DESIGN AND CFD ANALYSIS OF EXTERNAL GEAR PUMP

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Abstract— Gear pumps are commonly found in Turbine engine high pressure part of fuel system. Their objective is to start the engine, to provide take off flow at the required pressure and to prime a fuel system. Development of Electric Fuel Metering pumps for turbine engine has optimized the fuel system by eliminating the need for control valves. The correct prediction and control of flow rate in a gear pump is crucial for efficient fuel metering. In this paper an approach has been presented for obtaining the above mentioned prediction in a simple analytical and numerical method. Gear pump geometry is modeled in Pro/Engineer software. Pro/Mechanism module is used to avoid gear tooth overlapping. External Gear Pump mechanism is also studied using MATLAB/SIMULINK. The gear pump assembly is then simulated in Pumplinx software to obtain transient solution.

Keywords— Gear Pump, Turbine, values, Fuel System.

I. INTRODUCTION

External gear pump [1] is a positive displacement pump. It consists of two meshed gears, suitably housed with a shaft attached to one of the gears. Close tolerances are held between the housing and the gear sides and periphery to prevent excess leakage. Fluid is transported in the spaces between the teeth in the periphery of both gears, and the meshed segment of the of the gears serve as a seal between inlet and outlet. Unlike centrifugal pump [2] which produces constant pressure with flow rate depending on the restriction downstream, a gear pump produces constant flow rate with pressure depending on the restriction downstream.

Gear pump [3] has now become the main choice of fuel system designer due to its long life, minimum maintenance, high reliability, ability to operate on low lubricity fuel, low heat input to fuel, small size and low weight.

II. PUMP DESCRIPTION

Figure 1 shows [4] the nomenclature for the two meshed gears. Gears used in the analysis are spur gears with involute teeth. The number of teeth on gear 1 is given by \( N_1 \) ad the number of teeth on gear 2 is given by \( N_2 \). In the analysis that follows, the number of teeth on each gear is equal and therefore the gears will be identical. Outside diameter will be identical by \( D_o \), root diameter by \( D_r \), base dimater by \( D_b \) and the pitch diameter by \( D_p \). The center distance between the driving and driven gear is given by \( C \). The thickness of the gears is given the dimension \( W \) (not shown in figure). The subscript 1 and 2 denote the driving and driven gear respectively. The driving gear rotates at an angular velocity \( \omega_1 \). The driven gear rotates in the opposite direction at an angular velocity \( \omega_2 \).

External gear pump consists of two spur gears separated by a distance equal to pitch diameter. Pressure angle is selected on a basis of number of teeth on the driving and driven gear. Shaft is connected to one gear while the other gear rotates due to meshing of gears. Both the gears rotate within the housing. A clearance is made between tip of the gears and the housing which is known a tip clearance.

External gear pump is operated by filling the space surrounded by three sided wall (gear teeth and housing) with the fluid. The pump then imparts momentum to the fluid due to the rotational motion of gears. The fluid is then discharge at a constant flow rate when one of the sides (housing) opens up. The meshed portion in the middle separates the high pressure region on the outlet to the low pressure region in the inlet and is not used for imparting momentum to the fluid.
III. IDEAL FLOW RATE MODEL

Gears used in the study are spur gears with involute teeth. Geometric calculations of gear are carried out on an ANSI/AGMA [4] standard. The gears are hobbed out standard gear cutter. Gap between gear teeth and casing is maintained at a known value with a tolerance of 2mm. Both driving and driven gear has equal number of teeth. Standard pressure angle is selected for the gears. Gears are modeled on Pro/Engineer [5] 3D CAD software. Surface Area of the gear is measured using analysis module of CAD software. Area entrapped by single gear teeth was evaluated by subtracting total circular surface area by gear teeth surface area. Volumetric displacement for a single gear is found out by dividing total entrapped volume per gear by the angle in radians the single gear move to discharge total entrapped volume. The procedure is as follows:

Surface Area of Single Gear evaluated by CAD software\(S_G\)

Number of Teeth\(=N_1=N_2\)

Thickness of Gear\(=w\)

