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ANALYTICAL AND EXPERIMENTAL INVESTIGATION OF A SCROLL COMPRESSOR LUBRICATION SYSTEM

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ABSTRACT

Previous investigations have revealed the complex nature of the mechanisms involved in the lubrication system performance of refrigerant compressors. The entire system, rather than the oil pump alone, needs to be considered in order to gain an understanding of these mechanisms. A typical scroll compressor lubrication system contains several elements including oil pumps, galleries, grooved and non-grooved journal bearings, etc. Open literature (Refs. 1-4) may contain design criteria for each element individually, but not together as a system. It was for this reason that a model was developed to investigate the behavior of the lubrication system as a whole. Ultimately, this model can be used to optimize key design parameters in order to maximize the system performance.

A test rig was built to accurately measure the oil flow rates through a scroll compressor lubrication system so that a comparison could be made between predicted and experimental results. The test matrix variables included the operating speed, oil temperature, and geometry of the system. The ability to accurately predict the effects of subtle changes in geometry on the lubrication system performance is critical due to the low operating pressures and oil flow rates found in scroll compressor lubrication systems. A summary of the analytical model and experimental verification activities is presented.

NOMENCLATURE

Parameters:

- b Groove width
- g Gravitational acceleration
- h Head
- m Empirical correction factor
- z Height above datum

- A Area
- C Loss coefficient
- D Diameter
- F Force
- H Groove depth
- L Length
- N Running speed
- P Pressure
- Q Flow rate
- R Radius
- S Journal cicumferential speed
- U Groove circumference
- V Velocity

Subscripts:

- 0-8 Station numbers
- eff Effective
- f Friction
- gr Groove
- hydro Hydrodynamic
- or Offset radial pump
- os Orbiting scroll
- pt Pickup tube
- ref Reference
- rf Radial feed pump
- sb Scroll bearing
- sp Spiral groove
- st Straight groove
- t Total
- umb Upper main bearing

Greek:

- α Groove pitch
- β Offset angle
- γ Specific gravity
- ν Viscosity
- ρ Density
- ω Angular velocity

INTRODUCTION

A typical scroll-type refrigerant compressor contains a drive motor, drive shaft with eccentric crank, a fixed scroll element, and a moving (orbiting) scroll element. Vapor compression is achieved as a fluid contained within pockets created by meshing of the fixed and orbiting scroll wraps is displaced from the scroll outer pockets to the inner pockets with continuously decreasing volume. Scroll compressors possess high efficiency, low noise, and low vibration which make them an attractive alternative to reciprocating compressors.

Proper lubrication to bearings and thrust surfaces is essential for efficient and reliable operation of the scroll compressor. The potential severity of loss of lubrication, combined with the lack of relevant information on lubrication system design in open literature, prompted a thorough investigation of the lubrication system performance of the scroll compressor. An analytical model was developed to accurately predict the lubricant flow rates to all vital system elements. A test rig was subsequently built to measure the oil flow rates to the system elements so that a comparison could be made between predicted and experimental results.

This paper describes the operation of the lubrication system, the development of the analytical model and experimental test rig, and the comparisons between predicted and measured results.

LUBRICATION SYSTEM OPERATION

A typical lubrication system contains several elements including oil pumps, galleries, bearings, etc. Figure 1 shows a schematic diagram of a scroll compressor lubrication system. Following the lubricant path, the oil enters the oil pickup tube at station 1. The oil is rotated within the pickup tube resulting in a pressure rise due to centrifugal force. Some of the oil is then fed through a radial centrifugal pump to the lower bearing; the remaining oil enters the offset radial centrifugal pump at station 2, within which the fluid pressure is gradually increased as the radial distance from the shaft centerline increases. The pressure rise must be enough to overcome the static head and the friction losses in the gallery. At station 3, a portion of the oil is fed through a radial centrifugal pump to the upper bearing where it then enters a spiral groove located on the journal. Due to the pitch angle, the spiral groove acts as a viscous pump, thus promoting fluid flow. The total flow to this bearing will have a component due to the groove and also a hydrodynamic component. The remaining oil follows the path from station 3 to station 7 where it exits the shaft. The oil is then directed through the straight groove in the scroll bearing to station 8. The flow rates through the system will ultimately depend on the overall resistance through each system element.

