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Network Analysis of Engine Lubrication System

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Abstracts : A computer program for the analysis of engine lubrication systems has been developed. And a case study of a four cylinders gasoline engine is illustrated. This paper gives the mathematical models for oil flow through hydraulic tappet as well as those of oil jet and plain journal bearings. And the flow from an oil pump and the flow resistance through an oil filter is considered at various temperatures. In the analysis, the various design guidelines are applied. The distribution of flow and pressure of an engine lubrication system are calculated, and the pressure data compared with the experimental data at a few points in the engine lubrication system. This method is helpful to design of engine lubrication system efficiently.

Key words : engine lubrication system, network analysis, flow rate, pressure drop, centrifugal pressure

Introduction

The quality of engine lubrication is depending upon how much oil is supplied and how lubricant is pressurized to the lubricated components. This state of lubrication is closely related with the safe operation of an engine and the lifetime. Specially, in the concept design stage of an engine, the experimental verification of the engine lubrication system can not be performed. Therefore, the practically optimized analytical method has been required by engine designers. Some methods have been developed by several researchers[1-8]. Up to date, most of lubricated components of engine have successfully been developed the analytical model for the flow characteristics. However, the study on the flow through hydraulic tappet and bearings on camshaft with internal oil passage seem to be insufficient.

Therefore, this study is focused on the flow model through cam bearings and hydraulic tappet as well as periodical flow through oil jet on big end of connecting rod. Also, the pressure resistance and pressure gain as the lubricant approach to and leave from the oil bores on crankshaft and camshaft, are considered.

In this paper, the general flow network theory and the flow characteristics of each lubricated component are described. The flow characteristics are systematically structured in order to analytically simulate the flow network of engine lubrication system. In the analysis, the various design guidelines are applied. This paper gives the mathematical models for oil flow through hydraulic tappet as well as those of oil jet on connecting rod and plain journal bearings. The effect on inlet pressure of bearings by centrifugal pressure at rotating shaft is considered. For the calculation of bearing flow, the effective temperature is used. The distribution of flow and pressure of an engine lubrication system are calculated, and the pressure data compared with the experimental data at a few points in the engine lubrication system. This method is helpful to design efficiently of engine lubrication system at concept stage in terms of selecting oil pump size and choosing oil gallery size.

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Engine Lubrication System

The engine for a case study is a small 4 cylinder DOHC gasoline engine. The lubrication system as shown in Fig. 1 is composed of oil pan, suction pipe, oil pump(P), oil filter(F), main bearings(M1-M5), connecting rod bearings(B1-B4), oil jet on big end of connecting rod, cam bearings(CI1-CI5 and CE1-CE5), hydraulic tappets(TI1-TI8 and TE1-TE8), oil bores on crank shaft arm and cam shaft, and horizontal and vertical oil galleries. The lubricant is inhaled by oil pump from oil pan through suction pipe with strainer welded at the front. Then, the lubricant passes through an oil filter, and is distributed to

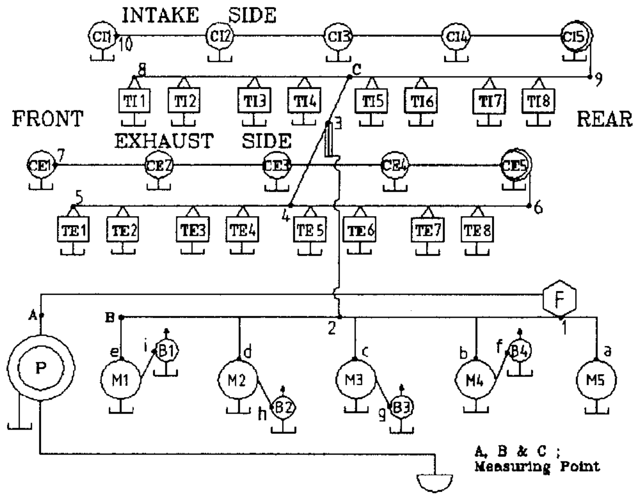

Fig. 1. The schematic diagram of engine lubrication system.

main and big end bearings, and continues to flow to jets on connecting rod. One of branch gallery after filter turns vertically toward the middle area of cylinder head. In cylinder head, there are two main oil galleries for the lubrication of cam bearings and hydraulic tappets at each intake and exhaust side respectively. First, the lubricant is supplied to the hydraulic tappets. Then, at the end of the cylinder head main oil gallery, the lubricant is supplied to the rear cam bearing. In the middle of the camshaft, there is a long oil bore for lubrication of other cam bearings. The lubricant passed through valve train at cylinder head is vertically down through several oil bores to the oil pan.

