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DEVELOPMENT OF AN ENCLOSED-IMPELLER VENTRICULAR ASSIST DEVICE USING SELF-BEARING MOTOR

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Abstract

Congestive heart disease is currently a major killer. Hence, there is an increasing need for implantable artificial hearts as an alternative to donor hearts which cannot meet the needs. The development of magnetically suspended Ventricular Assist Devices (VADs), alias blood pumps, has been a promising option for this purpose in recent years. In order to overcome the limitations of the current axial flow blood pumps available, such as low motoring and pump efficiencies as well as large size, an axial-flow blood pump featuring an enclosed-impeller and self-bearing motors is proposed in this thesis. Computational fluid dynamics (CFD) analysis is carried out to analyze the pump characteristics of the proposed blood pump. The results show that the proposed blood pump could provide 50 mmHg pressure head with 26 % pump efficiency at the operating point of 9,000 rpm rotating speed at 5 L/min flow rate. The maximum blood shear stress and maximum blood exposure time is 200 Pa and 146 ms respectively, which indicates lower blood trauma tendency in the proposed blood pump.

A prototyped Lorentz force type self-bearing motor test rig is designed with the aid of finite element analysis (FEA) to validate the control scheme for the proposed self-bearing motor. The self-bearing motor is successfully levitated by a Proportional-Integral-Derivative (PID) control scheme with a maximum displacement deviation of 0.3 mm. The maximum deviation is further reduced to 0.21 mm by the proposed PID-Robust Model Reference Adaptive Control (RMRAC). The stability control of the proposed PID-RMRAC under output disturbance is also verified through the experiments. These experimental results indicate that the proposed blood pump could be a better alternative for the future development of blood pumps.

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Chapter 1

Introduction

1.1 Background

Studies have shown that failure of the cardiovascular system is one of the most common health disorders causing premature death in society. The total annual death rate due to cardiovascular disease is almost one million globally. It is estimated that the prevalence of heart failure approximately doubles with each decade of life. This actually makes heart failure the number one cause of death in the world.

The type of treatment for heart failure depends on the extent and severity of the illness. Many patients can be cured with drug therapies. But there still are some forms of this disease that cannot be medically treated. One of the last options that remain for these patients is heart transplantation [1].

However, a serious problem with heart transplantation is that there is a large deficiency in the number of available organ donors. Fig. 1.1 shows 3,621 heart transplant candidates listed on the U.S.A. national waiting list of 2006 awaiting a heart transplant [2]. It also indicates the waiting time heart candidates spend on the waiting list. The long waiting time in heart transplants is due to the lack of availability of heart donors. This prevailing problem is also seen in other countries and leads to a need for

a viable substitute of heart organs that can bridge the gap between the supply and the demand of heart donors.

Fig. 1.1 Average waiting time for heart candidates in U.S.A.

Ventricular Assist Devices (VADs), also termed as blood pumps, are possible solution to this problem. Sometimes used as a "bridge-to-transplant" or as a permanent therapy, alternatively called "destination therapy," the VAD is a mechanical pump-type device that is surgically implanted to help maintain the pumping ability of a heart that is unable to function effectively on its own [3]. There are many commercially available blood pumps in the world. However, most of the VADs do not operate well in the long term. This may be due to the acute rejection from the body after transplantation, infection problems and thrombus problems. In addition, the cost for the commercial available VAD is also known to be very high and economically out of reach to average wage earners.

Thus, this study attempts to make a contribution to the development of new axial flow blood pumps by re-examining the present blood pump design problems and proposing a new type of impeller and a new type of pump supporting and driving mechanism through simulations and experiments.

1.2 Objectives

The specific objectives of the research are:

- 1) To provide a simple VAD design, small in size and low in cost.
- 2) To provide an enclosed-impeller axial flow blood pump with optimized pumping efficiency while restoring hemodynamics functions, minimizing hemolysis, preventing clot formation, infection and bleeding.
- 3) To provide a VAD with a Lorenz force type self-bearing motor to power and support the blood pump and reduce its size.
- 4) To provide efficient control algorithms to suspend the pump rotor with long term durability, low heating and minimized system vibrations.

1.3 Dissertation outline

This dissertation is organized as follows. Following the introduction, the history of development of VADs is reviewed, which highlights the background information and motivation of this research.

Chapter 3 presents the mechanical mechanism and pump components of the VAD system proposed in this study. In Chapter 4, the detailed design of the rotor and stator of the blood pump proposed is addressed. The computational fluid dynamics

(CFD) analysis of the blood pump is also carried out in this chapter. The flow characteristics, driving torque and axial forces and blood damage, etc. of the blood pump are discussed. Chapter 5 carries out the theoretical and finite element analysis (FEA) of the Lorentz force type self-bearing motor. An experimental test rig of the self-bearing motor is proposed based on the principle of the Lorentz force type self-bearing motor. In Chapter 6, the levitating of the self-bearing motor of the blood pump proposed is realized by using the PID control method. In Chapter 7, a hybrid control method by combining the PID control method and robust model reference adaptive control (RMRAC) method is proposed to realize the levitating of the self-bearing motor of the pump.

Finally, in Chapter 8, the whole work of the thesis is summarized along with some recommendations for future works.

1.4 Contributions

The original contributions to the development of the proposed blood pump, which include the concept design of VAD, pump simulation and development of the self-bearing motor of VAD, of which some research findings have been documented in international journals and conferences. The author has co-authored and contributed to the following publications:

Papers in International Journals

- 1) T. M. Lim, and D.S. Zhang, Development of Lorentz Force Type Self-Bearing Motor for an Alternative Axial Flow Blood Pump Design. *Artif Organs* 2006; 30: 347-353.

- 2) T. M. Lim, and D.S. Zhang, Control of Lorentz Force Type Self-Bearing Motors with Hybrid PID and Robust Model Reference Adaptive Control Scheme. Journal of Mechatronics 2008; 18: 35-45.

Papers in International Conferences

- 1) T. M. Lim, and D.S. Zhang, Development of Lorentz Force Type Self-Bearing Motor for an Alternative Axial Flow Blood Pump Design. 13th Congress of the International Society for Rotary Blood Pumps 2005, Tokyo, Japan.

The above paper was selected through blind peer review as one of the eight finalists for the “Prof. Helmut Reul Yong Investigator Encouragement Award Competition”.

- 2) T. M. Lim, and D.S. Zhang. Numerical Analysis of Blood Trauma In an Enclosed-impeller Axial Flow Pump. 13th Congress of the International Society for Rotary Blood Pumps 2005, Tokyo, Japan.
- 3) T. M. Lim, S. Cheng and D.S. Zhang, 2007: A Compact Magnetic Bearing System for Axial Flow Blood Pump, IMECS 2007: 21-23 March, 2007, Hong Kong, pp 1591-1596.

The above paper was selected as the International Association of Engineers (IAENG) Best Student Paper based on the originality, significance in scientific contributions, correctness, and clarity.

Paper in Local Conference

- 1) T. M. Lim, and D.S. Zhang. A Decoupled Control Method for the Magnetic Bearings of a Blood Pump. IBEC 2004, 8th September - 10th September 2004, Singapore.

Chapter 2

Literature review

2.1 Background of VADs

Heart disease causes more than 700,000 deaths each year in the United States alone. According to the American Heart Association (AHA) and the National Heart, Lung, and Blood Institute (NHLBI), about 3 million people in the U.S. are suffering from congestive heart failure (CHF), a chronic, highly debilitating and degenerative disorder where the heart is unable to pump sufficient blood to the body's organs. In the U.S., about 400,000 cases of CHF are diagnosed each year (Similar figures have been estimated for Europe and Japan together, though accurate worldwide figures are not available). The five-year survival rate for these CHF patients is approximately 50 % and it is estimated that 70,000 - 120,000 of these patients could benefit from a heart transplant [4]. However, in the U.S., only 2,185 transplants were performed in 2006, compared to a transplant waiting list that exceeded 3600 [5]. A severe shortage of donor hearts worldwide and high costs (\$250,000 - 400,000 per patient) are limiting factors for heart transplant [6].

Therefore, an alternative to heart transplants is needed to solve this problem. Total Artificial Heart (TAH) is the first of such substitute. TAH is a pneumatically driven pulsatile pump which simulates the ventricular action of the heart to circulate

the blood in the body. It is also called the first-generation blood pump because of its pulsatile feature. ABIOMED BVS 5000 blood pump [7], developed by Texas Heart Institute (THI), is such a blood pump. It is used for temporary left, right or biventricular (both ventricles) support in patients with potentially reversible heart failure. The structure of the pump is shown in Fig. 2.1. The pump houses two polyurethane chambers, an atrial chamber that is filled with blood through gravitational force and a ventricular chamber that pumps blood by air-driven power. The atrial chamber is vented outside the patient. The ventricular chamber is connected to the power console by a 6.4 mm pneumatic line. Two trileaflet valves separate the atrial and ventricular chambers. The pump can produce blood flow of up to 5 L/min. Cannulas of various designs (for blood drainage and return) are available to accommodate different patient anatomies.

Fig. 2.1 ABIOMED BVS-5000 blood pump.

TAH is a big help to patients. However TAH is large in size, requiring inflow and outflow valves and is prone to infection. Therefore, the survival rates of patients is very low [8].

However, failures of the left ventricle accounts for 80 % of heart failures. Those patients suffering from left ventricular failure only require left heart support for survival. Therefore, implantable left ventricular assist devices (LVADs) (also known as rotary blood pumps or non-pulsatile blood pumps) are developed to pump the blood continuously. These devices have been proven to be more versatile and practical than Total Artificial Hearts (TAHs), eliminating the need for cardiectomy and maintaining the possibility of device extraction if used as a bridge to recovery. This trend towards the utilization of Left Ventricular Assist Devices (LVADs) instead of TAHs is the major substitute for heart transplant.

2.2 History of development of rotary blood pumps

In 1980, the Cleveland clinic group reported on the survival of animals under blood flow completely generated by a nonpulsatile centrifugal pump which featured a pair of magnetically coupled motors and a shaft seal mechanism [9-11]. Although the animals in the experiment only survived 34 days due to pump malfunction caused by a blood clot formed around the shaft seal area, this pilot study proved that continuous flow did not limit the survival of the animals, provided that adequate arterial pressure and blood flow were maintained. The nonpulsatile blood pump, also termed as rotary blood pump, had a simple pump mechanism not requiring inflow and outflow valves. It was suggested that this type of pump might be a simpler way of realizing a compact and implantable blood pump system.

Since then, rotary blood pumps have been rapidly developed. The first types of nonpulsatile pumps rising on the horizon are axial or centrifugal turbo pumps with contacting mechanical bearings. Examples include the Jarvik 2000 blood pump [12] and DeBakey blood pump [13]. Those pumps are usually known as the second-generation pumps due to their contacting mechanical bearing feature.

Many advantages are associated with the use of nonpulsatile blood pumps. These advantages include:

Size: Nonpulsatile blood pumps are significantly smaller in volume than pulsatile ones. Their smaller dimensions enable the device to be placed in a more orthotic position, that is, in the thorax near the heart instead of in the upper abdomen. This eliminates the use of long cannula passing through the diaphragm.

Simplicity: Rotary blood pumps do not require complex components such as valves, diaphragms, blood sacs, vents and compliance chambers. Nonpulsatile continuous blood pumps are made of a simple motor which is coupled to an impeller located in the pump's housing. Therefore, nonpulsatile blood pumps are more reliable and durable.

Hemolysis: Hemolysis is the tearing of the red blood cells. The damage of nonpulsatile blood pumps to red blood cells is 10 to 20 times less than that of pulsatile ones [14].

Infection: The transcutaneous vent of the pulsatile pump easily causes infection. On the other hand, the rotary blood pump is fully enclosed in the body, thus reducing the possibility of infection [14].

Although the second-generation blood pumps are much better than the first-generation ones, they have a main problem, that is, hemodynamic reliability. Blood-immersed mechanical contact bearings of the second-generation blood pumps result in the wearing and tearing of materials as well as the generation of heat, leading to device malfunction and thrombus formation around the contact area. Thus the projected life span of the blood-immersed bearings of second-generation blood pumps is not longer than 5 years [15].

This fact makes the third-generation blood pumps which utilize magnetic suspension mechanisms (maglev) necessary. The maglev allows for the suspension of a moving element without mechanical contact, thus eliminating heat generation, as well as wear, that will take place at the contact surface. The technology of maglev was first applied to rotating machinery by the University of Virginia research group in

1950 [16]. Application for rotary blood pumps was first reported by Allaire et al. in 1996 [17]. The centrifugal rotary blood pump developed by Allaire et al. at the University of Virginia utilized a five-degree-of-freedom (DOF) levitation maglev system to suspend the impeller, while the impeller was rotated by a direct drive stator. This blood pump had a length of 196 mm, a diameter of 115 mm and a weight of 365 g. A flow rate of 6 L/min and a pressure head of 100 mmHg were obtained at a rotational speed of 2,400 rpm. A power of around 50 W was required to produce the bias flux and copper loss with a large current.

Since then, many blood pump systems have been developed following the successful application and implementation of the maglev concept, such as the DuraHeart centrifugal blood pump [18-21], the Berlin Heart INCOR axial flow blood pump [22-25], the Levitronix blood pump [26-30], the HeartMate III blood pump [31-33], the HeartQuest blood pump [34], the MagneVAD blood pump [35, 36], the Ibaraki University blood pump system [37-39] and the Tokyo Medical and Dental University/Tokyo Institute of Technology (TMDU/TIT) blood pump systems [40-43].

Among these pumps, the maglev systems can be categorized into three types: (a) external motor-driven system, (b) direct-drive system, and (c) self-bearing motor system. In the external motor-driven system, a motor is used to induce magnetic coupling force in the impeller, while a separate levitation system controls the impeller suspension. The shortcomings of this system are that mechanical bearings are required to support the rotation of the motor. In the direct-drive system, the impeller becomes the rotor of the motor, while a separate levitation system is built into the system to realize magnetic suspension. In the self-bearing motor system, the drive and the

levitation coils share the same stator core and provide both the driving torque and levitation force.