LUBRICATION SYSTEM ANALYTICAL MODEL

A computer model that accurately predicts the internal behavior of the scroll compressor lubrication system was developed. The major assumptions used in the analysis are as follows.

1. Incompressible working fluid
2. Constant angular speed
3. Friction factors are based on laminar pipe flow theory
4. The sump oil level remains constant
5. No heat transfer to or from the oil (constant viscosity)

Governing Equations

The conservation laws of mass and energy were used to characterize the behavior of the lubrication system. The general form of the continuity equation at a junction of several oil galleries can be expressed as

$$\sum_i (V_i A_i)_{in} = \sum_j (V_j A_j)_{out} \tag{1}$$

Energy conservation takes the form of the unsteady Bernoulli equation between any two points i and j along a streamline and is given as

$$\frac{1}{g}\int_i^j \frac{\partial V}{\partial t}\, ds + \frac{(P_j - P_i)}{\rho g} + \frac{1}{2g}(V_j^2 - V_i^2) + (z_j - z_i) + h_f + \Sigma h_k = 0 \tag{2}$$

where h_f is the friction head loss and Σh_k includes head changes due to pumps, flow losses, etc. The unsteady term in the equation is assumed to be zero everywhere in the system. The friction head loss is based on laminar pipe flow theory and is given as

$$h_f = \frac{32\nu LV}{gD^2} \tag{3}$$

<u>Lubrication System Elements</u>

The next step in the formulation of the model is to establish the characteristics of the lubrication system elements represented by the h_k terms. These include centrifugal oil pumps and bearing groove configurations. An expression is needed that relates the pressure differential across the system element to the flow rate, geometry, fluid properties, and operating conditions. After the characteristics of the lubrication system elements are known, the system equations can be formulated and solved for the flow rates.

Referring to Fig. 1, there are three types of centrifugal oil pumps in the system. The first type is the oil pickup tube. Fluid pressure develops as the oil is rotated within the pickup tube. The pressure rise is found by integrating the incremental centrifugal force that a fluid element experiences as it travels radially outward and is given as

$$\Delta P_{pt} = \frac{\rho\omega^2 R_{pt}^{\ 2}}{(2m+2)} \tag{4}$$

where m is an empirical factor and is due to the non-linearity of the fluid angular velocity within the pickup tube. The free surface of the oil within the pickup tube forms a paraboloid of revolution, concave upward as shown in Fig. 2, which allows the oil to "climb" the walls of the pickup tube and enter the second type of centrifugal pump labeled as the offset radial centrifugal pump. The offset angle is shown as β in Fig. 2. The fluid pressure in this pump is gradually increased as the radial distance of the pump from the shaft centerline is increased. The pressure rise in this pump can be expressed as (including friction loss and static head corrections)

$$\Delta P_{or} = \frac{\rho\omega^2 L^2 \cos^2\beta}{2} + \rho\omega^2 R_o L\cos\beta - \frac{32\nu LV}{gD_{eff}^{\ 2}} - \rho gL\sin\beta \tag{5}$$

where R_o is the initial offset of the pump and D_{eff} is an effective oil gallery line diameter based on the amount of oil present in the gallery. The third type of centrifugal pump is a radial feed centrifugal pump (stations 3-4 in Fig. 1) which delivers oil to the upper bearing. Vasser (Ref. 5) developed an empirical relation for the pressure rise through this pump which is given as

$$\Delta P_{rf} = P_E\left\{1 - \left[\frac{Q_{gr}+Q_{hydro}}{Q_{max}}\right]^{\xi}\right\} \tag{6}$$

where P_E is the pressure rise at zero flow rate (Euler pressure) and takes the form

$$P_E = 4.8\times10^{-5}\gamma(2\pi N)^2(R_4^{\ 2} - R_3^{\ 2}) \tag{7}$$

Q_{gr} is the oil flow through the groove (V_5A_5), Q_{hydro} is the hydrodynamic flow to the bearing, and Q_{max} is the flow rate at zero pressure which is a function of the pump geometry, running speed (N), oil viscosity, and suction head. The shape of the pump curve is determined by the empirical exponent ξ.