Theory

Flow Network Theory

In order to decide the flow rate and pressure drop through flow network, first, the theory of mass conservation may be imposed at each junctions of flow network. Therefore, the algebraic summation of all flow entering a junction is zero. For the incompressible flow, at a node, the continuity equation is

$$\sum_{i=1}^{n} Q_i = 0 \tag{1}$$

Second, the continuity of energy per unit mass may be applied along the same stream line. The energy potential between two nodes in a certain pipe is equal. For the incompressible flow on the same stream line, the energy balance equation is

$$\frac{P_a}{\gamma} + \frac{V_a}{2g} + Z_a = \frac{P_b}{\gamma} + \frac{V_b}{2g} + Z_b + h_f \tag{2}$$

where, P_a, P_b, and V_a, V_b, and Z_a, Z_b represent pressure, velocity and elevation from a datum at certain points, a and b. h_f is frictional head loss. g is specific gravity of the fluid, is acceleration of gravity.

Pipe Flow

For the incompressible flow, the pressure drop through a pipe is given by the Fanning equation induced by energy equation (2).

$$P_a - P_b - \rho f(Q,D)\frac{8Q^2}{D^4}\left\{\frac{L}{D} + \frac{K}{f(Q,D)}\right\} - \gamma(Z_b - Z_a) = 0 \tag{3}$$

where, K is dimensionless experimental coefficient accounting for head losses in bend, elbows, T-joints etc. For general elbow, $K = 0.5$, and for T-type joint, $K = 1.0$ [9]. ρ is the density of fluid. Q is volumetric flow rate. D and L are pipe diameter and length respectively. f is a friction factor for pipes of circular cross section. The friction factor for turbulent flow can be decided by Reynolds number(Re), velocity(V) and Hazen-William constant(C). Here, transient region is, in convenience, included into turbulent region. For the smooth pipe, C is 140[10].

(1) laminar flow, Re 2000

$$f(Q,D) = \frac{64}{Re} \tag{4}$$

(2)turbulent flow, Re>2000

$$f(Q,D) = 1304.56\frac{V^{0.0184}}{C^{1.852}Re^{0.1664}} \tag{5}$$

where, Re can be expressed by $4\rho Q/\pi\mu D$. μ is dynamic viscosity of lubricant.

Oil Pump Flow

The oil pump used in the sample engine is sickle type. The flow rate of this volumetric oil pump increases with the angular velocity. If the pressure of oil pump reached up to the opening pressure of by-pass valve, the valve is opened. Then, the flow rate suddenly decreases. The opening pressure on the sample engine is around 4.0 bar. The flow rate and pressure difference of the oil pump on sample engine are shown in Fig. 2. The relationship between flow rate and pressure difference can be expressed by the equation (6).

$$P_{in} - P_{out} - \beta_{1p} + \beta_{2p}Q^k = 0 \tag{6}$$

where, β_{1p}, β_{2p} and k are constants that decided by curve fitting for Fig. 2. As the design guide, the suction speed through suction pipe is recommended less than 3.0 m/sec.

Oil Filter Flow

An oil filter is a sort of flow resistance. The flow rate and pressure difference through an oil filter on the sample engine is shown on Fig.3. The oil filter has a relief valve to keep filter tissue safely under high pressure. The opening pressure difference is around 1 bar. So, the pressure drop through an oil filter can be sustained around 1 bar. The typical flow characteristics can be expressed by equation (7).

$$P_{in} - P_{out} - \alpha_f Q^{kf} = 0 \tag{7}$$

where, α_f and kf are constants that can be obtained by curve

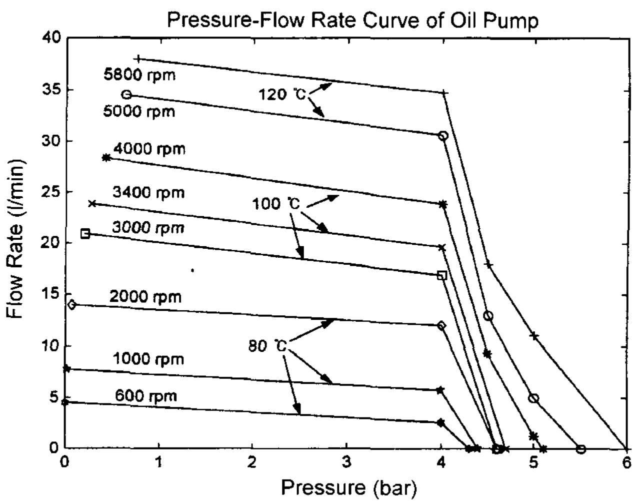

Fig. 2. The flow characteristics of oil pump.

