

Development of Analytical Model for Design of Gerotor Oil Pump and Experimental Validation

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ABSTRACT

Increasing the efficiency of the Engine parts and reduction in development time with good accuracy are the challenges in the Automotive Industry. Lubricating oil pump has been selected for this study. Existing literatures explain the methodology to generate the rotor profile from the given geometrical parameters of the rotor like eccentricity, tooth radius etc. Invariably the specifications to design the pump are provided in terms of pump performance at various operating conditions. The analytical model developed in this study uses the performance and boundary specifications to generate the rotor profile and to estimate the flow rate at various operating conditions of the pump. This methodology includes the generation of trochoidal profile for inner rotor and modified conjugate profile for the outer rotor and the volume calculation of number of chambers (N) which are created between the rotors during meshing. The flow is estimated by solving a system of $N+2$ coupled differential equations using a unique solver code. Various designs are carried out by modifying the rotor parameters and the optimum design is selected based on the flow ripples, geometrical interference, minimum flow margin etc., Based on the analytical model, an oil pump prototype is made and validated experimentally. Correlation between the analytical and experimental results established with in 5%. This analytical model can be used to design the gerotor oil pump for any application in a very short time.

INTRODUCTION

The main focus on today's automotive industry is the reduction in fuel consumption and emissions coming out from the engines. Lubricating oil pump systems can make a significant contribution in this direction and depending upon type of oil pump, the potential for reduction in fuel consumption could be up to 3% in the European MVEG-Cycle [1]. Gerotor (Generated Rotor) is one type of positive displacement lubricating pump, which is a critical part in the operation of an automotive engine. It utilizes the power developed in the engine to raise the oil from atmospheric to higher pressure, which is used for lubrication of engine parts. Gerotor pumps are generally designed using a trochoidal inner rotor and an outer rotor formed by a profile with intersecting circular arcs. The eccentrically mounted inner rotor is having one tooth less than the outer rotor and the teeth are having continuous sliding contact with outer rotor. The pump operates on a principle that the working fluid is drawn into and squeezed out of a cavity (sealed pocket) formed by two rotating, intermeshing gears. When one portion of the cavity opens, the liquid is filled in. As the rotation continues, that portion of the cavity closes and the liquid is squeezed out, and an adjacent portion opens. Continuous rotation of the rotors will raise the pressure in the delivery chamber and is pumped out of the delivery chamber to the delivery line where it is consumed by the oil consumers.

LITERATURE REVIEW

A great deal of the development of internal gear pumps was carried out by M.F.Hill and there are many patents in his name. The spur gear and the external gear are called by Hill the inner and outer rotors. He describes how the tooth shape of the outer gear can be chosen freely and the inner gear is formed using a tool, which is shaped like the tooth of the outer gear. While he recognizes that any oval shape will do for the outer gear tooth specifically mentions circles, cycloids, trochoids and lastly ellipses [1]. Colbourne [2] classified the trochoidal curves and their trochoidal envelopes to be potentially applicable in pumps or engines. He has used bond graph to predict the performance parameters. Yasuyoshi Saegusa et al [4] developed a system to design Oil pumps with trochoidal profile rapidly. Chiu-Fan Hsieh et al [11] explains the gerotor profile design with conjugate profiles and offset concept using a parametric model to achieve the high area efficiency without interference. The generation of conjugate inner rotor profile based on the outer rotor profile and the design optimization based on the performance index are discussed by Gamez-Montero et al [7], Kim et al [9] and Mimmi et al [12]. Colbourne [3] describes the effect of tooth shape on the flow rate. Baek-Hyun cho et al [8] suggested a theoretical model with moderate complexity to find effective bulk modulus considering entrained air effect at real operating conditions. Fabiani et al [6] developed the inner rotor with outer rotor parameters and the outer rotor is developed with three circular arcs. They used AMESim to carry out the simulation and the results are validated with experiment. P J Gamez et al [10] made analytical work on geometry and kinematics of trochoidal profiles and volumetric characteristics. The bond graph technique is applied to model a trochoidal-gear pump based upon an ideal representation in order to simulate the instantaneous flow and to optimize the trochoidal profiles.

In our present study, in order to design the oil pump based on the performance and boundary specifications, an analytical model is developed in which the flow rate at various operating conditions of the pump is estimated based on the generated rotor profile and the results are verified experimentally using the prototype.

METHODOLOGY

1. ROTOR PROFILE PARAMETERS

In the proposed analytical model, the rotor profile parameters are arrived using the theoretical flow capacity and the boundary specifications of the pump. The theoretical flow capacity (Q_t) is estimated from the performance specification and target flow margin of the pump. Usually the pump performance is specified in terms of the volume flow rate required at various engine speed and delivery pressure conditions of the pump. Based on the engine requirement,

considering the rotor profile parameters, theoretical flow is estimated. From the [figure 1](#), we could observe that the flow margin (f_m) is the difference in flow rate between the calculated (Q_c) and the engine requirement (Q_s) at different speed and pressure conditions. Usually this is represented as percentage of theoretical flow capacity. The target flow margin percentage is estimated considering the uncertainties like the expected variation between the analytical and the experimental results and the variation due to manufacturing tolerances etc. The theoretical flow capacity is calculated iteratively by ensuring the target flow margin percentage at all the specified operating condition of the pump.