There are two types of grooved journal bearings in the system. The first type is the cylindrical spiral groove located on the upper bearing journal. As the force vectors in Fig. 3 indicate, the spiral groove acts as a viscous pump which allows the oil to be distributed along the entire length of the bearing and also allows for any contaminants present in the system to be flushed through the bearing. For oil flow analysis purposes, the groove is considered as a nozzle, a laminar friction element, and a pump connected in series. The pressure differential across the groove is then given by Vasser as

$$\Delta P_{sp} = 1.145\times10^{-6}\gamma\nu\frac{L_{gr}U^2V}{A^2\cos\alpha} + 0.710\times10^{-3}\gamma V^2 - 0.0387\times10^{-6}\gamma\nu\frac{S^2L_{gr}b\tan\alpha}{VA(H/2)} \quad (8)$$

where A is the cross-sectional area of the groove, U is the groove circumference, and S is the circumferential speed of the journal. The second type of groove in the system is a straight cylindrical groove located on the scroll bearing journal. The pressure drop across this element (ΔP_{st}) is the same as that of the spiral groove with $\alpha = 0$.

Additional flow losses due to turning and discharge are represented as a reduction in kinetic energy

$$\Delta P = \frac{1}{2}\rho C_L V^2 \quad (9)$$

where C_L represents a loss coefficient.

System Equations

The system equations are formulated by applying the steady Bernoulli equation along the lubricant flow paths. There are two unknown flow rates in the system. They are.

1. The flow rate through the upper bearing (V_5A_5)
2. The flow rate through the scroll bearing (V_8A_8)

The flow rate to the lower bearing is calculated using journal bearing design theory. Therefore, two independent system equations involving the unknown flow rates are necessary. The first equation governs the flow through the upper bearing (from station 0 to station 5). The appropriate pressure differential relations for the system elements along this path are substituted into the steady form of the Bernoulli equation giving (with $V_0=0$)

$$(P_5-P_0) + \frac{1}{2}\rho V_5^2 + \rho g(z_5-z_0) + \rho g h_{f_{2-3}} - \Delta P_{pt} - \Delta P_{or_{2-3}} - \Delta P_{rf} - \Delta P_{sp} + \frac{1}{2}\rho C_{L_5} V_5^2 = 0 \quad (10)$$

The second equation is set up for the flow through the scroll bearing (from station 0 to station 8) and takes the form

$$(P_8-P_0) + \frac{1}{2}\rho V_8^2 + \rho g(z_8-z_0) + \rho g(h_{f_{2-3}}+h_{f_{3-7}}) - \Delta P_{pt} - \Delta P_{or_{2-7}} - \Delta P_{st} + \frac{1}{2}\rho C_{L_8} V_8^2 = 0 \quad (11)$$

The pressures P_5 and P_8 are known, which yields a system of two independent equations with two unknowns (V_5 and V_8). A Newton-Raphson iteration technique is used to solve the system.

EXPERIMENTAL APPARATUS

The development of a laboratory test rig for measurements of the lubricant flow rates through a scroll compressor was achieved in parallel with the analytical model development. The test rig, or "lube-rig" as it will be referred to, was used to verify the simulation code for a given system geometry. The lube-rig was designed to precisely control the oil sump temperature and compressor shaft speed, while measurements of the lubricant supply pressure and oil flow rates to the upper main bearing, orbiting scroll bearing, and scroll thrust surface were obtained. Various shaft oil gallery and journal bearing groove geometries were tested in the lube-rig for their effects on the total flow and flow proportioning to the sliding surfaces. Measurements at various lube-rig test conditions were compared with those predicted by the analytical model. Upon verification of the simulation code, a valid design tool was available for quickly determining the effects of oil viscosity, operating speeds, and manufacturing tolerances on the system flow rates without the need for a multitude of laboratory tests.

Description of the Lube-Rig

A schematic diagram of the lube-rig is shown in Fig. 4. Oil is pumped from a main supply reservoir to a heater tank that uses two 200W resistance heaters to maintain a desired oil temperature. The heated oil from the tank is then fed into a clear acrylic oil sump where the oil is picked up by the oil pickup tube and distributed throughout the compressor. The oil supply to the compressor sump is regulated such that at steady-state operation, a constant oil level is maintained. The delivery of the oil to the sump is controlled using a variable speed, magnetically coupled oil pump. Fine adjustments in the oil level are obtained using a metering valve on the return drain from the compressor sump to the main oil reservoir. Temperature regulation is accomplished by means of a temperature controller which is connected to a thermocouple located in the sump. The compressor shaft is driven by a variable speed ECM drive motor. Shaft speed is determined using a strobe light and accelerometer. Oil temperatures at the heater reservoir, oil sump, and flow measurement locations were measured. Oil supply pressure was measured at the upper main bearing using a digital absolute pressure gauge. Flow measurements were determined by monitoring the time required to fill a graduated cylinder to a specified level. Large diameter (1/2in I.D.) tubing was used to route the oil from the compressor to the graduated cylinders while capillary tubing (1/8in I.D.) was used for the pressure measurements.

