DESIGN AND DEVELOPMENT OF A MULTISTAGE SYMMETRICAL WOBBLE COMPRESSOR

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JUDUL: DESIGN AND DEVELOPMENT OF MULTISTAGE SYMMETRICAL WOBBLE PLATE COMPRESSOR

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DESIGN AND DEVELOPMENT OF MULTISTAGE SYMMETRICAL WOBBLE COMPRESSOR

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A thesis submitted in fulfilment of the

requirements for the award of the degree of

Master of Engineering

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Universiti Teknologi Malaysia

DECEMBER 2006

I declare that this thesis entitled, "***The Design and Development of Multistage Symmetrical Wobble Plate Compressor***" is the result of my own research except as cited in references. The thesis has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

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Specially Dedicated to My Beloved :

Wife (Harisaweni. ST),
Daughter (Nanila Salwa Ardiyansyah),
Parent (Syahrom) and (Rosni),
Parent-in-law (M. Nasir) and (Dra. Hernita Rais),
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(Chrisnawati) and (Heri Yanto)
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ABSTRACT

There are many types of compressor design based on variation applications from the low pressure to the high pressure compression. For the high pressure application, the horizontal opposed reciprocating compressor is the most popular. However, for the smaller flow-rate natural gas refueling appliance compressors, scotch-yoke type has just been introduced into the market. Judging from the advantages and disadvantages from these compressor types, the wobble-plate and swash-plate compressor were chosen to be the combined concept for development of the new compressor. Both compressor concepts are currently used only for low pressure application with single stage compression. For this new compressor design development, both compressor types were combined to develop into a new symmetrical multi-stage wobble-plate compressor. The new compressor design operates with the suction pressure of 3 bar and discharge pressure of 206 bar. This new compressor design inherits the advantages of the wobble-plate and the swash-plate compressor which are compact and able to operate at high operating speed. Main improvement in this new compressor design is the introduction of the symmetrical wobble-plate configuration which allows for higher compressor capacity and balanced horizontal forces. The rotor concept from the swash-plate compressor has also been adopted in this new design. The normal connecting rod with the two ended ball joints has been replaced by the connecting rod with standard end-joints at both ends. This has eased the manufacturing process as the end-joints are available on the shelves. However, this standard universal end joint has limit the tilting angle of the wobble plate to a maximum of 16º.

Against this limitation and for the compressor to operate with minimum possible operating torque and optimum pressure ratio, analysis conducted concludes that the optimum number of stages is five. Flow analysis was conducted to simulate pressure and gas velocity distributions. This has helped in the conceptual development and this design of the suction and discharge port, the value and the cylinder of each stage. Heat transfer analysis was also conducted to simulate the temperature distribution on the cylinder block. The predicted temperature is about 302ºC at the first stage. Temperature rise due to compression of the air for both prototypes was found to be insignificant. As such the inter-cooler and after-cooler provided were found unnecessary and were not used. Both prototypes operated with good stability at all speeds and noise generated was acceptably low. The 1.00 m^3/hr prototype compressor was run at 1100 rpm producing a discharge pressure of 260 bar and for flow rates of 10 m^3/hr was run at 400 rpm producing a discharge pressure of 180 bar.

ABSTRAK

Kebanyakan pemampat direkabentuk berdasarkan aplikasi bermula dari pemampat bertekanan rendah hinggalah ke pemampat bertekanan tinggi. Bagi aplikasi bertekanan tinggi, pemampat salingan berkedudukan mendatar adalah yang paling popular. Walaubagaimanapun, untuk kadaralir yang kecil pemampat jenis *scotch-yoke* lebih sesuai dan telah berada di pasaran. Setelah semua kebaikan dan keburukan bagi semua pemampat diambil kira, konsep pemampat jenis plat wobal dan plat *swash* telah digabungkan dan dipilih sebagai pemampat baru yang akan dibangunkan. Pada masa kini, kedua-dua konsep pemampat digunakan untuk aplikasi satu peringkat dan bertekanan rendah. Kedua-dua konsep pemampat ini digabungkan untuk membentuk satu konsep pemampat baru iaitu pemampat salingan plat wobal simetri berbilang peringkat. Pemampat baru ini direkabentuk untuk beroperasi dalam keadaan tekanan masukan 3 bar dan tekanan keluaran 206 bar. Pemampat baru ini lebih kecil dan boleh beroperasi dalam kelajuan tinggi. Penambahbaikan utama pemampat baru ini ialah dengan pengenalan ciri plat wobal simetri yang mana akan dapat menambahkan kapasiti pemampat dan mengimbangkan daya mendatar yang terhasil. Konsep rotor bagi pemampat jenis plat *swash* juga telah diadaptasi di dalam rekabentuk baru ini. Rod penyambung asal yang berbentuk bebola di kedua-dua hujung telah ditukar dengan dua *end-joint* piawai di kedua-dua hujung. Penggunaan komponen piawai ini akan memudahkan lagi proses pembuatan. Namun demikian komponen piawai ini mempunyai had sudut kemiringan maksimum tersendiri iaitu 16 darjah.

Bagi membolehkan pemampat beroperasi dengan daya kilas yang minimum dan nisbah tekanan yang optimum, analisis telah dijalankan dan didapati bilangan peringkat yang sesuai ialah pada 5 peringkat. Selain itu, analisa aliran juga dibuat untuk mensimulasikan tekanan dan pengagihan halaju gas. Ini telah membantu dalam membangunkan konsep yang baik terutamanya dalam merekabentuk bahagian masukan dan keluaran pada setiap blok silinder. Analisis pemindahan haba juga dijalankan untuk mensimulasi taburan suhu pada blok silinder. Suhu anggaran pada blok silinder pertama adalah setinggi 302 darjah Celsius. Bagi kedua-dua prototaip, didapati peningkatan suhu tidak disebabkan oleh tekanan. Oleh itu penggunaan penyejuk (*inter-cooler/after-cooler)*tidak diperlukan. Kedua-dua prototaip beroperasi dengan stabil dan pada kebisingan yang rendah. Prototaip pemampat bagi 1.00 m^3/jam beroperasi pada kelajuan 1100 ppm dan menghasilkan tekanan keluaran 260 bar dan bagi prototaip pemampat 10m^3/jam pula yang beroperasi pada 400 ppm telah menghasilkan tekanan keluaran setinggi 180 bar.

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NOMENCLATURES

A	-	Effective flow area
A_p	-	Area of piston
Ae		Geometric port area
A_{dn}	-	Effective flow area
b	-	Diameter
C	-	Flow pattern around the reed
c	-	Speed of Sound
c_p	-	Specific heat at constant pressure
D	-	Diameter piston
D_h	-	Hydraulic diameter
E	-	Young's modulus
F	-	Force
h	-	Convective heat transfer coefficient
h_e	-	Enthalpy
h_l	-	Valve lift height
K	-	Spring Constant
K_g	-	Gas thermal conductivity
k_s	-	Contraction coefficient
L	-	Circum pattern
l	-	Stroke of compressor

l_{cr}	-	Length connecting rod
$\dot{m}$	-	Mass flow rate
M_p	-	Mass of single piston
M	-	March number
N	-	Speed compressor/ Total number of piston
Nu	-	Nusselt number
n	-	Specific heat ratio
P_a	-	Second stage pressure
P_d	-	Discharge pressure
P_s	-	Suction pressure
Pr	-	Prandtl number
P_{dn}	-	Downstream pressure
P_{up}	-	Upstream pressure
ΔP	-	Pressure difference at valve ports
Q	-	Capacity
$\dot{Q}$	-	Instantaneous rate of heat transfer
R	-	Ideal gas constant
R_p	-	Radius of piston
R_n	-	Reaction force
R_w	-	Radius wobble plate
Re	-	Reynold number
r		Pressure ratio
s	-	Entropy
T_d	-	Discharge temperature
T_s	-	Suction temperature

T_{up}	-	Upstream temperature
T	-	Torque of compressor
t_w	-	Thickness wobble plate
t	-	Time/thickness of valve
u	-	Internal energy
V	-	Volume
v	-	Velocity
$\dot{V}$	-	Volume rate
W'	-	Work
$\dot{W}$	-	Instantaneous work done by the gas in the volume control
$\ddot{x}$	-	Acceleration

Greek

α	-	Tilting angle
μ	-	Absolute viscosity
ρ_d	-	Discharge density
ρ_s	-	Suction density
θ	-	Shaft rotation angle
λ	-	Volumetric efficiency
ς	-	Damping Coefficient
γ	-	Specific heat ratio
ϕ_n	-	Angle between the connecting rod and piston's z-axis
Ω	-	Shaft speed

LIST OF APPENDICES

CHAPTER 1

INTRODUCTION

1.1 Background

Malaysia has a huge reserve of natural gas as compared to that of oil. Most of the natural gas is exported to Japan and Korea, while the remaining substantial amount is consumed by local industries. A pipeline network has been installed by Gas Malaysia a subsidiary of national petroleum agency, PETRONAS, throughout the peninsular running through major industrial areas. This infrastructure is put in place to encourage industries to use natural gas as an alternative fuel.

To encourage automotive vehicles to use natural gas, PETRONAS has been instructed to build NGV refueling stations throughout the country. So far, 24 stations have been built in Klang Valley, 1 station in Negeri Sembilan and 4 stations in Johor.

Petronas is also embarking into developing domestic natural gas refueling facilities. The concept is that of slow refueling over a fairly long period of time. Petronas has drawn up a set of specifications where by the design is relatively small, light and produces low levels of noise and vibration. This challenge is now partly translated into a scope of the present work. A symmetrical swash wobble plate multistage reciprocating compressor is found to fulfil the specification and will be the subject of the research.

1.2 Research Scopes

The scope of this research which can be summarized as follows:

i. Review on literature, patents and existing models of wobble plate reciprocating gas compressor.
ii. Develop the new concept of a wobble plate compressor.
iii. Set the operating specification and conduct thermodynamic, heat transfer and flow analyses on wobble plate compressor.
iv. Design compressor and conduct design analysis
v. Analytical Simulation.
vi. Fabrication and testing
vii. Write report (thesis).

1.3 Objectives

The objectives of this study are as follows:

i. To develop a new concept of "Symmetrical Wobble Plate Multistage Reciprocating Compressor".
ii. To design a Symmetrical wobble plate multistage reciprocating compressor for compression natural gas from pressure 3 bar to 206 bar.
iii. To design a reciprocating compressor that is effective and efficient to the application of home Refueling.

1.4 Importance of Research

i. Malaysia has to fully utilize compressed natural gas.
ii. Universiti Teknologi Malaysia (UTM) together with Petronas Research & Scientific Services (PRSS) and Universiti Teknologi Petronas (UTP) are to

develop domestic natural gas refueling facilities. UTM is to develop the compression system.

iii. The compression system has to be small, compact, light and of low noise and vibration levels.

1.5 Research Problem

The problems of energy supply shortage, polluted and poor air quality and high energy costs have contributed to the importance of natural gas as an alternative to fossil oil based fuels. As a transportation fuel, the gas must be compressed to increase its storage capacity in order for the vehicle to travel a much longer distance but still using the standard size tank. The compressor therefore becomes important primary equipment to the natural gas (CNG) refueling station.

The present design of reciprocating compressor that is used in the NGV refueling station is relatively huge, heavy, and occupies a lot of space [22]. Alternative to this is a smaller, compact and low noise vibration levels compressors when installed in a modular arrangement which can also meet the specification of the present model large compressor. If a concept of home refueling is to be implemented a single module of this small compressor may be sufficient to meet the requirement of a slow refueling rate.

After exhaustive review of the open literature which includes journal, conference proceedings and patent it is concluded that more research should be carried out to develop a compressor which is small in size, compact in the assembly and stable in the operation. A scotch-yoke concept has already been developed but the compressors are still not available in the market probably because of the problem of stability.

Many wobble or swash plate compressors are used in the automotive sector especially for air conditioners, where the maximum operating pressure is relatively low at about 14 bar. The normal wobble plate or swash plate compressor models are designed with only one side compression mechanism which creates instability especially running at high speed. The design of the compressor is to achieve smaller size, compact and stable. Instability problem at the existing compressor can be solved by developing the same system on the opposite side. The symmetrical wobble plate piston-cylinder assembly is thought to produce a dynamically balance compression machine and further development work on the piston, piston rings and cylinder liner should be able to produce a system that can compress and discharge a natural gas up to a very high pressure of 206 bar.

However, it was expected that there would be a number of parameters needed to be investigated during the development of this new concept. These parameters are interdependent on each other that finding an optimum design will be a problematical but challenging task.

1.6 Research Design and Methodology

The work involved design and development new concept high pressure compressor, analysis and simulation, and experimental. The methodology of research showed Figure 1.1.

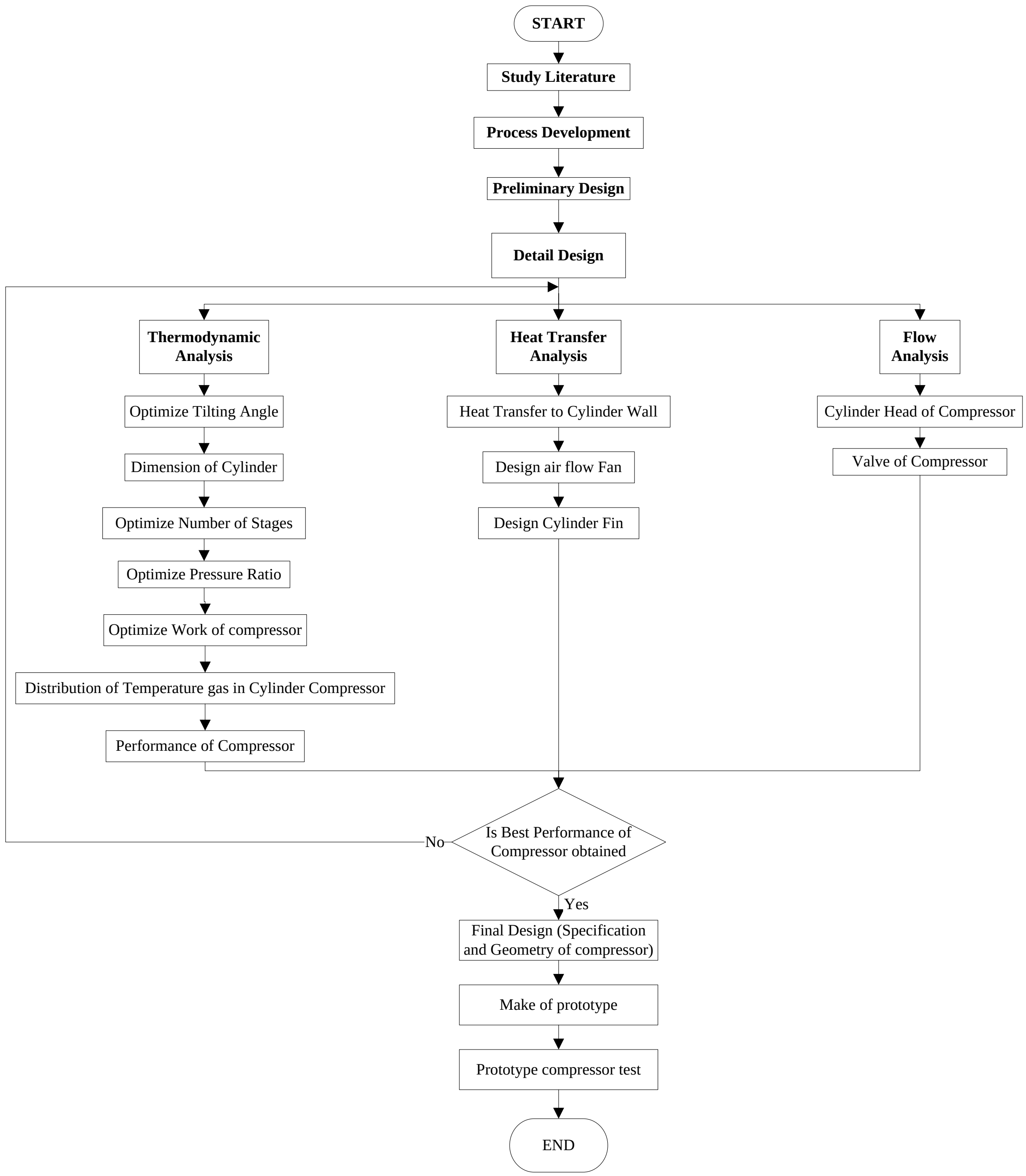

Figure 1.1 Methodology of research

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

This chapter discusses the literature review made on the development of the existing compressor design and on the report of the performance analysis of the respective prototype. Reviews on patent were also made and this had definitely triggered new ideas to improve and to enhance the development of the concept, design and fabrication process of the proposed new compressor. The review covers aspects like value design and performance, the concept of variable displacement mechanism, materials for the piston ring, heat transfer analysis and others which are thought relevant to the present work. Considering that the working principles and mechanisms are the same much review was done on refrigerant compressors.

2.2 Compressor Design

Woollatt Derek (2003) proposed a reciprocating compressor valve design: looking at optimizing valve lift and reliability. This is the primary consideration in designing a compressor that will operate 25,000 hours between scheduled maintenance shutdowns. Continuing advancement in compressor valve design, particularly valve materials, is critical to achieving this 25,000-hour operating target.

William C. Wirz (2003) presented a review of the design work that has been accomplished by Dresser-Rand Co. to develop and successfully apply modern medium-speed compressor with absorber power up to 8MW as an alternative to the slow-speed Integral Engine/compressor. The research focused on the development of the compressor on reliability, installation cost, and capacity control techniques. Properly designed and tested medium-speed compressors are a viable alternative to slow-speed compression equipment due to their operating efficiency, installation cost and ease of maintenance.

Ottitsch Franz and Scarpinato Paul (2000) carried out a work which compared the result of “state of the art” commercially available CFD software with that of wind tunnel testing for several common valve types (reed, poppet, ring and valve plate). The comparison which gives good agreement also take into account the different available turbulence models and shows how these models change the outcome of the calculations. In the present work the use of this type of tool for a valve design process is recommended and agrees that the commercially available CFD packages can predict the effective flow area of a valve.

Kazuhito Miyagawa and Hiroaki Kayugawa (1998) presented the development a new compressor for car air conditioners. The paper describes the development principles, structure, of the new type compressor. This compressor is of swash plate type with one-sided compression and having a continuously variable displacement mechanism. It has been developed from previous variable displacement compressors, and it encompasses a simple structure and attractive features, such as excellent noise and vibration characteristic, displacement controllability and reliability; while at the same time driving to a maximum speed of 9,200 rpm.

Hiroshi Toyada and Masaharu Hiraga (1990) provided a historical review of compressors in general and then of the wobble plate and scroll types specifically. The first wobble plate model was MC-508, the first 100 units that were made in the 1971 pilot production. There was a departure from the conventional concept of compressor, they are:

- Compactness was thoroughly sought

- The feature of smooth operation with low torque variation was attained with this 5-cylinder construction
- A universal mounting method was offered for the wide variety of market segments
- Durability was, of course, the first priority item, especially considering that internal space was limited to give the feature listed above.

  The top dead clearance volume was minimized by:

  - Providing a wing around the suction reed to occupy the space between the piston crown and valve plate and by controller.
  - Providing the piston extension tolerances and gaskets selection more precisely.

As a result of these changes, the volumetric efficiency, and the capacity per horsepower, was improved by 13% and 7% respectively. In the development of the 7-cylinder series in 1982, the major items introduced were:

i. The use of Tetrafloro Ethylene (TFE) piston rings to eliminate the cast cylinder liners.
ii. The reduction of the oil circulation ratio in the refrigerant circuit.
iii. The optimization of the bore to stroke ratio.

The 7-cylinder design was commonly selected in mid 1980's as a result of a compromise between capacity and smoothness. In faith since 1962, this swash plate compressor has been the major type used in the industry.

Kenji Tojo, Kunihiko Takao, Masaru Ito, Isao Hayase, and Yukito Takahashi (1990) presented an analytical model for evaluating the dynamic behavior of the variable displacement compression mechanism. The model gives detailed geometric and kinematics information regarding each element. It calculates gas torque fluctuation, constraint forces of each pair of machine elements and unbalanced force inertia. It also calculates the pressure differential between the crankcase pressure and the cylinder inlet pressure required for displacement control. This type of compressor gives the following advantages compared to a conventional fixed displacement compressor:

i. More comfortable environment
ii. Better drive ability
iii. Lower fuel consumption

iv. Improved reliability and durability.

The friction coefficient was assumed to be around 0.02. Inertial force and moment balance have direct effect on controllability of the variable displacement in the high-speed range.

Hiroshi Ishii, Yoshikazu Abe, Tatsuhisa Taguchi, Teruo Maruyana, and Takeo Kitamura (1990) examined the dynamic behavior of a variable displacement wobble plate compressor which makes it possible to control the cooling capacity continuously. The continuous cooling capacity control is achieved by the complicated motion of the piston, the piston rod, the wobble plate, the rotating journal and drive shaft. By analyzing the dynamic behavior of each moving element; design criteria were obtained for quiet operation and for durability of the compressor parts at high operating speeds.

Kenji Tojo, Kunihiko Takao, Youzou Nakamura, Kenichi Kawasima and Yukio Takahashi (1988) investigated the dynamic of the variable displacement mechanism and develops a mathematical model for evaluating stresses and bearing loads, and for optimizing inertia balance. The model gives detailed geometrical and kinematic information about the behavior of each element. The wobble plate is prevented from rotating by a guide-shoe arrangement. Actually the wobble plate/connecting rod joint traces an elliptical orbit and this creates side forces which act on the piston and wobble plate. Both the crankcase pressure control system and cylinder inlet pressure control system can regulate the compressor displacement at the required position. The crankcase control system requires a slightly large pressure differential. The increase in operating discharge pressure and the decrease in mutating angle of the wobble plate/journal require larger pressure differential for displacement control.

Keribar Rifat and Morel Thomas (1988) proposed a new methodology that they developed, which includes gas to wall heat transfer calculation based on in cylinder flow velocities and the model can be used to predict heat transfer in a compressor as a function of speed, pressure ratio, fluid properties and compressor valve and piston geometry. These are coupled with a finite element based calculation of heat conduction in the structure to provide simultaneous solution for component temperature, providing a complete performance and thermal characterization of the compressor.

Zhou Zicheng and F. Hamilton James (1986) developed a simulation model for predicting performance of multi cylinder reciprocating refrigerant compressors. The model takes into account the real gas properties; heat transfer between the gas and the cylinder wall during the working process; heat and mass transfer between the suction gas and the gas in the clearance volume; heat transfer between the gas and plenum wall; gas leakage through the clearance between piston ring and the cylinder wall; and pressure variation in the suction and discharge plenums. The use of real gas properties produced results closer to the real processes. The general cylinder method is necessary for modeling a multi cylinder compressor, which has different cylinder diameter. The gas parameters in the cylinder and the efficiencies are affected by the gas parameter in the suction and discharge plenums.

2.3 Performance of Compressor

A. Longo Giovanni and *Gasparella Andrea* (2003) developed a specific one-dimensional model of compressor valve. The mass and energy balance ware applied to the refrigerant inside the cylinder to determine the mass pressure and temperature behavior and the heat and work transfer though the compression cycle. The model was able to evaluate the refrigerant mass flow rate, the electric power input, the heat flow rates and the temperatures inside the hermetic unit, the characteristic parameters; trend during the compression cycle; the efficiencies of the compressor cycle and the hermetic unit.

Eric Winandy, Claudio Saavedrao, and Jean Lebrun (2002) proposed a detailed experimental analysis of an open-type reciprocating compressor equipped with internal sensors. The analysis reveals the main processes affecting the refrigerant mass flow rate, the compressor power and the discharge temperature. The refrigerant mass flow rate is affected by the clearance volume re-expansion, pressure drop occurring through a supply flow restriction and a temperature increase due to some heat transfer from a supposed-to-be isothermal wall. The friction power loss is transferred to this fictitious wall, which is also exchanging heat with the discharge gas and ambient.

Y.-C. Ma and O.-K. Min (2001) denote that pressure pulsation has a critical importance in the design of refrigerant compressor since it affect the performance by increasing over-compressor loss, and it acts as a noise and vibration source. Unsteady in suction and discharge pipes flow is generated by the reciprocating action of the piston, aided by the rapid opening and closing of pressure-actuated valve. A new pressure calculation method was proposed to include the gas inertia due to a decrease in the volume of cavity in the conventional helmholtz resonator model by a rolling piston movement. The comparisons with an experimental result show that the proposed MNHR is better than other conventional QS or NHR in predicting pressure over-shooting phenomena at an instant of valve opening state.

Ban Jong-ok, Lee Un-Seop, Ahn Byoung-Ha, and Kim Young-Soo (1998) presented computational fluid dynamic (CFD) analysis for the rotary compressor focusing on the valve environment including muffler and cylinder. From this analysis, it is possible to obtain flow pattern, pressure and temperature distribution from cylinder to muffler part in a rotary compressor. The CFD technique can be tried on various geometry changes to determine their deference using flow loss. Through the analysis, energy efficiency ratios (EER) increase without noise increment.

C. Arzano-Daurelle, D Clodic, and B. Hivet (1998) presented a compression model for open reciprocating compressor is elaborated. Relevant literature has been analyzed, assumption and equation related to gas flow through valves, characteristic of valves, choice of gas-wall heat transfer correlation are given. The model running deals with wall cooling during compression. Comparison between experimental and simulation result shows that usual calculation underestimate heat transfers. The model indicated that cooling of cylinder wall implies improvement of the compressor energy efficiency. This improvement is due to the increase of the mass flow rate and to the decrease of the input power.

K.Hashimoto, et all (1996) presented new valve plate assembly change utilizing a unique design arrangement. This new design change significantly improved compressor Noise, Vibration, and Harshness (NVH) characteristics due to reduction in the valve impact force as well as in-cylinder over-pressure. The over-pressure at the valve opening time was reduced.

Si-Ying Sun and Ting-Rong Ren (1995) proposed, comprehensive consideration of the various factors, such as heat transfer, leakage, gas pulsation and valve motion, that influence the working process of the compressor and establish all the mathematical simulation equations. By using numerical computation, the thermodynamic parameter which governs the working process of the compressor and the microscopic thermodynamic performance, such as capacity, power and specific power in the compressor can be found. The result of computation are in good agreement with practical measurement and the correctness and applicability of the proposed method.

M.L.Todescat, F. Fagotti, A.T. Prata, and R.T.S. Ferreira. (1992) presented the simulation model employed in the program is based on energy balances. For the refrigerant gas inside the compressor cylinder use was made of the first low of thermodynamic time variations of the mass and energy fluxes. The required temperature at the suction chamber, cylinder walls, discharge chamber, muffler, compressor shell, and ambient inside the compressor shell are obtained from steady state energy balances at various location within the compressor. A companion simulation program, which represents the compressor working features, was to calculate the mass fluxes at the suction, discharge, and the leakage flux. Simulation results are presented for a small compressor and compared with experimental result. The influence of different correlations for the heat transfer coefficient between the gas and the cylinder walls on the compressor performance. The influence of different correlations for the heat transfer coefficient between the gas and the cylinder walls is on the compressor performance. Except for the temperature of the cylinder walls, use of different correlations for the heat transfer coefficient, has little effect on the quantities. The rate of the heat transferred between the gas and the cylinder walls (including piston and valve plate), represent the least heat transfer contribution among those entering in the energy balances. Therefore, variations on heat transferred are expected to have little effect on the thermal performance compressor. That is the contributions that are more effected by changes on the temperature of the compressor shell.

Geral W. Recktenwald, James W. Ramsel and Suhas V. Patankar (1986) provide two numerical models are used to investigate the instantaneous heat transfer between the cylinder walls and gas in a reciprocating compressor. One model uses simple mass and energy balance to predict the bulk thermodynamic properties of the gas in the cylinder. Heat transfer between the cylinder walls and the gas is calculated with a widely used correlation for the heat transfer coefficient. The other models solve the unsteady continuity, momentum, and energy equation for the gas in the cylinder using a finite-difference technique. Results from the finite-difference model agree quite well with the published result from experiments and similar computations for compressors and non-firing reciprocating engines. The instantaneous heat transfer predicted by the simple model is an order of magnitude less than that predicted by the finite-difference.

2.4 Summary

Natural Gas Vehicle (NGV) technology is a familiar usage modern country such as; United State, Australia, and Asia (China, Japan, and Korea, etc). The technology and the prospect of NGV had been evaluated by many previous researches. Referring on those evaluation results, natural gas can be use as vehicle energy. However, there are still disadvantages using natural gas. The increase of weight of the vehicles and the decrease of the available space, normally people think that a CNG container with pressurized natural gas in a car is like a big bomb which is likely to explode at any time, developing CNG vehicles needs a large amount of capital investment are several of the disadvantages the used of the natural gas.

Symmetrical wobble plate compressor is kind of compressor that use to compressor this natural gas. It because of this wobble plate have so many advantages such as; the compactness, the feature so smooth, universal mounting and the durability. To design this compressor there are several things that have to consider. First, the pressure ratio and the number of the stages must be optimum. This is very important because it were needed to reach the geometry and the optimum work. Second is the heat transfer. It will affect to the compressor performance. Third is the gas velocity in cylinder. It will affect to heat transfer in cylinder wall and also will be effect to compressor valve design. Valve it self is the main component to determine the compressor performance. Fourth are the kinematical and dynamical. Those have correlate with the compressor motion, load and balancing. The simulation software can be use to design it in highly efficient, relatively in short time and in low cost if compare to conventional development process.

Based on the list of patents, given in Appendix 1, the development had been done on improving the wobble plate concept, swash plate compressor, piston, lubrication system, bearing, variable displacement and cylinder. So far, no one have done on symmetrical wobble plate concept. This new concept could be considered as our new invention in the compressor world.

CHAPTER 3

BASIC PRINCIPLE OF SYMMETRICAL WOBBLE PLATE COMPRESSOR

3.1 Introduction

The function of a compressor is to increase a gas pressure. There are several compressor design concepts such as centrifugal, rotary-vane, reciprocating, helical, and scroll. Speed, pressure ratio, discharge pressure, and mass flow rate are the most important parameters to be considered in each concept.