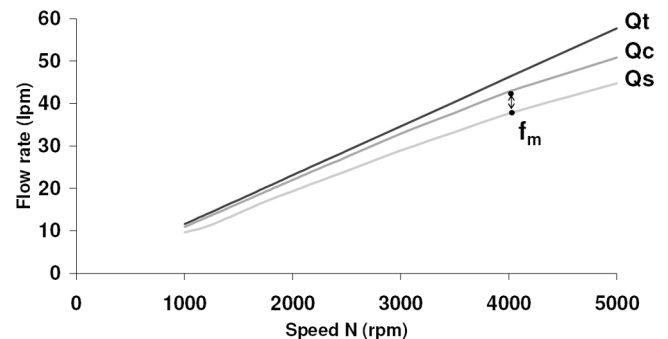


Figure 1. Estimation of theoretical flow capacity

Apart from the estimation of theoretical flow capacity, the pump boundary specifications are required to arrive at the inner rotor parameters. The boundary specification is the size of the space envelope within which the pump geometry to be constrained. The rotor profile boundaries namely inner (D_i) and outer (D_o) boundary diameters are calculated from the specification considering minimum wall thickness of rotor (Refer [Figure 2](#)). The relationship between the rotor profile parameters like Number of teeth of inner rotor (Z_2), eccentricity (e), base circle radius (r_{bi}), rolling circle radius (r_{ri}), cutter circle radius (r_c), and arm length (d_i) are derived based on the rotor assembly relationships (Refer [Figure 3](#)).

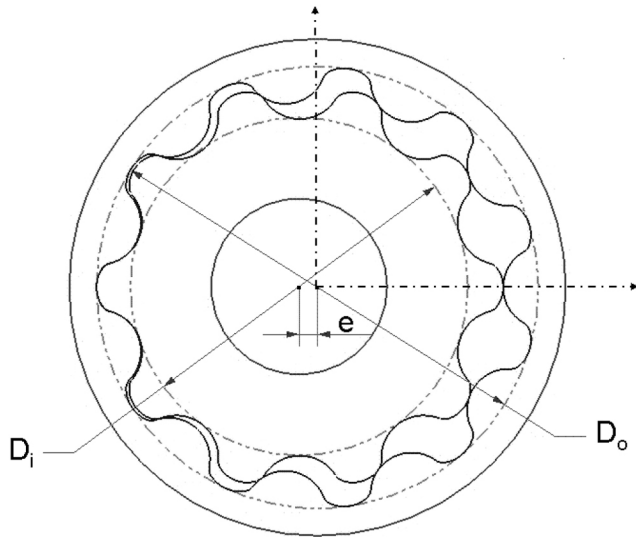


Figure 2. Inner rotor profile parameters

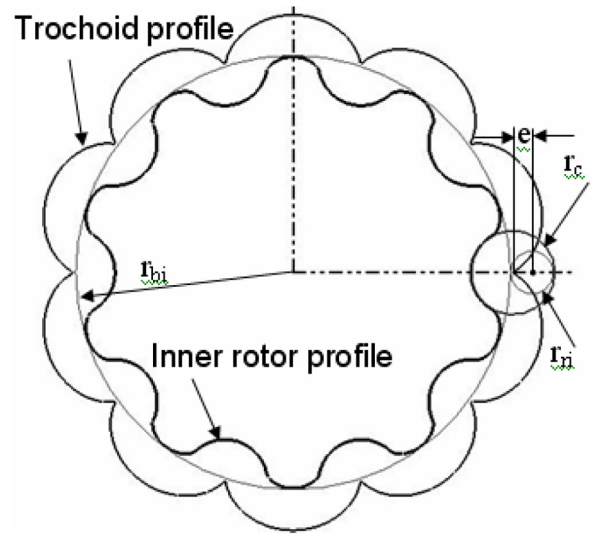


Figure 3. Inner rotor profile

Within the geometrical boundary limitations of the inner and outer profile boundary diameters, the feasible minimum and maximum number of teeth of inner rotor are calculated. The extreme limits of eccentricity (e) are calculated for each tooth in the feasible zone. The limits of cutter radius are calculated for each feasible tooth and eccentricity combination. The geometrical tooth interference is avoided by maintaining the cutter radius less than the allowable maximum value[7]. In case of undercutting, the profile can be modified by altering the arm length to certain extent to avoid the interference for each feasible number of teeth, eccentricity and tooth radius combination. Number of alternate profile designs are generated by varying the parameters successively with suitable increments, within the feasible limits. The optimum design is selected based on the geometrical interference, minimum flow margin, flow ripple etc and the same can be used as input for rotor profile design.

2. INNER ROTOR PROFILE

After calculating the rotor profile parameters based on the boundary specifications and theoretical flow capacity, the trochoidal path and the inner rotor profile are generated by using the inner rotor parameters. In the generation of inner rotor profile, it is considered that the rolling circle with radius r_{ri} rolls over a base circle of radius r_{bi} without any slip. The locus of a point located on the rolling circle with an eccentricity (e) from its centre forms the trochoid profile (Refer Figure 3).

The co-ordinates for generating the trochoidal profile is given by

$$x_{ti} = (r_{bi} + r_{ri}) \cos \theta + e \cos((Z_2 + 1)\theta) \quad (1)$$

$$y_{ti} = (r_{bi} + r_{ri}) \sin \theta + e \sin((Z_2 + 1)\theta) \quad (2)$$

Where, θ is the absolute angle of the rolling circle centre about the base circle centre.

In order to get the inner rotor profile, the trochoidal profile generated using the above equation is used along with the cutter radius, which forms the inner rotor profile.. From [figure 3](#), we could observe that this path is formed by the edge of a rotating cutter with radius (r_c) whose centre moves along the trochoidal path generates the inner rotor profile (Refer [Figure 3](#)). The co ordinates of the generated inner rotor profile is given by

$$x_i = x_{ti} \pm \frac{r_c}{\sqrt{(1 + u_i^2)}} \quad (3)$$

$$y_i = y_{ti} \pm \frac{u_i r_c}{\sqrt{(1 + u_i^2)}} \quad (4)$$