pump front cover (1) by means of two O-rings. This formed a sealed cavity at the back of the stator which could be pressurized to control the closing force and hence, the clearance between the stator and the flat face rotor (5). An eddy current proximity probe (6) was installed in the front cover. This probe allowed the measurement of the relative position of the stator with respect to the pump front cover thus permitting indirect measurement of the small clearance, c, between stator and rotor. The inlet to the pump allowed liquid to be introduced to the outer circumference of the rotor and stator.

Contributed by the Tribology Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS and presented at the Joint ASME/STLE Tribology Conference, Toronto, Canada, October 7-10. Manuscript received by the Tribology Division January 19, 1990; revised manuscript received July 8, 1990. Paper No. 90-Trib-58. Associate Editor: K. E. Rouch.

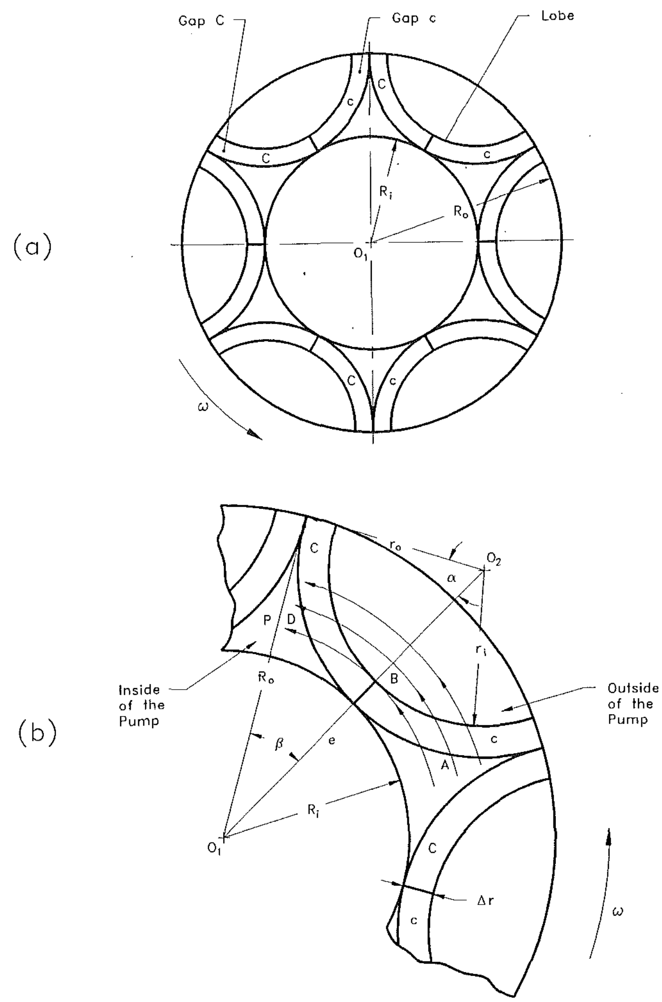

Fig. 1 Pump lobed stator: (a) Top view; (b) Streamlines crossing a lobe

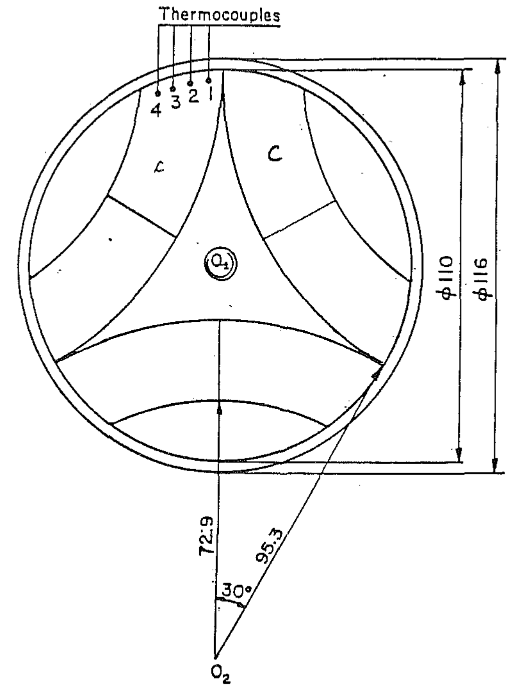

Fig. 2 Three lobes experimental stator

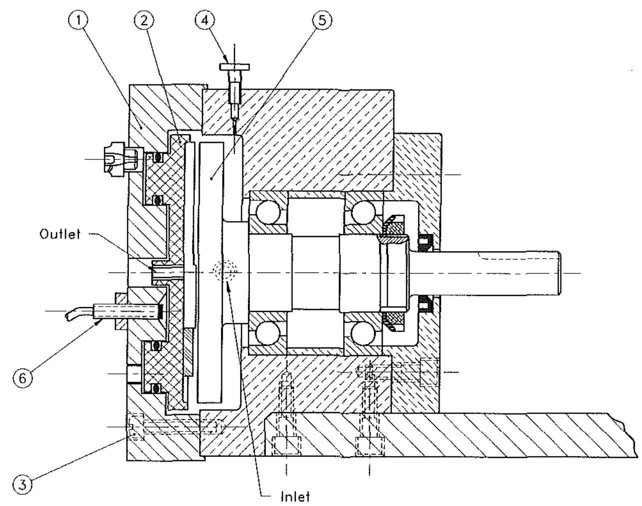

Fig. 3 Pump cross section

A vent (4) was used to bleed air out of the inlet cavity to assure priming of the pump and eliminate trapped air bubbles. The liquid that was pumped radially inward exited the pump through the outlet passage in the center of the stator and the front cover.

