Design of a miniature pump

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DESIGN OF A MINIATURE PUMP

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SCHOOL OF MECHANICAL AND AEROSPACE
ENGINEERING
2013**

DESIGN OF A MINIATURE PUMP

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ABSTRACT

The applications of pumps are becoming increasingly extensive and there are more application-specific constraints that are imposed on pump designs, including constraints in size and capacity. Thus, miniature pumps have gained their popularity in many industries such as pharmaceutical, medical, MEMS applications etc. The design and development of a miniature pump is necessary to serve these applications. However, miniaturization process is both difficult and challenging.

In this project, a miniature pump with better performance has been designed and developed. Several designs are discussed and explored and the reciprocating ball pump (RBP) is proposed. The working principle of the RBP is presented. The RBP is unique in that it does not require any valves to regulate flow and inherently produce unidirectional flow. In addition, the design of this pump is simple, compact and easy to fabricate. Besides that, this pump has potential in pumping fluid with minimal contamination because the pumping unit is small in size and it does not require a seal and bearing. Furthermore, it has the potential in generating a higher flow rate as compared to the conventional reciprocating pump. The final RBP is 19.5 mm long, has a diameter of 18.8 mm and weighs less than 25 grams.

In this research project, a mathematical model has been developed with some assumptions and simplifications to reveal the working principle and to evaluate the performance of the RBP. It also enables the prediction on the instantaneous flow rate and pressure variations upstream and downstream of the RBP. In addition, parametric studies have been conducted to determine the effects of the weight of the ball and the

size of the pumping unit on its performance. In addition, parametric studies show that the optimal RBP has a casing length of 11.5 mm, a ball of diameter 9.6 mm with a rear cap hole diameter of 8.5 mm.

CFD simulation with fluid-structure-interaction (FSI) of the RBP has been successfully conducted via ANSYS-FLUENT by using moving and deforming dynamic mesh model with user-defined-function (UDF). Besides from revealing the working principle and the performance of the RBP, CFD simulation also shows the working principle of the RBP in a more comprehensive manner through the pressure distribution and velocity flow field.

Subsequently, the functionality of the RBP has been verified by experimental studies. In addition, a modified test rig was used to conduct flow visualization experiments. The mathematical model and CFD simulation have been validated by the experimental performance characteristic and flow visualization studies as they agree qualitatively with the results obtained from both mathematical model and CFD simulation.

The three studies show that the RBP generates additional induced flow especially during its backward stroke even though the RBP is operated at a back pressure of as high as 1000 Pa. This proves that the RBP is superior to a conventional reciprocating pump having the same dimensions. The RBP has a flow efficiency of as high as 172.15%, 170.88% and 159.90% which are measured through mathematical model, CFD simulation and experimental studies respectively.

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CHAPTER 1

INTRODUCTION

1.1 Background

A pump can be defined as a machine that delivers a specific quantity of fluid while increasing its pressure. Pumps have existed for centuries and are used in many daily applications. They are probably the second most common machines in use nowadays; the electric motor being the most used machine [1]. Hence, it is not surprising to see that they are produced in many different types and sizes ranging from big capacities pumps for marine, oil and gas industry to micro-pump for MEMS or biotechnology application. Currently, the biggest pump is the progressive cavity pump that is manufactured by Seepex [2] and weighs at seven ton. It is capable of delivering 180 m³/hr of thickened sewage sludge. At the other end of the spectrum are small pumps of the size of micro scale. However, as the application of pumps has become increasingly extensive, there are more application-specific constraints that are imposed on pump designs, including constraint in size and capacity.

Pumps are needed for a wide range of applications. It is also used in complex devices whereby numerous components are packed within one single device. This implies that there is a size constraint on the design of the pump when it is integrated in a complex device. Hence, a miniature pump concept is popular among Original Equipment Manufacturers (OEMs) to ensure the competitiveness of their product. In field applications, miniature pumps can be applied in the liquid cooling system of small components, in printing technology as well as in medical or pharmaceutical applications.

The usage of a miniature pump is normally restricted to applications that require a relatively low flow rate. In many industries such as pharmaceutical, medical, MEMS etc, miniature pumps have become popular due to its scale, mobility and precision. Some diseases such as diabetes, dehydration, gastrointestinal diseases or even cancer require administration of medication or drug with varying volume and as little as 0.1 mL/hr at specified time interval throughout the day and over a prolong period of time. This would be impractically expensive or unreliable if it is performed manually by nursing staff. Besides that, the control on the amount or flow rate of the delivered drugs or medications is very crucial for a complete infusion therapy. Thus, an infusion pump that is convenient and cost effective is used to infuse or administer drugs, medications or nutrients into a patient's circulatory system intravenously. Generally there are two types of infusion pump that are widely used in the hospitals, namely, the large volume peristaltic pumps which used rollers to compress a silicone or polyurethane tube to push drugs or medication through the tube and small volume infusion pump that uses a computer-controlled motor turning a screw to push the plunger on a syringe to deliver drugs or medications into the patient's body. However, these infusion pumps are normally large in size and heavy in weight and thus restricted the mobility of the pump. As such there is an urgent need for portable pumps for enhanced mobility. Hence, many researchers are putting efforts in developing smaller and wearable devices so that patients can maintain their normal life style and perform daily activities with minimal disruption [3, 4].

The application of miniature pump as chemical and drug delivery system also implies that liquid contamination has to be minimized. One of the methods is to eliminate contact between the actuator and the liquid delivered. The current method to prevent

contamination is by either using peristaltic design or diaphragm pump [5-11]. The limitation of a peristaltic pump lies in its inability to operate at high frequencies due to constant rubbing between the roller and the elastic tube. In the case of the diaphragm pump, compactness in the design is difficult to achieve as allowance has to be made for the volumetric expansion of the diaphragm [5, 7, 10, 11]. Under high frequency operation, the diaphragm may also suffer from fatigue problems due to continuous expansion-contraction cycle.

In recent years, many researchers devoted their efforts in developing a miniature pump to serve these applications. However, miniaturization is not an easy task as they face problems in the miniaturization of existing pumps. Due to the constraint in size, in order to achieve reasonable flow rate for a particular application, the pump has to operate at a high frequency. However, this in turn causes fatigue problem on the pumping mechanism.

In view of the need for miniature pumps and the problems faced by current designs, a miniature pump with better performance using the concept of reciprocating ball has been developed and tested.

1.2 Objective

The objectives of this project are to design and develop an innovative design of a miniature pump that is small and portable. The scope of this project is as follows:

- i. The first stage is to review existing pump designs that are available in the market or under research for various kinds of applications and to identify the limitations of these pumps.

- ii. The second stage is to design and develop a miniature pumping mechanism that meets the requirements and overcomes existing limitations that were found during the first stage of the project. Once the concept has been proven and recognized as feasible, subsequent actions would be taken to verify its functionality and actual developmental procedures begin.
- iii. A CAD model is indispensable during the design stage. Thus, a CAD model of the new pumping mechanism will be created. The CAD model will be used in the preliminary investigation which will reveal the design characteristics in the fundamental working mechanism of the pump and assist in verifying its functionality.
- iv. Simulation will be carried out by using mathematical models and Computational Fluid Dynamics (CFD) to facilitate theoretical studies of the pumping cycle and its performance. These simulation programs can help to provide a better understanding of the miniature pump's operational characteristics.
- v. Fabrication and experimental studies on actual prototype will be carried out. The results obtained from the experimental studies will be used to validate the theoretical simulation work and CFD simulation work.

1.3 Organization of Thesis

This dissertation presents the design and development process of a new miniature pump, starting from the initial stage of conceptualization to prototype testing.

A thorough literature review of existing pumping mechanisms that are commercially available and miniature pumping mechanisms that are under research is presented in chapter 2. This helps to recognize the limitations of various kinds of pump in their respective area of application and serves to determine the area of application where the miniature pump can be applied. This also helps to draw a series of specifications needed for the pump design. Chapter 2 also includes a review on mathematical modeling, CFD simulation model and fluid structure interaction simulation works.

Chapter 3 presents the preliminary designs explored in this project and the final design adopted. The problems associated with the final design are also discussed. An improved design has been proposed to overcome these problems. The design variation and operation orientation of the final design will also be discussed.

Chapter 4 presents the theoretical analysis and mathematical modeling of the pumping cycle and its performance. Parametric studies on the pump performance will also be conducted. In chapter 5, Computational Fluid Dynamics (CFD) simulation coupled with fluid-structure-interaction (FSI) is presented. This includes the assumptions made during the simulation, the domain setup, mesh generation, dynamic mesh and the solution method.

Experimental works and the details on the experimental setup for performance characterization and flow visualization are reported in chapter 6.

Results obtained from mathematical modeling, CFD simulations and experiments are presented in chapter 7. Validations on the mathematical modeling and CFD simulation

are also presented by comparing the theoretical results with CFD results and experimental results.

Conclusions are drawn and presented in chapter 8. Possible future improvements and further development of the miniature pump will also be discussed.

CHAPTER 2

LITERATURE REVIEW

This chapter consists three main sections. The first section presents a review on the evaluation of existing pumps, classification of pumps and their applications in different industries. This provides an understanding on the current technologies in different industries and their respective advantages and limitations. The second section reviews the mathematical model, simulation studies and experimental works on reciprocating pump. This gives an idea on the formulations of modeling different aspects for the new pump, the design of the experimental test rig and the measurement apparatus. The third section illustrates the needs for miniaturization and the challenges involved. Finally, at the end of this section, it concludes the chapter with the motivation for the present project.

2.1 Review on the Existing Pumps and Pump Applications

This section gives an overview of the history of pumps and its evolution. This is important because it provides an understanding of the desirable traits as well as the limitations that exist with these designs. This also helps to identity the field of application of the miniature pump and draws the specifications and requirements of the miniature pump which will be further elaborated in chapter 3.

2.1.1 Evolution of Pumps

Pumps have existed for centuries and are one of the most commonly used machines in the world throughout the history of mankind. Even today, pumps are still widely used in

many industries serving different applications such as water supply, gasoline supply, air conditioning systems, refrigeration, chemical movement, sewage movement, flood control, drug delivery, lifting, marine services etc. Generally, a pump is a machine that is used to facilitate the movement of fluid, transporting it from a lower pressure to a higher pressure by adding energy in order to overcome losses and pressure difference. A gas pump is generally termed as a compressor and is commonly used for low pressure-rise application such as heating, ventilating and air-conditioning application while the one that is used for liquid is generally termed as a pump.

The earliest kind of pump, which is the Archimedes screw, was first used by Sennacherib [12], King of Assyria, for the water systems in the Hanging Gardens of Babylon and Nineveh in the 7th century BC. The pump was later described in details by Archimedes in the 3rd century BC [12]. In the 13th century AD, al-Jazari described and illustrated different types of pumps, including a reciprocating pump, double-action pump with suction pipes, water pump, and piston pump [13]. These pumps were basically invented for water supply purposes.

Over the centuries, pumps have extended their applications from pumping water to pumping other fluids such as air, gases, refrigerant, oil, petrol, bio-fluid, drugs, medicine, blood and etc. Pumps have also evolved from macro-scale pumps such as water pump, compressor, hydraulic pump to micropumps such as membrane pumps [14-17], electrohydrodynamic pumps [18-21], electrokinetic pump [22-26], viscous pumps [27-29], peristaltic pumps [6, 8, 9, 30-32], ultrasonic pumps [33-36] that are used for specific applications.

2.1.2 Classification of Pumps

One of the most widely adapted classifications of pumps is to define the principle by which energy is added to the fluid, identify the means by which this principle is implemented, and finally delineates specific geometries commonly employed. This system is therefore related to the pump itself and is unrelated to any consideration external to the pump even to the material from which it may be constructed.

By using this approach, pumps can be categorized into two major groups [1, 37, 38]; dynamic and positive displacement. Dynamic pumps produce flow by increasing the velocity of the fluid, whereby energy is continuously added to the fluid passing through the machine with the assistance of a rotating vane or impeller. Positive displacement pumps create flow by trapping a fixed amount of fluid and then forcing the trapped volume of fluid to move with each revolution of the pumping element. The pumping element includes gears, lobe, rotary pistons, vanes, screw and etc. Energy is periodically added by application of force to one or more movable boundaries of any desired number of trapped volumes, resulting in a direct increase in pressure.

Dynamic pumps may be further classified into radial flow, axial flow and mixed flow pumps; and other special-effect pumps. Figure 2-1 presents an outline of the main and sub classifications within this category.

Figure 2-1 Classification of Dynamic and Displacement Pumps [1]

Displacement pumps are basically classified into reciprocating and rotary types, depending on the nature of movement of the pressure-producing members. Each of these major classifications may be further subdivided into several specific types as indicated in Figure 2-1.

However, not all pumps can be classified into these two categories especially for micropumps which are gaining popularity in recent years. The positive displacement

pumps and dynamic pumps mentioned above are mechanical pumps that require moving parts such as check valves, oscillating piston, turbine or membrane for delivering a constant fluid volume in each cycle. At the micro scale level, viscous forces in microchannels increase with miniaturization and mechanical pumps cannot deliver enough energy to overcome the high fluidic impedance [39]. However, some researchers have made use of this effect and have developed a new type of pumps, viscous pump [27-29] where the working principle is different from that of the positive displacement and dynamic pumps. Apart from this, non-mechanical micropumps have also been developed. These pumps include electrohydrodynamic pumps [18-21], electrokinetic pumps [22-26], phase transfer pumps [40-43], electro wetting pumps [44-46], electrochemical pumps [47, 48], magnetohydrodynamic pumps [49-51] and etc.

2.1.3 Positive Displacement Pump

By definition, positive displacement pumps displace a known quantity of fluid with each revolution of the pumping element. The pumping element includes gears, lobe, rotary pistons, vanes, screw, piston, plunger and etc. Positive displacement pumps can be classified into rotary pumps and reciprocating pumps due to their nature of motion. A brief review on both rotary pumps and reciprocating pumps will be presented in this section.

a. Rotary Pumps

Rotary pumps displace liquid by creating a space between the pumping elements and trapping liquid in the space. The rotation of the pumping element then reduces the size of the space and moves the liquid out of the pump.

Figure 2-2 Rotary Vane Pump [52]

Rotary pumps can be further divided into two groups which is single rotor and multiple rotor pumps. Single rotor pumps consist of one rotating element only. Single rotor pumps include rotary vane pumps, rotary piston pumps, flexible membrane pumps, screw pumps and peristaltic pumps. Figure 2-2 shows a typical single rotor rotary vane pump. Rotary vane pump was first invented and patented by Barnes in 1874 [53]. The pump consists a stationary housing and an offset rotor. One or more vanes are mounted on the rotor, supported by spring so that they are always forced to touch the internal surface of the housing and thus crescent-shaped cavities are formed. As the rotor rotates, trapped fluid moves and is displaced through the outlet. However this pump is not suitable for high pressure and high viscosity operation. Large frictional force between the vane tip and the housing is another disadvantage. Furthermore, the pump cannot be operated for fluid carrying abrasive particles as the vanes will be damaged by the abrasives.

Multiple rotor pumps consist of more than one rotating elements in order to trap the liquid. These pumps include gear pumps, lobe pumps, circumferential piston pumps and

multiple screw pumps. Gear pump is one of the most commonly used multiple rotor pump. It uses the meshing of gears to pump fluid. There are two main types of gear pump namely external gear pumps which use two external spur gears (Figure 2-3) and internal gear pumps which use an external and an internal spur gear. Helical and herringbone gear sets are also used instead of spur gear.

As the gears rotate, they separate the intake and discharge sides of the pump, creating a void which is filled by incoming fluid. The fluid is carried by the gears to the discharge side of the pump, where the meshing of the gears displaces the fluid. The tight micro scale clearances, along with the speed of rotation, effectively prevent the fluid from leaking back to the suction side.

The design of the gears and housing allows very high pressures to be developed and the ability to pump highly viscous fluids. However, this pump cannot handle fluids transporting abrasive or solid particles.

Figure 2-3 External Gear Pump [54]

Rotary pumps are used in a wide range of applications - liquids, slurries, and pastes. They can handle liquids of different viscosity, a wide range of pressure, and can deliver minute doses to large capacities of fluid. Rotary pumps are self-priming and deliver a constant, smooth flow, regardless of pressure variations.

b. Reciprocating Pumps

The reciprocating pump is a mechanical device that uses pistons or plungers to impart a pulsating, dynamic flow to a liquid. The pistons or plungers in the liquid are normally driven by a rotating crank and connecting rod mechanism which converts rotary shaft motion into axial reciprocating motion. The motion produced is more or less harmonic in nature. The flow generated by this reciprocating motion is directed from the pump inlet (suction) to the pump outlet (discharge) by the selective operation of the self-acting check valves located at the inlet and outlet. This type of pump was used extensively in the early days of steam propulsion in the 19th century as boiler feed water pumps. Though still used today, reciprocating pumps are typically used for pumping highly viscous fluids including concrete and heavy oils as they are generally robust and very efficient in pumping many types of fluids at high delivery pressure.

The most common type of reciprocating pump is the single-acting piston pump which is shown in Figure 2-4. In a single-acting piston pump, the liquid in the cylinder is discharged only during a head-end or crank-end for one half of a revolution. This means that the pump is only pumping the liquid during its forward stroke and it is not pumping during the backward stroke. The direction of the flow is controlled by self-acting check valves located at the inlet and outlet. Due to the disadvantages of this half-stroke period of no delivery of single-acting piston pump, many other designs have been proposed to eliminate this problem.

Figure 2-4 Single-Acting Reciprocating Piston Pump [55]

The double-acting pump as shown in Figure 2-5 is designed to discharge the liquid in the cylinder during both head-end and crank-end stroke for a full revolution. This eliminates the problem of usual half-stroke period of no delivery of a single-acting pump. However, this design reduces the volumetric flow rate during the backward stroke, thus creating flow variation and pulsation to the flow. In order to solve this problem, uneven displacement of the rotating crank is required to achieve the same volumetric flow rate for both the forward and backward stroke. Unfortunately, the design of the uneven displacement rotating crank is tedious. To avoid the design of uneven displacement of the rotating crank, the Simplex single-double-acting pump was introduced. As shown in Figure 2-6, this arrangement allows a single-acting pump to perform like a double acting pump whereby it delivers liquid during both head-end and crank-end stroke. The delivery is divided into two equal flow rates per stroke, thus, permitting improved pulsation control. For greater improvement in flow variation of reciprocating pumps, the single-acting multi-cylinder type of piston pump was later developed.

Figure 2-5 Double Acting Piston Pump [56]

Figure 2-6 Simplex Single-Double-Acting Pump [56]

Over the past few decades, many other different types of reciprocating pumps have been developed. These include articulated pump, opposed pump, membrane pump, diaphragm and dual-disc pump [56].

2.1.4 Dynamic Pumps

In contrast to positive displacement pumps, dynamic pumps are pumps where kinetic energy is continuously added to the fluid by increasing the flow velocity. This increase in energy is converted to potential energy, i.e. pressure, when the flow exits the pump into the discharge pipe. Dynamic pumps can be further subdivided into centrifugal, axial

and mixed flow pump according to the impellers used and the direction of the flow [38, 57, 58].

Figure 2-7 shows a schematic diagram of the typical centrifugal pump. A typical centrifugal pump consists of two main parts, the impeller and the casing. The casing is in the form of a volute. As the impeller rotates, fluid is drawn into the pump through the centre of the pump. It is then accelerated by the rotating impeller and flows radially outward into a diffuser or volute chamber and discharges into the downstream piping due to centrifugal forces. Thus, centrifugal pumps are also known as radial flow pumps. Centrifugal pumps are generally suitable for high pressure but relatively low flow rate applications.

In some applications where high flowrates at low heads are required, centrifugal pumps are not suitable. In order to generate a higher flow rate without increasing the impeller diameter, the tendency is to increase the inlet diameter of the pump. This results in the changes to the shape of the vanes of the impeller in order to handle large volumes of flow. The modification makes the fluid to whirl around the rotating vanes. This type of pump is called axial flow pump. For this type of pump, the flow is primarily in the axial direction and is parallel to the axis of rotation of the shaft. Figure 2-10 shows a schematic diagram of an axial flow pump. As compared to centrifugal pump, axial flow pumps are capable of handling larger volume flow rate at relatively low pressure.

Figure 2-7 Schematic Diagram of a Typical Centrifugal Pump [57]

Figure 2-8 Schematic Diagram of a Typical Axial Flow Pump [58]

Between these two types of pumps, there is a class of pump which has both radial and axial components. This type of pump is called mixed flow pump. Mixed flow pumps function as a compromise between radial and axial flow pumps. The fluid experiences both radial and axial acceleration and exits the impeller at an angle between 0-90 degrees with respect to the axial axis. The exit angle of the flow dictates the pressure and discharge characteristics in relation to radial and mixed flow. Due to the construction of mixed flow pumps, it can operate at higher pressures than axial flow pumps while delivering higher discharges than radial flow pumps.

One of the advantages of dynamic pumps over positive displacement pumps is that dynamic pumps can operate under closed valve condition. In contrast, reciprocating pumps will result in a continual build up in pressure and eventually lead to mechanical failure if operated under closed valve condition. However, dynamic pumps usually have lower efficiencies than positive displacement pumps. Dynamic pumps are also able to operate at fairly high speeds and high fluid flow rates.

2.1.5 Pumps Application

Nowadays, billions of pumps are commercially available or under research serving different kinds of applications all over the world. In this section, the applications of pumps in some specific areas will be discussed namely in (1) Pharmaceutical Application, (2) Medical Application, and (3) MEMS Micropumps Application.

a. Pharmaceutical Applications

The syringe which is illustrated in Figure 2-9 is the most commonly used pump in pharmaceutical applications. It is classified as a positive displacement pump, whereby a fixed volume of drug or medication is trapped inside the barrel and the fluid is pushed and then ejected out from the needle opening when force is applied onto the plunger.

Figure 2-9 Syringe [59]

It is one of the simplest and most widely used devices to infuse drugs, fluids, medication or nutrients into the patients' circulatory system. However, this injection procedure is impractical and expensive if a patient requires continuous administration of the drug. In order to overcome these problems, a convenient and cost effective solution for delivering drug has been introduced. These devices include infusion pump [3, 4], syringe pump [60]. Figure 2-10 shows a commercially available infusion pump which is commonly used in most hospitals.

Figure 2-10 Baxter Colleague CX Infusion Pump [61]

However, these pumps are generally not portable and not wearable and they are not suitable for patients who require regular intervals of drug intake such as those suffering from diabetes and requiring regular intake of insulin. As such they require a portable pump for enhanced mobility. Researchers are putting efforts in developing a smaller and wearable device so that patients can maintain their normal life style and perform daily activities with minimal disruption. Some researchers are working on an infusion pump which is implantable or with embedded computer control [3, 4].

b. Medical Applications

Heart failure is a major cause of death. Due to the insufficient supply of donor heart all over the world, Total Artificial Heart (TAH) and Ventricular Assist Device (VAD) are introduced to replace or aid patients' failing heart in order to regain their body functionality. Generally these devices served as a bridge to transplantation, bridge to recovery or destination therapy.

Currently, there are two TAHs which are FDA approved and clinically available nowadays. One of them is the CardioWest Total Artificial Heart (TAH) [62] developed by SynCardia Systems and the other one is AbioCor Implantable Replacement Heart (IRH) by ABIOMED which are shown in Figure 2-11. Both TAHs can be classified as a positive displacement pump whereby a fixed volume of blood is trapped within the diaphragm inside the pump and the diaphragm is displaced forward by compressed air pushing blood out of the prosthetic ventricle [62, 63]. In general, the TAHs that are clinically available are served as a bridge to transplantation or as destination therapy. Nowadays intensive efforts have been conducted by many researchers on developing a TAH that is fully implantable for permanent usage and can serve as destination therapy [64-67].

Due to the size of TAHs, patients are normally required to remove their biological heart and let the TAH to completely take over their cardiac function. In contrast, a Ventricular Assist Device (VAD) is a mechanical device that is used to partially replace the function of a failing heart. Currently there are several VADs that have been granted FDA approval through the pre-market approval (PMA) process namely HeartMate,

Thoratec® Implantable Ventricular Assist Device (IVAD), Novacor LVAS, Abiomed BVS Biventricular Support System and DeBakey VAD® Child [68, 69].

Figure 2-11 AbioCor Implantable Replacement Heart (IRH) [70]

Early generation of VADs are generally positive displacement pump that are similar to TAHs. Basically these VADs use a diaphragm to hold the blood and a pusher plate to alter the diaphragm compartment to create suction and compression stage. An actuator such as ball screw, roller screw, linear drive actuator, swash plate, undulation shaft, helical cam or a hydraulic system is used to drive the pusher plate [65, 71-74] in order to provide pulsatile flow that replicates the pumping motion of a biological heart. These pulsatile VADs include Novacor LVAS, HeartMate LVAS and they are generally smaller in size as compared to TAH.

These pulsatile VADs normally require a motion converter to obtain pulsation motion such as a roller screw, a solenoid-beam spring, a helical cam or a hydraulic system [74]. The motion converter is a barrier in reducing the size of the device, thus making miniaturization a difficult task.

Figure 2-12 Novacor LVAS [10]

Figure 2-13 The DeBakey/NASA Axial Flow Ventricular Assist Device [69]

Second generation of VADs such as rotary VADs overcome this problem. They can be categorized into centrifugal pumps or axial pumps which provide continuous flow instead of pulsatile flow. These rotary pumps have the advantages of simplicity, smaller

in size and greater reliability. Figure 2-13 shows a typical rotary VAD that was developed in 1988 by DeBakey [69, 75]. However, the effect of using continuous flow VADs in human body is still not clear yet [74].

c. MEMS Micropumps Applications

In the early years of MEMS development, fluidic components were among the first devices developed at the microscale level using silicon technology. The most common components were flow sensors, microvalves and micropumps. Using micromachining technology, a wide range of micropumps has been developed. With the growing importance of genomics, proteomics and the discovery of new drugs, these microfluidic systems are rapidly gaining importance. Micropumps are commonly used in chemical and biomedical applications requiring the transport of small, accurately measured liquid quantities. When utilized in chemical applications, micropumps are often a key component of a lab-on-a-chip device. Such devices are envisioned as providing a reasonably inexpensive way in conducting laboratory experiments. Micropumps can also be utilized in biomedical applications, where micropumps can be used to administer a small amount of medication at regular intervals.

As mentioned in the previous section, micropumps can be classified into mechanical and non-mechanical micropumps. Mechanical pumps at the micro scale level include check valve pumps, peristaltic pumps, valveless rectifications pumps and rotary pumps. All these micropumps are positive displacement pumps except the rotary pumps which are equipped with a microturbine. All mechanical pumps require a mechanical actuator, which converts electric energy into mechanical work. These actuators include disk type or cantilever type piezoelectric actuator, stack type piezoelectric actuator, pneumatic

actuator, shape memory actuator, electrostatic actuator, thermo-pneumatic actuator and electromagnetic actuator [39]. Figure 2-14 shows a general structure of a micro check valve pump.

Figure 2-14 General Structure of a Micro Check Valve Pump [39]

d. Liquid Cooling Devices

In recent years, the demand for reducing the size and weight of electronic devices has been growing rapidly. This leads to extensive research on reducing the size of microprocessor as well as on increasing the performance of the microprocessor. This in turn increases the complexity and power density of the microprocessor. However, the development of high performance compact size microprocessor has been restricted by advances in technology to remove heat from these microprocessors. In the near future, microprocessors and electric components are projected to dissipate over 1000 W/cm^2 of heat which is not easy to be removed using existing cooling techniques such as heat sink [76]. Thus, liquid cooling device has become one of the interesting research topics in removing heat from these high power density microprocessors. In order to deliver fluid to the heated surfaces, miniature pumps or micropumps are generally needed in the liquid cooling device. Figure 2-15 shows a simple schematic diagram of a micro cooling device.

Figure 2-15 Schematic Diagram of Micro Cooling Device [77]

In 2001, Darabi [76] introduced his electrohydrodynamic polarization micropump for electronic cooling. The advantage of the electrohydrodynamic pumping is that it does not require any moving parts and thus minimal maintenance is required and this leads to high reliability while low cost and low power consumption are other benefits. In 2010, He [78] successfully developed an electrohydrodynamic micro pump based on MEMS technology. Besides electrohydrodynamic micro pump, electroconjugate fluid (ECF) micropump has also been realized for liquid cooling devices by Kim in 2010 [79].

The next section will review the different methods used in analysis the performance of a typical pump.

2.2 Characteristics of Pump Performance

In the design and development of a pump, mathematical models, computer simulation and experimental studies are often used to characterize the performance of the pump. In this section, research studies on mathematical models, CFD simulations and

experimental studies of reciprocating pump will be reviewed. In addition, information on fluid structure interaction and flow visualization will also be presented.

2.2.1 Mathematical Model

A mathematical model is used to represent the essential aspects of reality (process, phenomenon, object, element, system, etc.) with the help of mathematical constructions. Mathematical models are one of the most powerful tools in any research and developmental work as they reduce actual experimentation which can be time-consuming and costly. In the design of a pump, a mathematical modeling is very important for the pre-design and post-design analysis. Before the design of a pump, mathematical modeling can help to establish the working principle of a newly designed mechanism. A mathematical model can provide an understanding of the characteristics of the system, thus accelerating the design and developmental process. It can also help to predict the performance of the pump or compressor prior to the making of the prototype. Subsequently, parametric studies by using mathematical models help designers or researchers to understand the system's operational characteristics and its sensitivity to various parameters, thus eliminating the need of numerous prototypes and reduce developmental costs. Once the prototype has been made, experimental studies can be conducted to verify the predictions of the mathematical model results in order to validate the accuracy of the theoretical predictions. Design improvement and optimizations can be carried out using the mathematical model to conduct optimization processes to determine the optimal parameters of the system in a more directed manner rather than trial and error on actual prototypes.

Over the past few decades, numerous research studies on the mathematical modeling of reciprocating pumps, reciprocating compressors and other positive displacement pumps have been conducted. Figure 2-16 shows a schematic diagram of a one cylinder single-acting reciprocating pump. In practice, the operation of a pump may be affected by a large numbers of factors. Thus, the designer of the mathematical model has to decide the key parameter and to assume some factors can be neglected for simplicity sake. This can be achieved through a parametric study. Generally, the modeling of a reciprocating pump or reciprocating compressor includes the models on piston motion, variation of cylinder pressure, inclusion of cavitation, valve motion, thermodynamic, and heat transfer in evaluating the performance of the pump [80-84]. Some special modeling such as pressure pulsation modeling due to the reciprocating motion has also been carried out in examining the flow characteristics on reciprocating pumps [85, 86].

Figure 2-16 Schematic of one cylinder of a reciprocating Pump [85]

In 1984, Edge [81] developed a digital computer model of a multi-cylinder piston pump with self-acting valves. His modeling includes the flow characteristic, piston cylinder dynamic, delivery manifold, delivery valves and inlet valve modeling. The simulation results showed close agreement with experimental measurements. However, the valve flow characteristic that was based on the theoretical models does not correlate well with measurements. In addition, his model does not include cavitation effect. A detailed

model for the pump shown in Figure 2-16 has been presented by Johnston [84]. His modeling includes the piston motion, valve flow rate with cavitation, cylinder pressure and cavitation, valve motion and lumped parameter model of load. His simulated results show good overall correlation with experimental measurements. However, some discrepancies were observed particularly regarding the velocity of opening of the inlet and delivery valves. In 1990s, Richard et al. [87] developed a PC simulation tool, BATHfp for simulating hydraulic circuits and other fluid systems. BATHfp allows user to build the hydraulic circuit to be studied in much the same way as it would be drawn by a draughtsman and it has been used by many researchers in studying the performance and behavior of their positive displacement pumps [88-91]. In recent years, with new pumps' design, some of the researchers used virtual prototype technology (VPT) to study their pumps. With VPT, it is convenient and flexible to build a complicated 3D CAD virtual prototype based on real physical model.