There are three important sections in a reciprocating compressor where designer should give more attention in order to achieve a high volumetric efficiency of the compressor. There are the suction port, compression chamber and the discharge port. In the suction port, the input parameters play a very important role in determining a smooth efficient in flow of gas and preventing back flow during compression of the gas. In the compression chamber, piston rings play a very critical role in achieving a leak free compression process. In discharge port, the output parameters must be suitable to ensure smooth out flow of the gas, whereas a complete sealing capability of the value is vital to prevent back flow of high pressure gas into the compression chamber. A designer must also look at the mechanism to oscillate the piston in such a way that the mechanical efficiency is maximum, stable at all speeds, low level of noise and vibration.

The reciprocating concept has been selected since it gives very high volumetric efficiency although it needs some further improvement. A pair of a wobble plate assembled in mirror image arrangement was chosen to be our new mechanism for the new compressor.

This chapter discusses in detail the working principle of symmetrical wobble plate compressor of a reciprocating concept.

3.2 Positive Displacement Compressors

All positive displacement compressors work on the same principle and have the same forms of losses respectively. However, the relative magnitude of the different losses will be different in each type. For example, leakage losses will be low in a lubricated reciprocating compressor with good piston rings, but may be significant in a dry screw unit, especially if speed is low and the pressure increase is high. Cooling of the gas may be almost complete in a liquid flooded screw compressor.

Each of these compressor types has a clearance volume that contains gas at the discharge pressure at the end of the discharge process. This volume may be small in some designs but significant in others. Some types, of these reciprocating compressors may have a large clearance volume, recover the work done on this gas by expanding it back to suction in the cylinder.

Some compressor types, especially those that use fixed ports for the discharge, are designed to operate at a fixed volume ratio. As the ratio varies from this value, the compressor efficiency will be less than the optimum value. Other compressor types use either ports that can be varied with slides or they use pressure-actuated valve. These types are optimized at any pressure ratio. The following discussion deals especially with the application of reciprocating compressors, but similar considerations apply to other types.

3.3 Advantages of Symmetrical Wobble Plate Compressor

Symmetrical wobble plate compressor technology offers advantages in performance for a number of reasons. Some of the advantages will be discussed in the following section. Symmetrical wobble plate compressor is developed from a conventional wobble plate compressor. The symmetry is as shown in Figure 3.1. Each opposite pair of piston-cylinder assemblies represents a difference stage of compression.

The symmetrical wobble plate compressor is very stable dynamically compared to that of the existing the single wobble plate compressor. This is due to the fait that the forces that acted on each piston pair are equal and opposite. The inertia force on the wobbling plates are also cancelled and as a result the movement of the entire assembly is supposed to give a dynamically balance operation with low levels of noise and vibration. With the result of the analysis the compressor can be operated at reasonably high speed.

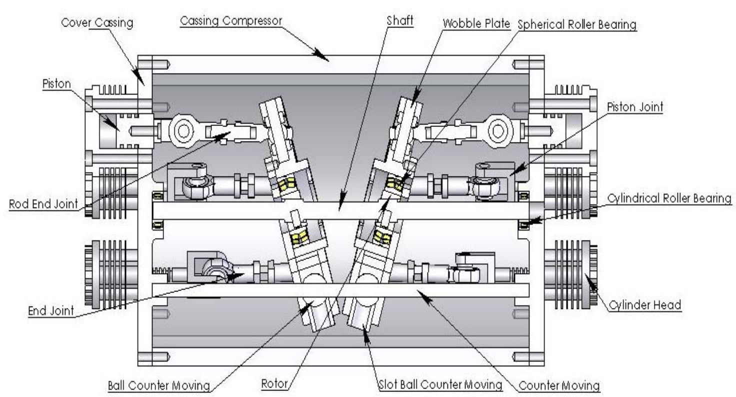

Figure 3.1 Symmetrical wobble plate compressor

Other advantages of the symmetrical wobble plate compressor model are that it is compact and most components are cylindrical in shape hence easy to manufacture. The end-joint which is one of the most critical components is available at affordable price. The system is fairly easily to dismantled due to the technical design.

3.4 General Description of Compressor

A rotor which comes the plate to wobble has same thickness as the wobble plate. A hole was drilled right through center of the rotor at a predetermined angle to the plane of the rotor. The rotor is locked to the drive shaft which planes though the hole. A standard spherical roller bearing which can take both axial and radial force is forced feed and locked around the bearing and all these finally form the tilted wobble plate-rotor-shaft assembly as shown in Figure 3.1. The tilting angle is proportional to the stroke of each piston as shown in Figure 3.2. Each piston connected to the wobble plate by a rod and end-joint.

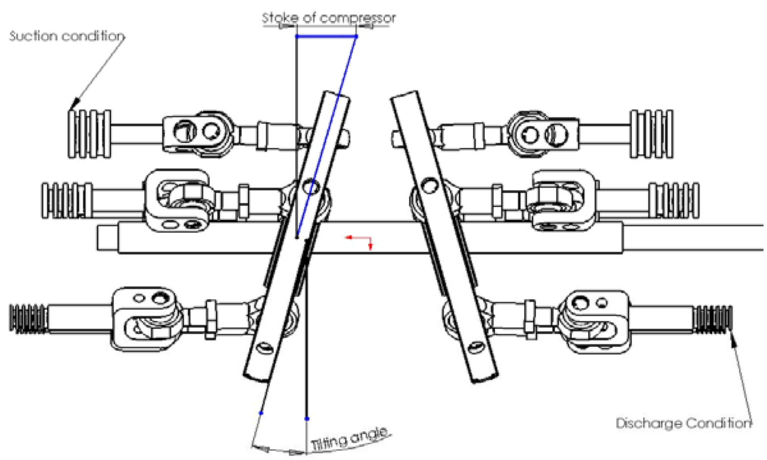

Figure 3.2 Description symmetrical wobble plate compressor

Figure 3.3 shows a section view of cylinder block that houses the liner and the valves. The liner in tight fitted to the cylinder block whereas the value mechanisms are sandwiched by the cylinder heat. Each cylinder block assembly is bolted to the compressor casing in a circular arrangement as shown in Figure 3.4. The working principle of either valve is purely by virtue of pressure difference across it, which automatically will close or open when the pressure difference is negative or positive respectively.

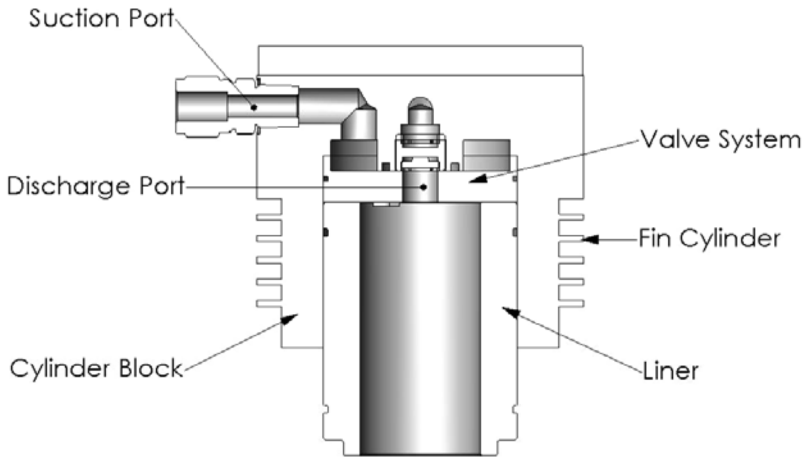

Figure 3.3 Cylinder block assembly

In the present work the discharge pressure of the gas is very high, at 206 bar, from a suction pressure of 3 bar. The compression process inevitably has to be carried out in five stages. The bore of each cylinder will be designed for each respective stage, being biggest for the first stage and smallest for the last stage. The gas is discharged to the succeeding cylinder or stage through as\ small but strong pipe. The presence of fins enhances the dissipation of heat generated by both compression and friction processes. Each stage has an intercooler and the last stage has an after cooler. However these coolers are not shown in Figure.3.4.

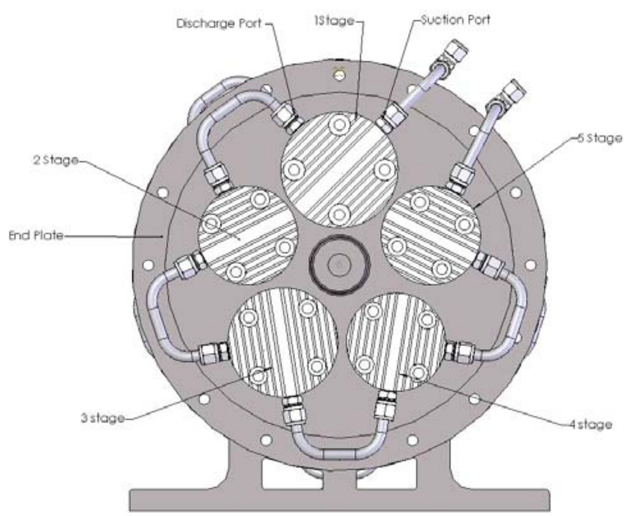

Figure 3.4 Multistage arrangement of cylinder block

3.5 Basic Principle

The complete working cycle of the reciprocating compressor can be illustrated as shown in Figure 3.5.

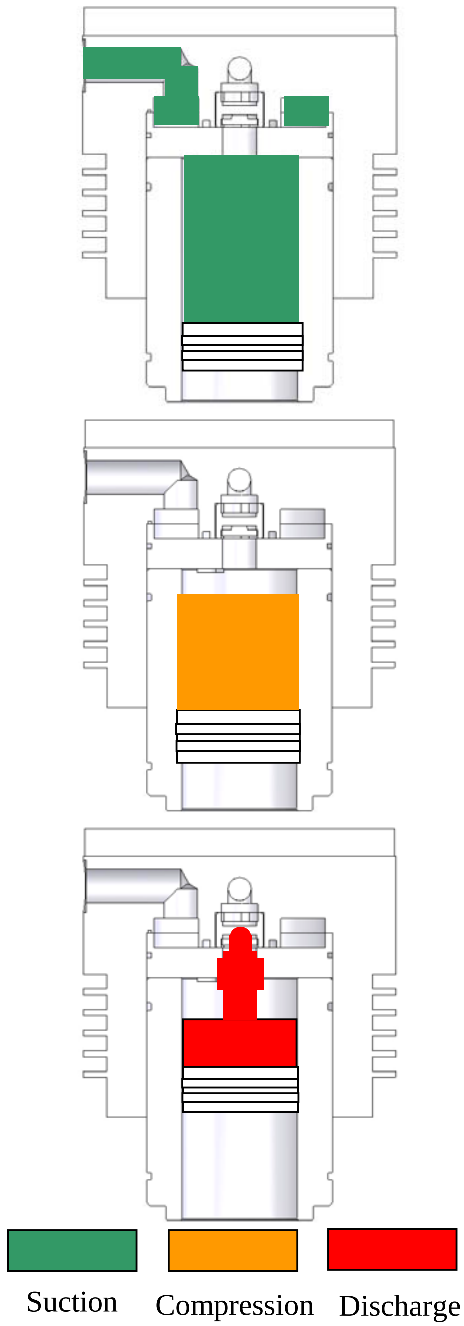

Figure 3.5 Working cycle of the symmetrical wobble plate reciprocating compressor

The suction and compression process could occurred at all consecutive cylinders simultaneously. The suction process in a cylinder stage n is related to stage (n-1) and stage (n+1). Their relationship the angle of shaft rotation could be quite complex and has to be analyzed. For example at the end of the suction stroke in stage (n+1), stage n is at the middle of suction stroke. At the same time stage (n-1) is at beginning of the suction stroke. At this time the gas flow pattern through the valve at all cylinders are so complicated. In stage (n+1), the discharge cylinder valve is remain closed until the suction process finished. Due to a different displacement with that of n^{th} cylinder a pressure difference is duly created and the magnitude is enough to cause the discharge valve of the n^{th} cylinder to open to allow the gas to flow out, even during suction mode. A similar phenomena occurs between n^{th} and $(n-1)^{th}$ cylinders. By visualizing the movement of the wobble plates in Figure 3.6, simultaneous suction modes occurred in all consecutive stages.

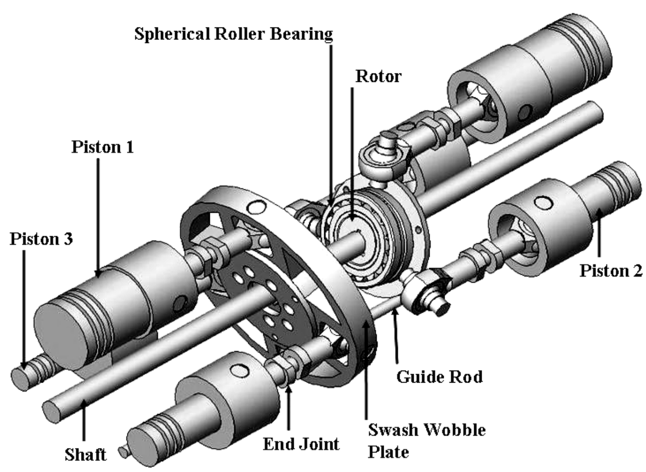

Figure 3.6 Working mechanism of the symmetrical wobble plate compressor

The suction, compression and discharge processes are normally describes on a P-V diagram as shown in Figure 3.7. The P-V diagram is drawn on the assumption that all processes are perfect.

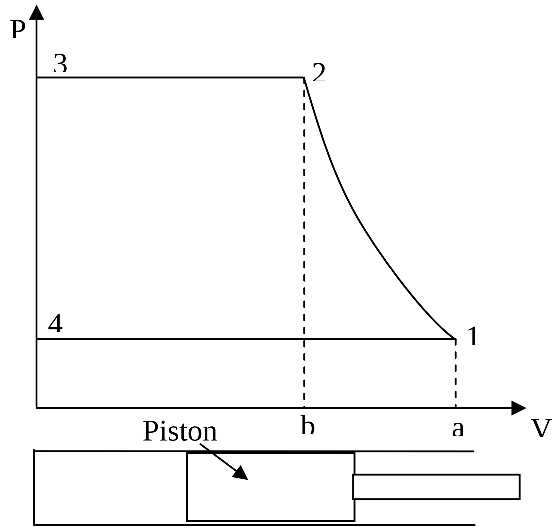

Figure 3.7 Simplified P-V diagram of ideal compressor cycle

CHAPTER 4

SYMMETRICAL WOBBLE PLATE COMPRESSOR ENGINEERING ANALYSIS

4.1 Introduction

Once the suction and discharge pressures, the suction gas temperature, the required flow rate and the gas composition are determined, a compressor can be selected to do the job. The selection will depend on the relative importance of efficiency, reliability and cost, but certain principles will always apply.

Whether the compressor performances is good or not also depends on the analysis of this geometry analysis, where it will be determined whether this compressor is under or over designed. Many things that will influence in the geometry symmetrical wobble plate compressor, such as suction and discharge pressure, capacity of compressor, tilting angle wobble plate, cylinder size, radius wobble plate, overall size of compressor, piston stroke, pressure ratio, number of stages, work of compressor, and indicated work. Number of stages and pressure ratio are the two things that are to be concerned much if a design is the kind of multistage compressor.

This chapter will analyse the affect of geometry and parameters in the development of the symmetrical wobble plate compressor.

4.2 Optimized Number of Stages

The number of stages must be selected. One consideration here is the allowed discharge temperature; another pressure ratio capability of the available cylinders as determined by their fixed clearance; end yet another is efficiency. If the calculated discharge temperature using one stage is too high, obviously more stages are needed. During preliminary sizing, the isentropic discharge temperature can be used, but if a certain number of stages create a marginal situation, the discharge temperature should be estimated more accurately. As a first estimate, it can be assumed that equal pressure ratios are used for all stages. In practice it is often good to take a higher-pressure ratio in the low-pressure ratio stages and unload the more critical higher-pressure stages a little.(Shen, 1997)

A gas cooling system has been used to cool the gas between stages. In this case, increasing the number of stages will increase the efficiency of the compressor. An alternative way to support this statement is by looking at a pressure volume diagram as shown in Figure 4.1.

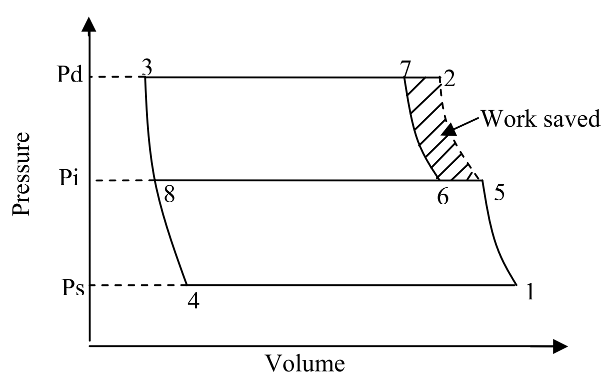

Figure 4.1 Effect of multistage

If too many stages used, the pressures losses in the valve and piping will offset the gains from inter cooling and the efficiency will reduced. The cost of the compressor to do a given task usually increases as the number of stages is increased because of the additional compressor cylinder, coolers and piping.

In this case, a symmetrical wobble plate compressor is suitable to use in compressing natural gas with an operating condition: suction pressure 3 bar, discharge pressure 206 bar, and flow capacity 10Nm^3/hr. The operation is considered equivalent to a high-pressure compressor type and therefore, the optimized number of stages has to be determined. There are several parameters involved such as load, torque, work, pressure ratio, and overall size of the compressor in order to obtain the optimum number of stages.

The number of stages has to be determined first. In this design the number of stages considered will be from 3 (three) to 7 (seven) stages. Based on these stages, the ones that are more optimized could be determined and after that the pressure ratio from each of this stage has to be calculated.

4.2.1. Pressure Ratio

If the compression ratio increased, the final compression temperature will increases as well and therefore the volumetric efficiency of the compressor will be decreased. A high compression temperature affects the operation of the delivery valve and diminishes the lubrication properties of the oil. The maximum compression ratio for small single-stage compressor is normally 8 but for large machines is only 5. In multi-stages compression, gas leaving the second stage is cool down by a second intercooler before it flows into the third stage. The process is repeated until required pressure is reached. Since the gas temperature is almost constant then the compression process is therefore approaching isothermal, thus less work required.

In Figure 4.2, the process of compression during the first stage is representing by the stages 1-2-3-4, and during the second stage, the stages 2-5-8-3. The cooling process condition is indicated by the shift from stages 2 to 6. Thus, the compression process during the second stage until into look is now represented by the new stages

6-7-8-3. Thus, using inter cooling has reduced, the work that have to do in each cycle by the area represented by stages 2-5-7-6.

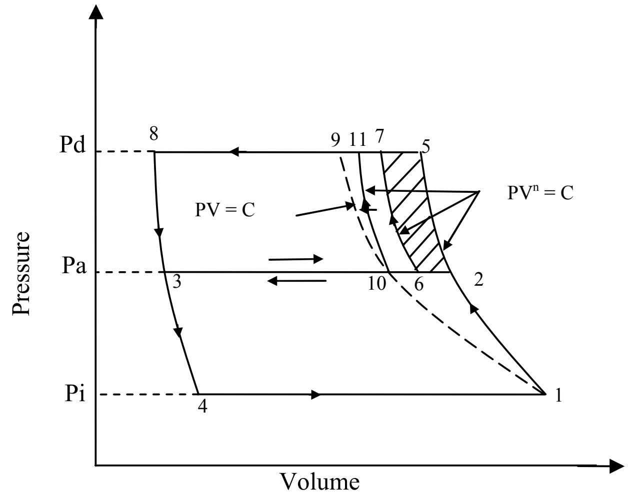

Figure 4.2 Theoretical pressure volume diagram of two stages compressor

If the gas that comes out from the first stage is cooled down until the initial temperature, it could be said that complete inter cooling has occurred. This process could be seen from 2-10 lines in Figure 4.2. In this case, 1, 10 and 9 are point at the same temperature satisfying the equation

$$PV = C$$

$$T_{10} = T_{1.}$$

In this case, the energy saved for the second cycle is represented by the area 2-3-11-10.

The total work is the sum of the low-pressure and the high-pressure work.

$$W^{'} = \dot{m}\, RT_1 \frac{n}{n-1}\left[\left(\frac{p_a}{p_i}\right)^{\frac{n-1}{n}} - 1\right] + \dot{m}\, RT_1 \frac{n}{n-1}\left[\left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}} - 1\right] \qquad 4.1$$

Or

$$W^{'} = \dot{m} RT_1 \frac{n}{n-1}\left[\left(\frac{p_a}{p_i}\right)^{\frac{n-1}{n}} + \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}} - 2\right] \quad 4.2$$

The condition for the total work to be a minimum is that the value of difference coefficient of the expression in the brackets with respect to P_a is zero. After substituting $z = \frac{n-1}{n}$, we obtain:

$$\frac{zP_a^{z-1}}{P_1^z} - \frac{zP_d^z}{P_a^{z+1}} = 0\,, \quad 4.3$$

And then from this;

$$P_a^{2z} = P_s^z P_d^z\,. \quad 4.4$$

Extracting the root leads to:

$$P_a = \sqrt{P_s P_d} \text{ or } \frac{P_a}{P_s} = \frac{P_2}{P_a}\,. \quad 4.5$$

The pressure ratio in each stages must be the same to make the total of work done smaller. For this, the equation 4.2 become:

$$W^{'} = 2\frac{n}{n-1}\dot{m} RT_1\left[\left(\frac{p_a}{p_i}\right)^{\frac{n-1}{n}} - 1\right] \quad 4.6$$

If the stages of the compressor are each subdivided further into two stages, we then obtain four-stage compression. The total work of the first and second stages will be minimum if the compression ratios are the same in both stages. This also applies to the third and fourth stages. Since the compression ratio in both stages of the original two-stage compression was equal, the compression ratios of all four stages will be equal. Let us now illustrate adiabatic four-stage compression on the

T-s diagram (Figure 4.4). Assuming perfect inters cooling, and then the final compression temperature for all stages will be equal. Hence the entropy changes in all stages will be equal i.e. $(s_0 - s_1) = (s_1 - s_2) = (s_2 - s_3) = (s_3 - s_4)$.

In general, for a compressor with n stage:

$$\frac{P_1}{P_0} = \frac{P_2}{P_1} = \ldots = \frac{P_n}{P_{n-1}} = \sqrt[n]{\frac{P_n}{P_0}} \qquad 4.7$$

$$\mathrm{P_n} = \mathrm{P_{n-1}}.\mathrm{r} \qquad 4.8$$

This equation will be used, for determining the compressor pressure in each after stage.

Figure 4.3 and 4.4 show that an increase in gas temperature during compression process i.e. temperature T_1 increase to T_2. A cooling system will bring the gas temperature T_2 down to T_2'. Thus, gas with temperature T_2' will flow to the next stage and again through compression process and increase the temperature to T_3. Then, aftercooler will bring temperature T_3 down to T_3' continuously. Generally, the temperature in delivery compressor could be derived by using a following equation;

$$T_n = T_{n-1}\left(\frac{P_n}{P_{n-1}}\right)^{\frac{n-1}{n}} \qquad 4.9$$

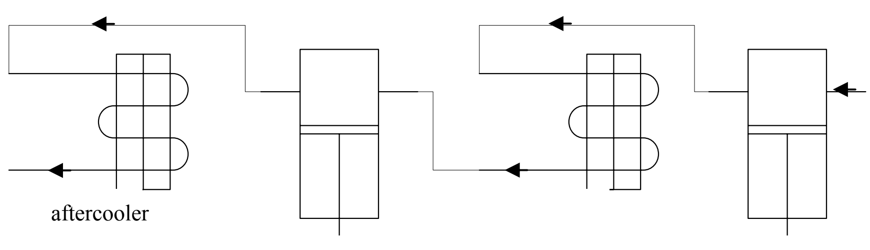

Figure 4.3 Intercooling and aftercooling between compressor stages

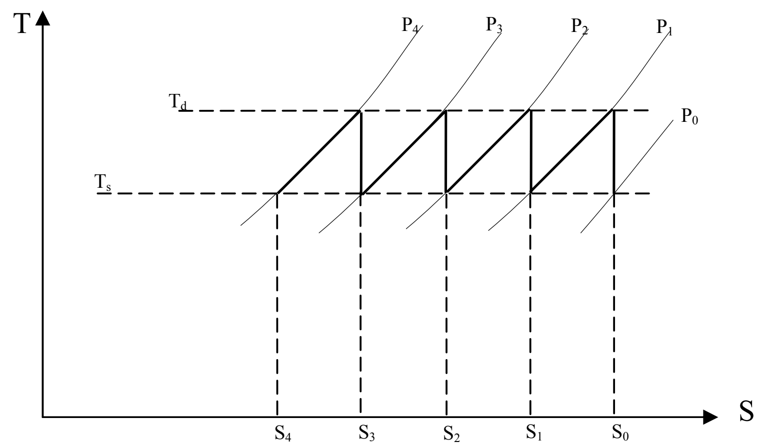

Figure 4.4 Adiabatic four-stage compression on the T-s diagram

4.2.2. Kinematics of Symmetrical Wobble Plate Compressor

The equations that govern the motion of piston, connecting rod, wobble plate and using standard coordinate transformation method derives the anti-rotation mechanism. In the concluding section, these equations are written and used to demonstrate the effects of anti-rotation mechanism and variation in stroke length. The complete reported by a co-worker from the same project.

4.2.2.1 Wobble Plate Motion

Wobble plate compressors exhibits complex motion compared to that of crankshaft reciprocating compressor and swash plate compressor. In a swash plate compressor, piston movement is sinusoidal and they only affected by the plate movement in shaft axis direction (z-axis) only. However, in a wobble plate compressor all the three directions of the wobble plate movement in the x-axis, y-axis, and z-axis will affect the movement of the piston due to the usage of ball joint at connecting rod to connect the plate to the piston.

Wobble plate kinematics are obtained from the movement of connecting rod. Connecting rod ball joint at wobble plate side will show the movement of plate while the connecting rod ball joint at piston side will show the movement of piston. The plate is constrained from rotating with rotor and shaft using the anti rotating mechanism. This constrain is needed to prevent connecting rod from being tangled and tied together.

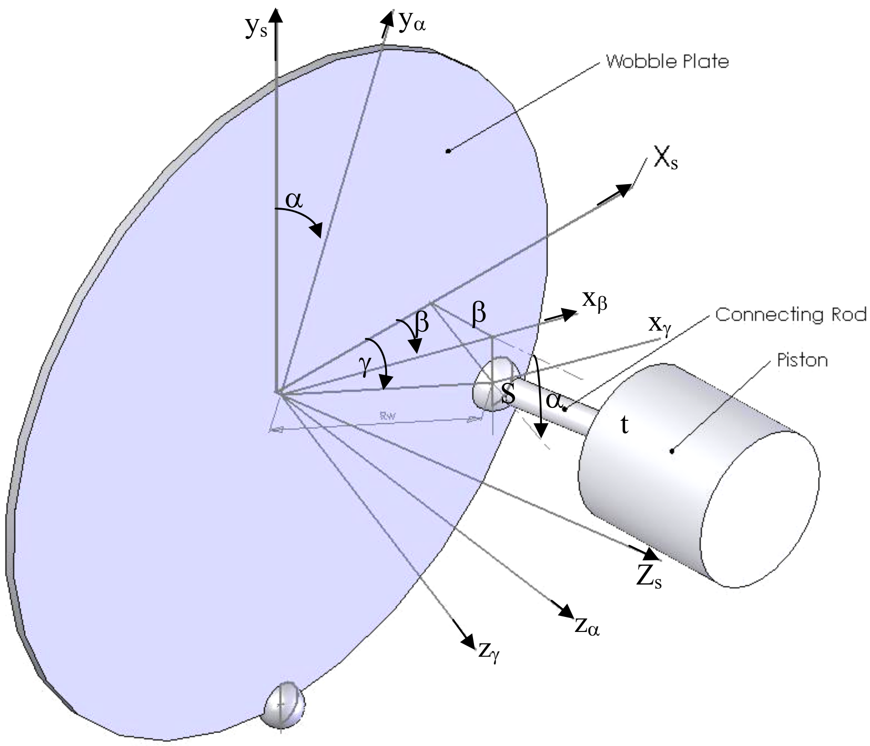

Figure 4.5 Geometric relationship that exist in wobble plate

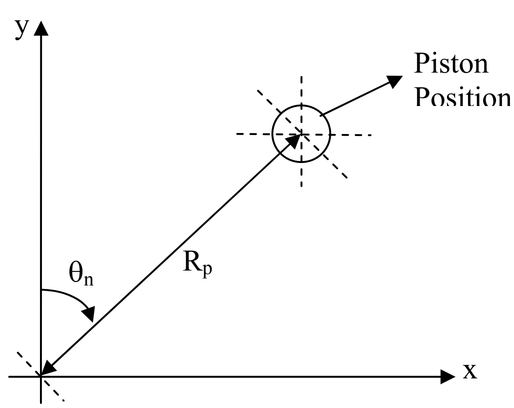

Figure 4.6 Location of connecting rod ball on piston side

From Figure 4.6 and Figure 4.7, the coordinate of connecting rod ball on piston side, t is represented by:

$$x_t = R_p \sin\theta_n$$
$$y_t = R_p \cos\theta_n$$
$$z_t = z_w + l_{cr} \cos\phi_n \qquad 4.10$$

The ϕ_n angle is the angle between the connecting rod and piston's z-axis obtained from Figure 4.7 as:

$$\phi n = \sin^{-1}\left[\frac{\left\{(x_s - x_t)^2 + (y_s - y_t)^2\right\}^{\frac{1}{2}}}{l_{cr}}\right] \qquad 4.11$$

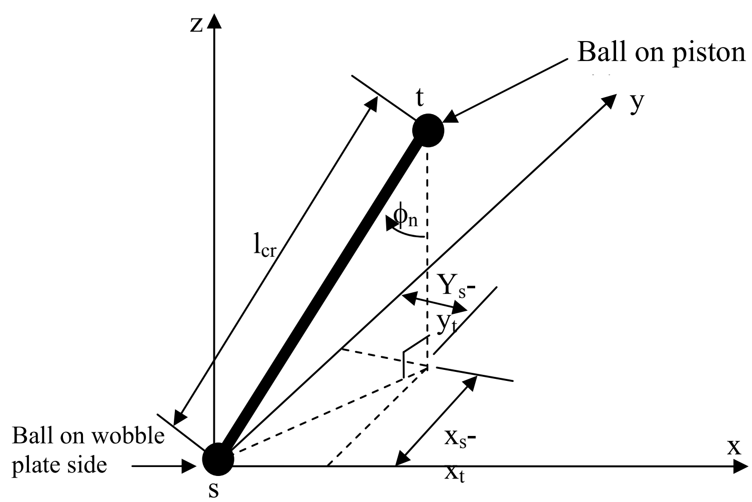

Figure 4.7 Location of connecting rod ball on piston and wobble plate side

$$z_t = ((t_w)\cos\gamma + (R_w \sin\theta_n)\sin\gamma)\cos\alpha + (R_w \cos\theta_n)\sin\alpha + l_{cr}\cos\phi_n \qquad 4.12$$

Thus, piston stroke is given by:

$$l = (z_t)_{max} - (z_t)_{min} \qquad 4.13$$

From previous sets of equation, piston stroke is a function of:

$$l = f(R_w,\ R_p,\ t_w,\ l_{cr},\ \theta_n,\ \tilde{\alpha}) \qquad 4.14$$

At all piston location, all the value of R_w, R_p, t_w, l_{cr}, θ_n, and $\tilde{\alpha}$ are the same. The only difference is in the value of piston angular location, θ_n. This difference cause each piston set to have a different value of stroke. However, the stroke value is symmetrical in the x-axis with the maximum value on θ_n equal to 45°, 135°, 225°, and 315° and minimum value on θ_n equal 0°,90°, 180°, and 270°. The stroke difference between maximum and minimum stroke location is less tan three percent for value of $\tilde{\alpha}$ les than 30° (Zair Asrar, 2006).