Figure 4 shows schematically the test system. Mineral oil (with viscosity of 8.69 mPa S at 40°C and 2.26 mPa S at 100°C) was supplied to the pump (1) by gravity from a container (2). The pump discharge was circulated into the container and the discharge pressure could be varied and measured by means of a discharge valve and a pressure indicator located on the return line. A two way valve located downstream of the discharge valve provided a mean of temporarily diverting all the pump outflow into a measuring beaker for flow-rate measurement. A heat exchanger utilizing tap water was used to remove shear induced heat from the oil in the container and to maintain a constant temperature. The pump was driven by a 0.75 kw variable speed motor. The cavity behind the stator was filled with oil and was pressurized by compressed air. This back pressure was controlled by an electropneumatic control valve. The oil temperature in the container was measured with a mercury thermometer and the face temperature of the stator was measured by means of thermocouples on one of the lobes as shown in Fig. 2. The thermocouples were inserted into holes drilled from the back of the stator and then lapped flat with the stator face. The reading of thermocouple no. 1 was used to determine the oil temperature in the clearance between stator and rotor so that the actual viscosity can be used in calculating P^* from equation (3).

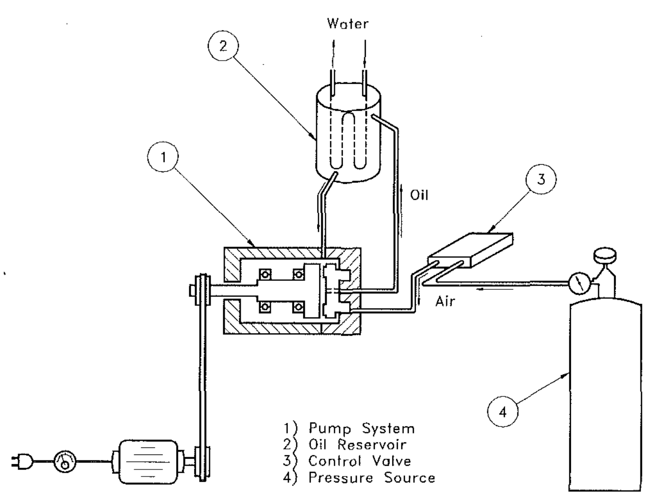

Fig. 4 Schematic of test system

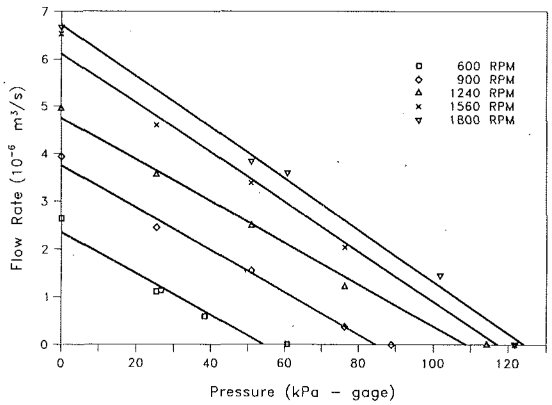

Fig. 5 Pressure-flow characteristics at various speeds and 9 μm nominal clearance

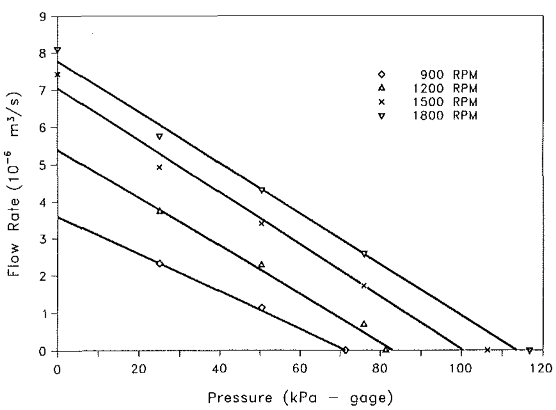

Fig. 6 Pressure-flow characteristics at various speeds and 15 μm nominal clearance