The lube-rig compressor assembly is shown in Fig.5. The assembly consists of the compressor shell, motor, upper and lower bearing mounts, and the orbiting scroll analog. This assembly bolts in place above the compressor sump. Precise alignment of the bearing mounts in the assembly is maintained via dowel holes through the bearing mount flanges and compressor shell receptacle flanges which also allows for ease of installation and removal of the compressor shaft. The ECM rotor is fixed to the shaft by means of snap rings, and is keyed to the shaft to allow torque transmission. This technique yielded the flexibility for testing various shaft configurations while allowing for testing of lubrication system design enhancements along with new product research and development. Shaft seals were employed to segregate the flow rates from the upper main bearing gallery and the orbiting scroll gallery.

RESULTS

Results were obtained in a logical sequence which progressed from obtaining flow data for the basic geometry of the centrifugal pumps in the shaft to detailed data on flow proportioning through grooves and feed galleries in the complete scroll compressor lubrication system. Simple experiments to measure the total oil flow and fluid pressure in the shaft main oil gallery were used to establish the empirical constant that characterizes the velocity profile of the oil within the oil pickup tube. Next, experiments were focused on determining the incremental flow rates in each of the system elements. The results will be presented in this chronology.

Shaft Main Oil Gallery Total Flow Rate

Measurements of the total oil flow rate through the system were made for several values of oil temperature, sump oil level, and shaft rotational speed. Figure 6 shows the variation in total flow rate with increasing oil temperature (decreasing oil viscosity). The total flow rate is defined in units Q/Q_{ref}, representing the total flow as a ratio of the flow at any temperature and shaft speed to a defined reference condition flow. The defined reference flow Q_{ref} was chosen to be at 150 deg F oil temperature, 60Hz shaft frequency, and an oil level of 2.75in above the datum (see Fig. 1 for the datum location). Figure 7 shows a comparison of analytical and experimental results for the total flow rate as a function of oil level. The total oil flow is shown to be very sensitive to the height of the oil within the oil pickup tube due to the axial variation of the free surface profile of the oil as it enters the offset radial centrifugal pump (see Fig. 2).

Journal Bearing Flow Rates

Efforts were directed towards measuring the flow ratio between the upper main bearing and the scroll bearing while varying the geometry of the spiral groove located on the upper bearing journal. The results were then compared to those predicted by the simulation code. Following verification of the code, the flow ratio was optimized based on the magnitudes of parameters such as bearing load, clearance, and operating temperature that occur during actual compressor operation. Table 1 compares both analytical and experimental results for various spiral groove geometries. Both the groove width (b) and depth (H) are non-dimensionalized with respect to a reference geometry. Note also that the flow rates are given as a fraction of the total flow for each condition. The analytical and experimental results are in close agreement and show that as the spiral groove dimensions are increased, the resistance to flow through the groove decreases until eventually no flow proceeds upstream to the orbiting scroll bearing. These data show that the analytical model can be used with a high level of confidence to design groove configurations that generate the desired flow ratio.

CONCLUSIONS

An analytical model was developed and verified to aid in the design of a scroll compressor lubrication system. The simulation code can be used as a design tool to gain an initial understanding of the lubrication system behavior under a variety of conditions without the need for extensive laboratory testing. New techniques were also developed for the accurate measurement of the flow rates and fluid pressures through the various lubrication system elements. Analysis of the results showed that the simulation code was in agreement with measurements. It was also found that the overall lubrication system performance is sensitive to small changes in geometry and operating condition. While the lubrication system model can be considered complete, further work is planned to expand the range of applicability of the equations that characterize critical lubrication system elements.

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Table 1 Comparison of Analytical and Experimental Flow Rates for Various Groove Geometries.