2.3 Typical rotary blood pumps

2.3.1 Jarvik 2000 axial flow blood pump

The Jarvik 2000 blood pump [12] (Jarvik Heart Inc, New York, U.S.A.) is a second-generation blood pump. As shown in Fig. 2.2, the blood pump weighs 90 g, measures 25 mm in diameter and displaces 25 mL in volume. The titanium pump shell is hermetically sealed and contains a brushless electromagnetic direct current (DC) motor. The rotor which is held in place by two blood-immersed ceramic bearings is made of a titanium impeller blade and neodymium-iron-boron permanent magnets. The pump also includes outflow stator blades downstream from the impeller. By means of electromagnetic force, the impeller can rotate at speeds of 8,000 to 12,000 rpm, yielding a maximal blood flow of 7 L/min against physiologic resistance.

Fig. 2.2 Jarvik 2000 axial flow blood pump.

The blood pump is implanted in the left ventricle through a circular incision and is secured with silicone-polyester sewing cuff (Fig. 2.3). The 16 mm hema-shield outflow graft is extended from the left ventricular apex to the descending aorta. After being brought out through the abdominal wall, the power cable is connected to the controller. The pump is powered by lithium-ion or sealed lead-acid batteries. The microprocessor-based controller responds to the cardiac cycle by sensing changes in the motor current, which corresponds to changes in the pump differential pressure. The controller also can be programmed to change the pump speed at different points in the cardiac cycle. External power can be transmitted to an implanted controller and battery by means of a transcutaneous energy transfer system (TETS), if necessary, backup power is supplied by a secondary TETS coil. For maximal freedom of movement, the patient wears an external controller and lithium-ion battery pack, which allows tether-free operation for 8 hours.

Fig. 2.3 Implantation of Jarvik 2000 blood pump.

The Jarvik 2000 blood pump has been successfully used for a small number of patients (around 50 cases) as of 2005. However, it frequently lasts for only slightly more than a year due to thrombosis and/or hemolysis and pump failure.

2.3.2 DeBaakey axial flow blood pump

The DeBaakey blood pump [13] (MicroMed Technology, Houston, TX, U.S.A.) is a miniaturized axial flow pumping device (Fig. 2.4). It is 86 mm in length, 25 mm in diameter and weighs 95 g. It pumps blood from the left ventricle through a titanium inflow cannula inserted into the heart's apex. After passing through the pump, blood flows back to the body through a Dacron outflow graft sewn to the ascending aorta. Within the flow tube, the components consist of an inducer-impeller (the only moving part); a fixed flow straightener that acts as the front-bearing support for the inducer-impeller; and a fixed rear diffuser which provides the rear bearing of the inducer-impeller. The diffuser retards the highly tangential blood flow velocity by redirecting it to flow axially, an action that results in an increased fluid pressure head. Rare earth magnets which are embedded in the blades of the impeller act as the rotor of a brushless motor and enable it to spin in a rotating magnetic field. This pump can produce a flow rate of 5 to 6 L/min against a 100 mmHg pressure. The operational speed is about 10,000 rpm with less than 10 W of power consumption.

Fig.2.4 DeBakey axial flow blood pump.

The pump is connected to the controller via a percutaneous cable that is passed through the skin just above the right iliac crest (Fig. 2.5). The controller which is designed to operate the pump is powered primarily by two 12-volt direct current (DC) batteries. The pump is powered at night by the Patient Home Support System, which includes an eight-foot power cable and eliminates the need for battery changes.

Fig. 2.5 Implantation of DeBakey blood pump.

Currently, this device has been implanted in almost 200 patients. However, significant hemolysis, device failure and a need for pump replacement have been reported [15].

2.3.3 DuraHeart blood pump

The DuraHeart centrifugal blood pump [18-21] in Fig. 2.6 (Terumo Heart Inc., Michigan U.S.A.) is a third-generation blood pump. The implantable DuraHeart system consists of a titanium pumping unit, a titanium inflow and outflow conduit and a drive and control console. The pump has a displacement volume of 180 mL, an external diameter of 72 mm, a height of 45 mm, and weights 540 g. The pumping unit consists of an upper housing with a levitation system, impeller and bottom housing. Three electromagnets and three position sensors are mounted in the upper housing. Three electromagnets and three position sensors are mounted in the upper housing.

Fig. 2.6 DuraHeart blood pump.

Fig. 2.7 shows the levitation and drive mechanisms of the DuraHeart system. Tilting and axial displacements of the impeller are monitored and controlled by using a three-DOF control. A levitated impeller is driven through a magnetic coupling force from the direct current brushless motor mounted outside. Radial impeller movement is passively suspended with a bias flux through electromagnetic rotor and drive magnet

rotor. Also, the pump design incorporating the double volute can reduce hydraulic force imbalance inside the pump housing, stabilizing the radial movement. The pump speed is controlled from 1,200 to 2,600 rpm and can produce a flow rate of 2 – 10 L/min.

Fig. 2.7 Levitation and drive system of DuraHeart blood pump.

The first clinical implant of the DuraHeart blood pump was performed in January 2004 in Germany. To date, about 20 implants have been completed at four heart centers in Europe. The support duration was 17 ~ 497 days.

2.3.4 Berlin Heart INCOR axial blood pump

The Berlin Heart INCOR axial flow blood pump [22-25] (Fig. 2.8) is developed by the Berlin Heart centre. The pump is made of titanium and weighs 200 g. It is 120 mm long in axial direction. The outer diameter and the volume of the pump are 30 mm and 80 mL respectively.

Fig. 2.8 Berlin Heart INCOR axial blood pump.

Fig. 2.9 Schematic structure of Berlin Heart INCOR blood pump.

Fig. 2.10 Levitation and drive schematic of Berlin Heart INCOR blood pump.

The schematic structure of Berlin Heart INCOR blood pump is shown in Fig. 2.9. Levitation and drive schematic of the system is shown in Fig. 2.10. The axial

impeller movement is monitored by a position sensor and controlled actively by two electromagnets mounted at both ends of the rotor. The radial and tilting impeller movements are passively controlled. The levitated impeller can be rotated at speeds between 5,000 and 10,000 rpm, generating up to 5 L/min of flow against a pressure of 100 mmHg with a maglev power consumption of 2 – 4 W. The stationary diffuser behind the rotor has specially aligned blades which reduce the rotational effects of the blood flow and adds additional pressure to assist the transport of the blood in the outflow cannula to the aorta.

The efficiency (over 90 %) and the power consumption (< 4 W) of the motor ensure that the pump does not generate a significant amount of heat. The control software enables the pressure gradient and the flow to be monitored by detecting the rotor's position in the magnetic field. This provides the patient and the user with important data about flow rates and pump performance. These values are further utilized to prevent suction through the pump in the left ventricle by detecting the residual pulsatility in the pump. The pump can automatically reduce its rotation speed and allow a renewed filling of the ventricle. The originally selected pump performance is then restored slowly and in a controlled way.

The first human clinical trial of the pump was performed in June, 2002 at the German Heart Institute, Berlin, Germany. Since then, the Berlin Heart INCOR axial flow blood pump has been used in 212 cases worldwide. The support time for the device varies from 162 days to 1071 days.

2.3.5 Levitronix blood pump

The Levitronix blood pump [26-30] (Fig. 2.11) is developed by Levitronix Incorporation. The schematic structure of the pump is shown in Fig. 2.12. The total pump system consists of a single-use disposable pump head (31 mL in volume), a motor/bearing drive unit (diameter $\times$ height = 87 $\times$ 70 mm), a cannulas and a controller. The rotor is suspended and rotated by eight (four pairs) L-shaped iron cores with drive and control winding together. The position and rotation of the rotor are continuously controlled by a feedback control system in radial directions as shown in Fig. 2.13. The Levitronix blood pump can deliver a pump flow of up to 9.9 L/min and a pressure head of up to 600 mmHg operating at a rotational speed of up to 5,500 rpm. The pump was specially designed for extracorporeal circulatory support applications as cardiopulmonary bypass or uni- or bi-ventricular assistance.

Fig. 2.11 Levitronix blood pump.

Fig. 2.12 Schematic structure of Levitronix blood pump.

Fig. 2.13 Levitation and motor system of Levitronix blood pump.

From 2003 to 2005, over 150 patients worldwide were supported with the Levitronix blood pump. All cases exhibited low hemolysis, and no technical problems occurred with the maglev system during the observation period.

2.3.6 HeartMate III blood pump

The HeartMate III blood pump [31-33] (Thoratec Corporation) shown in Fig. 2.14 is a totally implantable uni- / bi-ventricular assist system. The casing, vent port and inflow cannula of the pump are made of titanium alloy. The textured surface of

the pump housing adapted for blood-contacting surfaces reduces the risk of thromboembolism. The rotor is fabricated with smooth and polished surfaces. The external pump is approximately 69 mm in diameter, 35 mm in height and 535 g in weight (including the inflow cannula, recovery section, outflow graft, bend relief and connecting hardware).

Fig. 2.14 HeartMate III blood pump.

The HeartMate III blood pump adopts bearingless slice motor technology that combines drive, magnetic bearing and pump rotor functions in a single unit. The basic arrangement of the HeartMate III blood pump is shown in Fig. 2.15. The three spatial degrees of freedom are stabilized passively and the remaining three are stabilized by active control. The active control of the rotational and radial position of the rotor is provided by feedback electronics, implemented by appropriately placed sensors. The axial position and tilting are stabilized by passive magnetic forces, as shown in Fig. 2.16.

Fig. 2.15 Basic arrangement of HeartMate III blood pump.

Fig. 2.16 Passive stabilization of axial and angular displacement of HeartMate III blood pump.

Due to the passive stabilization of three spatial degrees of freedom, only one

motor phase and two control phases (which make a total of three power amplifiers) are needed as a minimum requirement for the operation of a bearingless slice motor. With an additional motor phase, a vector-controlled drive becomes feasible. It enables the motor to accelerate rapidly, which allows the pump to operate in a pulsating mode. Additionally, only two position sensors are required. This leads to straightforward electronic control schemes and simple motor assembly.

A flow rate of 7 L/min can be obtained at a head pressure of 135 mmHg with a power consumption of 8 W. At a rotational speed of 2,000 – 5,500 rpm, the pump can produce an output of 10 L/min under the left ventricular assist conditions. The percutaneous version of the HeartMate III blood pump demonstrated up to 120 days survival in animal tests.

2.3.7 MagneVAD blood pump

The MagneVAD blood pump [35, 36] (Gold Medical Technologies) shown in Fig. 2.17 is an ultra-small magnetically levitated axial pump system with a low volume of 25 mL and a weight of 82 g. Magnetic flux is produced between the inlet side electromagnet stator and the impeller. The position of the impeller is monitored with an ultrasonic position sensor in the axial direction (one DOF). Other impeller movements are passively suspended by magnetic flux. The washout hole is in the rotor to produce secondary flow for better anti-thrombogenicity. An outlet diffuser reduces the flow separation and obtains better washout in the outlet line junction. The rotor location shifts in response to differential pressure. A low-cost position sensor measures this motion to provide pump differential pressure.

Fig. 2.17 MagneVAD blood pump.

Fig. 2.18 Cross section of MagneVAD blood pump.

Fig. 2.18 shows the cross-section schematic through the blood pump. The pump is composed of three main assemblies: the inlet stator at the far right, the magnetically suspended rotor (impeller) at the center and the outlet stator at the left, which has two integral cones as shown. The ultrasonic position sensor housed in the inlet stator houses measures the axial position of the rotor at an update rate of 100 kHz and operates at a switching frequency of 8 MHz. It is coated for blood compatibility. The stator also houses a bearing coil used to stabilize the rotor axial position. The

outlet stator is not shown in section but it has a similar bearing coil. These coils can push or pull the rotor using opposite currents, which is particularly useful at initial lift off.

A 12-segment radial magnetized neodymium magnet is located at each end of the rotor. Only the bearing at the right end of the rotor is shown sectioned. The axial magnetic field created by this magnet is focused by the inner and outer permeable-iron fringing rings. The flux crosses the rotor blood gap into the matching fringing rings in the stator. This axial field produces a passive radial load capacity with high radial stiffness.

A typical pump flow rate is 6 L/min at a pressure head of 100 mmHg and a rotational speed of 12,500 rpm. The motor power consumption is typically 5 W and the magnetic suspension system consumes 0.33 W using virtually zero power control.

2.3.8 Ibaraki University maglev blood pumps

The Ibaraki University maglev blood pumps were developed by Ibaraki University, Japan [37-39]. Fig. 2.19 shows the structure of the pump. In the pump, the impeller is set between the active magnetic bearing and the motor stator. The active magnetic bearing uses four electromagnets to control the tilt and the axial position of the impeller (three DOF) as shown in Fig. 2.20. The radial movement of the levitated impeller is restricted with the passive stability dependent upon the top stator and the passive permanent magnetic bearing to reduce the energy consumption and the complexity of the control system. The levitated impeller is rotated with the bottom stator. This system is 56 mm in height, 95 mm in diameter and has an impeller

diameter of 60 mm. The maximum pressure head and the maximum flow rate are 200 mmHg and 10.5 L/min, respectively at the rotational speed of 2,000 rpm.

Fig. 2.19 Ibaraki University maglev blood pump.

Fig. 2.20 Maglev schematic of Ibaraki University blood pump.

2.3.9 TMDU/TIT maglev centrifugal blood pumps

An extracorporeal disposable centrifugal blood pump system and an implantable LVAD blood pump were developed by Tokyo Medical and Dental University and Tokyo Institute of Technology (TMDU/TIT) [40-43]. In both blood pump systems, the impeller is suspended with two pairs of electromagnets in two DOF controlled in x- and y-directions. Strong magnetic bias flux is also produced between the rotor and the magnetized steel in the outer pump housing to passively control the movement of the impeller in the z-axis and rotational axes. This system has a high passive stiffness of 15.9 - 17.0 N/mm in z-direction and 3.4 - 4.4 Nm/rad in tilt directions.

The disposable pump system consists of a pump head with an easy detach-and-insert feature and an actuator unit (Fig. 2.21 and Fig. 2.22). The impeller is rotated by the radial magnetic coupling force induced from the driver magnet which is mounted on the external motor shaft and placed inside the doughnut-shaped impeller. In the simulated mock loop testing with a pulsating ventricle, it is shown that the impeller stability was affected by the ventricular pulsation, indicating that from the level of impeller movement, the ejection strength of the ventricle could be diagnosed.

A flow rate of 5 L/min is obtained at a pressure head of 100 mmHg and a rotational speed around 1,800 rpm. The power consumption of the magnetic bearing is < 0.7 W at a rotational speed of 1,400 – 2,400 rpm. This system is intended as a one-month bridge-to-bridge extracorporeal heart assist device, bridging to various options including the implantable VAD, heart transplantation, medical treatment and other advanced therapies.

