STUDY OF A MINIATURE VANE PUMP

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We hereby recommend that the thesis presented under our supervision by *Mr. Sawan Kumar* entitled *"Study of a Miniature Vane Pump"* be accepted in partial fulfilment of the requirements for the degree of Master of Mechanical Engineering.

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ABSTRACT

In the field of Fluid Power, vane pumps possess the most simplified designs, and are the only ones capable of working at high pressures, besides possessing the best performance (efficiency) of the entire group of existing pumps. The ease of controlling the rate of discharge from a vane pump and maintaining a constant discharge makes it an automatic choice in most of the fluid control systems. The performance of a vane pump depends on the wear and leakage between the various sliding surfaces such as between the vane and slotted rotor. The use of vane pumps in hydraulic system applications has become widespread due to their efficiency advantages; however, this efficiency gain is often accompanied by a degradation of system stability. Vane pumps are often widely used for fuel loading terminals, fuel transport vehicles, an oil hydraulic vane pump for power steering systems, solvents, alcohol and even soft drinks and syrups.

The present study deals with the static and dynamic analysis of a miniature vane pump at different operating pressures. A set of mathematical equations have been presented to represent the flow rate and pressure distribution of vane pump for the design configurations. The vane pump also permits a substantial simplification of the control system with the elimination of complex metering valves, offering a significant reduction in fuel system cost. The dynamic behavior of the vane pump has been analyzed by solving the equations in MATLAB / SIMULINK environment. This program was initiated to develop a technology that embodied the versatility of the variable displacement vane pump. Thick metal vanes slide in the slotted rotor while the simple controlled ring provides the stability.

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NOMENCLATURE

A_d	Opening area of the delivery port	m^2
A_0	Opening area of the orifice	m^2
A_s	Opening area of the suction port	m^2
b	Starting angle of the suction or delivery port	rad
C_d	Co-efficient of discharge of flow	
c	Working angle of the suction or delivery port	rad
d	Width of the port at the end point of suction and starting point of discharge port	m
e	Eccentricity	m
f	Width of the port at the starting point of suction and end point of discharge port	m
L	Total length of the vane pump	
N	Rotational speed	rpm
P_d	Discharge pressure	Pa
p_i	Pressure inside the i^{th} chamber	Pa
P_s	Suction Pressure	Pa
Q_d	Discharge flow	m^3/sec
Q_{disi}	Discharge flow from i^{th} chamber	m^3/sec
Q_i	Kinematic flow of i^{th} chamber	m^3/sec

Q_{suci}	Suction flow to i^{th} chamber	m^3/sec
R	Radius of the casing	m
r	Radius of the rotor	m
r_c	Radius of the centre line of the delivery or suction port	m
V	Volume of the i^{th} pumping chamber	m^3
z	Number of vanes	
α	Angular extent between two vanes	$(^0)$
β	Bulk modulus of working fluid	GPa
μ	Viscosity of working fluid	Pa-sec
ω	Rotational speed	rad/sec
ρ	Density of working fluid	Kg/m^3
θ	Rotation angle	rad

1 INTRODUCTION

1.1 An Introduction to Pump

A pump is a device that moves fluids (liquids or gases), or sometimes slurries, by mechanical action. Pumps can be classified into three major groups according to the method they use to move the fluid: direct lift, displacement, and gravity pumps. Pumps operate by some mechanism (typically reciprocating or rotary), and consume energy to perform mechanical work by moving the fluid. Pumps operate via many energy sources, including manual operation, electricity, engines, or wind power, come in many sizes, from microscopic for use in medical applications to large industrial pumps. Mechanical pumps serve in a wide range of applications such as pumping water from wells, aquarium filtering, pond filtering and aeration, in the car industry for water-cooling and fuel injection, in the energy industry for pumping oil and natural gas or for operating cooling towers. In the medical industry, pumps are used for biochemical processes in developing and manufacturing medicine, and as artificial replacements for body parts, in particular the artificial heart and penile prosthesis.

Hydraulic Pump: An Introduction

Hydraulic pumps are sources of power for many dynamic machines. Hydraulic pumps are capable of pushing large amounts of oil through hydraulic cylinders or hydraulic motors. In this fashion, the pump converts the mechanical energy of the drive (i.e. torque, speed) into hydrostatic energy (i.e. flow, pressure).

Hydraulic pumps operate according to the displacement principle. This involves the existence of mechanically sealed chambers in the pump. Through these chambers, fluid is transported from the inlet (suction port) of the pump to the outlet (pressure port). The sealed chambers ensure that there is no direct connection between the two ports of the pump. As a result, these pumps are very suitable to operate at high system pressures and are ideal for hydraulics.

Different Types of Hydraulic Pumps

Hydraulic pumps are manufactured depending on different functional and hydraulic system requirements, such as operating medium, required range of pressure, type of drive, etc. A large range of design principles and configurations exists behind hydraulic pumps. Consequently, not every pump can fully meet all sets of requirements to an optimum degree. Two different types of hydraulic pumps exist, as shown in Figure 1.1. Here we discussed about the positive displacement pump only.

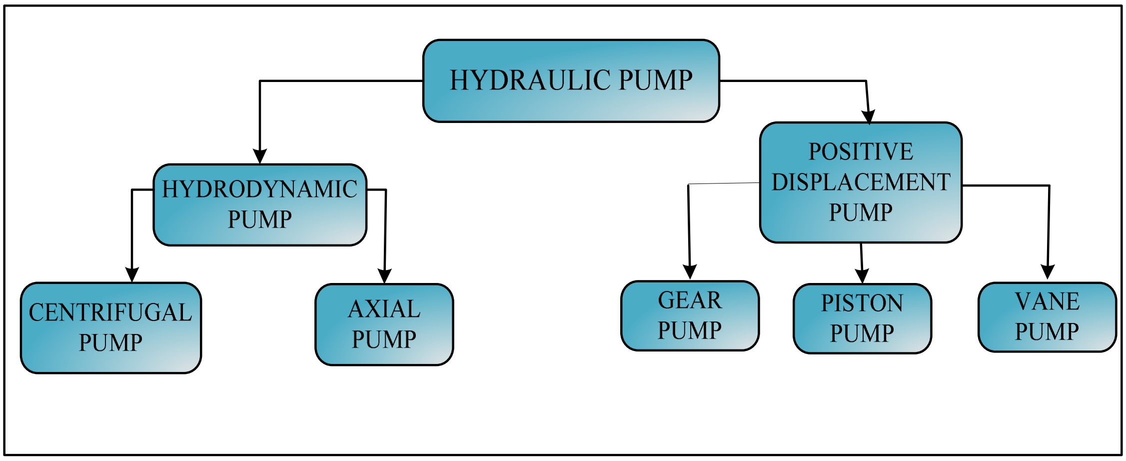

Figure 1.1 Brief classifications of hydraulic pumps.

1.2 Positive displacement pumps

A positive displacement pump has an expanding cavity on the suction side and a decreasing cavity on the discharge side. Liquid flows into the pumps as the cavity on the suction side expands and the liquid flows out of the discharge as the cavity collapses. If the displaced volume is constant for each cycle of operation the pump is known as fixed displacement pump.

A *relief or safety valve* on the discharge side of the Positive displacement pump is absolute necessary. The relief valve can be internal or external the pump. An internal valve should in general only be used as a safety precaution. An external relief valve installed in the discharge line with a return line back to the suction line or supply tank is highly recommended.

Classification of positive displacement pump

Gear pumps

a) External gear pump
b) Internal gear pump
c) Gear ring pump
d) Screw spindle pump

Piston pumps

a) Axial piston pump
b) Radial piston pump

Vane pumps

a) Fixed displacement vane pump
b) Variable displacement vane pump

1.2.1 Gear pumps

A gear pump is used in many hydraulic systems. It has few moving parts, works smoothly, and operates very well at pressures up to 250 bars. The displacement chambers are formed between the housing of the pump and the rotating gear wheel (or gear wheels, depending on model). Different types of gear pump are discussed below.

External gear pump

External gear pumps are used in industrial and mobile (e.g. log splitters, lifts) hydraulic applications. Typical applications are lubrication pumps in machine tools, fluid power transfer units and oil pumps in engines. These pumps have some unique features:

i. Low weight
ii. Relatively high working pressures
iii. Wide range of speeds
iv. Wide temperature and viscosity range (i.e. flexibility)
v. Low cost

In an external gear pump, only one of the gear wheels is connected to the drive. The other gear wheel rotates in the opposite direction so that the teeth of the rotating gear wheels interlock. With use of a bearing block, the gear wheels are positioned in such a way that they interlock with the minimum clearance. Volume is created between the gear tooth profiles, housing walls and surfaces of the bearing blocks. Typical parameters of external vane pump explained as below:

i. Displacement volume: 0.2 to 200 cc
ii. Maximum pressure: up to 300 bar (size dependent)
iii. Speed range: 500 to 6000 rpm

Internal gear pump

Internal gear pumps are primarily used in non-mobile hydraulics (e.g. machines for plastics and machine tools, presses, etc.) and in vehicles that operate in an enclosed space (electric fork-lifts, etc.). The internal gear pump is exceptionally versatile and also capable of handling thick fluids. Key features of internal gear pump explained as below:

i. Low flow pulsation
ii. Low operating noise
iii. High efficiency

In an internal gear pump, the gear rotor is connected to the drive. When the gear rotor and internal gear rotate, volume is created between the gear ring profiles, housing walls and filling piece. The space between the gear tooth profiles increases relatively slowly over an angle of about 120°. This causes operation to be exceptionally quiet with a constant flow. Typical parameters are explained as below:

i. Displacement volume: 0.5 to 500 cc (cubic centimetres)
ii. Maximum pressure: up to 300 bar (dependent on nominal size)
iii. Speed range: 500 to 3,000 rpm (dependent on nominal size)

Gear ring pump

The gear ring pump is primarily used as a pressure lubrication system for machines and combustion engines. They are also used in hydraulic power steering systems. This pump is often assembled with a high pressure pump, e.g. radial piston pump. The rotors of the gear ring pump can be directly built into the housing of the high pressure pump, which makes it possible to build very compact units. Such small double-pumps are often used for rapid traverse on large presses and tensioning equipment. The rotor has one tooth less than the inner stator. Planetary movement of the rotor results in compressing and decompressing of the displacement chambers within the housing.

Screw spindle pump

Similar to internal gear pumps, screw pumps possess an extremely low operating noise level. They are therefore used in hydraulic systems in such places as theatres and opera houses. The displacement volume of the screw spindle pump is the largest of all gear pumps. Screw pumps contain 2 or 3 worm gears within the housing and therefore also referred to as worm gear pumps. Typical parameters of screw spindle pump are explained below:

i. Displacement vol.: 15 to 3,500 cc
ii. Maximum pressure: up to 200 bar (dependent on nominal size)
iii. Speed range: 500 to 3,500 rpm (dependent on nominal size)

The worm gear that is connected to the drive has a clockwise thread. Rotary movement is transmitted to further worm gears, which have counter-clockwise threads. The displacement chamber is formed between the threads and the housing of the screw pump.

1.2.2 Piston Pumps

Hydraulic piston pumps can handle large flows at high hydraulic system pressures. Typical applications are mobile and construction equipment, marine auxiliary power, metal forming and stamping, machine tools and oil field equipment. In these pumps, the pistons accurately slide back and forth inside the cylinders that are part of the hydraulic pump. The sealing properties of the pistons are excellent. Key features of piston pumps are mentioned below:

i. Compact size
ii. High power density
iii. Optimum efficiency and reliability
iv. High speed and torque
v. High operating pressures

Piston pumps operate at very high volumetric efficiency levels due to low fluid leakage. The plungers may consist of valves at the suction and pressure ports or with input and output channels. Piston pumps with valves at the ports are better suited to operate at higher system pressures due to better sealing characteristics.

Axial piston pump

The design of an axial piston unit is based on two important principles. First, the design of the axial piston pump may be based on the swash plate principle or bent axis design. Secondly, hydraulic system parameters have to be taken into account. Whether the usage is to take place in an open or closed loop circuit is of great importance. In closed loop circuits, the return line (i.e. the suction line of the pump) is under pressure. This must be incorporated in the design of axial piston units used in closed loop systems. It is also imperative to have a variable displacement volume hydraulic pump in operation in these systems. In fixed displacement volume configuration, the axial piston unit can be used both as pump and as motor.

In bent axis design, the displacement volume is dependent on the swivel angle: the pistons move within the cylinder bores when the shaft rotates. In swash plate design, the rotating pistons are supported by a swash plate; the angle of the swash plate determines the piston stroke. Typical parameters of axial pump are explained below:

i. Displacement volume: 5 to 1,000 cc
ii. Maximum pressure: up to 450 bar
iii. Speed range: 1,500 to 11,000 rpm

Radial piston pump

Radial piston pumps are used in applications that involve high pressures (operating pressures above 400 bar and up to 700 bar), such as presses, machines for processing plastic and machine tools that clamp hydraulics. Radial piston pumps are the only pumps capable of working satisfactorily at such high pressures, even under continuous operation.