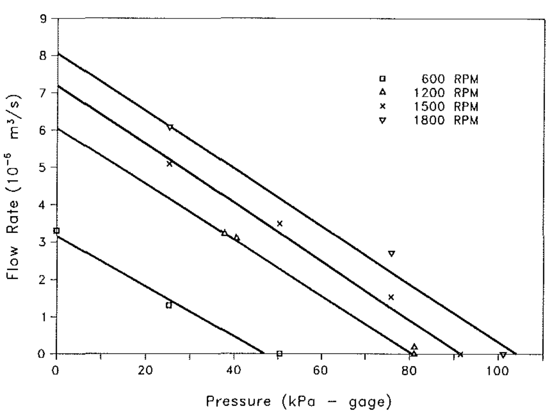

Fig. 7 Pressure-flow characteristics at various speeds and 21 μm nominal clearance

Test Procedure

Tests were conducted with the pump operating at five speeds (600, 900, 1200, 1500, and 1800 rpm) and three different small nominal clearances (9, 15, and 21 μm).

The clearance was measured using a proximity probe which determined the location of the stator relative to its position in contact with the rotor. It was found, however, that when the pump is pressurized, the rotor moves relative to its initial position within the pump housing. Thus, the position of the stator with respect to its initial contact position is not an entirely accurate measure of the clearance. In order to account for this movement of the rotor, a calibration procedure was utilized. First, the force balance relating the pump pressure and the back pressure was calculated. The stator was then placed in contact with the rotor (now stationary) and the position of the rotor was recorded. Next, a back pressure was applied to the stator, and the new location of the rotor was recorded. From this data, a calibration curve was generated so that the position of the rotor could be determined based upon the force generated by the pressure of the fluid under operating conditions.

Tests were conducted such that for a given clearance a certain speed was assigned. For every set of clearance and speed the flow was incrementally increased from no flow at all (maximum pressure) to full pumping capacity (zero gauge pressure) by means of a restricting valve on the outlet line. The flow was collected in a beaker, and the rate was determined by dividing the collected volume by the measured elapsed time. Simultaneously, the pressure was read off a gauge placed on the outlet line. Normally five flow/pressure readings were attempted for each clearance/speed set, although fewer points were obtained in some cases of high flow rates which were hard to handle by the simple technique outlined above. Tests were repeated for the five speeds at a given clearance. (In certain instances the speed drifted slightly away from the designed value.) Once the five speed tests were completed a new clearance was assigned and the tests were again repeated.

In the above procedure, as a result of a change in the flow and pump pressure, the opening force on the stator and rotor changed, thus causing the clearance to change. An attempt was made to keep the clearance value constant by regulating the back pressure on the stator. However, the task of precisely maintaining a constant clearance was not always successful due to, what is believed to be, the hysteresis of the O-rings that supported the stator, and the displacement of the rotor. At best the clearance could be maintained about an average or 'nominal' value. The oil temperature in the container was kept at 33°C and regulated by the flow rate of tap water through the heat exchanger.

Table 1 Linear regression analysis of the experimental results and calculated results (in parenthesis)

Nominal clearance (μm)	Speed (rpm)	Slope, Q_0/P^* ($10^{-11}m^5/Ns$)	Intercept, Q_0 ($10^{-6}m^3/s$)
9	600	4.33 (1.06)	2.35 (3.34)
	900	4.44	3.74 (5.01)
	1240	4.42	4.76 (6.90)
	1560	5.32	6.17 (8.69)
	1800	5.42	6.65 (10.02)
15	900	5.03 (1.49)	3.55 (5.01)
	1200	6.55	5.44 (6.68)
	1500	6.95	7.04 (8.35)
	1800	6.91	7.81 (10.02)
21	600	6.55 (2.04)	3.09 (3.34)
	1240	7.41	6.02 (6.90)
	1500	7.75	7.16 (8.35)
	1800	7.67	8.11 (10.02)

Test Results

The results of the tests, namely the measured flow and pressure at various test speeds, are presented in Figs. 5 to 7 for nominal clearance of 9, 15, and 21 μm, respectively. The data suggest a linear flow versus pressure relationship for a given speed and clearance. A simple linear regression curve has been fit to the data and is summarized in Table 1. A t-distribution statistical analysis of significance indicates that the linear regression is the best fit. Higher order regressions produced t values well below the lowest generally accepted level of statistical significance. This linear flow versus pressure relationship agrees well with equation (1).