4.2.2.2 Determination of Cylinders Volume

There are several parameters that influence the determination of the cylinder volume such as radius of wobble plate, tilting angle, stroke of compressor, speed of compressor, and space for optimum cylinder arrangement. The dependence on these parameters has to be studied simultaneously. The independence of these parameters on each other adds up to the complexity in the analysis. For example, changing the tilting angle in order to change the stoke will change the radius of the wobble plate.

The dimension of the cylinders in multistage symmetrical wobble plate compressor is depending on several factors. One of the major factors is the space availability for the cylinders as shown in Figure 4.8.

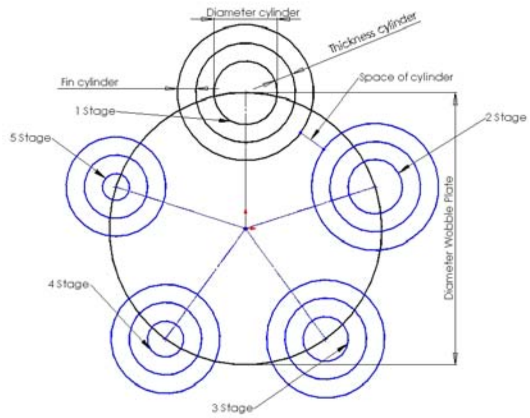

Figure 4.8 Cylinder configuration

The first step in determining the optimum number of stages is to determine the appropriate dimension and the configuration to fulfill the $10Nm^3/hr$ capacity.

Based on Figure 4.11, it could be seen that the inner diameter of the cylinder above is not enough to determine the arrangement of the cylinders. The thickness of the cylinder and the diameter of the fin would also have to be considered.

The next step in to determine the optimum of the tilting angle. This process will be repeat the same calculations as that required to determine the number of stages.

To start off the inner diameter of the first stage is determined first based on the compressor capacity that has been beside specified. It shows in the equation 4.15.

$$\dot{V} = D.l.N \quad 4.15$$

l is the stroke of the compressor and it could be calculated by using the equation 4.13.

$$V_1 = \frac{\pi}{4}(D_1)^2 \quad 4.16$$

The diameters of the piston for the higher stages are given by the equation below:

$$V_n = V_{n\text{-}1}\left(\frac{P_n}{P_{n\text{-}1}}\right)^{\frac{1}{\gamma}} \qquad 4.17$$

By obtaining the cylinder volumes using the equations 4.16 and 4.17, the diameter of the second stage cylinder may then be determined. After the cylinder dimensions are obtained the availability of space to put that cylinder can then be checked.

4.2.2.3 Force Acting on the Piston

The design of the wobble plate mechanism of the compressor could be started with the forces acting on the pistons. One of the causes the force of the piston is the pressure of the gas. The gas pressures inside the cylinder varies with the angle of rotation of the shaft, correspondence to the suction, compression and discharge process.

For the case of the multistage compressor the forces produced in each piston have variations appropriate with the pressure and the cylinder width, since the cylinders are of different geometries and also pressures. In single stage compressors, forces on the pistons are the same because the geometry and the pressure in each cylinder are the same too. Considering one wobble plate position at a time, it can be seen that the pressure in each compressor will be different because there is a different process in each cylinder i.e first cylinder in suction condition while the next cylinder (second cylinder) be in compression and discharge process. All of those things being the causes of the pressure differences for each cylinder. The force on each piston could be determined by using the following equation;

$$P = \frac{F}{A} \tag{4.18}$$

By knowing the pressure and the width of the cylinder, the force on the piston can be determined by:

$$F = P.A \tag{4.19}$$

Compressors for wide range of applications tend to run at about the same piston speed. That is compressors with a long stroke tend to run slow than those with a short stroke. Further, short stroke compressors tend to be of lighter construction with lower allowable loads. For the best efficiency and reliability at the expense of increased cost, a piston speed at the low end of the normal rang will be used. The compressor speed and the stroke will then be determined by the horsepower requirement. A low horsepower application will require a light, low stroke, high-speed compressor. A high horse power application will require a heavy, long strike, low speed compressor. If possible, large compressors are directly coupled to the drive. Thus the speed range of available drivers may influence the selection of the compressor.

4.2.2.4 Torque in Compressor

Torque in compressor can be determined after knowing the force in each piston and the distribution of force in each angle-rotating shaft. There are several the need to reasons for the torque in the compressor to be determined. This include, optimize the use of energy while the compressor is in running condition. Also, torque in required in the appropriate motor to be used or running the compressor. The optimum closing torque values will make the compressor perform better.

To determine the torque on the symmetrical wobble plate is very different from other kinds of compressors. It is described by force and torque diagram as follows:

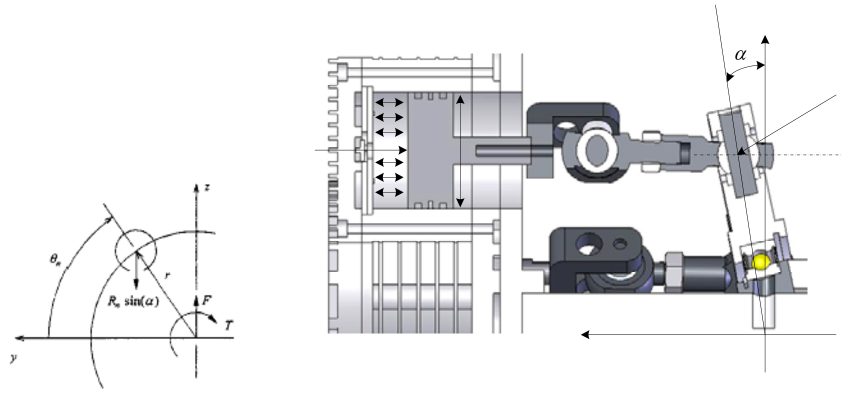

Figure 4.9 Force and torque diagram for loads exerted on the shaft

Figure 4.9 shows a sectioned view of a single piston within the cylinder block as it operates within the compressor. In this view, the reaction force between the wobble plate and the n^{th} slot is shown by the symbol, R_n. From the left hand side of Figure 4.9, it can be seen that the component of this force in the downward direction is given by (Manring, 2000): P

$$F_n = R_n \sin \alpha \qquad 4.20$$

Where α is the wobble plate angle.

Equation 4.20 represents the downward force exerted on the shaft by the nth piston. Summing these forces for all pistons within the compressor yields the total force exerted on the shaft in the downward direction. This result is given by:

$$F = \sum_{n=1}^{N} R_n \sin \alpha \qquad 4.21$$

Where N is the total number of pistons within the compressor.

The torque on the shaft is generated by the downward component of the reaction force between the *n*th piston and the wobble plate, multiplied by the distance of the n^{th} piston away from the *z*-axis. From Figure 4.10, it can be seen that the piston is located a distance away from the *z*-axis by the expression, $r \cos(\theta_n)$, where r is the piston pitch radius and θ_n is the circular position of the n^{th} piston. Multiplying this distance by the right hand side of Equation 4.20 yields the following result for the torque exerted on the shaft by the n^{th} piston:

$$T_n = R_n \sin \alpha \, r \cos\theta_n \qquad 4.22$$

Summing this torques for all pistons within the compressor yields the following result for the total torque exerted on the shaft.

$$T = \sum_{n=1}^{N} R_n \sin \alpha \, r \cos\theta_n \qquad 4.23$$

Figure 4.10 serves to graphically illustrate the total downward force given by Equation. 4.21 while the total of torque given by equation. 4.23.

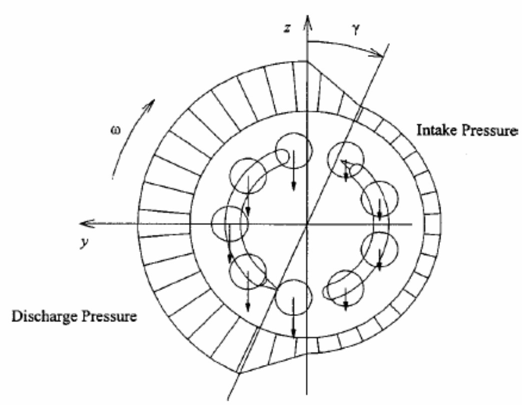

Figure 4.10 Piston pressure profile

The reaction between the n^{th} piston and the wobble plate may be determined by summing the forces which are acting on the piston in the *x*-direction and setting them equal to the piston's time rate-of-change of linear momentum. Writing this equation, and rearranging terms, yields the following result for the reaction force between the *n*th piston and the wobble plate(Manring, 2000):

$$R_n = \frac{M_p.\ddot{x}_n + P_n.A_p}{\cos\alpha} \qquad 4.24$$

where M_p is the mass of a single piston, $\ddot{x}n$ is the piston's acceleration in the *x*-direction, *Pn* is the fluid pressure within the n^{th} piston chamber, and A_p is the pressurized area of a single piston.

The general expressions for the downward force and torque exerted on the shaft are given in Equation 4.36 and 4.38. These equations describe the instantaneous loads that are exerted on the shaft, which tends to oscillate at certain dominant frequencies depending upon the number of pistons within the compressor and the rotational speed of the shaft. If the compressor is designed with a sufficiently large number of pistons, the amplitude of the oscillations can be reduced and the frequency of the oscillations can be increased.

Using Equation 4.21 and 4.23, the average quantities of force and torque may be computed using the integral-averaging technique. This technique yields the following general forms (Manring, 2000):

$$\overline{F} = \frac{N}{2\pi}\int_0^{2\pi} R_n.\sin\alpha.d\theta_n \qquad 4.25$$

$$\overline{T} = \frac{N}{2\pi}\int_0^{2\pi} R_n.\sin\alpha.r\cos\theta_n.d\theta_n \qquad 4.26$$

The results of equation 4.25, and performing the discontinuous integration of the pressure terms, yields the following results for the average force and torque exerted on the shaft:

$$\overline{F} = \frac{NA_p\left(P_d + P_i\right)\tan\left(\alpha\right)}{2} \qquad 4.27$$

$$\overline{T} = \frac{NA_p\left(P_d + P_i\right) r \tan\left(\alpha\right)}{\pi} \qquad 4.28$$

4.3 Tilting Angle of the Wobble Plate

The tilting angle of the wobble plate is very important in determining the compressor performance. By using the same methods as were used to determine the optimum number of stages, the tilting angle also can be determined. Several factors influence the choice of the tilting angle such as; the force on the piston, torque in the compressor, capacity and the overall size of the compressor. Another factor of concern is the availability the end of joint to be used.

4.4 Design of Compressor Valves

Compressor valve are devices placed in the cylinder to permit one-way flow of gas either into or out of the cylinder. There must be one or more for inlet and discharge in each compression chamber (cylinder end).

4.4.1. The Basic requirements of Compressor Valves

Basically, an automatic compressor valve requires only three components to do the job:

- Valve seat
- Sealing element
- A stop to contain the travel of the sealing element

A valve comprised of the above components installed in a modern compressor would not fulfill life and efficiency requirements. Due to the high sophistication level of today's reciprocating compressors, the demands on a compressor valve require a much more elaborate design than outlined above as follows:

i. Large passage area and good aerodynamics of flow for low throttling effect (pressure drop)
ii. Low mass of the moving parts for low impact energy
iii. Quick response to low differential pressure
iv. Small outside dimensions to allow for low clearance volume
v. Low noise level
vi. High reliability factor and long life
vii. Ease of maintenance and the service
viii. Tightness in closed position

Without a doubt, the valves in a reciprocating compressor have the greatest single effect on the operating performance of the machine from both an efficiency and mechanical standpoint.

4.4.2. Basic Functions of a Valve

A compressor valve regulates the cycle of operation in a compressor cylinder. Automatic compressor valves are pressure activated, and their movement is controlled through the compressor cycle. The valves are opened solely by the difference in pressure across the valve; no positive mechanical device is used. The only exception is where cam-drive engine type valves are used as suction valves in some of the portable units. The best illustration of compressor valve cycle is done in correlating the piston movement in relation to the pressure volume diagram.

4.4.3. Fundamentals of Compressor Valve Operation

4.4.3.1 The Essential Function

The essential functions of a compressor valve could be illustrated by aligning a schematic drawing of horizontal single-acting reciprocating compressor. It was directly above its piston speed vs stroke and its cylinder pressure volume PV – diagram as clearly illustrated in Figure 4.11. P_1 represents inlet pressure and P_2 represents delivery pressure (Hoerbiger Corporation of America, 1989).

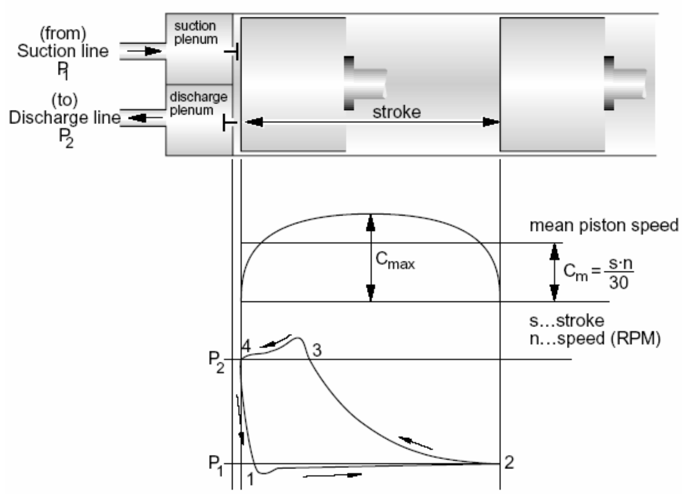

Figure 4.11 Essential functions of a compressor valve

The piston is shown at its top dead center, momentarily motionless at the end of its compression stroke (Point 4 in the PV-diagram). At this moment, the discharge valve has just closed and the suction valve has not yet opened.

4.4.3.2 Gas Intake

When the piston starts moving to the right (suction stroke), the small amount of gas remaining in the cylinder (Clearance Volume) is expanded from P_2 to P_1 and lower. The resulting slight under pressure permits the suction pressure P_1 to push the suction valve open and gas from the suction plenum is drawn into the cylinder (Point 1 in the PV-diagram). As the piston nears the end of its suction stroke, its deceleration decreases the gas speed through the open valve, and in a properly designed valve, the spring-force closes the valve at the moment the piston reaches its bottom dead center (Point 2 in the PV-diagram).

4.4.3.3 Compression

With the suction valve and the discharge valve are closed, the piston's return stroke to the left compresses the gas in the cylinder (reduces its volume while increasing its temperature and pressure) until the pressure exceeds the desired delivery pressure P_2 by the amount sufficient to open the discharge valve (Point 3 in the PV diagram). This excess pressure is necessary to overcome the equalization of static pressure on the valve plate and to lift the valve plate, against the spring force.

4.4.3.4 Gas Discharge

When the discharge valve opens, the excess pressure drops in diminishing waves to P_2. Just before the piston reaches the end of its leftward (compression) stroke (Point 4 in the PV-diagram), the discharge valve is automatically closed by its springs.

4.4.3.5 Schematic of Suction and Discharge Valves

The sequence of events is as follows:

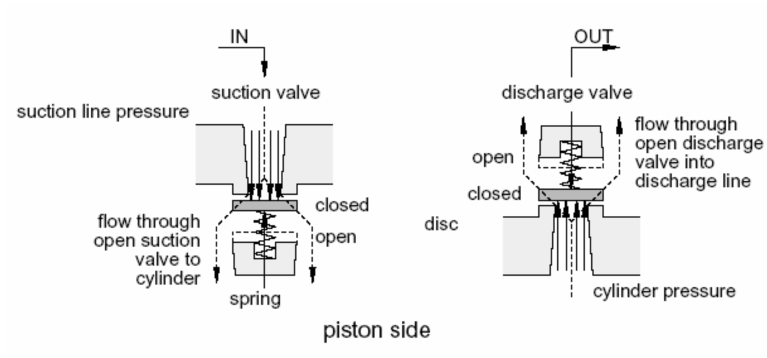

Figure 4.12 Schematic of Suction and Discharge Valve

4.4.3.6 A Pressure Differential is Necessary

On suction valves, the pressure has to be reversed. The above-mentioned factors explain why a pressure differential is necessary inside the cylinder versus outside the cylinder to lift the valve plate off the seat. The difference in area of a sealing element (valve plate or valve ring) is normally 15% to sometimes as high as 30% between exposure underneath (seat side) and exposure on top (guard side). Since there is always some leakage through the closed valve plate along the seat lands, there is a certain amount of pressure build-up in this area. Therefore, the actual pressure differential needed to break the valve open is only 5% to 15% over the line pressure and not higher, as would theoretically result from the abovementioned differential in area.

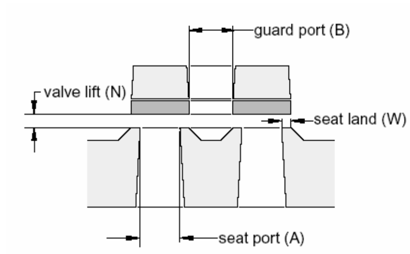

Figure 4.13 Sketch of compressor valve

As the valve plate lifts off the seat, it accelerates the valve plate rapidly against the spring-load toward the guard. The valve plate or sealing element impacts against the guard causing the so-called opening impact and, at this stage, the valve is considered fully open.

4.4.3.7 The Flow of the Gas

The flow of the gas out through the seat keeps the sealing element (valve plate) open. As the flow diminishes due to the decreasing piston speed, the springs or other cushioning elements found in most valves will force the sealing element to return to the seat and close the valve in time. Preferably, the valve is completely closed when the piston is near dead center.

4.4.4. Determine Geometry of Valve Compressor

4.4.4.1 Thermodynamic Consideration

Slow speed, water cooled air or gas compressor approach isothermal compression conditions. Using in general a polytrophic coefficient *n*, the discharge temperature and density can be estimated from:

$$T_d = T_s \left(\frac{P_d}{P_s} \right)^{\frac{n-1}{n}} \qquad 4.29$$

$$\rho_d = \rho_s \left(\frac{P_d}{P_s} \right)^{\frac{1}{n}} \qquad 4.30$$

The value of n is bracketed by n=1.0 for isothermal compression and $n=k$ for isentropic compression. If experimentally it is found that $n >> k$, it may be an indication that too much external heat finds its way in to gas.

4.4.4.2 Construction of Indicator Diagram, Valve Timing, and Velocity Estimates

After these preliminaries, it is advisable to construct an idealized pressure-volume diagram to aid in the determination of valve timing. It will be assumed here that the required basic size of the swept compressor volume has been determined and that the kinematics type of compressor has been selected, since this is not the subject of this treatise. However, it should be noted that when first sizing the compressor, a generous allowance for clearance volume should be made where its effect is of importance. Since the clearance volume will be a strong function of the valve design, a later revision in design dimension may have to be made.

In order to lay out a valve, it is necessary to determine first the average flow velocity. This is determined by the suction and discharge conditions and by the valve timing. The latter is a strong function of the kinematics design and is obtained with the help of the idealized pressure-volume diagram. For a reciprocating piston compressor, a typical diagram is show in Figure.4.14. At position 1, the piston is at bottom dead center. Both valves are closed as the piston starts to compress the gas. Discharge pressure is reached at 2 and the discharge valve opens. Assuming that the valve is ideal, that it has no flow losses, the gas is pushed out under constant pressure until the top dead center position of the piston is reached at 3. Thus, the volume of discharge gas pushed out is V_2-V_3.

To do the indicator plot, the well-known relationship is used.

$$P = P_o\left(\frac{V_o}{V}\right)^n \qquad 4.31$$

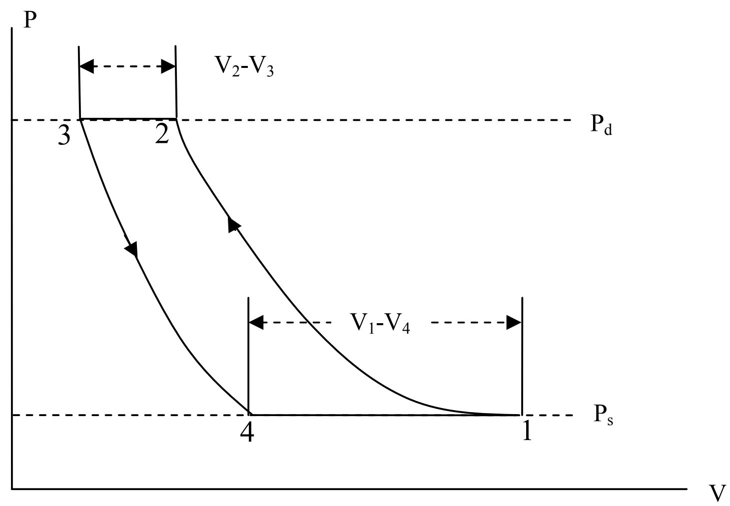

Figure 4.14 Idealized pressure-volume diagram for reciprocating compressor

From the kinematics of the drive, the establish next a relationship between volume and time, or preferably tilting angle wobble plate, since the idealized indicator diagram is independent of the shaft speed. This is shown in Figure 4.15. The shaft rotation angle during which the suction valve is open is θ_1-θ_4 and the discharge valve is open for θ_3-θ_2.

These opening angle can than be converted to opening times, assuming a constant shaft speed Ω [rad/s].

$$t_1\text{-}t_4 = \frac{1}{\Omega}\left(\theta_1\text{-}\theta_4\right) \qquad 4.32$$

$$t_3\text{-}t_2 = \frac{1}{\Omega}\left(\theta_3\text{-}\theta_2\right) \qquad 4.33$$

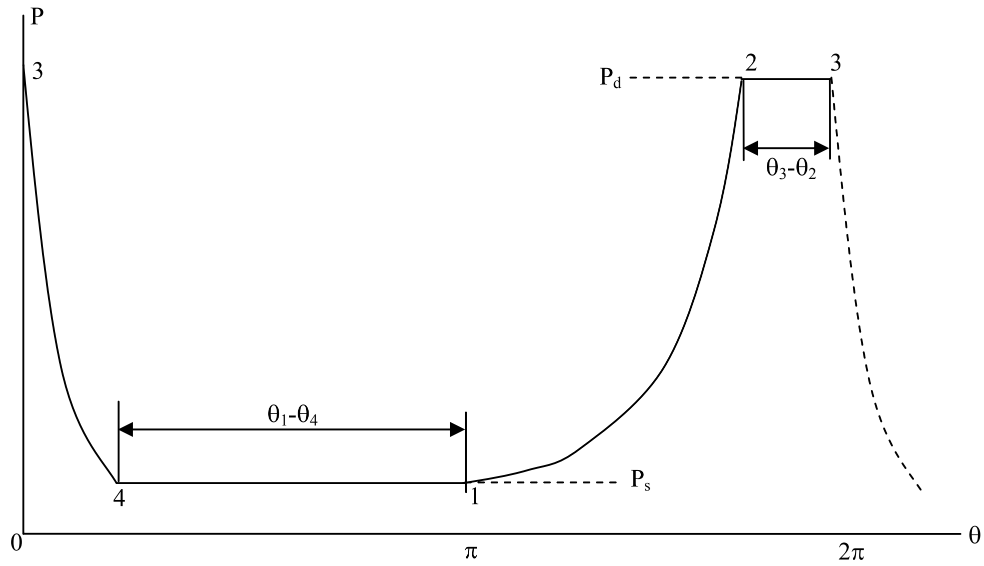

Figure 4.15 Pressure-shaft rotation angle diagram for valve opening time determination

While the diagram of Figure 4.15 is always the same, for a given kinematics design and a given suction and discharge pressure, the duration of valve opening is, obviously, inversely proportional to shaft speed. Thus, the average flow velocity of an ideal discharge valve of flow area A_d is:

$$V_d = \frac{V_2 - V_3}{(t_3 - t_2)A_d} = \frac{Q_d}{A_d} \qquad 4.34$$

Volume V_3 is the clearance volume. This volume needs to be estimated at first since it will be a function of valve design. Its presence affects the volumetric efficiency λ of the compressor. The volumetric efficiency is the ratio of the volume of gas entering through the suction valve (the mass at the suction condition is also the delivered mass at the discharge condition) to the swept volume of the piston. Because the clearance volume expands from 3 to 4, the gas entering the cylinder at suction condition is only V_1-V_4. Thus:

$$\lambda = \frac{V_1 - V_4}{V_1 - V_3} \qquad 4.35$$

which can be derived to:

$$\lambda = \frac{V_3}{V_1\text{-}V_3}\left[\left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} - 1\right] \qquad 4.36$$

There are other effects that influence volumetric efficiency, for instance pressure drops in the suction valve that delay closing, caused by pressure surges. The amount of mass delivered will be reduced if the suction gas is heated when passing through the suction manifold. However, at this point, it is best to ignore all effects except for the clearance volume expansion, and obtain an average suction velocity of

$$v_s = \frac{V_1\text{-}V_4}{(t_1\text{-}t_4)A_s} = \frac{Q_s}{A_s} \qquad 4.37$$

Again, all times and volumes are given by the proper kinematics relationships, which are obviously dependent on the type of compressor.

4.4.4.3 Sizing of Port Area

The pressure drops and flow losses in a valve according to:

$$\Delta p = \varsigma \frac{\rho v^2}{2} = \varsigma \frac{\rho M^2}{2} c^2 = \varsigma \frac{\rho}{2} kRTM^2 \qquad 4.38$$

Now, Mach number is an important parameter. It is recommended that $M \leq 0.2$ in order to avoid valve failure. Some authors distinguish between slow and fast, small and large compressor (Soedel, 1984). The allowable flow velocity is therefore:

$$v = Mc \qquad 4.39$$

Where:

$$c = \sqrt{kRT} \qquad 4.40$$

The required effective flow area is obtained by:

$$A = \frac{Q}{v} = \frac{Qc}{M} \qquad 4.41$$

Note that v is the allowable flow velocity, averaged over the opening time of the valve. The first order of business is therefore to design a valve port arrangement that gives this effective flow area. Introducing a contraction coefficient k_s, which may be taken, because information will in general be lacking at this point, as $k_s = 0.6$, the required geometric port area is:

$$\mathrm{Ae} = \frac{\mathrm{A}}{\mathrm{k_s}} \qquad 4.42$$

The same argument applies to suction and discharge. Since the volume of compressor increases with the cube of a typical dimension while the surface area available for valve ports increases only with the square, it become more and more difficult to find enough space for the valve as the size of a compressor increases, given that the compressor speed is held constant. Obviously, the flow area requirement increases proportionally to compressor speed also. Thus, large and fast compressors are the most difficult to design valves for. The smaller the compressor, the easier valve design becomes as far as space constraints are concerned.

4.4.4.4 Determination of Desirable Valve Lift

Once the port area is established, it can be argued that the lift height is established by dividing the port area by the effectively available circumference of the covering valve plate or reed. The term effectively available has been introduced since it is important at this point to sketch the reed design and visualize a flow pattern around the reed.

$$C = \pi(D) \tag{4.43}$$

Thus, the average required lift height is

$$h = \frac{A_e}{C} = \frac{A_e}{\pi(D)} \tag{4.44}$$

For a flexible ring valve, the gap area is of course not simply the circumference times the lift height. Rather, *h* has to be interpreted as an average value.

4.4.4.5 Expected Flow Force on the Valve and Selection of the Effective Stiffness

The effective stiffness, provided by springs, in the case of spring loaded plate valve or by the flexural resistance in the case of a flexing reed valve, is at this stage determined by the maximum required lift height h (Soedel, 1984).This height must be reachable by the action of the flow force on the valve. While the nature of the flow forces are fairly complicated when viewed over the entire valve opening cycle, it can be argued that as a rough approximation we can estimate them using the momentum-impulse law, ignoring Bernoulli effects due to wide valve seats, stream line detachment and reattachment, etc. Thus, the available average force to reach opening height h is

$$F = \rho A v^2 \tag{4.45}$$

Where A is the effective port area and since the admissible velocity has been already given as

$$v = Mc \tag{4.46}$$

To obtain;

$$F = kApM^2$$

For the case of a spring loaded plate valve, the required total spring rate K can now be determined from:

$$K = \frac{F}{h} \qquad 4.47$$

In case of flexible reed, the designer needs to decide, at this point, the general design of his reed. Because of the flexure, the force given by this simple approach represents now an approximate value only, in terms of a resultant. So, to determine the thickness of the valve the following equation can be used:

$$t = 2L\sqrt{\frac{K}{Eb}} \qquad 4.48$$

Assuming that we have also selected the type of material, the two variables we may play with are width b and length L. However since stress may not exceed a certain level, we introduce the condition that the maximum stress may not exceed a certain value, or use any other failure theory. Assuming that b is determined by the porthole size, the stress condition will give us the length L (Soedel, 1984)

.

4.5 Result and Discussion

Natural gas is use to simulate the design calculation taking γ as 1.27. The process of choosing the number of stages, and the operations involved in each stage have been described in sub-chapter 4.2. For this optimization the numbers of stages considered were from three to seven, with intercooling between the stages and aftercooling after the last stage. From here the best number of stages was determined for this compressor design.