Radial piston pumps are available in two different configurations. With eccentric cylinder block, the piston rotates within the rigid external ring. Eccentricity determines the stroke of the pistons or with an eccentric shaft, the rotating eccentric shaft causes radially oscillating piston movements to be produced. Most models have an odd number of pistons to reduce the flow pulsation. Typical parameters of radial piston pumps are explained as below:

i. Displacement volume: 0.5 to 100 cc
ii. Maximum pressure: up to 700 bar (size dependent)
iii. Range of speeds: 1,000 to 3,000 rpm (size dependent)

1.2.3 Vane pumps

The vane pump finds its use in die casting and injection moulding machines in industry, as well as in land and road construction machinery. Vane pumps operate with much lower flow pulsation, i.e. constant flow. As such, vane pumps produce less noise while maintaining a relatively high speed. The operating pressure of vane pumps does not normally exceed 175 bar. However, in specially designed vane pumps the operating pressure may go over 200 bar and up to 300 bar. Vane pumps are available as single chamber vane pumps or double chamber vane pumps. Both types use the same parts, i.e. they comprise a rotor and vanes. The vanes may be radially moved within the rotor, and the centrifugal force of the rotor pushes the vanes out to touch the housing. The difference between the two types is in the shape of the stroke ring that limits the stroke movement of the vanes. Key features of the vane pump are mentioned below:

i. Low flow pulsation
ii. Very low noise levels
iii. Wide range of speeds
iv. Wide viscosity range

Typical parameters of vane pump are as explained here:

i. Displacement volume: 6 to 640 cc
ii. Maximum pressure: up to 200 bar
iii. Speed range: 500 to 3,000 rpm

The operation of the vane pump is based on , the rotor which contain radial slots rotate by a shaft and rotate in cam ring (housing), each slot contain a vane design as to comes out from the slot as the rotor turns. During one half of the rotation the oil inters between the vane and the housing then this area starts to decrease in the second half which permit the pressure to be produced , then the oil comes out pressurizes to the output port.

Typical Applications for Vane Pumps

Vane pumps can be used in many different positive displacement applications. They can handle thin and low viscosity liquids, like water and petrol. They don't work particularly well with highly viscous fluids, as the higher viscosity prevents the vanes from moving freely in the slots. Due to the fact that they can handle a wide range of viscosities, Vane pumps are often widely used for fuel loading terminals, fuel transport vehicles, solvents, alcohol and even soft drinks and syrups.

Vane pumps are available in many different configurations and can also handle fluids with a wide range of temperatures and pressures. Since they often are used for pumping clean hydrocarbons including gas and light oils, vane pumps are normally constructed with ductile iron casings and metal rotors. The vanes are often made of carbon, which exhibits a good lubricity to keep the vanes sliding inside the slots and against the inside surface of the casing.

The type of fluid being handled plays a large part in whether this pump is right for a specific application. High-viscosity or thicker fluids will obviously greatly reduce the speed of the pump and a therefore vane pump might not be the best choice. In some cases *vane pumps* present a good alternative to gear pumps for pumping relatively low viscosity oils.

Applications of vane pump

a) Aerosol and Propellants

b) Aviation Service - Fuel Transfer, De-icing
c) Auto Industry - Fuels, Lubes, Refrigeration Coolants
d) Bulk Transfer of LPG and NH_3
e) LPG Cylinder Filling

Materials of construction options for vane pump

a) Externals (head, casing) - Cast iron, ductile iron, steel, and stainless steel.
b) Vane, Pushrods - Carbon graphite.
c) End Plates - Carbon graphite
d) Shaft Seal - Component mechanical seals, industry-standard cartridge mechanical seals, and magnetically-driven pumps.

Vane pumps generally work within the following ranges:

a) Flow rate ranges between 20 to 9500 lpm (litre per minute)
b) Total head (pressure) ranges between 1 to 14 Bar
c) Horsepower ranges between 1 to 300 hp

There are two types of vane pumps, they are explained below:

a) Fixed Displacement vane pump
b) Variable Displacement vane pump

Fixed Displacement vane pump

In this type of pump the eccentricity between pump cam-ring and rotor is fixed and pump discharge always remain same at a particular pressure. The Denison Vane Technology by Parker Hannifin [1] provides us with the best solution in the market of fixed displacement balanced vane pumps. The outflow of the pump may be controlled and adjusted mechanically (i.e. directly with an adjustment screw on the pump). You can also control the outflow by other means, such as a combination of hydraulic and electrical control.

Fixed Displacement vane pump characteristics

a) Typical pressures to 280 bar
b) Fixed displacement only
c) Provides prime mover soft-start
d) Simple double assemblies
e) Low noise
f) Good serviceability

There are two types of fixed displacement vane pump

i. Unbalanced Vane Pump

ii. Balanced Vane Pump

Unbalanced vane pump

In the unbalanced vane pump a cam ring shape is a true circle that is on a different centreline from a rotor. Pump displacement depends on how far a rotor and ring are eccentric. The advantage of a true-circle ring is that control can be applied to vary the eccentricity and thus vary the displacement. A disadvantage is that an unbalanced pressure at the outlet is effective against a small area of the rotor's edge, imposing side loads on the shaft. Thus there is a limit on a pump's size unless very large hearings and heavy supports are used. The schematic diagram is shown in Figure 1.2.

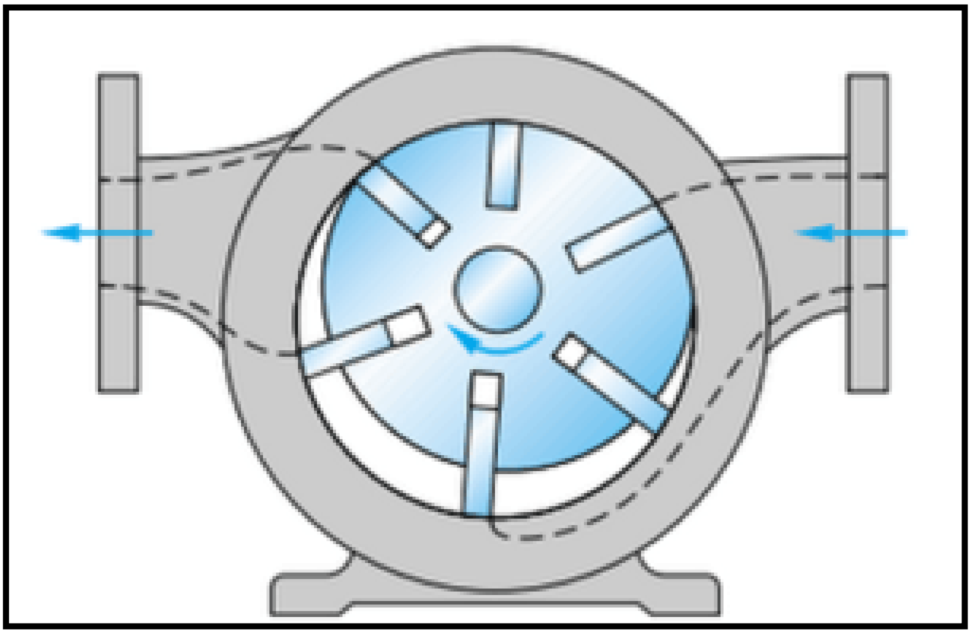

Figure 1.2 Unbalanced vane pumps.

As the rotor rotates and fluid enters the pump, centrifugal force, hydraulic pressure, and/or pushrods push the vanes to the walls of the housing. The tight seal among the vanes, rotor, cam, and side plate is the key to the good suction characteristics common to the vane pumping principle. The housing and cam force fluid into the pumping chamber through holes in the cam. Fluid enters the pockets created by the vanes, rotor, cam, and side plate. As the rotor continues around, the vanes sweep the fluid to the opposite side of the crescent where it is squeezed through discharge holes of the cam as the vane approaches the point of the crescent. Fluid then exits the discharge port.

Advantages of Unbalanced vane pump

a) Handles thin liquids at relatively higher pressures
b) Compensates for wear through vane extension
c) Can run dry for short periods
d) Can have one seal or stuffing box
e) Develops good vacuum

Disadvantages of Unbalanced vane pump

a) Complex housing and many parts
b) Not suitable for high pressures
c) Not suitable for high viscosity

Balanced vane pump

A balanced vane pump is one that has two intake and two outlet ports diametrically opposite each other. Pressure ports are opposite each other and a complete hydraulic balance is achieved. One disadvantage of the balanced vane pump is that it cannot be designed as a variable displacement unit. It has elliptical housing which formed two separate pumping chambers on opposite side of the rotor. This kind of pump gives higher operating pressure**.**

Advantages of balanced vane pump over unbalanced vane pump

a) it has bigger flow
b) it has bigger pressure
c) its life is bigger
d) constant volume displacement

Variable Displacement Vane Pump

In variable displacement the discharge of pump can be changed by varying the eccentricity between rotor and pump cam-ring. As eccentricity increases pump discharge increases. With decrease in eccentricity discharge decreases and oil flow completely stop when rotor becomes concentric to pump cam ring.

Only the single chamber vane pump is available as a variable displacement volume type. By moving the stator (stroke ring) with an adjustment screw, it is possible to adapt the size of the displacement chambers. When the axis of the rotor is in the centre position of the stroke ring, all formed chambers are of equal size and the outflow of the pump is nil. Three positioning devices may be used:

a) *Horizontal adjustment screw for the stroke volume*: displacement volume is directly determined by the distance between the stroke ring and the rotor of the pump.
b) *Height adjustment screw*: this changes the vertical position of the stroke ring which directly affects dynamic response of the pump and noise.
c) *Setting screw for maximum operating pressure*: it sets the amount of the spring pre-tensioning that determines the maximum operating pressure.

Advantage of variable displacement vane pump

a) Typical displacements to 100 cm^3/rpm
b) Typical pressures to 160 bar
c) Simple multiple assemblies
d) Range of pump controls
e) Low noise
f) Low cost.

Vane pump can also classify as mentioned below

Single Chamber Vane Pump

In a single chamber vane pump, the stroke movement of the vanes is limited by a ring with a circular internal track. The position of this so-called stroke ring is off-centre with respect to the rotor, resulting in change of volume in the displacement chambers. The displacement chambers are created by the rotor, two vanes, the internal surface of the ring and the control discs on one side.

In a single chamber vane pump, the system pressure is only on one side of the rotor. This causes a significant load on the bearings. To reduce this load, the forces acting on the rotor must be in balance. This is the reason why double chamber vane pumps were designed.

Double Chamber Vane Pump

For double chamber vane pumps, the process of filling the chambers (suction) and emptying is in principle the same as for single chamber vane pumps. In this case, however, the stroke ring (i.e. stator) has a double eccentric internal surface. The rotor can be placed in the axis of the stator because of these surfaces, which differentiates them from single chamber vane pumps.

This set up causes each vane to carry out two strokes per rotation of the shaft. All radial loads on the rotor are now neutralized (two pressure ports on each opposite side). The end result is that two pumps have been built together as one. Due to the twin cam forms of the stator, two displacement processes occur per revolution.

1.3 Development Stages in the Conceptual Design of an Electro Hydrostatic Actuator for Robotics:

As a scientific area, robotics and its application have achieved a very high level of development and are present in almost all human activities. The intention of mankind, to assign precise, heavy and dangerous routine and non-routine jobs to V. Karanović et al. [2] has prompted the development of science and engineering, robotics included. Application of robotics has allowed accurate, precise and highly efficient performance of such work tasks, with or without human interference with the aim of performing the designated tasks with utmost efficiency.

Robotic applications should have following characteristics

a) *Functional design* pertains to design solutions which allow simple robot control for the most efficient realization of tasks (functions).
b) *Optimized control units* should provide reliable, adjustable and efficient control of a robot in any situation.
c) *Power a drive unit* using one or a combination of energy sources (electric, pneumatic, and hydraulic), is tailored to match the given task and operating conditions.

In order to achieve maximum working performance and avoid drawbacks of classical drive units, various authors have contributed to research and development of combined robotic drive units. One such example is the development of the combined electro-hydrostatic drive (EHA – Electro Hydraulic Actuator). As a self-contained power device the Electro Hydraulic Actuator (EHA), was first developed and patented in 1950s to actuate landing gears [3] , wing flaps or other movable parts of an aircraft. Generally, the EHA device was designed to suit applications in which smaller size and weight-to-power ratio are required (for example hand and leg prosthesis, exoskeletons, etc.). Considering the advantages and drawbacks of EHA, scientists and engineers have subsequently investigated various possibilities to adapt EHA for different applications.

1.3.1 Electro hydraulic actuator for force assistive wearable robot

The most common actuators used in robotics are electromagnetic, hydraulic or pneumatic, i.e. they convert electric energy or the energy of a pressurized fluid into mechanical energy. While in industrial robotics [2] hydraulic and pneumatic actuators are widespread, in mobile robots most actuators are of the electric type, due to a number of reasons. Firstly, energy is mostly stored on board of moving Robots as electric energy, and the efficiency of using some sort of electric compressor or pump to operate pneumatic or hydraulic actuators is generally low. Other reasons are the more compact layout of the system and the fact that it is easier to control electric actuators directly from the main controller of the robot than to control actuators of different type. The case of electro-hydrostatic transmissions is an exception, but in this case the hydraulic transmission is not controlled, and all control action is performed by the electrical part of the system. In case of space robots the advantages of electric actuators over fluid based devices are even larger, owing to the difficulties of dealing with fluids in the harsh environment these devices must withstand.

In particular, pneumatic actuators are well suited where there is a practically unlimited supply of working fluid, but not where the working fluid must be carried on board and recycled indefinitely. Nevertheless, in some applications the higher compactness and lightness of hydraulic motors may make hydraulic or better electro hydraulic systems, very attractive. A number of other devices that can be used to actuate an active system must be added to these ‘conventional’ actuators. They are often called solid state or smart materials actuators, since they rely on materials that are able to change their properties under the effect of electric or magnetic fields or changes of temperature. Piezoelectric, magnetostrictive, electrotrictive and shape memory actuators these are the example of smart material actuators.