The flow Q_0 at zero (gauge) pressure (valve fully opened) is approximately proportional to the rotational speed of the pump (see equation (2)), as demonstrated in Table 2. Here, the meas-

Table 2 Maximum flow rate at zero gauge pressure

Nominal clearance (μm)	Speed (rpm)	Measured flow, Q_0 (10^{-6}m^3/s)	Scaled flow, Q_0 (10^{-6}m^3/s)	Percent difference
9	600	2.35	--	--
	900	3.74	3.52	− 5.9
	1240	4.76	4.86	+ 2.1
	1560	6.17	6.11	− 1.0
	1800	6.65	7.05	+ 6.0
15	900	3.55	--	--
	1200	5.44	4.73	− 13.0
	1500	7.04	5.92	− 16.0
	1800	7.81	7.10	− 9.1
21	600	3.09	--	--
	1240	6.02	6.39	+ 6.1
	1500	7.16	7.73	+ 8.0
	1800	8.11	9.27	+ 14.0

Table 3 Face temperature (in °C) for various speeds and clearances at two valve positions

Nominal clearance (μm)	Valve position	Speed (rpm) 600	900	1200	1500	1800
9	open	47	46	47	49	52
	closed	48	48	50	52	57
15	open	48	47	48	51	53
	closed	49	48	51	54	58
21	open	47	48	50	53	55
	closed	49	50	53	56	62

ured flow Q_0 at each speed is compared to the scaled flow which is the flow Q_0 at the lowest speed at which data was taken multiplied by the ratio of the two speeds. This is done for each nominal clearance. The measured flow given in Table 2 is actually based upon the linear regression described above.

Note that for a given nominal clearance the magnitude of the slopes listed in Table 1 tends toward a constant value as the speed increases. This slope, according to equation (1), represents the ratio Q_0/P^* where Q_0 and P^* are given by equations (2) and (3), respectively. This ratio is independent of speed as the experimental results confirm especially at the higher speeds.

A comparison between the experimental results of Q_0 and its calculated values as obtained from equation (2) is presented in Table 1 where the calculated Q_0 values appear in parenthesis. The agreement is better than 50 percent at the lowest nominal clearance of 9 μm and better than 23 percent at the highest clearance of 21 μm.

For calculating the slope Q_0/P^* by equations (2) and (3) the actual oil viscosity has to be known. Table 3 presents values of the face temperature obtained at the five test speeds for the three nominal clearances. Results are presented for conditions of maximum flow rate (valve open) and zero flow (valve closed). The measurements were taken after two minutes of operation for each case, at which time the temperature was almost steady, although it changed slightly with time. The temperatures measured at the surface of the lobe ranged from 46°C (valve fully open) to 62°C (valve fully closed).

In calculating P^* by equation (3) it was assumed that the oil viscosity corresponds to the temperature measured at the lobe surface. The results thus obtained for the ratio Q_0/P^* are shown in Table 1 in parenthesis as a single value for each one of the three clearances. The experimental results are about three to five times higher.

Discussion

Contrary to a good qualitative agreement the quantitative agreement between analytical and experimental results is poor, especially at high discharge pressure and low flow rates. This poor agreement may partially be attributed to indirect measurements of the clearance, pressure and fluid temperature. Moreover, at high discharge pressure flow losses may occur across the lobe ends at the outer circumference of the stator. These losses are not accounted for in equations (1) and (3). It is anticipated that such losses will reduce the magnitude of P^* thereby increasing the magnitude of the calculated slope Q_0/P^*. As the discharge pressure, and hence the end losses, decrease the agreement between measured and calculated results improves as shown by the corresponding values of Q_0 in Table 1.

The most important result of the tests is the proof that this new concept viscous pump is feasible. Actually, the pump has even more potential than initially revealed by the present results. An optimization of the pump geometry would yield a more efficient pump than the one used for the present tests. Moreover, a second stator could be added on the other side of the rotor to yield twice the flow.

With an eye to future work it is concluded that for better results the true clearance based upon the relative positions of the rotor and stator should be measured instead of just the motion of the stator relative to the housing. A better understanding of the flow field could be obtained with pressure and temperature gauges placed on both sides of the lobes. This could also serve to shed some light on the validity of the assumption that the pressure at the outer rim of the pump is indeed "ambient", and that the pump is fully primed. The theoretical analysis can be improved by considering the end losses described previously.

Acknowledgment

The tests reported in this work were conducted during a summer visit of the first author with the Dept. of Mechanical Engineering at Technion. The project was supported in part by a Georgia Tech. Foundation Grant, E25-549, made by Mr. Gilbert Bachman and by Technion VPR-Loewengart Research Fund. This support is gratefully acknowledged.

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