To begin with, for three stages, the pressure ratio is:

$$\text{Pressure ratio, R} = \sqrt[3]{\frac{P_d}{P_s}} = \sqrt[3]{\frac{206.8\ \text{bar}}{3.4\ \text{bar}}} = 3.91$$

With this pressure ratio we could determine the pressure in each stage. As with the pressure ratio, the pressure in each stage can be using the equation 4.7. For example, for this case of three stages:

$P_2 = P_1.r$

$P_2 = 3.4\ \text{bar} \times\ 3.91 = 13.5\ \text{bar}$

$P_3 = 13.5\ \text{bar} \times 3.91 = 52.8\ \text{bar}$

$P_4 = 52.8\ \text{bar} \times 3.91 = 206.8\ \text{bar}$

The pressure ratio and the pressure in each stage for the various cases of numbers of stages chosen are given in Table 4.1:

Table 4.1 Pressure ratio and pressure each stages

No	**Number of Stages**	**Pressure Ratio**	**P_1 (bar)**	**P_2 (bar)**	**P_3 (bar)**	**P_4 (bar)**	**P_5 (bar)**	**P_6 (bar)**	**P_7 (bar)**	**P_8 (bar)**
1	3	3.91	3.4	13.5	52.8	206.8	-	-	-	-
2	4	2.78	3.4	9.6	26.7	74.3	206.8	-	-	-
3	5	2.27	3.4	7.8	17.7	40.2	91.2	206.8	-	-
4	6	1.98	3.4	6.8	13.5	26.7	52.8	104.5	206.8	-
5	7	1.79	3.4	6.2	11.1	19.9	35.8	64.2	115.2	206.8

The temperature in each compression process can be determined using the equation 4.9. As a sample, take the case of the 3 stage of compressor:

$$T_2 = 303K\left(\frac{13.5\ Psi}{3.4\ Psi}\right)^{\frac{1.27-1}{1.27}} = 404.9K = 131.9^{o}C$$

Table 4.2 Suction and discharge Temperature for each stages

No	Number of stages	T_1 (°C)	T_2 (°C)	T2' (°C)	T3 (°C)	T3' (°C)	T4 (°C)	T4' (°C)	T5 (°C)	T5' (°C)	T6 (°C)	T6' (°C)	T7 (°C)	T7' (°C)
1	3	30	131.9	30	131.9	30	-	-	-	-	-	-	-	-
2	4	30	103.7	30	103.7	30	103.7	30	-	-	-	-	-	-
3	5	30	87.6	30	87.6	30	87.6	30	87.6	30	-	-	-	-
4	6	30	77.3	30	77.3	30	77.3	30	77.3	30	77.3	30	-	-
5	7	30	70.1	30	70.1	30	70.1	30	70.1	30	70.1	30	70.1	30

Note: if perfect inter-cooling and ufter-cooling

Table 4.3 Design input parameter for symmetrical wobble plate compressor

P_{in}	50	Psi
P_{out}	3000	Psi
T_{in}	303	^{0}C
Capacity	10	Nm^3/hr
Speed	1500	Rpm
Tilting Angle	5	^{0}

The temperatures at the suction and discharge condition are shown in Table 4.2.

Based on temperature list that had been described in Table 4.2, the compressor design with seven stages produced the lowest temperature discharge than with other number of stages. On the other hand, the design with three stages produced the highest discharge temperature than others. However, the decision on the best number of stages cannot be made yet as there are other concerns aside from temperature to be considered.

Table 4.3 is design input parameter for symmetrical wobble plate compressor that had been used for compress the natural gas in mini station. These specifications give by user Petronas Research Sciencetific and Service (PRSS). For the tilting angle 5º, it is the initial values for design and not the optimum. Table 4.4 shows the geometry that comes from the calculation in each stage. In determining this geometry, the overall size of the compressor has to be concern and it must be the optimal dimension in each stage.

Table 4.4 Geometry of symmetrical wobble plate compressor

	Radius Wobble Plate (mm)	Diameter^{1} (mm)	Stroke (mm)	Volume^{1} (mm^3)	Capacity (m^3/Hr)	mass flow rate (Kg/min)	Volume^{2} (mm^3)	Diameter^{2} (mm)	Volume^{3} (mm^3)	Diameter^{3} (mm)
3 Stage	75	75	13.1	57,756.3	10.4	0.68	19,719.4	43.8	6,732.7	25.6
4 Stage	88	70	15.3	59,032.9	10.6	0.69	26,367.3	46.8	11,777	31.3
5 Stage	105	63	18.3	57,054.	10.3	9.24	57,054.	63.0	57,054	63
6 Stage	120	60	20.9	59,142.5	10.6	0.69	34,557.8	45.9	20,192.6	35.1
7 Stage	138	55	24.1	57,150.5	10.3	0.67	36,058.1	43.7	22,750.3	34.7

Continued Table 4.4

Volume^{4} (mm^3)	Diameter^{4} (mm)	Volume^{5} (mm^3)	Diameter^{5} (mm)	Volume^{6} (mm^3)	Diameter^{6} (mm)	Volume^{7} (mm3)	Diameter^{7} (mm)
-	-	-	-	-	-	-	-
5,260.2	20.9	-	-	-	-	-	-
57,054	63	57,054	63	-	-	-	-
11,798.9	26.8	6,894.2	20. 5	4,028.4	15.7	-	-
14,353.9	27.6	9,056.3	21.9	5,713.9	17.41	3,605.1	13.8

4.5.1. Optimum design symmetrical wobble plate compressor

This part will discuss about the optimization of the symmetrical wobble plate compressor design. Like stated before the number of stages that we want to choose are only from 3 until 7 because more or less than that could cause the results of the compressor to be designed be unsatisfactory. It could be proven from the pressure ratio. It becomes higher if it only chooses 1 or 2 stages. If the pressure ratio to higher it will affect to the compressor performance. It will make the compressor become inefficient. Otherwise, if the number of stages is more than 7, the advantages of 7 stage make the efficiency of compressor more better while the disadvantages values more worse if compare with the advantages. Those disadvantages are as if dimension of compressor is made bigger, then the production cost becomes higher and a compact compressor can't be achieved.

Table 4.5 Specification symmetrical wobble plate compressor for 3 to 7 stage

Compressor Data						**Unit**
Number of stage	3	4	5	6	7	
Angle piston difference	120	90	72	60	51.428	Deg
$P_{Suction}$	3.4	3.4	3.4	3.4	3.4	BAR
$P_{Discharge}$	206.8	206.8	206.8	206.8	206.8	BAR
Isentropic coefficient	1.27	1.27	1.27	1.27	1.27	
Tilting angle	5	5	5	5	5	Deg
Plate radius	0.088	0.088	0.087	0.12	0.138	m
Stroke length	0.048	0.015	0.047	0.021	0.024	m
Compression ratio	2.268	2.783	2.268	1.979	1.795	
Speed of Compressor	1500	1500	1500	1500	1500	rpm
Diameter piston 1	0.075	0.07	0.039	0.06	0.055	m
Conrod length	0.125	0.125	0.125	0.125	0.125	m

Table 4.5 is the specification of symmetrical wobble plate compressor from the calculation that had been done before. Data of table 4.5 is used as a reference for analysis to get the angle position of piston, distribution pressure gas (suction and discharge) inside the cylinder, load of compressor in force mode that produce from gas pressure, stroke of compressor, and torque of compressor.

In Table 4.6 to Table 4.10 it could be seen that the maximum force in piston are last stage because due to the pressure in last cylinder was the higher pressure, even though the diameter is the smallest. So, the increasing of load in each cylinder is because of the increasing of the pressure and the reducing of the size of cylinder diameter is not straight matchless with the increasing of gas pressure. Load that acting upon the compressor is the pressure-balanced area from piston.

Appendix A to Appendix E show the analysis for compressor with 3 stage to 7 stage. Appendix A to Appendix E are for determining the changes of the tilting angle, variation stroke of compressor, pressure distribution in cylinder, force distribution on the piston, and variation torque of compressor in 1 (one) rotation shaft. The changes of angle shaft rotation will affect to the changes of tilting angle wobble plate compressor. It also will affect to the changes of stroke of compressor and pressure in cylinder will be increased due to the compression process. Load will also be increased because of the affect of increasing pressure and torque of the compressor be increasing too.

Figures 4.16 and 4.22 illustrate the change of shaft rotation angle and stroke of a compressor. Each stage has difference stroke where for the third-stage stroke of compressor is 13.1 mm, 15.3 mm for 4 stage, 15.2 mm for 5 stage, 20.09 mm for 6 stage and 7 stage is 24.1 mm. Maximum and minimum condition of the every piston difference base on the position of the piston. This Position is depending on the angle of rotation and therefore the maximum and minimum positions are different for each piston. For the example, piston 1 the maximum stroke is in the shaft rotation angle 180º, piston 2 the maximum stroke is in the angle shaft rotation 300º and piston 3 or the last piston, the maximum condition is in the angle shaft rotation 60º that for the 3 stage compressor.

Table 4.6 Data for analysis of symmetrical wobble plate compressor (3 Stage)

Piston	Angle Position of piston (deg)	Suction pressure (bars)	Discharge pressure (Bar)	Area (m2)	Diameter (m)	Max force (N)
1	0	3.4	13.5	0.004	0.075	5962.4
2	120	13.5	52.8	0.002	0.044	7969. 5
3	240	52.8	206.8	0.0005	0.026	10652.2

Table 4.7 Data for analysis of symmetrical wobble plate compressor (4 stage)

Piston	Angle Position of piston (deg)	Suction pressure (bars)	Discharge pressure (Bar)	Area (m2)	Diameter (m)	Max force (N)
1	0	3.4	9.6	0.004	0.07	3692.4
2	90	9.6	26.7	0.002	0.05	4590.1
3	180	26.7	74.3	0.0008	0.03	5705.9
4	270	74.3	206.8	0.0003	0.02	7093.1

Table 4.8 Data for analysis of symmetrical wobble plate compressor (5 stage)

Piston	Theta piston (deg)	Suction pressure (bars)	Discharge pressure (bars)	Area (m2)	Diameter (m)	Max force (N)
1	0	3.4	7.8	0.0012	0.04	933.9
2	72	7.8	17.7	0.0006	0.03	1111.6
3	144	17.7	40.2	0.0003	0.02	1322.9
4	216	40.2	91.2	0.0002	0.014	1574.5
5	288	91.2	206.8	9.0599E-05	0.01	1873.9

Table 4.9 Data for analysis of symmetrical wobble plate compressor (6 stage)

Piston	Theta piston (deg)	Suction pressure (bars)	Discharge pressure (bars)	Area (m2)	Diameter (m)	Max force (N)
1	0	3.4	6.8	0.0028	0.06	1928.6
2	60	6.8	13.5	0.0017	0.05	2229.7
3	120	13.5	26.7	0.0009	0.04	2577.8
4	180	26.7	52.8	0.0006	0.03	2980.3
5	240	52.8	104.5	0.0003	0.02	3445.6
6	300	104.5	206.8	0.0002	0.01	3983.5

Table 4.10 Data for analysis of symmetrical wobble plate compressor (7 stage)

Piston	Theta piston (deg)	Suction pressure (bars)	Discharge pressure (bars)	Area (m2)	Diameter (m)	Max force (N)
1	0	3.4	6.2	0.0024	0.06	1470.1
2	51.4	6.2	11.1	0.0015	0.04	1664.7
3	102.9	11.1	19.9	0.0009	0.03	1885.1
4	154.3	19.9	35.8	0.0006	0.03	2134.7
5	205.7	35.8	64.2	0.0004	0.02	2417.4
6	257.1	64.2	115.2	0.0002	0.01	2737.5
7	308.6	115.2	206.8	0.0001	0.01	3099.9

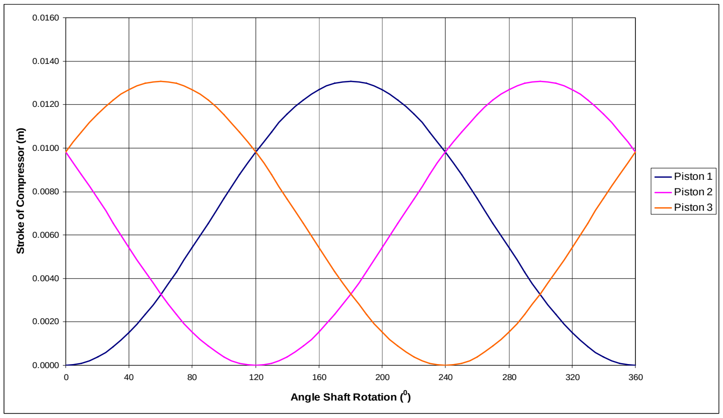

Figure 4.16 Angle shaft rotation vs stroke of compressor for 3 stage

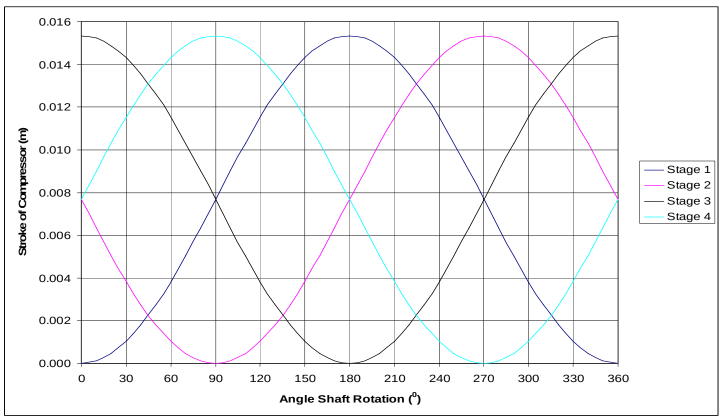

Figure 4.17 Angle shaft rotation vs stroke of compressor for 4 Stage

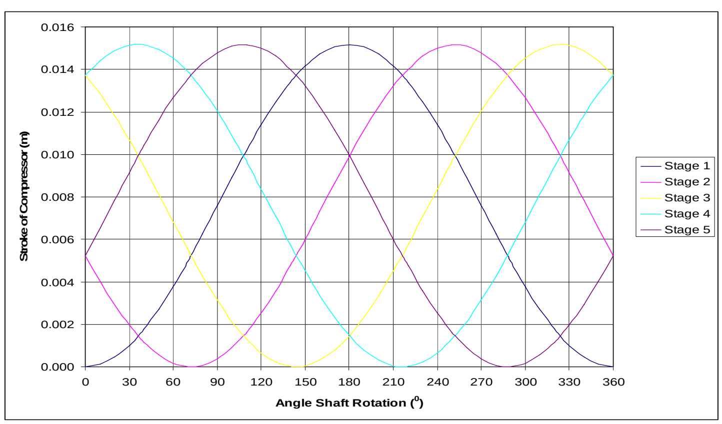

Figure 4.18 Angle shaft rotation vs stroke of compressor for 5 Stage

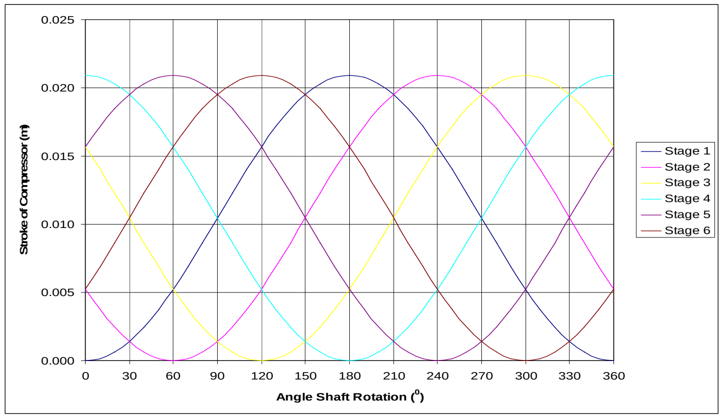

Figure 4.19 Angle shaft rotation vs stroke of compressor for 6 Stage

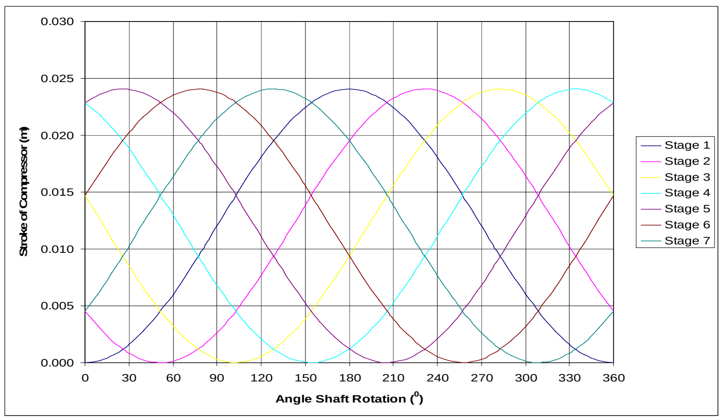

Figure 4.20 Angle shaft rotation vs stroke of compressor for 7 Stage

Figure 4.21 to Figure 4.25 shown the distribution pressure in each cylinder and every moving of piston with 1 (one) shaft rotation. The distribution of pressure in each moving of piston can be know after knowing about the stroke changes in every angle shaft rotation and the tilting angle wobble plate.

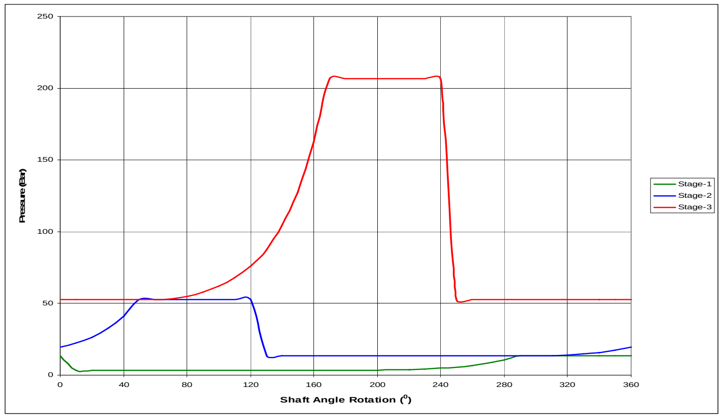

Figure 4.21 Pressure distribution of shaft angle rotation for 3 stage

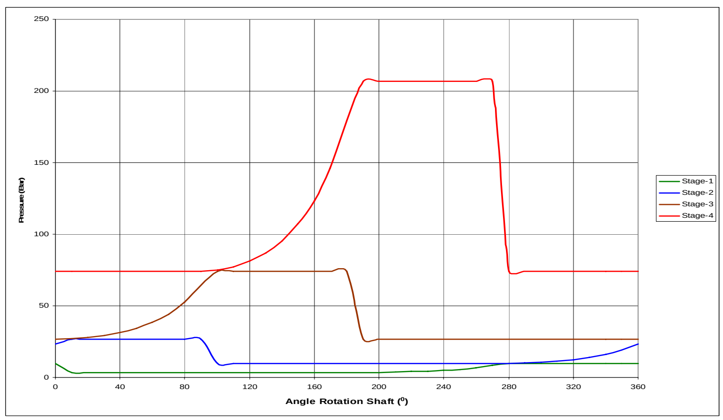

Figure 4.22 Pressure distribution of angle shaft rotation for 4 stage

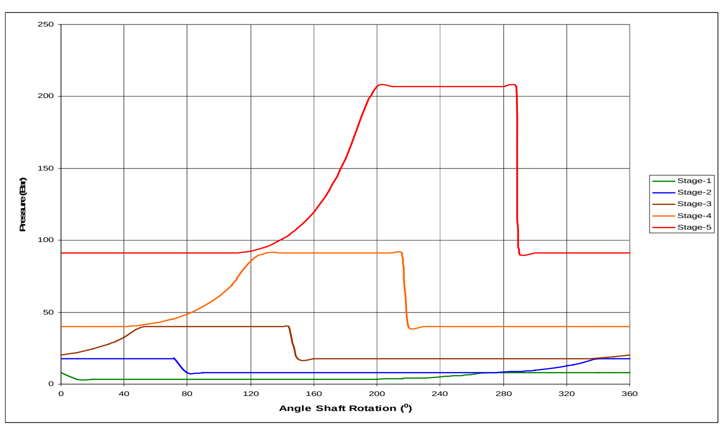

Figure 4.23 Pressure distribution of angle shaft rotation for 5 stage

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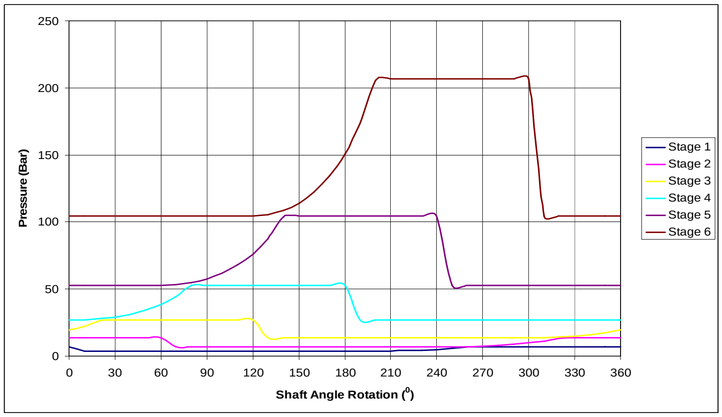


Figure 4.24 Pressure distribution of angle shaft rotation for 6 stage

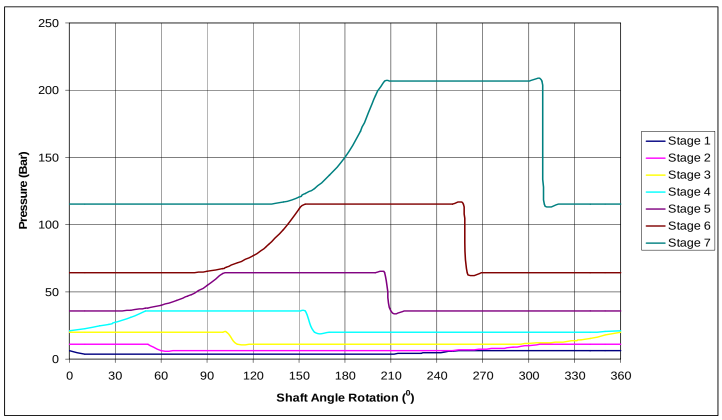


Figure 4.25 Pressure distribution of angle shaft rotation for 7 stage

By knowing the diameter in each piston, the wider of piston can be calculated and then load (force) in each piston can be determined. Based on the analysis that had been done, the results can be seen in Figure 4.26 to Figure 4.30. It can be seen that the changes of force are the same with the changes of the pressure. The last piston for every stage has the heavier force that causes by the gas pressure inside the cylinder are the bigger, it were the discharge processes. At 1st stage, the force and pressure on each piston are small. Hence this is a suction process condition.

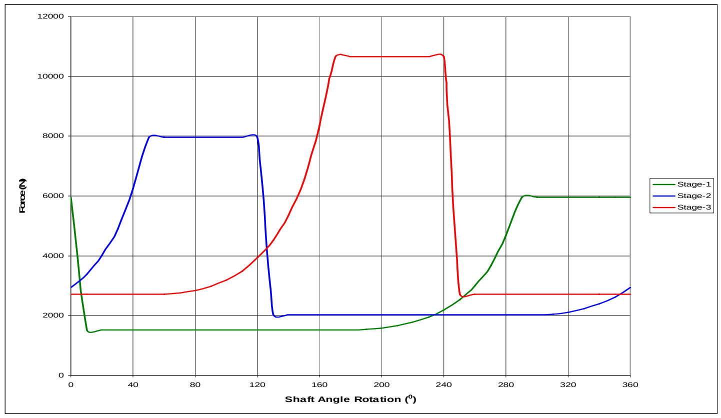


Figure 4.26 Force distribution of angle shaft rotation for 3 stage

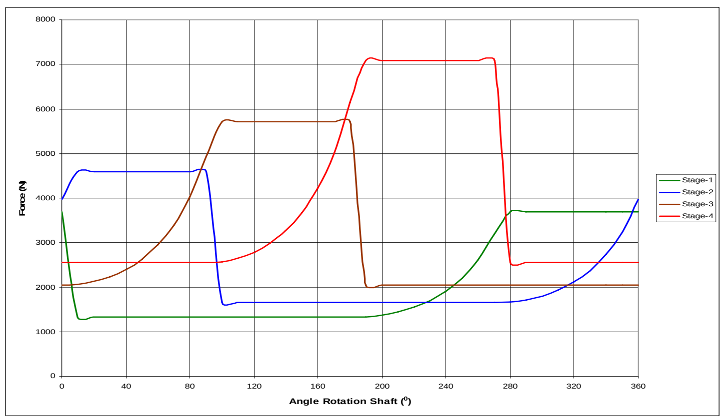


Figure 4.27 Force distribution of angle shaft rotation for 4 stage

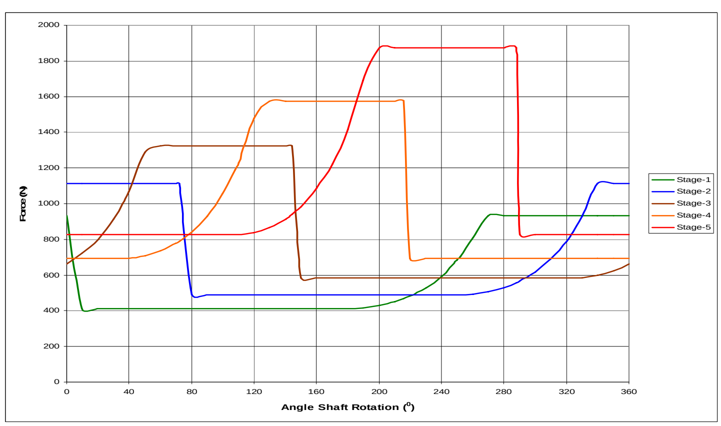


Figure 4.28 Force distribution of angle shaft rotation for 5 stage

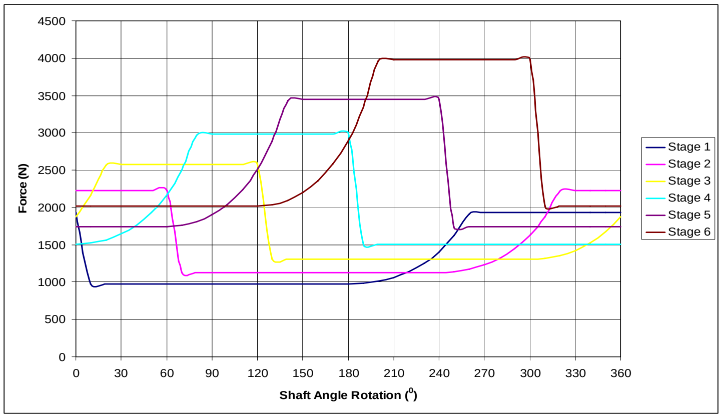


Figure 4.29 Force distribution of angle shaft rotation for 6 stage

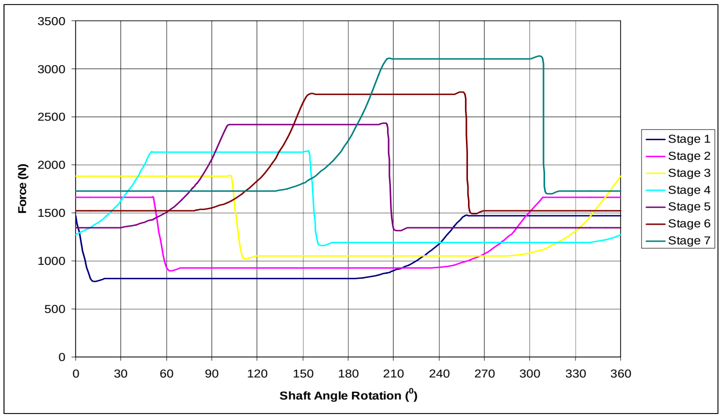


Figure 4.30 Force distribution of angle shaft rotation for 7 stage

In Figure 4.31 and 4.35 can be seen the torque distribution of each piston.

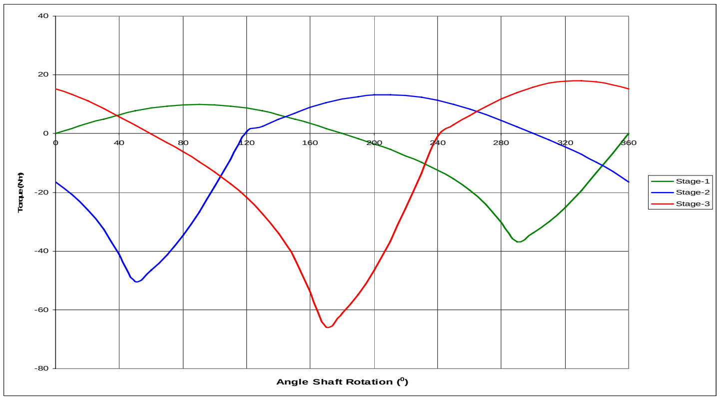


Figure 4.31 Torque distribution of angle shaft rotation for 3 stage

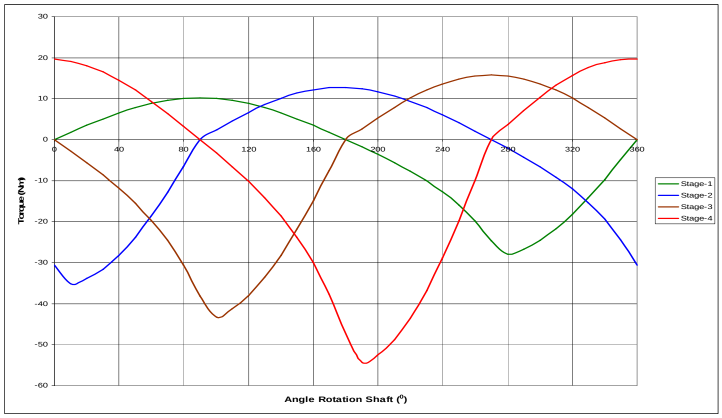


Figure 4.32 Torque distribution of angle shaft rotation for 4 stage

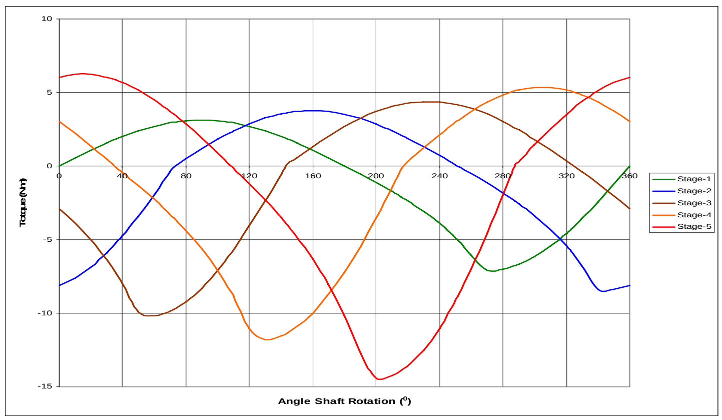


Figure 4.33 Torque distribution of angle shaft rotation for 5 stage

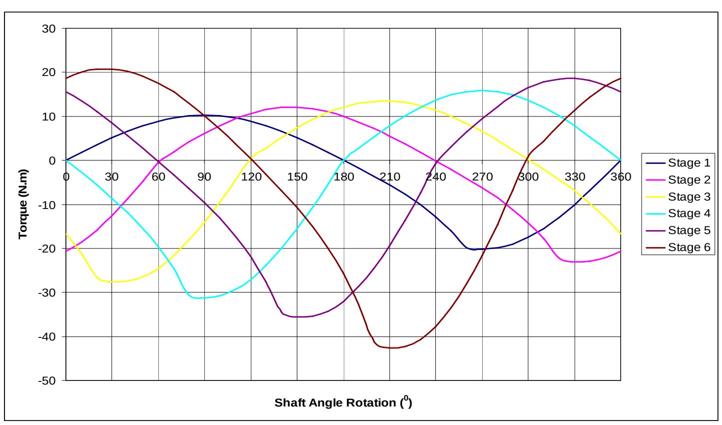


Figure 4.34 Torque distribution of angle shaft rotation for 6 stage

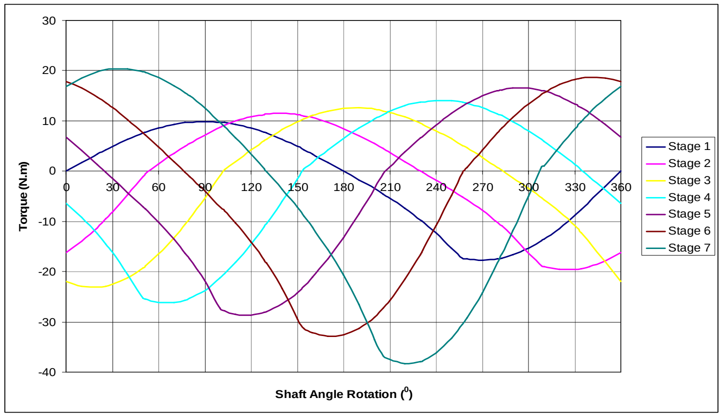


Figure 4.35 Torque distribution of angle shaft rotation for 7 stage

4.5.2. Optimum number of stage design symmetrical wobble plate compressor

The optimum designed of the compressor could be done by combining some of the stages and result from this combination trap has been analyzed.