In order to advance robot interaction with humans and environment, Kargov et al.designed a robotic hand prosthesis with hydraulic drive [4]. Although based on the use of a centralized hydraulic system, their investigation emphasizes the need to miniaturize hydraulic components and points out their efficiency (Figure 1.3).

The miniature components were made, enabling the hydraulic system consisting of the pump, reservoir, control valves, electric motor drive and control unit to be built into the natural-size robotic hand prosthesis. Moreover, its compact size and functionality allowed maximum performance.

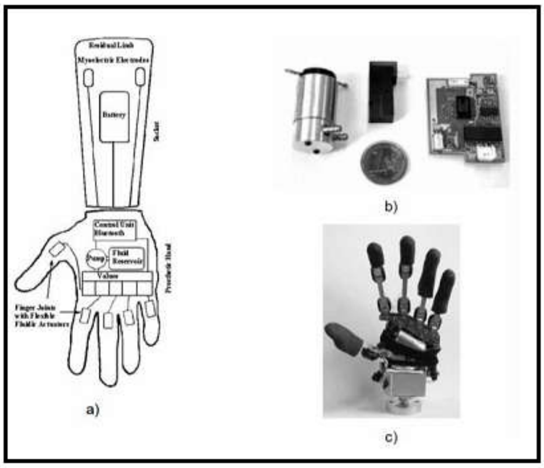

Figure 1.3 a) Schematic view of Kargov prosthetic hand b) Size comparison of hydraulic pump (left) micro valve (middle) and microcontroller (right) with a coin c) Look of prosthetic hand [5]

Parallel to Kargov's work, Kaminaga et al. experimented with the application of electro hydrostatic actuator on knee-joints and a robotic hand for a humanoid robot [6] as shown in Figure 1.4.

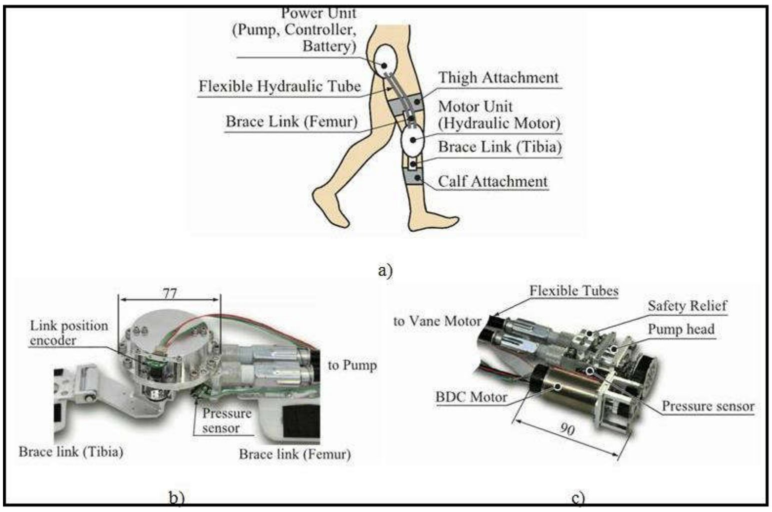

Figure 1.4 a) Knee power assist with EHA b) Hydraulic motor c) Hydraulic pump unit [7]

The goal of this investigation was to achieve certain degree of adaptability of the mechanisms which generate robot movements, in situations when the robot interacts with humans or objects within its environment. In order to achieve the desired degree of adaptability, which allow the robot to simulate human-like reactions, when it faced an obstacle. The built-in mechanisms generate human movements, with a convenient degree of back-drivability. Failing this requirement is liable to cause errors such as the displacement or breakage of the encountered object (obstacle), damage to the robot, positioning error, etc. The very ability of back-drivability gives the advantage to hydrostatic systems compared to their electric counterparts, since they are able to control the direction, sense and speed of movement under significant payloads.

1.4 Literature Review:

E.Mucchi [8] et al showed vane pumps exhibit pressure ripple in the pressure evolution trend during a complete shaft rotation. Pressure ripple can determine oscillating forces within the system leading to vibration and noise generation. He was focused on the experimental measurement of the pressure evolution in vane pumps by using two different methodologies. In the first method a pressure transducer was directly facing the volume between two vanes, in the second method the sensor was located inside an external chamber where the oil is transferred via a duct suitably designed in the rotor shaft. Briefly, the first method gave better results in terms of pressure evolution but involves some practical problems in the setup

Ihn Sung Cho and Jae Youn Jung [9] showed in the power steering system of an automobile, the balanced and fixed displacement type vane pump is principally applied to the main power source. It was driven by a pulley that was attached to the shaft of the vehicle's engine, and the pulley ratio was about 1:1. The vane pump continuously discharged an outlet flow rate proportional to the rotating speed of the engine. In a vane pump without the flow control valve, an increase in the discharge flow rate meant a linear increase in the actuator speed. At high speed when the driver rotated the steering wheel by even a small amount, there is a high risk of an accident due to the quick rotation of the tire joined to the actuator and links. He designed a flow control valve of the fixed displacement and pressure balanced type vane pump for power steering is designed and simulated using AMESim software. Its theoretical flow characteristics with respect to the rotating speed of the engine are compared and validated with experimental data. The result showed that the design parameters, such as the flow area, the under lap length, the spool clearance, the main spring stiffness and the spring initial force, largely affect the flow control characteristics.

Thn Sung Cho [10] studied in an oil hydraulic vane pump for power steering system; the force acting on the vane generally consists of oil hydraulic, inertial and viscous forces. The vane should not separate from the cam ring inner race during operation. Separation of the vane could occur during the initiation and delivery of the vane pump because the oil hydraulic force acting on the vane is zero and in equilibrium, respectively. If the vane was separated from the cam ring inner race, the delivery of the oil hydraulic vane pump could become unstable or volume efficiency could decrease rapidly. Therefore, the state of the contact between the vane and the cam ring was considered in this study. Results demonstrated that the rotating speed of the shaft, oil temperature, clearance between the vane and the rotor and mass of the vane significantly affect the contact state of the vane.

Jiang et al [11] presented a Computational Fluid Dynamic (CFD) model for the oil pump simulations aimed at better understanding the flow characteristics for improving their designs and reducing product development cycles. Several advanced numerical technologies have been developed to handle the complex geometries of oil pumps and the moving interfaces between the rotating and stationary parts. Two basic oil pump configurations, a vane oil pump and a gerotor oil pump, had been studied with the present method. The numerical results are compared with the existing experimental data.

D. Danardono [12] developed a roller vane type liquefied petroleum gas (LPG) pump for a liquid phase LPG injection (LPLi) engine. Most of the LPG pumps used in the current LPLi engines were installed inside of the LPG tank, but this pump was intended to be installed outside of the LPG tank to overcome the difficulty of fixing an in-tank pump. Because LPG had a low boiling point and high vapour pressure, it usually caused cavitations in the pump and consequently deteriorates the flow rate of the pump. The purpose of this work was to optimize the design of the roller vane pump in order to suppress cavitations and increase the fuel flow rate by using a computational fluid dynamics (CFD) analysis. Computation was performed for six different models to obtain the optimized design of the roller vane pump at a constant speed of 2600 rpm and a constant pressure difference between the inlet and outlet of 5 bar. The computation results showed that an increased intake port cross-section area can suppress cavitations, and the pump could achieve a higher flow rate when the rotor configuration was changed to increase its chamber volume. When the inlet pressure difference was 0.1 bar higher than the fluid saturation pressure, the pump reaches its maximum flow rate.

Tejinder Singh showed cavitation [13] can occur in positive displacement pumps at high operating speeds. High levels of noise and pitting damage to the pump components are the concerns. A finite difference fluid flow analysis of the suction port of a vane pump was presented as a powerful tool to optimize the pump design for eliminating the cavitations phenomenon. The test results of the original cavitating pump and the optimized design were discussed. A new design of the pump rotor was also introduced to fully utilize the potential of the optimized suction port. The approach was generic in nature and lends itself to other pump designs as well.

Manco, Armenio et. al [14] developed a model of a translational type VDOP(variable displacement oil pump) by evaluating geometric, kinematic and fluid dynamic properties of pump and fluid. They developed a kinematic analysis by evaluating vane position from rotor center to the vane's end point which is in contact with control ring. This kinematic calculation is integrated over a chamber to calculate the dynamically changing trapped volume between two adjacent vanes.

Modelling of variable displacement vane pumps are in interests of researchers and engineers due to the aforementioned certain outcomes on fuel economy. Karmel [15] generated a dynamical model of pivot type variable displacement vane pump with a regulator and resistive load. To apply stability criterion for the pressure regulation circuit the generated model had linearized. In his model by considering the motion of valve two operating modes are defined which are LR mode (regulation mode from Line to Regulation chamber) and RE mode (regulation mode from Regulation chamber to Exhaust). In LR mode the flow is regulated by transferring the fluid from main pressure line to regulation chamber however in RE mode the flow is regulated by transferring the fluid from regulation chamber to exhaust.

Wang et. al [16] developed a numerical model of a pivot type Variable displacement oil pump. Inputs of the vane pump kinetic motions have been derived from the CFD tool for each time step. At every new position of the control ring CFD model is re-meshed and oil flow rate results are numerically derived and a good alignment had been achieved with the experimental results at different engine speeds and different back pressures.

1.5 Scope of work and orientation of the thesis

A typical unbalanced vane pump of miniature size has been conceived for the present study. The size has been adopted from a similar type of pump of different category. The main objective of the thesis is to develop a mathematical model of the pump followed by simulation analysis under different operating condition. Using the simulation model, few parametric variations have been aimed to study the characteristics of the pump. The physical description of the pump along through its solid modelling and the pump testing system has been described in Chapter2. In Chapter3, the development of the mathematical model has been elaborated. Chapter4 details the performance study of the pump in MATLAB/SIMULINK environment along with the results of the parametric study. The summary and future scope of work has been detailed in Chapter5.

2 PHYSICAL DESCRIPTION OF THE VANE PUMP

2.1 Working Principle of the Vane Pump

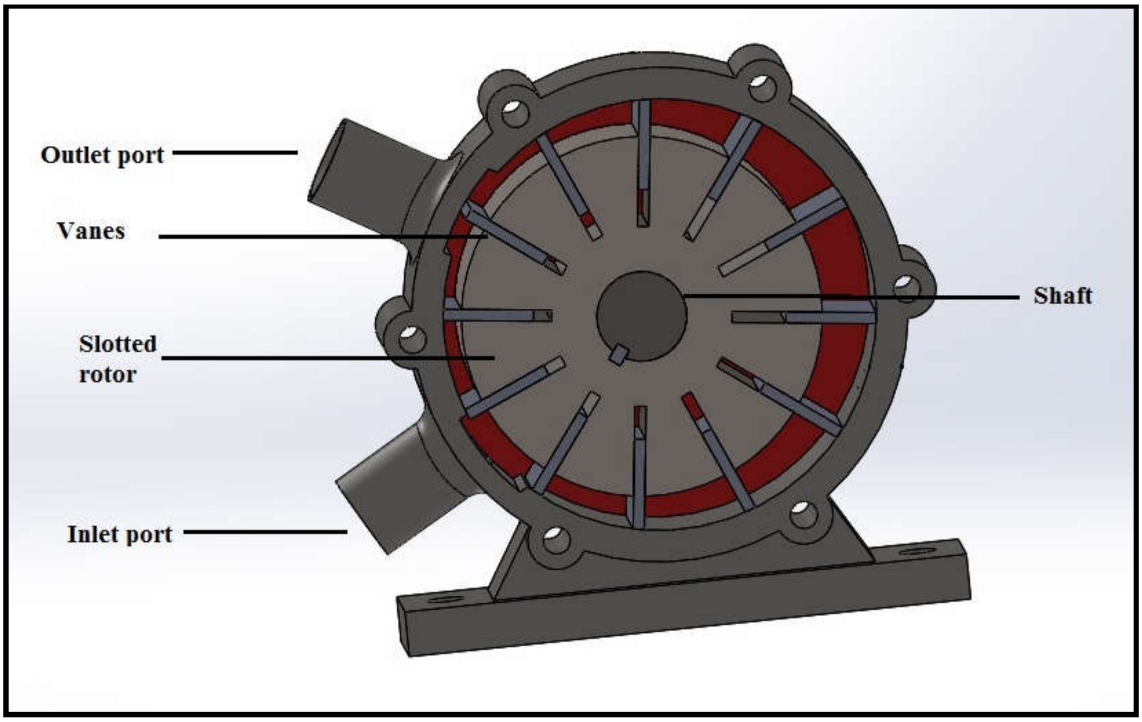

Figure 2.1 The assembled view of the vane pump (without the end cover)

Figure 2.1 shows a solid model of the vane pump (in solidworks). The main components in a vane pump are the slotted rotor, vanes, control ring and the end plate containing the delivery and suction manifolds. The vanes are inserted in the slots of the rotor which is coupled with a drive shaft.

The schematic of vane pump working principle is shown in Figure 2.2, in which the four main components of vane pump and the angular position of vane chamber are defined. Point A in Figure 2.2 is the centre of the pump housing as well as the control ring and point B is the centre of the rotor. There is an eccentricity e between these two centres. The driving rotor has 12 numbers of radial slots as shown. The vanes, placed inside these slots, can slide in radial direction up to the inner surface of the control ring. The enclosed volume due to two consecutive vanes, rotor and control ring surface and the end covers constitute a pumping chamber.