Fig. 2.21 TMDU/TIT extracorporeal disposable centrifugal blood pump.

Fig. 2.22 Levitation and motor system of TMDU/TIT extracorporeal disposable blood pump.

The implantable LVAD blood pump developed by TMDU/TIT reduces the overall pump height and makes it acceptable as an implantable system. A direct-drive stator is installed inside the doughnut-shaped impeller (inner stator drive) by eliminating the external motor of the disposable system (Fig. 2.23 and Fig. 2.24). The

ring-shaped permanent magnets embedded inside the doughnut-shaped impeller became the direct-drive motor rotor. The overall dimensions of the maglev implantable pump are 65 mm in diameter and 40 mm in height. An impeller stability of $< 20 \mu\text{m}$ is demonstrated as the pump is connected to the simulated left ventricular apex in the mock circulatory loop and under 8 g of acceleration testing.

Fig. 2.23 TMDU/TIT maglev implantable centrifugal blood pump.

Fig. 2.24 Levitation and motor system of TMDU/TIT maglev implantable centrifugal blood pump.

A flow rate of 6 L/min is obtained at a pressure head of 105 mmHg and a rotational speed around 1,800 rpm. The maglev and motor power consumption is 1.5 and 5 W respectively with an efficiency of 21 %.

As a summary, Table 2.1 lists the main characteristics for these blood pumps.

Table 2.1 Comparison of different blood pumps

Pump name	Pump type	Size (Diameter × Height) (mm)	Weight (g)	Support Bearing Type	DOF & Control Direction	Power (W)	Stage	Remark
Jarvik 2000	Axial	Φ25×55	90	Contact Bearing	-	-	Clinical Trial	Second Generation
DeBakey	Axial	Φ25×86	95	Contact Bearing	-	<10	Clinical Trial	Second Generation
University of Virginia	Centrifugal	Φ115×196	365	Magnetic Bearing	5 DOF	50	-	Third Generation
DuraHeart	Centrifugal	Φ72×45	540	Magnetic Bearing	3 DOF, Axial	8-10	Clinical Trial	Third Generation
Berlin Heart INCOR	Axial	Φ30×120	200	Magnetic Bearing	1 DOF, Axial	6-8	Clinical Trial	Third Generation
Levitronix	Centrifugal	Φ87×70	1700	Magnetic Bearing	-	-	Clinical Trial	Third Generation
HeartMate III	Centrifugal	Φ69×35	535	Magnetic Bearing	2 DOF, Radial	8	Animal Test	Third Generation
Magne VAD	Axial	Φ30×60	82	Magnetic Bearing	1 DOF, Axial	5	Animal Test	Third Generation
Ibaraki University	Centrifugal	Φ95×56	-	Magnetic Bearing	3 DOF, Axial	8	In Vitro Test	Third Generation
TMDU/TIT	Centrifugal	Φ75×45	100	Magnetic Bearing	2 DOF, Radial	5.7	In Vitro Test	Third Generation
TMDU/TIT	Centrifugal	Φ65×40	—	Magnetic Bearing	2 DOF, Axial	6.5	In Vitro Test	Third Generation

2.4 Key technologies in third-generation rotary blood pumps

The maglev system is the most critical component in the third-generation blood pump. The maglev system requires a rotor, position sensors, electromagnets, and an instrument control system. The maglev system also requires an active control method. The early-stage maglev rotary blood pump system was larger and more complex than the second-generation rotary blood pump system because the electromagnetic force per area was weaker. Many control axes were required to obtain stable and reliable mechanical noncontact operation. With the progress in understanding and technology, fewer axes are now required, as can be seen in the Berlin Heart INCOR blood pump, the Gold Medical Technologies MagneVAD blood pump and the HeartQuest blood pump, where only one-axis control is used to obtain a stable operation.

The factors for the improvement of the maglev system include (a) the design of suspension mechanism and testing (structure and magnetic field with finite element analysis (FEA)); (b) the control theory; and (c) the durability and stability of sensors and instruments.

As for the design of the suspension mechanism, FEA can be fully utilized to simulate conditions that are difficult to speculate and to minimize the development time and effort. Levitation control stability and robust performances should be evaluated as a blood pump system in the simulated actual environment rather than a levitation system alone. A better understanding of rotor dynamics together with pump design should help to improve the levitation mechanism and its theory. Because all

these devices will be used in the pulsatile environment receiving fairly high impact from the native heart, its control stability must be assured in the simulated pulsatile mock loop and in in-vivo animal studies.

As for the number of control axes, a simpler maglev system is preferred for a long-life blood pump, because as the number of control axes increases, the probability of failure increases. According to the theory of Earnshaw [44], zero-axis levitation control with the permanent magnet alone is not possible, and at least one-axis control is required to obtain a stable performance. To compensate for the shortcomings by reducing the number of axes involved in the control, a combination of the active and passive mechanisms is a possible solution.

Durability and stability of the sensors and control system are the third problem in the maglev system. As all the equipments are operated in the biological environment with a body temperature of 37 °C and with high humidity, the electronics and sealing performance will degrade and shorten the life span of the blood pump devices.

Chapter 3

Concept design of VAD

3.1 Design consideration of blood pump

The blood pump design takes into account anatomical and physiological consideration complying with mechanical, electrical and material requirements.

Firstly, bleeding is a significant problem associated with patients who have received VADs. In fact, 30 % of patients suffer from this problem. The risk of consequent infection is another important problem. For medical and surgical considerations, a completely intra-ventricular pump is the first choice. It eliminates the need for inflow and outflow grafts as well as their anastomoses reducing the risk of bleeding and infection.

Another important consideration for the design of the blood pump is hemolysis and thrombus. Hemolysis is the tearing of red blood cells, emptying the content of the cells in the blood stream resulting in freed hemoglobin (the normal level of plasma free hemoglobin is around 10 mg/dL). A blood pump with a normalized index of hemolysis (NIH) of 0.005 g/100 L and lower is considered to be almost athromatic for red blood cells. Thrombus refers to the formation of blood clots at some areas of low flow. In order to prevent blood clots, no contact between the rotating part and the stationary parts is preferred [15].

The required flow rate and pressure is also an important consideration. Depending on the severity of the failure, the blood pump is expected to work at flow rates of between 1 to 10 L/min against a pressure of as high as 200 mmHg, but it is more commonly used at a flow rate of between 3 to 5 L/min against a pressure of 100 mmHg.

The biocompatibility of the material for blood pumps also needs to be considered. Biocompatibility is important to minimize irritation, rejection and thrombogenesis. Because of high strength, good wear resistance and biocompatibility, titanium is frequently chosen as the material of blood pumps.

There are two flow types of rotary blood pumps: centrifugal flow blood pumps and axial flow blood pumps. In centrifugal blood pumps, the rotation of the impeller produces a centrifugal force that drags blood from the inlet port on the top to the outlet port at the bottom. To produce rotation of the impeller, the impeller is magnetically coupled to an electric motor by means of permanent magnets. This coupling is made either magnetically by means of permanent magnets located under the impeller and the rotor of the motor or mechanically by means of a shaft interposed between the impeller and the motor's rotor. A problem related to centrifugal pumps is that although they are much smaller than pulsatile pumps, they are still too large to be totally implantable in the human thorax thus eliminating the possibility of intra-ventricular implantation.

To overcome the above mentioned problem related to centrifugal pumps, axial flow ventricular assist blood pumps were developed. These axial flow pumps can decrease the hemolysis rate by decreasing the time of exposure of the blood to

frictional force and by reducing the intensity of these forces. Another advantage is that axial flow blood pumps are generally significantly smaller than centrifugal pumps. Therefore, they can be placed inside the heart to reduce the possibility of bleeding and infection.

3.2 Enclosed-impeller axial flow blood pump

After careful studies of the existing VAD systems, this thesis proposed a VAD with a novel design to achieve better performance and to simplify its manufacturing process. The proposed pump design is to achieve a pressure head of 100 mmHg with a flow rate of 5 L/min.

Fig. 3.1 Schematic structure of enclosed-impeller blood pump.

1. inlet cap 2. axial magnetic bearings 3. axial gap sensors 4. straightener 5. attractive disc 6. stator housing 7. permanent magnets 8. impeller 9. radial gap sensors 10. diffuser 11. outlet cap 12. axial magnetic bearing housing 13. casing 14. self-bearing motor housing 15. tachometer 16. motor spacer

Fig. 3.2 Basic dimension of the enclosed-impeller blood pump.

The cross-section schematic structure and basic dimension of the proposed blood pump are shown in Fig. 3.1 and Fig. 3.2 respectively. It is an axial flow type blood pump, about 60 mm in length and 24 mm in diameter. The blood pump is composed of two sub systems: enclosed-impeller pump system and self-bearing motor systems. The blood firstly flows into the pump through the inlet cap of the pump and is straightened by the straightener. Then the blood is pumped by the enclosed-impeller which is driven and levitated by two self-bearing motors and two axial magnetic bearings to increase its pressure head. After that, the blood is returned back to straight flow with further increase pressure head by the diffuser next to the impeller and flows out of the pump via outlet cap. With this structure and dimension, the proposed blood pump has the advantage to be implanted inside the heart as shown in Fig. 3.3 to reduce the possibility of bleeding and infection.

Fig. 3.3 Implantation of enclosed-impeller blood pump.

3.2.1 Enclosed-impeller pump system

The pump subsystem is mainly composed of an enclosed-impeller, a straightener, a diffuser, an inlet and an outlet cap.

3.2.1.1 Enclosed-impeller

Current axial flow blood pumps use permanent magnets inserted either in the central hub or in the impeller blades. However both methods require compromised considerations. Insertion of the permanent magnets (PMs) in the central hub requires a larger hub to locate the permanent magnets close to the motor windings for obvious electromagnetic coupling reasons; but a small hub is preferred as it can increase the volume of pumped blood. Insertion of the permanent magnets in the impeller blades also yields a compromise since the geometry of the blades must be curved for pumping efficiency. As a consequence, some of the embedded magnets move further away from the winding as the blade is curving [14].

In the proposed design, an enclosed-impeller is developed to eliminate the above drawbacks. Firstly, an impeller comprising four blades and a hub shaft is designed according to pump flow dynamic requirement (illustrated in Chapter 4). It is 12 mm in outer diameter and 16 mm in length and made of biocompatible titanium material. The model and basic dimension of the proposed impeller are shown in Fig. 3.4 and Fig. 3.5 respectively.

Fig. 3.4 3-D model of proposed impeller.

Fig. 3.5 Basic dimension of proposed impeller.

Then, the impeller is fixed into a casing to form the enclosed-impeller. At the same time, the outer surface of the casing is inserted eight elongated permanent magnets axially to form the rotor of the self-bearing motor. The whole structure of the enclosed-impeller is shown in Fig. 3.6. In this structure, the blades of impeller could be fully designed according to pump fluid dynamic optimal requirement and the permanent magnets of the motor rotor could be closer to the winding coils of the motor stator. Therefore, the blood pump could provide both better pumping efficiency and motor efficiency. And it is easy to manufacture because it needs not cut the blades or the hub of the impeller to locate the permanent magnets of the motor.

Fig. 3.6 3-D model of proposed enclosed-impeller.

3.2.1.2 Diffuser

The proposed diffuser for the blood pump is shown in Fig. 3.7 and Fig. 3.8. The diffuser is composed of four blades and a hub with coned head. It is 12 mm in diameter

and 6 mm in length and made of biocompatible material titanium. The function of the diffuser is to further increase the pressure of the blood and to reduce the blood damage. The coned head is used to prevent a dramatic change flow speed in the blood pump. The detail design and analysis will be explained in Chapter 4.

Fig. 3.7 3-D model of proposed diffuser.

Fig. 3.8 Basic dimension of proposed diffuser.

3.2.1.3 Straightener

Fig. 3.9 Model and dimension of proposed straightener.

The proposed straightener is composed of four straight blades and a cone type hub as shown in Fig. 3.9. It is 12 mm in diameter and 6 mm in length and made of titanium. The function of the straightener is to straighten the blood flow and reduce the flow turbulence. A proximity sensor locates at the center of the hub to measure the gap between the straightener and impeller.

3.2.1.4 Inlet cap and outlet cap

The function of the cap is to support the blood pump and form flow channel for the blood. The cap is also made of titanium. Figure 3.10 shows the model and dimension of the inlet and outlet cap.

Fig. 3.10 Model and dimension of proposed flow cap.

3.2.2 Self-bearing motor system

Self-bearing motor sub system is mainly composed of two self-bearing motors and two axis magnetic bearings. As shown in Fig. 3.11 and Fig. 3.12, whole self-bearing motor sub system is 24 mm in diameter, and 31.5 mm in length. Two self-bearing motors drive and levitate the enclosed-impeller in the blood pump. Two axial magnetic bearings keep the impeller stable in axial direction. A tachometer sensor is inserted into stator to sense the rotating angle and speed of the PMs. Four gap sensors locate at the upper and lower of the casing to detect the radial gap of the impeller. A gap sensor in the straightener detects the axis position of the impeller.

Fig. 3.11 3-D model of proposed self-bearing motor system.

Fig. 3.12 Basic dimension of proposed self-bearing motor system.

3.2.2.1 Self-bearing motor

As stated in Chapter 2, the contact bearings tend to cause thrombus formation in the blood pump. A self-bearing motor, which combines the motor and the radial magnetic bearings together as a single unit to drive and suspend the impeller, has been proposed by the authors to eliminate this drawback [45]. It comprises a 4-pole rotor and a 6-slot stator. The rotor of the self-bearing motor is formed by eight PMs bonded on the surface of the impeller casing which is made of soft iron. Both the PMs and casing are coated with titanium for biocompatible purpose. The model and basic dimension of the rotor are shown in Fig. 3.13 and Fig. 3.14 respectively.

Fig. 3.13 3-D model of proposed rotor of self-bearing motor.

Fig. 3.14 Basic dimension of proposed rotor of self-bearing motor.

The stator of the self-bearing motor is made of silicon steel, coating with titanium for biocompatible purpose. It is 22 mm in outer diameter, 16 mm in inner diameter and 6 mm in length as shown in Fig. 3.15 and Fig. 3.16. Six motoring coils and six levitation coils are wound on the six slots. A tachometer sensor is inserted into stator to sense the rotating angle and speed of the PMs. The detail principle of the self-bearing motor will be illustrated in Chapter 5.