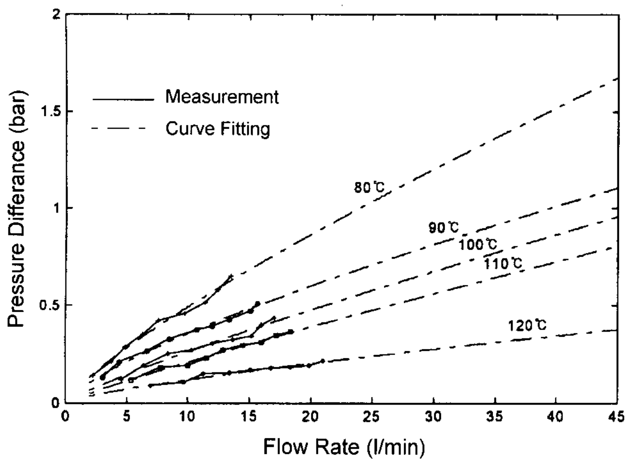

Fig. 3. The flow characteristics of oil filter

fitting for Fig. 3. Here, the data of 90℃ are used for 600 to 2000 rpm, those of 100℃ for 3000 to 3400 rpm, those of 110℃ for 4000 rpm, and those of 120℃ for 5000 to 5800 rpm, respectively. Even the assembly of internal oil cooler and an oil filter were attached on an engine, the flow characteristics could be expressed with the same equation (7) and different values of constants. Here, the original sample engine does not have an oil cooler. If more cooling were needed in order to increase the engine performance, the internal oil cooler may be attached before filter. In this circumstance, the pressure should be increased as much as the further pressure drops by oil cooler. So, the performance upgrade of oil pump may be needed.

Oil Jet Flow

The flow rate through oil jet may be described as the discharge of an incompressible non-cavitating fluid through an orifice.

$$Q = \frac{C_d A_o}{(1-m^2)^{1/2}}\left(\frac{2\Delta P}{\rho}\right)^{1/2} \tag{8}$$

where, $m = \frac{d_o^2}{D_p^2}$, and $A_o = \frac{\pi d_o^2}{4}$.

C_d is discharge coefficient which is described in detail on [8]. d_o is diameter of orifice or exit diameter for nozzle. D_p is diameter of approach pipe of nozzle. The periodical flow through oil jet on big end of connecting rod is considered. The flow rate is represented by time averaging to the oil flow during one cycle.

Centrifugal Pressure in Oil Bore

As the lubricant approach to and leave from the oil bores on rotating shaft, the pressure resistance and pressure gain are occurred. The general expression of centrifugal pressure is

$$P = \frac{1}{2}\rho\omega^2 r^2 \tag{9}$$

where, is angular velocity of rotating shaft. r is the distance between the center of rotating shaft and a certain point on oil bore. Here, It is investigated, in detail, on the various pattern of oil supply to and from rotating shafts.

Flow Resistance at Inlet of Crankshaft Oil Bore

In this paper, the average pressure of main bearing internal pressure is assumed to enter the inlet of oil bore. Further, the average pressure set equal to the unit load of bearing. As a design guide, at least, the average pressure of main bearing internal pressure has to be greater than the centrifugal pressure at the entrance of oil bore for the lubrication of big-end bearing.

The pressure(P_{bore_in})at the entrance of oil bore on crank shaft arm is as follow.

$$P_{bore\_in} = P_{main\_int} - P_{cen\_in} \tag{10}$$

where, P_{main_int} is the average internal pressure of a main bearing. $P_{cen\_in} = \frac{1}{2}\rho\omega^2 r_{cen\_in}{}^2$. r_{cen_in} represents the distance between the rotating center of crank shaft and the entrance of oil bore on crank shaft arm.

Pressure Gain at Inlet of Connecting Rod Bearing (or Exit of Crankshaft Oil Bore)

The pressure(P_{con_in}) at the exit of the oil bore on crank shaft arm can put equal to the inlet pressure of a big-end bearing. The pressure may express as follow.

$$P_{con\_in} = P_{bore\_in} - P_{fric} + P_{cen\_ex} \tag{11}$$

where, $P_{cen\_ex} = \frac{1}{2}\rho\omega^2 r_{cen\_ex}{}^2$. r_{cen_ex} is the distance between the center of rotating shaft and the exit at oil bore on crank shaft arm.

Flow Resistance at Oil Bore Inlet of Rear Cam Journal

The inlet pressure(P_{cam5_in}) at entrance of oil bore at rear cam journal in considering together with the centrifugal pressure can be expressed as follow.

$$P_{cam5\_in} = P_{camg\_in} - P_{cen\_cam5} \tag{12}$$

where, P_{camg_in} is the pressure at the end of main gallery of cylinder head. $P_{cen\_cam5} = \frac{1}{2}\rho\omega^2 r_{cam5}{}^2$. r_{cam5} is the radius of rear cam bearing. As a design guide, P_{camg_in} should be greater than P_{cen_cam5}.