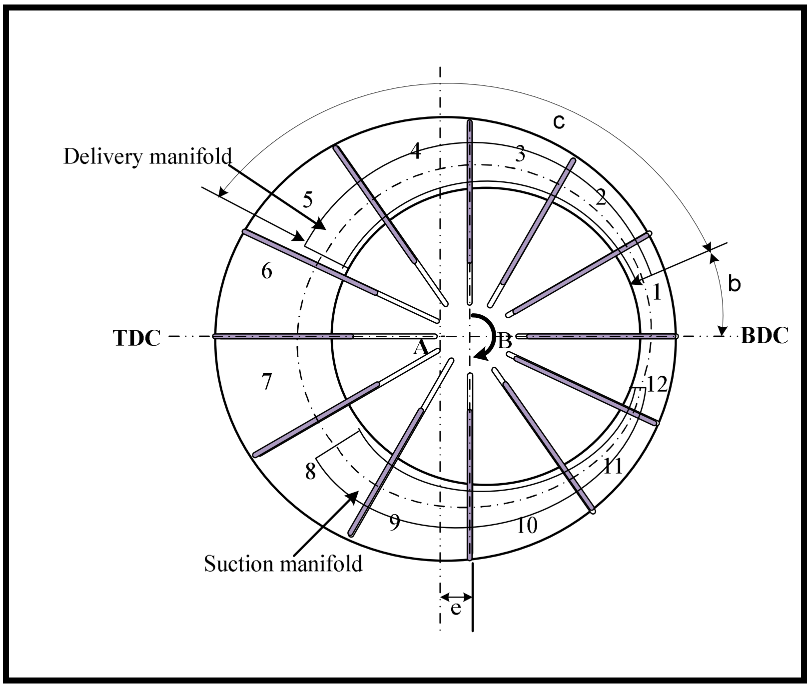

Figure 2.2 Schematic diagram of the vane pump

The rotor is rotated by the drive shaft by means of a constant speed electric motor. Due to its eccentricity with respect to the pump housing the length of the vanes from the centre of the rotor does not remain same at different position of the vane. Due to centrifugal force, the vanes come out radially through the slots and remain in contact with the control ring surface. As the rotor rotates anti-clockwise the pumping chamber volume at the bottom dead centre (BDC) gradually increases creating a vacuum and the fluid is sucked through the suction manifold located at one of the end covers shown in Figure 2.2. The expansion of the volume continues till it reaches the top dead centre (TDC). The fluid trapped in this volume is then pressurised since the volume between two consecutive vanes starts decreasing as it passes the TDC. This causes the oil to exit through the delivery manifold towards the delivery port till the chamber reaches BDC. The same operation continues for the consecutive pumping chambers formed by the next pair of vanes. The discharge of the pump is the algebraic sum of flow rates of all pumping chambers. This is the main working principle of vane pump.

The capacity of the pump depends upon the eccentricity, expansion of vanes, and width of vanes and speed of the rotor. In case of fixed displacement pump the eccentricity remains constant.

2.2 Component Details of the Vane Pump

The vane pumping system consists of

1. Housing
2. Slotted rotor
3. Vanes
4. Seal
5. Controlled ring
6. suction and delivery manifolds

2.2.1 Housing

The casing of a vane pump provides a pressure boundary for the pump and contains channels to properly direct the suction and discharge flow. The slotted rotor is fitted inside the casing.

2.2.2 Slotted rotor

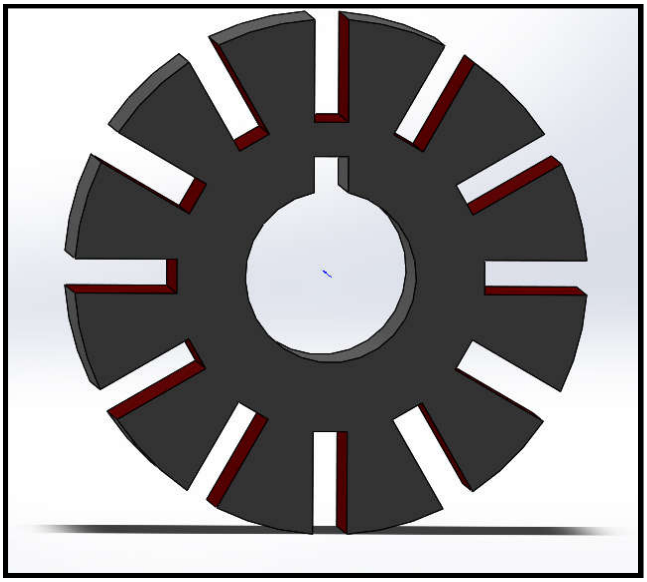

Figure 2.3 Front view of slotted rotor

The rotor is the main rotating part that provides the transfer of power from the mechanical drive to the fluid resulting in the increase in pressure. The rotor rotates inside the cam ring. Each radial slot contains a vane, which is free to slide in or out of the slots due to centrifugal force. This is shown in Figure 2.3.

2.2.3 Vanes

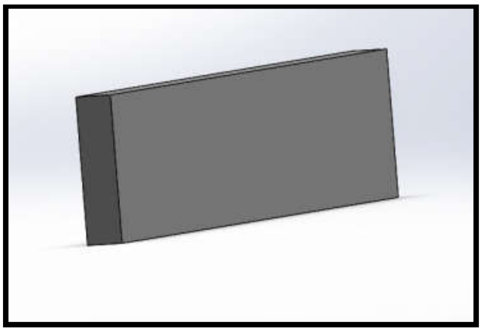

Figure 2.4 side view of vane

Vane is a broad blade type structure attached to a rotating axis or wheel which pushes or is pushed by fluid. A number of vanes that is free to slide into or out of slots in the pump rotor. When the pump driver turns the rotor, centrifugal force, push rods, and/or pressurized fluid causes the vanes to move outward in their slots and bear against the inner bore of the pump casing forming pumping chambers. This is shown in Figure 2.4.

2.2.4 Seal

Seal Chamber refers to a chamber, either integral with or separate from the pump case housing that forms the region between the shaft and casing where sealing components are installed. When the sealing is achieved by means of a mechanical seal, the chamber is commonly referred to as a Seal Chamber. When the sealing is achieved by means of packing, the chamber is referred to as a *Stuffing Box*. Both the Seal Chamber and the Stuffing Box have the primary function of protecting the pump against leakage at the point where the shaft passes out through the pump pressure casing.

2.2.5 Controlled ring

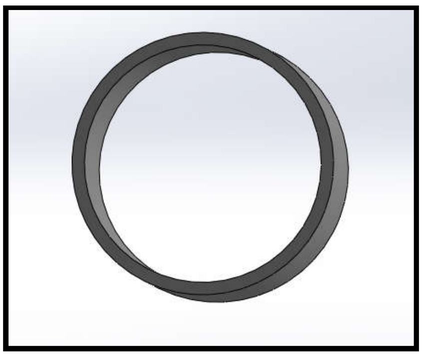

Figure 2.5 Front view of controlled ring

The controlled ring which is fixed in the housing or cover against rotation surrounds the rotor together with the lateral pressure/control plate or cover, respectively. The shape of the controlled ring means that one suction zone and one pressure zone always lie opposite each other. This is shown in Figure 2.5.

2.2.6 Suction and delivery manifolds

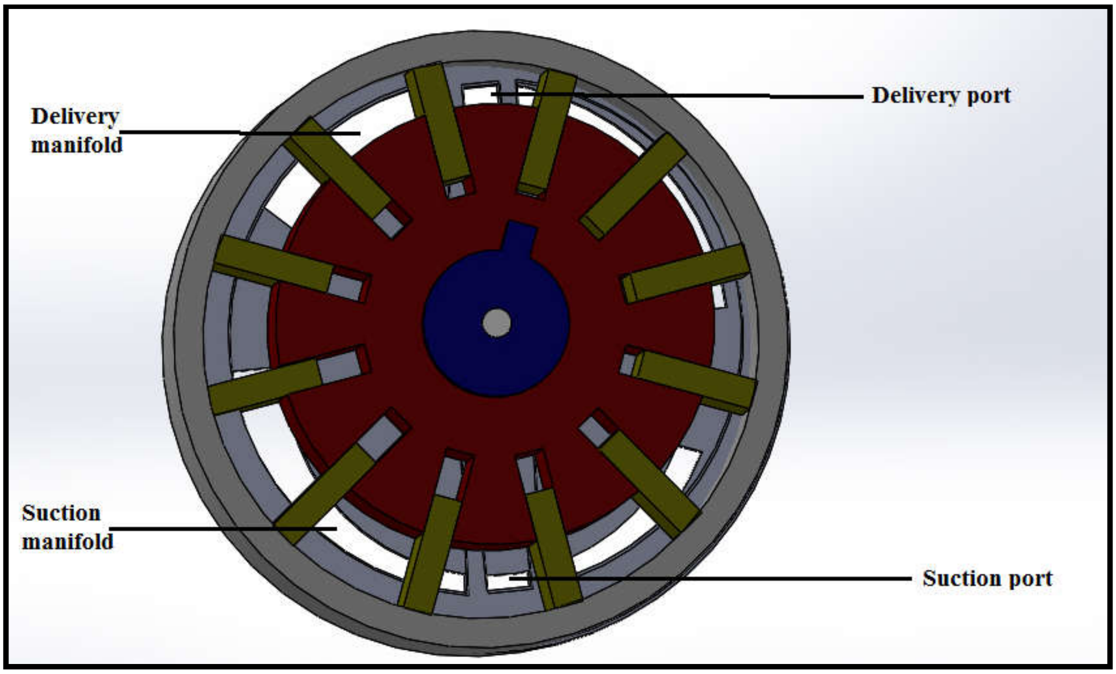

Figure 2.6 Delivery and suction manifold of vane pump

The suction manifold of the pump is designed to preserve fluid energy that will ensure complete filling of the cylinder in extreme pumping conditions. The suction manifold utilizes a chamber design positioned immediately below the suction valves, eliminating all connecting ducts. Fluid is expanded in suction manifold and when it reaches at delivery manifold the fluid is squeezes by vanes. Due to that we get high pressurises fluid at the delivery end. Both manifold portions having a contoured interior face is set to smooth flow of fluid. This is shown in Figure 2.6.

Pressure Relief valve (PRV)

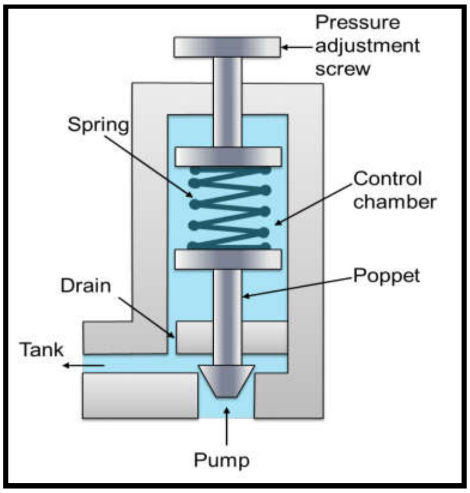

Figure 2.7 Schematic diagram of pressure relief valve

The pressure relief valves are used to protect the hydraulic components from excessive pressure. This is one of the most important components of a hydraulic system and is essentially required for safe operation of the system. Its primary function is to limit the system pressure within a specified range. It is normally a closed type and it opens when the pressure exceeds a specified maximum value by diverting pump flow back to the tank. The simplest type valve contains a poppet held in a seat against the spring force as shown in Figure 2.7. The fluid enters from the opposite side of the poppet. When the system pressure exceeds the preset value, the poppet lifts and the fluid is escaped through the orifice to the storage tank directly. It reduces the system pressure and as the pressure reduces to the set limit again the valve closes. This valve does not provide a flat cut-off pressure limit with flow rate because the spring must be deflected more when the flow rate is higher.

Flow control orifice (Needle valve)

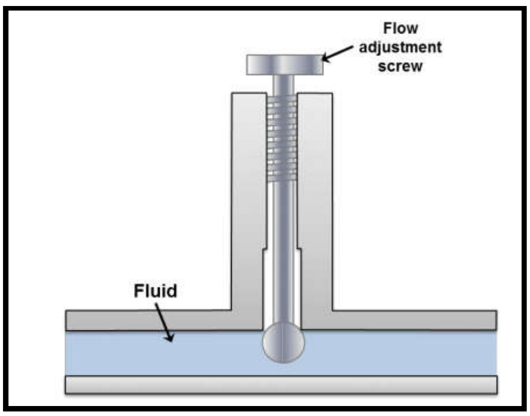

Figure 2.8 Flow control orifice (needle valve) of the vane pump

Flow Control Valve In practice, the speed of actuator is very important in terms of the desired output and needs to be controlled. The speed of actuator can be controlled by regulating the fluid flow. A flow control valve can regulate the flow or pressure of the fluid. The fluid flow is controlled by varying area of the valve opening through which fluid passes. The fluid flow can be decreased by reducing the area of the valve opening and it can be increased by increasing the area of the valve opening. A very common example to the fluid flow control valve is the household tap. Figure 2.8 shows the schematic diagram of a flow control valve. The pressure adjustment screw varies the fluid flow area in the pipe to control the discharge rate.