During the operation of a reciprocating pump, it is well known that pressure pulsation occurs in the pipeline. These pipeline pressure pulsations are a source of noise and vibration. This may have a significant influence on the reliability of the pump and its pipeline installation. It is strongly recommended to predict the pressure pulsations at the design stage of an installation in order to minimize their influence.

The flow in the pipeline can be classified as oscillatory flow of a viscous incompressible flow. Numerous theoretical works have been conducted by Atabek and Chang [92-94] on the entrance region in the oscillating pipe flow. Their analysis was based on linearization of the Navier-Stokes' equations by assuming that the inertia terms in the boundary layer equation remained the same as it was at the entrance section and that the

velocity distribution at the entrance cross-section was uniform [93]. Experimental work was performed by Gerrard and Hughes [95] on entrance region in oscillating pipe flow. They showed that laminar flow in front of an oscillating piston is fully developed at a distance $L = 0.3\delta R_\delta$, where R_δ is Reynolds number defined by Stokes layer thickness δ as a characteristic length.

In 1997, Shu et al [85] developed a new model for reciprocating pump which account for the presence of air pockets. The pipeline models can be coupled to existing model of pumping dynamics to allow time-domain simulations of pressure pulsation in both suction and delivery lines. The model was then validated in their experimental studies [86]. Based on the experimental results, the computer model predicts the pressure pulsation behavior with acceptable accuracy. Although pressure pulsation remains one of the critical phenomena that can be observed during the operation of a reciprocating pump, it is reported that the wave dynamics equations would be needed only if the time required for waves to travel the length of the pipe is much greater than the pumping period [96].

2.2.2 Experimental Investigation of Pump Performance

Experimental investigation is one of the best ways to study and evaluate the pump performance. Researchers propose hypotheses to predict the working principle and performance of a new pump and design experimental studies to test these hypotheses. Generally a pure mathematical study is not sufficient to predict and explain the phenomena, thus, experimental studies are needed to validate these assumption and the mathematical model.

Positive displacement pumps have existed for centuries. It is difficult to identify who pioneered experimental studies in this area. The earliest review was probably conducted by Isherwood [97] on a reciprocating pump, a rotary pump and a steam siphon pump in 1889. Since then, many experimental works have been conducted on studying the performance and other characteristics of positive displacement pumps. Some recent experimental works on positive displacement pumps include the experimental measurements conducted by Bachmann et al [98] on their pediatric ventricular assist device, the studies by Burton and Short on induced flow reciprocating pumps [99, 100], investigation of valves impedance pump by Hickerson et al [101], experimental investigation on a valveless pump conducted by Bringley et al [96] and etc. Most of these experimental investigations were conducted to study the performance characteristic of the pump that developed by each research group. In addition, the results obtained from their experimental studies were also used to validate the theoretical model.

Besides studying the performance characteristic of positive displacement pump, numerous experimental studies were also conducted in studying other characteristics of positive displacement pumps such as the pressure ripple or pressure pulsation of reciprocating pumps. In 1987, Vetter and Schweinfurter [102] developed a computational model to predict the pressure pulsation in the piping of reciprocating pumps. The data obtained from the computational model was compared with experimental data and good agreement was observed. Edge et al [86, 90, 103] have also conducted several experimental investigations on pressure ripple or pressure pulsation characteristics in order to validate their model.

2.2.3 Computational Fluid Dynamics (CFD)

Theoretical studies and experimental investigation have been the main approaches in the area of fluid dynamics since the seventeenth century [104]. However, with the advent of the high-speed digital computer combined with the development of accurate numerical algorithms for solving physical problem in recent years, computational fluid dynamics (CFD) has become a new “third approach” in the development of the fluid dynamics.

CFD involves the solution of the governing laws of fluid dynamics numerically. The complex sets of partial differential equations are solved in a geometrical domain that is divided into small volumes, commonly known as a mesh (or grid) [105, 106]. CFD can be used as a quantitative tool for narrowing down the choices between various designs. Designers and analysts can study the prototypes numerically, and then select the most promising prototype for experimental studies. In addition, CFD also allows observation of flow properties without disturbing the flow which is not always possible with conventional measuring instrument or observation of flow properties at locations which may not be accessible for the measuring instruments. However, CFD is not yet at the level where it can be blindly used by designers or analysts without sufficient knowledge of the flow physics. Despite increasing speed of digital computation, CFD is not matured enough to be used for real time computation yet. Numerical analyses require significant time to be set up and performed. Thus, CFD is an aid to experimental tools and it is used in conjunction with them.

Although numerous mathematical modeling works have been conducted on reciprocating pump, limited CFD studies can be found on reciprocating pump. In 2005, Gupta and Kshirsagar [107] performed numerical and experimental investigations of

cavitation in a pump by using a CFD package, CFX, with the newly introduced cavitation module. The overall performance prediction of the pump matched well with the measured results. In 2010, Jeong et al. [108] study the hydraulic fluid flow within axial piston pumps by using the 3-D numerical method and the process of generating discharge pressure ripples. Besides that, for micropumps, the working principle of a valveless micropump using piezoelectric actuator as the servo actuator was analyzed based on a CFD program ANSYS/Flotran [109]. In recent years, CFD software has been gaining its popularity in pumps' design and development due to the advancement in CFD software, the introduction of unstructured methodologies and new physical models which has greatly enhanced the versatility of CFD.

2.2.4 Fluid-Structure-Interaction (FSI)

Fluid-structure interaction (FSI) is one of the most challenging multi-physics problems which involve the interaction of some movable or deformable structure with an internal or surrounding fluid flow [110]. This is a very interesting topic for resolving many multi-physics problems that cannot be handled separately by a structural or a fluid point of view. One of the most common FSI problems found in reciprocating pump is the valve dynamics problem. In general, FSI problems are often too complex to solve analytically and they have to be analyzed by means of experiments or numerical simulation. Basically there are two main approaches for FSI simulation [111-115]:

➤ **Monolithic approach:**

The governing equations of the fluid flow problem and the structural problem are combined into a single system and solved simultaneously with a single solver. This method works well for multi-physics problem with strong physics coupling. However, the monolithic approach requires a code developed for this

particular combination of physical problems and has its own set of algorithmic issues/challenges. Figure 2-17 shows the FSI coupling spectrum which indicates the strength of physical coupling and the model complexity of various classic FSI problems.

➤ Partitioned/ Staggered approach:

The governing equations of the fluid flow problem and the structural problem are solved independently and different codes are used for each of the problems. These equations can be solved explicitly or implicitly. The load vectors or information are exchanged in an outer iteration loop at synchronization points only. This method is recommended for multi-physics problem with weak physics coupling. The advantage of the partitioned/staggered approach is that each physical field can be treated by discretization techniques and solution algorithms that are known to perform well for the standalone subsystem. In addition, the partitioned/staggered approach allows independent modeling of sub problems.

CFD package – FLUENT has been widely used in simulating FSI problem with its flexibility of using User-Defined-Function (UDF) and dynamic mesh model. A CFD dynamic simulation on the performance of a non-return valve of a car hydraulic control unit has been conducted by Valdes [116] using FLUENT with UDF and deforming mesh model. The velocity and displacement of the moving valve are calculated by means of a UDF that evaluates the forces over the moving valve. In 2005, Vierendeels [111] performed an analysis on the FSI algorithm for rigid body motion with the help of

FLUENT. The objective was to study the stabilization of the FSI algorithm on explicit partitioned/ staggered coupling and implicit partitioned/ staggered coupling.

Partitioned/ staggered coupling is normally used for FSI problem with weak physical coupling such as valve dynamics, engine head thermal stress analysis, fuel tank sloshing and etc. However, for FSI problems such as heart valve analysis, blood flow in artery veins, aircraft/ fan blade flutter and etc, a monolithic approach with strong physical coupling is needed. One of the worth mentioning work on monolithic approaches was conducted by Heil [112] in 2004. He developed an efficient solver for the fully coupled solution of large displacement FSI problems. The preconditioning technique adapted allows a rapid iterative solution of the linear systems that arise in the fully coupled (monolithic) solution of steady and unsteady large-displacement FSI problems with Newton's method. It also demonstrated the importance of consistent stabilization for the accurate simulation of FSI problems. In 2010, Barker et al. also developed a monolithically coupled FSI algorithm in solving FSI problem on blood flow in arteries [114]. Their results showed that their algorithm is robust and scalable for variety of physical parameters, scaling to hundreds of processors and millions of unknowns.

In recent years, research in the area of computational FSI has seen tremendous progress due to the maturity and the integration in the field of CFD and computational structural dynamics. The development of multi-physics solver such as COMSOL and ANSYS Workbench; the integration between CFD package and computational structural dynamics software such as FLUENT with ANSYS Finite Element Analysis (FEA), FLUENT with Abaqus FEA, StarCCM with Abaqus FEA and etc has made FSI modeling easily accessible to researchers.

Figure 2-17 FSI Coupling Spectrum [113]

2.2.5 Flow Visualization

Flow visualization is the art and science of obtaining a clear image of a physical flow field and the ability to capture it by means of photograph, a sketch or on other video storage devices for display or further processing [117]. The objective of conducting flow visualization is to have an understanding of the flow field. The foremost interest of flow fields are velocity and regions of flow separation. Fields of density and other thermodynamic variables are additional interest in compressible flows. However, most fluids like air, water and etc, are transparent, and therefore their flow patterns are invisible to humans' naked eyes when they are in motion. Thus, special methods need to be used to make these flow patterns visible. Flow visualization is important as one can derive quantitative data from the flow pictures obtained. By observing a flow pattern, which may be either steady or unsteady, one can have a good understanding of the physical flow field. Flow visualization techniques provides information on the complete flow field without physically interfering with the fluid flow [118]. Some common

visualization techniques include (i) tracer techniques, (ii) surface flow visualization techniques, (iii) infrared thermography, (iv) laser interferometry and holography, (v) tomography, (vi) large field flow visualization, and (viii) other optical techniques [119]. Flow visualization aims at the discovery, description and parametric investigation of new flow phenomena and at the educational presentation of established ones [117].

It is difficult to pinpoint who first pioneered the study on flow visualization, however, Leonardo da Vinci was generally considered as the father of flow visualization [120]. He was the first to sketch or describe some of the typical phenomena such as formation of eddies at abrupt expansions and in the wakes; the profiles of free jets; the velocity distribution in a vortex; the propagation, reflection and interference of waves; and the hydraulic jump. Sketches on transition to turbulence in pipes by means of a single streak line; sketches of vortex bursting in water-spouts; sketches of wing tip vortex; sketches of the vortex street in the wake of a circular cylinder were reproduced by Lugt, Lanchester and Benard in eighteenth and nineteenth century [117]. Subsequently, photography was introduced into flow visualization in 1879. The first photograph of fluid mechanics with some relevance to flow visualization is the photograph of water jet decaying into droplets taken by Rayleigh [117]. Mach and Salcher also used photography to visualize the shock wave and wake of a fast bullet [117]. Until today, high speed camera or high speed photography has become one of the most important tools for the recording of most of the flow visualization results. Some commercial digital single-lens reflex camera can produce up to 8 frames per second while some dedicated high speed camera for flow visualization can produces up to a 1 Mhz rate [119]. Over the past few decades, high speed videography has also improved rapidly and thus has served as an important tool for flow visualization.

In recent years, fluid mechanics has undergone rapid changes due to the advent and advancement of computer. With the latest computer technology, mathematical modeling or CFD simulation has been used extensively for solving partial differential equation or Navier-Stokes equation in fluid mechanics problems. However, computer simulations may not accurately replicate the actual problem in consideration, so the computed results need to be verified by comparison with direct observations of fluid behavior by using flow visualization [119]. Thus, flow visualization remains as one of the most essential tools in fluid mechanics.

In this section, reviews on mathematical modeling, CFD simulation, experimental investigations, FSI and flow visualization have been conducted. The need for pump miniaturization and the challenges on miniaturization will be discussed in the next two sections.

2.3 The Need for Miniaturization

In recent years, miniature fluidic pumps have been in great demand in the fields of pharmaceutical devices, cardiac assist devices, liquid-cooling-devices, liquid dispensing systems, fuel delivery systems for miniature fuel cells and etc.

In pharmaceutical applications, patients who are suffering from diabetes need insulin injections for every 3 to 8 hours daily. It is both a troublesome and tedious for patients. Infusion pumps have been developed to overcome this problem. Infusion pumps shown in Figure 2-10 are normally seen in hospitals, but it is not portable. For diabetic patients, a portable, smaller size and reliable infusion pump is needed so that they can have a normal life with minimal disruption.

An illustration of a portable infusion pump that delivers insulin to the body through a thin plastic tube is shown in Figure 2-18. Even though the technology of insulin infusion pumps is quite established today, researchers are still putting their efforts in making the device smaller and more portable. The goal is to develop an infusion pump which is implantable or with embedded computer control [3, 4]. With the advent of micro fabrication, it is possible that in the near future, more mini or microscale infusion pump can be developed.

Figure 2-18 Illustration of an Insulin Pump [121]

As mentioned above, pumps that are used as cardiac assist device are still relatively large in size [62, 66, 122] which make it difficult to implant into patients of a smaller body built. Currently, ventricular assist devices, especially the pulsatile blood pumps, are still limited to patients of a larger body frame [66]. This is because pulsatile pumps require a motion converter to obtain pulsation motion such as a roller screw, a solenoid-beam spring, a helical cam or a hydraulic system [74]. In 1999, Daiki reported that their implantable motor driven ventricular assist device has failed the animal test as the pump is too big to be implanted in either the abdomen or the thoracic cavity [122]. Such

devices are not suitable for some men or women with smaller body frame, kids or even neonates, thus miniaturization of such device is needed in order to benefit a wider range of the patients group. Besides that, a smaller artificial heart or VAD means that it is less invasive to patients and hence reduces the risk of infections [123]. In addition, a miniature pump is lighter, and hence reduces the burden of the patient who has undergone the implantation.

Due to the increasing demand on compact size and portable electronic device, compact and high power density microprocessor has been introduced. However, high heat generation of the compact and high power microprocessor is one of its disadvantages. Thus, liquid cooling device has become an interesting research topic in removing the heat generated by these compact and high power density microprocessor.

In 2001, Darabi [76] developed an electrohydrodynamic polarization micropump for electronic cooling purpose. However, dryout was observed during high heat flux levels due to the evaporation rate being higher than the rate of liquid supplied by the electrohydrodynamic pump. Similarly in 2002, Jiang [124] demonstrated an electro-osmotic microchannel cooling system for VLSI circuits. The models predict that the cooling system performance can be improved by increasing the pump flow rate. Thus, the development of a miniature pump or micropump with higher flow rate is needed in order to improve the performance of the liquid cooling device.

2.4 Challenges on Miniaturization

Miniaturization is not an easy task. In recent years, many researchers face problems in the miniaturization of existing pumps.

First of all, the main problem encountered is the method of fabrication or assembly and the strict accuracy required in fabricating the devices and the mechanism [73]. While devices are getting smaller, conventional machining method can no longer be applied. This can be overcome by using the micromachining technology and a wide range of micro devices has been realized. These micromachining techniques include bulk micromachining, surface micromachining and LIGA (Lithography, Electroplating and Molding) technology [39]. Aside from the fabrication method, the size of the fasteners that are commercially available restricts the miniaturization of the device. If the devices are at the micro-scale level, fasteners that are generally used in conventional machine can no longer be used. This poses a great challenge to researchers working on the design of miniature pumps.

All positive displacement pumps or dynamic pumps require a mechanical motor, which generally converts electric energy into mechanical work. Thus, miniaturization of the unit is always restricted by the size of the electric motor [125]. Aside from the electric motor, the positive displacement pump, for example the pulsatile TAHs or VADs require a motion converter to obtain pulsation motion such as a roller screw, a solenoid-beam spring, a helical cam or a hydraulic system [74]. The use of a motion converter in pulsatile pump is an obstacle to miniaturization. Kazuyoshi [74] tried to overcome this problem by using a linear oscillatory actuator (LOA) to drive his pulsatile pump. Some researchers tried to avoid using mechanical actuator in their miniature pumps or micropumps design in order to overcome this problem, and thus, non-mechanical micropumps which mentioned in the previous section have been developed. These non-mechanical pumps include electrohydrodynamic pumps, electrokinetic pumps, phase transfer pumps, electro wetting pumps, electrochemical pumps, magneto hydrodynamic

pumps and etc [39]. However, the details of these non-mechanical pumps will not be further discussed in this report.

As mentioned in the previous section, at very small scales, viscous forces are significant, and result in large fluidic impedance [27, 39]. This effect is undesirable when pumping some bio-fluid such as blood because red blood cells may be damaged when subjected to high shear level larger than 10^5 s^{-1} [27]. Thus, for bio-fluid or blood pumping applications, researchers should try to avoid scaling down their devices to the extent that viscous effect becomes dominant.

For the past few decades, miniature pumps have gaining their popularity in many industries such as applications in pharmaceutical, medical, and MEMS. Miniature pumps have the advantages of small in size, mobility and precision control of flow rate. Thus, development of a miniature pump is required to serve these applications. However, miniaturization process is not an easy task. This literature review gives an overview of the market's needs as well as some concepts and guidelines in developing the miniature pump.

In this chapter, a review on the classification of pumps, its working principle and applications are presented. A brief review on the mathematical and CFD simulation of reciprocating pumps has been carried out. A review on fluid-structure-interaction (FSI) and flow visualization have also been conducted. There is a need to miniaturize devices to serve the pharmaceutical and medical industries and the challenges facing many researchers in the process of miniaturization are also discussed. These serve as a motivation for the current project where the aim is to develop a miniature pump with better performance to address the issues raised. In addition, this also helps to identify the field of application of the miniature pump and draws the specifications and requirements of the miniature pump which will be further elaborated in chapter 3.

CHAPTER 3

DESIGN OF A MINIATURE PUMP

The design and development process of a miniature pump is a long term continuous process. Before the actual design was adopted, several conceptual designs have been examined to reveal the potential of the miniature pump in achieving an improved performance over existing pumps. Once the actual design was decided, efforts were made to improve its performance. Over the past few years, several designs have emerged, namely (i) the U-tube peristaltic pump, (ii) twin roller peristaltic pump, (iii) cam profile peristaltic pump and etc. Although all of them were discarded after a preliminary experimental study, they provided invaluable inputs in recognizing the potential performance of the final miniature pump design, namely the reciprocating ball pump (RBP).

In this chapter, the evaluation of the design objectives will be presented. A brief introduction of the miniature pump – RBP, followed by its working principle will be presented. In addition, evolution of the RBP and its design will also be discussed.

3.1 Evaluation of Design Requirements

Before the creation of any new designs, it is important that the design requirements are defined clearly. By having a clear perception of what the device is to achieve, the designer can then propose appropriate designs during the design and development process. To determine the desired design requirement, an evaluation of the existing

pump designs presented in section 2.1 is conducted, where the pros and cons of each design are carefully analyzed. The finalized design criteria are listed as follows.

i. Targeted Application and Flow Rate

From the literature review presented in Chapter 2, it is noted that there is an urgent need for portable pumps for enhanced mobility in medical or pharmaceutical applications such as infusion system, drug delivery or even ventricular assist device. Thus, the miniature pump will be targeted for these applications. In addition, it is noted that most miniature devices mentioned in the literature review above are operating with a flow rate of 0.5 - 3 L/min [9, 11]. For proof of concept purpose, the miniature pump has to be reasonably small that does not require special fabrication method and is able to deliver a flow rate of about 3 L/min.

ii. Simplicity in Working Mechanism

A machine with many moving components or complicated motion might fail due to part failure or jammed motion. Thus, simplicity is one of the design requirements of the proposed miniature pump. Simplicity in the operating mechanism such as a minimal number of moving parts and simple motion characteristics means that the pump is less likely to fail during operation. In addition, with lesser moving components and simpler motion, less maintenance will be required as well.

iii. Design Simplicity

Highly complex geometries and shapes bring about difficulties in manufacturing process especially when the part is very small. As mentioned in section 2.4, one

of the challenges encountered during miniaturization of a pump is the method of fabrication or assembly. Thus, keeping the parts or components geometries as simple as possible can simplify the fabrication process and reduce the difficulty in the miniaturization process.

iv. Small Size and Lightweight

Nowadays, miniature pumps concept is popular among Original Equipment Manufacturers (OEMs) to ensure the competitiveness of their product. Miniature pumps are used in complex devices whereby numerous components are packed within a single device. This implies that there is a size constraint on the design of the pump when it is integrated in a complex device. In field applications, miniature pumps can be applied in medical or pharmaceutical applications. Thus, the size and weight of the pump have to be taken into consideration during the design and development process. The objective is to develop a miniature pump that is less than 100 grams in weight.

v. Minimal Contamination

Design and development of miniature pumps for drug delivery system, pharmaceutical application, or even ventricular assist device and heart pump have been discussed extensively [3, 4, 67-69]. The application of miniature pump in these areas implies that liquid contamination has to be minimized. Thus, contamination free have been selected as one of the design requirements such that the miniature pump is not only applicable to application where contamination is not an issue, but also applicable to pharmaceutical application,

chemical application or drug delivery system where contamination is a key concern.

3.2 Reciprocating Ball Pump (RBP)

The reciprocating ball pump (RBP) is a positive displacement pump which uses the working principle of trapping a fixed volume of fluid and forcing the fluid to move towards the outlet. The pumping unit moves in a reciprocating manner and energy is periodically added to the fluid to generate flow.

Figure 3-1 Schematic Diagram of Reciprocating Ball Pump

Figure 3-1 shows a schematic diagram of the RBP. It consists of (i) an outer casing, (ii) sets of electro-magnetic coils, (iii) an inner tube/ pipe and (iv) the pumping unit. The pumping unit of the RBP is placed inside the inner tube/ pipe. The outer part of the tube/ pipe is covered with several sets of electromagnetic coils. The whole assembly of the inner tube/ pipe is covered by an outer casing. By sequentially activating the electromagnetic coils, the pumping unit can achieve reciprocating motion and perform its unique pumping cycle whereby fluid is pumped during the forward stroke and continues to flow in the forward direction even during the backward stroke.

3.2.1 Pumping Unit of RBP

Figure 3-2 shows CAD 3D model of the pumping unit of the RBP. In its basic form, the pumping unit of the RBP consists of four main components, namely a rear cap, a casing, a front cap and most importantly a ball as illustrated in Figure 3-2. The rear cap, casing and front cap are made of magnetic material or non-magnetic material with permanent magnet embedded inside such that the rear cap, casing and front cap assembly will be attracted by magnetic or electromagnetic force to undergo reciprocating motion. On the other hand, the ball is made of non-magnetic material such as ceramic, rubber, polymers and etc. The ball is housed in the chamber of the casing while the two ends of the casing were secured by the rear cap and the front cap to prevent the ball from escaping from the whole pumping unit assembly. Three supporting legs are used to restrict the motion of the ball such that it can only travel in one direction (axial direction). The rear cap and the front cap are specially designed such that the pumping unit can perform effective pumping action during the forward stroke while allowing fluid to pass through the pumping unit during the backward stroke.

Figure 3-2 Pumping Unit of Reciprocating Ball Pump

3.2.2 Working Principle of RBP

The uniqueness of the RBP lies within the pumping unit of the RBP. By driving the pumping unit in reciprocating motion, the RBP can deliver flow in both forward and backward strokes.

Figure 3-3 Working Principle of Reciprocating Ball Pump

Referring to Figure 3-3, the working principle of the RBP is as follows:

- i. At the bottom dead center (BDC) which indicates the beginning of the forward stroke, the position and orientation of the pumping unit and its ball are shown in Figure 3-3 (i). The ball is pressed onto the front cap due to the previous pumping motion. The RBP is in 'open' condition and fluid continues to flow in the forward direction due to its own inertia through the pumping unit of the 'open' RBP by flowing via the hole on the rear cap, the gap between the rear cap and the

ball, the flow area between the casing and the ball and finally through the slots on the front cap.

- ii. As the pumping unit starts to move in the forward direction, the ball inside the pumping unit will be pressed onto the rear cap as shown in Figure 3-3 (ii) due to its own inertia as well as the pressure acting on it. From this instance onwards, the hole on the rear cap is sealed by the ball and the RBP is in the 'closed' condition. A trapped volume of fluid is formed in front of the pumping unit.
- iii. As the pumping unit moves further forward (Figure 3-3 (iii)), the trapped volume of fluid will be displaced in the forward direction, thus creating flow in the forward direction.
- iv. The fluid continues to be displaced in the forward direction until the pumping unit reaches the end of the forward stroke or top dead center (TDC) as illustrated in Figure 3-3 (iv). After that, the pumping unit starts to move in the backward direction.
- v. When the backward stroke starts, the ball will then be pressed onto the front cap as shown in Figure 3-3 (v) due to its own inertia and the pressure acting on it. The RBP is in 'open' condition and fluid is allowed to flow in the forward direction due to its inertia as there are slots on the front cap of the RBP.
- vi. The fluid continues to flow through the pumping unit through the slots on the uniquely designed front cap (Figure 3-3 (vi)). The fluid will then slowly lost its

inertia as the pumping unit is moving towards the BDC (Figure 3-3 (i)). Subsequently, the pumping cycle will start all over again to generate flow in the forward direction.

For proof-of-concept purpose, the experimental setup uses a permanent ring magnet instead of the electro-magnetic coils to drive the pumping unit of the RBP. The permanent ring magnet is placed outside the pipe and is connected to a slider crank mechanism in order to drive the pumping unit of the RBP inside the pipe in reciprocating motion. The details of the experimental setup for the RBP will be presented in Chapter 6.

3.2.3 Advantages of RBP

Due to the design of the RBP, its simplicity and its unique working principle, the RBP design possesses several advantages over existing pump mechanisms.

First of all, the motion of the pumping unit of RBP is achieved by using magnetic or electromagnetic force. Thus, the pumping unit of the RBP is separated from its actuation mechanism. The contactless design of the RBP with its actuation mechanism achieves contamination free pumping process. Conventional turbomachines such as centrifugal or axial pumps require bearing to hold the rotating shaft in place and concern about potential contamination hazard during operations exist. Conventional reciprocating pumps which utilize check valves to regulate the flow might also be exposed to some risk of contamination at the valves as compared to the RBP.

Secondly, although the RBP is a reciprocating positive displacement pump, no external valve is required to regulate the flow as compared to conventional reciprocating pump.

Conventional reciprocating pump requires at least two check valves, one at the inlet and one at the outlet are needed to ensure that the fluid is pumped out from the chamber through the outlet check valve during its pumping stroke and ensure that the fluid is sucked into the chamber through the inlet check valve during its suction stroke. The uniqueness of the RBP design is that no external valves are required and their function is replaced by the reciprocating ball which is part of the pumping unit. Due to the special design of the slots in the casing and the front cap, it eliminates the use of external valves while ensuring uni-directional flow.

In addition, the unique pumping action of the RBP ensures that flow is generated even during the backward stroke. Comparing with conventional single acting reciprocating pumps where the liquid in the cylinder is discharged only during the forward stroke, the RBP has the advantage of inducing flow during the backward stroke. Figure 3-4 illustrates the additional flow induced by the RBP as compared to conventional single acting reciprocating pump. A higher flow rate is generated by the RBP due to the design of the pumping unit of RBP that allows liquid to flow through the pumping unit in the forward direction during the backward stroke due to its momentum gained during the forward stroke. In contrast, a conventional single acting reciprocating pump, where suction and discharge valves are employed, does not produce any flow during its backward stroke.

Figure 3-4 Flow Rate Comparisons between RBP and Conventional Reciprocating Pump

The RBP also has the advantage that it can operate at high frequencies. In contrast, peristaltic pumps that are widely used in existing miniature and portable pumps cannot operate at high frequency due to constant rubbing between the roller and the elastic tube. Hence, the RBP has a wider range of applications as compared to peristaltic pumps.

3.2.4 Evolution of RBP and Design Variation

The design and development of the miniature pump is a continuous process. The final version of the miniature pump adapted in this project is the reciprocating ball pump (RBP) because there is a ball inside the pumping unit of the RBP. However, the first few versions of this miniature pump which uses the same pumping principle do not have a ball inside its pumping unit.

a. Reciprocating Flap Pump

The first version of this miniature pump uses a flap to regulate flow. Figure 3-5 shows the CAD 3D model of the first version of RBP, namely the reciprocating flap pump.

Figure 3-5 CAD 3D Model of the Pumping Unit of Reciprocating Flap Pump

The pumping unit of reciprocating flap pump consists of two major components; a casing and most importantly the flap assembly which is located at the front of the casing. A hinge was used in the flap assembly such that it will either be in the ‘closed’ or ‘open’ condition as illustrated in Figure 3-6 to perform the unique pumping action similar to the RBP. The working principle of reciprocating flap pump is similar to the RBP as illustrated in Figure 3-3 and the operation sequence is similar to the RBP that is presented in section 3.2.2. When the pump is moving in the forward direction, due to the pressure acting onto the flap, it will be pressed onto the casing. At this instance, the reciprocating flap pump is in the ‘closed’ condition as illustrated in Figure 3-6 (a) and trapped volume is formed in front of the pumping unit. As the pumping unit moves forward, the trapped volume of fluid will be displaced in the forward direction. When the pumping unit starts to move in the backward direction, due to the momentum gained during the forward stroke, fluid continues to flow in the forward direction. The fluid

flow in the forward direction pushes and flips the flap open as illustrated in Figure 3-6 (b). The reciprocating flap pump is in the ‘open’ condition and fluid is allowed to pass through the pumping unit during the backward stroke. Thus, an inherent unidirectional flow is generated like RBP.

Figure 3-6 Schematic Diagram of Reciprocating Flap Pump in (a) ‘Closed’ Condition, (b) ‘Open’ Condition

The working principle of the reciprocating flap pump was tested experimentally.

However, several possible problems were identified:

- i. During the forward stroke, the flap might flip upward as illustrated in Figure 3-7 instead of flipping downward as illustrated in Figure 3-6 (a). Hence, the reciprocating flap pump will be in ‘open’ condition during the forward stroke and thus, cannot trap and displace fluid in the forward direction.

Figure 3-7 Reciprocating Flap Pump with Over Flipped Flap

- ii. Due to wear and tear, the hinge mechanism used in the flap assembly might fail or jam during the operation, thus, causing the pump to malfunction.
- iii. The hinge mechanism requires a minimum clearance for the flap to work. The clearance for the motion may lead to severe leakage flowing from downstream back to the upstream of the pumping unit, thus, reducing the effectiveness of the pump.

b. Reciprocating Disc Pump

In the second version, a disc was used to regulate the flow. Figure 3-8 shows the CAD 3D model of the first version of RBP, namely the reciprocating disc pump. The pumping unit of reciprocating disc pump consists of three main components; a front cap, a rear cap and most importantly the reciprocating disc which is located inside the compartment between the caps. The pumping unit is designed in such a way that when the pumping unit is moving forward, the reciprocating disc is pressed onto the rear cap so that the hole on the rear cap is blocked. Fluid is pumped or displaced forward due to the blockage of the hole and the forward motion of the pumping unit. When the pumping unit is moving backward, the reciprocating disc is pressed onto the front cap and there is no blockage of flow during this motion. Thus, fluid is allowed to pass through the pumping unit during the backward stroke.