Pressure Gain from Cam Shaft Oil Bore to Other Cam Bearings

The inlet pressure(P_{camoth_in}) for other cam bearings can be calculated as follow.

$$P_{camoth\_in} = P_{cam5\_in} - P_{fric} - P_{cen\_camoth} \tag{12}$$

where, $P_{cen\_camoth} = \frac{1}{2}\rho\omega^2 r_{camoth}{}^2$. r_{camoth} is the radius of other cam bearings.

Bearing Flow

Engine bearings have several types of oil groove for oil supply and the distribution. A circumferential groove with circular holes and a single circular hole without any groove are the typical type. The flow rates for those bearing types were described on [8]. The detail expressions of flow rate applied to each bearing of the sample engine are as follows.

Main Bearing Flow

The general expression of flow rate for a circumferentially grooved bearing is shown on reference[8]. However, the main bearing of the sample engine has a fully circumferential groove. Therefore, the corresponding flow rate [8] is simply given by,

$$Q = \frac{\pi c^3 D}{6\mu(L-a)}(2+3\varepsilon^2)\Delta P \tag{14}$$

Here, the eccentricity (ε) was chosen as the average value of the real eccentricity during one cycle. ΔP is the bearing inlet pressure(P_{bin}, Pa) minus ambient pressure(P_a, Pa). is the width of groove(m). μ is the viscosity of lubricant(Pa.s). is the width of a bearing(m). D is the diameter of a bearing(m). c is the radial clearance(m).

Big-End Bearing Flow

The big-end bearing of the sample has only a small circular hole without circumferential groove for the supply of lubricant. The oil flow rate for the non-grooved bearing with a small circular hole [8] is given by,

$$Q = Q_F + Q_H - 0.3(Q_F Q_H)^{1/2} \tag{15}$$

where, the flow rate(m^3/sec) based on feed pressure only,

$$Q_F = 0.675\frac{hg^3}{\mu}\left(\frac{d}{L}+0.4\right)^{1/2}\Delta P \text{ with } h_g = c(1+\varepsilon cos\delta)$$

at the position of oil hole, d is the diameter of oil hole(m). The flow rate(m^3/sec) based on hydrodynamic effect,

$Q_H = \frac{4c^3\vec{W}}{\mu LD}\left(\frac{L}{2R}\right)\vec{M}$, $\vec{W}$ is the bearing load(N), $\vec{M}$ is the mobility number $\left(\frac{LD\mu/c}{|\vec{W}|(c/R)^2}\dot{e}\right)$.

Cam Bearing Flow

At the rear cam bearing area, the journal has a fully circumferential groove and a vertically drilled oil bore as shown on Fig. 4. The oil bore is connected with a long oil bore in the middle of camshaft. The half of the journal put on the cam bore of cylinder head and a bearing cap is assembled on the journal. For this mechanism of rear cam bearing, the equation (14) is used for the calculation of the flow rate. And the bearing of other cams has only a small circular hole without circumferential groove for the supply of lubricant. So, the flow rate may be expressed by equation (16). However, in this paper, for the flow rate calculation, the term of Q_F is only considered because the change rate of eccentricity on cam shaft bearing less severe than those of big-end bearings.

Hydraulic Tappet Flow

The expression of flow between eccentric cylinders can be used for that through hydraulic tappets. The flow through

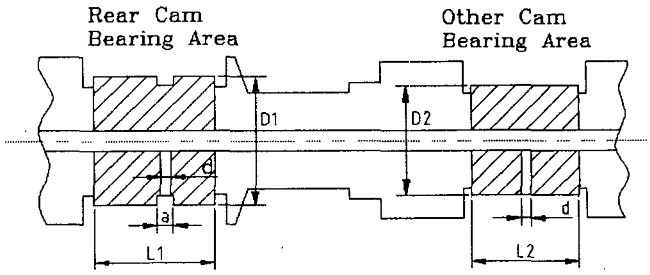

Fig. 4. The schematic drawing of camshaft around bearing area.

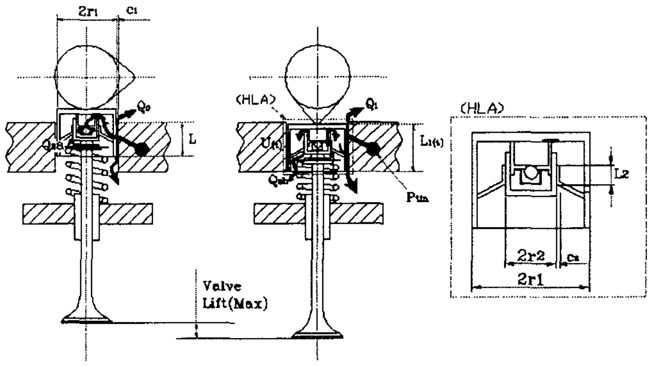

Fig. 5. The geometry of hydraulic tappet and the flow schematic drawing.

tappet exists between a tappet and the housing, and between a plunger and the guide funnel. The tappet is pushed by cam lobe. But, at stage of cam base circle, the tappet does not move. At the phase of cam lobe, the tappet moved with a certain speed. Therefore, the flow rate will be considered for the two stages separately as shown on Fig. 5.