2.3 Description of the system under circuit diagram

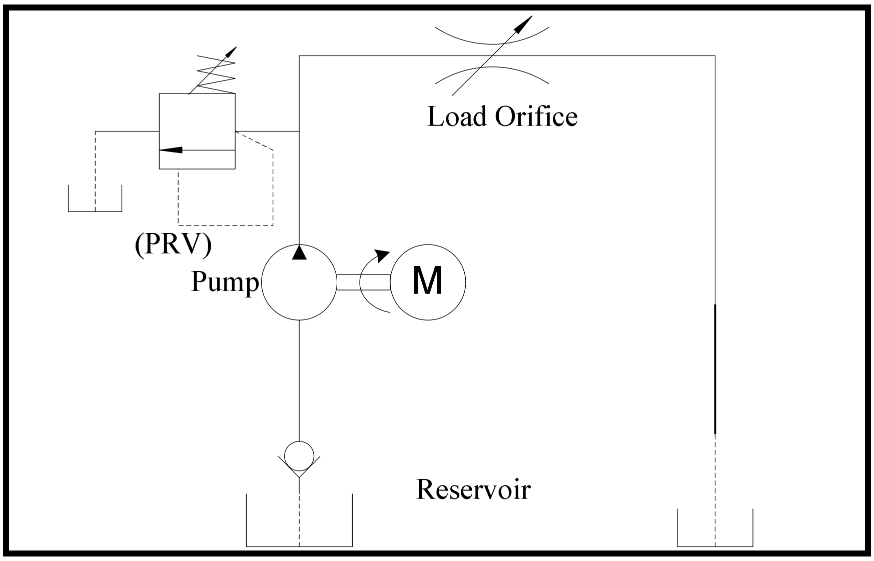

Figure 2.9 Circuit diagram for performance analysis of the pump

(PRV = Pressure relief valve)

Circuit diagram for performance analysis of the pump is as shown in Figure 2.9. The pump volumetric flow rate has been controlled by varying the area of flow control orifice (load orifice i.e. needle valve).

3 SIMULATION MODELING OF THE SYSTEM

Objectives

This chapter mainly focus on the simulation strategy of the modelling of vane pump which is explained in details in this chapter. The flow and pressure developed in the system is predicted at a certain speed if the other parameters such as viscosity, temperature, circulating fluid etc. is kept constant. A model is developed for performance analysis of simulation results of vane pump in the MATLAB/simulink environment. Outputs of the sub-systems have been determined and discussed in details in chapter 4.

3.1 Introduction of Simulink

Simulink is a software package for modelling, simulating, and analyzing dynamical systems. It supports linear and nonlinear systems, modelled in continuous time, sampled time, or a hybrid of the two. Systems can also be multi rate, i.e., have different parts that are sampled or updated at different rates. For modelling, simulink provides a graphical user interface (GUI) for building models as block diagrams. Simulink includes a comprehensive block library of sinks, sources, linear and nonlinear components, and connectors. This approach provides insight into how a model is organized and how its parts interact.

3.2 Mathematical Modelling of the Vane Pump

Mathematical models are developed for various components of the vane pump in this chapter. Mathematical modelling involves in representing the vane pump components in the form of equations. These mathematical models help in representing the vane pump components in Simulink Software. This mathematical modelling is done by considering the component properties such as flow properties.

In order to estimate the performance of the vane pump in the design, relationship between pump geometry and pump flow rate has to be developed. A schematic of the cross-section of the vane pump is shown in Figure 2.2 with the centre of the fixed rotor above and below the centre of the movable stator respectively. As a first step the flow rate generated by the oil pump should be determined. Wang.S et al [17] has modelled and controlled a vane pump for a valve less actuation system in this study the distance between the rotor centre and the vane end point is identified with equation (3.2).

A single pumping chamber, denoted by i^{th} chamber has been modelled first. The angular displacement of the i^{th} pumping chamber due to the angular speed ω of the rotor can be written as

$$\theta_i = \omega t + (i-1)\alpha \tag{3.1}$$

Where, α is the offset angle between two consecutive pumping chambers which in terms of the number of pumping chamber, z is 2π/z.

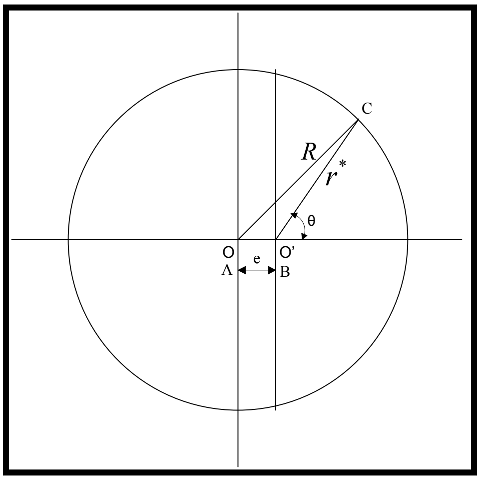

Figure 3.1 Definition of the variable length

As shown in Figures 3.1, R is the radius of the inner surface of the stator, r^* is the outer radius of rotor and θ is the angle that the vane has swept through relative to the initial up-right position. Considering figure 3.1, we can find the variable length r^* in terms of angular rotation, θ and the housing radius, R which is fixed. This variable length r^* causes the change in volume, from which the suction and delivery flow rate can be found. This flow is termed as kinematic flow. Triangle law is utilised in the aforementioned triangle ABC to derive the following relation:

$$\cos(180-\theta) = \frac{{r^*}^2+e^2-R^2}{2r^*e},$$

Results the expression of r^* as,

$$r^* = -e\cos\theta + \sqrt{R^2 - e^2 sin^2\theta} \tag{3.2}$$

After an initial rotation, vane 1 is at an angle θ_i with respect to the horizontal axis, and vane 2 is at angle $(\theta_i+\alpha)$.

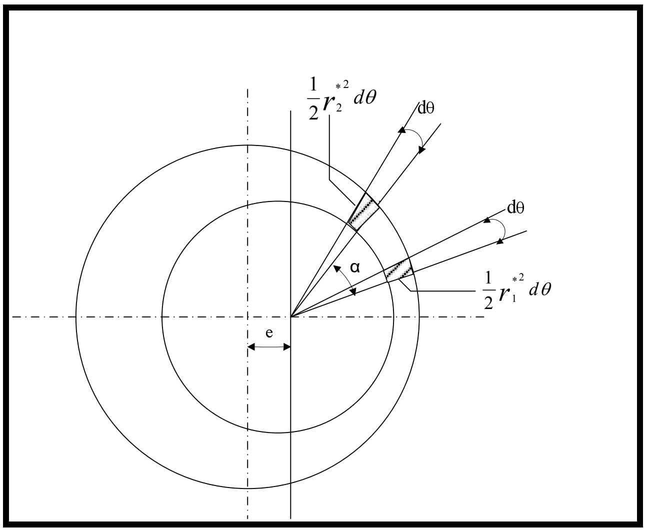

Figure 3.2 Representation of the differential volume

Now, considering a rotation by a differential $d\theta$, volume enclosed by the two vanes will change by an amount dV. As the angle of rotation is infinitesimally small, the change in volume can be approximately represented by the following equation with reference to Figure 3.2

$$dV = dA.L$$

$$= \frac{1}{2}\left(r_2^{*2} - r_1^{*2}\right)d\theta.L \tag{3.3}$$

Where, from equation (3.2)

$$r_1^* = -e\cos\theta + \sqrt{R^2 - e^2 sin^2\theta} \tag{3.4a}$$

and,

$$r_2^* = -e\cos(\theta+\alpha) + \sqrt{R^2 - e^2 sin^2(\theta+\alpha)} \tag{3.4b}$$

The volume entrapped between two adjacent vanes can be found by integrating the expression of dV with respect to time, as shown in equation (3.3),

$$V = L\left[\frac{1}{2}\int_{\theta}^{\theta+\alpha}(r_2^{*2} - r_1^{*2})d\theta\right] \tag{3.5}$$

and, $$Q_i = \frac{dV}{dt} \tag{3.6}$$

Suction flow to a single chamber through the port cut in the end cover is given by the following equation.

$$Q_{suci} = C_d A_s \sqrt{\frac{2}{\rho}|P_s - p_i|} sgn(P_s - p_i) \tag{3.7}$$

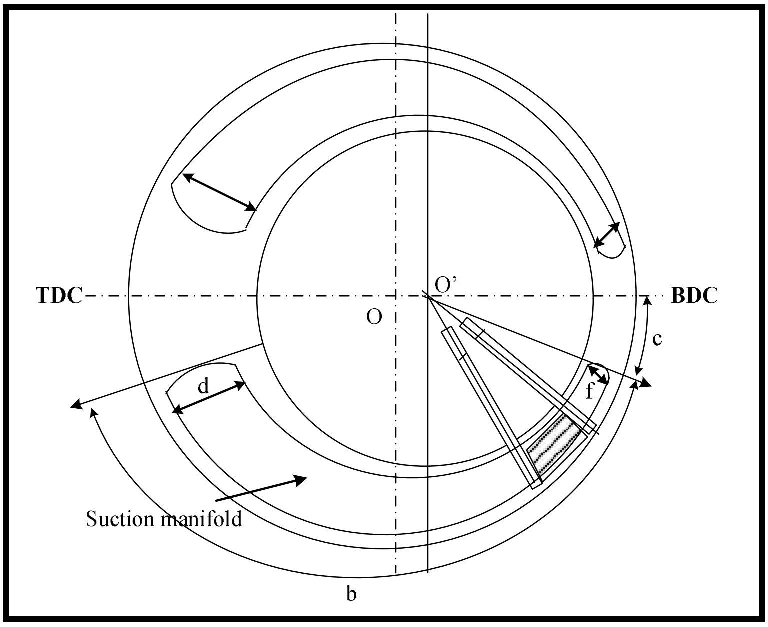

Figure 3.3 Schematic to understand the flow area between a pair of vanes

As shown in Figure 3.3, the suction flow area A_s can be described by Equation (3.8) with respect to angle rotated by the rotor is given by the following equation.

$$A_s = \int r_c \left[\frac{f-d}{c}(\theta - b) + d\right] d\theta \qquad \text{for } b - \alpha < \theta < b + c$$

$$= 0\ ; \qquad \text{for } \theta \leq b - \alpha \text{ and } \theta \geq b + c \qquad (3.8)$$

Where, r_c = Radius of the centre line of the delivery or suction port

f = Width of the port at the starting point of suction and end point of discharge port

d = Width of the port at the end point of suction and starting point of discharge port

c = working angle of the suction or delivery port

b = Starting angle of the suction or delivery port

Discharge flow from a single chamber through the port cut in the end cover is given by the following equation.

$$Q_{suci} = C_d A_d \sqrt{\frac{2}{\rho}|p_i - P_d|}\, sgn(p_i - P_d) \qquad (3.9)$$

In a similar way, area A_d of the port cut in end cover for collecting the discharge flow with respect to angle rotated by the rotor is given by the following equation.

$$A_d = \int r_c \left[f - \frac{f-d}{c}(\theta - \pi - b)\right] d\theta \text{ for } \pi + b - \alpha < \theta < \pi + b + c$$

$$= 0\ ; \qquad \text{for } \theta \leq \pi + b - \alpha \text{ and } \theta \geq \pi + b + c \ (3.10)$$

Pressure in the i^{th} chamber is generated due to compressibility effect for the difference between kinematic flow and pressure driven flow and is given by the following equation.

$$\frac{dp_i}{dt} = \beta \frac{Q_i - Q_{disi} - Q_{suci}}{V} \tag{3.11}$$

Where, p_i = Pressure inside the i^{th} chamber

Q_i = Kinematic flow of i^{th} chamber

V = Volume of the i^{th} pumping chamber

β = Bulk modulus of working fluid

P_s = Suction pressure

P_d = Discharge pressure

Expression for flow through the load orifice is given by the following equation.

$$Q_{suci} = C_d A_o \sqrt{\frac{2}{\rho} |P_d - P_s|} sgn(P_d - P_s) \tag{3.12}$$

Pressure P_d in the delivery pipe generated due to compressibility of fluid is given by the following equation.

$$\frac{dP_d}{dt} = \beta \frac{\sum_1^z Q_{disi} - Q_d}{V_{res}} \tag{3.13}$$

Pressure P_s in the suction pipe generated due to compressibility of fluid is given by the following equation.

$$\frac{dP_s}{dt} = \beta \frac{Q_d - \sum_1^z Q_{suci}}{V_{res}} \tag{3.14}$$

3.3 Structure of the Simulation Model

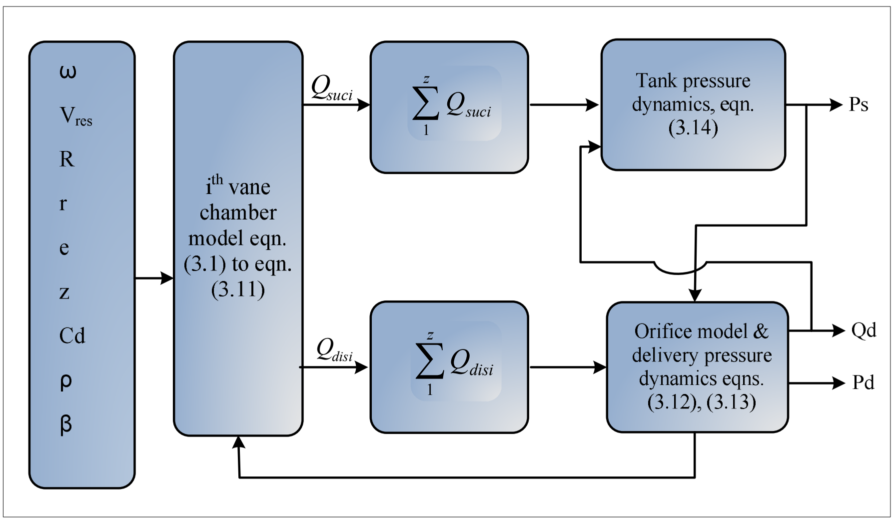

Figure 3.4 Simulation block diagram of vane pump

Explained equations and relations have been inserted to the MATLAB/simulink in order to solve the equations and simulate the system with different sets of parameters. Most of the parameters in vane pump are related with each other thus the output of the each block inserted in the simulink blocks connected to the related sub-block. Figure 3.4 shows the created MATLAB/simulink model of vane pump. As seen on the Figure 3.4 pump speed, length of vane pump, rotor radius and casing radius are basic design parameters and provided to the model as inputs.