Table 4.11 The maximum force every stage and every cylinder

Piston	Max force for 3 Stage (N)	Max force for 4 Stage (N)	Max force for 5 Stage (N)	Max force for 6 Stage (N)	Max force for 7 Stage (N)
1	3,454	3,692	933	1,928	1,470
2	4,110	4,590	1,111	2,229	1,664
3	4,892	5,705	1,322	2,577	1,885
4	-	7,093	1,574	2,980	2,134
5	-	-	1,873	3,445	2,417
6	-	-	-	3,983	2,737
7	-	-	-	-	3,099

Table 4.11 above was shown the maximum force that has by each piston. In table above it also shown that if the force become higher the pressure in compressor cylinder also high.

Table 4.12 The maximum and total one rotation shaft: force, torque and work of symmetrical wobble plate compressor

Number of Stage Compressor	Pressure Ratio of Compressor	Maximum Torque of Shaft Rotation Angle Compressor (Nm)	Maximum Force of Shaft Rotation Angle Compressor (N)	Maximum Work of Shaft Rotation Angle Compressor (Watt.s)
3	3.91	-53.7	14,210	53.7
4	2.78	-41	12,132	41
5	2.27	-17. 4	6,590	17. 4
6	1.98	-43.4	12,355	43.4
7	1.79	-34.1	11,464	34.1

Table 4.12 shows the maximum torque, load and work that have been done by compressor. That table could be seen based on the number or stage and the pressure ratio compressor.

Table 4.13 The maximum force every position of symmetrical wobble plate compressor for any stage with shaft angle rotation.

Shaft Angle Rotation (^{0})	Total Force of Compressor (3 Stage) N	Total Force of Compressor (4 Stage) N	Total Force of Compressor (5 Stage) N	Total Force of Compressor (6 Stage) N	Total Force of Compressor (7 Stage) N
0	11,618	12,268	5,630	11,297	10,888
10	7,623	10,535	5,011	10,643	10,317
20	8,249	10,596	5,118	11,088	10,432
30	9,150	10,703	5,266	11,181	10,596
40	10,491	10,864	5,475	11,299	10,845
50	12,213	11,095	5,787	11,469	11,182
60	12,213	11,416	5,867	11,705	10,530
70	12,240	11,864	5,925	10,948	10,654
80	12,322	12,497	5,182	11,465	10,831
90	12,465	13,409	5,302	11,576	11,099
100	12,681	11,255	5,468	11,714	11,496
110	12,989	11,332	5,701	11,910	10,772
120	13,417	11,465	6,045	12,185	10,906
130	8,079	11,667	6,211	11,310	11,117
140	8,917	11,954	6,275	11,907	11,438
150	10,124	12,354	5,384	12,029	11,841
160	11,920	12,912	5,516	12,189	11,189
170	14,211	13,697	5,701	12,416	10,919
180	14,211	14,828	5,958	12,732	11,144
190	14,226	12,132	6,327	11,708	11,464
200	14,271	12,171	6,590	12,356	11,721
210	14,350	12,241	6,619	12,429	11,245
220	14,470	12,345	5,490	12,506	11,310
230	14,641	12,494	5,552	12,615	11,402
240	14,880	12,702	5,638	12,768	11,537
250	7,282	12,992	5,757	11,288	11,741
260	7,751	13,401	5,931	11,623	10,686
270	8,429	13,992	6,113	11,694	10,756
280	9,441	9,956	6,144	11,783	10,856
290	10,719	10,006	4,797	11,910	11,001
300	10,719	10,092	4,866	12,088	11,220
310	10,739	10,223	4,962	10,377	10,052
320	10,800	10,408	5,095	10,763	10,126
330	10,907	10,668	5,286	10,842	10,234
340	11,068	11,028	5,544	10,945	10,388
350	11,298	11,536	5,578	11,092	10,622
360	11,618	12,268	5,630	11,297	10,888

Table 4.13 was shown the force maximum in each number of stages of compressor. The torque that has from the table above is the total force in each piston and the changes of the shaft angle rotation. Total force of compressor in 3 stage; force in the first piston until the third piston be sum up in the same condition or in one shaft angle rotation position.

Table 4.14 The maximum torque of symmetrical wobble plate compressor for any stage with shaft angle rotation.

Shaft Angle Rotation (°)	Max Torque of Compressor (3 Stage) N.m	Max Torque of Compressor (4 Stage) N.m	Max Torque of Compressor (5 Stage) N.m	Max Torque of Compressor (6 Stage) N.m	Max Torque of Compressor (7 Stage) N.m
0	-1.2	-11.0	-1.9	-3.2	-17.0
10	-5.3	-17.0	-2.5	-7.3	-15.9
20	-10.9	-18.0	-3.2	-12.8	-14.4
30	-18.2	-18.6	-4.2	-14.4	-11.8
40	-27.8	-19.0	-5.4	-15.6	-8.7
50	-38.5	-19.4	-7.1	-16.9	-4.2
60	-36.5	-20.1	-7.2	-18.5	-2.8
70	-33.4	-21.5	-6.9	-22.5	-0.4
80	-29.3	-23.8	-7.2	-27.9	0.2
90	-24.5	-27.8	-7.7	-28.4	0.3
100	-19.6	-34.1	-8.4	-28.3	0.8
110	-14.9	-33.7	-9.3	-28.2	-3.2
120	-10.9	-32.6	-10.7	-28.6	-5.9
130	-15.7	-31.3	-10.8	-32.5	-10.6
140	-21.6	-30.1	-10.0	-37.6	-16.3
150	-29.7	-29.2	-9.9	-37.4	-21.6
160	-40.9	-29.3	-10.1	-36.7	-29.2
170	-53.4	-30.8	-10.5	-36.0	-29.4
180	-49.6	-34.6	-11.2	-35.7	-33.2
190	-44.2	-41.0	-12.3	-38.8	-34.1
200	-37.5	-39.0	-12.5	-43.4	-32.7
210	-29.8	-35.8	-11.0	-40.8	-36.1
220	-21.4	-31.7	-9.5	-36.8	-32.3
230	-12.7	-27.0	-8.7	-31.9	-28.7
240	-4.2	-21.9	-7.7	-26.4	-20.8
250	-5.3	-16.8	-6.8	-23.6	-14.8
260	-8.2	-12.2	-5.9	-21.6	-6.4
270	-10.9	-8.8	-4.9	-16.3	-1.5
280	-9.9	-10.9	-2.9	-10.6	-1.7
290	-8.6	-9.0	-1.3	-5.2	6.6
300	-7.0	-7.2	-0.9	-0.5	7.7
310	-5.3	-5.6	-0.8	-1.2	6.0
320	-3.6	-4.5	-0.9	-3.1	4.7
330	-2.2	-4.2	-1.5	-2.2	0.1
340	-1.2	-4.9	-2.5	-1.8	-2.7
350	-0.8	-7.0	-2.2	-2.0	-9.5
360	-1.2	-11.0	-1.9	-3.2	-18.8

Table 4.14 above shows the maximum torque in each number of compressor stages. Torque in that table was the total of torque in each piston and the changes of the shaft angle rotation.

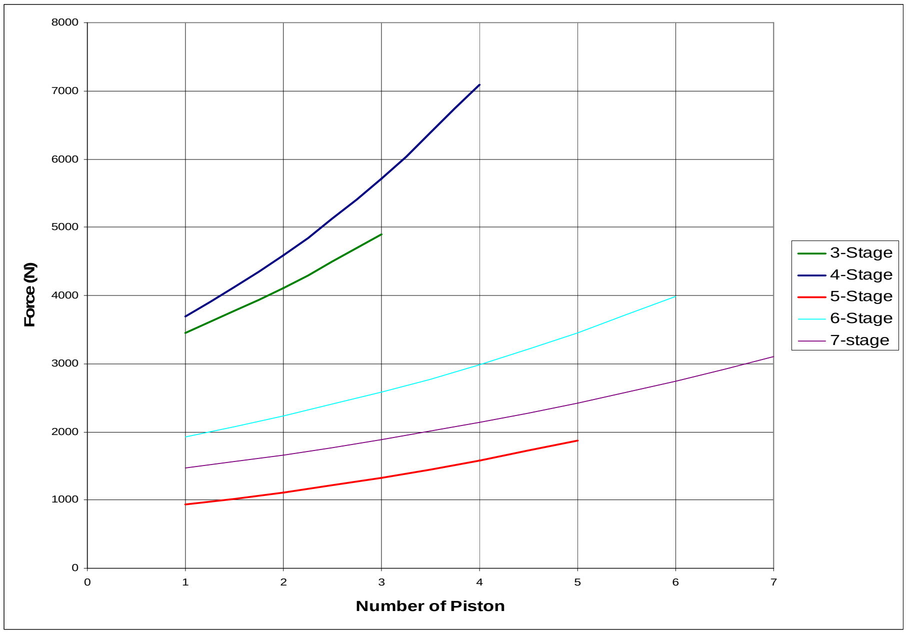


Figure 4.36 Load in each piston for each number of stages.

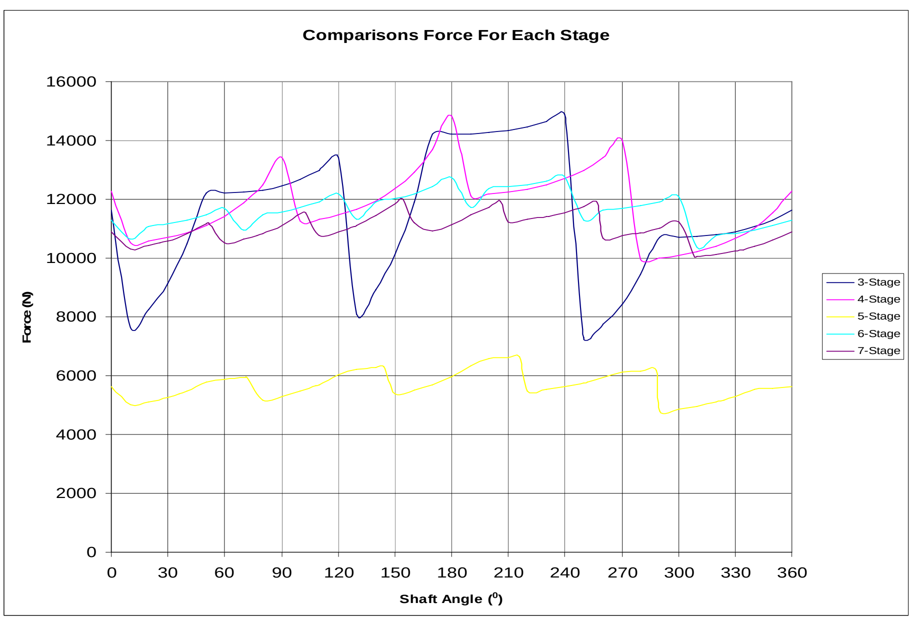


Figure 4.37 Compressor total force in each shaft angle rotation with the number stage of compressor.

To determine the optimum number of stages it can be do by check the force or load on the compressor. Load on each piston are shown Figures 4.36 and 4.37. That figures shown that the maximum load had by stage 3 and the minimum on the stage 5 and the measures of load depends on the pressure ration on each stage; the pressure ratio become bigger, load or force on the piston also become bigger. If it shows on the pressure ratio only, the minimum load should be on the stage 7 of compressor because in this compressor design not only the pressure ratio that is the determinant factors. The availability of the space for the configuration of cylinder and the size of the last piston has to be considered. By considered others factor except the pressure factors, the minimum load could be achieved on the stage 5 of the compressor.

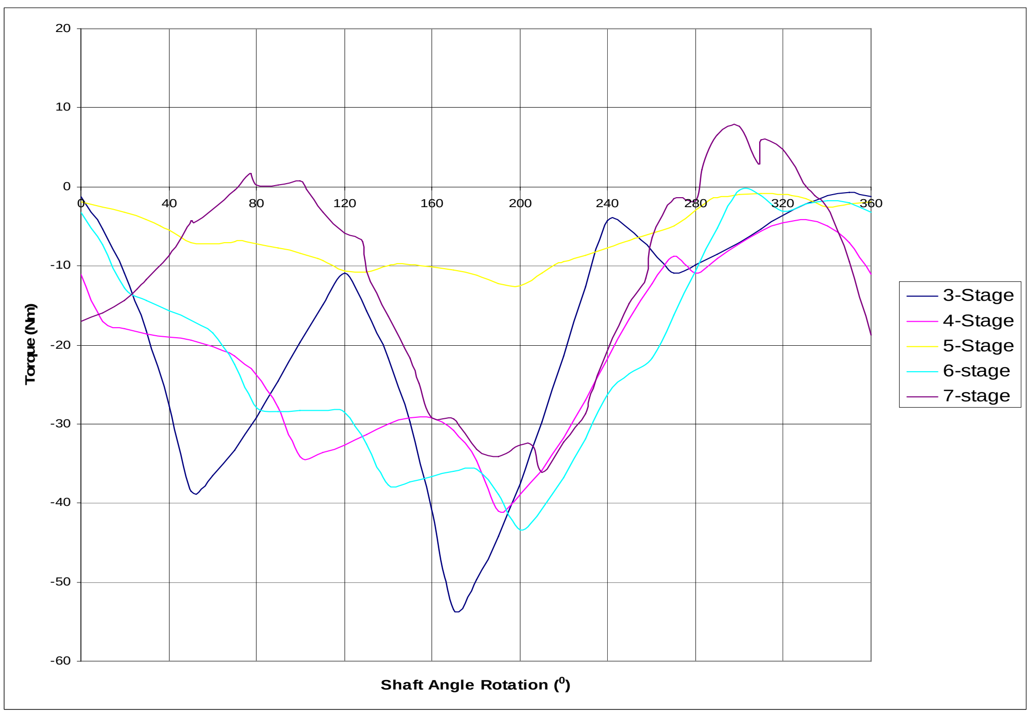


Figure 4.38 Total torque at the compressor in each shaft angle rotation with number of compressor stage

Based on the force that has by each piston, piston number 5 has the optimum force. In Figures 4.36 and 4.37 it can be seen or determined the number of stages that more optimum. Torque in 3 stage of compressor has the highest torque and lowest torque in stage 5. As usual, if the pressure ratios become smaller the torque does too. But for this case it is not happen because radius or diameter wobbles plate become bigger. The decrease of pressure ratio and piston diameter is not proportional with the enlargement of the diameter wobbles plate as shown in Figure 4.39. In Figure 4.39 it can be seen that if the number of stage become bigger, the dimension of wobble plate also become bigger but not for the diameter of piston, where it become smaller.

The diameter of piston in the 6th and 7th stages were bigger than in the 1st stage. If in the 6th and 7th stages the diameter of the 1st piston smaller than the 5th piston, the last piston was the smallest. It is impossible for process production and the availability of grove to give the piston ring and raider ring or guide ring. If the number of stages become bigger so that the space that was needed for the cylinder also bigger. It is impossible to reduce the wobble plate radius. In this analysis the dimension for the size of parts is already maximized for each stage.

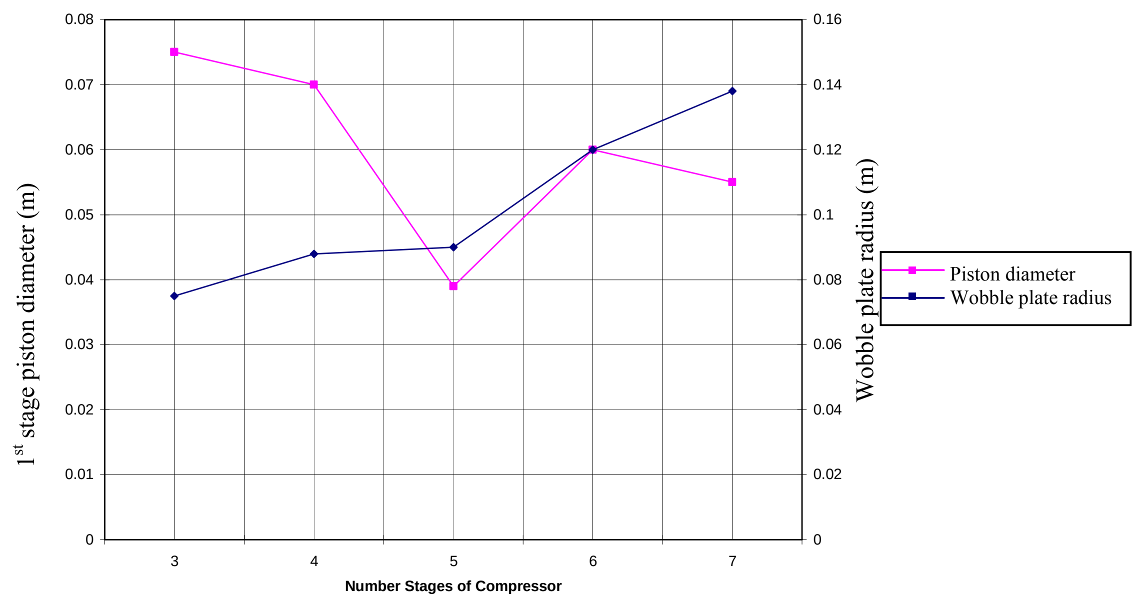


Figure 4.39 Correlation diameter piston, radius wobble plate, and number of stage of compressor

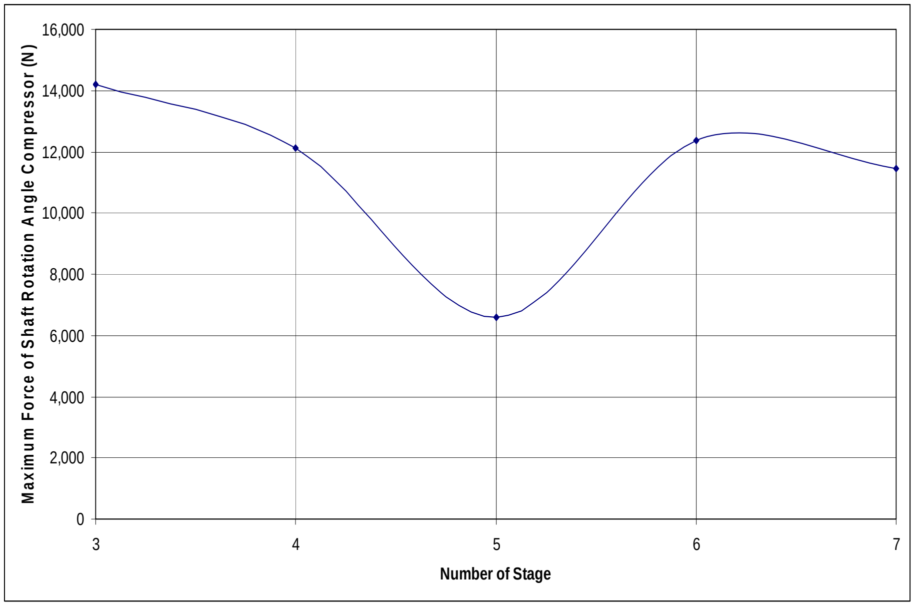


Figure 4.40 Force maximum on the compressor

Figure 4.40 shows the maximum load or force occurred at stage 3 and minimum at stage 5.

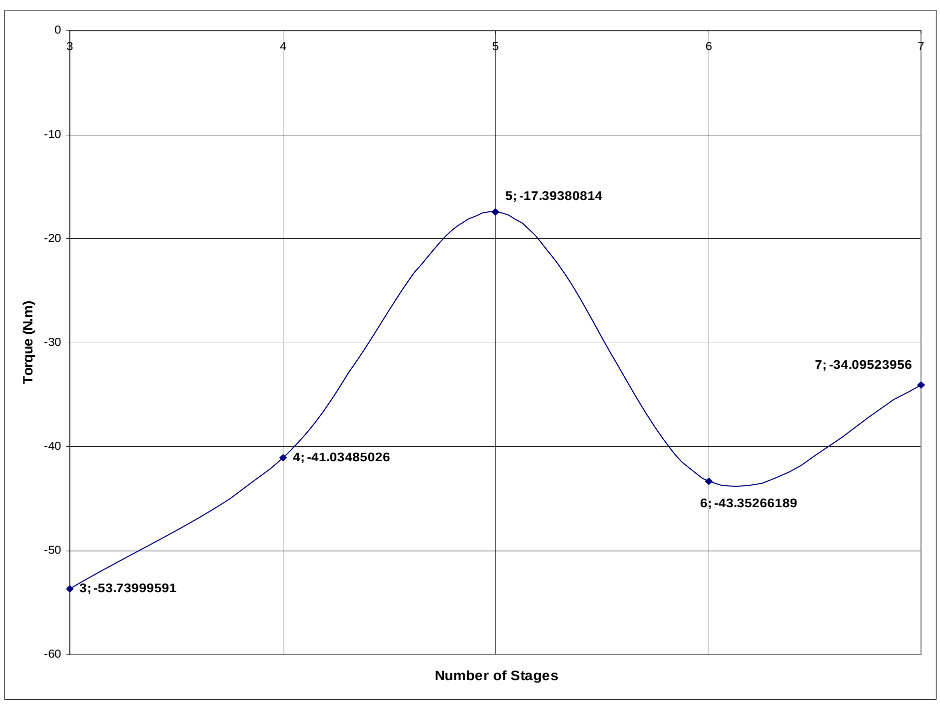


Figure 4.41 Torque maximum on the compressor

Figure 4.41 shows the torque that happen in each number of stages of compressor. It shown that the 5th stage has the smallest torque in the compressor. Before choosing the number of stages it is better to check the torque in compressor. Compressor that has the smallest torque will have better performance than compressor that have the biggest torque. This is has relation with the energy or the work that done by compressor. It can be seen in the Figure 4.42. The torque become smaller, the work done by the compressor also becomes smaller. So that the efficiency of compressor more increasing and its means that this compressor has better performance. Beside that the compressor torque has relation with the drive of compressor (motor).

Motor that has smaller torque will have smaller geometry and the price is cheaper. This is suitable with the aims of this compressor developing in term has home refueling and mini station. One of factor that have to deliberate is the total of geometry compressor include with the drive. This geometry must be small and compact. Other important thing is the motor price. The motor price must be straight proportional with that motor's torque; the torque higher, the motor price will more expensive.

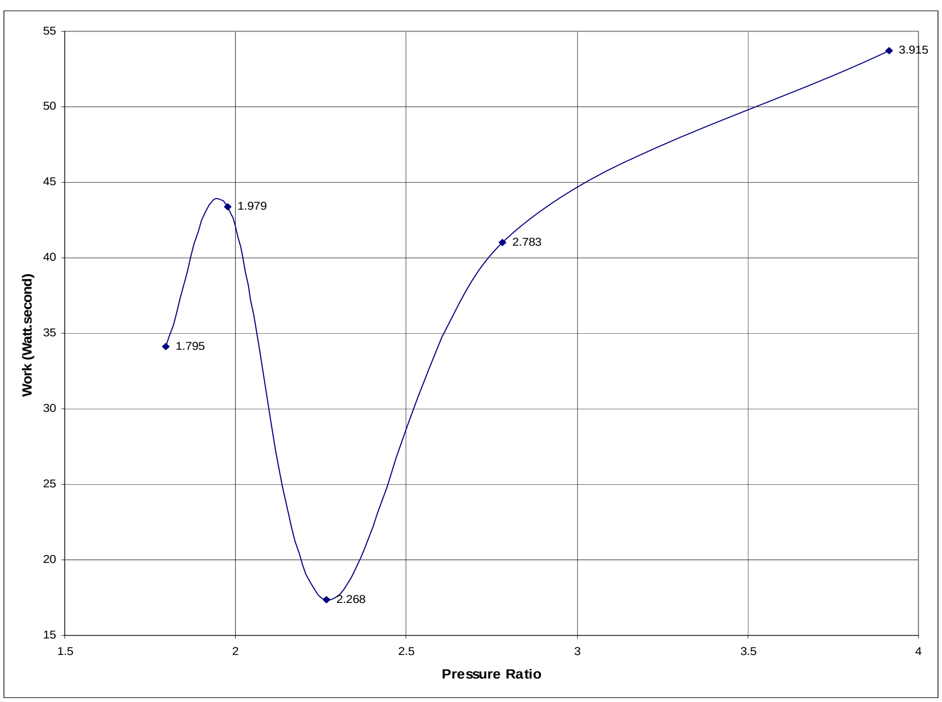


Figure 4.42 Work of compressor vs pressure ratio

4.5.3. Optimum tilting angle symmetrical wobble plate compressor

Appendix F shows the torque value that has by compressor with it variation-tilting angle of wobble plate and shaft angle of rotation. The analysis was done with constant capacity 10 m^3/hr and constant stages of compressor (5 stages). The aim of this analysis is to determine the optimum of tilting angle wobble plate compressor. In Figures 4.43 and 4.44 shows the changes of torque of compressor based on the tilting angle wobble plate of compressor. The changes of tilting angle wobble plate compressor to become bigger also caused the increase of compressor torque as shown in Figure 4.44. This analysis cannot assist the determination of the optimum of tilting angle. The choosing of tilting angle wobbles could be done based on the availability of end joint in the market.