Flow Rate at Base Circle Phase

The flow rate through between a tappet and the housing is

$$Q_o = \frac{\pi r_1 c_1^3 \Delta P}{6\mu L}(1+1.5E^2) \tag{16}$$

The flow rate through between a plunger and the guide funnel is

$$Q_{2a} = \frac{\pi r_2 c_2^3 \Delta P}{6\pi L_2} \tag{17}$$

Flow Rate at Cam Lobe Phase

The flow rate through between a tappet and the housing is

$$Q_1 = \frac{\pi r_1 c_1 U(t)}{2} + \frac{p r_1 c_1^3 \Delta P}{6mL_1(t)}(1+1.5E^2),\ \theta_s \le \theta \le \theta_e \tag{18}$$

The flow rate through between a plunger and the guide funnel is

$$Q_{2b} = \frac{\pi r_2 c_2^3 \Delta P}{6\mu L_2},\ \theta_s \le \theta \le \theta_e$$

where, θ_s and θ_e are the cam angle at valve opening point and valve closing point respectively. ΔP is the supply pressure(P_{tin}) minus the ambient pressure(P_a). E is the eccentricity between a tappet and the housing. Here, the average value is used for the eccentricity between a tappet and the housing, i.e. E = 0.33. And, the eccentricity between a plunger and the guide funnel is neglected. r_1 and r_2 are the nominal radius of tappet and plunger respectively. c_1 and c_2 are the clearances between a tappet and the housing, and between a plunger and the guide funnel, respectively. L and $L_1(t)$ are the sealing length between a tappet and the housing at base circle phase and at cam lobe phase, respectively. L_2 is the sealing length between a plunger and the guide funnel. Here, the movement of the plunger is neglected.

Oil Viscosity and Density

The density(kg/m^3) and kinematics viscosity(cst) of engine lubricant can be expressed by equation (20) and equation (21) with constants, a, b and c of which values are varied depending on SAE oil grade.

$$\rho = 0.0361(a - 0.000354T_f) \cdot 27680 \tag{20}$$

$$v = 10^{10(b - c\log_{10}(T_r))} - 0.6 \tag{21}$$

where, T_f and T_r represent the Fahrenheit temperature and Rankin temperature, respectively. The values of a, b, and c [11] are used 0.9071, 7.7649, and 2.7360 for SAE 5W30 engine oil.

In general, the lubricant temperature at oil gallery on engine lubrication system can be assumed to be equal to that in the oil pan. However, by the bearing rotation, the fluid shear resistance is occurred, and frictional heat is generated into the lubricant between the journal and bearing shell. Further, the power loss is occurred. The power loss turns out to be increasing on the temperature of the lubricant in bearings. Therefore, the increment on the temperature can be considered in the expression of an effective temperature [12] as follow.

$$T_{eff} = T_{in} + \left(\frac{0.8 \bullet 10^6 \bullet P_{loss}}{Q\rho C_o}\right) \tag{22}$$

where, T_{in} is the inlet temperature(°C) of a bearing. T_{loss} is the power loss(watt) of a bearing. Q is the flow rate(cc/sec) through a bearing. ρ is the density(kg/m^3) of lubricant. C_o is the specific heat(J/kg. °C) of the lubricant. Here,

$$C_o = 1796 + \frac{691}{160}T_{eff} \text{ for engine oil [13].}$$

System Analysis

In general, under a certain velocity, the flow rate of oil inside pipes increases as the oil pressure increases. Therefore, the higher the discharging pressure of oil pump is, the more the oil is needed. On the other hand, the flow rate of oil pump is inversely proportional to the discharging pressure. To satisfy the flow rate needed by an engine, the flow rate from oil pump is equal to the total flow rate needed by each lubricated components of an engine.

Therefore, as the first step to solve the whole system of flow network, the flow rare of suction by oil pump is assumed. The discharge flow rate is calculated using measuring volumetric efficiency of oil pump. Then, the discharge pressure is determined by the flow characteristics curve of oil pump. Further, it is calculated the pressure drop through each passage and the consumed oil flow rate by each lubricated parts. The consumed flow rate is checked whether it is in the same range of the assumed discharge flow rate or not. If the results were not satisfied within a certain range of error, the procedure is to be iterated with a better-adjusted value of the discharge flow rate till the value is satisfied. Finally, the discharge flow rate is decided, and the pressure and flow distribution at every points and components can be obtained.