3.4 Solution Methodology

Solution of equations (3.1) to (3.11) gives the flow rate for a single vane chamber. Same set of equations have repeated for z number of vane chamber incorporating the phase difference between two consecutive vane chambers which is equal to the angular extent between two vanes, α. The flow rates from the individual vane chambers have been algebraically summed and input to the next blocks respectively to evaluate the delivery and suction pressure of the pump as well as the load flow rate.

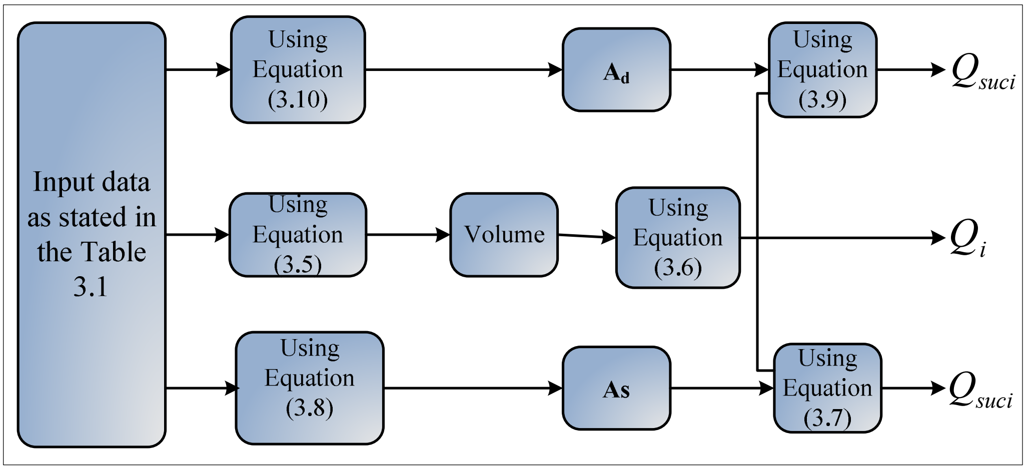

Figure 3.5 Block diagram of single chamber of vane pump

Figure 3.5 shows the simulation block diagram of single chamber of the vane pump. Solution of Equations (3.1) to (3.11) gives the flow rate for a single vane chamber. Same set of equations have repeated for z number of vane chamber incorporating the phase difference between two consecutive vane chambers which is equal to the angular extent between vanes, α.

The flow rates from the individual vane chambers have been algebraically summed and input to the next blocks respectively to evaluate the delivery and suction pressure of the pump as well as the load flow rate. This is shown in the Figure 3.4.

Solution of Equations (3.12) to (3.14) gives the flow rate and discharge pressure. Simultaneous solution of these equations provided the pressure and flow characteristics of an individual chamber as well as the overall pump. All parameters related to the pump is taken according **Table 3.1** is the input of the vane pump simulation model.

Table 3.1 Design parameters

Parameters	Value
Eccentricity, (e)	5e-4m
Inner radius of the casing, (R)	2.5e-3m
Inner radius of the rotor, (r)	1.5e-3m
Total length of the vane pump, (L)	0.01m
Rotational speed, (N)	3000rpm
Coefficient of discharge, (Cd)	0.6
Bulk modulus of working fluid, (β)	1.4e9Pa
Density of the working fluid, (ρ)	800Kg/m^3
Number of vanes, (z)	12
Angular extent between two vanes, (α)	30^0
Reservoir volume, (V_{res})	1e-6
Nominal needle valve opening, (A_0)	1.25e-7

The simulation is performed using Matlab - Simulink version 7.12.0. The 'Runge Kutta fourth order (ODE4)' solver with a fixed time step of 1×10^{-6} second is employed in the simulation. Simulations are performed for a maximum time of 0.10 seconds, required for about 5 revolutions of the pump at a rotational speed of 3000 rpm.

Summary

a) A set of equations have been developed to represent the chamber flow and pressure distribution of the vane pump chamber from eqns. (3.1) to (3.11). The equations have been derived considering a single groove cut on the slipper but can be extended to any number of grooves.
b) The different design parameters for vane pump have been identified.
c) The eqns. (3.12) to (3.14) give the delivery pressure and flow rate of the vane pump.

4 RESULTS AND DISCUSSIONS

4.1 Working Principle through Simulation Result

The model has been tested with a constant rotational speed profile and 100% needle valve opening. The different output is shown below. The pressure dynamics obtained for the vane pump is shown in Figure 4.1. The chamber pressure dynamics for a single cycle has been found to be repeated multiple times to achieve a pressure dynamics for multiple cycles. The distribution of pressure in the different lands has been modelled mathematically in the previous chapter. Solving the equations for the pressure distribution in MATLAB the variation of pressure with respect to time is evaluated for all chambers. Adding algebraically and again solved in MATLAB/simulink and get the different results is discussed below. Delivery pressure, suction pressure and 1st chamber pressure have plotted in single graph with respect to the time as shown in Figure 4.1.The delivery pressure distribution in the vane pump becomes steady after 2 revolutions. Delivery pressure curve lied on 1st chamber pressure at different point. In the same manner suction pressure also lied at some point of 1st chamber pressure which is clearly observed by Figure 4.4.

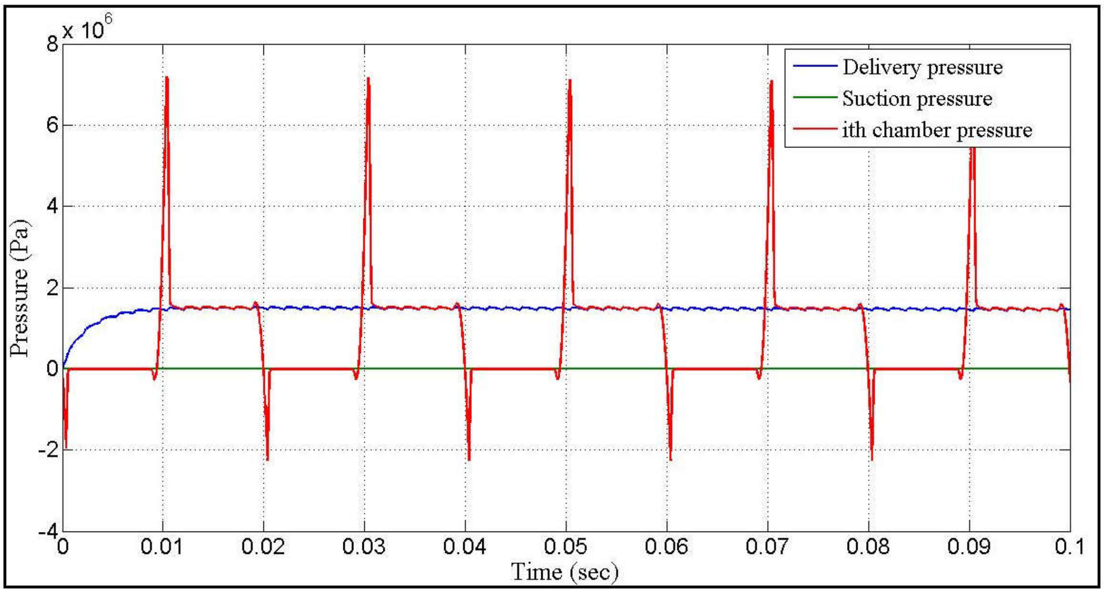

Figure 4.1 Pressure dynamics of the vane pump

Now, we plotted the graph of flow rate at different condition in 1st chamber. Figure 4.2 represents the kinematic flow with respect to time of 1st chamber that gives sinusoidal curve. Suction flow rate and discharge flow rate of 1st chamber of vane pump clearly show they are similar but opposite in nature as Figure 4.2.

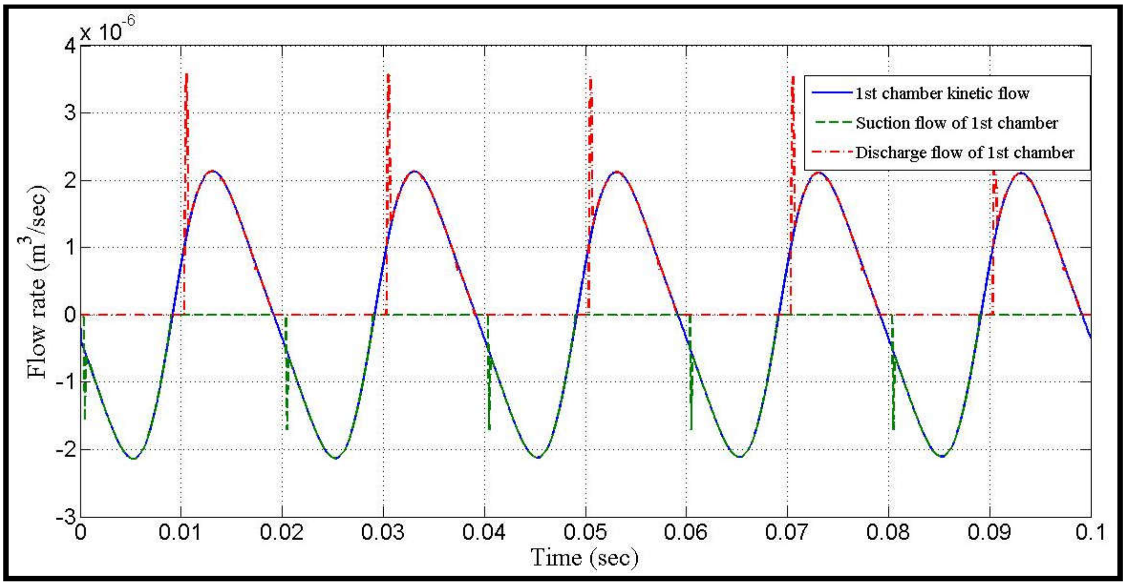

Figure 4.2 kinematic flow rate of 1st chamber of vane pump

Figure 4.3 shows the load flow rate which is obtained on 100% needle valve opening. Now we add algebraically of delivery flow rate of chamber 1 to chamber 12 then we see that it lies between the certain ranges which cause the steady flow rate of the vane pump. This is clearly observed in the Figure 4.3. Delivery flow rate becomes steady after 2 revolutions.

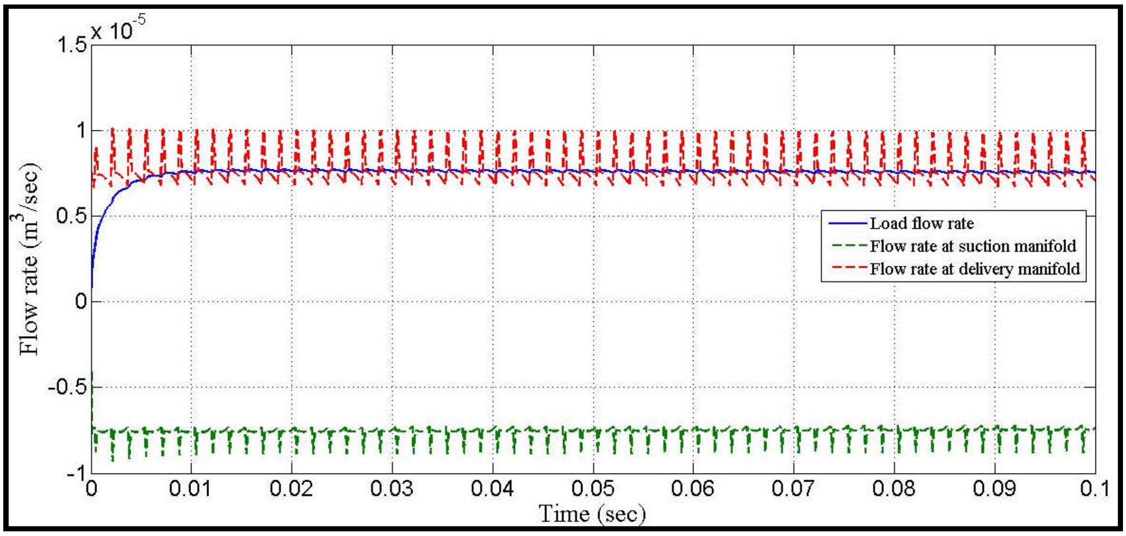

Figure 4.3 discharge flow rate from vane pump

The variation of suction flow areas of the port cut in end cover for suction flow with respect to time and delivery flow areas of the port cut in end cover for collecting the discharge flow of 1st chamber is shown in Figure 4.4. We see the when suction port area is open; delivery port area is closed and vice versa.