Fig. 3.15 3-D model of proposed stator of self-bearing motor.

Fig. 3.16 Basic dimension of proposed stator of self-bearing motor.

3.2.2.2 Axial magnetic bearings

Two axial magnetic bearings are located at rear and front of the impeller. Each axial magnetic bearing is composed of an attractive solenoid and an attractive disc on the casing. Both the solenoid housing and the attractive disc are made of soft iron and coated with titanium for biocompatible purpose. The model and dimension of proposed axial magnetic bearing are shown in Fig. 3.17. By detecting the gap between the straightener and the impeller through an axial gap sensor inserted in the straightener, the magnetic force generated by the two axial magnetic bearings is adjusted to overcome the axial force generated by the impeller during pumping blood, and keep the impeller stable in axial direction.

Fig. 3.17 Model and dimension of proposed axial magnetic bearings.

Chapter 4

Pump design of VAD

4.1 Euler pump principle

Fig. 4.1 Diagram of generalized pump.

Figure 4.1 shows the diagram of a generalized pump, where ω is the angular velocity of the pump, T is the applied driving torque, r_1 and r_2 are the inlet and outlet diameters of the pump, and $c_{\theta 1}$ and $c_{\theta 2}$ are the tangential velocities at inlet and outlet respectively. The fluid flowing into the pump from inlet is sped up by the rotating blades of the pump and flow out of the outlet with increased pressure head.

For the pump system shown in Fig. 4.1, according to Newton's second law, we have

Applied torque = rate of change of angular momentum.

The torque can also be expressed analytically as

$$T = \dot{m}(r_2 c_{\theta 2} - r_1 c_{\theta 1}), \quad (4-1)$$

where

T - torque,

$\dot{m}$ - mass flow rate,

r_1 - radius of inlet,

r_2 - radius of outlet,

$c_{\theta 1}$ - tangential velocity at inlet,

$c_{\theta 2}$ - tangential velocity at outlet.

According to the energy equation for steady incompressible flow, and ignoring the potential energy and temperature effects, we have

$$W = \dot{m}((p_2 - p_1) / \rho + (c_2^2 - c_1^2) / 2) = \dot{m}(p_{02} - p_{01}) / \rho, \quad (4-2)$$

where

W - flow energy,

p_1 - inlet pressure,

p_2 - outlet pressure,

ρ - fluid density,

c_1 - inlet velocity,

c_2 - outlet velocity,

p_{01} - inlet stagnation pressure,

p_{02} - outlet stagnation pressure.

At the same time, the flow energy W can be expressed as

$$W = T\omega, \quad (4-3)$$

where ω is the angular velocity.

Combining Eq. (4-1), Eq. (4-2) and Eq. (4-3), we get

$$(p_{02} - p_{01}) / \rho = \omega(r_2 c_{\theta 2} - r_1 c_{\theta 1}) = U_2 c_{\theta 2} - U_1 c_{\theta 1}, \quad (4-4)$$

where $U = r\omega$ is the blade speed. Eq. (4-4) is known as the Euler pump equation. It expresses the relationship between the pressure and the blade speed and is the basic principle of the pump.

For an axial flow pump, the blade radii of inlet and outlet are the same, that is, $r_1 = r_2 = r$ and the inlet tangential velocity is zero, $c_{\theta 1} = 0$. Therefore, the Euler pump equation (4-4) for the axial flow pump can be simplified as

$$(p_{02} - p_{01}) / \rho = U_2 c_{\theta 2}. \quad (4-5)$$

From Fig. 4.1, it is known that if the inlet velocity c is axial, there must be tangential velocity $c_{\theta 2}$ downstream from the impeller. In order to improve the efficiency, a diffuser is added to restore the flow at outlet to the axial direction and convert the dynamic pressure into static pressure. Therefore, an axial flow pump usually consists of a impeller and a diffuser. The pressure difference across the axial flow pump can be expressed as

$$(p_{02} - p_{01}) = p_3 - p_1 = \Delta p, \quad (4-6)$$

Where p_3 is the pressure at the outlet of the diffuser and Δp is the pressure difference across the pump. Therefore, the Euler pump equation (4-4) for the axial flow pump with an impeller and a diffuser can be further expressed as

$$\Delta p / \rho = U_2 c_{\theta 2} . \quad (4-7)$$

4.2 Two-dimensional analysis of axial flow pump

The simplest approach to the study of the axial flow pump is two-dimensional analysis method [46]. It assumes that the flow conditions prevailing at the mean radius fully represent the flow at all other radii. Another assumption is that the fluid will flow along the blade. Hence, the flow velocity of an axial flow pump at mean radius is shown as in Fig. 4.2.

Fig. 4.2 Velocity diagrams of axial flow pump.

In order to simplify the design of the axial flow pump, we define two dimensionless coefficients, the flow coefficient ϕ and the work coefficient ψ . The flow coefficient ϕ is formulated as

$$\phi = \frac{c_x}{U}, \quad (4-8)$$

where c_x is the axial flow velocity. The work coefficient ψ is formulated as

$$\psi = \frac{(p_{02} - p_{01})}{\rho U^2} = \frac{\Delta p}{\rho U^2}. \quad (4-9)$$

With these two coefficients and combining the velocity diagrams together, we can get the assembled velocity diagram for both the impeller and the diffuser of an axial flow pump as shown in Fig. 4.3.

Fig. 4.3 Assembled velocity diagram for axial flow pump.

From Fig. 4.3, we can calculate the inlet angle β_1 and outlet angle β_2 of the impeller and inlet angle α_2 and outlet angle α_3 of the diffuser. For the impeller, we get

$$\beta_1 = \tan^{-1}\left(\frac{1}{\phi}\right), \quad (4-10)$$

$$\beta_2 = \tan^{-1}\left(\frac{1-\psi}{\phi}\right). \quad (4-11)$$

For the diffuser, we get

$$\alpha_2 = \tan^{-1}\left(\frac{\psi}{\phi}\right), \quad (4-12)$$

$$\alpha_3 = 0. \quad (4-13)$$

Eqs. (4-10) - (4-13) are basic equations for the design of the axial flow pump.

4.3 Pump design of blood pump

As discussed in Chapter 3, the requirements of the blood pump are pressure difference $\Delta p = 100 \text{ mmHg}$ and flow rate $Q = 5 \text{ L/min}$. The outlet angle of the impeller of the blood pump is usually designed to be zero ($\beta_2 = 0$) to reduce the damage to the blood. Considering the volume of the heart, the length of the impeller would be 17 mm, diffuser 6 mm, radius of blade hub $r_h = 2 \text{ mm}$, tip of the blade $r_t = 6 \text{ mm}$.

According to Eq. (4-9), the work coefficient of the blood pump is $\psi = 1$. The required rotating speed at mean radius is

$$U = \sqrt{\frac{\Delta p}{\rho \psi}} = \sqrt{\frac{13.6 \times 9.8 \times 0.1}{1.055 \times 1}} = 3.55 \text{ m/s},$$

where $\rho = 1.055 \text{ kg/m}^3$ is the blood density. The required angular velocity of the impeller is

$$\omega = U / r_m = 3.55 / 0.004 = 887.5 \text{ rad/s} = 8,479 \text{ rpm},$$

where $r_m = (r_t + r_h) / 2$ is the mean radius of the impeller. The flow rate is

$$Q = 5 \text{ L/min} = 8.33 \times 10^{-5} \text{ m}^3/\text{s}.$$

Hence, the axial flow velocity is

$$c_x = 1.2 \times \frac{Q}{\pi(r_t^2 - r_h^2)} = 1.2 \times \frac{8.33 \times 10^{-5}}{3.14 \times (6^2 - 2^2) \times 10^{-6}} = 1 \text{ m/s},$$

where 1.2 is the experimental coefficient considering the thickness of the blades [46].

Thus, the flow coefficient of the blood pump is

$$\phi = \frac{c_x}{U} = \frac{1}{3.55} = 0.28.$$

Therefore, the inlet angle of the impeller is calculated as

$$\beta_1 = \arctg\left(\frac{1}{\phi}\right) / \pi \times 180^\circ = 74^\circ.$$

In addition, the inlet angle of the diffuser is calculated as

$$\alpha_2 = \arctg\left(\frac{\psi}{\phi}\right) / \pi \times 180^\circ = 74^\circ.$$

In order to simplify the fabrication of the blade, it is designed to be parabolic in shape. The thicknesses of the impeller blade and the diffuser blade are 0.6 mm and 0.3 mm respectively. The designed impeller for the blood pump is shown in Fig. 4.4. The impeller is inserted into a casing. Thus an enclosed-impeller is gotten as shown in Fig. 4.5.

Fig. 4.4 Designed impeller for blood pump.

Fig. 4.5 Enclosed-impeller of blood pump.

The designed diffuser for the blood pump is shown in Fig. 4.6. The coned head is used to prevent a dramatic change in flow speed in the blood pump.

Fig. 4.6 Designed diffuser for blood pump.

The design parameters of the blood pump are listed at Table 4.1.

Table 4.1 Design parameters of blood pump

Length of impeller	17 mm
Length of diffuser	6 mm
Radius of blade hub	2 mm
Radius of blade tip	6 mm
Thickness of casing	1 mm
Inlet angle of the impeller (β_1)	74°
Outlet angle of the impeller (β_2)	0°
Inlet angle of the diffuser (α_2)	74°
Outlet angle of the diffuser (α_3)	0°
Thickness of the impeller blade	0.6 mm
Thickness of the diffuser blade	0.3 mm

4.4 CFD simulating of blood pump

4.4.1 Simulation model of blood pump

In order to get the pump characteristics of the proposed blood pump, a simulating model of the blood pump (Fig. 4.7) is created based on the dimensions in Table 4. A computational fluid dynamics (CFD) analysis of the pump is carried out to analyze the pump by using the CFD simulation software CFdesign [47].

Fig. 4.7 Section view of axial flow blood pump model.

4.4.2 Simulation condition

4.4.2.1 Blood property

The fluid stresses dominating the flow paths in the blood pump are large enough to consider the blood as Newtonian fluid [48]. The blood is assumed to have a viscosity of 0.0035 Pa·s and a density of 1,055 kg/m³ [49].

4.4.2.2 Flow model

The k - ε turbulence flow model [50] has been widely used in the design of axial flow blood pumps [51]. Thus, it is applied in the present simulation. The k - ε turbulence model solves the nonlinear Reynolds stress terms in the Reynolds Averaged Navier-Stokes equations by approximating turbulent flow conditions for the turbulent kinetic energy k and the dissipation rate of the kinetic energy ε .

4.4.2.3 Blood trauma

Blood trauma is primarily determined by the exposure time of the blood cell and the shear stress exerted on it. Bludszuweit [52] developed a scalar shear stress value from the comparative stress theory of fluids that is analogous to the Mises yield criterion for solid materials. The scalar shear stress value includes the six components of the stress tensor and represents the level of shear experienced by the blood. This scalar stress estimation provides information about regions of high stress levels (possible hemolysis) and extremely low levels (likely stasis risking thrombosis). The same approach is adopted in the present study to account for the three-dimensional

shear field and calculate the scalar stress (ss) according to Bludszuweit's stress formula:

$$ss = \left[\frac{1}{3}(\tau_{ii}^2 + \tau_{jj}^2 + \tau_{kk}^2) - \frac{1}{3}(\tau_{ii}\tau_{jj} + \tau_{jj}\tau_{kk} + \tau_{kk}\tau_{ii}) + (\tau_{ij}^2 + \tau_{jk}^2 + \tau_{ki}^2) \right]^{1/2} \quad (4-11)$$

The blood damage threshold is a shear stress of 450 Pa for an exposure time of 620 ms [53].

4.4.2.4 Boundary conditions

Flow through the pump is assumed to be steady with constant boundary conditions for these simulations. The no-slip boundary condition is applied to the stationary walls such that the fluid velocity values along the boundary would be equal to zero. Zero pressure is set at the inlet of the pump and flow rate is set to 3 L/min, 5 L/min or 7 L/min at the outlet of the pump.

4.4.2.5 Mesh quality and convergence criterion

The full computational model of the axial flow blood pump consists of approximately 264,397 fluid nodes, 1,008,673 fluid elements, 38,825 solid nodes and 263,328 solid elements. The convergence criterion is that the results change less than 5 % over the last 20 % of the steps of the total iterations.

4.4.3 Simulation results and analysis of blood pump

Fifteen different cases were simulated. Flow rate ranges from 3 - 7 L/min and rotating speed ranges from 8,000 - 10,000 rpm.

4.4.3.1 Flow characteristics

The pressure versus flow rate curve for the blood pump is shown in Fig. 4.8.

Fig. 4.8 Pressure versus flow rate curve of blood pump.

Fig. 4.8 shows that the differential pressure across the pump for a given rotating speed decreased with increasing flow rate because of the flow loss. Differential pressures ranging from 8 – 76.6 mmHg were generated for this pump. The differential pressures generated at rotating speed of 8,000 rpm, flow rate of 7 L/min and at rotating speed of 9,000 rpm, flow rate of 7 L/min were lower than 30 mmHg.

The results show that the pump should be operated at a rotating speed of above 9,000 rpm and a flow rate of below 7 L/min.

4.4.3.2 Flow profile

The flow pattern of the internal flow in the blood pump needs to be studied in detail, as irregular flow pattern through the pump can lead to regions of high shear stresses and possible flow stasis.

Fig. 4.9 shows the pressure distribution across the pump at a rotating speed of 9,000 rpm and a flow rate of 5 L/min. Fig. 4.10 shows the velocity distribution across the pump. Fig. 4.11 and Fig. 4.12 show the particle track inside and outside the casing respectively. From these figures, it is found that the impeller region appears to have a smooth flow profile and the diffuser region demonstrates no signs of irregular flow patterns (separations or vortices).

Fig. 4.9 Pressure distribution across blood pump (9,000 rpm, 5 L/min).

Fig. 4.10 Velocity distribution across blood pump (9,000 rpm, 5 L/min).

Fig. 4.11 Particle track inside outer casing (9,000 rpm, 5 L/min).

Fig. 4.12 Particle track outside outer casing (9,000 rpm, 5 L/min).