Outer Diameter\(=D_O\)

\[A_{DO} = \pi \left(\frac{D_O}{2}\right)^2\]

Total entrapped area per gear\(=A_{etr(G)} = S_G - A_{DO}\)

Total entrapped Volume per gear\(=V_{etr(G)} = A_{etr(G)} \times \omega\)

\[\text{Entrapped Area in a tooth} = A_{etr(T)} = \frac{A_{etr(G)}}{N}\]

\[\text{Entrapped Volume in a tooth} = V_{etr(T)} = A_{etr(T)} \times \omega\]

Volumetric displacement\(=V_D = 2 \times \frac{N}{2} \times \frac{V_{etr(T)}}{\pi}\)

Ideal Flow rate is then calculated by the product of Volumetric Displacement and angular velocity and is given by

\[Q_{ideal} = V_D \times \omega\]

Flow rate calculated using this method has a linear relation with rotational speed. This model is independent of the pressure difference between gear inlet and outlet.

IV. PRO MECHANISM SIMULATION

The gear pump assembly was simulated in Pro Engineer Mechanism [5] module in order to find out the over lapping of gear teeth. Pin joint was selected for both the driving and driven gear. Gear connections was chosen with gear ratio of 1:1. Servo motor was applied to driving gear with a constant angular velocity of 1 deg/sec. Main aim is to avoid overlapping of gear tooth. The external gear pump mechanism was analysed using start time 0, end time 100, Frame rate of 100, Minimum interval of 0.1. Gear circular tooth thickness was reduced to some reasonable value in order to avoid gear tooth overlapping.

V. MATLAB/SIMULINK MECHANISM SIMULATION

MATLAB/SIMULINK [6] model was set up which shows the 2D representation of 3D External Gear Pump Assembly. Here Solver Configuration defines solver settings to use for simulation. World frame provide access to the world or ground frame. Mechanism Configuration sets mechanical and simulation parameters that apply to an entire machine. Rigid Transform block was included to transform slave gear by a distance of 42mm. Revolute Joint was selected...
for both driving and driven gear. A constant torque of 20 was applied to revolute joint of driving gear. Common Gear constraint was selected for both driving and driven gear. The Base and Follower gears rotate about the Base and Follower z-axes respectively. Center distance and gear ratio option was selected for Common Gear Constraint. See Appendix A for MATLAB/SIMULINK [6] Model of an external Gear Pump.

VI. CFD SIMULATION

For Computational Fluid Dynamics simulation the gear pump was first modeled in GAMBIT [7]. Grid is then made in Simerics/PUMPLINX code [9]. Numerical simulation is performed in Simerics/PUMPLINX code [9]. The results thus obtained are compared with analytical solution.

VII. GOVERNING EQUATION OF FLUID FLOW

The most general form of fluid flow and heat transfer equations of compressible Newtonian fluid with time dependency [10] is given as follows:

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{V}) = 0 \tag{1}
\]

\[
\frac{\partial (\rho u)}{\partial t} + \nabla \cdot (\rho u \vec{V}) = \frac{\partial \rho}{\partial x} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} + \frac{\partial}{\partial x} (\rho \tau_{xx}) + \frac{\partial}{\partial y} (\rho \tau_{xy}) + \frac{\partial}{\partial z} (\rho \tau_{xz}) + S_t \tag{2}
\]

\[
\frac{\partial (\rho v)}{\partial t} + \nabla \cdot (\rho v \vec{V}) = -\frac{\partial \rho}{\partial y} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} + \frac{\partial}{\partial x} (\rho \tau_{xy}) + \frac{\partial}{\partial y} (\rho \tau_{yy}) + \frac{\partial}{\partial z} (\rho \tau_{yz}) + S_t \tag{3}
\]

\[
\frac{\partial (\rho w)}{\partial t} + \nabla \cdot (\rho w \vec{V}) = -\frac{\partial \rho}{\partial z} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} + \frac{\partial}{\partial x} (\rho \tau_{xz}) + \frac{\partial}{\partial y} (\rho \tau_{yz}) + \frac{\partial}{\partial z} (\rho \tau_{zz}) + S_t \tag{4}
\]

\[
\frac{\partial (\rho h)}{\partial t} + \nabla \cdot (\rho h \vec{V}) = -p \nabla \cdot \vec{V} + \nabla \cdot (k \nabla T) + \phi + S_h \tag{5}
\]