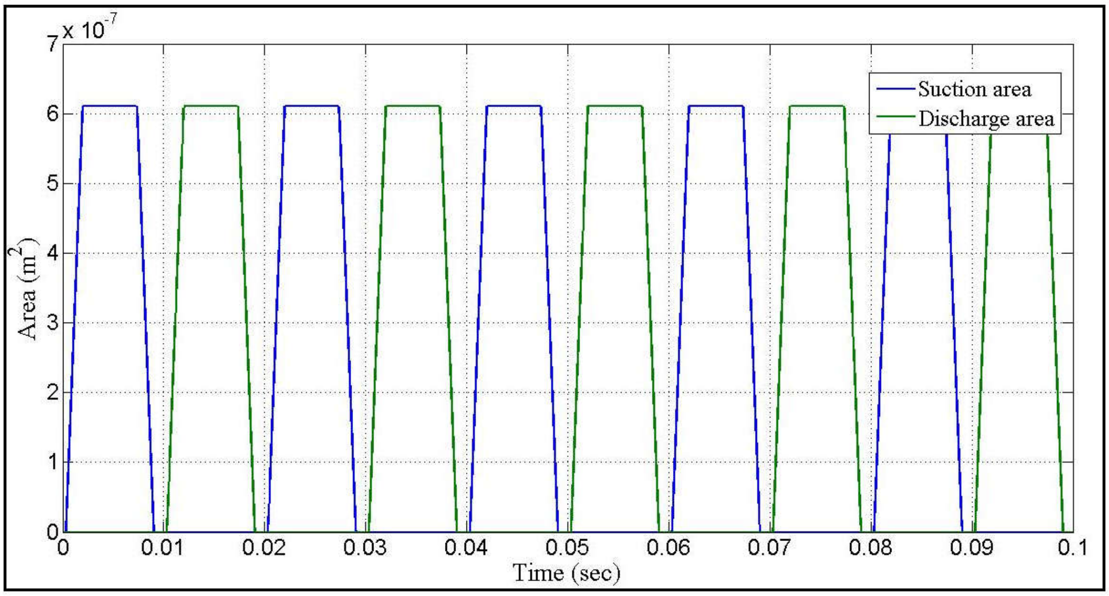

Figure 4.4 Suction port area and delivery port area of 1st chamber

4.2 Sensitivity Analysis of the Pump

4.2.1 Effects of different eccentricity values

Three different values of eccentricity are taken as a) e= 2.5e-4m b) e= 5e-4m c) e= 10e-4m and the performance of the pump in terms of delivery, suction and 1st chamber pressure have been plotted in figure 4.5.

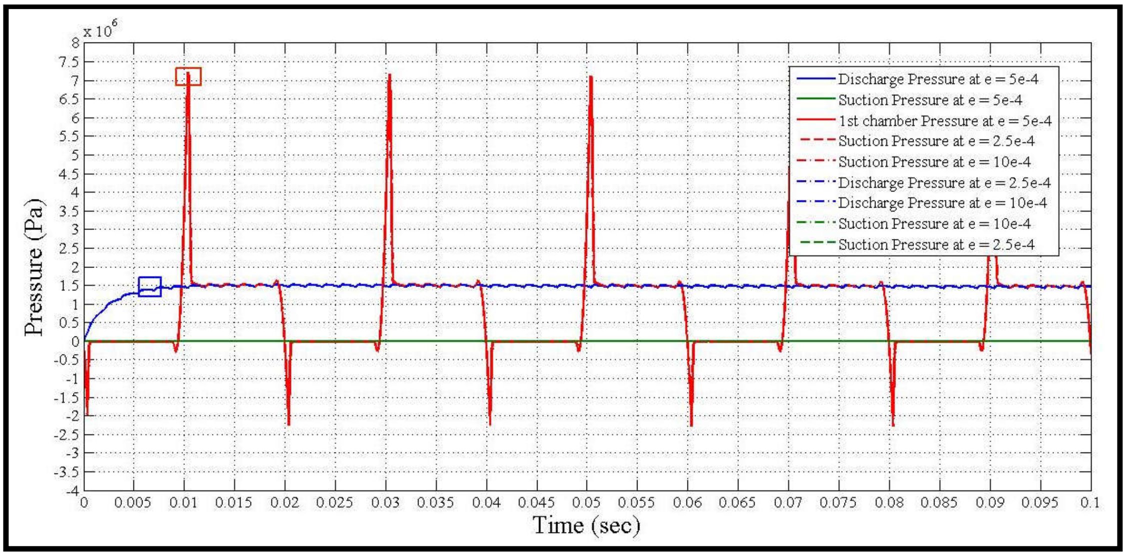

Figure 4.5 Pressure dynamics at different eccentricity values.

We see the decreasing value of eccentricity causes the slight increase in the value of delivery pressure which is clearly viewed in figure 4.6. The graph is magnified in figure 4.6 to show clear view of delivery pressure (blue colour box of figure 4.5 is magnified).

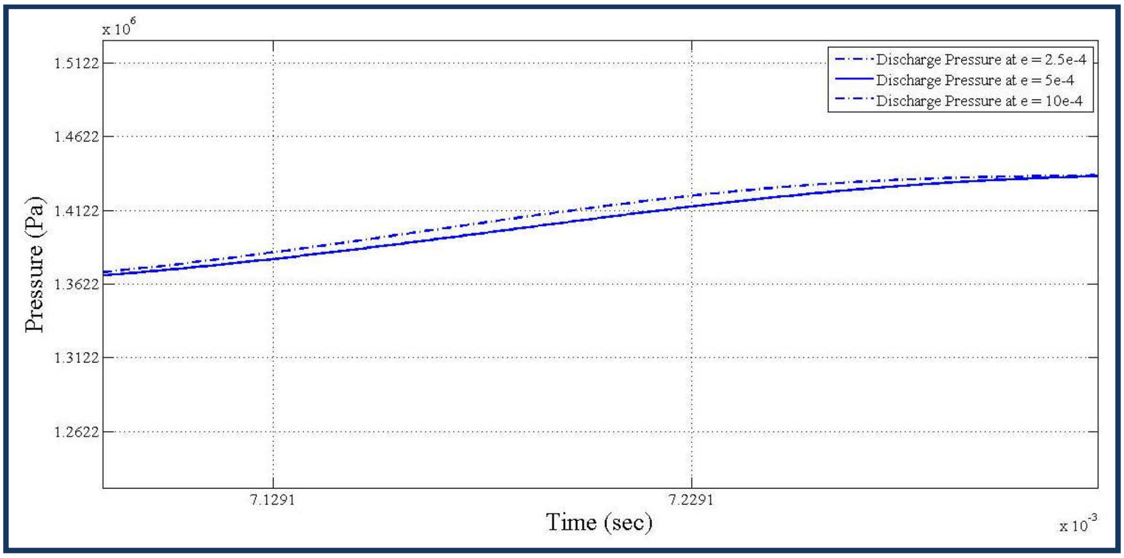

Figure 4.6 Magnified view of delivery pressure at different eccentricity

Again, the graph is magnified in figure 4.7 to show clear view of chamber pressure (red colour box of figure 4.5 is magnified). We see that the increasing eccentricity value slight decreases the spikes of chamber pressure curve. Which can clearly observed in figure 4.7.

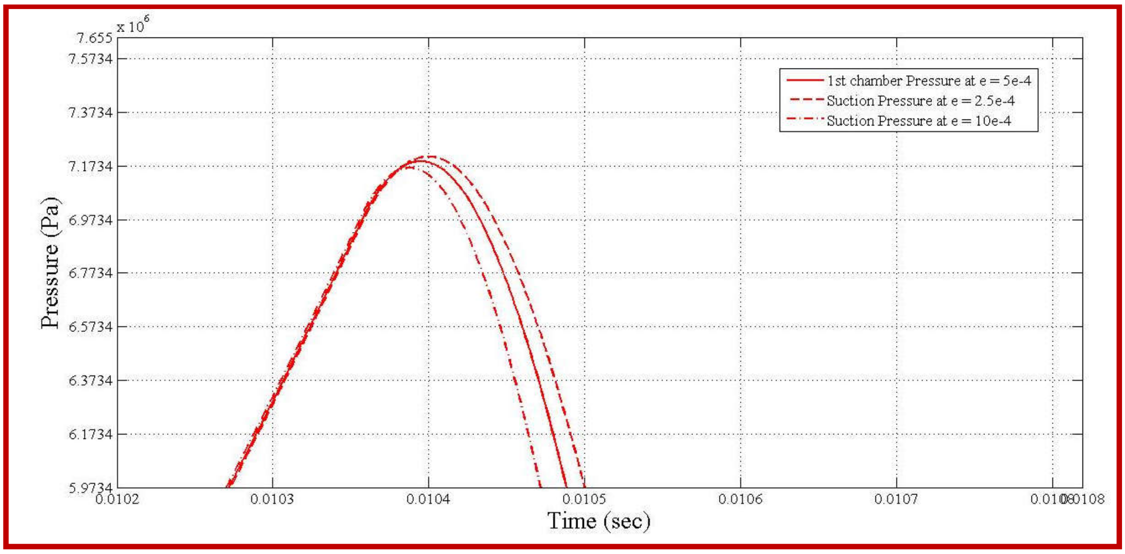

Figure 4.7 Magnified view of 1^{st} chamber pressure at different eccentricity

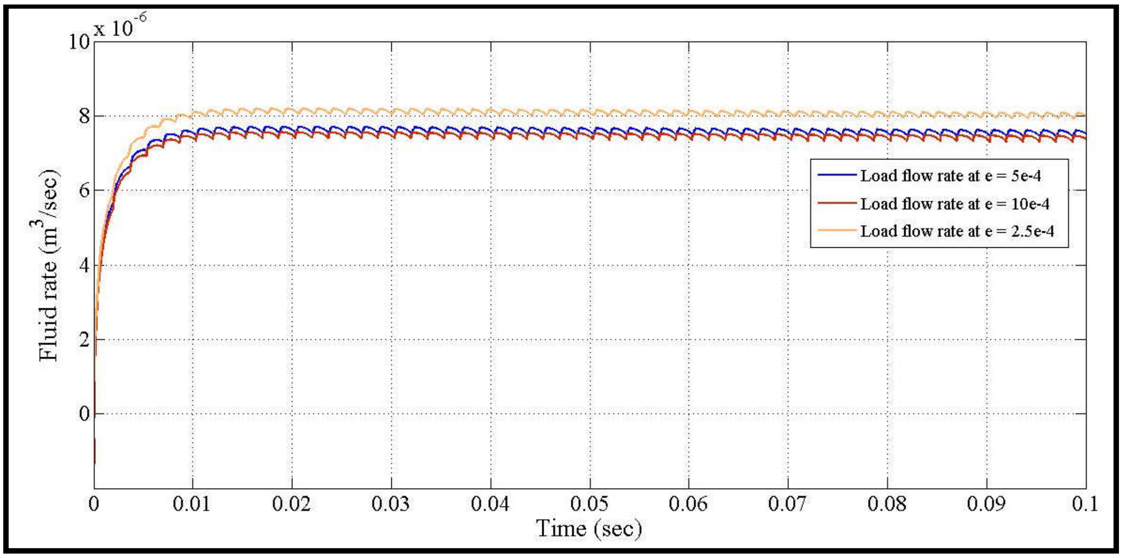

Figure 4.8 Discharge flow rate from vane pump at different eccentricity value.

We observed that increasing the eccentricity decrease the load flow rate. This can be seen in figure 4.8. But there is limitation of eccentricity values because it can causes the stop of pump mechanically.

4.2.2 Effects of different angular extent of manifold

We plotted the graph between the three different values of angular extent of manifold described below (c = angular extent of manifold) at c= 105^0, 115^0 and 140^0. This is shown in the Figure 4.9. We see the increasing the value of 'c' causes the decreasing the value of chamber pressure. The spikes of chamber pressure at c = 140^0 is relatively high with respect to spikes at c = 105^0.

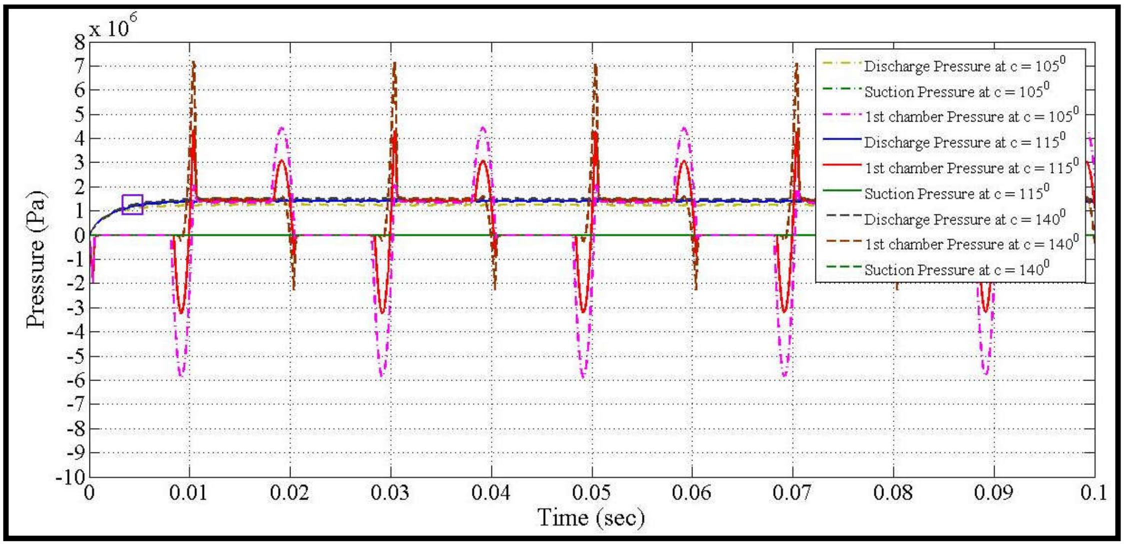

Figure 4.9 pressure dynamics of the vane pump at different angular extent of manifold

Figure 4.10 shows the magnified view (purple colour box of figure 4.9 is magnified) of discharge pressure as box is denoted in figure 4.9. We see the increasing the value of c causes the increase in value of delivery pressure but difference of increasing value is decreased.