4.4.3.3 Driving torque and pump efficiency

The torque needed to drive the pump is estimated by the following equation:

$$T = \int_A P \vec{r} \cdot d\vec{A}, \tag{4-13}$$

where P is the pressure on the impeller, $\vec{r}$ is the distance from the axis of the impeller and $\vec{A}$ is the surface of the impeller.

Fig. 4.13 shows computed torques at different flow rates and rotating speeds. It shows that the required torques ranging from 1.5 N.mm to 3 N.mm are within the practical range.

Fig. 4.13 Driving torque of blood pump.

With the computed torques from Eq. (4-13), the pump efficiency can be obtained according to the formula,

$$\eta = \frac{\Delta P Q}{T \omega}, \tag{4-14}$$

where ΔP is the pressure head, Q is the flow rate, T is the applied torque and ω is the rotating speed.

Fig. 4.14 shows the estimated pump efficiencies under different flow rates and rotating speeds. The estimated efficiencies for these operating conditions range from 6 - 30 %. The results indicate that the pump should be operated at a rotating speed above 9,000 rpm and a flow rate below 7 L/min.

Fig. 4.14 Pump efficiency of blood pump.

4.4.3.4 Axial force

The axis force exerted on the impeller must be estimated to ensure an effective magnetic bearings suspension design. The axial force is calculated by the following equation:

$$F = \int_A P \vec{n} \cdot d\vec{A}, \tag{4-15}$$

where $\vec{n}$ is the unit vector along the axis of impeller.

Fig. 4.15 shows that the axial forces under different flow conditions range from 0.18 N to 0.82 N, which are within the practical range.

Fig. 4.15 Axial force of blood pump.

4.4.3.5. Shear stress and residence time

Fig. 4.16 shows that the highest level of shear stress (about 200 Pa) is located at the gap between the hollow-cylindrical casing of the impeller and the housing at the condition of 9,000 rpm rotating speed and 5 L/min flow rate. The maximum residence time of the blood particles in this region is below 1.5 ms as shown in Fig. 4.17. The average exposure time of blood particle across the pump is 146 ms. Based on the blood damage criterion, the blood cells subjected to such a level of shear stress for such a short duration of time will result in lesser blood trauma risk.

Fig. 4.16 Shear stress at outer casing (9,000 rpm, 5 L/min).

Fig. 4.17 Residence time of blood at outer casing (9,000 rpm, 5 L/min).

4.4.3.6. Flow leakage

The flow leakage between the impeller casing and the pump house is 0.3 L/min under the condition of 9,000 rpm rotating speed and 5 L/min flow rate. It is about 6 % of the total flow rate.

4.4.4 Conclusion

In this section, the flow characteristics of the blood pump are successfully simulated and analyzed by CFdesign software. The driving torques and axial forces of the pump are determined to facilitate the design of a magnetically suspended enclosed-impeller driven by the self-bearing motor. In addition, the low shear stress and short residence time of the pump flow indicate that the pump has low blood trauma and thromboembolic tendencies. The above analytical results validate the proposed enclosed-impeller axial blood pump to be a better alternative design for a long term, effective and reliable implantable device.

Chapter 5

Design and development of self-bearing motor

As stated in Chapter 3, the self-bearing motor is the key component in the proposed blood pump. Hence, in this chapter, the background and principle of self-bearing motor are introduced. An experimental test rig for the proposed self-bearing motor is also designed and set up to validate the principle and develop the control scheme.

5.1 Introduction of magnetic bearing system

5.1.1 Active magnetic bearings

The application of active magnetic bearings is based upon the principle that an electromagnet will attract ferromagnetic material. A ferromagnetic rotor can thus be supported in a magnetic field generated in the bearing electromagnet stator as shown in Fig. 5.1. It is a natural tendency for the stator to attract the rotor when it is energized, therefore a control system is required to modulate the magnetic field and to maintain the rotor at a desired position.

Fig. 5.1 Principle of active magnetic bearings.

Hence, an active magnetic bearing system consists of electromagnet bearing actuators, position sensors, a control system and power amplifiers, as shown in Fig. 5.2. The bearing actuators and sensors are located in the machine assembly, while the control system and amplifiers are generally located externally. Non-contact position sensors monitor shaft position and feed this information to the control system. This, in turn, based on the desired response of the system commands, drives currents to the actuator via current amplifiers. These currents are converted to forces by the actuator and act on the rotor to adjust its position and provide damping to the rotor.

Fig. 5.2 Control system of active magnetic bearings.

The active magnetic bearing systems for application consist of two radial magnetic bearings, one motor and one axial magnetic bearing as shown in Fig. 5.3.

Fig. 5.3 Schematic structure of active magnetic bearings systems.

The technology of active magnetic bearings was first applied for rotating machinery by the University of Virginia research group in 1950 [16]. Since then active magnetic bearings have been widely developed. Today it is a matured technology and commercial magnetic bearings are available. An example of a

commercial magnetic bearing is the SKF magnetic bearings developed by SKF company [54] as shown in Fig. 5.4.

Fig. 5.4 SKF magnetic bearing.

However, active magnetic bearings systems require a separate driving motor to drive the rotor, therefore a relatively long shaft is required which leads to bending vibrations at relatively low frequency. Hence, the performance of the active magnetic bearing system in high speed is affected.

5.1.2 Self-bearing motor

To overcome the problem of traditional active magnetic bearings, self-bearing motors (also termed as bearingless motors) are developed. Self-bearing motors combine active magnetic bearing and motoring capabilities into a single actuator i.e. to introduce both radial force and torque production. The schematic structure of self-bearing motor systems is shown in Fig. 5.5. It is composed of two self-bearing motors and one axial magnetic bearing. The advantages of such a design are higher power

density and shorter shaft length characteristics, making them advantageous for high-speed applications.

Fig. 5.5 Schematic structure of self-bearing motor systems.

The first self-bearing motor was invented and created in the mid 1970's. A primitive electromagnet with stator windings having pole number p and $(p+2)$ was proposed by Hermann [55]. This electromagnet was proposed as a motor with a radial magnetic bearing function. In 1985, a stepping motor which was magnetically combined with a magnetic bearing was proposed by Higuchi [56]. It included an inherently decoupled structure for torque and radial force, while taking advantage of the motor exciting current.

In 1989, Chiba et al. proposed a general concept for the self-bearing motor [57]. Based on vector control theory, it is concluded that most electrical machines can be used in a self-bearing drive with additional suspension windings driven by a 3-phase inverter. The concept has been developed theoretically with the strong support of Chiba et al. [58]. Since then, the concept has been applied to synchronous reluctance, induction, permanent magnet and disc-type self-bearing drive, as well as

homopolar, hybrid and consequent-pole self-bearing drives. By the middle of the 1990's, the self-bearing motor had been widely researched throughout the world.

There are two main types of self-bearing motor designs currently under intensive research: a) Reluctance force type and b) Lorentz force type.

5.1.2.1 Reluctance force type self-bearing motor

It has been studied by Chiba, et al. [57]. One common feature of the designs is that attractive forces between the rotor and stator (reluctance-type forces) provide the bearing function, and magnetic forces on current carrying conductors (Lorentz-type forces) produce the motoring torque.

Fig. 5.6 shows the principles of the reluctance force type self-bearing motor. When the rotor is positioned in the center, symmetrical 4-pole fluxes ϕ_p are generated by the four permanent magnets. When the machine is driven at no load, the 4-pole fluxes produced by the currents of 4-pole windings N_a and N_b are small enough to be neglected. In addition to the conventional 4-pole motor windings, 2-pole radial force windings N_α and N_β are also wound in the stator. If the current of radial force winding N_α is positive, 2-pole magnetic fluxes ϕ_α are generated. As a result, the flux density is increased in air-gap 2, but it is decreased in air-gap 1. Therefore, a radial force F whose direction is negative in the α -axis is generated. If the current of the winding N_α is negative, a radial force can be produced in the positive direction of α -axis. With current in the N_β winding, β -axis direction force can be produced.

As a result, radial magnetic forces can be generated in any direction by regulating N_α and N_β winding currents.

Fig. 5.6 Principle of reluctance force type self-bearing motors.

The drawback of the reluctance force type self-bearing motor is its low efficiency and long term reliability. Non-linearity of the magnetic material also causes undesirable coupling between the two orthogonal directions. The surface permanent magnets secured onto the rotor surfaces must be thin. This results in higher manufacturing costs, weaker rotor strength and possible demagnetization problems in long term application.

5.1.2.2 Lorentz force type self-bearing motor

It was first studied and employed by Okada et al. [59] and Kim and Stephens [60] in 2002. Many researchers have followed their studies since then [45, 61]. This method uses Lorentz type forces to produce both the bearing levitation forces and the

motoring torque. In the Lorentz force type self-bearing motor, two type coil windings are wound in the stator. The coil windings have the same poles. However, they are arranged in a different way, so that the force generated by one set coil winding is only in the radial direction, meaning the levitation force, the force by the other set coil winding is only in the peripheral direction, meaning the torque. The Lorentz force type self-bearing motor eliminates the normal tradeoff between the torque and bearing force; i.e. both the torque and bearing force increase as the thickness of the permanent magnet increases. They also have the advantages of ease in fabricating, reduced demagnetization and better dynamic performance.

5.2 Theoretical analysis of self-bearing motor

The schematic of the proposed self-bearing motor in this research is shown in Fig. 5.7 and Fig. 5.8. The rotor is secured with 8 permanent magnets (PMs) magnetized in the radial direction. Two sets of coil windings are installed in the same positions inside the stator core. One set of coil windings is for motoring purposes, the other is for levitation. The flux entry and return paths of each coil are positioned at $\pi/4$ radian apart. The arrangement of motoring coils are $U_m, W_m, V_m, U_m, W_m, V_m$. The arrangement of levitation coils are $U_b, -V_b, W_b, -U_b, V_b, -W_b$.

Fig. 5.7 Schematic of self-bearing motor.

Fig. 5.8 Linear arrangement of six-pole stator with two sets of windings and eight-magnetic-pole rotor. (Note: $\odot$, current flow out; $\otimes$, current flow in.)

The rotor is assumed to produce the following sinusoidal form flux density at the air gap

$$B_g = -B\sin(\omega t + 4\theta), \tag{5-1}$$

where

$$B = \frac{B_r l_m}{\mu_r g_m}, \tag{5-2}$$

$$g_m = kg_a + \frac{l_m}{\mu_r}. \tag{5-3}$$

The variables in this chapter are defined in Table 5.1.

Table 5.1 Definition of variables

A	magnitude of motoring current
B	magnitude of flux density
C	magnitude of levitation current
θ	mechanical angle
ω	electric frequency
φ	phase of motoring current
ϕ	phase of levitation current
B_g	flux density in the air gap
l_m	thickness of permanent magnet
g_a	air gap
g_m	efficiency gap
B_r	residual flux density
μ_r	relative permeability of air
l	length of stator
r	radius of rotor
k	efficiency coefficient of air gap
N	number of turns of coils

5.2.1 Motoring torque

Fig. 5.9 Schematic of generating motoring torque.

The principle for generating the motoring torque is illustrated in Fig. 5.9. The forces and direction experienced by the current carrying conductors of the motoring coils can be easily determined by the Lorentz relationship. The motoring torque is driven by the three-phase currents:

$$\begin{cases} I_{U_m} = A \cos(\omega t + \varphi) \\ I_{V_m} = A \cos(\omega t + \frac{2}{3}\pi + \varphi) \\ I_{W_m} = A \cos(\omega t + \frac{4}{3}\pi + \varphi) \end{cases} \quad (5-4)$$

Referring to the arrangement of the motoring coils as shown in Fig. 5.8, the current distribution in the circumferential direction can be expressed by Dirac's delta function as

$$\begin{aligned} i_m = & I_{U_m} \left[\delta\left(\theta - \left(-\frac{1}{8}\pi\right)\right) - \delta\left(\theta - \frac{1}{8}\pi\right) + \delta\left(\theta - \frac{7}{8}\pi\right) - \delta\left(\theta - \frac{9}{8}\pi\right) \right] \\ & + I_{W_m} \left[\delta\left(\theta - \frac{5}{24}\pi\right) - \delta\left(\theta - \frac{11}{24}\pi\right) + \delta\left(\theta - \frac{29}{24}\pi\right) - \delta\left(\theta - \frac{35}{24}\pi\right) \right] \\ & + I_{V_m} \left[\delta\left(\theta - \frac{13}{24}\pi\right) - \delta\left(\theta - \frac{19}{24}\pi\right) + \delta\left(\theta - \frac{37}{24}\pi\right) - \delta\left(\theta - \frac{43}{24}\pi\right) \right] \end{aligned} \quad (5-5)$$

Hence, by applying the Lorentz force principle, the torque produced by the motoring currents is

$$T = 2Nrl \int_{-\frac{\pi}{8}}^{\frac{7\pi}{8}} B_g i_m d\theta = 6NrlAB \cos \varphi \quad (5-6)$$

Eq. (5-6) shows that the torque is only controlled by the motoring current magnitude A and phase φ , and that it is independent of rotor position and time.

5.2.2 Levitation force

Fig. 5.10 Schematic of generating levitating force.

The schematic principle of generating levitating force is shown in Fig. 5.10. The levitating force is driven by the three-phase currents:

$$\begin{cases} I_{U_b} = C \cos(\omega t + \phi) \\ I_{V_b} = C \cos(\omega t + \frac{2}{3}\pi + \phi) \\ I_{W_b} = C \cos(\omega t + \frac{4}{3}\pi + \phi) \end{cases} \quad (5-7)$$

Again, referring to the arrangement of the levitating coils as shown in Fig. 5.8, the current distribution in the circumferential direction can be expressed by Dirac's delta function as

$$\begin{aligned}
 i_b = & I_{U_b} [\delta(\theta - (-\frac{1}{8}\pi)) - \delta(\theta - \frac{1}{8}\pi) - \delta(\theta - \frac{7}{8}\pi) + \delta(\theta - \frac{9}{8}\pi)] \\
 & + I_{V_b} [-\delta(\theta - \frac{5}{24}\pi) + \delta(\theta - \frac{11}{24}\pi) + \delta(\theta - \frac{29}{24}\pi) - \delta(\theta - \frac{35}{24}\pi)] \\
 & + I_{W_b} [\delta(\theta - \frac{13}{24}\pi) - \delta(\theta - \frac{19}{24}\pi) - \delta(\theta - \frac{37}{24}\pi) + \delta(\theta - \frac{43}{24}\pi)]
 \end{aligned} \quad (5-8)$$

The levitating forces in the x and y directions can be calculated as follows:

$$\begin{cases} F_x = NI \int_{\frac{\pi}{8}}^{\frac{15\pi}{8}} B_g i_b \sin \theta d\theta = -\frac{3\sqrt{2+\sqrt{2}}}{2} NBI C \sin \phi \\ F_y = NI \int_{\frac{\pi}{8}}^{\frac{15\pi}{8}} B_g i_b \cos \theta d\theta = \frac{3\sqrt{2+\sqrt{2}}}{2} NBI C \cos \phi \end{cases} \quad (5-9)$$

From Eq. (5-9), it is observed that the levitating force is independent of rotor angular position and time. It is controlled only by the levitating current magnitude C and phase ϕ .