Upper Bearing Groove Configuration b/b_{ref}	$\times H/H_{ref}$	Q_{umb}/Q_t Code	Q_{umb}/Q_t Exp.	Q_{sb}/Q_t Code	Q_{sb}/Q_t Exp.
0.50	0.50	0.11	0.05	0.89	0.95
2.00	0.50	0.22	0.18	0.78	0.82
1.00	1.00	0.34	0.31	0.66	0.64
1.00	1.25	0.44	0.48	0.56	0.52
1.00	1.60	0.57	0.65	0.43	0.35
2.00	2.00	1.00	1.00	0.00	0.00

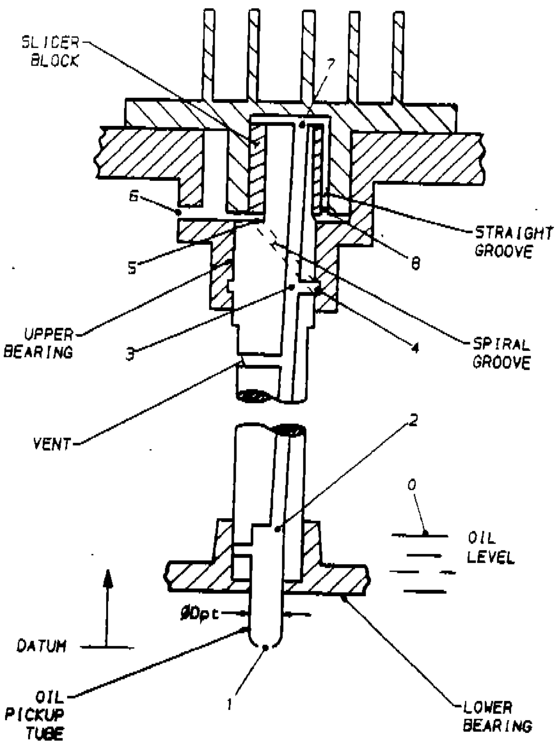

Fig. 1 Schematic Diagram of a Typical Scroll Compressor Lubrication System.

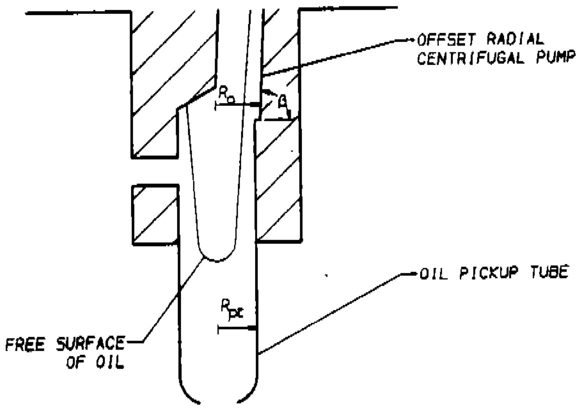

Fig. 2 Oil Pickup Tube Showing the Free Surface of the Oil.

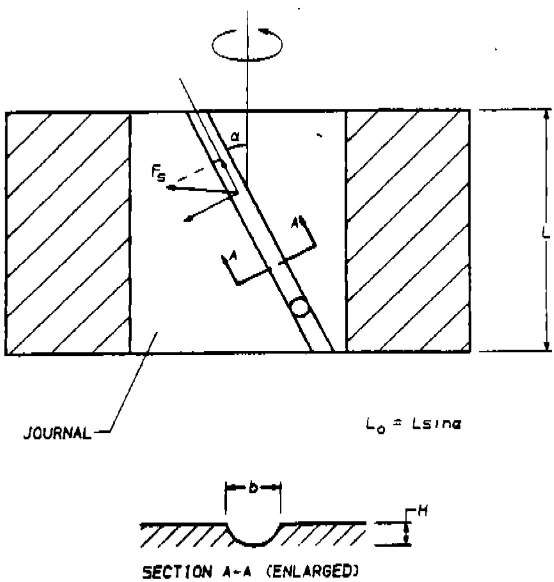

Fig. 3 Journal with a Spiral Groove.