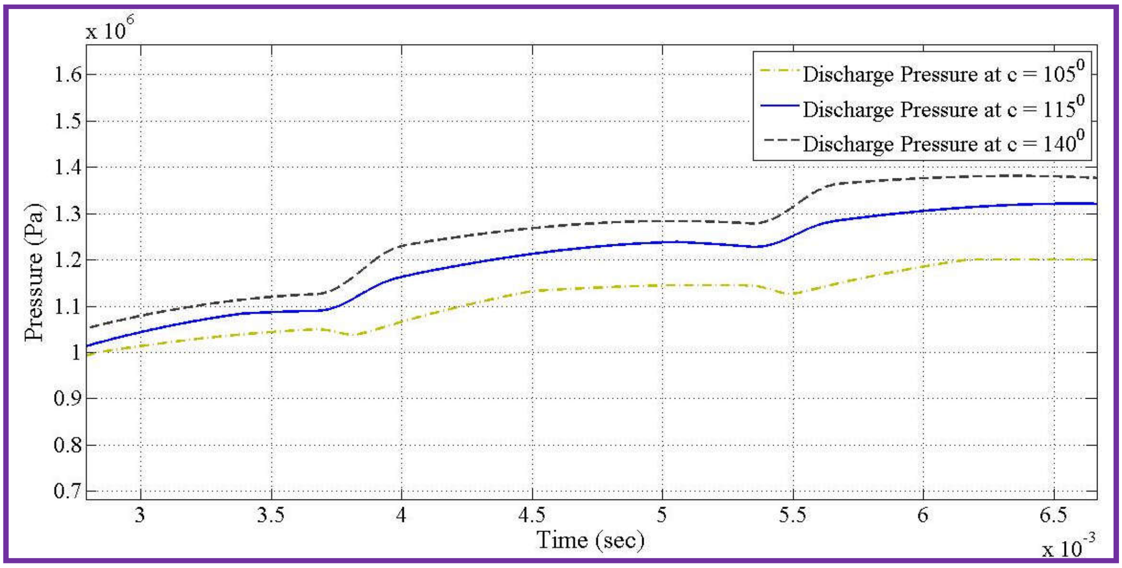

Figure 4.10 magnified view of delivery pressure at different angular extent of manifold

Also the same phenomenon occurs in flow rate as delivery pressure behaviour. As difference in increasing the value is decreased. It occurs due to increasing the value of manifold which causes the different flow rate. This can be seen in figure 4.11.

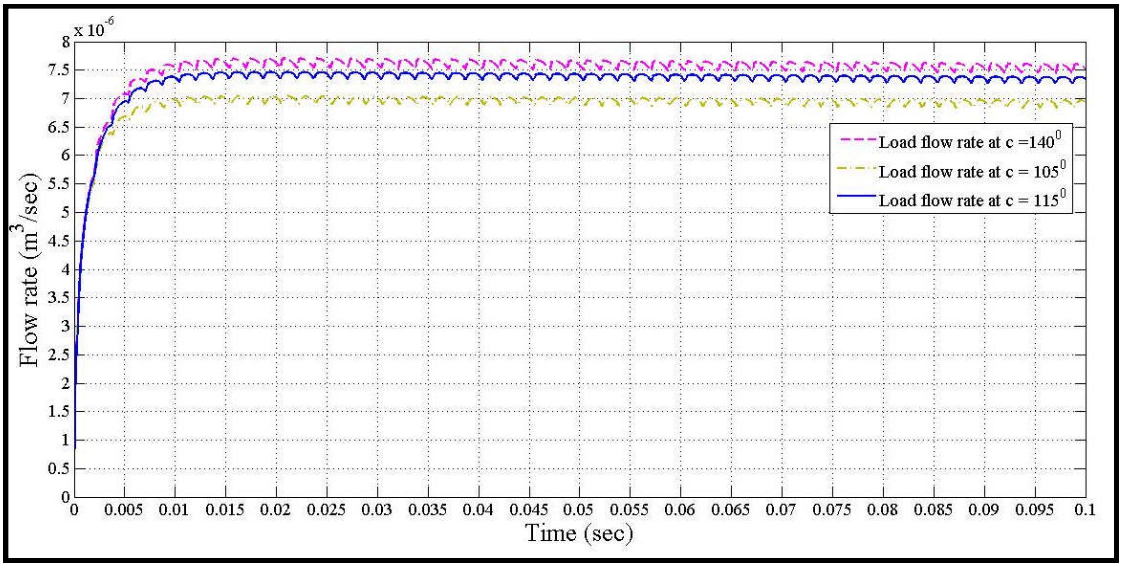

Figure 4.11 delivery pressure of the vane pump at different angular extent of manifold

4.3 Loading Effect on Pump Performance

Plotting graph at different valve area opening for pressure dynamics and load flow rate. We see the valve area opening is reducing and pressure gets increasing. The value of delivery pressure almost 4th time to reducing the half of the valve area opening. This is shown in Figure 4.12.

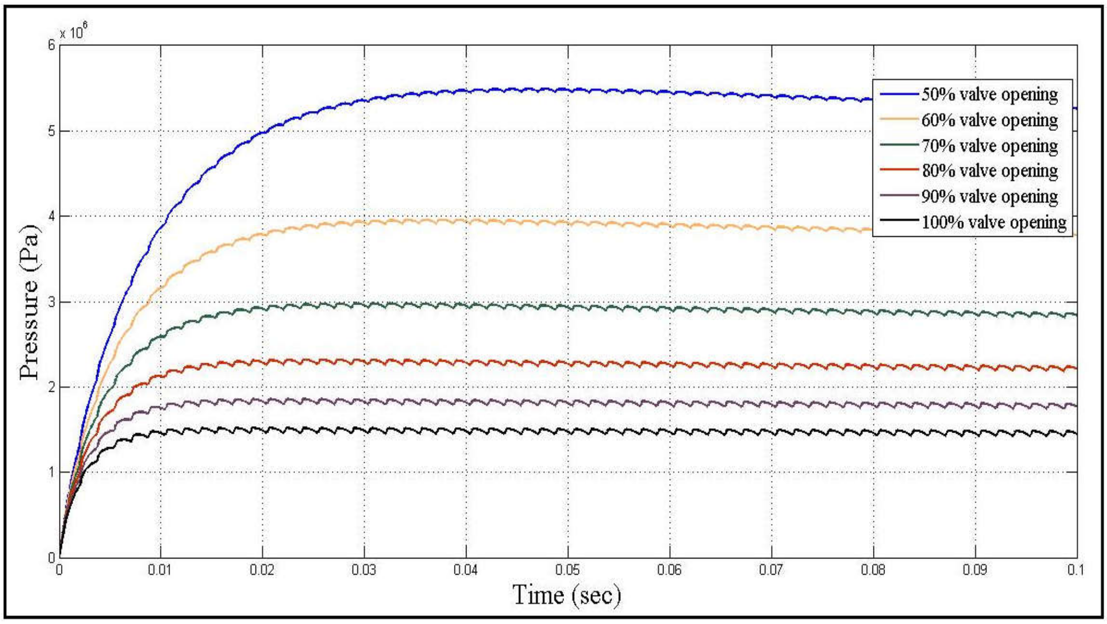

Figure 4.12 pressure dynamics of the vane pump at different loading condition

We see the valve area opening is reducing and load flow rate gets decreasing. But there is slight changes of load flow rate which can observed in the Figure 4.13.

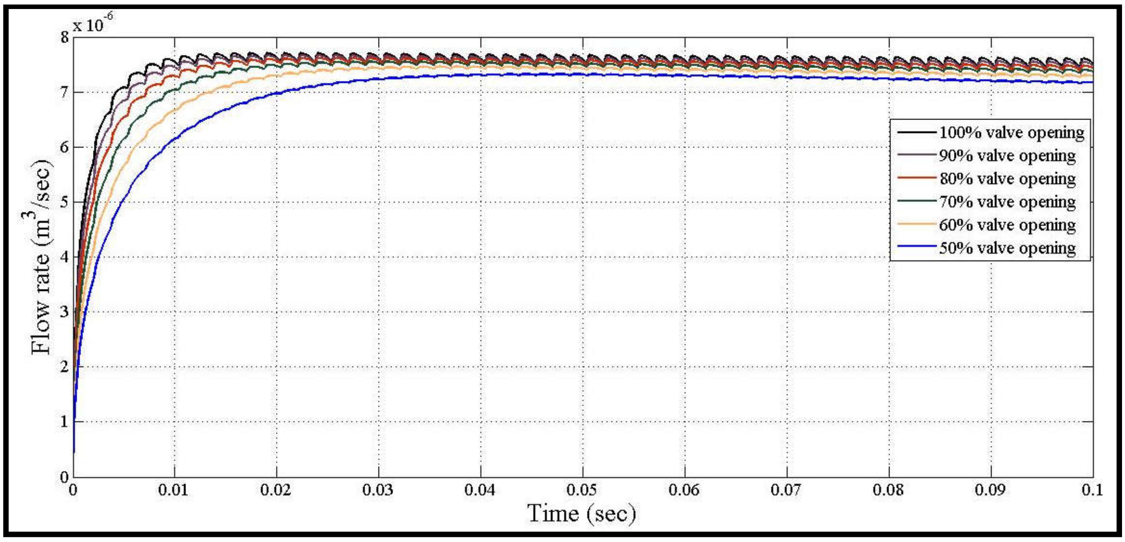

Figure 4.13 discharge flow rate from vane pump at different loading condition

Pressure versus steady state discharge curve

Positive displacement pumps (vane pump) do not utilize fluid momentum, meaning that flow rate is relatively independent of pump outlet pressure. Thus, (unlike dynamic pumps) positive displacement pumps have a definite capacity across a wide range of delivery pressures as shown in the figure 4.14 characteristic curve below. This indicates that small variation in pressure almost give the constant flow that is the main characteristics of the vane pump.

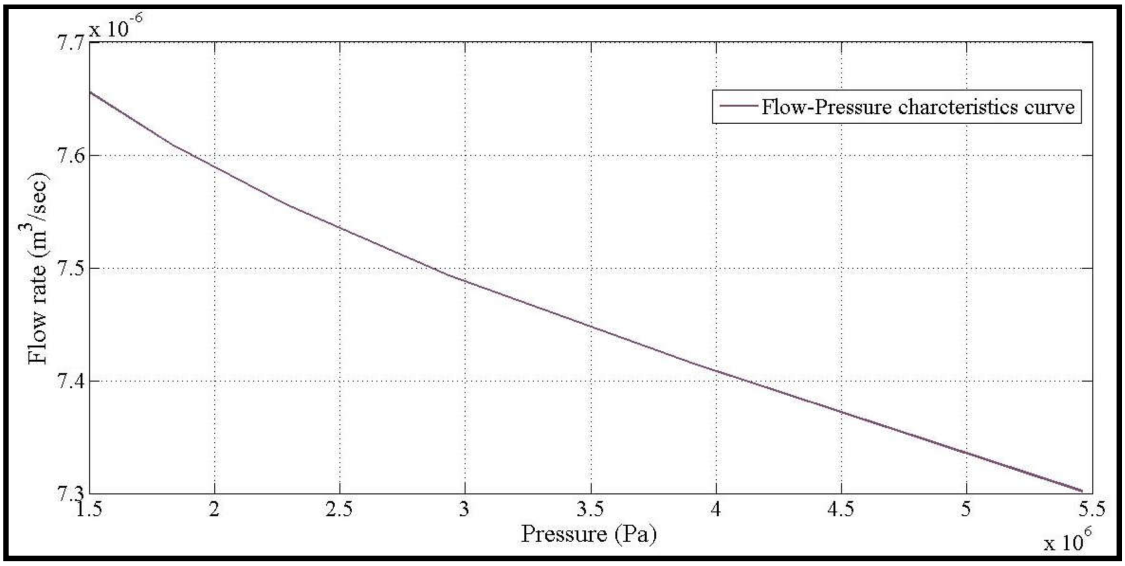

Figure 4.14 Delivery pressure versus flow curve

5 CONCLUDING REMARKS

5.1 Summary of the work

In this study a modelling of all subparts of a vane pump is discussed. A MATLAB/SIMULINK model is developed which includes all of the sub components of vane pump. All of the relations between these subsystems have been determined and connected to the related sub-block. This is achieved by the geometrical relations between rotor and vanes. As a first step a function of an algebraic equation from rotor centre to control ring have been determined to evaluate the area and then the volume between two adjacent chambers. Flow through the needle valve has been modelled with the orifice equations. It has been understood that all sub-systems of the vane pump operates compatible with each other.

As a conclusion a vane pump model which takes constant engine speed and constant oil pump parameters as input and provides oil flow rate on the pump outlet and through needle valve, pressure and flow rate at pump outlet as an output. This model also provides a quick review of the pump response to a new designed vane pump by changing the different parameters.

5.2 Future Scope of Work

The presented research is part of an ongoing work in constant development. There is a vast set of equipments found in typical vane pumping systems (e.g., other protection devices) that were not included in the model nor mathematically formulated. Despite having obtained a very good correlation between experimental and most simulation results, further comparisons should be performed, namely with field tests of real working vane pump systems with different characteristics, flow conditions, as well as for large, small and micro-hydro. Having this in mind, some suggestions are presented that seem relevant to future developments:

5.2.1 Numerical implementations

a) Mathematical formulations of the vane pump chamber area.
b) Implementation of the more relevant unsteady friction models;

c) Research and implement an adequate formulation for predicting cavitations problems caused by localized and distributed vapour opening/collapsing and consequent effects on the flow dynamics.

5.2.2 Experimental studies

a) Compare simulation results for pipelines with different materials (linear elastic and non-linear properties).

It is also desirable to develop a user friendly interface so that the programme is easier to use.

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