The verification of Eqs. (5-6) and (5-9) using Matlab can be found in Appendix A.

5.3 Finite element analysis (FEA) of self-bearing motor

The theoretical model of the self-bearing motor is an ideal model based on some assumptions, such as the angle between coils being exactly $\pi/4$ and a sinusoidal distribution of flux density in the air gap. The actual model is more complicated than the analytical assumptions. Hence, in this section, Finite Element Analysis (FEA) is performed to validate the actual design of the self-bearing motor by using ANSYS software.

5.3.1 FEA model

Fig. 5.11 shows the 2-D FEA model of the self-bearing motor. The model comprises five parts: stator, rotor shaft, air gap, permanent magnets (PMs) and coils.

Fig. 5.11 2-D FEA model of self-bearing motor.

The permanent magnet is made of rare earth magnet (NdFeB 45M) with 1.05 T residual flux density and 9,000 H/A Coercivity. The material of the stator is silicon steel. H-B characteristic curve of the silicon steel is shown in Fig. 5.12. The flux density saturation point of the stator is about 2.1 T. The material of the rotor shaft is iron with a relative permeability of 9,000. The relative permeability μ_r of air is 1.

Fig. 5.12 B-H curve of silicon steel.

The coils have two models, the motoring model and the levitation model. In the motoring model, the current distribution is defined as $U_m, W_m, V_m, U_m, W_m, V_m$. In the levitation model, the current distributed is defined as $U_b, -V_b, W_b, -U_b, V_b, -W_b$. The turn of the coils is assumed to be 30 and the amplitude of the coils is assumed to be 2 A.

5.3.2 Flux density analysis

The flux density is directly related to the produced motoring torque and levitation force. Hence, the flux density in air gap needs to be examined in detail. Fig.

5.13 shows the flux distribution of the self-bearing motor. The flux density distribution of the self-bearing motor in the radial direction is shown in Fig. 5.14.

Fig. 5.13 Flux density distribution of self-bearing motor.

Fig. 5.14. Flux density distribution in radial direction.

Fig. 5.15 shows the flux density in the air gap and the theoretical value. With about 0.45 T in magnitude and 4 periods in one rotation, the simulated flux density coincides with theoretical sinusoidal waveform, except for some small deformations in the peak which are caused by the slots of the stator. Hence, the sinusoidal assumption of the magnetic flux density distribution is appropriate and validated.

Fig. 5.15 Flux density in air gap.

5.3.3 Motoring torque analysis

Fig. 5.16 shows the torque generated by the motoring coils at different rotating positions. The torque changes sinusoidally in one rotation, with 38 N.mm in magnitude and 4 signal periods. The deformations in the torque waveform are the cogging torques caused by the slots of the stator.

Fig. 5.16 Torque generated by motoring coils.

Fig. 5.17 shows the torque generated by the levitation coils. It shows that the maximum torque generated by the levitation coils is only 0.8 N.mm, 2 % of the maximum torque produced by the motoring coils. It proves that the levitating coils have almost no contribution to the motoring torque.

Fig. 5.17 Torque generated by levitation coils.

5.3.4 Levitation force Analysis

Figs. 5.18 to 5.20 show the levitation force in the x direction, levitation force in the y direction and total levitation force generated by the levitation coils respectively. The levitation forces in the x direction and the y direction vary sinusoidally in one rotation, with 1.35 N in magnitude and 4 signal periods. The total levitation force is constant at different positions, which is coincided with theoretical prediction.

Fig. 5.18 Levitation force generated by levitation coils in x direction.

Fig. 5.19 Levitation force generated by levitation coils in y direction.

Fig. 5.20 Total levitation force generated by levitation coils.

Fig. 5.21 and Fig. 5.22 show the levitation force in the x direction and the y direction generated by the motoring coils in different positions. The maximum value is 0.15 N, about 1.2 % of that produced by the levitating coils. It proves that the motoring coils have almost no contribution to the levitation force of the self-bearing motor.

Fig. 5.21 Levitation force generated by motoring coils in x direction.

Fig. 5.22 Levitation force generated by motoring coils in y direction.

5.3.5 Conclusion

The FEA simulation results show that the torque and levitation force generated by the motoring coils and levitation coils coincide with theoretical predictions, though there are some variations caused by the stator slots. The results validate the actual design of the Lorentz force type self-bearing motor.

5.4 Experimental setup of self-bearing motor test rig

5.4.1 Test rig structure

According to the operational principle of the self-bearing motor as illustrated before, a prototype experimental test rig of self-bearing motor is fabricated as shown in Fig. 5.23. It is mainly composed of a rotor, two stators, a support shaft, two eddy current proximity probes and a tachometer. The detailed assembly and part drawings of the test rig can be found in Appendix B.

Fig. 5.23 Experimental setup of self-bearing motor test rig.

Fig. 5.24 shows the schematic arrangement of the experimental test rig. In order to facilitate validation of the theoretical developments of the motoring torque and levitating force, the rotor is designed as a hollow shaft with a support shaft for emergency landing purposes. The outer surface of the rotor is secured with 2 mm thick permanent magnets as

shown in Fig. 5.25, which are radially magnetized into 8 poles to generate a sinusoidal magnetic flux field.

Fig. 5.24 Schematic arrangement of experiment test rig.

Fig. 5.25 Rotor of self-bearing motor.

The air gap between the stator and the outer surface of the rotor is 1 mm. The clearance between the rotor and the support shaft is 0.5 mm. The support shaft serves as an emergency bearing for the rotor at the preliminary stage in case of touch down due to control failure. One end of the rotor is simply supported by a brass bushing. The other end of the rotor is free. Therefore the rotor has two degrees of freedom in x and y directions.

Two sets of eddy current proximity probes are installed near the stator of the self-bearing motor to measure the rotor motion in the x and y directions. The 5 mm proximate transducer system (Bently Nevada Inc.) is used for this application. It is a non-contacting, gap-to-voltage transducer system with fast dynamic response and high bandwidth.

A simple tachometer is mounted at the top of the test rig in-line with the rotation axis of the rotor to detect the magnetic poles' angular positions as shown in Fig. 5.26. This device is constructed using fine copper wires wrapped around a ferromagnetic rod. When the rotor is rotated about its axis, the change in the magnetic flux in the sensing coil induces a Faraday's voltage. Then, this voltage is fed to the controller to monitor both the angular position and the speed of the rotor.

Fig. 5.26 Tachometer of self-bearing motor.

The stator is made of laminated silicon steel plates of 0.25 mm thickness to minimize eddy current and hysteresis losses. The length of the stator is 20 mm and it is manufactured according to the design shown in Fig. 5.8. The manufactured stator is shown in Fig. 5.27. Two sets of coil windings are mounted in the inner and outer radii of the slots of the stator, respectively. One is for motoring torque and the other for levitating force. Each coil has 30 turns of copper wire with diameter of 0.5 mm. Although two stators are installed in the test rig, only one stator is used in present experiments. The other stator will be used to fully levitate the rotor in the future.

Fig. 5.27 Stator of self-bearing motor.

5.4.2 Test rig control system

The control system of the self-bearing motor test rig is mainly composed of a host computer together with xPC TargetBox from MathWorks Inc. and 2 sets of 3-phase sinusoidal brushless servo amplifiers from Advanced Motion Controls Inc.

xPC TargetBox from MathWorks is used for modeling, simulation and implementation for the self-bearing motor. It allows rapid prototyping where a controller design can be validated without the need for custom target hardware to be built. It can also be used as a stand-alone control box for real applications. These key features make the xPC TargetBox the ideal solution for the application in this project.

Fig. 5.28 shows the picture of the xPC TargetBox from MathWorks.

Fig. 5.28 xPC TargetBox.

SX30A Series PWM servo amplifiers from Advanced Motion Controls are used for generating the motoring torque and levitation force of the self-motoring motor. They can change the current according to the control signals. The amplifiers are fully protected against over-voltage, under-voltage, over-current, over-heating, and short-circuits. The SX30A series amplifiers require only a single unregulated DC power supply (all logic and control voltages are generated internally). Fig. 5.29 shows the SX30A Series PWM servo amplifiers from Advance Motion Controls.

Fig. 5.29 SX30A Series PWM Servo Amplifiers.

The diagram of the control system of the self-bearing system is shown in Fig. 5.30. The rotor speed, angular, gap displacements and currents sensings are input into xPC TargetBox via 16-bit A/D converters. The control algorithm is written and embedded in the xPC TargetBox to process these input signals and to calculate the suitable control signals. These control signals are then fed to the PWM servo amplifiers via 16-bit D/A converters to produce the required currents, which are then fed into the appropriate coils to generate the desired motoring torque and levitating force.

Fig. 5.30 Diagram of control system of self-bearing motor test rig.

Chapter 6

PID control of self-bearing motor

The structure of the self-bearing motor test rig is introduced in Chapter 5. In this chapter, we introduce how to realize the control of the self-bearing motor. The motoring control part uses an open-loop control and thus there is no feedback to the controller. Hence, the speed of the motor is controlled by the input frequency of the current through the controller. The levitation of the self-bearing motor needs a control scheme to be realized. Thus, the levitation control methods of the self-bearing motor are studied in detail in this chapter and the next chapter. In this chapter, PID control method is introduced.

6.1 Levitating control model of self-bearing motor

6.1.1 Rotordynamic model

The rotor shaft can be treated as a rigid body at low rotating speed. Assuming the contact point as a pure hinge constraint and neglecting the gyroscopic effect and rotor unbalance, the dynamic function for the rotor is

$$\begin{bmatrix} J_x & 0 \\ 0 & J_y \end{bmatrix} \begin{bmatrix} \ddot{x}_m \\ \ddot{y}_m \end{bmatrix} = \begin{bmatrix} l_m & 0 \\ 0 & l_m \end{bmatrix} \begin{bmatrix} F_x \\ F_y \end{bmatrix} \quad (6-1)$$

where $J_x = J_y = J$ is the transverse moment of inertia of the rotor around the contact point, x_m is the rotor displacement in the x direction at the center of the stator, y_m is the rotor displacement in the y direction at the center of the stator, and l_m is the distance between the center of the stator and the contact point.

Therefore, the transfer function of the rotordynamic model between the rotor displacements and levitation forces is

$$G_r(s) = \frac{X(s)}{F(s)} = \begin{bmatrix} \frac{l_m^2}{Js^2} & 0 \\ 0 & \frac{l_m^2}{Js^2} \end{bmatrix}, \quad (6-2)$$

where $X = [x_m \quad y_m]^T$, $F = [F_x \quad F_y]^T$.

6.1.2 Sensor model

Due to the high bandwidth of the eddy current displacement probes, they are treated as pure gains. The equation for the sensors is

$$\begin{bmatrix} v_x \\ v_y \end{bmatrix} = \begin{bmatrix} k_{s,x} & 0 \\ 0 & k_{s,y} \end{bmatrix} \begin{bmatrix} x_m \\ y_m \end{bmatrix}, \quad (6-3)$$

where v_x is x position sensor voltage, $k_{s,x} = 10 \text{ V/mm}$ is x position sensor gain, v_y is y position sensor voltage and $k_{s,y} = 11 \text{ V/mm}$ is y position sensor gain.

The transfer function between sensor voltages and rotor displacements is

$$G_s(s) = \frac{V_s(s)}{X(s)} = \begin{bmatrix} k_{s,x} & 0 \\ 0 & k_{s,y} \end{bmatrix}, \quad (6-4)$$

where $V_s = [v_x \quad v_y]^T$.

6.1.3 Actuator model

The power amplifier is treated as a first-order delay system. The transfer function between output current and input control voltage is

$$G_a(s) = \left[\frac{k_{p,a}}{1 + k_{i,a}s} \right], \quad (6-5)$$

where $k_{p,a}$ is the amplifier gain, and $k_{i,a}$ is the delay time constant of the amplifier.

The input control voltage of the amplifier is determined by the control command voltages as follows:

$$\phi_a = \tan^{-1} \left(\frac{-V_{c,x}}{V_{c,y}} \right), \quad (6-6)$$

$$V_a = \sqrt{V_{c,x}^2 + V_{c,y}^2}, \quad (6-7)$$

where ϕ_a is the initial phase of input control voltage of the amplifier, and V_a is the magnitude of input control voltage. $V_{c,x}$ and $V_{c,y}$ are control command voltages for the x and y positions respectively.

Considering the relationship between control command voltages, input control voltage of the amplifier, levitating currents and levitating forces, the transfer function between levitating forces and control command voltages can be described as

$$G_F(s) = \frac{F(s)}{V_c(s)} = \begin{bmatrix} \frac{k_{p,a}k_F}{1 + k_{i,a}s} & 0 \\ 0 & \frac{k_{p,a}k_F}{1 + k_{i,a}s} \end{bmatrix}, \quad (6-8)$$

where $V_c = [V_{c,x}, V_{c,y}]^T$, k_F is the levitating force constant coefficient.

6.1.4 Uncertainties in levitating control model of self-bearing motor

The uncertainties in the rotordynamic model are mainly the lateral forces from contact point, the out-of-balance force acting on the rotor and the gyroscopic effect. The uncertainties in the actuator model are mainly cogging forces due to the stator slots and flux leakage, Maxwell forces when the rotor is deviated from the center of the stator, and force coupling between x and y directions. The uncertainty in the sensor model is mainly the runout error of the sensors.

All those uncertainties and other unknown uncertainties are treated as multiplicative and additive unmodeled dynamics of the nominal levitating control model of the self-bearing motor. Therefore, the levitating control model of the self-bearing motor with uncertainties can be described as in Fig. 6.1.