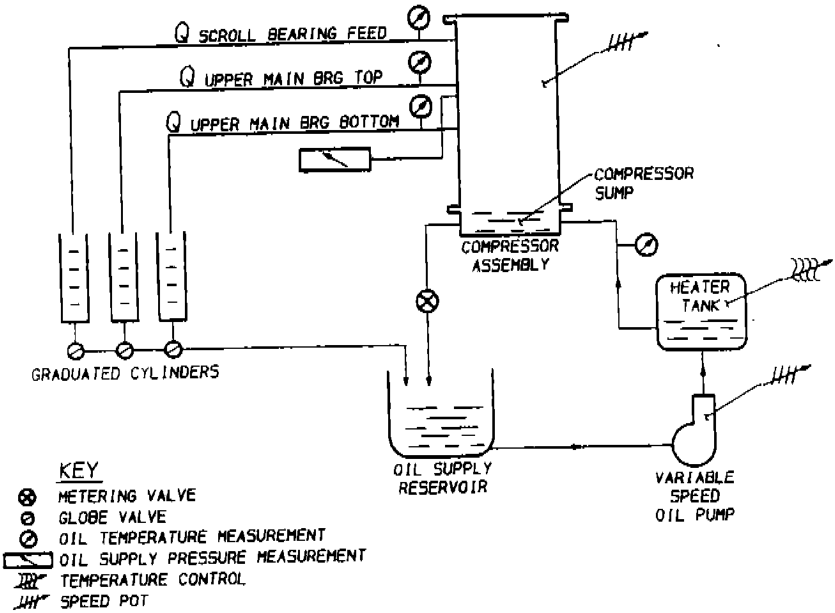

Fig. 4 Schematic Diagram of the Lube-Rig.

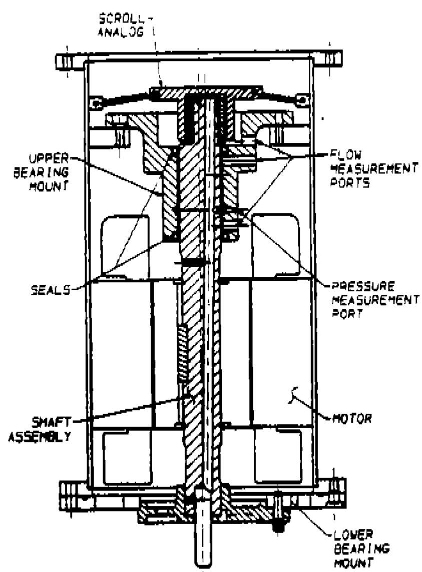

Fig. 5 Lube-Rig Compressor Assembly.

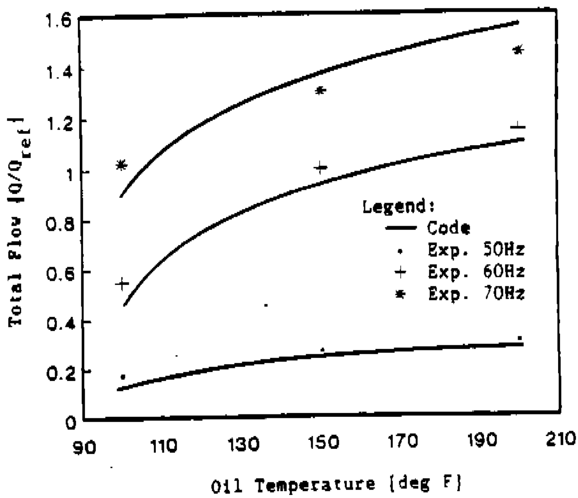

Fig. 6 Total Oil Flow Rate vs Temperature at Various Speeds.

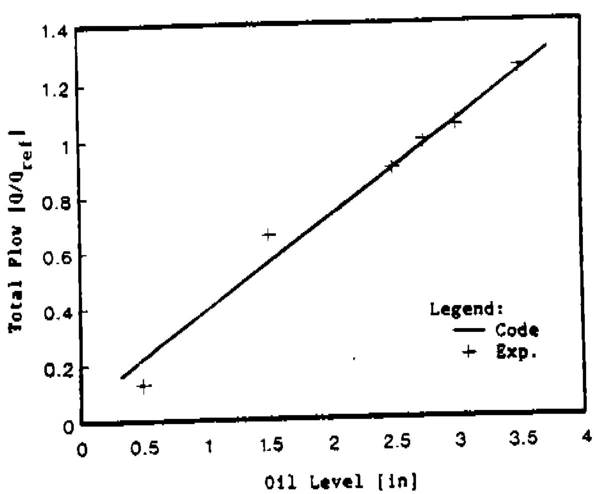

Fig. 7 Total Oil Flow Rate vs Oil Level.