Figure 3-8 CAD 3D Model of the Pumping Unit of Reciprocating Disc Pump

The working principle of the reciprocating disc pump is similar to the reciprocating ball pump as illustrated in Figure 3-3. The operation sequence of the reciprocating disc pump is exactly the same as the RBP that is presented in section 3.2.2. When the pump is moving forward, due to the pressure and inertia force acting on the disc, it will then be pressed onto the rear cap which is similar to the situation as illustrated in Figure 3-3 (ii). During this stage, flow is blocked and a trapped volume of fluid is formed in front of the pumping unit. As the pumping unit moves forward, fluid will be displaced from the control volume as illustrated in Figure 3-3 (ii) to (iii). As the reciprocating ball disc pump is moving in the backward direction, the disc is then pressed onto the front cap as illustrated in Figure 3-3 (v). During this stage, fluid is allowed to flow through the opening on the rear cap, the slots on the disc and the front cap. Thus, when the pumping unit is moving in the backward direction, fluid is not trapped and hence, an inherently unidirectional flow like the RBP is generated.

The working principle of the reciprocating disc pump was tested experimentally.

However, several possible problems were identified:

- i. When the valve is pressed onto the rear cap during the forward stroke, leakage will occur if the contact surfaces are not flat. Hence, this will lead to ineffective pumping effect.
- ii. The reciprocating valve might be trapped or jammed inside the compartment between the casing and cap when it is tilted at a certain angle during the operation as shown in Figure 3-9. This will cause the pump to malfunction.

After several iterations, an improved design has been developed to minimize the problems mentioned above, namely the RBP. The RBP design replaces the flap or the reciprocating valve with a ball. With the new design, only line contact is maintained when the ball comes into contact with the casing or cap. Due to the spherical shape of the ball valve, the problem of trapping or jamming inside the compartment between the casing and cap during the operation is eliminated.

Figure 3-9 Reciprocating Valve Trapped/Jammed Inside the Compartment between Casing and Cap

3.3 Double Acting RBP

Unfortunately, when the RBP operates under high pressure condition, the RBP will be in ‘closed’ condition throughout the whole pumping process due to the high pressure force acting on the ball in the backward direction and no net pumping effect is achieved. This renders the RBP ineffective. Hence, the double acting reciprocating pump (DRBP) is introduced to address this issue. The DRBP is similar to the normal RBP. However, it consists of one extra pumping unit which moves in the opposite direction to the other pumping unit during the operation. The schematic diagram of the DRBP is shown in Figure 3-10. The working principle of the DRBP is further illustrated in Figure 3-11.

Figure 3-10 Schematic Diagram of Double Acting Reciprocating Ball Pump

For high pressure operation, to prevent the fluid from flowing in the backward direction during the backward stroke due to the high pressure downstream, the second pumping unit was added to the DRBP design as shown in Figure 3-10. The working principle of the DRBP is as follows:

- i. At the beginning of the forward stroke of pumping unit A, the orientation and the position of the ball of pumping unit A and pumping unit B is illustrated in Figure 3-11(i). The pumping unit A is in the ‘open’ condition whereby the ball of the pumping unit A is pressed onto the front cap while the pumping unit B is in the

‘closed’ condition whereby the ball of the pumping unit B is pressed onto the rear cap.

- ii. When the pumping unit A starts to move in the forward direction, the ball will be pressed onto the rear cap of pumping unit A due to its own inertia and the pressure force acting on it (Figure 3-11(ii)). The ball in pumping unit A seals the hole on the rear cap and the pumping unit A becomes ‘closed’. A trapped volume of fluid is formed in front of pumping unit A. This causes the fluid ahead of pumping unit A to move in the forward direction (Figure 3-11(ii) & (iii)) when the pumping unit A is moving in the forward direction. The fluid behind pumping unit A will also be drawn to fill up the empty space created by the forward motion of the pumping unit A. On the other hand, pumping unit B is moving in the opposite direction with respect to pumping unit A. Due to the pumping action of pumping unit A, fluid is pushed and displaced in the forward direction. The ball inside pumping unit B will then be pushed onto the front cap (Figure 3-11(ii)) and the pumping unit B is in the ‘open’ condition. Due to the slots on the front cap, fluid is allowed to flow through pumping unit B when pumping unit A is performing the effective pumping action.
- iii. Pumping unit A remains in the ‘closed’ condition and continues to move in the forward direction while displacing the trapped volume of fluid in the forward direction. On the other hand, pumping unit B remains in the ‘open’ condition, allowing fluid to flow.

Figure 3-11 Working Principle of Double Acting Reciprocating Ball Pump

- iv. The pumping unit A is still in 'closed' condition upon reaching the end of its forward stroke. It then begins to move in the backward direction while pumping unit B, still in the 'open' condition, begins to move in the forward direction as shown in Figure 3-11(iv).
- v. Pumping unit A starts to move in the backward direction while pumping unit B starts to move in the forward direction. Due to the inertia of the ball inside pumping unit B and the pressure force acting on it, the ball inside pumping unit B will be pressed onto the rear cap of pumping unit B as shown in Figure 3-11(v).

The ball seals the hole on the rear cap of the pumping unit B and it is ‘closed’. As pumping unit B moves in the forward direction, the fluid ahead of it is pushed and displaced in the forward direction. Due to the forward motion of the pumping unit B, the fluid behind the pumping unit B will be sucked to fill up the empty space created by its motion. On the other hand, as pumping unit A moves in the backward direction, the ball inside pumping unit A will be pressed onto the front cap due to its inertia and the suction pressure force acting on it due to the forward motion of the pumping unit B (Figure 3-11(v)). The pumping unit A is ‘open’ and the fluid that is being sucked is allowed to pass through the pumping unit A and effectively fill up the empty space created by the forward motion of pumping unit B.

- vi. Pumping unit A remains in the ‘open’ condition and continues to move in the backward direction while allowing fluid that was pumped to flow through to achieve effective pumping action. On the other hand, the pumping unit B remains in the ‘closed’ condition, displacing the trapped volume of fluid in forward direction until the end of its forward stroke (Figure 3-11(i)).

This chapter describes the design and working principles of the RBP. In the next chapter, theoretical analysis and the mathematical modeling of the reciprocating ball pump (RBP) will be presented.

CHAPTER 4

THEORETICAL STUDY OF RECIPROCATING BALL PUMP

To study the performance of the RBP, it is necessary to determine the pressure and the flow rate generated during the pumping operation. The fundamental changes to these properties can be described by a set of differential equations derived from the motion dynamics of the reciprocating mechanism and the conservation of mass and momentum. In this chapter, the derivation of the reciprocating motion dynamics, the pressure and flow rate generated and the simplified mathematical model of RBP will be presented.

4.1 Derivation of Reciprocating Motion Dynamics

Before one can study the performance of a positive displacement or reciprocating pump, it is essential to understand its motion dynamics [84, 126]. For proof-of-concept purpose, the experimental setup uses a permanent ring magnet instead of the electro-magnetic coils that was shown in section 3.2 to drive the pumping unit of the RBP in reciprocating motion. The permanent ring magnet is placed outside the pipe and is connected to a slider crank mechanism to drive the pumping unit of the RBP inside the pipe in a reciprocating motion. A schematic diagram of the slider crank mechanism is shown in Figure 4-1.

Figure 4-1 Reciprocating Slider Crank Mechanism

According to cosine law,

$$s^2 + r^2 - 2 \cdot r \cdot s \cdot \cos \phi = b^2 \quad (4-1)$$

The distance s is

$$s = r \cdot \cos \phi + \sqrt{r^2 \cos^2 \phi - r^2 + b^2} \quad (4-2)$$

It can then be simplified as

$$s = r \cdot \cos \phi + \sqrt{b^2 - r^2 \sin^2 \phi} \quad (4-3)$$

Where s is the displacement of the slider L with reference to the slider crank mechanism center C ,

r and b are the length of the linkages used for the reciprocating mechanism

Point O is the point when the slider is at its bottom dead center (BDC). The distance between point O and the slider crank mechanism center C is equal to $(b - r)$ when

$\phi = 180^\circ$. Introduce displacement z

$$z = s - (b - r) \quad (4-4)$$

Where z is the displacement travelled by the slider L with reference to point O or BDC of the slider such that the displacement z range from 0 to $2r$ when ϕ varies from 0° to 360° . Hence, displacement z can be expressed as

$$z = r \cos \phi + \sqrt{b^2 - r^2 \sin^2 \phi} - (b - r) \quad (4-5)$$

Let $\phi = \omega t$, where $\omega = 2\pi N/60$, N [rpm] is the revolution per minute of the motor and t [s] is the time.

Equation (4-5) becomes,

$$z = r \cos(\omega t) + \sqrt{b^2 - r^2 \sin^2(\omega t)} - (b - r) \quad (4-6)$$

The velocity of the permanent ring magnet driven by the slider crank mechanism can be obtained by differentiating equation (4-6) with respect to time. Hence,

$$v = \frac{dz}{dt} = -r\omega \sin(\omega t) - \frac{r^2 \omega \sin(\omega t) \cos(\omega t)}{\sqrt{b^2 - r^2 \sin^2(\omega t)}} \quad (4-7)$$

The acceleration of the permanent ring magnet can be found by differentiating equation (4-7) with respect to time. Hence,

$$a = \frac{dv}{dt} = -r\omega^2 \cos(\omega t) - \frac{r^2 \omega^2 \cos^2(\omega t)}{\sqrt{b^2 - r^2 \sin^2(\omega t)}} - \frac{r^4 \omega^2 \sin^2(\omega t) \cos^2(\omega t)}{\sqrt[3]{b^2 - r^2 \sin^2(\omega t)}} + \frac{r^2 \omega^2 \sin^2(\omega t)}{\sqrt{b^2 - r^2 \sin^2(\omega t)}} \quad (4-8)$$

The pumping unit of the RBP is driven by the permanent ring magnet, and thus, the pumping unit of the RBP is assumed to move with the same displacement, velocity and acceleration as the permanent ring magnet. Hence, the displacement, velocity and acceleration of the pumping unit of the RBP can be represented by z , v and a as well.

Figure 4-2, Figure 4-3 and Figure 4-4 show the displacement, velocity and acceleration of the pumping unit of the RBP respectively.

Figure 4-2 Displacement of the Pumping Unit of RBP vs Time

Figure 4-3 Velocity of the Pumping Unit of RBP vs Time

Figure 4-4 Acceleration of the Pumping Unit of RBP vs Time

4.2 Modeling of the RBP

For a positive displacement pump, the pressure and flow rate generated are two important properties that determine its performance. The relationship between the pressure and the flow rate, i.e. the velocity of the fluid can be described by the Navier-Stokes equations and continuity equation. By taking the motion dynamics of the pumping unit of the RBP as boundary condition, the Navier-Stokes' and continuity equations can be solved to determine the pressure and flow rate generated by the RBP. Due to the geometry of the RBP, the Navier-Stokes' equations and the continuity equation in cylindrical coordinates have been adopted.

Navier-Stokes' Equation in Cylindrical Coordinates

r -direction:

$$\begin{aligned} \rho \left(\frac{\partial u_r}{\partial t} + u_r \frac{\partial u_r}{\partial r} + \frac{u_\theta}{r} \frac{\partial u_r}{\partial \theta} + u_z \frac{\partial u_r}{\partial z} - \frac{u_\theta^2}{r} \right) \\ = -\frac{\partial P}{\partial r} + \mu \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u_r}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 u_r}{\partial \theta^2} + \frac{\partial^2 u_r}{\partial z^2} - \frac{u_r}{r^2} - \frac{2}{r^2} \frac{\partial u_\theta}{\partial \theta} \right] \\ + \rho g_r \end{aligned} \quad (4-9)$$

θ -direction:

$$\begin{aligned} \rho \left(\frac{\partial u_\theta}{\partial t} + u_r \frac{\partial u_\theta}{\partial r} + \frac{u_\theta}{r} \frac{\partial u_\theta}{\partial \theta} + u_z \frac{\partial u_\theta}{\partial z} + \frac{u_r u_\theta}{r} \right) \\ = -\frac{1}{r} \frac{\partial P}{\partial \theta} + \mu \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u_\theta}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 u_\theta}{\partial \theta^2} + \frac{\partial^2 u_\theta}{\partial z^2} + \frac{2}{r^2} \frac{\partial u_r}{\partial \theta} - \frac{u_\theta}{r^2} \right] \\ + \rho g_\theta \end{aligned} \quad (4-10)$$

z-direction

$$\begin{aligned} \rho \left(\frac{\partial u_z}{\partial t} + u_r \frac{\partial u_z}{\partial r} + \frac{u_\theta}{r} \frac{\partial u_z}{\partial \theta} + u_z \frac{\partial u_z}{\partial z} \right) \\ = -\frac{\partial P}{\partial z} + \mu \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u_z}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 u_z}{\partial \theta^2} + \frac{\partial^2 u_z}{\partial z^2} \right] + \rho g_z \end{aligned} \quad (4-11)$$

Due to axis-symmetric boundary condition, the following assumptions are applied:

$$u_\theta = 0 \quad (4-12)$$

$$\frac{\partial}{\partial \theta}(f) = 0 \quad (4-13)$$

$$\frac{\partial^2}{\partial \theta^2}(f) = 0 \quad (4-14)$$

Where f is an arbitrary function.

Under these assumptions, the Navier-Stokes' equation in the θ -direction can be neglected and the Navier-Stokes' equation can be reduced to:

r -direction:

$$\rho \left(\frac{\partial u_r}{\partial t} + u_r \frac{\partial u_r}{\partial r} + u_z \frac{\partial u_r}{\partial z} - \frac{u_\theta^2}{r} \right) = -\frac{\partial P}{\partial r} + \mu \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u_r}{\partial r} \right) + \frac{\partial^2 u_r}{\partial z^2} - \frac{u_r}{r^2} \right] \quad (4-15)$$

z-direction

$$\rho \left(\frac{\partial u_z}{\partial t} + u_r \frac{\partial u_z}{\partial r} + u_z \frac{\partial u_z}{\partial z} \right) = -\frac{\partial P}{\partial z} + \mu \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u_z}{\partial r} \right) + \frac{\partial^2 u_z}{\partial z^2} \right] \quad (4-16)$$

And the continuity equation for an incompressible fluid can be written as:

$$\frac{1}{r} \frac{\partial}{\partial r} (r \cdot u_r) + \frac{1}{r} \frac{\partial u_\theta}{\partial \theta} + \frac{\partial u_z}{\partial z} = 0 \quad (4-17)$$

Or

$$\frac{1}{r}u_r + \frac{\partial u_r}{\partial r} + \frac{1}{r}\frac{\partial u_\theta}{\partial \theta} + \frac{\partial u_z}{\partial z} = 0 \quad (4-18)$$

Based on the assumption made in equations (4-14) and (4-15), the continuity equation can be further reduced to

$$\frac{1}{r}\frac{\partial}{\partial r}(r \cdot u_r) + \frac{\partial u_z}{\partial z} = 0 \quad (4-19)$$

Or

$$\frac{1}{r}u_r + \frac{\partial u_r}{\partial r} + \frac{\partial u_z}{\partial z} = 0 \quad (4-20)$$

Rearranging equation (4-20), it can be written as

$$\frac{\partial u_z}{\partial z} = -\frac{1}{r}u_r - \frac{\partial u_r}{\partial r} \quad (4-21)$$

Equation (4-21) is differentiated with respect to z and this yields

$$\frac{\partial^2 u_z}{\partial z^2} = -\frac{1}{r}\frac{\partial u_r}{\partial z} - \frac{\partial^2 u_r}{\partial r \partial z} \quad (4-22)$$

Equation (4-16) consists of terms in $\frac{\partial u_r}{\partial z}$ and $\frac{\partial^2 u_r}{\partial r \partial z}$. These two terms can be replaced by equation (4-21) and equation (4-22) and hence equation (4-16) becomes

$$\begin{aligned} \rho \left[\frac{\partial u_z}{\partial t} + u_r \frac{\partial u_z}{\partial r} + u_z \left(-\frac{1}{r}u_r - \frac{\partial u_r}{\partial r} \right) \right] \\ = -\frac{\partial P}{\partial z} + \mu \left(\frac{\partial^2 u_z}{\partial r^2} + \frac{1}{r}\frac{\partial u_z}{\partial r} - \frac{1}{r}\frac{\partial u_r}{\partial z} - \frac{\partial^2 u_r}{\partial r \partial z} \right) \end{aligned} \quad (4-23)$$

To further simplify the problem, 1D flow assumption is made assuming u_r is small and can be ignored. Based on this assumption, the Navier-Stokes' equation in the r -direction,

which is equation (4-15) can be neglected. Then, equation (4-16) can be further reduced to

$$\rho \frac{\partial u_z}{\partial t} = -\frac{\partial P}{\partial z} + \mu \left(\frac{\partial^2 u_z}{\partial r^2} + \frac{1}{r} \frac{\partial u_z}{\partial r} \right) \quad (4-24)$$

These are partial differential equations and the unknowns in the equations can only be solved using numerical techniques. There are several CFD codes or software that are commercially available to solve these partial differential equation by taking the motion of the pumping unit as boundary condition. These CFD codes or software include ANSYS FLUENT, ANYSYS CFX, COMSOL, STAR-CCM+, OpenFOAM and etc. However, these CFD codes or software are not user friendly for conducting parametric studies because of the laborious procedures involved in domain setup, mesh generation and grid independency test. Thus, instead of using CFD codes or software for parametric studies, a simple mathematical model has been developed to provide preliminary estimation of the performance of the RBP with reasonable accuracy that facilitates the parametric studies. However, CFD studies were also conducted and will be presented in Chapter 5 so that a better understanding on the flow field and pressure field generated by the RBP can be obtained.

4.3 A Simplified Mathematical Model

The purpose of developing this simple mathematical model is to facilitate parametric study of the RBP while capturing the essential qualitative features of the pumping motion. Figure 4-5 shows the components and variables in this model. The model consists of 5 regions, namely

- I. Rigid tube/pipe region located at the upstream of the pumping unit

- II. Pumping region located at the upstream of the pumping unit
- III. Pumping Unit
- IV. Pumping region located at the downstream of the pumping unit
- V. Rigid tube/pipe region located at the downstream of the pumping unit

The constants or variables in this model are shown in Table 4-1.

P_A	Constant pressure upstream at a distance away from the pumping unit of the RBP [Pa]
P_{up} P_{down}	Transient pressure in the pumping region II located upstream of the RBP [Pa] Transient pressure in the pumping region IV located downstream of the RBP [Pa]
P_B	Constant pressure downstream at a distance away from the pumping unit of the RBP [Pa]
V_{o1}	Initial volume in the pumping region II located upstream of the RBP [m^3]
V_{o2}	Initial volume in the pumping region IV located downstream of the RBP [m^3]
Q_1	Volume flow rate flowing into pumping region II [m^3/s]
Q_{pu}	Volume flow rate flowing out from pumping region II or flowing into pumping region IV via the opening on the pumping unit [m^3/s]
Q_2	Volume flow rate flowing out from pumping region IV [m^3/s]
$Z_p(t)$	Displacement of the pumping unit [m]

Table 4-1 Constants and Variables in the Simple Model

Figure 4-5 Components and Variables in the Simple Model

4.3.1 Assumptions

To have a simplified model, certain assumptions were made. They are:

i. Fluid compressibility in the pumping regions

Fluid compressibility was taken into consideration in the pumping regions (region II & IV) to estimate the transient pressure variation during the pumping operation. When the pumping unit moves in the forward direction (positive z -direction), the volume in pumping region II starts to expand while the volume in pumping region IV starts to compress. Due to the expansion of the volume in pumping region II, suction pressure will be generated and fluid will be drawn into the pumping region II. Conversely, due to the compression of the volume in pumping region IV, a compression pressure will be generated and fluid will be pumped from the pumping region IV. Within region II and IV, the pressures are assumed to be uniform in the respective regions.

ii. One-dimensional pressure driven unsteady flow in the rigid tubes

One-dimensional unsteady flow is assumed in the rigid tubes in region I and V. The flow within the rigid tubes is assumed to be pressure driven flow. In rigid pipe region I, the pressure difference between P_A and P_{up} drives the fluid in either the positive or the negative z direction. Similarly, in rigid pipe region V, the flow will be generated due to the pressure difference between P_{down} and P_B .

iii. Replacing viscous terms with frictional losses term

Equation (4-24) is a non-homogeneous partial differential equation whereby u_z is a function of both time, t , and radial coordinate, r , or $u_z = f(t, r)$. In order to simplify the partial differential equation into an ordinary differential equation,

assumption has been made by representing the viscous terms by frictional losses term. By doing this, u_z is no longer dependent on r , but depends on time, t only, or $u_z = f(t)$.

iv. Simplification of the Pumping Unit's Geometry

The geometry of the pumping unit of the RBP is simple and easy to fabricate. However, it is relatively complicated when trying to model it in this simplified model. Thus, instead of using the actual geometry of the pumping unit of the RBP in this model, a simplified geometry that imitates the pumping action of the pumping unit of the RBP is assumed. Figure 4-6 illustrates the geometry of the simplified pumping unit of the RBP which is modeled as a moving piston with a variable opening that replicates the fully 'closed' RBP, partially 'open' RBP and fully 'open' RBP. The diameter of the variable opening, $D_{pu}(t)$ or the area of the variable opening, $A_{pu}(t)$, is determined based on the relative distance between the ball inside the pumping unit and the rear cap of the pumping unit. The derivation of the diameter of the variable opening, $D_{pu}(t)$ or the area of the variable opening, $A_{pu}(t)$ is discussed in section 4.3.3.

Figure 4-6 Simplification on the Geometry of the Pumping Unit in the Simple Model

4.3.2 Derivation of the Simplified Mathematical Model

Based on the assumptions made in section 4.3.1, pumping regions II and IV are the regions where fluid compressibility is taken into consideration. Due to the motion of the pumping unit, these regions are subjected to expansion or compression. The relationship between the change in pressure with respect to time and the change in volume inside the pumping region can be expressed by equation (4-25) below [82, 85, 127].

$$A_p \dot{Z}_p - Q = \frac{V_o - A_p Z_p}{B} \frac{dP}{dt} \quad (4-25)$$

Or the change in pressure with respect to time can be expressed as:

$$\frac{dP}{dt} = \left(\frac{B}{V_o - A_p Z_p} \right) (A_p \dot{Z}_p - Q) \quad (4-26)$$

where A_p is the cross sectional area of the RBP or piston

Q is the net outflow from the pumping region

$\dot{Z}_p$ is the velocity of the pumping unit of the RBP or piston

Z_p is the displacement of the pumping unit of the RBP or piston

V_o is the initial volume of pumping region II or IV

B is the bulk modulus of the fluid

P is the pressure within the pumping region II or IV

Figure 4-7 Pumping Regions in the Simple Model

Referring to Figure 4-7, when the pumping unit is moving in the positive z -direction, the volume in pumping region II starts to expand. Fluid is drawn into this region with flow rate Q_1 to fill up the empty space created by the motion of the pumping unit. Due to the pressure difference between the pumping regions II and IV, flow might flow in or out from region II, through the pumping unit and flow out or into the region IV with flow rate Q_{pu} as illustrated in Figure 4-7. Thus, the pressure change in region II can be expressed as:

$$\frac{dP_{up}}{dt} = \left(\frac{B}{V_{o1} + A_p Z} \right) (-A_p \dot{Z} + Q_1 - Q_{pu}) \quad (4-27)$$

On the other hand, from Figure 4-7, when the pumping unit is moving in the positive z -direction, the volume in pumping region IV starts to decrease. Fluid is pumped out from

region IV with flow rate Q_2 due to the motion of the pumping unit. Thus, the pressure change in pumping region IV can be expressed as:

$$\frac{dP_{\text{down}}}{dt} = \left(\frac{B}{V_{o2} - A_p Z} \right) (A_p \dot{Z} + Q_{\text{pu}} - Q_2) \quad (4-28)$$

For positive displacement pumps, it should be noted that they do not function as turbo-machines. Instead they produce fluid flow and the resistance to this flow, produced by the hydraulic system, is what determines the pressure [37, 128]. Thus, P_{up} and P_{down} , which are dependent on the motion of the pumping unit related by equations (4-27) and (4-28) are also dependent on the resistance to the flow inside the rigid tube/pipe regions I and V as illustrated in Figure 4-5. The resistance to the flow can be due to viscous effect, frictional losses and minor losses inside the tube/pipe. Based on the assumption made in section 4.3.1, viscous terms are replaced with frictional losses term. Thus, equation (4-24) becomes

$$\rho \frac{\partial u_z}{\partial t} = -\frac{\partial P}{\partial z} - \frac{1}{2} \frac{f}{D} \rho u_z^2 \quad (4-29)$$

To express equation (4-29) in terms of volume flow rate, Q , the velocity in the z -direction, u_z , is substituted with

$$u_z = \frac{Q}{A_c} \quad (4-30)$$

Thus, equation (4-30) becomes

$$\frac{\partial Q}{\partial t} = -\frac{A_c}{\rho} \frac{\partial P}{\partial z} - \frac{f}{2DA_c} Q|Q| \quad (4-31)$$

Where Q is the volumetric flow rate

A_c is the cross sectional area of the tube/pipe

f is the frictional factor of tube/pipe

D is the diameter of the tube/pipe

From Figure 4-5, for rigid tube/pipe region I, equation (4-31) can be written as

$$\frac{\partial Q_1}{\partial t} = -\frac{A_c}{\rho} \frac{(P_{up} - P_A)}{(z_1 - z_o)} - \frac{f}{2DA_c} Q_1 |Q_1| \quad (4-32)$$

On the other hand, for rigid tube/pipe region V, equation (4-31) can be written as

$$\frac{\partial Q_2}{\partial t} = -\frac{A_c}{\rho} \frac{(P_{down} - P_2)}{(z_3 - z_2)} - \frac{f}{2DA_c} Q_2 |Q_2| \quad (4-33)$$

To solve for P_{up} , P_{down} , Q_1 and Q_2 , equations (4-27), (4-28), (4-32) and (4-33) must be solved simultaneously with Q_{pu} . Similarly, Q_{pu} can be found by using the unsteady momentum equation (4-31), which can be written as

$$\frac{\partial Q_{pu}}{\partial t} = -\frac{A_{pu}}{\rho} \frac{(P_{down} - P_{up})}{L_{pu}} - \frac{f_{pu}}{2D_{pu}A_{pu}} Q_{pu} |Q_{pu}| \quad (4-34)$$

Where A_{pu} is area of the variable opening of the moving piston in the simple model

L_c is the length of the pumping unit

f_{pu} is the frictional factor within the pumping unit

D_{pu} is the diameter of the variable opening of the moving piston in the simple model

4.3.3 Derivation of the Area or Diameter of the Variable Opening of the Moving Piston

To determine the flow rate, Q_{pu} , it is necessary to determine the area, A_{pu} or the diameter, D_{pu} , of the opening on the moving piston in the simple model. They are determined by the relative distance between the ball and the rear cap of the pumping unit of the RBP. Figure 4-8 shows the 3D CAD drawing and 2D schematic diagram that indicate the cross sectional area where the fluid is flowing through the pumping unit of the RBP during operation. By determining the cross sectional area for the fluid flow through the pumping unit of the RBP, it is possible to determine the equivalent area, A_{pu} , or diameter, D_{pu} , of the variable opening on the moving piston in the simple model.

Figure 4-8 3D CAD Drawing and 2D Schematic Diagram Showing the Cross Sectional Area for Fluid Flow through the RBP

Figure 4-9 Geometrical Relationship in Determining the Fluid Flow Area

Figure 4-9 shows the geometrical relationship between the rear cap and the ball inside the pumping unit of the RBP. $R_{\text{rear cap}}$ is the radius of the hole on the rear cap, R_{ball} is the radius of the ball, L is the relative distance between the rear cap and the center of the ball and S is the distance from the hole on the rear cap to the center of the ball. Given the radius of the hole on the rear cap and the radius of the ball, by knowing the relative displacement between the rear cap and the center of the ball, L , the distance, S , can be determined by:

$$S = \sqrt{R_{\text{rear cap}}^2 + L^2} \quad (4-35)$$

Subsequently, R_1 can be expressed as

$$R_1 = R_{\text{ball}} \cdot \frac{R_{\text{rear cap}}}{S} \quad (4-36)$$

Finally, by obtaining S and R_1 , the cross sectional area for the fluid flow through the RBP can be calculated by subtracting the surface area of the bigger cone with the surface area of the smaller cone as shown in Figure 4-9. The cross sectional area for fluid flow is assumed to be equal to the area of the variable opening on the moving piston, A_{pu} , can be written as:

$$A_{pu} = \pi \cdot R_{\text{rear cap}} \cdot S - \pi \cdot R_1 \cdot R_{\text{ball}} \quad (4-37)$$

While the diameter of the opening on the moving piston can be expressed as:

$$D_{pu} = \sqrt{\frac{4A_{pu}}{\pi}} \quad (4-38)$$

The relative distance between the rear cap and the center of the ball, L , can be determined by the motion of the pumping unit and the drag force acting onto the ball, whereby

$$F_{\text{drag}} = \frac{1}{2} C_d \rho A_{\text{ball}} \left(\frac{Q_{pu}}{A_{pu}} \right)^2 \quad (4-39)$$

Where ρ is the density of ball

A_{ball} is the projected area of ball

C_d is the drag coefficient of ball where it is a function of Reynolds number as shown in Figure 4-10.

After obtaining the drag force acting on the ball, the velocity at time, t , can be obtained by solving the differential equation below

$$\frac{dv}{dt} = \frac{F_{\text{drag}}}{m_{\text{ball}}} \quad (4-40)$$

where m_{ball} is the mass of the ball.

The new relative distance between the centre of the ball and the rear cap of the pumping unit can then be calculated using the equation below

$$L_{t+\Delta t} = L_t + v_t \cdot \Delta t \quad (4-41)$$

Where $L_{t+\Delta t}$ is the position of the ball at the next time step and L_t is the position of the ball at the current time step.

Figure 4-10 Drag Coefficient As a Function of Reynolds Number For a Smooth Circular Cylinder And a Smooth Sphere [58]

This mathematical model is programmed and written in FORTRAN. Six ordinary differential equations (ODE), namely equations (4-27), (4-28), (4-32), (4-33), (4-34) and (4-40) were solved simultaneously using Runge-Kutta 4th order method to obtain the theoretical flow rate and pressure that indicates the performance of the RBP.