Fig. 6.1 Levitation control model of self-bearing motor with uncertainties.

6.1.5 PID controller

The levitating control of the self-bearing motor is realized by using the PID method. The control diagram is shown in Fig. 6.2.

Fig. 6.2 PID control diagram for self-bearing motor.

6.2 Matlab’s realization of PID control for self-bearing motor

The PID control diagram realized by Matlab Simulink is shown in Fig. 6.3. The rotor speed, angular, gap displacements and currents’ sensings are input into xPC TargetBox via 16-bit A/D converters. The PID control algorithm is written and embedded in xPC TargetBox to process these input signals and calculate the suitable control signals. These control signals are then fed to the PWM power amplifiers via 16-bit D/A converters to produce the required 3 phase currents, which are then put into the coils to generate the desired levitating force.

Fig. 6.3 PID control diagram in Matlab Simulink.

The diagram mainly comprised three subsystems: the motor current generation subsystem, the levitation current generation subsystem and the PID controller subsystem.

6.2.1 Motor current generation

The motor current generation subsystem is shown in Fig. 6.4.

Fig. 6.4 Motor current generation subsystem.

The two generated motoring currents signals are as follows:

$$I_{U_m} = A \cos(\omega t + \varphi), \tag{6-10}$$

$$I_{V_m} = A \cos(\omega t + \frac{4}{3}\pi + \varphi). \tag{6-11}$$

The magnitude A , initial phase φ and frequency ω are set by control input. Time t is provided by a digital clock. The two generated current signals are the input of the amplifier and the amplifier will calculate the third phase current automatically.

6.2.2 Levitation current generation

The levitation current generation subsystem is shown in Fig. 6.5.

Fig. 6.5 Levitation current generation subsystem.

The two generated levitation current signals are as follows:

$$I_{U_b} = C \cos(\omega t + \phi) , \tag{6-12}$$

$$I_{V_b} = C \cos(\omega t + \frac{2}{3} \pi + \phi) . \tag{6-13}$$

The magnitude C and phase difference ϕ between the levitation current and the flux density of the rotor are decided by the control input. In order to make sure the phase difference, the levitation current signal is needed to synchronize with flux density signal. Fig. 6.6 shows the method to realize the synchronization of the levitation current and the flux density.

Fig. 6.6 Synchronization of levitation current and flux density.

The flux density phase is first passed through a square wave generator to generate a square wave. The rising edge of the square wave then triggers the levitation current phase to zero. Thus, the levitation current is synchronized with the flux density.

6.2.3 PID controller

The PID controller subsystem for the levitation control of the self-bearing motor is shown in Fig. 6.7.

Fig. 6.7 PID controller subsystem.

In order to prevent the saturation of the levitation currents, the output of the PID controller is limited. The parameters K_p , K_i and K_d of the PID controller are tuned by experiments.

6.3 Experimental results of PID control

The rotating speed of the self-bearing motor is directly controlled by the exciting frequency of the motoring current. However, currently, the highest speed of the rotor attained is 1,500 rpm in the experiments. This is attributed to the back-emf voltage acting on the supply voltage, thus reducing the coil’s current substantially and limiting the motoring torque.

The levitation of the rotor is controlled by the PID controller. The tuned control parameters are shown in Table 6.1.

Table 6.1 Control parameters of PID controller

K_p	Proportional gain	0.001 A/mm
K_d	Derivative gain	0.0018 A.s/mm
K_i	Integral gain	1 A/(s.mm)
T_d	Derivative time constant	1 ms
τ	Sampling time	0.3 ms

Fig. 6.8 and Fig. 6.9 show the x and y displacements of the rotor about the stator geometric center without levitating control force at the rotating speed of 1,050 rpm. The maximum x displacement is about 0.5 mm indicating that the rotor is in contact with the noncontact support shaft.

Fig. 6.8 x displacement of rotor's center without control.

Fig. 6.9 y displacement of rotor's center without control.

Fig. 6.10 and Fig. 6.11 show x and y displacement respectively with PID control at the rotating speed of 1,050 rpm. The maximum rotor perturbations are observed to be about 0.3 mm in the x direction and 0.2 mm in the y direction respectively.

Fig. 6.10 x displacement of rotor's center with PID control.

Fig. 6.11 y displacement of rotor's center with PID control.

The x and y displacements of the rotor can be added to monitor the orbital motions of the rotor. Fig. 6.12 shows the orbital motions of the rotor with PID control. Again, it indicates that the maximum displacement of the rotor is about 0.3 mm from the geometric center of the stator.

Fig. 6.12 Rotor’s orbit track with PID control.

6.4 Conclusion

In this chapter, the levitation control model of the self-bearing motor is established and a PID method is developed to realize the levitating control of the self-bearing motor. The experimental results indicate that the self-bearing motor could be successfully levitated and controlled by manipulating the levitating coil's current using PID control method, though the observed rotor amplitude response is 0.3 mm. It could be further improved by rotor balancing, optimizing the geometry of the stator, and adopting adaptive control strategy.

Chapter 7

PID-robust model reference adaptive control of self-bearing motor

7.1 Background of PID-RMRAC control for self-bearing motor

In the control of the self-bearing motor, it is difficult to obtain the exact mathematical models of the self-bearing motors because of the influence of unbalance mass in the rotor, the sensor runout, the saturation of flux density and coil current limit, etc. Due to these problems, most self-bearing motors are controlled by the PID scheme. However, it is difficult for the PID control scheme to realize the system's optimal control, as its tuning is based on trial and error techniques without prior knowledge of the plant's mathematical model [62].

Hence, the adaptive control method is adopted to control the self-bearing motor. Adaptive control can help deliver both stability and good response. The approach changes the control algorithm coefficients in real time to compensate for variations in the environment or in the system itself. In general, the controller periodically monitors the system transfer function and then modifies the control algorithm. It does so by simultaneously learning about the process while controlling its behavior. The goal is to make the controller robust to a point where the

performance of the complete system is as insensitive as possible to modeling errors and to changes in the environment.

A main approach to adaptive feedback-control design is the model reference adaptive control (MRAC) method. In MRAC, a reference model is established to describe system performance. An adaptive controller is then designed to force the system or plant to behave like the reference model. Model output is compared to the actual output, and the difference is used to adjust feedback controller parameters.

Most work on MRAC has focused on the design of the adaptation mechanism. This mechanism must note the output error and determine how to adjust the controller coefficients. It must also remain stable under all conditions.

However, there are still a lot of problems with adaptive control. Adaptive control is sensitive to the assumptions such as known relative degree and freedom of disturbances. Instability has been observed when some of the assumptions are violated. Therefore, the robust model reference adaptive control (RMRAC) method has been proposed to overcome the drawback. The adaptive part can evaluate the model of the plant while the robust part can overcome the uncertain disturbance.

In 1992, Fu presented a robust continuous-time model reference adaptive control (RMRAC) scheme for single-input-single-output (SISO) linear plants with relative degree two, a class which encompasses a huge number of linearized mechanical systems [63]. The adaptive feature of RMRAC can evaluate the model of the plant, and its robust feature can overcome the uncertainties in the plant. The features of the RMRAC scheme are the drastic improvement of both transient behavior and tracking performance compared to the conventional adaptive schemes.

However, deficiencies still exist in the RMRAC control method when applied to the levitation control of the self-bearing motor. Theoretically, the output of RMRAC control can be of any value. However, in reality, the value is limited by the capacity of the amplifier and the maximum current of the coils. Under these conditions, especially when the system is an unstable system, the limits of output make the control unstable. The reason is that to control the unstable system directly, the input values need to change dramatically, but the actual current does not change in this manner.

However, through simulation, we observed that for stable systems, the RMRAC could realize the control of the plant with limited control input. Hence, we are proposing a hybrid PID-RMRAC control scheme here to realize the levitating control of the self-bearing motor. The control strategy is to: 1) make the self-bearing motor system stable by introducing PID control, and 2) introduce the hybrid PID-RMRAC to reduce the influence of the uncertain disturbance and achieve better control.

7.2 Principle of RMRAC control

The aim of employing RMRAC is to choose a suitable adaptive law such that for arbitrary bounded reference input $r(t)$, the overall system is stabilized despite the existence of unmodelled dynamics and bounded output disturbance, that is, the plant output y_p will try to follow the model output y_m as closely as possible.

The aim is achieved by a gain adaptive controller whose parameters are adjusted by an adaptation scheme based on some measurements of the error e_0 . The RMRAC method used in this project was initially adopted by Fu [63]. The block diagram of the RMRAC is shown in Fig. 7.1.

Fig. 7.1 Block diagram of RMRAC.

7.2.1 Plant model

The plant model is assumed to be

$$\hat{P}_u(s) = \hat{P}(s)(1 + \mu\Delta\hat{P}_1(s)) + \mu\Delta\hat{P}_2(s) = k_p \frac{\hat{n}_p(s)}{\hat{d}_p(s)}(1 + \mu\Delta\hat{P}_1(s)) + \mu\Delta\hat{P}_2(s), \quad (7-1)$$

where $\hat{P}(s)$ represents the nominal plant transfer function of order n , and $\mu\Delta\hat{P}_1(s)$ and $\mu\Delta\hat{P}_2(s)$ are the multiplicative and additive unmodeled dynamics respectively

with some $\mu \in \mathbb{R}$. $\hat{n}_p(s)$ and $\hat{d}_p(s)$ are monic coprime polynomials. There are two assumptions made on the plant model,

- 1) the sign of k_p is known and is assumed to be positive, and
- 2) the nominal plant transfer function $\hat{P}(s)$ is minimum phase, i.e. $\hat{n}_p(s)$ is a Hurwitz polynomial.

7.2.2 Reference model

The reference model is assumed to be

$$\hat{M}(s) = k_m \frac{\hat{n}_m(s)}{\hat{d}_m(s)}, \quad (7-2)$$

where $\hat{n}_m(s)$ and $\hat{d}_m(s)$ are monic but not necessarily coprime polynomials of degree $n-2$ and n respectively. There are two assumptions made on the reference model:

- 1) $\hat{M}(s)$ is assumed to be stable and with minimum phase. In addition, there exists $\hat{L}(s) = (s + a)$ for some $a > 0$ such that $\hat{M}(s)\hat{L}(s)$ is strictly positive real (SPR), and
- 2) The sign of k_m is the same as that of k_p .

7.2.3 Controller structure

The basic controller structure is shown in Fig. 7.1. The dynamical compensators blocks F_1 and F_2 are two identical single input, $n-1$ output systems described by the following transfer function:

$$\hat{F}_1(s) = \hat{F}_2(s) = \frac{1}{\hat{\Lambda}(s)} \begin{pmatrix} 1 \\ s \\ \vdots \\ s^{n-2} \end{pmatrix} \in R^{n-1}(s), \quad (7-3)$$

where the monic characteristic polynomial $\hat{\Lambda}(s)$ is chosen as

$$\hat{\Lambda}(s) = \hat{n}_m(s)\hat{L}(s). \quad (7-4)$$

The parameter vector θ is defined as:

$$\theta = [c_0, C^T, d_0, D^T]^T, \quad (7-5)$$

where the parameters $C \in R^{n-1}$ in the pre-compensator block serve to locate the closed loop plant zeros, $D \in R^{n-1}$ and $d_0 \in R$ in the post-compensator block assign the closed loop plant poles, and the parameter $c_0 \in R$ adjusts the overall gain of the closed loop plant.

7.2.4 Adaptive law

The signal vectors w and $\hat{w}$ are defined as

$$\hat{w} = \begin{pmatrix} r \\ v_p^{(1)} \\ \hat{y}_p \\ v_p^{(2)} \end{pmatrix} = \begin{pmatrix} r \\ v_p^{(1)} \\ y_p + \xi_0 \\ v_p^{(2)} \end{pmatrix} = w + \begin{pmatrix} 0 \\ 0 \\ \xi_0 \\ 0 \end{pmatrix}. \quad (7-6)$$

ξ_0 in Eq. (7-6) is a bounded output disturbance given by

$$\xi_0 = \hat{y}_p - y_p. \quad (7-7)$$

and

$$\hat{\xi} = \hat{L}^{-1}(s)(\hat{w}), \quad (7-8)$$

$$\xi =: \hat{L}^{-1}(s)(w), \quad (7-9)$$

as the filtered version of w and $\hat{w}$ respectively through $\hat{L}^{-1}(s)$.

The adaptive law is given by

$$\dot{\phi} = \dot{\theta} = -\Gamma(\hat{e}_0 \hat{\xi} + \sigma \theta), \quad (7-10)$$

where $\phi =: \theta - \theta^*$ is the parameter error vector, θ^* is the constant parameter vector, and $\Gamma(> 0)$ is the adaptation gain matrix.

The estimated output error $\hat{e}_0$ in Eq. (7-10) is defined as

$$\hat{e}_0 = \hat{y}_p - y_m. \quad (7-11)$$

σ in Eq. (7-10) is given by

$$\sigma = \begin{cases} 0 & |\theta| < \theta_0 \\ \sigma_0 \left(\frac{|\theta|}{\theta_0} - 1 \right) (1 + |\hat{\xi}|^2) & \theta_0 \leq |\theta| < 2\theta_0 \\ \sigma_0 (1 + |\hat{\xi}|^2) & |\theta| \geq 2\theta_0 \end{cases}, \quad (7-12)$$

where $|\cdot|$ denotes the Euclidian vector norm. $\sigma_0(> 0)$ and $\theta_0(\geq |\theta^*|)$ are some design parameters.

7.2.5 Control input

The control input is defined as

$$u_p = (\theta^T \hat{w} + \dot{\theta}^T \hat{L}^{-1}(s)(\hat{w})) + v_p, \quad (7-13)$$

where v_p is an additional input for overcoming the disturbance. It is chosen as

$$v_p = af(\hat{e}_0)\pi(|\hat{\xi}|) + \frac{d}{dt}(f(\hat{e}_0)\pi(|\hat{\xi}|)), \quad (7-14)$$

with

$$f(\hat{e}_0) = \begin{cases} -\text{sgn}(\hat{e}_0) & |\hat{e}_0| \geq \varepsilon \\ -\frac{\hat{e}_0}{\varepsilon} & |\hat{e}_0| < \varepsilon \end{cases}, \quad (7-15)$$

and

$$\pi(|\hat{\xi}|) = \begin{cases} \beta_1 |\hat{\xi}| + \beta_2 & |\hat{\xi}| \geq h \\ \beta_1 \left(\frac{|\hat{\xi}|^2}{2h} + \frac{h}{2} \right) + \beta_2, & |\hat{\xi}| < h \end{cases}. \quad (7-16)$$

where $\beta_1 \geq 0$, $\beta_2 > 0$, h and ε are very small positive numbers.