Computer Program

The computer program is composed of a main program and several subprograms. In the main program, all the data of lubrication system are recorded. Those are the physical geometry of the related components and the experiment data of engine and components, for example, the flow characteristics of oil pump and oil filter. And the network frame of lubrication system is organized. The whole network is divided into several small networks, for examples, the bearing group 1 for rear main bearing, the bearing group 2 to 5 for each set composed of a main bearing, a big-end bearing and a oil jet, the hydraulic tappet group 1 and 2 for each intake side and exhaust side, and the cam bearing group 1 and 2 for each intake side and exhaust side. The involved subprograms are of the calculation of basic equations for pipe flow, oil pump flow, oil filter flow, bearing flow including bearing load, effective temperature, eccentricity and power loss calculation with pressure data of combustion chamber, centrifugal pressure of rotating parts, oil jet flow, hydraulic tappet flow, and the expression of density and viscosity depending on temperature. During calculation, a certain assumed flow rate for each group is forced to converge by iteration. If the iteration were successfully performed for the whole lubrication system, then all the calculated data are written on the output files.

Results

In this case study, a small 4 cylinder DOHC gasoline engine is handled. The oil pump is a sickle type. The SAE 5W30 engine oil is used. The temperatures of the engine oil for each RPM are shown on Fig 6.

The measuring volumetric efficiency of oil pump at each rpm for the given pressure is shown on Fig. 7. Here, the efficiencies at 600 to 2000 rpm are of 80°C, those at 3000 to 4000 rpm of 100°C, and those at 5000 rpm to 5800 rpm of 120°C, respectively. For the better results of network analysis, it is necessary to get the exact data of volumetric efficiency corresponding to the temperature at each rpm.

From the given physical input data and operation conditions, the flow rate and pressure distribution are calculated.

In the Fig. 8, the calculated suction flow rate is compared to

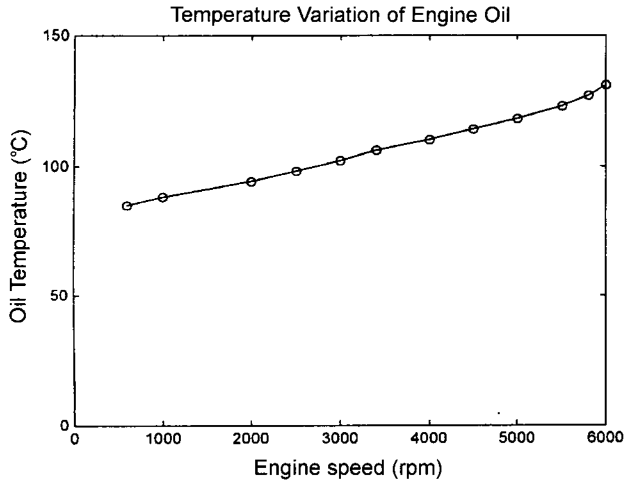

Fig. 6. Temperature of engine oil.

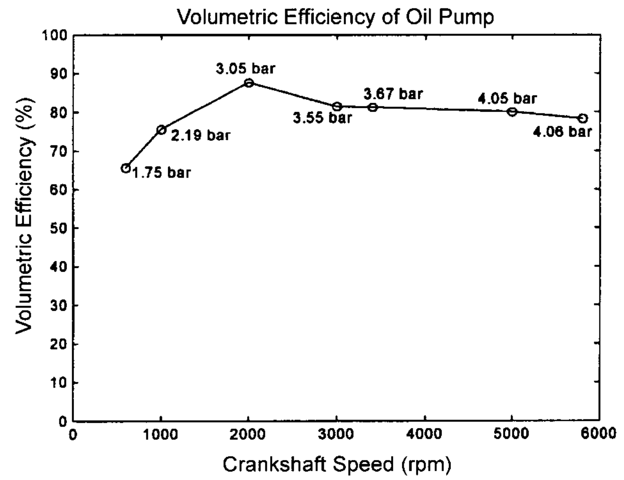

Fig. 7. Volumetric efficiency of oil pump.