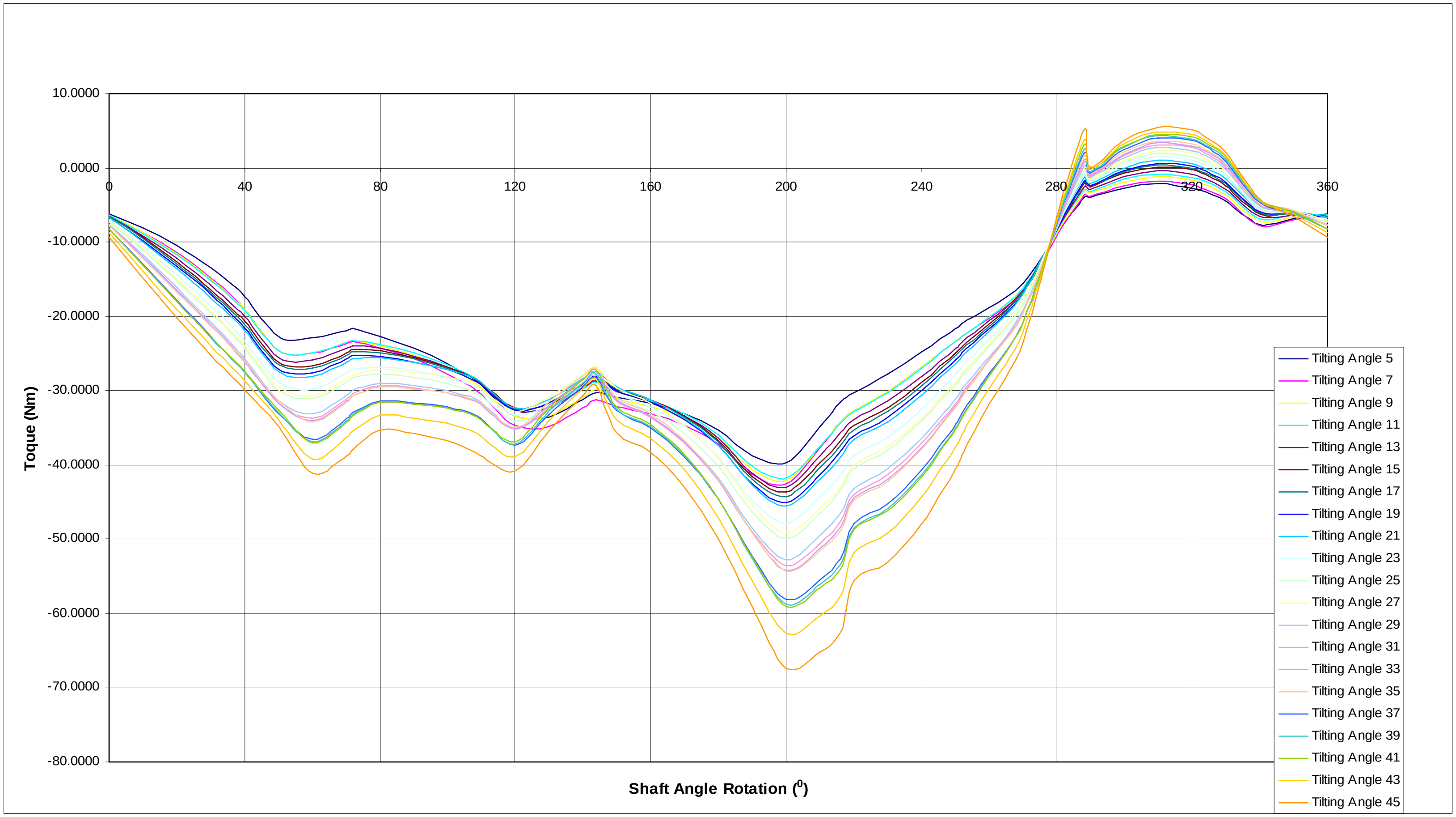


Figure 4.43 Variation torque of compressor with shaft angle rotation

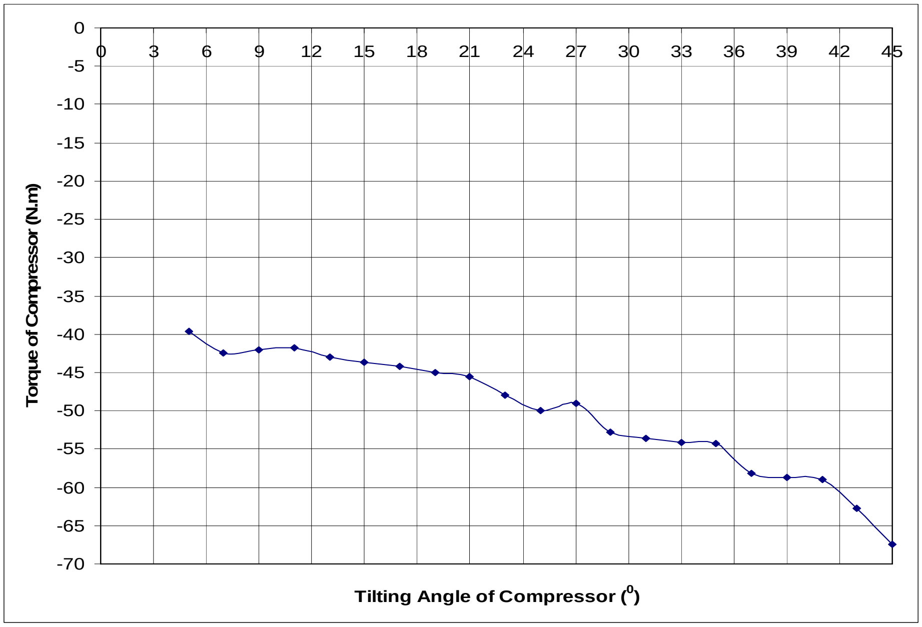


Figure 4.44 Tilting angle of compressor vs torque of compressor

Initially, the designed started with an appropriate steps and a tilting wobble plate angle as VRA compressor. Hence designer process is reenacted as according to the requirements. The design based on 16 º tilting angle.

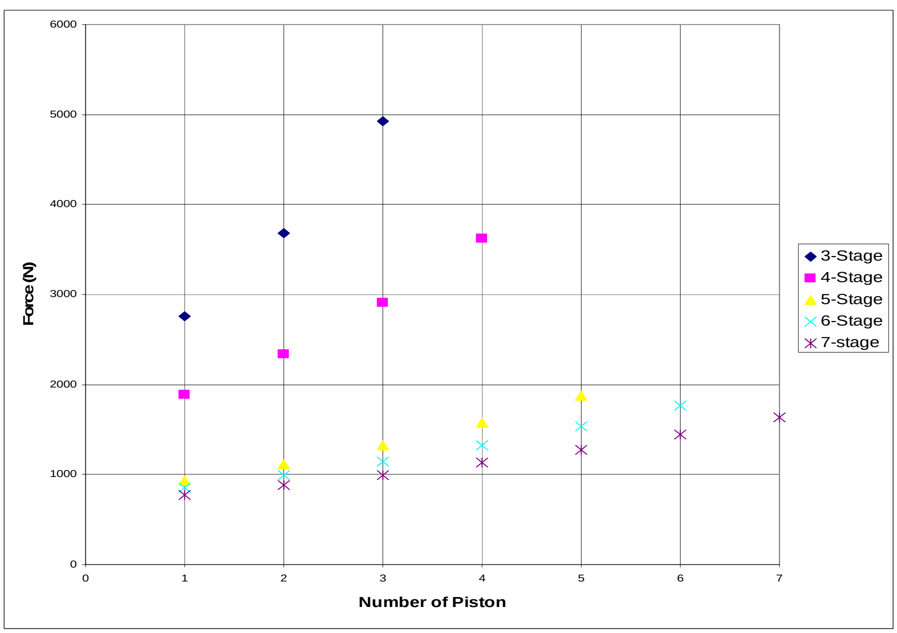


Figure 4.45 Load in each piston for each number of stages at tilting angle 16º

Figure 4.45 shows that the maximum load experienced by piston at stage 3 while minimum at stage 7. This is due to piston diameter at stage 7 is smaller compare to other stages. On top of it, the pressure in cylinder block stage 7 is also smaller compare to others due to smaller pressure ratio.

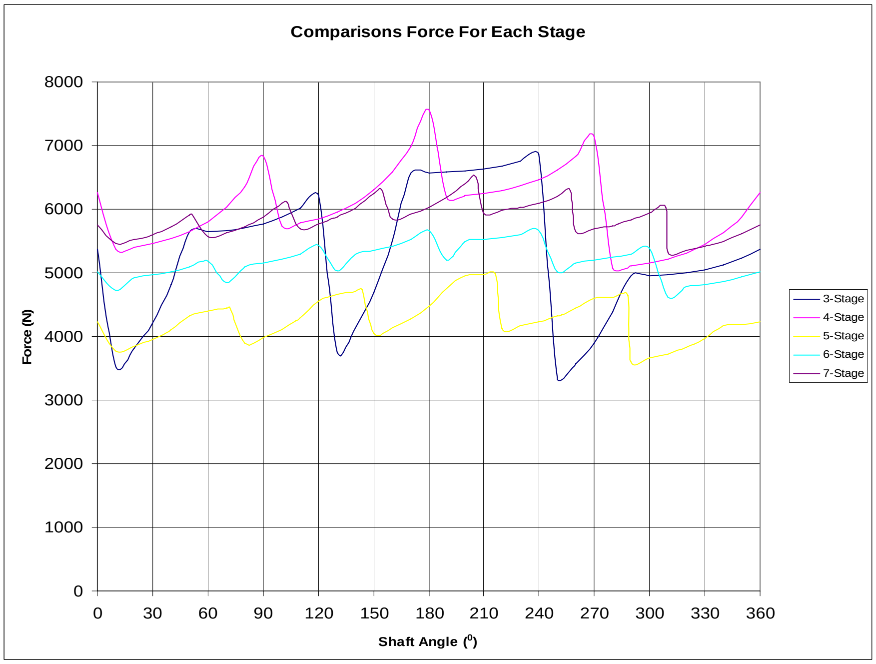


Figure 4.46 Compressor total force in each shaft angle rotation with the number stage of compressor at tilting angle 16°

Total load of the compressor could be determined when we know maximum load in each stage. Figure 4.46 shows a distribution total of load for difference of shaft rotation angle. Minimum load was found at stage 3 while maximum at stage 4.

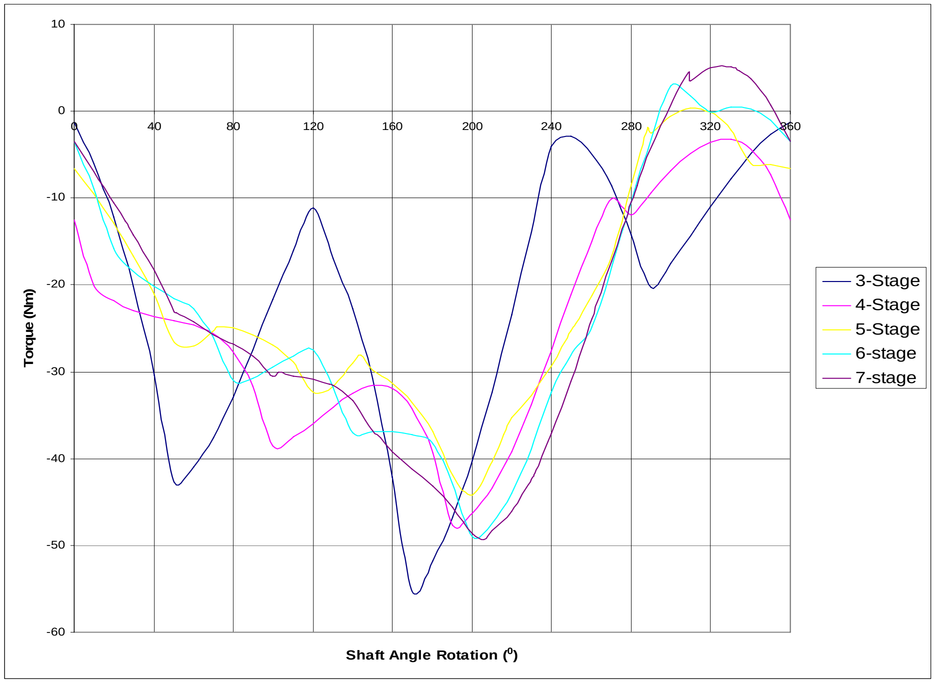


Figure 4.47 Total torque at the compressor in each shaft angle rotation with number of compressor stage at tilting angle 16º

The optimum number of stages is 5. Stage 3 has the biggest twist as shown in Figure 4.47. Even though a total load of stage 3 is small, the torque is big due to bigger radius of the wobble plate.

4.6 Conclusion

Based on analysis, it can be concluded that the optimum number of stages for capacity of 10 m^3/hr is 5 stage compressors with specification as following:

Table 4.15 Optimum specification of symmetrical wobble plate compressor

Input Data	**Calculated Value**				
	First Stage	**Second Stage**	**Third Stage**	**Fourth Stage**	**Fifth Stage**
Cylinder diameter	39 mm	28.25 mm	20.47 mm	14.83 mm	10.74 mm
Suction Pressure	3.4 bar	7.8 bar	17.7 bar	40.2 bar	91.2 bar
Discharge Pressure	7.8 bar	17.7 bar	40.2 bar	91.2 bar	206.8 bar
Diameter of wobble plate	87 mm				
Stroke	47.96 mm				
Fluid	Natural gas				
Rotating Speed	1500 rpm				
Tilting Angle	16^o				
Pressure ratio	2.7				
Capacity	10 Nm^3/hr				
Mechanical efficiency	85%				
Volumetric efficiency	90.5%				

Complete engineering drawing and patent filing (PI 20055456) for new multistage symmetrical wobble plate compressor can be shown in Appendix G and Appendix H

CHAPTER 5

THERMODYNAMIC, FLOW AND HEAT TRANSFER ANALYSIS

5.1 Introduction

The thermodynamic analysis, done on the compressor was based on many parameters such as the volume ratio, leakage, discharge porting area, valve dimension, and a heat transfer. In this symmetrical wobble plate compressor, the volume ratio (or the compression ratio) depends on number of stages, stroke length, and diameter of each cylinder. A leakage flow simulation modeling is used to demonstrate the mass flow rate through all clearances. Two different types of leakage namely; the flank and the tip leakages had been modeled. The discharge model porting process depends on the motion of the piston and the valve actions affect the discharge flow process losses. The lumped heat transfer model was used to evaluate the amount of heat gained in suction process. This chapter will discuss a complete analysis of the suction, compression and discharge processes. The leakage and heat transfer modeling are used for analytical simulation.

5.2 Thermodynamic Properties within the Cylinder Block

It is essential to analyze the thermodynamic properties within the pocket, in order to determine the pressure, temperature, and natural gas flow rate in the operating process. The possibility of leakage, heat transfer, over-compression, and under-compression conditions were also investigated.

In the analysis, the working fluid which is the natural gas was assumed as a perfect gas. The gas properties were assumed uniform and steady throughout the process in the control volume. The inlet and outlet velocities and leakage opening to the control volume all are assumed constant.

The inputs to the thermodynamic model are the state of gas at the start of the suction process i.e., pressure and temperature. In addition the speed of compressor and the discharge pressure are also the input parameters to the model. The other inputs are geometrical which includes the parameters that would limit capacity of the compressor. These parameters are number of stages, tilting angle, diameter of piston, and stroke length.

The outputs of the model are the prediction of properties at all intermediate states of the compressor process, the pressure and temperature, the mass flow rate, the compressor work, the average wall temperature of the suction and discharge. The model also predicts the discharge temperature, instantaneous torque, the adiabatic efficiency, the coefficient of performance and volumetric efficiency.

5.2.1. Suction Process

The gas flow into the compressor cylinder block by opened and closed valve the operation at the inlet. The volume in the suction pockets and in the inlet cross section area into the pocket will vary with the shaft angle of rotation. As shown in Figure 5.1, during the suction process cylinder volume of each stages increases and the gas was sucked into the cylinder. It takes about 175º of rotation for each stage to complete the suction process. The compression process is discussed in 5.2.2.

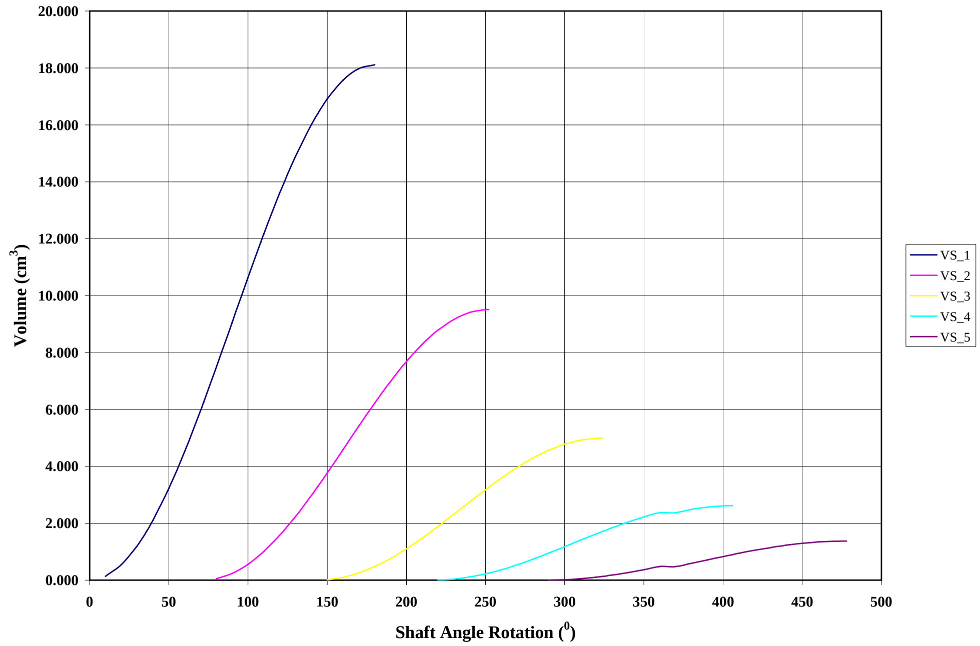


Figure 5.1 Suction volume at various rotation angle

5.2.1.1 Suction Mass Flow Rate

The suction process can be divided into two working conditions with reference to the value as presented by Zhu (1990). The first one is the steady suction when the value is fully open the volume of the cylinder increases. The second is a negative suction process when the value is closing. Suction end when the piston is at the bottom dead center. As compression stroke begins pressure of gas increases and suction value (discharge value al ready closed during suction) starts to close. Compression only begins when suction value is completely closed. Therefore during closing of suction value some gas is rejected and this defined as "negative suction process" during which the volume of gas decreases.

The suction mass flow rate of symmetrical wobble plate compressor depends on the cylinder diameter, displacement of compressor and temperature of the gas. As mentioned before, the suction process modeling could be done in two stages increase of volume followed by instantaneous decrease in volume of gas. In the first stage, the process was assumed a quasi-static filling condition such that the mass flow rate could be evaluated by multiplying volume rate to the density of the gas. The following equation could be applied for this stage. The volume rate can be computed from the finite difference procedure using the numerical volume equation (4.15) given in chapter 4.

The suction mass flow rate is given by:

$$\dot{m} = \dot{V} \ x \, \rho \qquad 5.1$$

Where, $\dot{V}$ is the volume rate.

In the second stage of suction, equation for the instantaneous steady isentropic flow will be applied as follows:

$$\dot{m} = A_{dn}.P_{up}\sqrt{\frac{2\gamma}{(\gamma\text{-}1)RT_{up}}}\sqrt{\left(\frac{P_{dn}}{P_{up}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{dn}}{P_{up}}\right)^{\frac{\gamma+1}{\gamma}}} \qquad 5.2$$

This is valid for an un-choked flow but for a critical flow when setting mach number, M = 1, then

$$\frac{P_{critical}}{P_{up}} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma\text{-}1}} = r_c \qquad 5.3$$

For both critical and sub-critical flow it is assumed that $P_d = P_{critical}$, where $P_{critical}$ is the downstream pressure at M=1 i.e. if the pressure inside the pocket is greater than the discharge pressure then $P_{critical}$ is the discharge pressure and if there is a back flow then the downstream pressure is the pocket pressure. The critical pressure ratio r_c is a constant for a given value of γ and the flow is chocked for pressure ratio less than the critical ratio. The mass flow rate under the chocked condition is

$$\dot{m}_{critical} = A_{dn}.P_{up}\sqrt{\frac{2\gamma}{(\gamma\text{-}1)RT_{up}}}\sqrt{\left(r_c\right)^{\frac{2}{\gamma}} - \left(r_c\right)^{\frac{\gamma+1}{\gamma}}} \qquad 5.4$$

At unchoked condition,

$$r = \frac{P_{dn}}{P_{up}} \qquad 5.5$$

Where r is the pressure ratio.

5.2.1.2 The Average Rate of Heat Transfer at Suction

The gas that flows into the cylinder block or chamber will mix with the temperature of gas where the main source of the heat and the cylinder wall. The gas that flows into the cylinder through the suction port will circulate inside the cylinder. Gas flows into the cylinder with high temperature and therefore there will be a heat transfer from the gas to the cylinder wall. When the suction value is completely closed the suction process finished, and the compression process begins. The thermodynamic of the gas from beginning of suction to the beginning of compression processes could be evaluated as follows. The schematic diagrams of the suction processes are shown in Figure 5.2.

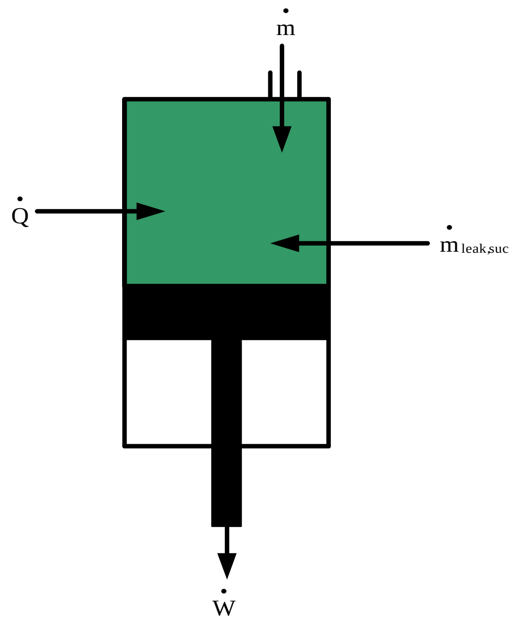


Figure 5.2 Schematic diagram for suction process

Application of the first law of thermodynamic to the control volume of the suction pocket, will give:

Change of internal energy = (energy into system)
- (energy out of system)

$$Rate\ of\ change\ of\ internal\ energy = \dot{Q} - \dot{W} + m_{in} h_{in} + \dot{m}_{leak,in}\, h_{leak,in} \qquad 5.6$$

Where:

$\dot{Q}$ = Instantaneous rate of heat into the volume

$\dot{W}$ = Instantaneous work done by the gas in the volume control

h_{in} = The Specific enthalpy of the gas at the suction control volume

$$h_{leak} = \frac{(h_{dis} + h_{su})}{2} \qquad 5.7$$

In steady state condition:

$$Rate\ of\ change\ of\ internal\ energy = \left[\left(\dot{m} + \dot{m}_{leak}\right)\left(u_{cv}\right)_{t1} - \left(\dot{m}\right)\left(u_{cv}\right)_{t0}\right] \quad 5.8$$

And at the beginning of the cycle $\left(\dot{m}\right)\left(u_{cv}\right)_{t0} = 0$

Thus

$$\left(\dot{m} + \dot{m}_{leak}\right)\left(u_{cv}\right)_{t1} = \dot{Q}_{suc} - \dot{W}_{suc} + \dot{m}\,h_{in} + \dot{m}_{leak,in}\,h_{leak,in} \qquad 5.9$$

The work interaction in the suction process could be divided into two parts; first is work in the expansion process of fluid and second is the work to overcome friction. If the frictional work be ignored, the work input in the suction process could be determined as follows:

$$\dot{W}_{suc} = \dot{W}_{\exp antion} = \omega \int_{0}^{V_1} P.dv$$

$$= \omega.P_L.V_1 \qquad 5.10$$

By substituting $\dot{W}_{suc}$, equation 5.3 becomes

$$\left(\dot{m} + \dot{m}_{leak}\right)\left(u_{cv}\right)_{t1} = \dot{Q}_{suc} - \omega.P_L.V_1 + \dot{m}\,h_{in} + \dot{m}_{leak,in}\,h_{leak,in} \qquad 5.11$$

Putting volume at the end discharge (V_1) = (mass) t_1 x specific volume (ν)

$$\left(\dot{m} + \dot{m}_{leak}\right)\left(u_{cv}\right)_{t1} = \dot{Q}_{suc} - \omega\left(m_{cv}\right)_{t1}.P_L\left(\upsilon_{cv}\right)_{t1} + \dot{m}\,h_L \qquad 5.12$$

$$\because \omega\left(m_{cv}\right)_{t1} = \dot{m} + \dot{m}_{leak,in}\text{, and } P_L = P_1\text{, specific volume } (\nu_{cv})_{t1} = \nu_1$$

$$\dot{Q}_{su} = \left(\dot{m} + \dot{m}_{leak,in}\right)u_1 + \left(\dot{m} + \dot{m}_{leak,in}\right)P_1\upsilon_1 - \dot{m}\,h_L - \dot{m}_{leak,in}\,h_{leak,in}$$

$$\dot{Q}_{su} = \dot{m}\,u_1 + \dot{m}_{leak,in}\,u_1 + \dot{m}\,P_1\upsilon_1 + \dot{m}_{leak,in}\,P_1\upsilon_1 - \dot{m}\,h_L - \dot{m}_{leak,in}\,h_{leak,in}$$

$$\dot{Q}_{su} = \left(\dot{m}\right)\left(u_1 + P_1 \upsilon_1\right) + \left(\dot{m}_{leak,in}\right)\left(u_1 + P_1 \upsilon_1\right) - \dot{m}\, h_L - \dot{m}_{leak,in}\, h_{leak,in}$$

when $\left(u_1 + P_1 \upsilon_1\right) = h_1$, then, the average rate of heat transfer to the suction will be

$$\dot{Q}_{su} = \dot{m}\left(h_1 \text{-} h_L\right) + \dot{m}_{leak,in}\, h_1 - \dot{m}_{leak,in}\, h_{leak,in}$$

$$\dot{Q}_{su} = \dot{m}\left(h_1 \text{-} h_L\right) + \dot{m}_{leak,in}\left(h_1 - h_{leak,in}\right) \qquad 5.13$$

5.2.2. Compression Process

In the compression process both ports in the cylinder were closed. As the cylinder volume decreases the pressure is increased as shown in Figure 5.3. The mass inside the closed system is the suction mass minus the leakage mass. Due to the gas leakage, the temperature and pressure increase inside compression chamber. At the end of the closed process, the gas in the pocket (in general) is not equal to the discharge pressure. Therefore, the equilibrium process comes out to adapt with the discharge pressure as described in Figure 5.4. The thermodynamic governing equations for the compression process are given below:

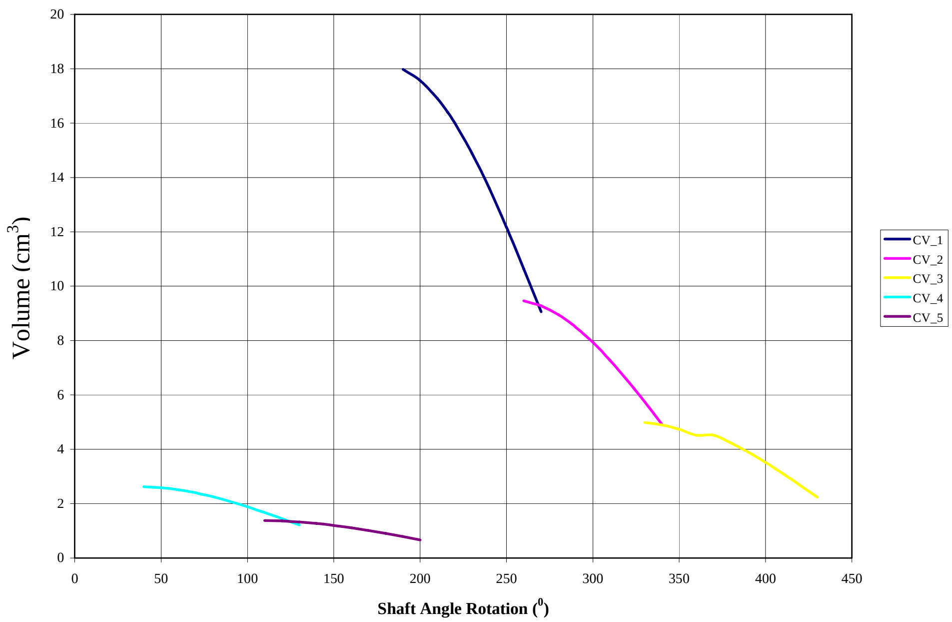
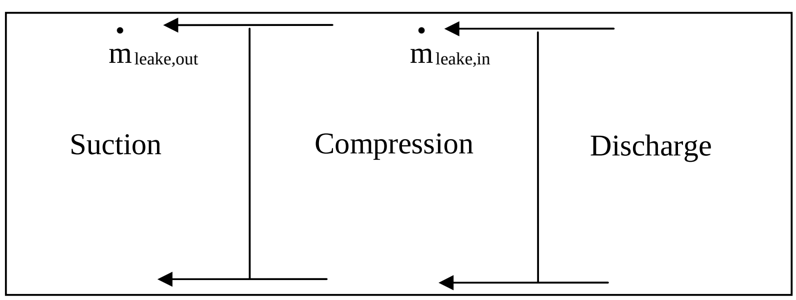


Figure 5.3 Compression volume at various rotation angle

Continuity equation

$$\frac{\partial m_{cv}}{\partial t} = \dot{m}_{leak,in} - \dot{m}_{leak,out}$$



$$\int dm_{cv} = \int_{t_1}^{t_2} \left(\dot{m}_{leake,\ in} - \dot{m}_{leake,\ out} \right) dt$$

$$m\ (cv)_{t2} - m\ (cv)_{t2} = \int_{t_1}^{t_2} \dot{m}_{leake,\ in}\, dt - \int_{t_1}^{t_2} \dot{m}_{leake,\ out}\, dt$$

$$m\,(c\,v)_{t2} = m\,(c\,v)_{t2} + \int_{t_1}^{t_2} \dot{m}_{leake,\,in}\,dt - \int_{t_1}^{t_2} \dot{m}_{leake,\,out}\,dt$$

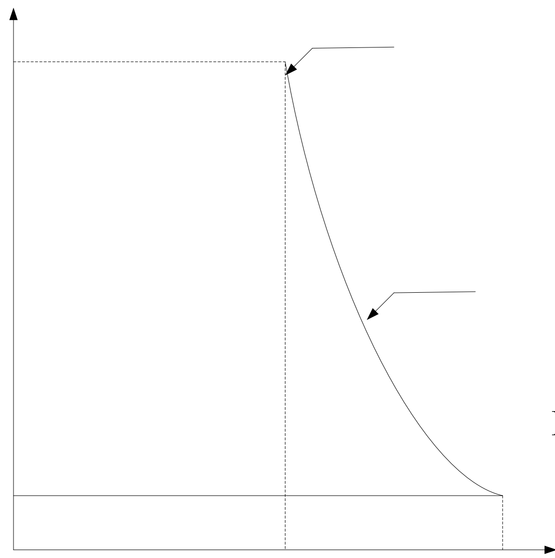


Figure 5.4 Equilibrium process

5.2.2.1 Pressure and Temperature in Closed Process

The first law of thermodynamic states that the change of the total energy (kinetic, potential, and internal) of a control mass is equal to the heat transfer to the control mass minus the work done by the control mass. This can be applied on the control volume mathematically in order to evaluate the pressure distribution in the closed cylinder.

$$E_{in} = E_{out} + \frac{du}{dt} \qquad 5.14$$

Where:

E_{in} = the amount of heat or energy entering the control volume

E_{out} = the amount of work done by the control volume and, or the energy leaving the control volume

$\frac{du}{dt}$ = the rate of change of internal energy of the control volume

$$E_{in} = \frac{dQ}{dt} + \sum \frac{dm_{in}}{dt}\left(u_{in} + \frac{p_{in}}{\rho_{in}} + \frac{v^2}{2} + g\, z_{in}\right) \qquad 5.15$$

The schematic diagram for the compression process can be shown in Figure 5.5 if potential and kinetic energy are neglected because these are small relative to the enthalpy term:

$$\left(u_{in} + \frac{p_{in}}{\rho_{in}}\right) = Cp_{in}\, T_{in}$$

then,

$$E_{in} = \frac{dQ}{dt} + \sum \frac{dm_{in}}{dt} cp_{in} T_{in}$$

$$E_{out} = P\frac{dV}{dt} + \sum \frac{dm_{out}}{dt} cp_{out} T_{out} \qquad 5.16$$

$$\frac{du_c}{dt} = \frac{d}{dt}\left(m_c u_c\right) \qquad 5.17$$

By substituting E_{in}, E_{out}, and $\frac{du}{dt}$ in equation 5.8 and rearranging the equation to get the following relation:

$$\frac{dp}{dt} = \frac{1}{V}\left[(\gamma\text{-}1)\frac{dQ}{dt} + R\gamma\left(\sum \frac{dm_{in}}{dt} T_{in} - \sum \frac{dm_{out}}{dt} T_{out}\right) - \gamma p \frac{dV}{dt}\right] \qquad 5.18$$