The robust stability of RMRAC was proved in reference [63] and some numerical simulation examples were given to illustrate the effectiveness of this method. However, when applied to a practical self-bearing motor system, RMRAC cannot realize the levitation control of the self-bearing motor, because the magnitude of control input is limited by the capacity of the power amplifier and the maximum allowable coil current.

7.3 Principle of PID-RMRAC hybrid control

In order to overcome the problems encountered in using the PID control and RMRAC, a hybrid PID-RMRAC control strategy is proposed here. The main idea is to make the PID controller and the plant form a new stable system, that is, to use the PID controller to control the plant and make the system stable, and then use the robust MRAC to control this new stable system precisely. The PID-RMRAC control model is shown in Fig 7.2. The RMRAC controller realized by Matlab Simulink is shown Fig. 7.3. The PID-RMRAC control diagram realized by Matlab Simulink is shown Fig. 7.4.

Fig. 7.2 Block diagram of PID-RMRAC.

Chapter 7 PID-robust model reference adaptive control of self-bearing motor

Fig. 7.3 RMRAC controller in Matlab Simulink.

Fig. 7.4 PID-RMRAC control diagram in Matlab Simulink.

7.4 Experimental results of PID-RMRAC control

7.4.1 PID-RMRAC control

The reference model is chosen as $\frac{1}{s^2 + 2s + 1}$. The other parameters are selected as: $a = 1$, $\theta_0 = 40$, $\sigma_0 = 3$, $\Gamma = 1$, $\varepsilon = 0.2$, $\beta_1 = 40$, $\beta_2 = 1$, $h = 0.1$. Fig. 7.5 and Fig. 7.6 show the x and y displacements of the rotor's center respectively under PID-RMRAC control at the rotating speed of 1,050 rpm. The rotor's center perturbs about 0.17 mm and 0.13 mm in the x and y directions respectively from the geometric center of the stator.

Fig. 7.5 x displacement of the rotor's center under PID-RMRAC.

Fig. 7.6 y displacement of the rotor's center under PID-RMRAC.

With the x and y displacements of the rotor, the rotor's orbit can be monitored. Fig. 7.7 shows the rotor's orbit motions under PID-RMRAC control. The maximum observed deviating displacement is about 0.2 mm from the geometric center of the stator.

Fig. 7.7 Rotor orbit motions under PID-RMRAC.

As compared to the PID control orbit's response, the vibrational amplitudes of the rotor for the PID-RMRAC control in the x and y directions are reduced by 57 % and 65 % respectively.

7.4.2 PID-RMRAC control with output disturbance

In order to test the robust ability of PID-RMRAC control, a sinusoidal disturbance signal with magnitude 1 V and frequency 35 Hz is added into both the x and y sensor signals as an output disturbance. With this disturbance, the PID control cannot control the self-bearing motor system. On the other hand, with this disturbance, the controllability of the system is noted on the x and y displacements of the rotor under PID-RMRAC as shown in Fig. 7.8 and Fig. 7.9 respectively. The rotor's center perturbs about 0.2 mm and 0.15 mm in the x and y directions respectively from the geometric center of the stator.

Fig. 7.8 x displacement of the rotor's center under PID-RMRAC with disturbance.

Fig. 7.9 y displacement of the rotor’s center under PID-RMRAC with disturbance.

The rotor center track under PID-RMRAC with disturbance is shown in Fig. 7.10. The maximum deviating displacement is about 0.21 mm from the geometric center of the stator.

Fig. 7.10 Rotor orbit motions under PID-RMRAC with disturbance.

These Figures show that the vibrational amplitudes in the x and y direction only increase by about 18 % and 15 % under this disturbance. It, therefore, supports the effectiveness of the PID-RMRAC control in providing control stability of the rotor for the self-bearing motor.

7.5 Conclusion

In this chapter, a PID-RMRAC method is proposed and developed to realize the levitating control of the Lorentz force-type self-bearing motor. This method overcomes the drawback of both the PID controller and RMRAC controller. The experimental results indicate that the hybrid control method can deliver better control force onto the rotor, reducing the rotor vibrational amplitudes by as much as 65 %, as compared to the PID controller. Furthermore, it shows better robust stability with the added disturbances. This control scheme could be a better alternative for the future control of self-bearing motors of the proposed blood pump.

Chapter 8

Conclusions and recommendations

8.1 Conclusions

In conclusion, this thesis consists of three parts. The first part gives an introduction to the background of VAD development and the objectives of the thesis. In the second part, a review is carried out covering a pool of literature on the characteristics and limitations of existing VADs. There are many existing commercially available blood pumps. However, most of the VADs do not operate well in the long term. This may be due to the rejection, infection and thrombus problems associated with VADs. In addition, the cost for the commercially available VAD is also known to be very high and economically out of reach to average-wage earners. To overcome these limitations, this thesis aims to design a compact and low-cost axial flow blood pump featuring an enclosed-impeller and a self-bearing motor, and to provide efficient control algorithms to suspend the pump rotor with long term durability, high efficiency and minimized system vibrations. The third part presents the author's original contributions to the development of the proposed blood pump, which include the concept design of the VAD, pump simulation and development of the self-bearing motor of the VAD.

Part three contains the main contributions of the thesis, of which some research findings have been documented in international journals and conferences (Section 1.4). A brief summary of the main academic contributions is listed as follows:

- 1) A novel blood pump is proposed to achieve a pressure head of 100 mmHg with a flow rate of 5 L/min. The blood pump features an enclosed-impeller and a self-bearing motor. The enclosed-impeller can provide better electromagnetic coupling between the coil windings and the permanent magnets of the pump motor, as well as larger pumping volume for the blood pump. The self-bearing motor, which combines the motor and magnetic bearings to provide both driving torque and levitation forces for the blood pump, not only avoids thrombus formation in the blood pump, but also reduces the size of the blood pump. The other advantage of the self-bearing motor is that it is easy to manufacture.
- 2) The principle of the Lorentz force type self-bearing motor proposed in the blood pump is reviewed and studied. As there are many practical considerations and constraints for the implementation of this device in the VAD, an experimental self-bearing motor test rig has been designed and fabricated with the aid of FEA to verify its control scheme.
- 3) The levitation control model of the self-bearing motor is established and a PID controller is developed to realize the levitating control of the self-bearing motor. The experimental results indicate that the self-bearing motor could be successfully levitated and controlled by manipulating the levitating coil's current using the PID control method, though the observed rotor amplitude

response is 0.3 mm. In order to further improve the control accuracy of the self-bearing motor, a PID-RMRAC method is proposed and developed to realize the levitating control of the Lorentz force type self-bearing motor. This method overcomes the drawback of both the PID controller and RMRAC controller. The experimental results indicate that the hybrid control method can deliver better control force onto the rotor, reducing the rotor vibrational amplitudes by as much as 65 %, as compared to the PID controller. Furthermore, it shows better robust stability with the added disturbances.

- 4) The flow characteristics of the blood pump are successfully simulated and analyzed by CFdesign software. The results show that the proposed blood pump could achieve 50 mmHg pressure head at a flow rate of 5 L/min and a pump rotating speed of 9,000 rpm. The highest level of shear stress located at the gap between the hollow-cylindrical casing of the impeller and the housing is about 200 Pa under the conditions of 9,000 rpm rotating speed and 5 L/min flow rate. The average exposure time of blood particles across the pump is 146 ms. Based on the blood damage criterion, subjecting the blood cells to such a level of shear stress for such a short duration of time will result in a lower risk of blood trauma. The driving torques and axial forces of the pump are also determined to facilitate the design of a magnetically suspended enclosed-impeller driven by the self-bearing motor.

In short, the series of original works may provide new ways for the development of VADs featuring long term durability, high efficiency, minimized

system vibrations and low cost. It may have potential applications in artificial heart transplantations for patients suffering from congestive heart disease.

8.2 Recommendations

In this thesis, several simplifications and assumptions have been made to conduct research conveniently. Some further explorations and improvements are recommended in the future.

- 1) **Three-dimensional theory design for the blood pump:** The simulated result shows that the blood pump only achieves 50 mmHg pressure head at a flow rate of 5 L/min and a pump rotating speed of 9,000 rpm. This is due to the fact that the simplified two-dimensional theory adopted for blood pump design could not precisely describe the actual flow in the pump. Thus the designed pump is not fully optimized to reduce the flow energy loss. A natural extension to this work would be a three-dimensional analysis for better flow visualization and higher efficiency of the blood pump.
- 2) **Decoupling control technology:** In this research, only one end of the self-bearing motor is levitated. To realize the complete levitation of the self-bearing motor, five degrees of freedom (DOFs) of the motor need to be controlled. The control of different DOFs will influence each other. Hence, a decoupling control method may be explored further in order to decouple this influence. Thus, the precise control of the enclosed-impeller blood pump could be realized.
- 3) **Design of small self-bearing motor with high speed:** The current self-bearing motor designed is mainly for validating the control scheme. It cannot be directly used for the proposed blood pump due to its low speed ($< 1,050$ rpm) and large size (60 mm in outer diameter). The low speed is mainly attributed to the back-emf voltage acting on the supply voltage, thus reducing the coil's armature current

substantially and limiting the motoring torque. It may be improved in future by reducing the turns of the motor coils, and thereby reducing coil's inductance and resistance. At the same time, the size of the motor may be further reduced by optimization of the motor mechanical design. Thus, a miniature self-bearing motor (20 mm in outer diameter, 10,000 rpm rotating speed) could be realized.

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Appendix A

Verification of the Principle of Lorentz Force Type Self-bearing Motor

The Matlab code for the verification of the principle of Lorentz force type self-bearing motor (Eqs. (5-6) and (5-9)) is as follows:

syms t sit fan

% t- time

%sit- phase of motor coils

% fan- phase of levitating coils

%%%%%%%%%%%% coil poistion

f11=-pi/8

f12=pi/8

*f13=pi/8*7*

*f14=pi/8*9*

*f21=pi/24*5*

*f22=pi/24*11*

*f23=pi/24*29*

Appendix A Veification of the Principle of Lorentz Force Type Self-bearing Motor

$$f24=\pi/24*35$$

$$f31=\pi/24*13$$

$$f32=\pi/24*19$$

$$f33=\pi/24*37$$

$$f34=\pi/24*43$$

%%%% torque calculation

$$T1=\cos(t+sit)*(\sin(t+4*f11)-\sin(t+4*f12)+\sin(t+4*f13)-\sin(t+4*f14))$$

$$T2=\cos(t+sit+4/3*\pi)*(\sin(t+4*f21)-\sin(t+4*f22)+\sin(t+4*f23)-\sin(t+4*f24))$$

$$T3=\cos(t+sit+2/3*\pi)*(\sin(t+4*f31)-\sin(t+4*f32)+\sin(t+4*f33)-\sin(t+4*f34))$$

$$T=T1+T2+T3 \text{ \% motoring torque}$$

figure(1)

ezplot(T)

d=subs(T,t,0)

figure(2)

ezplot(d)

%%%% force calculation

$$Fx1=\cos(t+fan)*(\sin(t+4*f11)*\sin(f11)-\sin(t+4*f12)*\sin(f12)-$$

$$\sin(t+4*f13)*\sin(f13)+\sin(t+4*f14)*\sin(f14))$$

$$Fx2=\cos(t+fan+2/3*\pi)*(-$$

$$\sin(t+4*f21)*\sin(f21)+\sin(t+4*f22)*\sin(f22)+\sin(t+4*f23)*\sin(f23)-$$

$$\sin(t+4*f24)*\sin(f24))$$

Appendix A Veification of the Principle of Lorentz Force Type Self-bearing Motor

$$F_{x3} = \cos(t + f_{an} + 4/3 * \pi) * (\sin(t + 4 * f_{31}) * \sin(f_{31}) - \sin(t + 4 * f_{32}) * \sin(f_{32}) - \sin(t + 4 * f_{33}) * \sin(f_{33}) + \sin(t + 4 * f_{34}) * \sin(f_{34}))$$

$$F_x = F_{x1} + F_{x2} + F_{x3} \text{ \% levitating force in x direction}$$

figure(3)

ezplot(Fx)

dx=subs(Fx,t,0)

figure(4)

ezplot(dx)

$$F_{y1} = \cos(t + f_{an}) * (\sin(t + 4 * f_{11}) * \cos(f_{11}) - \sin(t + 4 * f_{12}) * \cos(f_{12}) - \sin(t + 4 * f_{13}) * \cos(f_{13}) + \sin(t + 4 * f_{14}) * \cos(f_{14}))$$

$$F_{y2} = \cos(t + f_{an} + 2/3 * \pi) * (-\sin(t + 4 * f_{21}) * \cos(f_{21}) + \sin(t + 4 * f_{22}) * \cos(f_{22}) + \sin(t + 4 * f_{23}) * \cos(f_{23}) - \sin(t + 4 * f_{24}) * \cos(f_{24}))$$

$$F_{y3} = \cos(t + f_{an} + 4/3 * \pi) * (\sin(t + 4 * f_{31}) * \cos(f_{31}) - \sin(t + 4 * f_{32}) * \cos(f_{32}) - \sin(t + 4 * f_{33}) * \cos(f_{33}) + \sin(t + 4 * f_{34}) * \cos(f_{34}))$$

$$F_y = F_{y1} + F_{y2} + F_{y3} \text{ \% levitating force in y direction}$$

figure(5)

ezplot(Fy)

dy=subs(Fy,t,0)

figure(6)

ezplot(dy)

Appendix B

Drawings of Test Rig

The detailed assembly and part drawings of the test rig produced in Pro/E for the self-bearing motor are attached in the following:

Appendix B Drawings of Test Rig

Appendix B Drawings of Test Rig

Appendix B Drawings of Test Rig

Appendix B Drawings of Test Rig

Appendix B Drawings of Test Rig

material:Al
quantity:1

511

UNLESS OTHERWISE STATED, ALL DIMENSIONS IN MM

Title: speed sensor holder

Scale: 1:1	No: _____
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Appendix B Drawings of Test Rig

Appendix B Drawings of Test Rig