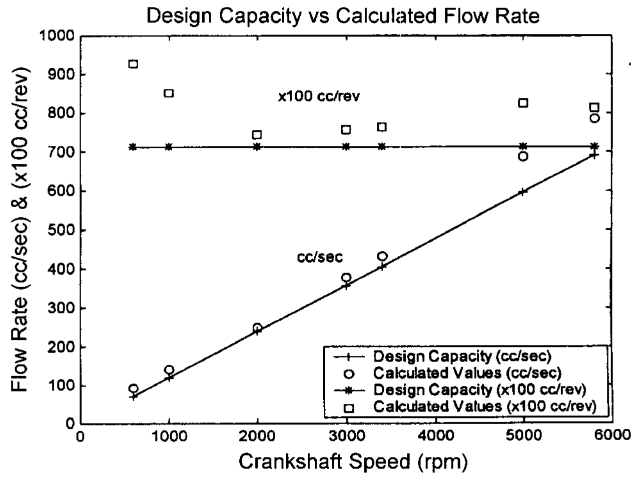

Fig. 8. Theoretical vs. calculated flow rate.

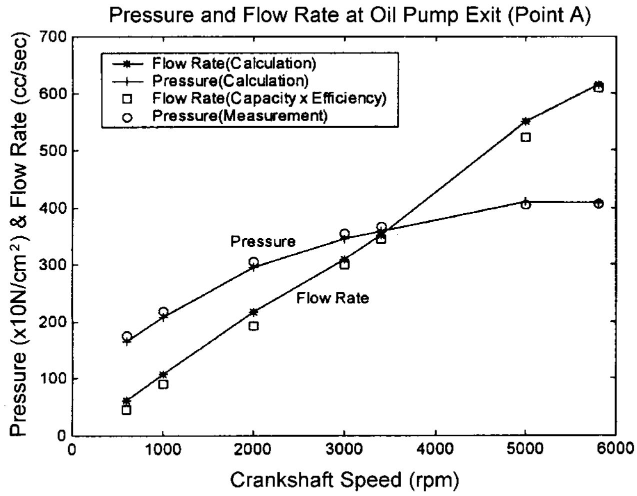

Fig. 9. Pressure and flow rate at oil pump exit.

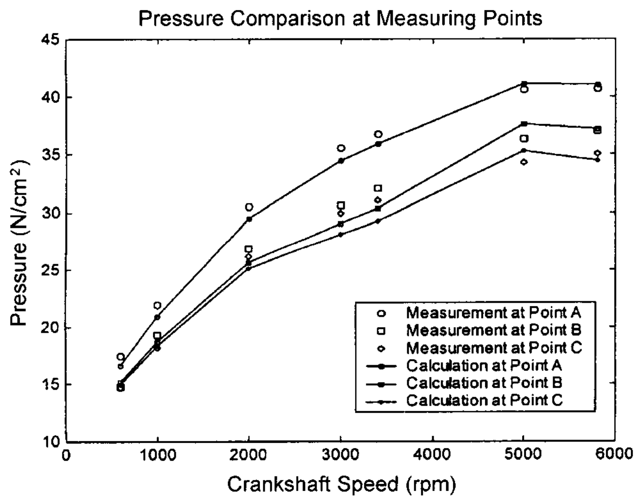

Fig. 10. Pressure vs. RPM at point A, B & C.

the theoretical flow rate. From the results, the capacity improvement of oil pump is required to all rpm ranges. The maximum required oil flow capacity of oil pump is about 9.30cc/rev at 600 rpm.

The relationship of pressure and flow rate at oil pump exit is shown on Fig. 9. The real flow rate is compared to the calculated values at every measuring rpm. Also, the calculated and real pressure distributions are plotted together. It turns out that the required flow rates for the whole lubrication system are greater than the real flow rate for each rpm like the suction flow rate in Fig. 8. The pressure at oil pump exit is lower than the measured pressure at most rpm. But, at 5000 and 5800 rpm, the calculated pressure is higher than the measured value. The error in oil pump exit pressure between the calculation and the measurement is within 5.2%.

The measuring data of pressure at point A, B and C in Fig. 1 are compared to the calculated values at each RPM in Fig. 10.

The Point A is the oil pump exit. The pressure data at point A is same as those in Fig. 9. The calculated pressures at points B and C are lower than the measured values except the pressure at 5000 rpm. The error between the calculated and measured values is within 6.0%.

The oil pressure distributions of engine lubrication system at major calculated points are plotted on Fig. 11. The maximum

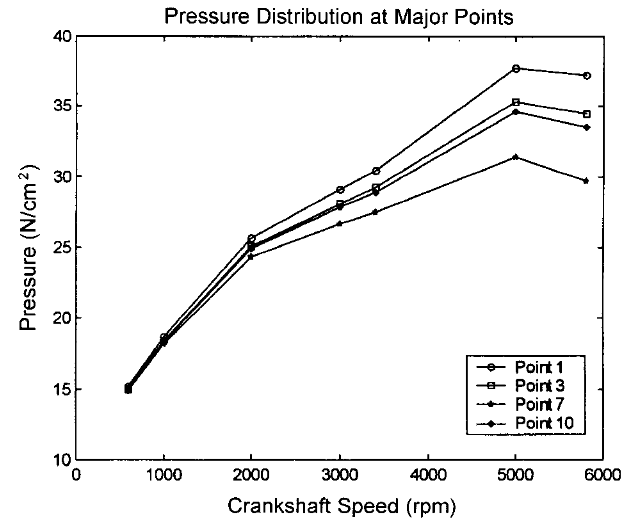

Fig. 11. Pressure vs. RPM at main calculated points.