Where \( \rho \) is the density, \( u, v \) and \( w \) are velocity components, \( \vec{V} \) is the velocity vector, \( p \) is the pressure, \( S \) terms are the source terms and \( \tau \) terms are the viscous stress components which are defined for a Newtonian fluid as:

\[
\tau_{xx} = \lambda \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + 2\mu \frac{\partial u}{\partial x} \tag{7}
\]

\[
\tau_{yy} = \lambda \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + 2\mu \frac{\partial v}{\partial y} \tag{8}
\]

\[
\tau_{zz} = \lambda \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + 2\mu \frac{\partial w}{\partial z} \tag{9}
\]

\[
\tau_{xy} = \tau_{yx} = \mu \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \tag{10}
\]

\[
\tau_{xz} = \tau_{zx} = \mu \left( \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right) \tag{11}
\]

\[
\tau_{yz} = \tau_{zy} = \mu \left( \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right) \tag{12}
\]

Here \( \mu \) is the dynamic viscosity; \( \lambda \) is the second viscosity and a good approximation for gases is taking \( \lambda = -(2/3) \) and a good approximation for gases is taking \( \lambda = -(2/3) \).

In Equation \( \phi \) is the viscous dissipation term. It is always positive and represents the dissipation of mechanical energy into heat. This dissipation term is usually very small except for high Mach Number flows.

\[
\phi = i \left( \frac{\rho}{2} \left( \frac{\partial u}{\partial x} \right)^2 + \frac{\partial v}{\partial y} \right) + \frac{1}{\rho} \left( \frac{\partial p}{\partial x} \right) \left( \frac{\partial u}{\partial x} \right)^2 + \frac{1}{\rho} \left( \frac{\partial p}{\partial y} \right) \left( \frac{\partial v}{\partial y} \right)^2 + \frac{1}{\rho} \left( \frac{\partial p}{\partial z} \right) \left( \frac{\partial w}{\partial z} \right)^2 \tag{13}
\]

\[
h_g = h_e + \frac{1}{2} (u^2 + v^2 + w^2) = i + \frac{p}{\rho} + \frac{1}{2} \left( u^2 + v^2 + w^2 \right) = E + \frac{p}{\rho} \tag{14}
\]

Where \( h_e \) is the enthalpy, which is defined as

\[
h_e = \frac{1}{C_p} \int_{T_1}^{T_2} C_p dT, \text{ where } i \text{ is the internal thermal energy, } E \text{ is the total energy of the fluid which is the sum of internal thermal energy and kinetic energy.}
\]

VIII. GRID MODELING

Figure below shows the size of the inlet and outlet duct for gear pump. The inlet and outlet duct size is 20mm in depth. Gears center are separated by distance of 42mm. Isometric view of external gear pump is shown in Figure. To create a mesh domain, gears are subtracted from double round duct.

![Figure 2Gear Pump Mesh Domain](image-url)
IX. MESHING

CFD Domain once modeled is then transferred to simerics/PUMPLINX software [9] where hexahedron grid is generated. This hex grid is updated after every time step. Figure shows the hex grid for external gear pump.

![Figure 3: Gear Pump Mesh](image)

Boundary conditions are defined to be pressure inlet for inlet and pressure outlet for outlet of duct. Inlet pressure is 101325 Pa and outlet pressure is 500000 Pa. Wall condition is selected for remaining surfaces.

X. FLOW CONFIGURATION AND BOUNDARY CONDITION

The compressibility effects and turbulence inside the gear pump mechanism are the parameters changing the governing equations to be solved. The role of radiation effect, interior wall and boundary conditions on the flow are studied.