After obtaining the theoretical instantaneous flow rate, the average flow rate per cycle can be found by using the equation below:

$$Q_{avg} = \frac{A_c \int_0^T u_z(t) dt}{T} \quad (4-42)$$

where Q_{avg} is the average flow rate (m^3/s)

u_z is the fluid velocity in axial direction (m/s)

A_c is the cross sectional area of the pipe (m^2)

T is the time taken per period or cycle (sec)

As mentioned in the section 3.2.2, the RBP generates flow in the forward direction during its backward stroke due to the inertia of the fluid. Thus, a higher flow rate is produced by the RBP as compared to a conventional single acting reciprocating pump where the liquid in the cylinder is discharged only during its forward stroke. In order to quantify the flow rate generated by the RBP, the volumetric efficiency, ϕ was introduced, whereby

$$\phi = \left(\frac{Q_{avg}}{V_{swept}/T} \right) \times 100\% \quad (4-43)$$

where ϕ is the volumetric efficiency (%)

Q_{avg} is the average flow rate (m^3/s)

V_{swept} is the swept volume of a single acting reciprocating pump with the same diameter & stroke length with RBP (m^3)

T is the time taken per period or cycle (sec)

In addition, parametric study can be conducted by changing the dimensions or specifications of the RBP or the pumping unit of the RBP, the rotational speed of the slider crank mechanism and inlet or outlet condition.

In the next chapter, an introduction to Computational Fluid Dynamics (CFD) will be presented. The modeling of RBP by using CFD with dynamic mesh model coupled with fluid-structure-interaction (FSI) will be discussed. The details of the dynamic mesh model and grid independency for the fluid flow problem of the RBP will also be presented prior to obtaining the solution for the problem using commercial CFD software (FLUENT).

CHAPTER 5

CFD SIMULATION OF RECIPROCATING BALL PUMP

The mathematical modeling presented in the previous chapter was simplified in such a way that it facilitates the parametric study of the RBP. It provides a preliminary estimation of the performance of the RBP by making certain assumptions such as fluid compressibility, one dimensional unsteady flow, replacing viscous term with frictional losses term to simplify the Navier-Stokes equation into a single order ODE. In addition, simplification of RBP geometry as a moving piston with variable opening was also made. To evaluate the performance of the pump by taking the viscous losses into consideration and to study the flow field and pressure field generated by the RBP, it is required to solve the non-linear PDE Navier-Stokes' equations for the velocity and the pressure distribution across the pump. However, it is almost impossible to solve the Navier-Stokes' equation analytically. Thus, computational fluid dynamics (CFD) is used to solve these governing equations numerically.

There are many CFD codes or software available in the market. These softwares include ANSYS FLUENT, ANSYS CFX, COMSOL, STAR-CCM+, OpenFOAM, FIDAP, POLYFLOW and etc. In this project, ANSYS FLUENT is used to simulate the flow and evaluate the performance of the RBP. Before the actual simulation can be carried out, pre-processing of the geometry is needed. The pre-processor, Gambit is used to create

the geometry, generate mesh, examine mesh quality, assign boundary zone and etc. The mesh file will then be imported to FLUENT to run the simulation.

For the calculation of the flow field, the commercial CFD code: FLUENT 13.0.0 (ANSYS, Inc) was used to solve the conservation equation of mass and momentum by means of a finite volume approach. FLUENT uses a control-volume-based technique to convert the governing equations into algebraic equations that can be solved numerically.

5.1 Model Description

To replicate the motion of the RBP, a simplified 2D-axis-symmetric model domain that is shown in Figure 5-1 has been created. The 2D-axis-symmetric model consists of an open-ended pipe with the pumping unit and the ball inside the pipe. It is impossible to represent the three supporting legs of the casing (see Figure 3-2) in 2D-axis-symmetric model, thus the three supporting legs have been removed. However, the volume occupied by the three supporting legs are calculated and added to the wall thickness of the pumping unit such that the cross sectional area of the casing remains the same. The front cap has also been simplified in a similar way.

Figure 5-1 2D Axis-symmetric RBP Model and Boundaries

User Defined Function (UDF) is a function that is written in the C programming language to enhance the standard features of the FLUENT code [129]. Users can use a

UDF to define their own boundary condition, material properties, source term and etc. for their flow regime. In this simulation, due to the periodic motion of the pumping unit, an unsteady solver has been used. The motion of the pumping unit is prescribed using a UDF that is coupled with the FLUENT solver. The motion of the ball is determined by FSI interaction between the fluid flow and the ball's displacement. This is also achieved using an UDF that is coupled with the FLUENT solver to evaluate the forces acting onto the ball and to determine its motion. The details of the FSI interaction and the force calculation are presented in section 5.3. Due to motion of the pumping unit and the ball, a dynamic mesh model which is movable and deformable has been used. The details of the dynamic mesh model are discussed in section 5.4.

5.2 Assumptions

In the CFD simulation, some assumptions made are:

- (1) According to section 4.2, due to the axis-symmetric boundary condition, the Navier-Stokes equations can be reduced to the r and z directions only. Thus, in this CFD simulation, instead of using a 3D CFD domain, a 2D axis-symmetric model is used to simplify the problem and to reduce the computational demand.
- (2) In the experimental studies, saline solution has been used as the working fluid. However, due to the negligible difference of the density and viscosity of pure water and saline solution, water has been selected as the working fluid for the CFD simulation.
- (3) Experimental works were conducted at pumping frequency of 1.667, 2.5, 3.333 and 4.167Hz, i.e. 100, 150, 200 and 250 rpm respectively. The corresponding maximum Reynolds number are 5090, 7636, 10182 and

12741 respectively. The velocity of the pumping unit gradually increases from 0 m/s to its corresponding maximum velocity based on the operating frequency. This covers a wide range of Reynolds number and thus, $k - \omega SST$ (shear stress transport) turbulence model was selected in the simulation in order to cover the range of Reynolds number during the simulation.

- (4) The pumping unit of the RBP pump is driven by the permanent ring magnet. It is assumed that the pumping unit will move at the same displacement velocity as the permanent ring magnet. The displacement or velocity of the pumping unit of the RBP at different time steps can be determined by equations (4-6) or (4-7), and this motion can be prescribed using a UDF. However, the motion of the ball inside the casing is based on the motion of the pumping unit and the forces acting on it. This is typically a Fluid Structure Interaction (FSI) problem. It is assumed that the forces acting onto the ball are primarily the pressure and viscous force.
- (5) According to section 3.2.1, the function of the three supporting legs is to constrain the ball to move in the axial direction, i.e. z-direction, only. Thus, in the simulation, the ball is assumed to travel in the axial direction, i.e. z-direction, and no rotation is involved. Hence, when evaluating the displacement or velocity of the ball, only the forces acting in the axial direction, i.e. z-direction, will be considered.

- (6) Physically, when the RBP is in operation, the ball will repeatedly come in contact with either the front or the rear cap. However, it is impossible to perform a full closure of the ball with the rear cap or front cap in the CFD simulation. This is because when both the wall surface of the rear cap or front cap comes in contact with the wall surface of the ball, the adjacent computational cells are forced to collapse and the cell volume will be set to zero or become negative when cells overlap each other. This will lead to a discontinuity in the computation domain and cause computational errors during the simulation. Thus, a constraint has been made on the motion of the ball such that there is always a minimum clearance gap between the wall surface of the rear cap or front cap and the wall surface of the ball in order to prevent this problem.

5.3 Fluid-Structure-Interaction (FSI) and Forces Calculation

Fluid-structure-interaction (FSI) is one of the most challenging multi-physics problems which involve the interaction of some movable or deformable structure with an internal or surrounding fluid flow [110]. During the RBP pumping operation, the pumping unit is driven in reciprocating motion by magnetic or electromagnetic forces outside the pipe. Due to the motion of the pumping unit and the ball, the flow field will change accordingly during the pumping process. The changes in the flow field will also affect the forces acting on the ball and thus alter the ball's velocity and position with respect to the pumping unit. Thus, this is typically a fluid-structure-interaction (FSI) problem which involves a movable structure with a surrounding fluid flow. In the CFD simulation, the reciprocating motion of the pumping unit has been prescribed by means of a UDF. However, the ball inside the pumping unit is not attracted by the magnetic or

electromagnetic forces. The ball is free to move in the axial direction depending on the forces acting on it. These forces include the force from the pumping unit, pressure and viscous forces due to the flow field around the ball. The forces acting on the ball are shown in Figure 5-2. In the CFD simulation, a UDF with a FSI algorithm is used to evaluate the forces acting on the ball and determine its displacement and velocity during the pumping process.

Figure 5-2 Forces Acting on the Ball when (a) Ball In Contact with Rear Cap during Forward Stroke, (b) Ball In Between Rear and Front Cap during Forward Stroke, (c) Ball In Contact with Front Cap during Backward Stroke, (d) Ball In Between Rear and Front Cap during Backward Stroke

As shown in Figure 5-2, there are three possible forces acting on the ball during the pumping operation, namely:

1. F_{casing} , the force by the pumping unit that pushes the ball in either forward or backward direction. This force is only present when the ball is in contact with either rear cap or front cap.
2. F_{pressure} , pressure forces acting on the ball in the z-direction, i.e. axial direction.

The equation for calculating the pressure force in FLUENT is given as

$$\vec{F}_{\text{pressure}} = \sum_{i=1}^n (P - P_{\text{ref}}) A \hat{n} \quad (5-1)$$

Or

$$\vec{F}_{\text{pressure}} = \sum_{i=1}^n P A \hat{n} - \sum_{i=1}^n P_{\text{ref}} A \hat{n} \quad (5-2)$$

where $\vec{F}_{\text{pressure}}$ is the pressure force vector acting on the ball

n is the number of faces on the ball in the CFD domain

A is the area of the face

$\hat{n}$ is the unit normal to the face

P is the pressure acting on the face

P_{ref} is the reference pressure used to normalize the cell pressure for computation of the pressure force

Due to the assumption made in section 5.2, only forces in the z -direction, i.e. axial direction, will be taken into consideration during the simulation, thus, the pressure force acting on the ball in the z -direction can be expressed as

$$F_{\text{pressure}} = \vec{F}_{\text{pressure}} \cdot \vec{z} \quad (5-3)$$

Where $\vec{z}$ is the unit vector in z -direction.

3. F_{viscous} , viscous forces acting on the ball in the z -direction. The equation for calculating the viscous force in FLUENT is given as

$$\vec{F}_{\text{viscous}} = \sum_{i=1}^n \vec{\tau} A \quad (5-4)$$

Where $\vec{\tau}$ is the shear stress

n is the number of faces on the ball in the CFD domain

A is the area of the face

Due to axis-symmetric boundary condition, the components in $\vec{\tau}$ are the stress acting on a plane perpendicular to the z -direction but in the radial direction, τ_{zr} , and the stress acting on a plane perpendicular to radial direction but in the z -direction τ_{rz} . Thus, $\vec{\tau}$ can be written as

$$\vec{\tau} = \tau_{zr} \vec{r} + \tau_{rz} \vec{z} \quad (5-5)$$

where $\vec{r}$ is the unit vector in r -direction, i.e. radial direction

$\vec{z}$ is the unit vector in z -direction, i.e. axial direction

In addition,

$$\tau_{zr} = \tau_{rz} = \mu \left(\frac{\partial u_r}{\partial z} + \frac{\partial u_z}{\partial r} \right) \quad (5-6)$$

Hence, the viscous force that acting on the ball in z -direction can be expressed as

$$F_{\text{viscous}} = \vec{F}_{\text{viscous}} \cdot \vec{z} \quad (5-7)$$

From Figure 5-2 (a), when the pumping unit is moving in the forward or positive z -direction, the rear cap of the pumping unit comes in contact with the ball and pushes it to move in the positive z -direction. At this instance, the pumping unit of the RBP is in the ‘closed’ condition and no fluid is allowed to flow through the pumping unit. In addition, the pressure force and viscous force is pressing the ball in the negative z -direction so that the ball will always be in contact with the rear cap. In summary, the forces acting on the

ball when it is in contact with the rear cap, i.e. a ‘closed’ RBP, during the forward stroke can be written as:

$$F_{\text{total}} = F_{\text{casing}} - F_{\text{pressure}} - F_{\text{viscous}} \quad (5-8)$$

Referring to Figure 5-2 (b), when the pumping unit is moving in the forward or positive z-direction, there is an instance where the upstream pressure is larger than the downstream pressure. At this instance, due to the larger pressure force upstream, the ball tends to move at a faster speed as compared to the pumping unit, thus, it detaches from the rear cap of the pumping unit and it is floating between the rear cap and front cap. At this moment, there is no direct contact of the ball with either the rear cap or front cap. Thus, the pushing force from the pumping unit is absent in this case. The forces acting on the ball are pressure and viscous forces as shown in Figure 5-2 (b). The total forces acting on the ball can be written as:

$$F_{\text{total}} = F_{\text{pressure}} + F_{\text{viscous}} \quad (5-9)$$

When the pumping unit is moving in the backward direction, the front cap of the pumping unit comes in contact with the ball and pushes it to move in the negative z-direction as shown in Figure 5-2(c). Meanwhile, the pressure and viscous forces are acting on the ball in positive z-direction. Thus, the forces acting on the ball when it is in contact with the front cap during backward stroke can be written as:

$$F_{\text{total}} = -F_{\text{casing}} + F_{\text{pressure}} + F_{\text{viscous}} \quad (5-10)$$

Referring to Figure 5-2 (d), when the pumping unit is moving in the backward or negative z-direction, there is an instance where the downstream pressure is larger than the upstream pressure. At this instance, due to the larger pressure force downstream, the

ball tends to move at a faster speed as compared to the pumping unit in the negative z -direction, thus, it detaches from the front cap of the pumping unit and it is located between the rear cap and front cap. At this moment, there is no direct contact of the ball with either the rear cap or front cap. Thus, the pushing force from the pumping unit is absent in this case. The forces acting on the ball are pressure and viscous forces as shown in Figure 5-2 (b). The total forces acting on the ball can be written as:

$$F_{\text{total}} = -F_{\text{pressure}} - F_{\text{viscous}} \quad (5-11)$$

After obtaining the total force acting on the ball, the velocity at time, t , is calculated using the total force in the explicit Euler formula as shown below:

$$v_t = v_{t-\Delta t} + \left(\frac{F_{\text{total}}}{m_{\text{ball}}} \right) \cdot \Delta t \quad (5-12)$$

where $v_{t-\Delta t}$ is the velocity at the previous time step is, m_{ball} is the mass of the ball and Δt is the time step size. The new position or displacement of the centre of gravity of the ball can then be calculated using the below equation

$$z_{t+\Delta t} = z_t + v_t \cdot \Delta t \quad (5-13)$$

Where $x_{t+\Delta t}$ is the position of the ball at the next time step and x_t is the position of the ball at the current time step.

5.4 Movable and Deformable Dynamic Mesh

In order to reproduce the reciprocating motion of the pumping unit of RBP and the relative motion between the pumping unit and the ball, the FLUENT user-defined-function (UDF) has been used in the simulation. The whole 2D-axis-symmetric domain has been separated into different fluid zones to cater for the motion of the RBP. Figure 5-3 shows the different dynamic fluid zones that cater for the reciprocating motion of the pumping unit of the RBP. Starting from the left is (I) Stationary Inlet Fluid Zone, (II) Upstream Layering Fluid Zone, (III) RBP Moving Fluid Zone, (IV) Downstream Layering Fluid Zone and (V) Stationary Outlet Fluid Zone. During the simulation, fluid zone III – RBP Moving Fluid Zone that consists of the pumping unit and the ball is moving in a reciprocating motion as prescribed in the UDF based on the slider crank mechanism used in the experimental works. Due to the motion of fluid zone III, fluid zone II – Upstream Layering Fluid Zone and fluid zone IV – Downstream Layering Fluid Zone are performing layering dynamic mesh deformation to accommodate the motion of fluid zone III. Fluid zone I – Stationary Inlet Fluid Zone and fluid zone V – Stationary Outlet Fluid Zone are stationary throughout the simulation. The effect on meshes after dynamic layering in fluid zone II and IV during Bottom Dead Center (TDC) and Top Dead Center (TDC) of the reciprocating motion are shown in Figure 5-4.

During the simulation, due to the pressure and viscous forces acting on the ball, the ball actually travels at a displacement that is slightly different from the pumping unit. In order to cater for the relative motion between the pumping unit and the ball, fluid zone III is further subdivided into several fluid zones as shown in Figure 5-5.

Figure 5-3 2D Axis-symmetric RBP Model Boundary Conditions and Fluid Zone. From left to right (I) Stationary Inlet Fluid Zone, (II) Upstream Layering Fluid Zone, (III) RBP Moving Fluid Zone, (IV) Downstream Layering Fluid Zone, (IV) Stationary Outlet Fluid Zone

Figure 5-4 Dynamic Layering Effect on Fluid Zone II and Fluid Zone IV during (a) BDC (b) TDC

Figure 5-5 Subdivided Fluid Zones in RBP Moving Fluid Zone III

Figure 5-6 Dynamic Layering Effect on Subdivided Fluid Zones iv and vi when (a) Ball In Contact with Rear Cap (b) Ball in the Middle of the Pumping Unit (c) Ball In Contact with Front Cap

Subdivided Fluid Zone	Motion	Dynamic Mesh Scheme
i	Pumping Unit Displacement/ Velocity	No Dynamic Mesh Scheme
ii	Pumping Unit Displacement/ Velocity	No Dynamic Mesh Scheme
iii	Pumping Unit Displacement/ Velocity	No Dynamic Mesh Scheme
iv	Ball Displacement/Velocity	Dynamic Layering Scheme
v	Ball Displacement/Velocity	No Dynamic Mesh Scheme
vi	Ball Displacement/Velocity	Dynamic Layering Scheme

Table 5-1 Summary of the Motion and Dynamic Mesh Scheme of Subdivided Fluid Zones in Fluid Zone III

The effect on meshes after dynamic layering in the subdivided fluid zone (iv) and (vi) when the ball is in contact with the rear cap, in the middle of the pumping unit and in contact with the front cap are shown in Figure 5-6.

By using the dynamic layering in fluid zone II and IV as well as the subdivided fluid zone (iv) and (vi), the dynamic mesh model can replicate the motion of the reciprocating ball pump accurately.

5.5 Grid Independency Study

A converged solution is not necessarily a correct solution. One of the ways to determine the accuracy of the solution is to ensure that the solution is grid independent. The solution is said to be grid independent when further refinement of the grid will not yield any significant changes to the solution.

Grid independency test can be carried out by creating different mesh sizes for the given fluid domain ranging from coarser mesh with lesser number of cells to a finer mesh with larger number of cells. This can be conducted using Gambit.

Simulations can be carried out with the coarser mesh first. After that, the finer meshes will be used for the same setup. After several runs on different meshes, if the solution is found to be very close to the one that is obtained previously, the solution is said to be a grid independent solution where refinement of grids does not change the solution significantly.

In this CFD simulation, two sets of grid independency studies have been carried out in deformable zones (fluid zone II, III and IV) and stationary zones (fluid zone I and V) respectively in order to determine the most appropriate meshing topology for RBP. The first set of grid independency test consists of five different meshes. The details of these meshes for these two set of grid independency test are shown in Table 5-2 and Table 5-3. The total number of cells, faces and nodes are directly proportional to the mesh size within the deformable zone.

The grid independency test is carried out by comparing several parameters over different meshes. These parameters include total number of iterations, total forces acting onto the ball, the pressure at the upstream and downstream of the pumping unit. The total number of iteration was selected for comparison because it is directly proportional to the computational load. The total forces acting on the ball were selected because it is the most important parameter in solving the coupled FSI problem. The pressures at upstream and downstream of the pumping unit were selected because they are one of the most important parameter which determines the performance of the pump.

Mesh	Mesh Size (Deformable Size)	No. of Cells	No. of Faces	No. of Nodes
A	0.300	84471	171104	86632
B	0.200	103656	209726	106069
C	0.150	133539	269795	136255
D	0.100	191110	385352	194241
E	0.075	261395	526311	264915

Table 5-2 Meshes for Grid Independency Test for Deformable Zones

The first set of grid independency test was carried out to determine the optimal mesh size of the deformable zones (fluid zone II, III and IV). The details of the meshes used in the first set of grid independency test are shown in Table 5-2. Figure 5-7 (i) shows the total number of iterations needed to achieve the convergence criteria for different

meshes; Figure 5-7 (ii - iv) shows the percentage errors of the total force acting on the ball, the pressure at the upstream and downstream of a particular mesh when it is compared to the finer mesh. Figure 5-7 (ii - iv) shows that coarser meshes (mesh A and B) have larger errors as compared to finer meshes (mesh C, D and E). It can be observed that from mesh C onwards, as the total no of cells increase, the percentage errors on the total force, upstream and downstream pressures are almost zero. Thus, meshes C, D and E are said to be grid independent when further refinement of the grid does not yield any significant changes to the solution. In addition, according to Table 5-2 and Figure 5-7 (i), the number of meshes (cells, faces and nodes) and the number of iterations used for achieving convergence criteria for meshes D and E are significantly higher than mesh C. Hence, mesh C is selected for subsequent CFD simulations in order to reduce the computational load.

Figure 5-7 Grid Independency Test for Deformable Fluid Zones

The second set of grid independency test was carried out to determine the optimal mesh size of the stationary zones (fluid zone I and V). The grid independent mesh selected from the first set of grid independency test – Mesh (C) was also used in the second set of grid independency test. In the second set of grid independency test, the selected mesh from the first set of grid independency test, mesh (C) is identified as mesh (a). In the second set of grid independency test, the meshes inside the deformable zone (fluid zone II, III and IV) remains unchanged, while the meshes in the stationary zone were refined to different sizes to conduct the second set of grid independency test. The details of the meshes used in the second set of grid independency test are shown in Table 5-3.

Mesh	Mesh Size (Stationary Zones)	Cells	Faces	Nodes
a	size function (0.5 - 5)	133539	269795	136255
b	4	218658	441702	223043
c	2	357123	721347	364223
d	1	419751	847831	428079
e	0.5	524046	1058466	534419

Table 5-3 Meshes for Grid Independency Test for Stationary Zones

Figure 5-8 (i) shows the total number of iterations needed to achieve the convergence criteria for different meshes; Figure 5-8 (ii - iv) shows the percentage errors of the total force acting onto the ball, the pressure at the upstream and downstream of a particular mesh when it is compared to the finer mesh. It can be observed that for meshes (a)-(e), as the total no of cells increases, the percentage errors on total force, upstream and downstream pressures are almost equal to zero. Thus, meshes (a)-(e) are said to be grid independent when further refinement of the grid does not yield any significant changes to the solution. Generally, the computational load is proportional to the total number of cells, faces or nodes in the mesh and the total number of iteration to achieve convergence. Figure 5-8 (i) shows that mesh (a) requires less iteration to achieve convergence. In addition, Table 5-3 indicates that mesh (a) has the least number of cells,

faces or nodes. Thus, to reduce the computational load, mesh (a) was selected as the mesh for conducting further simulation.

Figure 5-8 Grid Independency Test for Stationary Fluid Zones

5.6 FLUENT Solver with UDF Coupled FSI Algorithm

In this project, a commercial CFD software, FLUENT 13.0.0 (ANSYS, Inc) is used to solve the flow and pressure field. The flow chart of the RBP CFD simulation with UDF coupled FSI algorithm is shown in Figure 5-9.

From Figure 5-9, it is noted that the RBP CFD simulation is divided into 2 parts. The major part is the FLUENT solver that solves the conservation equation of mass and momentum by means of a finite volume approach. The other part is the user defined code – FSI algorithm that is used to calculate the motion of the pumping unit and the

ball. The UDF FSI algorithm is compiled and linked with the FLUENT solver so that information can be exchanged between the FLUENT solver and the FSI algorithm. The details of the flow of RBP CFD simulation are shown in Figure 5-9.

Figure 5-9 Flow Chart of RBP CFD Simulation with UDF coupled FSI in FLUENT

To obtain a correct solution for the fluid flow problem of the RBP, the parameters should be set correctly according to the physical problem. FLUENT provides two

numerical methods: Pressure based (previously known as segregated solver) and density based (previously known as coupled solver) solver. The two numerical methods employ a similar discretization process (finite-volume), but the approach used to linearize and solve the discretized equations is different. The pressure based solver has traditionally been used for incompressible and mildly compressible flows, while the density based solver may give a performance advantage in solving high-speed compressible flows. Therefore, in this study, the pressure based solver is selected.

The working fluid used in the simulation is water, which is a Newtonian fluid that is incompressible as stated in section 5.2. Axis-symmetry boundary condition is adopted for the simulation of the RBP due to its geometry. No-slip condition is applied to all the pipe and pumping unit walls so that the fluid velocity along the wall boundary would be equal to zero. Constant gauge pressures corresponding to the water level in the experiments is set at the inlet and outlet as boundary conditions.

After setting up these parameters, the solver commences by initialization of all variables at $t = 0s$. A time step size that satisfies the Courant–Friedrichs–Lewy (CFL) condition is calculated for the simulation to ensure convergence. However, a smaller time step is selected to provide 1000 data per cycle in the solution. Before the iteration starts, the FLUENT solver reads the information of the displacement and velocity of the pumping unit and ball from UDF FSI algorithm. At $t = 0s$, these data are all set to zero. However, at $t > 0s$, the information of the displacement and velocity of the pumping unit and ball will be passed to the FLUENT solver after the calculation in the UDF FSI algorithm. Subsequently, the mesh will move or deform based on the information obtained from the UDF FSI algorithm. By taking the motion of the pumping unit and

ball as boundary condition, the iteration starts to solve the discretized Navier-Stokes equation and obtain the solution for the pressure and flow field. The iterations are completed when all the variables converge to a specified convergence criterion. When convergence is achieved, all the discrete conservation equations such as conservation of momentum, energy continuity and etc are observed to reach a specified tolerance in all cells. Generally, a converged criterion or tolerance from 10^{-5} to 10^{-6} is needed to obtain a satisfactory solution. After obtaining the converged solution in the FLUENT solver, information on the pressure and flow field will be passed into the UDF FSI algorithm. The FSI algorithm evaluates the forces acting on the ball based on the information obtained from the FLUENT solver and calculates the velocity and displacement of the ball in the next time step. After advancing to next time step, the information of the displacement or velocity of the pumping unit and the ball obtained from the UDF FSI algorithm will again be passed to the FLUENT solver to update the mesh and boundary condition. The calculation repeats until the ending time, t_e , when the complete cycle of RBP simulation is reached.

The theoretical and CFD studies have been discussed in detail in chapters 4 and 5 respectively. In the next chapter, the experimental setup for the RBP will be presented. The test rig is designed to measure the performance of the pump. In addition, an improvised test rig designed specifically for flow visualization experiment will also be presented.

CHAPTER 6

EXPERIMENTAL SETUP

The mathematical modeling and CFD studies of the RBP have been presented in chapters 4 and 5. In this chapter, the experimental setup for the RBP will be presented. The experimental setup, equipment, measurement instruments and the experimental procedures will be introduced. The calibration and uncertainties analysis of the measurement instruments will also be discussed. In addition, the improvised test rig design specifically for flow visualization experiments will also be presented.

6.1 Experimental Test Rig for Performance Characteristics

To prove the concept and the working principle of the RBP and to validate the mathematical modeling as well as the CFD simulations, a test rig was designed and constructed to conduct experiments to determine the performance of the pump. The schematic diagram and the actual test rig of the experimental setup for the RBP are illustrated in Figure 6-1. In summary, the experimental setup consists of, in accordance to the flow sequence:

a) Reservoir

In this experimental setup, a closed loop test rig has been designed. The reservoir is used to supply water to the pump. The reservoir is separated into two sections for inlet and outlet respectively in order to create different pressure heads between the upstream and downstream of the RBP.

Figure 6-1 Experimental Setup for RBP (i) Schematic Diagram (ii) Actual Test Rig

b) Nihon Kohden Electromagnetic FlowMeter MFV-3200

The Nihon Kohden electromagnetic flow meter MFV-3200 shown in Figure 6-2 is a non-intrusive flow meter. It consists of (i) the main unit and (ii) the flow

probe. The flow probe is installed upstream of the pump and is connected to the main unit to display the measured average flow rate. Alternatively, the analog voltage output from the main unit can be connected to a DAQ card to obtain the instantaneous flow rate generated by the pump via a computer. The flow meter is capable of measuring a maximum flow rate of up to 19.9 l/min .

Figure 6-2 Nihon Kohden Electromagnetic Flow Meter MFV-3200 (i) Main Unit & (ii) Flow Probe

c) Setra Pressure Transducers Model 230

Two Setra pressure transducers model 230 as shown in Figure 6-3 with an output voltage of 0 to 5VDC for a range of ± 5 PSID or $\pm 34.47 \text{ kPa}$ are used in the experimental studies. In the experiments, the pressure transducers are used to measure the instantaneous pressure at the upstream and downstream of the RBP with reference to the atmospheric pressure. For upstream pressure measurement, one end of the pressure transducer is connected to the connector located $2D$ (where D is the diameter of the tube) upstream from the RBP according the British Standard 848 [130] while the other end was left open to the atmosphere. For downstream pressure measurement, the pressure tapping point is located $4D$ downstream of the RBP.

Figure 6-3 Setra 230 Pressure Transducer

d) RBP with Slider Crank Mechanism and Limit Switches

To create reciprocating motion of the RBP, and to prevent any potential contamination to the fluid, a neodymium permanent ring magnet is used to drive the RBP. The pumping unit of the RBP is placed inside the pipe while the neodymium permanent ring magnet is placed outside the pipe. The rear cap, the front cap and the casing of the pumping unit are made up of magnetic grade stainless steel as listed in Table 6-1 such that the casing assembly will be attracted by the Neodymium magnet to provide reciprocating motion. The ball of the pumping unit of the RBP is made up of ceramic Al_2O_3 such that it is not attracted by the magnetic force and is free to move within the casing assembly in order to perform the unique pumping operation of the RBP. The details of the specifications of the pumping unit of the RBP are listed in Table 6-2. A holder is fabricated to hold the magnet and sits on a linear slider guide to ensure that it is moving in a linear motion and is shown in Figure 6-4. The holder is also connected to a slider crank mechanism driven by a DC brushless motor. With this combination, the magnet can drive the pumping unit of the RBP in a reciprocating motion when the motor rotates. The specifications of the slider crank mechanism are listed in Table 6-3. Two limit switches were installed on

the test rig. The purpose of the limit switches is to identify the position of the slider, i.e. the position of the pumping unit that is inside the pump. Hence, the experimental data can easily be mapped with the simulation data by comparing the position of the pumping unit.

Material List of the Pumping Unit of the RBP	
Rear Cap	AISI 410 Stainless Steel
Front Cap	AISI 410 Stainless Steel
Casing	AISI 410 Stainless Steel
Ball	Al ₂ O ₃

Table 6-1 Material List of the Pumping Unit of the RBP

Specifications of the Pumping Unit of the RBP	
Outer Diameter	Ø18.8 mm
Rear Cap Hole's Diameter	Ø8.5 mm
Rear Cap Thickness	4 mm
Front Cap Thickness	4 mm
Casing Length	11.5 mm
Ball Diameter	Ø9.6 mm
Weight	25 grams

Table 6-2 Specification of the Pumping Unit of the RBP

Specifications of the Slider Crank Mechanism	
Crank	25 mm
Connecting Rod	115 mm
Stroke Length	50 mm

Table 6-3 Specification of the Slider Crank Mechanism

Figure 6-4 Slider Crank Mechanism

e) National Instruments USB-6221 M Series Multifunction DAQ

National Instruments USB-6221 M Series Multifunction DAQ which is shown in Figure 6-5 is used to acquire the signals generated by the pressure transducers, electromagnetic flow meter and limit switches and to store them in the computer for future data processing.