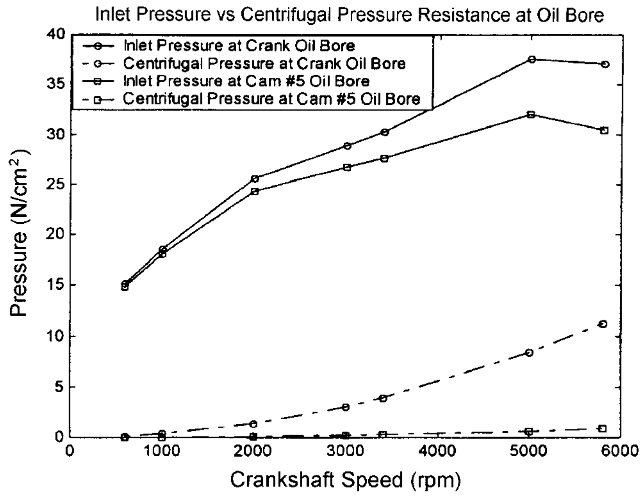

Fig. 13. Bearing entrance pressure vs. centrifugal pressure at big-end bearing and cam bearing.

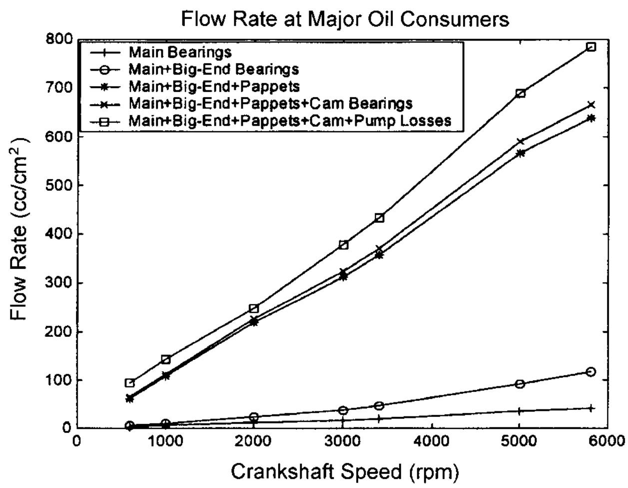

Fig. 12. Flow rare vs. RPM at main oil consumer.

pressure drop through the lubrication system after Point A is about 11.22 N/cm^2 at Point 7and 5800 rpm. The pressure drop through oil filter after Point A is about maximum 3.73 N/cm^2 at 5800 rpm. The pressure drop through the vertical gallery is about maximum 2.68 N/cm^2 at 5800 rpm. The maximum pressure difference between Point 7 and Point 10 is about 3.86 N/cm^2 at 5800 rpm. The reason of this big pressure drop is the big clearance between hydraulic tappet and the bore at exhaust side. The radial clearances of exhaust and intake side are 145.5 μm and 28.5 μm, respectively. The big clearance induces the increase of flow rate through hydraulic tappet at exhaust side, and brings in the pressure drop.

The flow rate distribution of main oil consumer in the lubrication system is plotted on Fig. 12. The biggest consumer of oil is the hydraulic tappets at exhaust side. The flow rate through tappets at exhaust side is higher around 35 to 40 times at each rpm than that at intake side. The reason is the big clearance as described above.

In Fig. 13, the centrifugal pressure resistances are compared to the inlet pressure at the oil bore of rotating shafts. Both inlet pressures to the oil bore at crankshaft arm and the entrance oil bore at rear cam are much higher than the centrifugal pressure of oil inside oil bore by shaft rotating. Therefore, the centrifugal pressure resistance does not block the oil flow in the lubrication system.

Conclusion

In this paper, the way of the network analysis of engine lubrication system is described orderly and in detail. For the analysis, the more in detail the flow-pressure data of oil pump and oil filter based on temperature by component rig tests are, the better the calculation results can be obtained.

As the results of the network analysis of the sample engine lubrication system, the suction flow rate is lower than the required flow rate by all the consumer of oil in the lubrication system. The pump capacity needs to be improved up to 9.30 cc/rev. Other design guides are satisfied. It can be said that the calculation results are within 10% errors comparing to the measurements. It may say that the errors are brought from the various assumptions, for examples, effective temperatures inside bearings, average eccentricities of bearings and no thermal expansion of material for decisions on clearances.

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