1) Compressibility: The fluid in the domain is oil. The compressibility effects are ignored due to the low speeds. For air speed of 4.32 m/s moving inside the duct of dimensions 22 mm x 20 mm, hydraulic diameter is calculated to be:

\[
D_h = \frac{2 \times 22 \times 20}{22 + 20} = 21 \text{ mm or } 0.21 \text{ m}
\]

The Reynolds number is given by:

\[
Re = \frac{(800) \times (4.32) \times (0.21)}{0.079} = 9186.84
\]

where \( \rho_{oil} = 800 \text{ kg/m}^3 \); \( \mu_{oil} = 0.079 \text{ kg/m s} \)

Mach number based in inlet condition is calculated to be:

\[
Ma = \frac{V}{c} = \frac{19}{1460} = 0.0130
\]

Since Mach number is much less than 0.3 therefore incompressible flow assumption will be used in this study.

2) Turbulence: The flow inside the duct is turbulent regardless the Reynolds number. Turbulent Intensity based on Reynolds number at the inlet is calculated to be

\[
I = 0.16 \times \frac{Re_{Ph}^{\frac{1}{8}}}{Re_{Ph}^{\frac{1}{8}}}
\]

\[
I = 0.051
\]

3) Boundary Conditions: Since Navier-Stokes equations are solved inside the domain, no-slip boundary condition is applied to all walls in the domain. Therefore, at all surfaces

\[
u = v = w = 0
\]

4) Numerical Scheme: Flow Convergence criteria is chosen to be 0.001. Upwind numerical scheme is selected for both velocity and pressure. SIMPLE algorithm is chosen for Pressure Velocity coupling method. For turbulence model numerical scheme Upwind is selected for Turbulent Kinetic Energy and Turbulent Energy Dissipation Rate. The convergence criteria is set to 0.001 for turbulence model.

XI. RESULTS OF CFD ANALYSIS

By completion of all the test runs in Pumplinx [9], several key performance indicators were studied to understand the flow characteristics and trends for external gear pump configuration. To understand results, Velocity results and Pressure based results were studied.

1) Total Pressure Contours at 5000 RPM: In this study effect of total pressure of fluid on gear pump mechanism was studied. Both driven and driving gear rotational speed was kept at 5000 RPM. Total pressure rises in the middle of gear pump mechanism up to certain value and then drops off. Total pressure at inlet is lower than outlet. Figure below shows total pressure contour at 5000 RPM rotational speed. The total pressure ranges from 2030 Pa to \( 1.31422 \times 10^6 \text{ Pa} \).
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2) Pressure Contours at 5000 RPM: Figure below shows pressure effect of fluid on gear pump mechanism. It is clear that pressure at the outlet of gear pump assembly increases to certain value and then drops off. Also gear pump mechanism is designed such that the pressure does not increase to the critical value. Pressure Contours ranges from 171 Pa to $3.5762 \times 10^6$ Pa.

3) Velocity Contour at 5000 RPM: Figure below shows velocity contour at 5000 RPM. It is clear that the velocity contours remain constant within gear pump mechanism. Velocity contours ranges from 0 Pa to 44 m/s.

4) Volumetric Flux at 5000 RPM: Figure below shows volumetric flux of gear pump assembly with time. The volumetric flux remains within the range of $0.001 \frac{m^3}{s}$ to $0.0015 \frac{m^3}{s}$. Rippling is produced due to the tooth of gear pump geometry.

5) Total Pressure contours at 2500 RPM and 7500 RPM: Figure shows total pressure contours at a rotational speed of 2500 RPM and 7500 RPM. The total pressure increases at the center of meshed spur gears and then drops out. The total pressure should remain within a certain limit to avoid gear tooth failure. Total pressure ranges from 2030 Pa to $1.31422 \times 10^6$ Pa for 2500 RPM and ranges from 243 Pa to $2.4655 \times 10^6$ Pa for 7500 RPM.
XII. DISCUSSIONS

Three simulations were conducted in order to calculate the volumetric flow rate of external gear pump at rotational speed of 2500 RPM, 5000 RPM and 7500 RPM. After convergence of iterations, volumetric flow rate was found to be $0.0015\frac{m^3}{s}$ at 5000 RPM whereas that calculated analytically is $0.0019\frac{m^3}{s}$. External Gear pump geometry is revisited in order to lower the total pressure rising in the middle of gear tooth. In our case the circular tooth thickness is reduced to reasonable value. Velocity is found to be constant throughout the domain. Unsteady simulation of external gear pump shows that the total pressure changes with the rotational speed of driving and driven gear while flow rate remain constant throughout the domain.