Figure 6-5 National Instruments USB-6221 M Series Multifunction DAQ

f) Computer

The National Instruments USB-6221 M Series Multifunction DAQ is connected to a computer via USB port. The signals acquired by the DAQ card will first be

processed by the National Instruments Signal Express software and stored in the computer for further data processing.

g) 24VDC Power Supply

Both pressure transducers require a 24VDC operating voltage. Thus a 24VDC power supply is used to power these devices.

h) Tachometer

A handheld tachometer is used to measure the rotational speed of the motor shaft when the experiment is being carried out.

Figure 6-6 Tachometer

6.1.1 Calibration and Uncertainties Analysis

Calibration is an important process prior to any actual experimental measurements. Calibration is a comparison between the measurements of a standard device with known magnitude and another measurement of the device under test or being calibrated. There are 3 measurement devices used in the experimental studies of RBP. They are the upstream and downstream pressure transducers for measuring instantaneous pressure and electromagnetic flow meter for measuring instantaneous flow rate.

In addition to the calibration, the uncertainty estimation of an experimental result should always be determined. The uncertainty estimates the limits of experimental errors.

When determining the uncertainty of a measurement, basically standard deviations (or their squares, called variances) are evaluated. Standard uncertainty of a single component can be determined in two ways [131, 132] as described below:

- a. Calculating uncertainty by statistical means from repeated measurements (Type A)

Type A uncertainty is a statistical way to determine the uncertainty in measurement. By taking repeated measurements of a single input, the mean output value can be calculated by using equation (6-1).

$$\bar{q} = \frac{1}{n} \sum_{i=1}^n q_i \quad (6-1)$$

Where $\bar{q}$ is the arithmetic mean of series of n independent measurements

n is the number of independent measurements

q_i is the value of the measurement

Subsequently, the experimental standard deviation of the mean or the standard uncertainty can be estimated by equation (6-2) [131-133].

$$s(\bar{q}) = u = \sqrt{\frac{1}{n(n-1)} \sum_{i=1}^n (q_i - \bar{q})^2} \quad (6-2)$$

Where $s(\bar{q})$ is the experimental standard deviation of the mean

u is the standard uncertainty

- b. Estimating uncertainty from other sources such as manufacturer's data (Type B)

In Type B, standard uncertainty is obtained from a heuristically derived probability that errors from the source will be contained within stated limits. For uniformly distributed errors, contained within symmetrical two sided tolerance

with offsets of $\pm L$, the standard uncertainty can be found by equation (6-3) [131, 132].

$$u = \frac{L}{\sqrt{3}} \quad (6-3)$$

I. Pressure Transducers

Figure 6-7 Air Dead-Weight Tester for Pressure Transducers Calibration

The pressure transducers for measuring upstream and downstream instantaneous pressures of the RBP are manufactured by Setra (USA), model 230. The pressure transducers have been calibrated using air dead-weight tester as shown in Figure 6-7. The pressure-voltage relationship is obtained based on the best fit line of the mean values of 50 voltage measurements (output) at different pressure (input) which is determined by the total weights added onto the air dead-weight tester. The calibration

results are shown in Figure 6-8 which indicates that a good linearity of the pressure-voltage relationship is obtained for the two pressure transducers.

At different pressures, which are determined by the total weights added onto the air dead-weight tester, 50 voltage measurements are taken. The voltage readings are then converted into pressure readings by using the linear equation obtained during the calibration process. The type A standard uncertainties can be obtained from these repeated measurements and calculated by equation (6-2). The maximum Type A standard uncertainties found for pressure transducer A and pressure transducer B is 0.0025 PSID or 17.073 Pa and 0.0032 PSID or 22.272 Pa respectively. However, according to the manufacturer's data, the accuracy of the pressure transducer is stated as $\pm 0.25\%$ of full scale which is equivalent to ± 0.025 PSID (± 172.35 Pa). Thus, type B standard uncertainties can be estimated by equation (6-3) where

$$u = \frac{L}{\sqrt{3}} = \frac{0.025}{\sqrt{3}} = 0.0144 \text{ PSID}$$

Or

$$u = \frac{L}{\sqrt{3}} = \frac{172.35}{\sqrt{3}} = 99.51 \text{ Pa}$$

Figure 6-8 Pressure-Voltage Relationships of (a) Upstream (b) Downstream Pressure Transducer

Since the standard uncertainties found statistically (Type A) is smaller than the standard uncertainties found heuristically (Type B), it is recommended to use Type B standard uncertainties for estimating the uncertainties of the pressure transducers in order to cater for the other sources of error that are unaccounted for during the repeated measurements conducted in determining the Type A uncertainties.

II. Electromagnetic Flow meter

Nihon Kohden Electromagnetic Flow meter MFV-3200 is used to measure the instantaneous flow rate generated by the RBP during the experiment. The electromagnetic flow meter has been calibrated using the setup shown in Figure 6-9. A centrifugal pump was used to generate a constant flow rate for the calibration. The amount of flow rate can be controlled by adjusting the throttle valve. The calibration of the Nihon Kohden electromagnetic flow meter is carried out by using a Kobold magnetic inductive flow meter with an accuracy of 0.21% of full scale which is equivalent to $\pm 0.105 \text{ l/min}$. The flow rate-voltage relationship was obtained based on the best fit line of the mean values of 50 voltage measurements (output) at different flow rates (input). The calibration results are shown in Figure 6-10 which indicates that good linearity of the flow rate-voltage relationship are obtained in the electromagnetic flow meter.

Figure 6-9 Experimental Setup for Flow Meter Calibration

At different flow rates, which are controlled by the throttle valve, 50 voltage measurements were taken. The voltage readings are then converted into flow rate readings by using the linear equation obtained during the calibration process. The type A standard uncertainties can be obtained from these repeated measurements and calculated by using equation (6-2). The maximum Type A standard uncertainties of the electromagnetic blood flow meter was found to be 0.0253 l/min . However, according to the manufacturer's data, the accuracy of the electromagnetic flow meter is stated as $\pm 2\%$ of rate of flow. The maximum flow rate of the magnetic flow meter is 19.90 l/min . Thus, the maximum error can be calculated by taking $\pm 2\%$ of 19.90 l/min which is equal to $\pm 0.380 \text{ l/min}$. Thus, type B standard uncertainties can be estimated by using equation (6-3) where

$$u = \frac{L}{\sqrt{3}} = \frac{0.38}{\sqrt{3}} = 0.219 \text{ l/min}$$

Figure 6-10 Flow Rate-Voltage Relationships of Electromagnetic Flow Meter

Since the standard uncertainties found statistically (Type A) is smaller than the standard uncertainties found heuristically (Type B), it is recommended to use Type B standard

uncertainties for estimating the uncertainties of the flow meter in order to cater for the other sources of error that are unaccounted for during the repeated measurements conducted in determining the Type A uncertainties.

6.1.2 Experimental Variables

According to the working principle of the RBP, reciprocating motion is generated by the slider crank mechanism which is connected to a DC brushless motor. When the motor is started, flow and pressure will be generated based on the motion of the pumping unit. The main objective of the project is to analyze the performance of the RBP by measuring the instantaneous flow rate and the instantaneous pressure variation at the upstream and downstream of the RBP. In the experiment studies, there are five important variables that are considered, namely the rotational speed of the motor, downstream back pressure, the weight of the ball, the rear cap hole diameter and the length of the casing of the RBP.

a. Variation of Rotational Speed

Based on the working principle of the RBP, the flow and pressure is generated due to the motion of the pumping unit. It is obvious that the rotational speed of the motor which controls the reciprocating motion of the pumping unit has a significant effect on the instantaneous flow rate and the instantaneous pressure variation at the upstream and downstream of the pump. However, to avoid unnecessary level of vibration during high rotational speed operation, low motor speed was used throughout the experiment. In this project, experiments with motor speed of 100, 150, 200 and 250 rpm are conducted in order to analyze the effect of rotation speed on the performance of the pump.

b. Downstream Back pressure

Generally, pumps are required to operate against certain pressure in order to deliver the fluid to the desired location. In this project, experiments are conducted by prescribing certain back pressure at the downstream of the pump unit to examine the effect of back pressure on the pump performance. The downstream back pressure can be achieved by maintaining a difference in the water levels between the inlet and outlet with the reservoir.

c. Different Material of Balls with Different Density and Weight

The ball inside the pumping unit is one of the most important components of the RBP. Figure 6-11 shows balls of different materials with different densities and weights. During the RBP operation, the weight of the ball may affect the performance of the RBP. In this project, experiments are carried out by using different materials to evaluate the effect on the performance of the RBP due to different balls' weight. The detail of the balls used in the experiments is shown in Table 6-4.

Figure 6-11 Five Different Material of Balls with Different Density and Weight

Material	Weight (grams)	Density (kg/m ³)
Al ₂ O ₃	1.750	3867.63
PA66	0.520	1149.24
Si ₃ N ₄	1.460	3226.70
ZrO ₂	2.700	5967.19
Tungsten Carbide	6.825	15083.74

Table 6-4 Weight and Density of the Balls Used in the RBP Experiments

d. RBP Rear Cap with Different Hole Diameter

The diameter of the hole on the rear cap will affect the distance travelled by the ball within the pumping unit of the RBP. In addition, the diameter of the hole on the rear cap may also affect the amount of induced flow rate flowing through the pumping unit of the RBP during the backward stroke. Thus, in the experimental studies, five rear caps with different hole diameter are used to study the effect on the performance of the RBP due to different hole diameter. The rear caps which are shown in Figure 6-12 were manufactured with hole diameter of 7.0, 7.5, 8.0, 8.5 and 9.0 mm respectively.

Figure 6-12 RBP Rear Cap with Different Hole Diameter

e. RBP Casing with Different Length

The length of the RBP casing will affect the distance travelled by the ball within the pumping unit of the RBP. In addition, it may also affect the amount of induced flow rate flowing through the pumping unit of the RBP during the backward stroke. Thus, in the experimental studies, five casings with different length are used to study the effect on the performance of the RBP due to different casing length. The casings which are shown in Figure 6-13 were manufactured with lengths of 9.5, 11.5, 13.5, 15.5 and 17.5 mm respectively.

Figure 6-13 RBP Casing with Different Length

6.1.3 Experimental Procedures

The experiments are carried out with different operating variables as discussed above. The purpose of the experiment is to verify the working principle of the RBP, to determine the performance of the RBP and to evaluate its sensitivity to the change in the variables. Before actual measurements, the RBP is run for several minutes to prime and remove air bubbles inside the pipe line. It is noted that the experiments are conducted after the pressure transducers have been switched on for about 20 minutes to allow stabilization of the pressure reading after the priming process. The experimental procedures are as follows:

1. Switch on the power supply for both the flow meter and the pressure transducers.
Switch on the computer and activate the NI signal express software.
2. Switch on the motor and adjust the motor speed to the desired speed with the help of a tachometer.
3. Press the “Run/Record” button in NI signal express software to start acquiring instantaneous pressures data, instantaneous flow rate data and limit switches data for 1 or 2 minutes.
4. Pressed “Stop” button on NI signal express software to finalize the recorded instantaneous pressures data, instantaneous flow rate data and limit switches data.
Turn off the motor after the finalization of data.

5. Set to another operating variable that is introduced in section 6.1.2 and repeat steps 2-4 to collect another set of readings.

After the experiments have been conducted, the data gathered by NI signal express software can be post processed using Microsoft Excel or Origin. Graphs of instantaneous pressure variation at the upstream and downstream, instantaneous flow rate are plotted and will be presented in the next chapter. These data will also be compared with the data obtained from the simple mathematical modeling and CFD simulation for validation purposes.

6.2 Experimental Test Rig for Flow Visualization

CFD simulation of the RBP has been presented in Chapter 5. To verify the simulation results, flow visualization can be used to validate the predicted flow pattern. The flow visualization technique used in this experiment is tracer method. For this technique, the state of flow is examined by observing the motion of tracer particles or seeding particles that are dispersed in the fluid. In order to make the tracer particles or seeding particles visible, a laser sheet optics as shown in Figure 6-14 is required and a laser sheet can be formed by passing a laser beam through a Dantec divergence lens. The tracer particles or seeding particles become visible when they are illuminated by the laser sheet. By using photography technique, the path lines due to the motion of the tracer particles or seeding particles can be recorded for further study or analysis. The seeding particles used in the experiment are Dantec 50 μm polyamid seeding particles.

In this experimental setup, the following equipment has been added to the existing test rig for the flow visualization experiment:

1. Brimrose Laser Source
2. Holder for Laser Source
3. Dantec Divergence Lens
4. Stand of holding Dantec Divergence Lens
5. Nikon DSLR D90
6. Nikon AF Micro-Nikkor 60 mm F2.8D
7. Velbon Tripod

Figure 6-14 Laser Sheet Optics for Flow Visualization Experiment

6.2.1 Modification on the Existing Test Rig

According to the physical experimental test rig that is presented in Figure 6-1, the neodymium permanent ring magnet actually covers the whole pumping unit inside the

pipe/tube. It blocks the whole pumping unit and makes visualization impossible. Furthermore, the neodymium permanent ring magnet hinders the illumination of the flow field of interest that is illustrated in Figure 6-14 for flow visualization experiment. Thus, in order to study the flow pattern or path lines generated by the pumping unit of the RBP by flow visualization method, some modifications have to be made on the existing test rig.

Figure 6-15 Modified Experimental Test Rig for Flow Visualization

Figure 6-15 shows the modified experimental test rig that is used for flow visualization experiment. The modifications include:

- i. Modified Pipe Section

To allow the laser sheet to illuminate the flow field, while ensuring light refraction and distortion to be kept minimal. Thus, the original circular pipe/tube was replaced with the modified pipe section that is shown in Figure 6-16. The

top and the front surfaces of the modified pipe section are flat and perpendicular to each other. The top flat surface allows laser sheet to pass through and shine into the flow field of interest, while the front surface allows the observer to record the flow pattern. The back of the modified pipe section is circular so that the new magnet holder with a crescent shape magnet can slide along the modified pipe section during the actuation operation. The inner part of the modified pipe section is machined to the same inner diameter as the original circular pipe/tube that has been used in the previous setup such that the pumping unit of the RBP can fit in nicely to perform the pumping motion during the flow visualization experiment.

Figure 6-16 Modified Pipe Section

ii. Modified Magnet Holder & Crescent Shape Neodymium Magnet

As the ring magnet blocks the view of the pumping unit inside the pipe/tube and hindered the entrance of the laser sheet for flow visualization experiment, a crescent magnet shown in Figure 6-17 (i) is used to replace the original ring magnet to drive the pumping unit of the RBP in reciprocating motion. Hence, a new magnet holder shown in Figure 6-17 (ii) is designed and fabricated. The

crescent shape neodymium magnet and the magnet holder are designed so that it can accommodate the circular shape at the back of the modified pipe section, shown in Figure 6-16, and to maintain the attraction force between the magnet and the pumping unit.

Figure 6-17 (i) Crescent Neodymium Shape Magnet, (ii) Modified Magnet Holder

iii. Modified Pumping Unit of the RBP

To observe the motion of the ball inside the pumping unit and to observe the flow field within the pumping unit, the pumping unit of the RBP has been modified into a one-third stainless steel and two-third acrylic pumping unit as shown in Figure 6-18. One-third of the casing assembly is made of stainless steel to maintain the magnetic attraction between the crescent shape magnet and the pumping unit so that the pumping unit can move in a reciprocatory motion. Two-thirds of the pumping unit is made of acrylic material so that the motion of the ball and the flow field inside the pumping unit can be observed during the experiment.

Figure 6-18 Modified Pumping Unit

6.2.2 Experimental Variables

During the preliminary flow visualization experiment, the flow visualization photos that are taken include the whole flow field that covers the full stroke length of the pumping unit of the RBP. One of the preliminary flow visualization photos is shown in Figure 6-19. It can be observed that the flow field and path lines in this photo are of poor quality. To have a clearer view of the flow field and path lines that are generated by the pumping unit of the RBP, a close-up view on the flow field around the pumping unit is needed. Thus, instead of taking the flow visualization photos of the whole flow field that covers that full stroke length of the pumping unit, flow visualization photos were taken at three different zones along the stroke length.

Figure 6-19 Preliminary Flow Visualization Photo

Figure 6-20 shows the three zones where zoom-in photographs of the flow field around the pumping unit are taken. Zone 1 is located at the BDC. Zone 2 located when the

pumping unit is at the middle of the pumping stroke. Lastly, zone 3 is located at the TDC.

Figure 6-20 Zones and Schematic Diagram of the Pumping Unit of RBP

6.2.3 Experimental Procedures

The flow visualization experiments are carried out at different zones as discussed above. The purpose of this experiment is to capture the flow pattern around the pumping unit of the RBP during its forward or backward stroke at a particular zone so that these observations can be used to validate the CFD results. The experimental procedures for the flow visualization experiment are as follows:

1. Add Dantec seeding particles into the water in the reservoir. Stir the water so that the seeding particles scatter evenly in the water.
2. Align the laser beam so that laser sheet is produced and illuminated the flow field of interest.
3. Adjust the position of the camera to zone 1 as described in the previous section.
4. Run the motor at 100 rpm and take photos on the flow field and path lines generated by the pumping unit of the RBP by using Nikon DSLR at a setting of F3.5 and 1/80 seconds.
5. Adjust the position of the camera to zones 2 or 3 and repeat step 4 to obtain the flow field and path lines photos for zones 2 or 3.

After the experiments have been conducted, the photos of the flow field and path lines generated by the pumping unit of the RBP can be post processed by Photoshop CS5. The post-processed photos at different zones will be presented and compared with the data that obtained from the CFD simulation.

Mathematical modeling and CFD simulation of the RBP have been discussed extensively in the previous chapters. The experimental test rig for the RBP is presented in this chapter. In the next chapter, results obtained from mathematical modeling, CFD simulation and experiment will be presented. The results of the mathematical modeling and CFD simulation will also be compared with the experimental measurements to validate the mathematical model and the CFD simulation.

CHAPTER 7

RESULTS AND DISCUSSIONS

In this chapter, results obtained from the mathematical modeling, CFD simulation and experimental studies will be presented. In the first three sections of this chapter, the performance characteristics of the RBP will be discussed. The results of the experimental measurements will also be used to validate the findings obtained from the mathematical modeling and CFD studies.

Subsequently, parametric studies which were conducted using the mathematical model and experiment will also be presented. In addition, the pressure flow field generated by the CFD simulation and the experimental flow visualization photos will also be presented.

7.1 Transient Pressure and Instantaneous Flow Rate in the RBP

In this section, prediction of the performance of the RBP is presented. The transient pressure variations and instantaneous flow rate obtained from the mathematical model, the CFD studies and experimental investigation are discussed and compared.

The specifications of the pumping unit of the RBP used in these three studies are (i) rear cap hole of $\varnothing 8.5$ mm, (ii) casing length of 11.5 mm and (iii) Al_2O_3 ceramic ball. For discussion purposes, only two cycles of these data will be presented to have a closer view on the changes of the transient pressure variations and the instantaneous flow rate as well as to show the periodic characteristics of these data.

7.1.1 Mathematical Model Prediction of the Transient Pressure and Instantaneous Flow Rate of the RBP

Figure 7-1 (a) shows the transient pressure variations at the upstream, P_{up} , and downstream, P_{down} , of the RBP and Figure 7-1 (b) shows the instantaneous flow rate generated by the RBP when the slider crank mechanism is operating at 100 rpm with 250 Pa back pressure. The chart in Figure 7-1 (a) and (b) are divided into two sections, (A) forward stroke – from BDC to TDC and (B) backward stroke – from TDC to BDC respectively. The pressure variations during each stroke will be discussed according to a summarized pumping process diagram as shown in Figure 7-1 (c). In addition, the pumping process is divided into different time sequence for better clarity. For the same reason, the theoretically velocity profile of the motion of the pumping unit is embedded with the instantaneous flow rate chart in Figure 7-1 (b).

Figure 7-2 (a) shows the velocity of the pumping unit and the velocity of the ball in RBP while Figure 7-2 (b) shows the normalized relative displacement of the ball with respect to the displacement of the pumping unit where 0 refers to ball in contact with the rear cap (fully closed condition) and 1 refers to ball in contact with the front cap (fully open condition).

Forward Stroke

At time t_0 , the pumping unit reaches its BDC, i.e. the end of its backward stroke or the beginning of its forward stroke, the pumping unit is at stage (i) as shown in Figure 7-1 (c), where the RBP is in the ‘open’ condition. At this moment, it can be observed from Figure 7-2(a) that the ball is moving backwards in the negative z -direction. The ball is not in contact with either the rear cap or the front cap at this moment. From Figure 7-2 (b) the RBP is in the partially ‘open’ condition as the normalized relative displacement

of the ball is neither 1 nor 0 and fluid flows through the ‘open’ RBP due to its inertia and the pressure difference between upstream and downstream. At this instance, P_{down} is negative while P_{up} is positive as shown in Figure 7-1 (a) due to the previous stroke pumping process. The instantaneous flow rate is in the positive direction as shown in Figure 7-1 (b). However, the instantaneous flow rate is almost zero as the pumping unit reaches the BDC and the fluid has lost most of its inertia and will gradually come to rest.

At the next instance, the pumping unit begins its forward stroke and the velocity of the pumping unit increases in the positive z -direction as shown in Figure 7-1 (b). Due to the forward motion of the pumping unit, P_{up} decreases while P_{down} increases as shown in Figure 7-1 (a). After a fraction of a second, at time t_1 , P_{up} and P_{down} intersect and subsequently, P_{down} continues to increase while P_{up} decreases. The differences between P_{up} and P_{down} gives rise to a force acting on the ball in the negative z -direction. Due to the relative displacement between the pumping unit and the ball, the ball will come into contact with the rear cap and the ball is pushed forward by the pumping unit. They then move together in the positive z -direction. Thus, a sudden increase in the ball velocity is observed at time t_1 as shown in Figure 7-2 (a). Figure 7-2 (b) also shows that starting from time t_1 , the normalized relative displacement of the ball with respect to the displacement of the pumping unit is equal to 0 which means that the ball is in contact with the rear cap. At this instance, the RBP is in ‘closed’ condition as illustrated in Figure 7-1 (c) stage (ii). A trapped volume of fluid is formed in front of the pumping unit. As the pumping unit continues to move in the positive z -direction, the fluid in front of the pumping unit is being pushed forward by the motion of ‘closed’ RBP. Thus, the fluid is accelerated and the flow rate starts to increase as shown in Figure 7-1 (b). Due

to the sudden acceleration on the fluid, P_{up} continues to decrease while P_{down} increases significantly.

Figure 7-1 (a) Theoretical Pressure Variations (b) Instantaneous Flow Rate of Simple Model and (c) Summarized Pumping Process of RBP

Figure 7-2 (a) Velocity and (b) Normalized Relative Displacement of the Ball in RBP

After the sudden acceleration from time t_1 to t_2 , the fluid reaches its maximum acceleration and thus P_{up} reaches its minimum while P_{down} reaches its maximum. It can be observed from Figure 7-1 (a) that P_{down} increases to approximately 5000 Pa while P_{up} decreases to approximately 2000 Pa. After time t_2 , the fluid is still accelerating in the positive z-direction but with a decreasing magnitude, thus, P_{up} increases while P_{down} decreases as observed in Figure 7-1 (a). The flow rate also increases due to the acceleration.

The ‘closed’ pumping unit, upon reaching its maximum velocity at time t_3 as shown in Figure 7-2 (b), starts to decelerate; therefore the velocity of the pumping unit starts to decrease. Based on the working principle illustrated in section 3.2.2, the RBP should be in ‘closed’ condition even though the pumping unit is decelerating. Due to the deceleration of the pumping unit, the fluid velocity decreases too. However, from Figure 7-1 (b), the instantaneous flow rate is still increasing even though the pumping unit is decelerating. Thus, it is suspected that the RBP does not work exactly as the proposed working principle as illustrated in section 3.2.2. Due to the deceleration of the pumping unit and the inertia of the fluid, fluid continues to flow in the positive z -direction and at time t_4 , P_{up} and P_{down} intersect and subsequently, P_{down} continues to decrease while P_{up} increases as shown in Figure 7-1 (a). The difference between P_{up} and P_{down} gives rise to a net force acting on the ball in the positive z -direction and causes it to detach from the rear cap. The RBP becomes partially ‘open’ instead of being in the ‘closed’ condition until the end of the forward stroke. It can be observed from Figure 7-2 (a) that after time t_4 , the ball is travelling at a higher velocity as compared to the pumping unit. Figure 7-2 (b) shows that after time t_4 , the normalized relative displacement starts to increase which means that the ball is detached from the rear cap. This allows fluid to flow through the pumping unit due to its own inertia as a result of its acceleration. This is consistent with Figure 7-1 (b) where a higher instantaneous flow rate is obtained. This flow rate corresponds to a higher velocity as compared to the velocity of the pumping unit. The deceleration of the fluid causes P_{up} to increase while reducing P_{down} as shown in Figure 7-1 (a). The net force on the ball causes it to move in the positive z -direction. Hence at time t_5 , the ball comes into contact with the front cap and the pumping unit is fully ‘open’. Figure 7-2 (a) shows that subsequent to time t_5 , the velocities of the ball and the pumping unit are the same, indicating that they move as a single unit. The normalized relative displacement of the ball in Figure 7-2 (b) is 1

which indicates that the pumping unit of RBP is in the fully ‘open’ condition. This allows the fluid to continue to flow through the ‘open’ RBP. The fluid starts to lose its inertia and decelerates. From Figure 7-1 (b), the instantaneous flow rate is observed to decrease.

Backward Stroke

When the pumping unit reaches its maximum displacement at the end of the forward stroke, i.e. TDC, the velocity of the pumping unit is zero. After time t_6 , the pumping unit starts its backward stroke, i.e. moving from TDC to BDC. Inertia effect causes the fluid to move in the forward direction while the pumping unit starts to move in the backward direction. The fluid inertia pushes the ball forward and the ball comes in contact with the front cap and the pumping unit is in the fully ‘open’ condition. Fluid flows forward through the pumping unit during the backward stroke. From Figure 7-1 (b), it is noted that the instantaneous flow rate is decreasing continuously as fluid is losing its inertia until the end of the BDC. Upon reaching the BDC at time t_7 , the whole pumping process repeats itself as described above.

7.1.2 CFD Simulation of Transient Pressure and Instantaneous Flow Rate of the RBP

The transient pressure variations at the upstream, P_{up} and downstream, P_{down} of the RBP and the instantaneous flow rate generated were computed using CFD simulation and are shown in Figure 7-3 (a) and (b) respectively. Similar to section 7.1.1, only two cycles of these data will be presented. It can be observed from Figure 7-3 (a) and (b) that the charts are divided into two sections, (A) forward stroke – from BDC to TDC and (B) backward stroke – from TDC to BDC respectively. The summarized pumping process diagram is also included in Figure 7-3 (c). Similarly Figure 7-4 (a) shows the velocity of

the pumping unit and the velocity of the ball in RBP while Figure 7-4 (b) shows the normalized relative displacement of the ball with respect to the displacement of the pumping unit.

In general, the transient pressure variations and instantaneous flow rate obtained from CFD simulation are very similar to the results predicted by mathematical model. It can be observed from Figure 7-3 and 7-4 that similar events happen from time t_0 to t_7 as compared to the scenario discussed in section 7.1.1. This shows that the mathematical model agrees qualitatively with the CFD results. The key differences between the two results will be discussed in the subsequent section.

Forward Stroke

At time t_0 , the pumping unit reaches its BDC and is at stage (i) as shown in Figure 7-3 (c). From Figure 7-4 (a), the ball is moving backwards while the RBP is partially ‘open’ and fluid flows through the ‘open’ RBP due to its inertia.

After a fraction of a second, at time t_1 , the ball comes in contact with the rear cap and the ball is pushed forward by the pumping unit. They then move together in the positive z -direction. At this instance, the RBP is in ‘closed’ condition as illustrated in Figure 7-3 (c) stage (ii). A trapped volume of fluid is formed in front of the pumping unit. As the pumping unit continues to move in the positive z -direction, the fluid in front of the pumping unit is being pushed forward. The trend of the transient pressure variation and the instantaneous flow rate predicted is similar to that obtained by the mathematical model.

Figure 7-3 (a) CFD Transient Pressure (b) CFD Instantaneous Flow Rate and (c) Summarized Pumping Process of RBP

Figure 7-4 (a) Velocity and (b) Normalized Relative Displacement of the Ball in RBP

After the sudden acceleration from time t_1 to t_2 , the fluid reaches its maximum acceleration and thus P_{up} reaches its minimum while P_{down} reaches its maximum. It can be observed from Figure 7-3 (a) that P_{down} increases to approximately 5000 Pa while P_{up} decreases to approximately 2000 Pa. After time t_2 , the fluid is still accelerating in the positive z-direction but with a decreasing magnitude, thus, an increment in P_{up} and a decrement in P_{down} is observed in Figure 7-3 (a). The flow rate of the fluid is observed to increase due to the positive acceleration.

The ‘closed’ pumping unit, upon reaching its maximum velocity at t_3 as shown in Figure 7-4 (b), starts to decelerate; therefore the velocity of the pumping unit starts to

decrease. However, it can be observed from Figure 7-3 (b) that the instantaneous flow rate continues to increase even though the pumping unit is decelerating and the velocity of the pumping unit is decreasing. As discussed in section 7.1.1, due to the deceleration of the pumping unit and the inertia of the flowing fluid, the fluid continues to flow in the positive z-direction and at time t_4 , P_{up} and P_{down} intersect and subsequently, P_{down} continues to decrease while P_{up} increases as shown in Figure 7-3 (a). The difference between P_{up} and P_{down} gives rise to a force acting on the ball in the positive z-direction and causes it to detach from the rear cap. The RBP becomes partially ‘open’ instead of being in the ‘closed’ condition until the end of the forward stroke as illustrated in section 3.2.2. It can be observed from Figure 7-4 (a) that after t_4 , the ball is travelling at a higher velocity as compared to the pumping unit. It can also be observed from Figure 7-4 (b) that after t_4 , the normalized relative displacement starts to increase which means that the ball is detach from the rear cap. This allows the fluid to flow through the pumping unit due to its own inertia as a result of the previous acceleration. However, due to the deceleration of the fluid, P_{up} continues to increase while P_{down} decreases as shown in Figure 7-3 (a).

At t_5 , the RBP becomes fully ‘open’ as observed from Figure 7-4 (a) that there is a sudden drop in the ball velocity and subsequently the ball follows the pumping unit velocity. This allows the fluid to continue to flow through the ‘open’ RBP inertially. In the meanwhile, the fluid starts to lose its inertia and decelerates. It can be observed from Figure 7-3 (b) that the instantaneous flow rate continues to decrease. In addition, P_{up} continues to increase while P_{down} continues to decrease as shown in Figure 7-3 (a) due to the deceleration of the fluid until the end of the forward stroke.

Backward Stroke

When the pumping unit reaches its maximum displacement at the end of the forward stroke, i.e. TDC, the velocity of the pumping unit is zero. At t_6 , the pumping unit starts its backward stroke, i.e. moving from TDC to BDC. Inertia effect causes the fluid to move in the forward direction while the pumping unit starts to move in the backward direction. The fluid inertia pushes the ball forward and the ball comes in contact with the front cap resulting in a fully ‘open’ condition. Fluid continues to flow forward through the pumping unit during the backward stroke. From Figure 7-3 (b), it is noted that the instantaneous flow rate is decreasing gradually regardless of the motion of the pumping unit. This can be explained by the fact that during the backward stroke, the fluid is losing its forward inertia until the end of the BDC. Upon reaching the BDC at t_7 , the whole pumping process starts all over again as described above.

7.1.3 Experimental Transient Pressure and Instantaneous Flow Rate of RBP

The experimental results of the transient pressure variations at the upstream, P_{up} , and downstream, P_{down} , of the RBP as well as the instantaneous flow rate generated by the RBP are presented in Figure 7-5 (a) and (b) respectively when the RBP is operating at 100 rpm with 250 Pa back pressure. Similar to section 7.1.1 and 7.1.2, two cycles of these data will be presented for the discussions and the figures are organized in the same manner as the previous two sections for the sake of clarity and consistency.

In general, the experimental results are in agreement with those obtained theoretically and computationally though there are differences in magnitudes of the pressures and flow rate obtained which will be discussed in the subsequent section.

Figure 7-5 (a) Experimental Transient Pressure and (b) Instantaneous Flow Rate Generated by the RBP (c) Summarized Pumping Process

Forward Stroke

At time t_0 , similar to the scenarios discussed in section 7.1.1 and 7.1.2, the pumping unit reaches its BDC, the pumping unit is at stage (i) as shown in Figure 7-5 (c), where the ball is pressed onto the front cap. The RBP is in the ‘open’ condition and fluid flows through the pumping unit. At this instance, P_{down} is negative while the P_{up} is positive as shown in Figure 7-5(a) due to the previous stroke pumping process. Though the instantaneous flow rate is positive as shown in Figure 7-5(b), its magnitude is nearly zero as the fluid has lost most of its inertia. At the next instance, the pumping unit begins its forward stroke and the velocity of the pumping unit increases in the positive z -direction as shown in Figure 7-5(b).

The forward motion of the pumping unit causes P_{up} to decrease and P_{down} increases. After a fraction of a second, at time t_1 , it can be observed from Figure 7-5(a) that P_{up} and P_{down} intersect and subsequently, P_{down} continues to increase while P_{up} decreases. The differences in the pressures results in a net force acting onto the ball in the negative z -direction. This will cause the ball to come into contact with the rear cap. They will then move forward together. At this instance, the pumping unit of the RBP is in the ‘closed’ condition as illustrated in Figure 7-5 (c) (ii). Fluid will be pushed forward as a result the flow rate starts to increase as shown in Figure 7-5(b) from time t_1 onwards.

After the sudden acceleration from time t_1 to t_2 , the fluid reaches its maximum acceleration and thus P_{up} reaches a minimum while P_{down} reaches its maximum as shown in Figure 7-5(a). After time t_2 , the fluid is still accelerating in the positive z -direction, but with a decreasing magnitude, thus, an increment in P_{up} and a decrement in P_{down} was observed in Figure 7-5(a). Furthermore, as shown in Figure 7-5(b), the

instantaneous flow rate continues to increase due to the positive acceleration, but, at a lesser rate due to the decrease in acceleration.

The pumping unit, upon reaching its maximum velocity at t_3 as shown in Figure 7-5(b), starts to decelerate, i.e. the velocity of the pumping unit starts to decrease. According to the working principle illustrated in Figure 7-5 (c), the RBP should be in a ‘closed’ condition and the ball should be in contact with the rear cap until the end of the forward stroke even though the pumping unit is decelerating, and due to the deceleration of the pumping unit, the fluid should suffer from deceleration as well. However, it can be observed from Figure 7-5(b) that the instantaneous flow rate is still increasing even though the pumping unit is decelerating and the velocity of the pumping unit is decreasing. Thus, it is suspected that the RBP does not work exactly as the proposed working principle as illustrated in Figure 7-5 (c). Due to the deceleration of the pumping unit and the inertia of the flowing fluid, the fluid continues to flow in the positive z-direction and it pushes the ball in the positive z-direction, thus causing it to detach from the rear cap. The RBP becomes partially ‘open’ instead of being in the ‘closed’ condition until the end of the forward stroke, as illustrated in Figure 7-5 (c). This allows the fluid to flow through the pumping unit due to its own inertia as a result of the previous acceleration. Thus, it can be observed from Figure 7-5(b) that the instantaneous flow rate continues to increase even though the pumping unit has slowed down. The acceleration of the fluid is decreasing and thus P_{up} continues to increase and P_{down} continues to decrease as shown in Figure 7-5(a).

After t_4 , the fluid starts to lose its inertia and decelerates. It can be observed from Figure 7-5(b) that the instantaneous flow rate decreases from t_4 onwards. It can also be

observed that P_{up} increases while P_{down} decreases as shown in Figure 7-5(a) due to the deceleration of the fluid until the end of the forward stroke.

Backward Stroke

When the pumping unit reaches its maximum displacement at the end of the forward stroke at t_5 , i.e TDC, the velocity of the pumping unit is zero. Inertia effect continues to cause the fluid to move forward while the pumping unit starts to move in the backward direction. The fluid inertia continues to push the ball forward and the ball comes in contact with the front cap and the pumping unit is in a fully ‘open’ condition. Fluid continues to flow forward through the pumping unit during the backward stroke. From Figure 7-5(b), it is noted that the volume flow rate is decreasing gradually regardless of the motion of the pumping unit. This can be explained by the fact that during the backward stroke, the fluid is losing its forward inertia. Upon reaching the BDC at t_6 , the whole pumping process starts all over again as described above.

Based on the transient pressure variations and instantaneous flow rate obtained from the three studies in section 7.1.1, 7.1.2 and 7.1.3 respectively, it can be observed that the actual working principle of the RBP is slightly different from the proposed working principle as discussed in section 3.2.2. In general, results from both theoretical and computational models are in good agreement with experimental results. The unique pumping characteristics of the RBP allows fluid to flow in the forward direction even when the pumping unit is decelerating during the forward stroke or when it is moving in the backward direction. This allows the RBP to induce additional flow rate as compared to the conventional reciprocating pump.

7.1.4 Pressure Variations and Instantaneous Flow Rate Relationship

To facilitate the explanation of the changes of transient pressure variations and instantaneous flow rate or instantaneous fluid velocity of the RBP, several equations that have been discussed in chapter 4 are further simplified to show the relationship between the transient pressure and flow rate.

As discussed in chapter 4, the Navier-Stoke equation can be reduced to z-direction only as shown in equation (4-24) where

$$\rho \frac{\partial u_z}{\partial t} = -\frac{\partial P}{\partial z} + \mu \left(\frac{\partial^2 u_z}{\partial r^2} + \frac{1}{r} \frac{\partial u_z}{\partial r} \right)$$

In the simplified mathematical model, viscous terms are replaced with frictional losses term and thus equation (4-24) becomes equation (4-29) where

$$\rho \frac{\partial u_z}{\partial t} = -\frac{\partial P}{\partial z} - \frac{1}{2} \frac{f}{D} \rho u_z^2$$

To illustrate the relationship between the transient pressure variations and the flow rate, equation (4-29) can be further simplified by neglecting the frictional losses term and thus equation (4-29) becomes

$$\rho \frac{\partial u_z}{\partial t} = -\frac{\partial P}{\partial z} \quad (7-1)$$

Referring to Figure 4-5 discussed in Chapter 4, the pressure variation upstream, P_{up} , and pressure variation downstream, P_{down} , of the RBP can then be approximated by the following equations.

$$P_{up} = -\rho \left(\frac{\partial u_z}{\partial t} \right) (z_1 - z_0) \quad (7-2)$$

$$P_{down} = \rho \left(\frac{\partial u_z}{\partial t} \right) (z_3 - z_2) \quad (7-3)$$

Refer to equations (7-2) and (7-3), the pressure variation at the upstream, P_{up} , and downstream, P_{down} , of the RBP can be approximated by using the rate of change of the fluid velocity.

Based on the instantaneous velocity obtained from the mathematical model or CFD simulation, $\frac{\partial u_z}{\partial t}$ can be obtained. Experimentally, the instantaneous fluid velocity can be obtained based on the instantaneous flow rate as shown in Figure 7-5 (b). Curve fitting using Fourier series can be used to determine the instantaneous fluid velocity and hence the rate of change of the instantaneous fluid velocity can be obtained.

By substituting the actual length of the pipeline at the upstream, $(z_1 - z_0)$ and downstream, $(z_3 - z_2)$ of the RBP, an approximation of P_{up} and P_{down} can be obtained.

Figure 7-6 Comparison of the Pressure Variations Obtained from Mathematical model and Equations (7-2) & (7-3) at (a) Downstream and (b) Upstream

Figure 7-7 Comparison of the Pressure Variations Obtained from CFD and Equations (7-2) & (7-3) at (a) Downstream and (b) Upstream

Figure 7-8 Comparison of the Pressure Variations Obtained from Experiment and Equations (7-2) & (7-3) at (a) Downstream and (b) Upstream

Figure 7-6 compares the P_{up} and P_{down} obtained from the mathematical model and that obtained from equations (7-2) and (7-3) respectively. The graph of P_{up} and P_{down} obtained from the CFD simulation and equations (7-2) and (7-3) are shown in Figure 7-7 while Figure 7-8 shows the P_{up} and P_{down} measured experimentally with the P_{up} and P_{down} computed using Equations (7-2) and (7-3).

It is noted that the P_{up} and P_{down} that are obtained directly from the three studies agree qualitatively with their respective P_{up} and P_{down} that are obtained from equations (7-2) and (7-3) with their respective $\frac{\partial u_z}{\partial t}$ in terms of the trend and the order of magnitude. The discrepancy between them might be due to the absence of the viscous terms in equations (7-2) and (7-3).

Basically, the pressure variations follow the first derivative of the instantaneous fluid velocity. Thus, in the subsequent section, the discussion will focus on the instantaneous flow rate, the average flow rate and the volumetric efficiency instead of the pressure variations at the upstream, P_{up} and the downstream, P_{down} of the RBP.

7.1.5 Comparison of Instantaneous Flow Rate

Figure 7-9 shows the comparison of instantaneous flow rates obtained the three studies. It can be observed that the instantaneous flow rates obtained from mathematical model and CFD simulation agrees very well. For the first half of the forward stroke, the results from the mathematical model and CFD simulation are identical. During the later stage of the forward stroke, there is a slight discrepancy between the mathematical model and CFD simulation and this could be due to the simplification of the losses in the mathematical model. The mathematical model has underestimated the viscous effect.

When the pumping unit becomes partially ‘open’ during the later stage of the forward stroke, fluid flows through the pumping unit due to its inertia. However, underestimation of the viscous effect in the mathematical model results in a higher theoretical instantaneous flow rate as compared to that obtained from CFD simulation.

Figure 7-9 Comparison of Instantaneous Flow Rates Obtained from Mathematical model, CFD Simulation and Experiment

From Figure 7-9, the experimental instantaneous flow rate agrees qualitatively with instantaneous flow rate obtained from both mathematical model and CFD simulation. Small oscillations in the experimental instantaneous flow rate were observed during the first half of the forward stroke and it is suspected that when the ball comes into contact with the rear cap, the hole of the rear cap might not be fully sealed by the ball due to fabrication imperfection e.g. uneven surfaces on the rear cap, imperfect hole on the rear cap and etc. Another reason is that flow leakage occurs during the forward stroke. Leakage may be due to the impact force acting on the ball during the collision of the ball and rear cap. As the ball comes in contact or collides with the rear cap, the impact force might cause the ball to detach from the rear cap momentarily. However, due to the high pressure at the downstream, the ball is then pushed in the negative z-direction again. This phenomenon might happen repeatedly, thus causing fluctuation in the

experimental instantaneous flow rate during the first half of the forward stroke. This is not observed in both mathematical model and CFD simulation as the collision is not modeled in both these studies. The effect of the multiple collisions may also be the cause of a lower experimental instantaneous flow rate as compared to results obtained from the mathematical model and CFD simulation.

Figure 7-10 Comparison of (a) P_{down} and (b) P_{up} Obtained from Mathematical model, CFD Simulation and Experiment

Figure 7-10 shows the comparison of the P_{up} and P_{down} obtained from the three studies, namely, mathematical model, CFD simulation and experimental studies. Basically the predicted results are in good agreement with the experimental studies both in terms of trend and order of magnitude. The magnitudes of P_{up} and P_{down} obtained experimentally are smaller. From section 7.1.4, since P_{up} and P_{down} are directly proportional to the first

derivative of the instantaneous fluid velocity, smaller magnitudes of P_{up} and P_{down} in experimental studies are expected.

7.2 Effect on the Performance of RBP at Different Pumping

Frequency

Three studies have been conducted for different operating speeds of 100, 150, 200 and 250 rpm to evaluate the effect on generated flow rate with different operating speeds. In addition, for each speed, the studies have also been conducted with different back pressure ranging from 0, 250, 500 and 1000 Pa. The average flow rate and volumetric efficiency can be calculated by using equations (4-42) and (4-43) after obtaining the instantaneous flow rate.

Figure 7-11 shows the average flow rate against different operating speeds for different back pressures for three studies.

As the operating speed increases, the average flow rate is expected to increase. For the mathematical model, it can be observed from Figure 7-11 (a) that the average flow rate at back pressure 0 Pa increases from 2.44 to 5.78 L/min with increasing operating speed. The average flow rate increases from 2.42 to 5.65 L/min at 250 rpm for CFD simulation while for experimental studies, the average flow rate increases from 2.27 to 5.25 L/min respectively. For all the three different studies, similar patterns can be observed at higher back pressures. In addition, it can be noted that the average flow rate at a particular speed is lower when the back pressure increases. This could be due to the higher adverse pressure that the pumping unit has to overcome to generate flow. This effect is more significant at lower operating speed between 100 and 200 rpm while at higher speed of 250 rpm, the differences in the average flow rate generated are much smaller.

This can be explained by the fact that at a higher rotational speed, the pressure generated by the pumping unit is much higher and the effects of back pressure are reduced.

Figure 7-11 Average Flow vs Rotational Speed at Different Back Pressures for (a) Mathematical model (b) CFD Simulation and (c) Experiment

Figure 7-12 shows comparison of average flow rate obtained from three studies at different rotational speeds. In general, it can be observed that the average flow rates obtained from the three studies agree qualitatively in terms of trend and magnitude. The mathematical model tends to overestimate the average flow rate at all operating conditions and this is due to an underestimation of the viscous effect. The collision phenomena and the effect of leakage as presented in section 7.1.5 have resulted in the lowest experimental average flow rate.

Figure 7-12 Comparison of Average Flow Rate Obtained from Mathematical model, CFD Simulation and Experiment at Different Rotational Speed

In addition, the differences in the average flow rate between the mathematical model and CFD simulation results are negligible for all rotational speeds. However, as the rotational speed increases, the differences between the experimental results and theoretical results become bigger. It is suspected that mathematical model and CFD simulation tend to underestimate the losses within the entire system. These losses are

proportional to the square of fluid velocity. Thus, as rotational speed increases, these losses increase as well. Thus, the results obtained from mathematical model and CFD simulation are always higher than those obtained experimentally.

Figure 7-13 Mathematical model Volumetric Efficiency vs Rotational Speed at Different Back Pressures

Figure 7-13 shows the volumetric efficiency against different operating speeds at different back pressures for the three studies. The average flow rate and volumetric efficiency can be calculated by using equations (4-42) and (4-43) after obtaining the instantaneous flow rate.

The results show that at back pressures lower than 250 Pa, the volumetric efficiency decreases gradually with increasing operating speeds. For the mathematical model, the volumetric efficiency decreases from 172.15% to 162.98% at back pressure of 0 Pa. For

the CFD model, at the same back pressure, it decreases slightly from 170.88% to 159.31% while for the experimental studies, it decreases from 159.89% to 148.12%. The results obtained from the three studies show that at low back pressures, the volumetric efficiency decreases with increasing rotational speed. As the rotational speed increases, the flow rate and the fluid velocity increases, thus, resulting in higher frictional losses, hence, reducing the volumetric efficiency.

However, at higher back pressures such as 750 and 1000 Pa, it can be observed that the volumetric efficiency is lower at rotational speeds such as 100 rpm. As the rotational speed increases to 150 rpm, the volumetric efficiency increases slightly. Subsequently as the rotational speed increases further to 200 or 250 rpm, the volumetric efficiency decreases. At low rotational speed such as 100 rpm, the pressure generated by the pumping unit is insufficient to overcome higher adverse pressure thus reducing the flow generated during the backwards stroke, leading to a lower volumetric efficiency. However, as rotational speed increases to 150 or 200 rpm, the pressure generated is sufficient to overcome the adverse pressure and thus the volumetric efficiency increases. As the rotational speed increases further, the viscous effect and frictional losses increases with increasing flow rate, thus leading to a decrease in volumetric efficiency. However, the decrease in volumetric efficiency is smaller than those observed for operations at lower back pressure. This shows that for higher adverse pressure, the effect of higher rotational speed does not significantly affect the volumetric efficiency.

7.3 Effect on the Performance of RBP at Different Back Pressure

Three studies have been conducted for different back pressures ranging from 0, 250, 500, 750 to 1000 Pa to determine the flow rate generated. In addition, for each back pressure, the rotational speed is varied from 100 to 250 rpm respectively.

Figure 7-14 shows the variation of the average flow rate at different back pressures at different operating speeds for the three studies.

Figure 7-14 Average Flow vs Back Pressures at Different Rotational Speed for (a) Mathematical model (b) CFD Simulation and (c) Experiment

In general, it is noted that for a given speed, for example 100 rpm, as the back pressure increases from 0 to 1000 Pa, the results from the mathematical model show that the average flow rate decreases monotonically from 2.44 to 1.75 L/min. Similar trend can also be observed for the results obtained by the other two studies. From the CFD

`simulation results for the same operating condition, the average flow rate decreases from 2.44 to 1.75 L/min for increasing back pressure while for the experimental studies, it decreases from 2.27 to 1.72 L/min respectively. Similar trend can be observed when the RBP operates at higher rotational speeds.

Figure 7-15 shows comparison of the average flow rate obtained from the three methods at different back pressures. In general, it can be observed that the average flow rates obtained from the three studies agree qualitatively in terms of trend and magnitude. Consistent with the previous observations, the mathematical model overestimates the average flow rate at different operating condition due to the underestimation of the viscous effect. The collision phenomenon and leakage problem result in the lowest average flow rate measured experimentally.

In addition, it can also be observed from Figure 7-15 that differences of the average flow rate between the mathematical model and CFD simulation results are very small for different back pressures. However, there is an observable difference between the experimental results and the theoretical results due to the reasons mentioned above. Nevertheless, as the back pressure increases, the results obtained from all the three methods decrease at almost the same rate.

The graphs of volumetric efficiency as a function of back pressure are shown in Figure 7-16 and it is observed that as the back pressure increases, the volumetric efficiency decreases. Generally, the back pressure acts as a resistance to flow during the pumping process and as it increases, the fluid encounters higher flow resistance, thus resulting in a lower volumetric efficiency.

Figure 7-15 Comparison of Average Flow Rate Obtained from Mathematical model, CFD Simulation and Experiment at Different Back Pressures

For the mathematical model, it can be observed from Figure 7-16 (a) that as the back pressure increases, the volumetric efficiency decreases from 172.15% to 123.17% when the RBP is running at 100 rpm. For the same rotational speed and decreasing back pressure, the CFD simulation shows that the volumetric efficiency decreases from 170.88% to 123.66% and for experimental studies, it decreases from 159.89% to 121.42% respectively.

It can also be observed from Figure 7-16 that the back pressure effect is more significant when the RBP is operating at a lower rotational speed as the volumetric efficiency decreases at a steeper rate. At lower rotational speed such as 100 rpm, the pressure generated may not be sufficient to overcome the back pressure, thus leading to lower volumetric efficiency as back pressures increases. However, when the RBP is

operating at higher rotational speeds, the pressure generated by the pumping unit is much higher and the effects of back pressure are reduced.

Figure 7-16 Volumetric Efficiency vs Back Pressures at Different Rotational Speed for (a) Mathematical model (b) CFD Simulation and (c) Experiment

Figure 7-17 shows that the volumetric efficiencies obtained from the three studies at different back pressures are in good agreement though in general, the value obtained from experimental studies are consistently lower due to reasons explained above.

Figure 7-17 Comparison of Volumetric Efficiency Obtained from Mathematical model, CFD Simulation and Experiment at Different Back Pressures

7.4 Parametric Studies

In addition to the performance characteristics presented in sections 7.1.1 to 7.3, parametric studies were also conducted by using the mathematical model and experimental studies to evaluate the effect on the performance when different specifications of the RBP were used. These factors include the ball weight, the diameter of rear cap hole and the length of the RBP casing. Parametric studies using CFD simulation were not conducted as different meshes need to be generated for different geometries and it is very laborious. In addition, the results on performance characteristics presented in sections 7.1.1 to 7.3 shows that the discrepancy between the mathematical model and CFD simulation is negligible.

7.4.1 Effect on the Performance of RBP Due to Different Balls' Weight

As discussed in section 6.1.2c), balls of different materials and densities are used as part of the RBP mechanism. In this section, the rear cap hole of diameter 8.5 mm and casing length of 11.5 mm are kept constant while varying the density and the weight of the ball to evaluate its effect on the performance of the RBP.

In this study, mathematical simulations and experimental studies were conducted by using different types of ball. The materials of the ball used in this study are PA66, Si_3N_4 , Al_2O_3 , ZrO_2 and tungsten carbide respectively where PA66 being the lightest ball while tungsten carbide being the heaviest ball. In addition, for each type of ball, the simulations and experiments were conducted at different operating speeds namely 100 and 150 rpm and at back pressure of 0 and 1000 Pa respectively. Figure 7-18 shows the relationship between the average flow rate generated by the RBP and the types of ball at different operating conditions while Figure 7-19 shows the relationship between the volumetric efficiency and the type of ball at different operating conditions.

Figure 7-18 Average Flow vs Different Ball Weight for
(a) Mathematical model and (b) Experiment

Figure 7-19 Volumetric Efficiency vs Different Ball Weight for
(a) Mathematical model and (b) Experiment

Figure 7-18 (a) and Figure 7-19 (a) show the prediction of the flow rate and the volumetric efficiency by mathematical model. In general, the results obtained are similar and consistent with each other. The flow rate and volumetric efficiency remain almost constant regardless of the material used for the ball. However, a closer examination of the results reveals that the average flow rate and volumetric efficiency actually decrease slightly with increasing ball weight. For example, at 100 rpm with a back pressure equal to 0 Pa, the volumetric efficiency decreases from 171.88% when using the PA66 to 169.23% when using the tungsten carbide. Similarly at a 150 rpm with back pressure equal to 1000 Pa, the volumetric efficiency decreases from 148.46% when using the lightest ball to 147.46% when using the heaviest ball. Similar trend can be observed at different operating conditions.

Similarly, Figure 7-18 (b) and Figure 7-19 (b) show that the flow rate and the volumetric efficiency decrease slightly as the weight of the ball used in the experiment increases. At 100 rpm with a back pressure equal to 0 Pa, the volumetric efficiency decreases from 159.48% when using the PA66 to 149.23% when using the tungsten carbide. At 150 rpm with a back pressure equal to 1000 Pa, the volumetric efficiency

decreases from 135.48% when using the lightest ball to 130.58% when using the heaviest ball.

Accordingly to Newton's second law, the force acting on an object is equal to the mass of the object multiply by the acceleration experienced by the object. If the force remains almost constant, the heavier the ball, the lesser the acceleration experienced by the ball. In this case, it is expected that a heavier ball would take a longer time to move to the rear cap or front cap respectively during the pumping operation, thus affecting the flow rate generated or the volumetric efficiency. However, the results show that the density of the ball does not affect significantly the volume flow rate or the volumetric efficiency. As the internal dimension of the pumping unit is relatively short, it is speculated that the reduction in the acceleration due to an increase in mass of the ball does not affect the results significantly.

Closer examination of the results reveals that for increasing ball weight, the theoretical average flow rate and volumetric efficiency still remain almost the same while that obtained from the experiments show a slight decline.

The mathematical model may have overestimated the forces acting on the ball due to the simplification made on the geometry of the pumping unit. In reality, a heavier ball will take a longer time to transit from fully 'closed' to fully 'open' condition. Due to a slower transition, lesser flow will be induced and thus a lower average flow rate or volumetric efficiency is obtained. However, an overestimation of the forces acting on the ball in the mathematical model will result in a shorter transition period than the actual experiment. Thus, as heavier balls are used, the theoretical results do not show any significant decrease in the average flow rate.

7.4.2 Effect on the Performance of RBP Due to Different Diameter of the Rear Cap Hole

As discussed in section 6.1.2d), the hole on the rear cap is an important feature in the design of RBP. Different rear cap hole diameters may affect the performance of the RBP. In this section, the casing length of 11.5 mm and the weight of ball are kept constant while varying the diameter of the hole on the rear cap to evaluate its effect on the performance of the RBP.

Figure 7-20 Average Flow vs Rear Cap Hole Diameter for (a) Mathematical model and (b) Experiment

Figure 7-21 Volumetric Efficiency vs Rear Cap Hole Diameter for (a) Mathematical model and (b) Experiment

It can be observed from Figure 7-20 (a) and Figure 7-21 (a) that for the mathematical model, the average flow rate and the volumetric efficiency increase with increasing rear cap hole diameter. For the RBP operating at 100 rpm and 0 Pa back pressure, as the

diameter of the hole is increased from $\varnothing 7.0$ mm to $\varnothing 8.5$ mm, the average flow rate increases from 2.14 to 2.44 L/min and the volumetric efficiency increases from 151.26% to 172.15% respectively. Similar trend can be observed for increasing operating speeds. According to Figure 7-20 (b) and Figure 7-21 (b), the experimental results also show that the average flow rate and the volumetric efficiency are the lowest when the rear cap hole diameter of $\varnothing 7.0$ mm was used at four different operating conditions. For increasing rear cap hole diameter, the experimental average flow rate and volumetric efficiency increase until the diameter of the rear cap hole is equal to $\varnothing 8.5$ mm. For both mathematical model and experimental results, it can be observed that when rear cap hole diameter increases from $\varnothing 8.5$ to $\varnothing 9.0$ mm, the average flow rate and volumetric efficiency remain almost the same or decreases slightly. In addition, the results obtained from the two studies are in good agreement with each other.

It is speculated that as the diameter of the hole on the rear cap increases, the area on the rear cap for the fluid to flow through the pumping unit increases, thus resulting in a higher flow rate induced especially during the backward stroke. However, this is only valid when the diameter of the rear cap hole is less than 8.5 mm. As discussed in section 3.2.2, during the backward stroke, fluid continues to flow in the forward direction through the 'open' RBP due to its inertia. The flow path of the induced flow during the backward stroke of the RBP depends on three regions namely (i) the flow area of the hole on the rear cap, (ii) the flow area between the hole of the rear cap and the ball as discussed in section 4.3.3 and (iii) the flow area between the casing and the ball as shown in Figure 7-22. Thus, the induced flow rate, the average flow rate and the volumetric efficiency are not only affected by the diameter of the hole on the rear cap, but also depend on the relative distance between the rear cap and the ball as illustrated in section 4.3.3 and the cross sectional area between the casing and the ball as illustrated

in Figure 7-22 below. As fluid passes through the pumping unit, the average flow rate will be affected by the region with the smallest area, i.e. the critical flow area.

Figure 7-22 Cross Sectional Area for Fluid Flow of the RBP Casing

Diameter of Rear Cap Hole (mm)	(i) Flow Area of the hole on the Rear Cap (m^2)	(ii) Flow Area between the hole on the rear cap and the ball of 'open' RBP (m^2)	(iii) Flow Area between the casing and the ball as shown in Figure 7-22 (m^2)
7.00	3.848×10^{-05}	7.739×10^{-05}	5.554×10^{-05}
7.50	4.418×10^{-05}	7.739×10^{-05}	5.554×10^{-05}
8.00	5.027×10^{-05}	7.739×10^{-05}	5.554×10^{-05}
8.50	5.675×10^{-05}	7.739×10^{-05}	5.554×10^{-05}
9.00	6.362×10^{-05}	7.739×10^{-05}	5.554×10^{-05}

Table 7-1 Comparison Between of Flow Area at Different Diameter of Rear Cap Hole and Flow Area of RBP Casing

From Table 7-1, it can be observed that for rear cap hole diameters of less than 8.50 mm, the flow area on the rear cap hole is the smallest among the three regions as shown. Thus, the induced flow rate is restricted by the smallest flow area which is the flow area (i). Hence, as the diameter of rear cap hole increases, the critical flow area increases correspondingly, leading to an increase in the average flow rate and the volumetric efficiency. When the diameter of the rear cap hole is larger than 8.50 mm, the flow area (iii) becomes the smallest among the three flow areas. Hence, even though the diameter of the rear cap hole increases beyond Ø8.50 mm, the flow rate and volumetric efficiency remains almost the same.

7.4.3 Effect on the Performance of RBP Due to Different Length of the Casing

As discussed in section 6.1.2e), the length of the casing of the RBP may affect the performance of the RBP. In this section, the rear cap hole diameter is fixed at 8.5 mm and the weight of ball is kept constant while the length of the RBP casing is varied to evaluate its effect on the performance of the RBP.

The results obtained from mathematical model results are presented in Figure 7-23 (a) and Figure 7-24 (a). As the length of the RBP casing increases from 9.50 to 11.50 mm, the average flow rate increases from 1.97 to 2.44 L/min and the volumetric efficiency increases from 138.64% to 172.15% when the RBP is running at 100 rpm at 0 Pa back pressure. Similar trend can be observed at other operating conditions.

The results obtained from the mathematical model are consistent with those obtained experimentally. From Figure 7-23 (b) and Figure 7-24 (b), as the length of the RBP casing increases from 9.50 mm to 11.50 mm, the experimental average flow rate increases from 1.95 to 2.27 L/min and the volumetric efficiency increases from 137.78% to 159.88% when the RBP is running at 100 rpm at 0 Pa back pressure.

Figure 7-23 Average Flow vs Casing Length for
(a) Mathematical model and (b) Experiment

Figure 7-24 Volumetric Efficiency vs Casing Length for
(a) Mathematical model and (b) Experiment

Length of RBP Casing (mm)	(i) Flow Area of the hole on the Rear Cap (m^2)	(ii) Flow Area between the hole on the rear cap and the ball (m^2)	(iii) Flow Area between the casing and the ball as shown in Figure 7-22 (m^2)
9.50	5.675×10^{-05}	4.287×10^{-05}	5.554×10^{-05}
11.50	5.675×10^{-05}	7.739×10^{-05}	5.554×10^{-05}
13.50	5.675×10^{-05}	1.040×10^{-04}	5.554×10^{-05}
15.50	5.675×10^{-05}	1.333×10^{-04}	5.554×10^{-05}
17.50	5.675×10^{-05}	1.621×10^{-04}	5.554×10^{-05}

Table 7-2 Comparison Between of Flow Area at Different Casing Length with Flow Area of the hole on the rear cap and Flow Area of RBP Casing

For a fully ‘open’ RBP, as the length of the RBP casing increases, the distance between the hole on the rear cap and the ball increases, thus the flow area between the hole of the rear cap and the ball, namely flow area (ii) as discussed in section 7.4.2 will increase as well. As flow area (ii) increases, the induced flow is expected to increase. However, as discussed in section 7.4.2, the induced flow depends on three different flow areas, namely flow area (i), (ii) and (iii) as discussed previously. Table 7-2 shows that when the length of RBP casing is equal to 9.50 mm, flow area (ii) is the critical flow area that restricts the induced flow. As the length of the RBP casing increases to 11.50 mm, flow area (ii) is larger than flow area (iii), thus, flow area (iii) becomes the critical flow area which restricts the flow rated generated. Hence, it can be observed from Figure 7-23 and Figure 7-24 that the average flow rate and volumetric flow efficiency increases as the

length of the RBP increases from 9.50 mm to 11.50 mm because the critical flow area increases from $4.287 \times 10^{-05} \text{ m}^2$ to $5.554 \times 10^{-05} \text{ m}^2$. As the length of the RBP casing increases further, the average flow rate and the volumetric efficiency decreases gradually. It is suspected that as the length of the RBP casing increase, frictional losses within the pumping unit of the RBP increase, thus leading to a decrease in the average flow rate and the volumetric efficiency even though the critical flow area remains constant.