For rotational speed of 2500RPM the volumetric flow rate was found to be $0.0006\frac{m^3}{s}$ whereas that calculated analytically is $0.00099179\frac{m^3}{s}$. For rotational speed of 7500RPM the volumetric flow rate was found to be $0.0025\frac{m^3}{s}$ whereas that calculated analytically is $0.0029754\frac{m^3}{s}$.

![Figure 9: Total Pressure Contour at 7500 RPM](image)

<table>
<thead>
<tr>
<th>Speed</th>
<th>2500 RPM</th>
<th>5000 RPM</th>
<th>7500 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Numerical</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\left(\frac{m^3}{s}\right)$</td>
<td>0.0006</td>
<td>0.0015</td>
<td>0.0025</td>
</tr>
<tr>
<td>Analytical</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\left(\frac{m^3}{s}\right)$</td>
<td>0.00099179</td>
<td>0.0019</td>
<td>0.0029754</td>
</tr>
</tbody>
</table>

Table 1: Comparison of Analytical and Numerical Solution

XIII. CONCLUSIONS

In the present study, a calculation of external gear pump mechanism is performed using analytical and numerical technique. The main goal is to calculate circular tooth thickness to avoid damage to geometry. First the gears are modeled in Pro/Engineer module according to the given pitch diameter, number of teeth and width of the assembly. The external gear pump geometry is analyzed for overlapping of gear tooth. The geometry is then analyzed in Pro/Engineer Mechanism module [5]. Both the driving and driven gear are simulated at different rotational speed to predict overlapping of gear tooth. MATLAB/SIMULINK [6] module is used to study the behavior of external gear pump mechanism for a particular torque applied. Finally the gear pump geometry is simulated using Simerics/PUMPLINX software [9]. Study is conducted in order to calculate flow rate at different pump rotational speed. Also total pressure between gear pump meshed tooth is analyzed to avoid damage to the external gear pump assembly.

XIV. NOMENCLATURES

- $S_G$: Surface Area of Gear Face
- $N$: Number of teeth on driving and driven gear
- $w$: Thickness of Gear
- $A_{DO}$: Area of Circle with Outer Diameter
- $A_{etr(G)}$: Entrapped Area per Gear
- $V_{etr(G)}$: Entrapped Volume per Gear
- $A_{etr(T)}$: Entrapped Area per Tooth
- $A_t$: Tip Area
- $A_{Inlet}$: Area before Orifice
- $V_D$: Volumetric displacement
- $Q_{ideal}$: Ideal Flow Rate
- $Q_{Predicted}$: Predicted Flow Rate
- $Q_{Leak}$: Leakage Flow Rate
- $D_O$: Outer Diameter of Gear
- $D_R$: Root Diameter of Gear
- $D_B$: Base Diameter
- $D_P$: Pitch Diameter
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ω  Angular Velocity of Gear

ϕp  Gear Pressure Angle


h  Heat Transfer Coefficient  [5]  Pro/Engineer Wildfire 4.0 Help Center, Parametric Technology Corporation, USA.


T  Temperature  [7]  GAMBIT® software for Grid generation and processing

k  Thermal Conductivity  [8]  FLUENT® Software for CFD Analysis


μ  Viscosity  [12]  Autodesk AutoCAD® software for drawing generation


c  Speed of sound  [14]  www.kingston.com, Kingston Technology


Nu  Nusselt Number

R  Real Gas Constant

λ  Second viscosity

ϕ  Viscous Dissipation term

h0  Total enthalpy

i  Internal thermal energy

E  Total energy of the fluid

Main Subscript

1  Driving Gear

2  Driven Gear

In  Inlet

Out  Outlet

Greek Letters

Δ  Difference

XV. REFERENCES


XVI. ANNEX A

Figure 10MATLAB/SIMULINK Gear Pump Simulation