In this section, the performance characteristics of the RBP have been discussed based on the results obtained from the mathematical model presented in Chapter 4. The transient pressure variations, instantaneous flow rate and the average flow rate generated by the RBP at different pumping speeds and back pressures are discussed in details. In addition, parametric studies with different ball's weight, different rear cap hole diameters and different casing lengths have also been investigated. In the next section, results obtained from CFD simulation will be presented.

7.5 CFD Pressure Flow Field and Experimental Flow Visualization

In this section, results on the CFD pressure distribution and flow pattern will be presented. CFD simulations were conducted to investigate the pressure distribution and flow pattern generated by the RBP. By understanding the pressure distribution and the flow field generated by the RBP at different instances during its operation, the unique pumping action of the RBP can be explained in a more comprehensive perspective. Subsequently, experimental flow visualizations were conducted to validate the results obtained from CFD simulation.

7.5.1 CFD Pressure Flow Field

The pressure contour and the streamlines generated by the RBP at different instances during the forward and backward stroke are shown in Figure 7-25 and Figure 7-28 respectively.

Forward Stroke

At $t=0$ sec as shown in Figure 7-25, fluid is flowing in the forward direction even though the pumping unit is moving backwards and reaches its BDC. The pressure is distributed uniformly at the upstream and downstream of the pumping unit. At this instance, the ball is neither in contact with the front cap nor the rear cap. It can be observed that the fluid rushes out from the middle of the front cap and vortices are formed near to the pipe wall at the downstream of the pumping unit. Further downstream, there is another pair of vortices which is being formed due to the backward motion of the RBP. This will be discussed in the later part of this section.

At $t=0.024$ sec, the pressure downstream of the pumping unit becomes slightly higher than the pressure upstream of the pumping unit due to the forward motion on the

pumping unit. This gives rise to a net force acting on the ball in the negative z -direction. Due to the relative displacement between the pumping unit and the ball, it can be observed from Figure 7-25 at $t=0.024$ sec that the ball comes into contact with the rear cap and the RBP is now in the fully 'closed' condition.

The ball is then pushed by the pumping unit and they move together in the positive z -direction. From this instance onwards, a trapped volume of fluid is formed in front of the 'closed' pumping unit. As the pumping unit continues to move in the positive z -direction, the trapped volume of fluid is pushed forward by the motion of the 'closed' RBP. Due to the sudden acceleration of the fluid, it can be observed from Figure 7-25 that the pressure downstream of pumping unit increases significantly due to the compression effect while the pressure upstream of the pumping unit decreases due to the suction effect caused by the moving RBP. It can also be observed that a pair of vortices is formed upstream of the 'closed' RBP. Before the ball comes into contact with the rear cap, fluid is allowed to flow through the hole or opening on the rear cap. However, at $t=0.06$ sec, after the ball has come into contact with the rear cap, the hole for fluid flow is sealed. The fluid that is originally flowing through the hole is forced to flow in the negative z -direction and hence form a pair of vortices at the upstream of the 'closed' RBP.

Figure 7-25 Pressure and Flow Field Generated by RBP during Forward Stroke

The pumping unit continues to move in the forward direction in the 'closed' condition until $t=0.204$ sec. The fluid continues to accelerate in the positive z -direction after $t=0.060$ sec, but with a decreasing magnitude. Figure 7-25 shows that the pressure downstream of the 'closed' RBP decreases from $t=0.060$ sec to $t=0.204$ sec while the pressure at the upstream of the 'closed' RBP increases slightly. The vortices upstream of the 'closed' RBP continue to expand due to the forward motion of the pumping unit while the vortices downstream of the 'closed' RBP disappear as fluid is being pushed in the positive z -direction by the 'closed' RBP.

The 'closed' RBP, upon reaching its maximum velocity, starts to decelerate. Due to the deceleration of the pumping unit, the ball is pushed forward by the fluid in the positive z -direction. At this instance, the hole on the rear cap opens and fluid continues to flow in the positive z -direction due to its inertia. Thus, a higher flow rate can be achieved by this unique pumping action. As the ball is being pushed towards the front cap, the pumping unit becomes fully 'open' at $t=0.276$ sec. It can be observed from Figure 7-25 that the size of the vortices at the upstream of the pumping unit decrease gradually and subsequently disappear at $t=0.300$ sec as fluid continues to flow in the forward direction. An additional two pairs of vortices are formed downstream of the fully 'open' RBP at $t=0.300$ sec. At this instance, the fully 'open' pumping unit has come to rest at the end of the forward stroke while fluid is still flowing forward due to its inertia. The first pair of vortices is formed near the front cap and the pipe wall due to separation caused by the fluid flow through a sudden expansion which is similar to the example shown in Figure 7-26. The second set of vortices is formed near the center of the pipe due to the separation caused by fluid flow over the sphere and the supporting legs of the front cap which is similar to the example shown in Figure 7-27 where a pair of vortices is observed downstream of the sphere.

Figure 7-26 Flow Through a Sudden Expansion [55]

Figure 7-27 Flow Past a Cylinder [54]

Backward Stroke

At $t=0.300$ sec, the pumping unit of RBP reaches the TDC. As explained earlier, there are two pairs of vortices formed at the downstream of the pumping unit. At this instance, the pumping unit starts to move in the negative z -direction. From Figure 7-28, the pressure at the upstream of the pumping unit is higher than the pressure at the downstream of the pumping unit. This gives rise to a net force acting in the positive z -direction which pushes the ball onto the front cap of the pumping unit. This keeps the RBP in a fully 'open' condition during the backward stroke. Fluid continues to flow through the pumping unit due to its inertia. Thus, additional flow is induced as compared to the conventional reciprocating pump.

It can also be observed from Figure 7-28 that as the velocity of the pumping unit continues to increase in the negative z-direction, the two pairs of vortices start to expand from $t=0.300$ sec to $t=0.540$ sec. As the pumping unit travels in the negative z-direction, while fluid flows in the positive z-direction, the relative velocity between the pumping unit and the fluid increases, thus causing the vortices to grow.

Upon reaching its maximum velocity in the negative z-direction, the pumping unit continues to move in the same direction but with a slower velocity until it comes to rest at BDC. At the same time, the fluid gradually loses its inertia and its velocity decreases. Thus, the relative velocity between the pumping unit and the fluid decreases as well. It can be observed from Figure 7-28 that from $t=0.576$ sec onwards, the vortices downstream of the RBP become smaller as the relative velocity between the pumping unit and the fluid decreases.

The analysis of the pressure distribution and flow field of the RBP provides a better understanding of the unique pumping action of the RBP which enables it to induce a higher average flow rate as compared to the conventional reciprocating pump.

Figure 7-28 Pressure and Flow Field Generated by RBP during Backward Stroke

7.5.2 Experimental Flow Visualization

The flow fields around the RBP were captured using the experimental flow visualization setup as discussed in section 6.2. The flow patterns were taken with a camera setting of aperture F4.2 and shutter speed 1/60 seconds. As mentioned in section 6.2.2, the flow pattern will be captured at three different zones. At each zone, there are three possible scenarios for the pumping unit of the RBP, namely (i) the ball is pressed onto the rear cap, (ii) the ball is in the middle of the casing and (iii) the ball is pressed onto the front cap. Table 7-3 summarized the possible scenarios at different zones.

	Zone 1	Zone 2	Zone 3
Scenario (i) (Ball in contact with rear cap)
Observation	Observed	Observed	Observed
Scenario (ii) (Ball is neither in contact with rear cap nor front cap)
Observation	Observed	Not Observed	Observed
Scenario (iii) (Ball in contact with front cap)
Observation	Observed	Observed	Observed

Table 7-3 Summary of Flow Patterns Captured in Flow Visualization Experiment

The photos obtained from the flow visualization experiments will also be used to validate the flow field predicted by the CFD simulations.

Based on the results obtained from the CFD simulations, the forward stroke can be divided three zones. The pumping unit of the RBP will be in zone 1 (corresponds to zone 1 in the flow visualization experiment) from $t=0$ sec to $t=0.096$ sec. Subsequently, the pumping unit of the RBP will move into zone 2 from $t=0.096$ sec to $t=0.204$ sec. Finally the pumping unit of the RBP will enter zone 3 from $t=0.204$ sec to $t=0.300$ sec.

Similarly, the backward stroke can also be divided into three zones. The pumping unit of the RBP will be in zone 3 (corresponds to zone 3 in the flow visualization experiment) from $t=0.300$ sec to $t=0.396$ sec. Subsequently, the pumping unit of the RBP will move into zone 2 from $t=0.396$ sec to $t=0.504$ sec. Finally the pumping unit of the RBP will enter zone 1 from $t=0.504$ sec to $t=0.600$ sec.

Forward Stroke – Zone 1

From Figure 7-25, from $t=0$ sec to $t=0.096$ sec, the pumping unit is in zone 1. Two possible scenarios can happen during this period; (i) - ball in contact with rear cap and scenario (ii) ball is neither in contact with rear cap nor front cap.

Figure 7-29 compares the CFD flow field extracted at $t=0.012$ sec and experimental flow visualization photograph taken at zone 1 with scenario (ii), i.e. the ball is neither in contact with the rear cap nor the front cap. It can be observed that the flow patterns between Figure 7-29 (a) and (b) are similar. At $t=0.012$ sec, the pumping unit has just started its forward stroke and fluid is still flowing through the partially ‘open’ pumping unit due to its inertia. It can be observed from Figure 7-29 (a) and (b) that fluid is rushing through the opening on the rear cap and flows downstream.

Figure 7-29 (a) CFD Pressure Flow Field at $t=0.012$ sec vs (b) Flow Visualization Photos Taken at Zone 1 –Scenario (ii) Ball is in Contact with neither Front Cap nor Rear Cap

Figure 7-30 shows the CFD flow field at $t=0.060$ sec and the experimental photograph taken at zone 1 with scenario (i), i.e. the ball is in contact with the rear cap. The flow patterns between Figure 7-30 (a) and (b) are almost identical. At $t=0.060$ sec, the RBP is in fully ‘closed’ condition. As it moves forward, the trapped volume of fluid will be pushed in forward direction and a pair of vortices is formed upstream of the ‘closed’ RBP. Before the ball comes into contact with the rear cap, fluid is allowed to flow through the hole on the rear cap. However, at $t=0.06$ sec, as the ball comes into contact with the rear cap, the hole for fluid flow is sealed. The fluid that is originally flowing through the hole is forced to flow in the negative z -direction, forming a pair of vortices upstream of the ‘closed’ RBP.

Figure 7-30 (a) CFD Pressure Flow Field at $t=0.060$ sec vs (b) Flow Visualization Photos Taken at Zone 1 –Scenario (i) Ball in Contact with Front Cap

Forward Stroke – Zone 2

Based on Figure 7-25, from $t=0.096$ sec to $t=0.204$ sec, the pumping unit is in zone 2. There is only one possible scenario that can happen during this period of time which is scenario (i) - Ball in contact with rear cap.

Figure 7-31 shows a direct comparison between the CFD pressure flow field extracted at $t=0.180$ sec and the picture of flow visualization taken at zone 2 with scenario (i), i.e. the ball is in contact with the rear cap. It can be observed that the flow patterns between Figure 7-31 (a) and (b) are similar. At this instance, the ‘closed’ RBP continues to push the trapped volume of fluid in the positive z -direction. All the fluid particles are observed to be flowing in the positive z -direction.

Figure 7-31 (a) CFD Pressure Flow Field at $t=0.180$ sec vs (b) Flow Visualization Photos Taken at Zone 2 –Scenario (i) Ball in Contact with Rear Cap

Forward Stroke – Zone 3

Figure 7-25 shows that from $t=0.204$ sec to $t=0.300$ sec, the pumping unit is in zone 3. There are three possible scenarios that can happen during this period of time which is scenario (i) - Ball in contact with rear cap, (ii) ball is in contact with neither front cap nor rear cap and (iii) ball in contact with front cap.

The CFD pressure flow field and the experimental results at $t=0.204$ sec, at zone 3 with scenario (i), i.e. the ball is in contact with the rear cap, are shown in Figure 7-32. As it is just 0.024 sec ahead of the flow pattern discussed in Figure 7-31, the ‘closed’ RBP continues to push the trapped volume of fluid in the positive z-direction. It can be observed from Figure 7-32 that the flow pattern is similar to that in Figure 7-31.

Figure 7-32 (a) CFD Pressure Flow Field at $t=0.204$ sec vs (b) Flow Visualization Photos Taken at Zone 3 –Scenario (i) Ball in Contact with Rear Cap

Figure 7-33 (a) CFD Pressure Flow Field at $t=0.240$ sec vs (b) Flow Visualization Photos Taken at Zone 3 –Scenario (ii) Ball is in Contact with neither Front Cap nor Rear Cap

As discussed in section 7.1, the pumping unit starts to decelerate upon reaching its maximum velocity. The inertia of the fluid pushes the ball, thus causing the RBP to become partially ‘open’. Figure 7-33 shows the results obtained at $t=0.240$ sec at zone 3 with scenario (ii), i.e. the ball is neither in contact with the rear cap nor front cap. From

this figure, upstream of the pumping unit, fluid rushes through the partially ‘open’ pumping unit and continues to flow in positive z-direction.

Figure 7-34 (a) CFD Pressure Flow Field at $t=0.300$ sec vs (b) Flow Visualization Photos Taken at Zone 3 –Scenario (iii) Ball in Contact with Front Cap

As fluid inertia continues to push the ball in the positive z-direction, the ball is then pressed onto the front cap as shown in Figure 7-34 which shows results extracted at $t=0.300$ sec taken at zone 3 with scenario (iii), i.e. the ball in contact with the front cap. It can be observed that fluid continues to flow through the fully ‘open’ RBP through the hole on the rear cap due to its inertia. One pair of vortices is formed near the front cap and the pipe wall due to separation caused by the fluid flow through a sudden expansion as discussed in section 7.5.1. This is clearly shown in both Figure 7-34 (a) and (b). The other pair of vortices is formed near the center of the pipe due to the separation caused by fluid flow over the sphere and the supporting legs of the front cap which is similar to the example shown in Figure 7-27. However, this pair of vortices is not clearly seen in the experimental flow visualization picture as shown in Figure 7-34 (b).

Backward Stroke – Zone 3

According to Figure 7-28, from $t=0.300$ sec to $t=0.396$ sec, the pumping unit is in zone 3. There is only one possible scenario that can happen during this period of time which is scenario (iii) ball in contact with front cap as discussed in section 7.5.2.

Figure 7-35 (a) CFD Pressure Flow Field at $t=0.348$ sec vs (b) Flow Visualization Photos Taken at Zone 3 –Scenario (iii) Ball in Contact with Front Cap

The results at $t = 0.348$ sec are shown in Figure 7-35 at zone 3 with scenario (iii), i.e. the ball is in contact with the rear cap. At this instance, the pumping unit is in the fully ‘open’ condition while the fluid flows through the pumping unit when the pumping unit is moving in the negative z -direction. Similar to the flow pattern discussed in Figure 7-34, it can be observed from Figure 7-35 (a) and (b) that fluid continues to flow through the fully ‘open’ RBP through the hole on the rear cap due to inertia effect while the vortices formed during $t=0.300$ sec as shown in Figure 7-34 continue to grow as the pumping unit is moving in negative z -direction.

Backward Stroke - Zone 2

From $t=0.396$ sec to $t=0.504$ sec, the pumping unit is in zone 2 as shown in Figure 7-28.

There is only one possible scenario that can happen during this period of time which is scenario (iii) ball in contact with front cap as presented in section 7.5.2.

Figure 7-36 (a) CFD Pressure Flow Field at $t=0.468$ sec vs (b) Flow Visualization Photos Taken at Zone 2 –Scenario (iii) Ball in Contact with Front Cap

Figure 7-36 shows the results extracted at $t=0.468$ sec at zone 2 with scenario (iii), i.e. the ball is in contact with the rear cap. The pumping unit continues to move in the negative z -direction and enters zone 2. At this instance, the pumping unit is still in the fully ‘open’ condition. The flow patterns in Figure 7-36 (a) and (b) are similar to those presented in Figure 7-35. Fluid rushes through the fully ‘open’ pumping unit via the hole on the rear cap upstream of the pumping unit and flows downstream. Similarly, vortices are also observed in Figure 7-36 (a) and (b), and are further from the pumping unit as the pumping unit is moving away from the vortices in negative z -direction.

Backward Stroke – Zone 1

According to Figure 7-28, from $t=0.504$ sec to $t=0.600$ sec, the pumping unit is in zone 1. There are two possible scenarios that can happen during this period of time which is scenario (ii) ball is in contact with neither front cap nor rear cap and (iii) ball in contact with front cap as discussed in section 7.5.2.

Figure 7-37 (a) CFD Pressure Flow Field at $t=0.540$ sec vs (b) Flow Visualization Photos Taken at Zone 1 –Scenario (iii) Ball in Contact with Front Cap

The pumping unit continues to move in the negative z -direction and reaches zone 1. Similar flow patterns are observed in Figure 7-37 (a) and (b) which are results extracted at $t=0.540$ sec at zone 1 with scenario (iii), i.e. the ball is in contact with the front cap.

Figure 7-38 shows results extracted at $t=0.600$ sec at zone 1 with scenario (ii), i.e. the ball is in contact with neither front cap nor rear cap. As the pumping unit comes to the end of its backwards stroke, the fluid has gradually lost its inertia. At this instance, the force due to fluid inertia may be insufficient to press the ball onto the front cap and thus it detached from the front cap. However, fluid still continues to flow in the positive z -

direction through the partially ‘open’ pumping unit and vortices are still observable at the downstream in both Figure 7-38 (a) and (b).

Figure 7-38 (a) CFD Pressure Flow Field at $t=0.600$ sec vs (b) Flow Visualization Photos Taken at Zone 1 –Scenario (ii) Ball is in Contact with neither Front Cap nor Rear Cap

In this chapter, the performance of the RBP has been evaluated based on the transient pressure variation and the instantaneous flow rate generated by the RBP by using the mathematical model, CFD simulation and experimental studies. In addition, the performance characteristics of the RBP at different operating speeds and different back pressures have also been studied by using the three methods mentioned above. Parametric studies using the mathematical model and experimental studies are used to evaluate the effect on the performance of the RBP due to different pumping unit configuration such as the weight of the ball, diameter of the hole on the rear cap and the casing length. Lastly, comparison between the flow field obtained from the CFD studies and experimental flow visualization has been presented to have a better understanding on the working principle of the RBP. In the next chapter, conclusion will be drawn based on the results mentioned above.

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS

In this project, comprehensive studies have been carried out on the reciprocating ball pump (RBP). These studies include mathematical modeling, CFD simulation and experimental studies. All these studies provided a better understanding on the working principle and the performance of the RBP.

In this chapter, the key findings of this research are summarized and presented. In addition, recommendations will be proposed for further improvement and future development of the RBP.

8.1 Conclusion

This dissertation recorded the design and development process of a new miniature pump, starting from the initial stage of conceptualization to prototype testing.

This dissertation begins with a comprehensive literature review of existing miniature pumping mechanisms that are commercially available or under research for different applications in some specific areas namely, (1) pharmaceutical application, (2) medical application, and (3) MEMS micropumps application. The literature review highlights the limitations of various kinds of pumps in their respective area of application and serves to determine the area of application where the miniature pump can be applied. In addition, the literature review also included a review on mathematical modeling, CFD simulation model and fluid structure interaction simulation works. At the end of the

literature review, a series of specification for the miniature pump design has been drawn in order to proceed with the subsequent development process namely (i) design of the prototype, (ii) the mathematical modeling, (iii) CFD simulation and (iv) experimental testing. The design and development process of this project are summarized categorically as follows.

(i) Design of the prototype

The mechanisms and the limitations of various types of miniature pumping mechanism are reviewed. Particular attentions have been paid to focus on the simplicity in the working mechanism and simplicity in component geometry in order to achieve a pump that is small in size and light in weight. In addition, contamination free pumping mechanism has also been included in the design criteria to cater for pharmaceutical application, medical application, MEMS micropumps application and etc.

During the preliminary design phase, several attempts have been made to design a contamination free miniature pump. Preliminary designs include reciprocating flap pump and reciprocating disc pump which have similar working principle but with different geometry and different components. However, several drawbacks have been identified. They are (i) possible malfunctioning due to complicated geometry or mechanism, (ii) wear and tear issue due to the mechanism used and (iii) leakage problem due to the mechanism used or uneven surface. After identification of these issues, the final RBP design which is simpler and has an orientation free ball that act as a valve has been introduced. Subsequently, the focus of the research is on the development of the mathematical model, CFD simulation and experimental studies of the RBP to prove its working principle and to predict its performance.

(ii) Mathematical Modeling

In this project, a mathematical model has been successfully developed with some assumptions and simplifications. These include fluid compressibility, one-dimensional pressure driven unsteady flow, replacing viscous terms with frictional losses term and simplification of the pumping unit geometry. By using the fluid compressibility equation and one-dimensional pressure driven unsteady flow equation with viscous terms replaced by the frictional losses terms together with the reciprocating pumping motion dynamics, the pressure variation at the upstream and downstream can be predicted. The dynamics of the ball inside the RBP could also be predicted.

After the mathematical model has been developed, simulations have been conducted to reveal the working principle and the performance of the RBP. In addition, parametric studies have been conducted to evaluate the effects of the weight of the ball and the size of the pumping unit on its performance.

(iii) CFD Simulation

CFD simulation with fluid-structure-interaction (FSI) of the RBP has been successfully conducted using ANSYS-FLUENT. Moving and deforming dynamic mesh model with user-defined-function (UDF) have been incorporated to handle the unsteady dynamics of the RBP. By using UDF, the pumping motion dynamics of the pumping unit can be prescribed during the simulation. The UDF can also be applied to describe and calculate the motion of the ball inside the RBP, which is governed by the force balance between the casing, pressure and viscous forces acting onto the ball. The displacement of the ball can be calculated from the velocity of the ball, which is computed using an explicit Euler formula from the force balance acting on the ball in the axial direction.

In addition, the simulation reveals the working principle and the performance of the RBP. CFD simulation shows the working principle of the RBP in a more comprehensive manner through the pressure distribution and velocity flow field at different instances throughout the pumping process.

(iv) Experimental Studies

The functionality of the RBP has been verified through a smooth running of the prototype. The RBP operates successfully using water as the working fluid. In addition, a modified test rig was used to conduct flow visualization experiments. The experimental studies served to prove the working principle of the RBP, evaluate the performance of the RBP and to validate the mathematical model and CFD simulation.

The three studies have lead to the following conclusions:

- (a) The RBP has been proven to function reliably. Simulation and measurements have been conducted to evaluate the pump performance for motor speeds varying from 100 to 250 rpm and for different pressure heads varying from 0 to 1000 Pa.
- (b) Mathematical model and CFD simulations indicated that the RBP has a superior performance as it can deliver a higher flow rate as compared to a conventional reciprocating pump with the same dimension and stroke length. Experimental studies also confirm this finding. The instantaneous flow rate of the three studies agrees qualitatively in terms of trend and magnitude. The instantaneous flow rate vs time graph of the three studies showed that additional flow rate was induced during the backward stroke of the pumping motion of the RBP, thus leading to a higher flow rate as compared to the conventional reciprocating pump.

(c) Small fluctuations were observed in the instantaneous flow rate obtained from experimental studies. It is believed that the fluctuations were caused by the leakage due to the repeated collisions between the ball and the rear cap. This phenomenon happens when the collision force causes the ball to detach from the rear cap momentarily while the high pressure force downstream pressed the ball back onto the rear cap during the forward stroke pumping action. However, the fluctuation in the instantaneous flow rate was not observed in mathematical modeling and CFD simulation as the collision dynamics was not included in these two studies.

(d) Results obtained from mathematical model and CFD simulation revealed that the motion of the ball is slightly different from the expected working principle. This is due to the inertia of the fluid and the pressure difference across the ball. The simulation results show that the ball detaches from the rear cap before the pumping unit reaches its TDC. The RBP opens earlier than expected and this actually helps to induce more flow during the pumping process as there is more time for fluid to flow through the RBP inertially.

(e) A maximum volumetric efficiency of 172.15% and 170.88% was observed from mathematical model and CFD simulation respectively when the RBP is running at 100 rpm with 0 Pa back pressure. Experimental studies showed a maximum volumetric efficiency of 159.90% at the same condition. The percentage error between the predicted and the measured volumetric efficiency is less than 8%.

(f) The three studies revealed that the pressure variations at the upstream and the downstream of the RBP are directly proportional to the rate of change of the fluid velocity.

(g) The performance data obtained from the three studies agrees qualitatively. All three studies show that as the rotational speed increases, the flow rate increases. However, the results also showed that at low back pressures, the volumetric efficiency decreases with increasing rotational speed due to higher frictional losses experienced by the fluid at higher velocity. In addition, the results also reveal that the back pressure has an adverse effect on the volumetric efficiency.

(h) The mathematical model tends to overestimate the flow rate generated by the RBP as compared to the CFD simulation due to the underestimation of the viscous forces. However, the discrepancy between the results obtained from mathematical model and CFD simulation is less than 3%.

(i) Parametric studies have been conducted via mathematical modeling and validated by the experimental studies. Both studies revealed that the weight of the ball in the RBP has negligible effect on the performance of the RBP. The results also show that the average flow rate generated by the RBP increases as the dimension of the hole on the rear cap increases. In addition, both studies also show that as the casing length increases, the average flow rate generated by the RBP decreases as a result of higher frictional loss within the pumping unit of the RBP.

(j) Mathematical modeling and experimental parametric studies shows that the larger the critical area, the higher the RBP average flow rate and volumetric efficiency. A larger critical area can be achieved by maximizing 3 flow areas, namely, flow area (i) which is the flow area of the hole on the rear cap, flow area (ii) which is the flow area between the hole on the rear cap and the ball of 'open' RBP and finally flow area (iii) which is the flow area between the casing and the ball as shown in Figure 7-22. However, maximizing flow area (i), i.e. the diameter of the rear cap hole results in using a longer casing and a ball of larger diameter. A longer casing leads to a higher internal resistance within the pumping unit which reduces the induced flow rate and volumetric efficiency while a ball with larger diameter results in a smaller flow area (ii) which will then become the critical area that restrict the induced flow. As such, the parametric studies show that the optimal design of the pumping unit given the above restrictions is (a) a rear cap hole of 8.5 mm diameter, (b) a casing length of 11.5 mm and (c) a ball of diameter 9.6 mm.

(k) CFD simulation has been conducted to study the pressure flow field generated by the RBP to have a better understanding on the working principle of the RBP in a more comprehensive aspect. The results show the interaction between the pressure flow field and the ball and vice versa. In addition, flow visualization photos of the RBP at different positions across the stroke length were taken. These flow visualization photos agree well with the flow fields obtained from the CFD simulation at different instances during the pumping process. The flow visualization photos were also used to validate the CFD results.

8.2 Recommendation

This dissertation has ended with the mathematical model, CFD simulation and experimental studies of the RBP. However, the development of the RBP does not end here. For future development, the following studies are recommended.

(a) Collision modeling

Small fluctuations were observed in the instantaneous flow rate obtained from experimental studies while it was not observed in mathematical modeling and CFD simulation as the collision dynamics was not included in these two studies. Thus, the collision modeling can be included in the future studies to have a better prediction on the performance of the RBP.

(b) Vertical Pumping Mechanism and Orientation Free Mechanism

Current studies were conducted with the RBP operating in the horizontal orientation. Future studies can be conducted to reveal its potential and performance in the vertical orientation. However, the performance of the RBP in the vertical orientation might be affected by the gravitational force as it will pull the ball downwards and hinders its role to act as a ball valve during the pumping process. Studies can be conducted by using a ball which has a smaller density as compared to the working fluid to determine its functionality as a ball valve. In addition, certain design parameters of the existing RBP can be modified to cater for the vertical pumping process. Subsequent effort can be done in making the RBP an orientation free mechanism.

(c) Double Acting RBP

As mentioned in chapter 3, when the RBP operates under high back pressure condition, the RBP might always be in the ‘closed’ condition throughout the whole pumping process due to the high pressure force acting onto the ball in the backward direction and no net pumping effect can be achieved. This renders the RBP ineffective. Similar phenomena might also be experienced when the RBP is operated in vertical orientation. Hence, the double acting reciprocating pump (DRBP) is introduced to address this issue. The DRBP is similar to the normal RBP. However, it consists of one extra pumping unit which moves in the opposite direction to the other pumping unit during the operation so that the two pumping unit take turns to perform their effective pumping action and eliminate the issues encountered. Mathematical modeling can be modified to simulate the double acting RBP design and CFD simulation can also be conducted to reveal its pressure distribution and the velocity flow field. Finally, experimental setup can also be modified to prove the working principle of the double acting RBP.

(d) Further optimization of the RBP on the outlet geometry to enhance its performance

Parametric studies have been conducted to reveal the effect on how certain parameters of the RBP such as the diameter of the hole on the rear cap, the casing length and the weight of the ball may affect the performance of the RBP. The effects of the outlet geometry and how they will affect the performance of the RBP have not been examined. Further studies on these can be conducted via CFD simulation or experimental studies to further optimize the design and to enhance the performance of the RBP.

The study carried out has revealed the potential of the reciprocating ball pump and it is believed that there are more exciting potential to be unveiled in the future.

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APPENDIX A

Detail Drawings of RBP

APPENDIX B

Fabricated Components

Figure B-1 Reciprocating ball valve pump and its components

APPENDIX C

Catalogues and Specifications of Instruments

Technical Sales
Singapore
1 800 226 5886
asean@ni.com

NI USB-6221

Legacy USB DAQ Device

- National Instruments recommends the NI USB-6341 X Series DAQ device for all new applications.
- 16 analog inputs (16-bit, 250 kS/s)
- 2 analog outputs (16-bit, 833 kS/s); 24 digital I/O (8 clocked); 32-bit counters

Overview

This product is scheduled for obsolescence, with a last-time-buy date of September 30, 2012. National Instruments recommends the NI USB-6341 X Series multifunction DAQ device for all new applications.

The NI USB-6221 is a USB high-performance M Series multifunction DAQ module optimized for superior accuracy at fast sampling rates.

It is designed specifically for mobile or space-constrained applications. Plug-and-play installation minimizes configuration and setup time, while direct screw-terminal connectivity helps keep costs down and simplifies signal connections.

Each module also features an OEM version. Check the Resources tab or use the left navigation to get pricing and technical information.

Driver Software

NI-DAQmx driver and measurement services software provides easy-to-use configuration and programming interfaces with features such as the DAQ Assistant to help reduce development time. Browse the information in the Resources tab to learn more about driver software or download a driver. M Series devices are not compatible with the Traditional NI-DAQ (Legacy) driver.

Application Software

Every M Series DAQ device includes a copy of NI LabVIEW SignalExpress LE so you can quickly acquire, analyze, and present data without programming. In addition to LabVIEW SignalExpress, M Series DAQ devices are compatible with the following versions (or later) of NI application software - LabVIEW 7.1, LabWindows™/CVI 7.x, or Measurement Studio 7.x. M Series DAQ devices are also compatible with Visual Studio .NET, ANSI C/C++, and Visual Basic 6.0.

The mark LabWindows is used under a license from Microsoft Corporation. Windows is a registered trademark of Microsoft Corporation in the United States and other countries.

Specifications

Specifications Documents

- Specifications
- Data Sheet

Specifications Summary

General	
Product Name	USB-6221
Product Family	Multifunction Data Acquisition
Form Factor	USB
Part Number	779808-01 , 779808-04 , 779808-03 , 779808-02 , 779808-07 , 779808-06
Operating System/Target	Windows
DAQ Product Family	M Series
Measurement Type	Quadrature encoder , Voltage
RoHS Compliant	Yes
Analog Input	
Channels	16 , 8
Single-Ended Channels	16
Differential Channels	8
Resolution	16 bits
Sample Rate	250 kS/s
Max Voltage	10 V
Maximum Voltage Range	-10 V , 10 V
Maximum Voltage Range Accuracy	3100 μ V
Maximum Voltage Range Sensitivity	97.6 μ V
Minimum Voltage Range	-200 mV , 200 mV
Minimum Voltage Range Accuracy	112 μ V
Minimum Voltage Range Sensitivity	5.2 μ V
Number of Ranges	4
Simultaneous Sampling	No
On-Board Memory	4095 samples
Analog Output	
Channels	2
Resolution	16 bits

Max Voltage	10 V
Maximum Voltage Range	-10 V , 10 V
Maximum Voltage Range Accuracy	3230 μ V
Minimum Voltage Range	-10 V , 10 V
Minimum Voltage Range Accuracy	3230 μ V
Update Rate	833 kS/s
Current Drive Single	5 mA
Digital I/O	
Bidirectional Channels	24
Input-Only Channels	0
Output-Only Channels	0
Number of Channels	24 , 0
Timing	Software , Hardware
Max Clock Rate	1 MHz
Logic Levels	TTL
Input Current Flow	Sinking , Sourcing
Output Current Flow	Sinking , Sourcing
Programmable Input Filters	Yes
Supports Programmable Power-Up States?	Yes
Current Drive Single	24 mA
Current Drive All	448 mA
Watchdog Timer	No
Supports Handshaking I/O?	No
Supports Pattern I/O?	Yes
Maximum Input Range	0 V , 5 V
Maximum Output Range	0 V , 5 V
Counter/Timers	
Counters	2
Buffered Operations	Yes
Debouncing/Glitch Removal	Yes
GPS Synchronization	No

Maximum Range	0 V , 5 V
Max Source Frequency	80 MHz
Pulse Generation	Yes
Resolution	32 bits
Timebase Stability	50 ppm
Logic Levels	TTL
Physical Specifications	
Length	26.67 cm
Width	17.09 cm
Height	4.45 cm
I/O Connector	Screw terminals
Timing/Triggering/Synchronization	
Triggering	Digital

Resources

Additional Product Information

- [Manuals \(7\)](#)
- [Dimensional Drawings](#)
- [Product Certifications](#)

Related Information

- [NI USB Data Acquisition for OEM](#)
- [NI Data Acquisition Drivers](#)
- [NI LabVIEW SignalExpress Interactive Data-Logging Software](#)
- [NI Signal Streaming: Bidirectional High-Speed Data Streams Over USB](#)

Pricing

Price does not include custom duties and taxes.

Part Number	Description	Est Ship	Singapore Dollar*	Qty
Device and Professional Accessory Set				
779808-06	NI USB-6221 M Series, 26 cm, Screw Term, U.K. (240V)	8 - 13	SGD 1,634.00	0
Optional Accessories				
780315-01	Rugged Carrying Case for Portable Instrumentation	8 - 13	SGD 234.00	0
780214-01	Externally Powered USB M Series Panel Mounting Kit	8 - 13	SGD 76.00	0

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- 2 . Once you have added your items to your cart, see the "Your Cart Options" section to place your order, obtain a quote, or print a fax form.

Order by Phone

Call 1 800 226 5886 to place your order or obtain a quote.

Services

Extended Warranties

National Instruments designs and manufactures all products to minimize failures, however unexpected failures can still occur. Extended warranties provide a fixed economical price at the time of system purchase, covering any repair costs for up to three years. In addition, they offer the following benefits:

- Significant cost savings compared to individual repair incidents
- Fault location, diagnostics, and repair by NI any time the system product fails
- All parts and labor costs covered as well as any adjustments needed to restore the hardware to manufacturing specifications

For more information about your warranty options:

- Learn More About Warranty Services [<http://www.ni.com/services/warranty.htm>]
- Talk to an Expert About Extended Warranties [[javascript:openCallMeWindowCTA\(document.referrer,'%20'US'\)](http://www.ni.com/services/warranty.htm#cta)]
- View Warranty Repair Policies [http://www.ni.com/services/warranty_repair_policies.htm]

Calibration

NI recognizes the need to maintain properly calibrated devices for high-accuracy measurements. NI provides manual calibration procedures, services to recalibrate your products, and automated calibration software to calibrate many NI measurement products.

- Learn More About Calibration Services [<http://www.ni.com/services/calibration.htm>]

Training

NI training is the fastest, most certain route to productivity with NI tools and successful application development.

- Learn More About NI Training and Certification [<http://www.ni.com/training/>]
- Find a Course Near You and View Schedules [<http://sine.ni.com/apps/utf8/nisv.custed>]

Repair Services

Return your registered product under warranty at no additional labor and parts cost. NI offers fault location, diagnostics, and repair any time the system fails as well as any adjustments needed to restore the hardware to manufacturing specifications.

- Learn More About Repair Services [<http://www.ni.com/services/warranty.htm>]
- Contact NI to obtain a Return Material Authorization (RMA) form and shipping instructions. [http://sine.ni.com/apps/utf8/nicc.call_me]
- View your RMA support request status online. [<http://www.ni.com/support/servicereq/>]
- Register your product [<http://www.ni.com/register/>].

Technical Support

[ni.com/support](http://www.ni.com/support/) [<http://www.ni.com/support/>]

ELECTROMAGNETIC BLOOD FLOWMETERS

models MFV-3100
MFV-3200

A Quick probe Change by the CAL PACK
Design Concept on The IEC Class I, Type CF Safety Configuration

FIGHTING DISEASE WITH ELECTRONICS

NIHON KOHDEN

There are four major blood flow measuring methods commercially available today—ultrasonic (Doppler and transient), plethysmographic, and electromagnetic methods.

Each method has advantages and disadvantages compared to the others, but overall, the electromagnetic method is the best to determine the rate of blood flow in a particular part of the cardio-vascular system (see the table below). Therefore, the MFV-3100/-3200 electromagnetic blood flowmeters can be utilized as standard instruments to determine the flow rate and the flow waveform of specific vessels measured by an instrument based on other measuring principle.

The majority of electromagnetic blood flowmeters have been applied to physiological laboratory/animal experimental applications because the clinical field requires extremely skillful techniques for flow probe applications, complexity in measuring instrument operation, and safety against patient leak current.

However, the development of isolated probe excitation power supply, CAL PACK*-equipped flow probes, a zero stabilization circuit, etc., has made the clinical applications simpler, safer, and more realistic. Recently, the applications of the MFV-3100/-3200 electromagnetic blood flowmeters have been significantly extended to the clinical fields—cardiac, brain, abdominal, etc., to verify the effect of surgery.

The difference between the two models is that the MFV-3200 has more data display modes and output signal modes than the MFV-3100. For details, refer to page 3 and the specifications section.

* CAL PACK: The CAL PACK is a unique system to simplify calibration and balancing for zero stabilization required when initializing and/or changing a flow probe. The system is composed of a data carrying IC card box, provided with each flow probe, and a data reading and matching circuit incorporated in the MFV-3100/-3200 unit. When the box is plugged into the receptacle on the MFV's panel, followed by the touch of BAL '1' key switch, it automatically reads and memorizes both the calibration number and the balancing data for that probe. This makes it possible to immediately set a probe into operation.

Shown is the FR Series acute/clinical flow probe—one of 11 different flow probe configurations available.

Each flow probe is supplied with an auto CAL PACK.

For use with an existing flow probe, a manual CAL PACK is supplied with each blood flowmeter.

Comparison List Among Four Different Blood Flow Measuring Principles

Measuring Principle \ Performance	Wave-form Measure-ment	Mean Value	Flow Rate		Ab-solute Value	Accu-racy	Measuring Condition	
			Large	Small			Non-inva-sive	Inva-sive
Plethysmograph	○	○	○	×	×	△	○	×
Electromagnetic	◎	○	◎	◎	◎	◎	×	○
Ultrasound (Doppler)	○	×	◎	△	×	△	○	○
Ultrasound (Transient)	○	○	△	△	○	○	×	○

◎ Excellent

△ Fair

○ Good

× Impossible

The MFV-3200 is more versatile in the signal output facilities than the MFV-3100. Photo shows the MFV-3200's rear panel.

Secured Safety by Circuit Configuration

Quick Probe Ch

Blood Flow and Direction Monitoring

An array of LEDs is provided to indicate the changing blood flow and direction. This provision makes it possible for an experienced operator to visualize the blood flow waveform without connecting a recorder or monitor.

IEC Class I, Type CF Electrical Safety

Due to an isolated configuration of the flow probe excitation power and amplifier circuits, patient leak current is significantly reduced to a level as low as $10\mu\text{A}$ even under the possible worst condition concerning the instrument and flow probe. This safety design clears the requirements of IEC Class I and Type CF regulation.

Automatic Probe Balancing

The CAL PACK system simplifies and speeds initialization of a flow probe.

Zero Flow Checking without Blood Vessel Occlusion

A zero stabilization circuit is incorporated to enable the zero blood flow of the probe immersed in physiological saline solution to agree with the zero blood flow of the probe actually applied onto the blood vessel.

This permits the absolute measurement of a blood flow within the vessel which cannot be occluded. Such vessels include aortas, coronary artery, or those whose inside membrane may be easily damaged.

Simultaneous Measurement of Multiple Sites

If a number of electromagnetic blood flowmeters are provided, the blood flows at various sites on the same subject can be simultaneously measured. This permits analysis of the distribution of blood flow at different each parts of the body, the equilibrium input blood flow versus output, blood flow disturbance, etc. Max. number of sites for simultaneous measurement:

MFV-3100: 2

MFV-3200: 6

Supplied with A Manual CAL PACK to Allow The Use of Existing Flow Probes

The MFV-3100/-3200 have been incorporated with the new CAL PACK to enable immediate probe changing and application. A manual CAL PACK is a standard accessory with the MFV-3100/-3200 so that old flow probes may be used. Additionally, each new flow probe comes with its own automatic CAL PACK.

External Noise Level Checking

An external noise level, such as AC interference and/or ECG incoming to the flow probe, can be checked while disconnecting the probe exciting current by pushing the NOISE CHECK switch.

Auto Polarity Selection

The blood flowmeter is incorporated with a circuit to detect the blood flow direction and compare it against the standard direction. If it is judged incorrect, the polarity of the signal is automatically reversed. Thus, the output signal's polarity is kept correct irrespective of the direction of the probe applied onto the vessel.

Sheet Switch Control Panel Design

Employment of sheet switches on the front control panel keeps foreign matters such as solution, blood, particles, etc. from entering into the apparatus, and makes cleaning extremely easy.

ange by the CAL PACK Design Concept

Extended Measuring Range to A Minute Flow Volume Due to Low Noise Level Circuit Configuration
Development of low noise circuit configuration and 0.3mm ϕ flow probe makes it possible to determine blood flow in a small experimental animal or bypass grafting.

Minimum Measurable Flow Rates

Signals available with either MFV-3100 or MFV-3200

Signals available with MFV-3200, only

PRINCIPLE OF ELECTROMAGNETIC BLOOD FLOWMETERS

The principle used in the electromagnetic flowmeter is based on Faraday's law of electromagnetic induction, which is explained as follows: As shown in the illustration below, when a current is flowed through an excitor coil, a magnetic field is generated. If a blood flow is directed in a perpendicular direction to the magnetic field, an electromotive force is produced in a perpendicular direction to both the magnetic field and blood flow.

Multiple Output Signals

The MFV-3200 is equipped with various types of output signals: Instantaneous blood flow volume*, mean volume*, and one stroke volume.

* Available with the MFV-3100.

GP-IB Interface for Data and Signal Communications between the MFV-3200 and A Personal Computer

The MFV-3200 is equipped with a GP-IB interface to send the mean flow data, one stroke volume to a personal computer for data processing. As well as instrument's control settings from a personal computer. This capability is not available with the MFV-3100.

The magnitude of electromotive force can be expressed by the following equation,

$$ef = BD\bar{V} \times 10^{-8}$$

$$\bar{V} = \frac{ef}{BD} \times 10^8$$

where ef: electromotive force [V]
B: magnetic flux density [gauss]
D: distance between electrodes [cm]
 $\bar{V}$: mean blood flow velocity [cm/s]

Flow rate Q [mL/min] can be expressed by the following equation,

$$Q = \frac{\pi D^2}{4} \bar{V} \times 60$$

$$= \frac{15\pi D}{B} ef \times 10^8$$

Therefore, a measurement of electromotive force (ef) discloses the flow rate (Q).

Clinically Proven Applications Reveal A Variety of Blood Flows of Interest

• Ascending Aorta

- * A wide slot angle (70°) of the probe (the FR series) offers ease of application on the vessel close to a heart.
- * An automatic balancing feature enables measuring procedures to complete measurement in three minutes.

Flow Probe used: FR-190T

• Saphenous Vein Graft between Aorta and Coronary Artery (Left Anterior Descending)

- * Handle and slide-slot-enclosure equipped flow probe allows one-hand held probe application to a soft by-pass vessel, where flow measurement is apt to be affected by external force deforming the shape of the vessel.
 - * An automatic balancing feature of the instrument enables measurement for completion in a desirably short time.
- It is possible to have reliable data even in a small flow rate due to a successful low noise circuit design in the flowmeters.

Flow Probe used: FG-040T

• Internal Carotid Artery/Middle Cerebral Artery

- * A variety of probe configurations and compact design made it possible to measure blood flow in brain arteries.
- a) A wide slot opening for safe application to the fragile wall of a brain artery.
- b) A minimum slot length for easy application to the branchy brain artery thereby minimizing bothersome adjacent tissue separation.
- c) A slim probe body design for easy access to a deep-lying vessel in a narrow operating site.

Flow Probes used: FL-020T

• Right Common Iliac Artery

- * Negligible noise level is observed even in a mean blood flow as small as 140 mL/minute against a satisfactory flow volume of the calibration signal of 500 mL/minute.
- * As seen in the record below, noise on the waveform is within a negligible range even in a low mean flow rate of 140 mL/min.
- * An automatic balancing feature of the flowmeter enables measurement for test completion in three minutes.

Flow Probe used: FR-060T

SPECIFICATIONS

Probe Calibration Value Setting:

Automatic when an AUTO CALPACK-equipped flow probe is used.

Calibration Voltage: 0 to 5.3 V (adjustable by the GAIN control)

Flow Rate Measuring Range:

0.2 mL/min to 19.99 L/min

Balancing System: Automatic balancing

Noise Level: 0.3 μ Vp-p or less, referred to input (within a frequency range from DC to 30 Hz)

Linearity: $\pm 2\%$

Max. Output Voltage: ± 7 V, or greater

Max. Output Current: ± 3 mA, or greater

Zero Line Stability: Within $\pm 3\%$ of a calibrated flow rate

Output Impedance: 100 ohms, $\pm 20\%$

Input Impedance: $\oplus - E - \ominus$

5 Mohms - 5 Mohms, $\pm 5\%$

Patient Protection Circuit: The patient circuit is isolated against the chassis ground.

Frequency Response: DC to 30 Hz

(-3dB) $\pm 20\%$, pulse signal

Probe Exciting Current: 250 Hz pulse excitation, 1 A, $\pm 2\%$

Patient Leak Current: 10 μ A, or less

Chassis Leak Current: 100 μ A, or less

Input Circuit Protection:

The input circuit is protected against an excessive input pulse not greater than 1 kV with a half-wave width of 2.5 ms.

Excitation Coil Protection:

The excitation coil of a flow probe is protected from an excessive voltage by automatically reducing the voltage to zero.

Safety Design: Designed to comply with IEC class I, type CF

Power Requirements: 100, 110, 117, 125, 220, or 240 V, AC 50/60 Hz, approx. 100 VA

Dimensions and Net Weight:

218 (W) $\times$ 170 (H) $\times$ 350 (D), mm
approx. 8.2 kg

Difference between the MFV-3100 and the MFV-3200

	MFV-3100	MFV-3200
Flow Rate Display	Mean flow rate	Mean flow rate or one stroke volume
Outputs Facilities on Rear Panel	One Output Instantaneous flow rate or mean flow rate	Five Outputs 1) Instantaneous flow rate, mean flow rate, or one stroke volume 2) Mean flow rate (Time Constant: Approx. 1s) 3) Mean flow rate (Time Constant: Approx. 3s) 4) One stroke volume 5) GP-IB output
Multi-channel Synchronized Operation	Synchronized operation with an MFV-3100/ MFV-3200/ MFV-1100/ MFV-1200/ MFV-2100	Synchronized operation with an MFV-3100/ MFV-3200/ MFV-1100/ MFV-1200/ MFV-2100/ MF-27
Synchronization with an ECG	—	An ECG signal of 200 mV p-p or greater with a positive polarity

STANDARD ACCESSORIES

- 1) Power cord 1 pc.
- 2) Ground lead 1 pc.
- 3) Synchronization cord 1 pc.
- 4) Output cord
(for MFV-3100) 1 pc.
(for MFV-3200) 4 pcs.
- 5) Power fuse, slow-blow 2 pcs.
1 A for 100 to 125 V line
0.5 A for 220 to 240 V line
- 6) Manual CAL. PACK 1 pc.

ORDERING INFORMATION

When ordering, please specify;

- 1) Power line voltage and frequency
- 2) Types and quantities of flow probes

FLOW PROBE INFORMATION

Please refer to the FLOW PROBES leaflet for details of the flow probes for use with the MFV-3100/3200 blood flowmeters.

ELECTROMAGNETIC BLOOD FLOWMETER

FLOW PROBES

Exclusive for Nihon Kohden Blood Flowmeters, MFV-1100 MFV-1200 MFV-2100

17 Different Configurations
Experienced Professional Technology
100% Delivery Guarantee

FIGHTING DISEASE WITH ELECTRONICS

NIHON KOHDEN

Design and specifications are subject to change without notice.

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CAT. No.
MFV-3100/-3200-2

Printed in Japan

SK2B-0789-5
(1087)

Model 230

Wet-to-Wet Pressure Transducer

Optional

3-Valve Manifold Assembly

Setra Systems Model 230 is a high output, low differential pressure transducer designed for wet to wet differential pressure measurements of liquids or gases. A fast-response capacitance sensor and signal conditioned electronic circuitry provide a highly accurate, linear analog output proportional to pressure. Both unidirectional and bidirectional pressure ranges are available for applications with line pressure up to 250 psig.

A unique isolation system transmits the motion of the differential pressure sensing diaphragm from the high line pressure environment (e.g. corrosive liquids) to the dry (air) enclosure where it moves one of a pair of capacitance plates proportionally to the diaphragm movement. This system responds to pressure changes approximately 20 times faster than conventional fluid-filled transducers. The electronic circuit linearizes output vs. pressure and compensates for thermal effects of the sensor.

The 230 has a NEMA 4/IP65 rated package to withstand environmental effects.

3-VALVE MANIFOLD

The Model 230 can be supplied with an optional 3-Valve Manifold assembly for ease of installation and maintenance. The 3-Valve Manifold is a machined brass body requiring no internal pipe connections, thereby eliminating the risk of internal leaks. The manifold's rugged, yet compact, construction requires minimum space for installation and use. The 230 bleed ports allow for total elimination of air in the line and pressure cavities. If the Model 230 is ordered with the 3-Valve Manifold, the system is shipped completely assembled and ready for wall or pipe mounting. (Order as Pressure Fitting Code 3V.)

Model 230 Specifications

Performance Data

Accuracy RSS* (at constant temp)	±0.25% FS
Non-Linearity, BFSI	±0.20% FS
Hysteresis	0.10% FS
Non-Repeatability	0.05% FS
Thermal Effects**	
Compensated Range °F(°C)	+30 to +150 (-1 to +65)
Zero shift %FS/100°F(%FS/50°C)	2.0 (1.8)
Span Shift %FS/100°F(%FS/50°C)	2.0 (1.8)
Line Pressure Effect	Zero shift ±0.004% FS/psig line pressure
Resolution	Infinite, limited only by output noise level (0.02%FS)
Static Acceleration Effect	2%FS/g (most sensitive axis)
Natural Frequency	500 Hz (gaseous media)
Warm-up Shift	±0.1% FS total
Response Time	30 to 50 milliseconds
Long Term Stability	0.5%FS/1 YR

Maximum Working Pressure 250 psig

*RSS of Non-Linearity, Non-Repeatability and Hysteresis.

**Units calibrated at nominal 70°F. Maximum thermal error computed from this datum.

Environmental Data

Temperature	
Operating* °F (°C)	0 to +175 (-18 to +80)
Storage °F (°C)	-65 to +250 (-54 to +121)
Vibration	5 g from 5 Hz to 500 Hz
Acceleration	10 g
Shock	50 g

*Operating temperature limits of the electronics only.

Pressure media temperatures may be considerably higher or lower.

Physical Description (Model 230)

Case	Stainless Steel/Aluminum
Electrical Connection	Barrier strip terminal block with conduit enclosure & 0.875 DIA conduit opening.
Pressure Fittings	1/4"-18 NPT internal
Weight (approx.)	14.4 oz
Sensor Cavity Volume	0.27 in ³ Positive Port, 0.08 in ³ Negative Port
(With 1/4"NPT external fittings installed - does not include cavity volume of 1/4"NPT external fittings.)	

Specifications are subject to change without notice.

Physical Description (3-Valve Manifold Assembly)*

Manifold Block	Brass
Valves (3)**	V1 for connection to + port V2 for connection to - port V3 for equalizing pressure

Valve Type	90° On/Off
Process Connections	1/4"-18 NPT Internal Thread
Dimensions	7.05"W x 6.25"H x 2.16"D
Weight	<2.5 lbs.

*Order assembled with the Model 230 (Code 3V) or separately as Option 891. (Manifold can only be mated with Setra's Model 230.)

**Refer to drawing on back page.

Electrical Data (Voltage)

Circuit	3-Wire (Exc., Out, Com)
Excitation	9 to 30 VDC for 0-5 VDC output 13 to 30 VDC for 0-10 VDC output
Output*	0 to 5 VDC** 0 to 10 VDC**

Output Impedance 100 ohms

*Calibrated into a 50K ohm load, operable into a 5000 ohm load or greater.

**Zero output factory set to within ±25mV (for 5 VDC output) or ±50mV (for 10 VDC output).

**Span (Full Scale) output factory set to within ±25mV (for 5 VDC output) or ±50mV (for 10 VDC output).

Electrical Data (Current)

Circuit	2-Wire
Output*	4 to 20 mA**
External Load	0 to 1000 ohms
Minimum loop supply voltage (VDC) = 9 + 0.02 x	(Resistance of receiver plus line).
Maximum loop supply voltage (VDC) = 30 + 0.004 x	(Resistance of receiver plus line).

*Calibrated at factory with a 24 VDC loop supply voltage and a 250 ohm load.

**Zero output factory set to within ±0.08mA.

**Span output factory set to within ±0.08mA.

Pressure Media

For the Model 230

Gases or liquids compatible with 17-4 PH Stainless Steel, 300 Series Stainless Steel, Viton and Silicone O-Rings.

Note: Hydrogen not recommended for use with 17-4 PH stainless steel.

Optional Buna-N O-rings are recommended for hydrocarbon applications.

For the 3 Valve Manifold

Gases or liquids compatible with 360 brass, Copper 122, Acetal plug valves and Nitrile O-rings.

Applications

- Energy Management Systems
- Process Control Systems
- Flow Measurement of Various Gases or Liquids
- Liquid Level Measurement of Pressurized Vessels
- Pressure Drop Across Filters

Features

- NEMA 4/IP65 Rating
- No Liquid Fill Diaphragm
- Available with 3-Valve Manifold Assembly Option
- Low Line Pressure Effect
- Fast Response
- Gas and Liquid Compatible
- Low Differential Ranges
- CE and RoHS Compliant

When it comes to a product to rely on - choose the Model 230. When it comes to a company to trust - choose Setra.

ISO
9001:
2000
Certified

Visit Setra Online:
<http://www.setra.com>

setra
800-257-3872

Pressure Ranges

UNIDIRECTIONAL		
Pressure Range PSID	Proof Pressure High Side* PSI	Proof Pressure Low Side* PSI
0 to 1	20	2.5
0 to 2	40	5
0 to 5	100	12.5
0 to 10	100	25
0 to 25	250	62.5
0 to 30	250	75
0 to 50	250	125
0 to 100	250	250

BIDIRECTIONAL		
Pressure Range PSID	Proof Pressure High Side* PSI	Proof Pressure Low Side* PSI
0 to ±0.5	20	1.25
0 to ±1	40	2.5
0 to ±2.5	100	6.25
0 to ±5	100	12.5
0 to ±10	200	25
0 to ±25	250	62.5
0 to ±50	250	125

*The zero will shift slightly when high differential overpressure is applied. The shift may be as much as ±10% FS with overpressure applied to the low pressure port. Other parameters (sensitivity, linearity, etc) will not shift. If the overpressure is normally only in one direction, the user may apply this overpressure to preset the sensor. Subsequent overload of less magnitude will not cause additional shift. The unit is pre-zeroed at the factory after application of maximum overload pressure to the high pressure port.

NOTE: Setra quality standards including ISO 9001 are based on ANSI-Z540-1. The calibration of this product is NIST traceable. U.S. Patent nos. 4054833

DMI MAGNETIC INDUCTIVE FLOWMETER

Flow
Pressure
Level
Temperature
measurement
monitoring
control

- Measuring Ranges: 0.7-10 Through 25-320 GPM
- PEEK or PVDF Body
- LCD Rate/Total Display
- Accuracy: $\pm 3\%$ of Reading, $\pm 1.5\%$ Optional
- Fully Flexible, Menu-Driven

S5

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www.kobold.com

Model:
DMI

Features

- Measuring Ranges: 0.7-10 Through 25-320 GPM
- PEEK or PVDF Body
- LCD Rate/Total Display
- Accuracy: $\pm 3\%$ of Reading, $\pm 1.5\%$ Optional
- Fully Flexible, Menu-Driven

The KOBOLD series DMI operates on the magneto-inductive principle. With no protrusions into the pipe, conductive liquids and slurries can be measured with virtually no system pressure loss. The DMI is available with a PEEK or PVDF flow tube and Hastelloy C measuring electrodes are standard. These features make the DMI series light-weight, economical and suitable for many types of chemical solutions and aggressive media. The DMI series has no moving parts and is therefore virtually maintenance free. The electronics package includes an LCD rate/total display which is switchable from U.S. to metric units. The standard open collector output is configurable as a pulse flowrate transmitter or alarm switch. An optional 4-20 mA flow rate transmitter is also available. Setup is easy with menu-driven programming which can be set by the user in one of seven different languages.

DMI Magnetic Inductive Flowmeter

Specifications

Body Material

3/8" through 1"

diameter: PEEK

2" diameter: PVDF

Electrodes: Hastelloy C

Max. Pressure: 145 PSIG

Temperature: $-30^{\circ}\text{F} \dots +230^{\circ}\text{F}$

Conductivity: $> 50 \mu\text{S/cm}$

Minimum Inlet
and Outlet Straight
Pipe Req:

10 x D inlet

5 x D outlet

Accuracy: (for $> 0.07 \times Q_{\text{max}}$)
 $\pm 3\%$ of measured
value
(Option: $\pm 1.5\%$ of
measured value)

Repeatability: $< 0.2\%$ of
measured value

Creep

Suspension: Adjustable 0-10%
of measuring range

Response Time: 0-99% jump > 5 sec. minimum.
setable between
5-40 sec.

Protection: NEMA 4X/IP 65

Electronics

Power Supply: 18-30 VDC or
18-26 VAC
2-line LCD

Display:
Electrical

Connections: Hirschmann plug
acc. to DIN 43650
1 pulse/gallon(liter)
fixed set.

Pulse Width: 20 msec

Pulse Frequency: Max. 20 Hz

Output

DMI-...A...

The optocoupler output can be set via display as a pulse output or as an alarm switch.

DMI-...B...

Additional 0/4...20 mA scalable output (optional)

Programmable Features

- Language (7 Available)
- Open Collector Function (Flow Transmitter or Switch)
- Switch Setpoint
- Delay Time
- Creep Suspension
- 0(4)-20 mA Transmitter Span (Optional)
- Measuring Units

DMI Magnetic Inductive Flowmeter

Programmable Features

- Language (7 Available)
- Open Collector Function (Flow Transmitter or Switch)
- Switch Setpoint
- Delay Time
- Creep Suspension
- 0(4)-20 mA Transmitter Span (Optional)
- Measuring Units

Applications

- For conductive liquids and slurries which are compatible with wetted parts.

Ordering Information

Range (GPM)	Body Material	Base Model	Process Connection	Flow Tube Diameter*	Output	Power Supply	Options
0.7 to 10	PEEK	DMI-2002...	..N20..=3/4 NPT	3/8"	..A..=Pulse or alarm ..B..=Pulse or alarm and (0) 4-20 mA Output	..3..= 24 VDC/VAC	...0=none ...G=1.5% of measuring value
2 to 25	PEEK	DMI-2004...	..N20..=3/4 NPT	1/2"			
6 to 80	PEEK	DMI-2006...	..N32..=1-1/4 NPT	1"			
25 to 320	PVDF	DMI-2508...	..N65..=2-1/2 NPT	2"			

Example: DMI-2302 N20 A30

* In order to meet the DMI series straight piping requirements, the inlet and outlet piping should be the same nominal I.D. as the flow tube diameter for the model selected. Reducers can be used at the inlet and outlet fittings to adapt to the larger process connections.

DIMENSIONS

Type	L	D	φ	H
DMI-...02	3.3	2.1	3/4" NPT	5.9
DMI-...04	3.3	2.1	3/4" NPT	5.9
DMI-...06	3.9	2.5	1-1/4" NPT	6.3
DMI-...08	5.1	3.6	2-1/2" NPT	6.9