The effect of the heat transfer in the closed process is negligible as found experimentally by Sankar (1997), therefore, the model developed in this analysis neglects the effect of the heat transfer in the compression process. For the temperature it can be expressed with respect to time from the differential on the equation of the state and can be rearranged to be as follows:

$$\frac{dT}{dt} = T\left[\frac{1}{V}\frac{dV}{dt} + \frac{1}{p}\frac{dp}{dt} + \frac{1}{m}\frac{dm}{dt}\right] \qquad 5.19$$

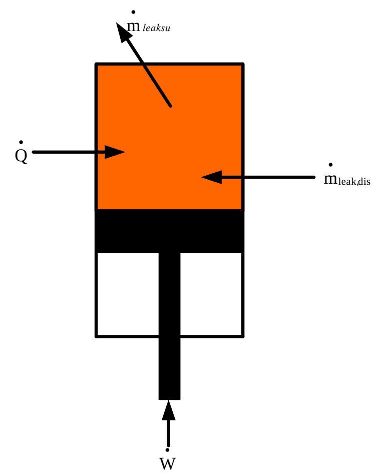


Figure 5.5 Schematic diagram for compression process

5.2.3. Discharge Process

This is the end of the compression process, where the pressure in cylinder is higher than that of the discharge port. The schematic diagram for the discharge process is shown in Figure 5.6 and the discharge volume in Figure 5.7. The average rate of heat transfer out from the discharge fluid can be evaluated as in suction process from the first law of thermodynamic as follows:

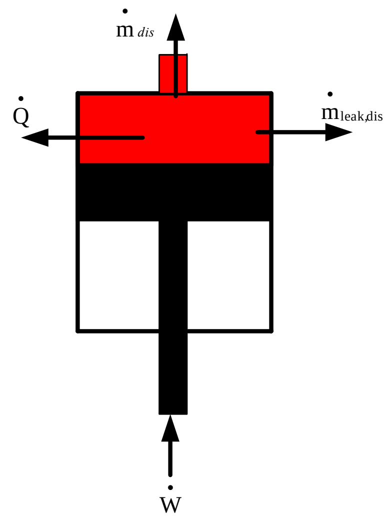


Figure 5.6 Schematic diagram for discharge process

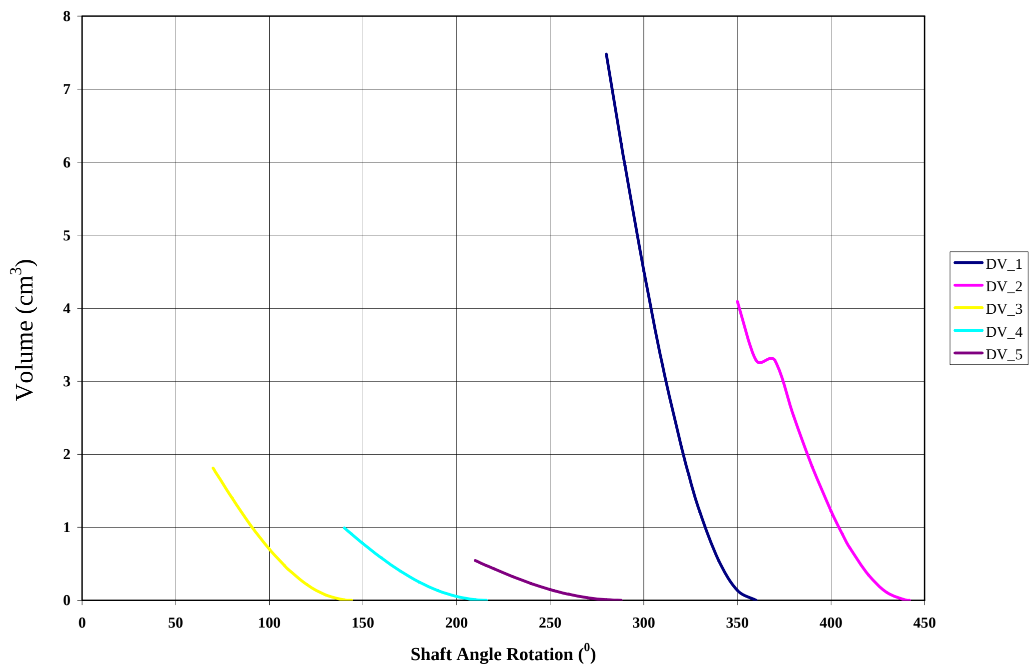


Figure 5.7 Discharge volume at various rotation angle

$$\text{Rate of internal energy} = -\dot{Q} + \dot{W} - \dot{m}_{out}\, h_{out} - \dot{m}_{leak,out}\, h_{leak,out}$$

A steady state condition, the average rate of change of internal energy is:

$$= \left[\left(\dot{m} \right) \left(u_{cv} \right)_{t1} - \left(\dot{m} + \dot{m}_{leak} \right) \left(u_{cv} \right)_{t0} \right] \qquad 5.20$$

And $\left(\dot{m} \right) \left(u_{cv} \right)_{t1} = 0$ at the end of the cycle then,

$$- \left(\dot{m} + \dot{m}_{leak} \right) \left(u_{cv} \right)_{high} = -\dot{Q}_{suc} + \dot{W}_{dis} - \dot{m}\, h_{out} - \dot{m}_{leak,out}\, h_{leak,high} \qquad 5.21$$

Rearranging the equation then the rate of heat transfer at suction will be:

$$\dot{Q} = \dot{m} \left(h_{high} - h_{out} \right) \qquad 5.22$$

The spring loaded valve could be useful for preventing the back flow because it closes the discharge port to prevent back flow into the discharge port. The spring loaded valve opens only when the pressure of the discharge pocket is greater than the pressure of the discharge and permits the gas to flow from the discharge port.

The discharge process can be modeled into two processes the first is before opening the valve and the second after opening. The process before opening the valve can be treated as the compression process for evaluating the pressure, temperature and mass inside the pocket. As soon as the valve opens mass flow from the discharge port hole and the same equation in compression can be used.

5.2.3.1 Flow through Spring Loaded Valve

A flow through the discharge port when the spring loaded valve is open as shown in Figure 5.8. To assist in the design of the complete discharge system, a simple model of the flow process has to be developed.

Generally, the principle work of the spring loaded valve is almost the same with others. The differences are in the suction and the discharge of the valve where the spring loaded valve depends on the differences of the pressures. If the pressure in the cylinder smaller than in the suction and discharge port, the discharge port will be closed by the valve plate, the suction port will be open and the suction process will be occur. Otherwise, if the pressure in the cylinder is higher than the suction port, the suction port will be close, the discharge port will be open and the discharge process will occur.

The design process and the analysis of this spring loaded valve can be refereed to Section 4.4. To assist in the analysis the COSMOS Flow Work version (2004) which is based on advanced Computational Fluid Dynamics (CFD) can be used as a tool.

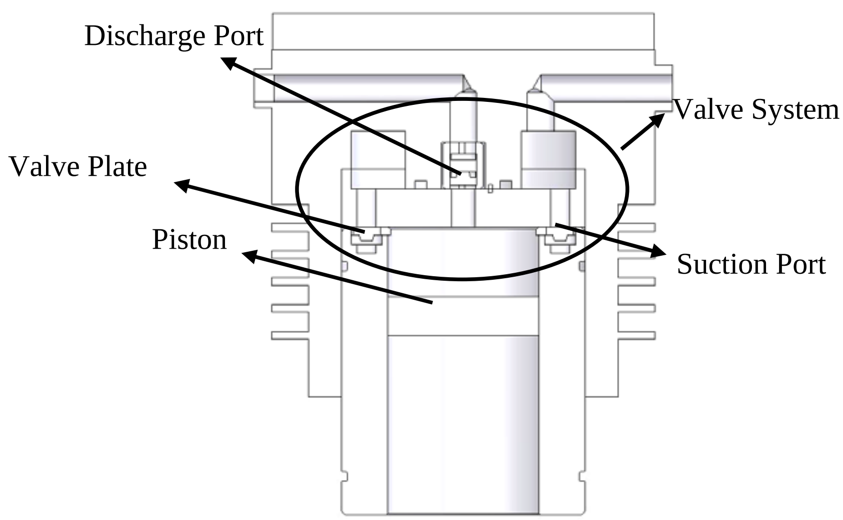


Figure 5.8 Spring loaded valve

5.2.3.1.1 Discussion on Flow Simulation and Analysis

Flow simulation could be used to predict the flow parameter's field such as pressure distributions, velocity distribution, Mach number and temperature distribution of the flowing gas. This information is required to assist in the design work. Figure 5.9 to 5.28 shows the respective results of the flow simulation for each cylinder stage during suction and discharge. The simulation in 3-D on Figure 5.9 for cylinder of stage 1 for example has its average values at and along the core (free stream) of the flow plotted on graphs shown in Figure 5.10 for the cylinder of stage 1 on each graph. In Figure 5.10 (a) the pressure difference at suction is only about (3.456-3.450) = 0.006 bar. For cylinder 5, the pressure difference at suction is about 3 times higher, i.e. (9.1400-9.1200) = 0.02 bar. Important information to be noted is that during suction and discharge process, the Mach number M must be less than 0.2. This condition is required in order to avoid choking at suction and discharge valve of all cylinders. In cylinder 1, gas temperature during suction is about 30ºC at the valve and about 29ºC in the cylinder.

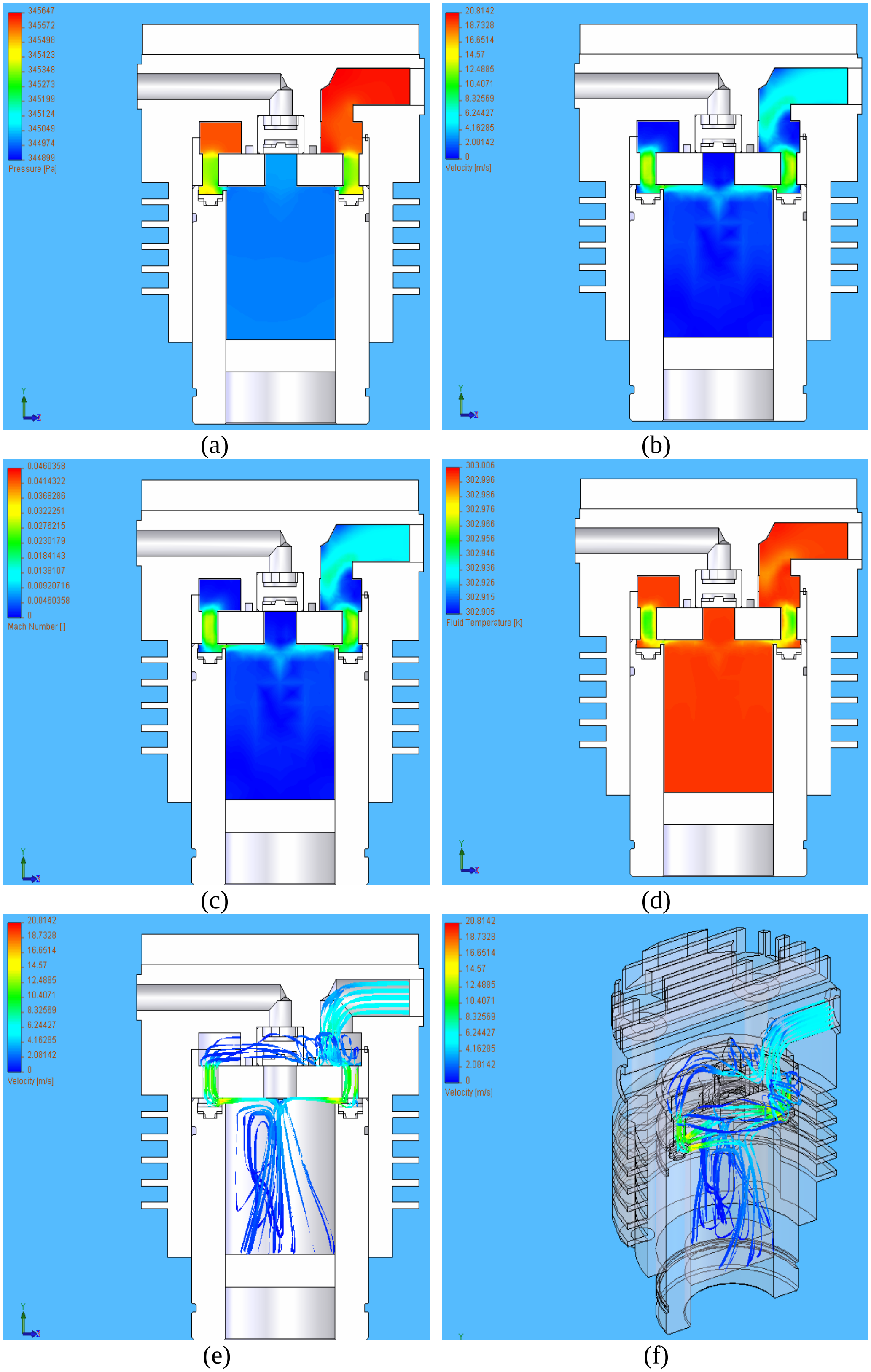
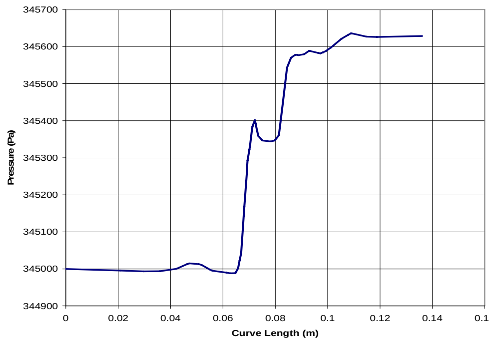
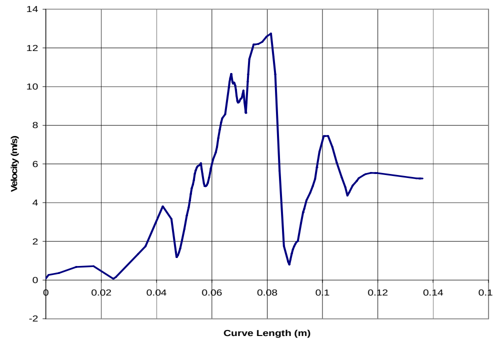


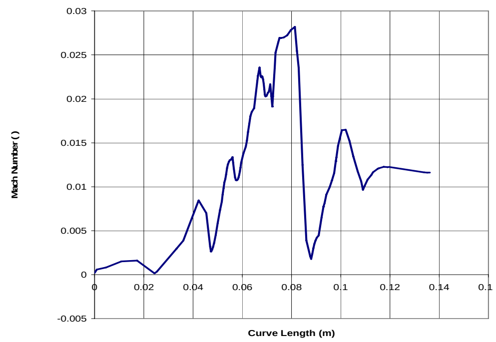
Figure 5.9 Flow analysis of cylinder 1 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Flow Trajectories



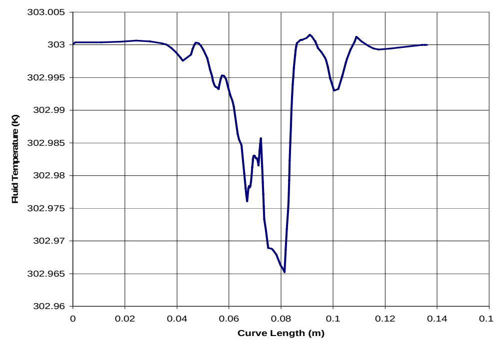
(a)



(b)



(c)



(d)

Figure 5.10 Graph flow analysis of cylinder 1 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature

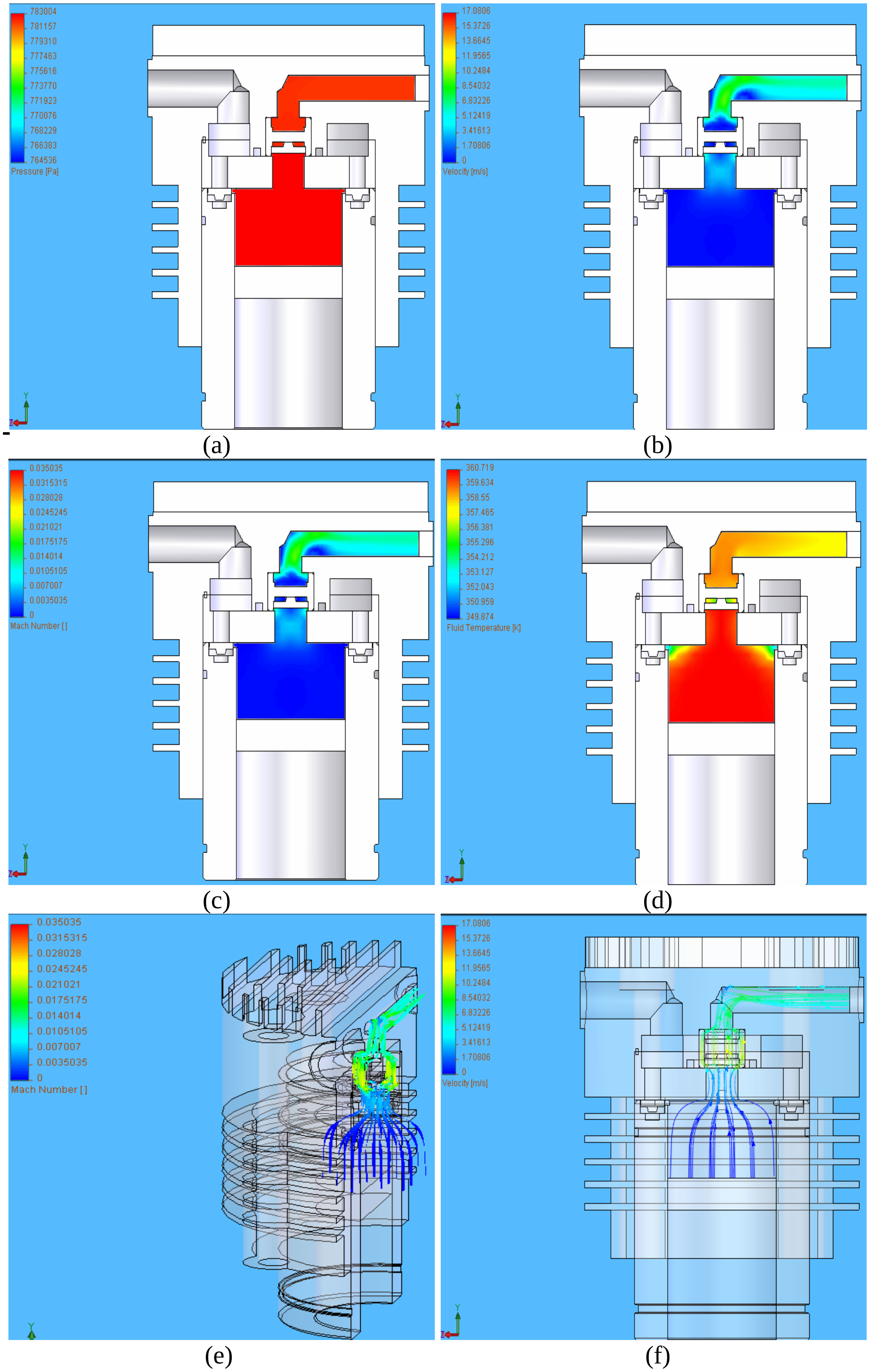
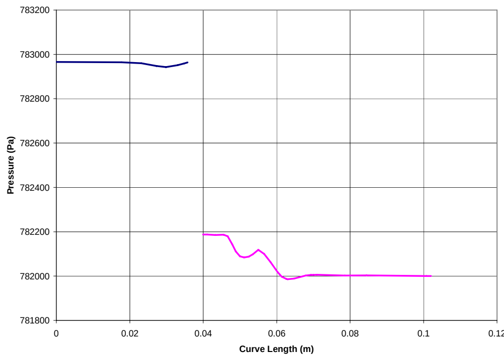
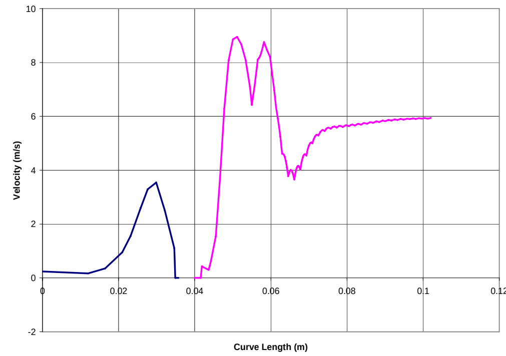


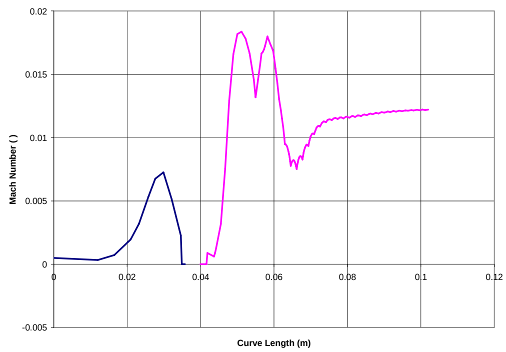
Figure 5.11 Flow analysis of cylinder 1 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Isometric view Flow Trajectories (f). Flow Trajectories



(a)



(b)



(c)

Figure 5.12 Graph flow analysis of cylinder 1 (discharge)
(a). Pressure (b). Velocity (c). Mach number

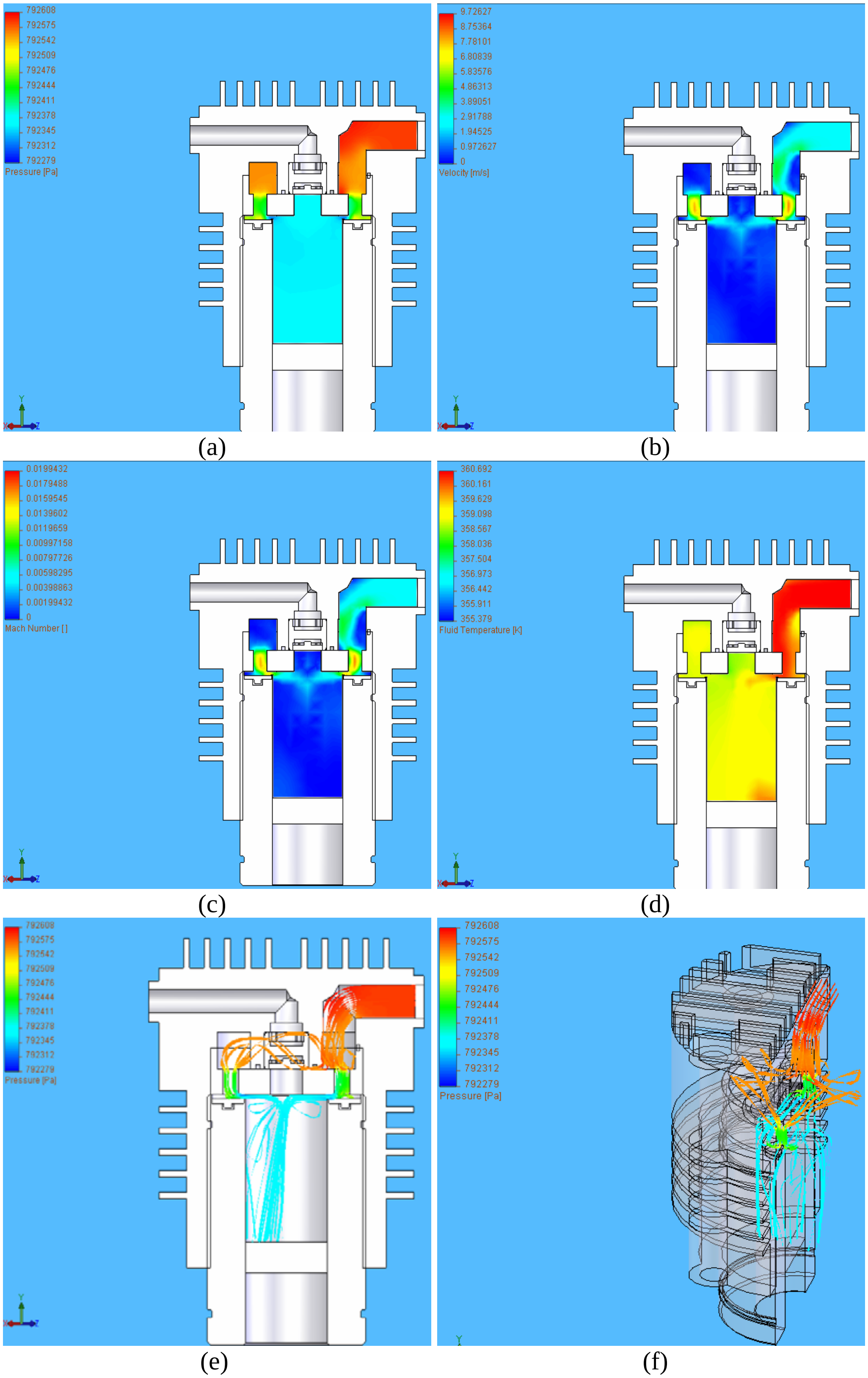


Figure 5.13 Flow analysis of cylinder 2 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Flow Trajectories

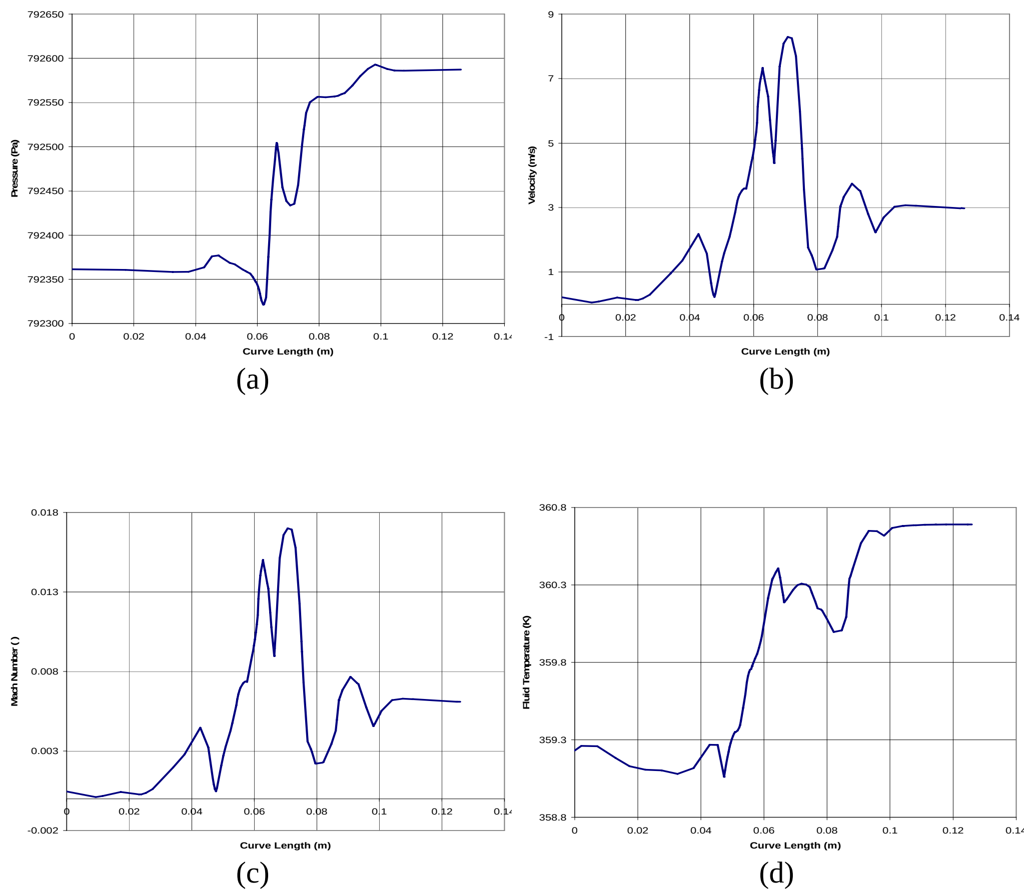


Figure 5.14 Graph flow analysis of cylinder 2 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature

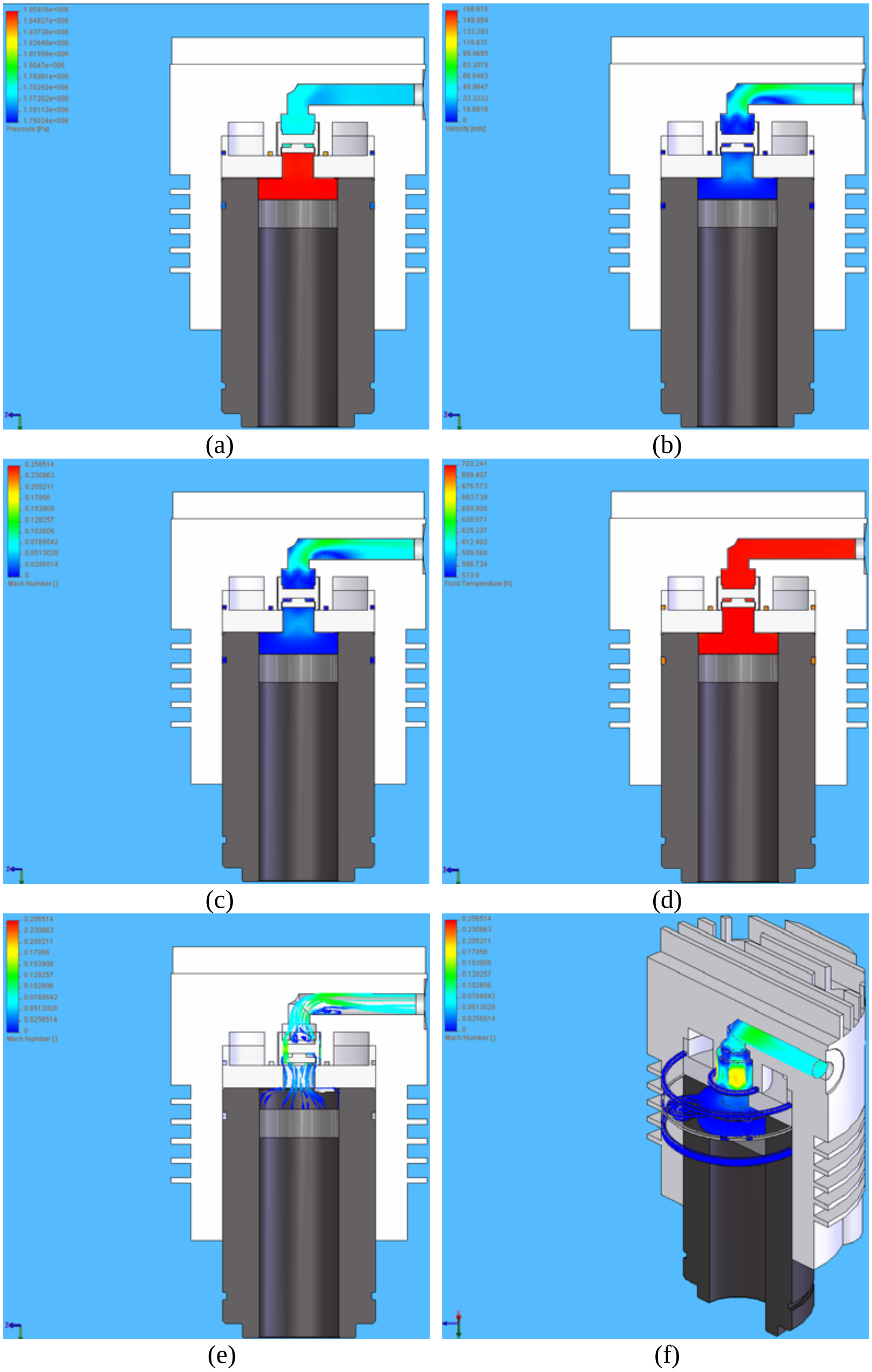
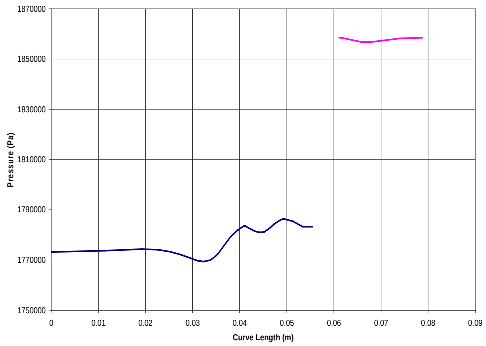
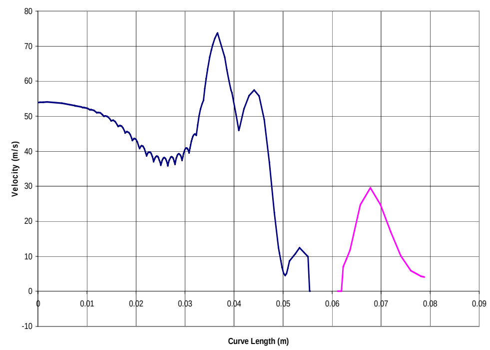


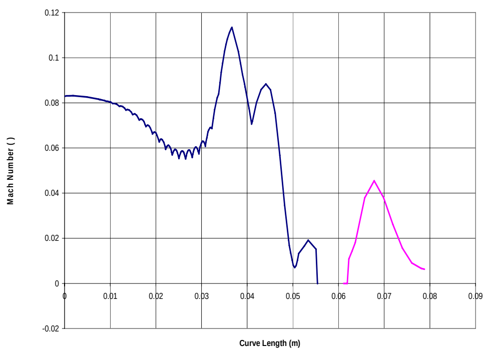
Figure 5.15 Flow analysis of cylinder 2 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Surface Plot



(a)



(b)



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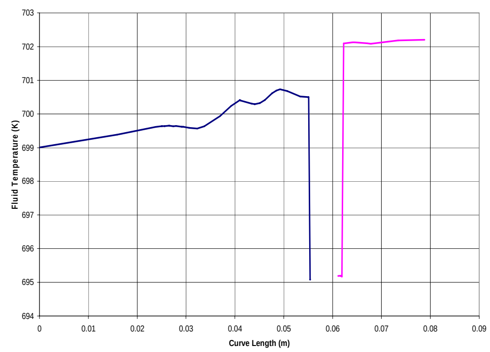


(d)

Figure 5.16 Graph flow analysis of cylinder 2 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature

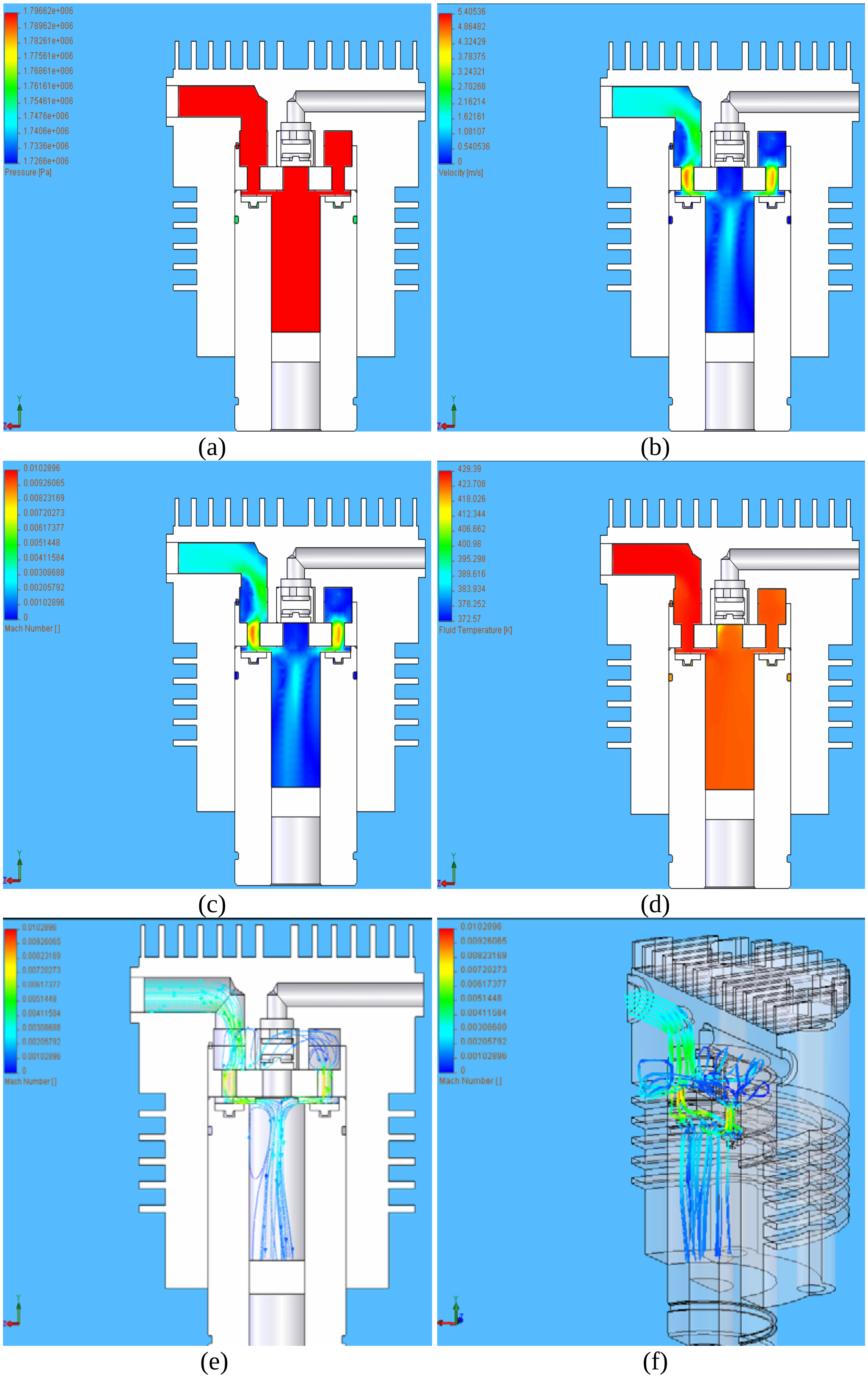


Figure 5.17 Flow analysis of cylinder 3 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Flow Trajectories

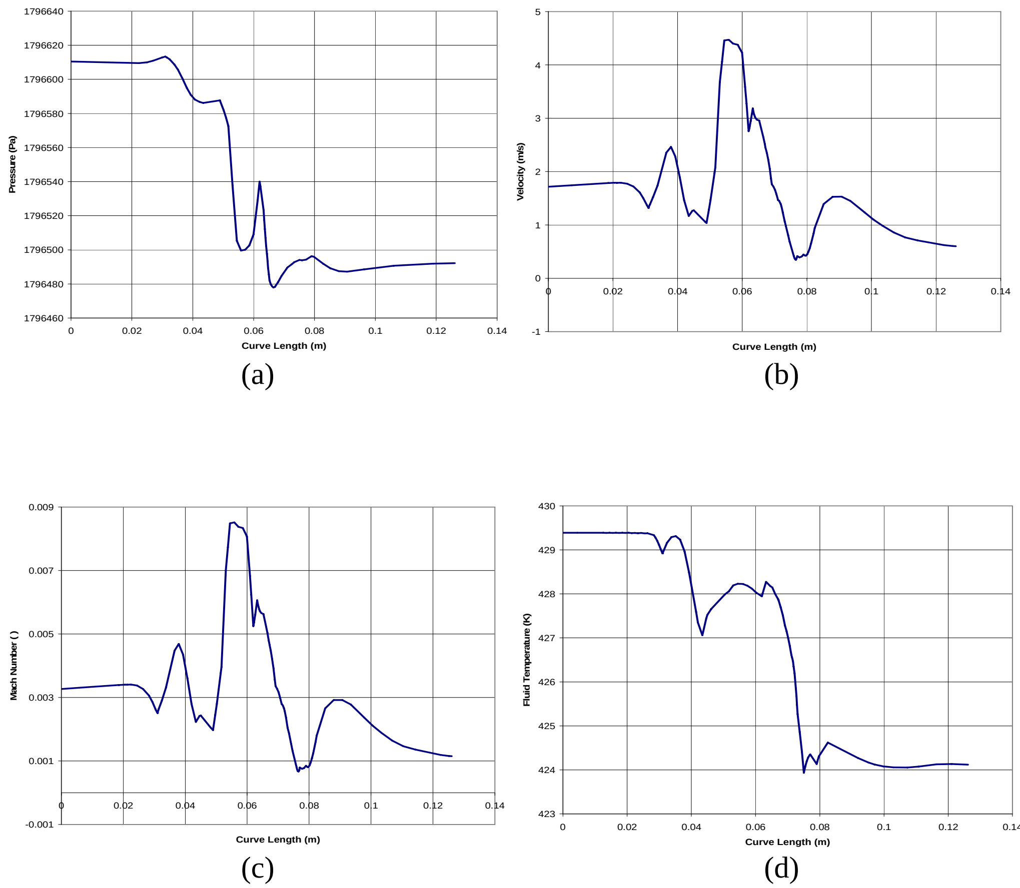


Figure 5.18 Graph flow analysis of cylinder 3 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature

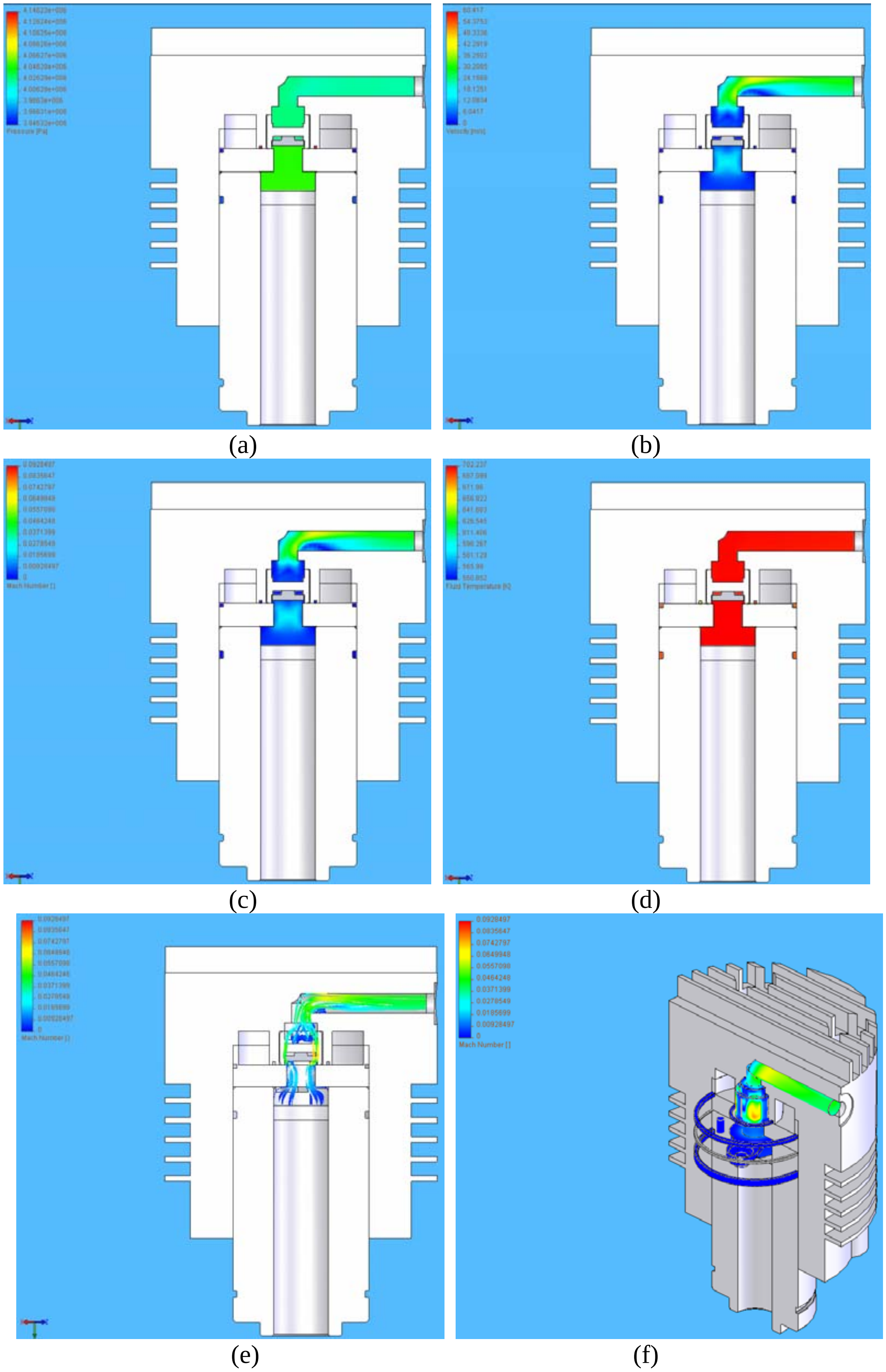


Figure 5.19 Flow analysis of cylinder 3 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Surface Plot

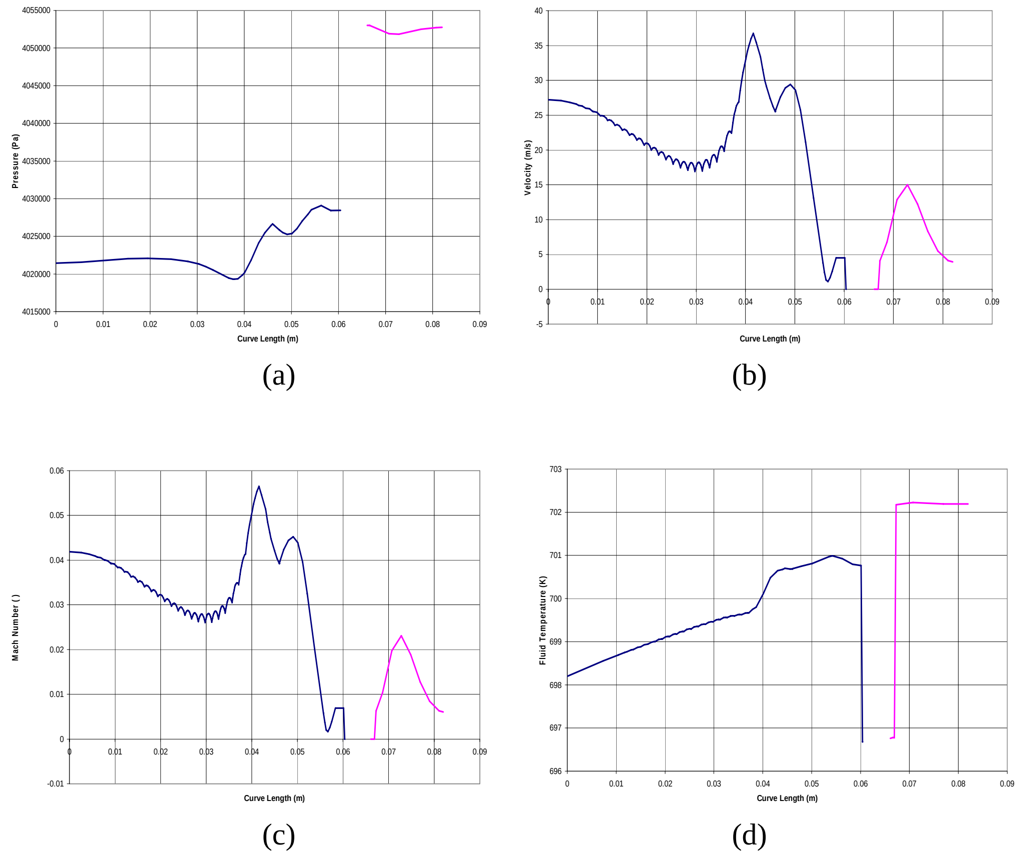
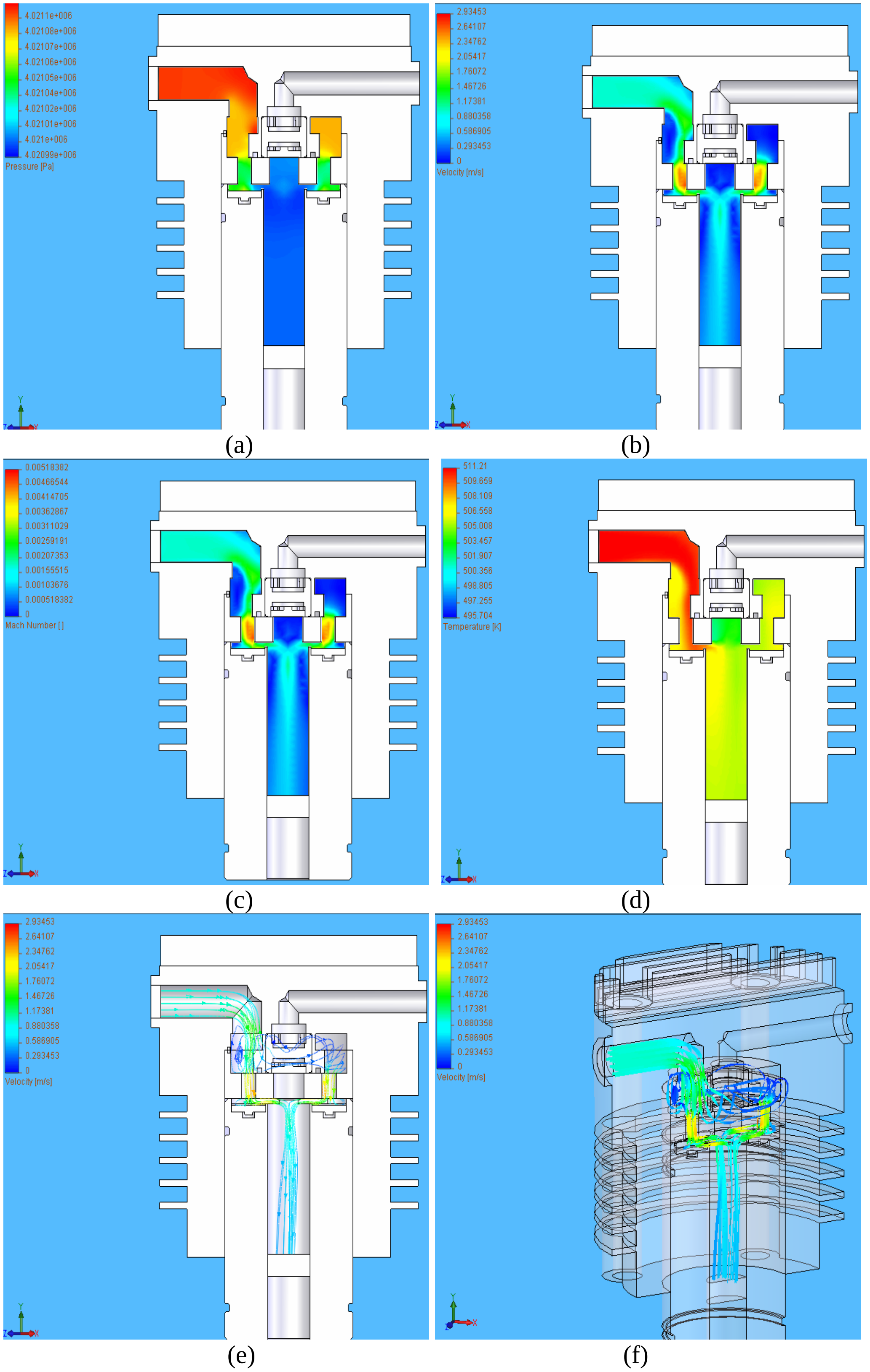
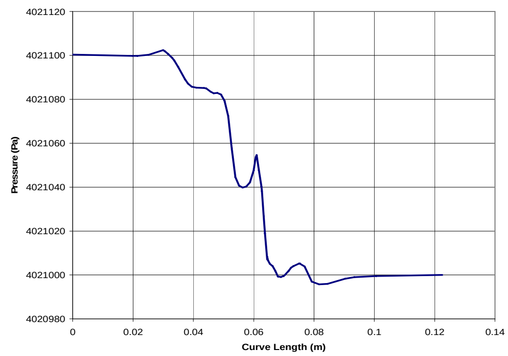


Figure 5.20 Graph flow analysis of cylinder 3 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature

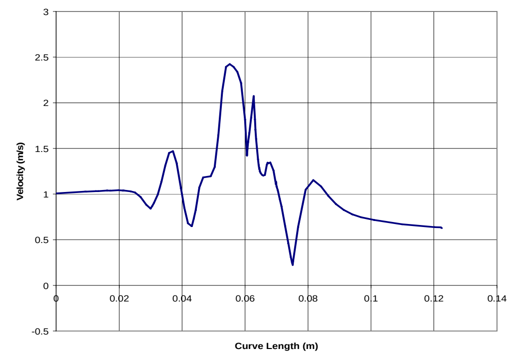


(a) (b) (c) (d) (e) (f)

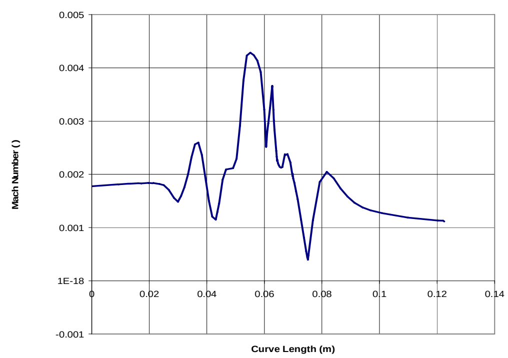
Figure 5.21 Flow analysis of cylinder 4 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Flow Trajectories



(a)

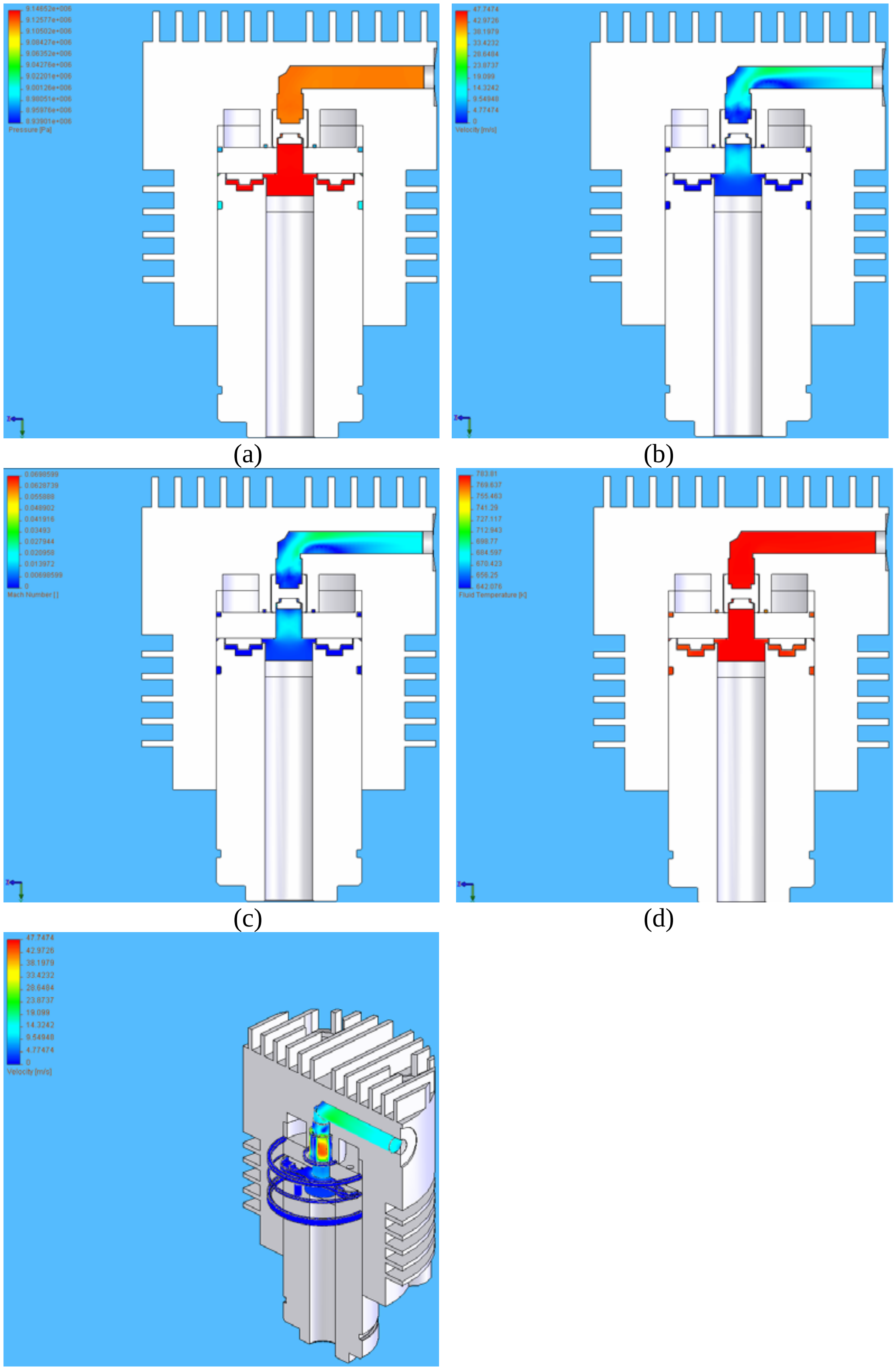


(b)



(c)

Figure 5.22 Graph flow analysis of cylinder 4 (suction)
(a). Pressure (b). Velocity (c). Mach number



(a) (b) (c) (d) (e)

Figure 5.23 Flow analysis of cylinder 4 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Isometric view Surface Plot

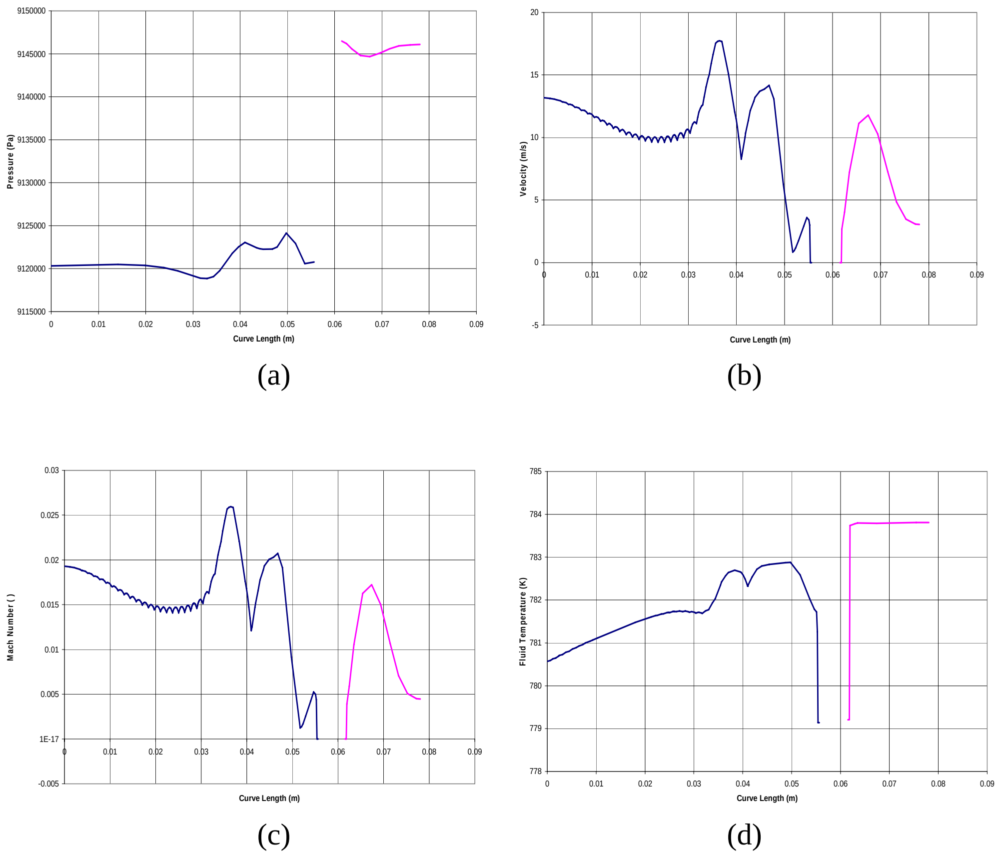


Figure 5.24 Graph flow analysis of cylinder 4 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature

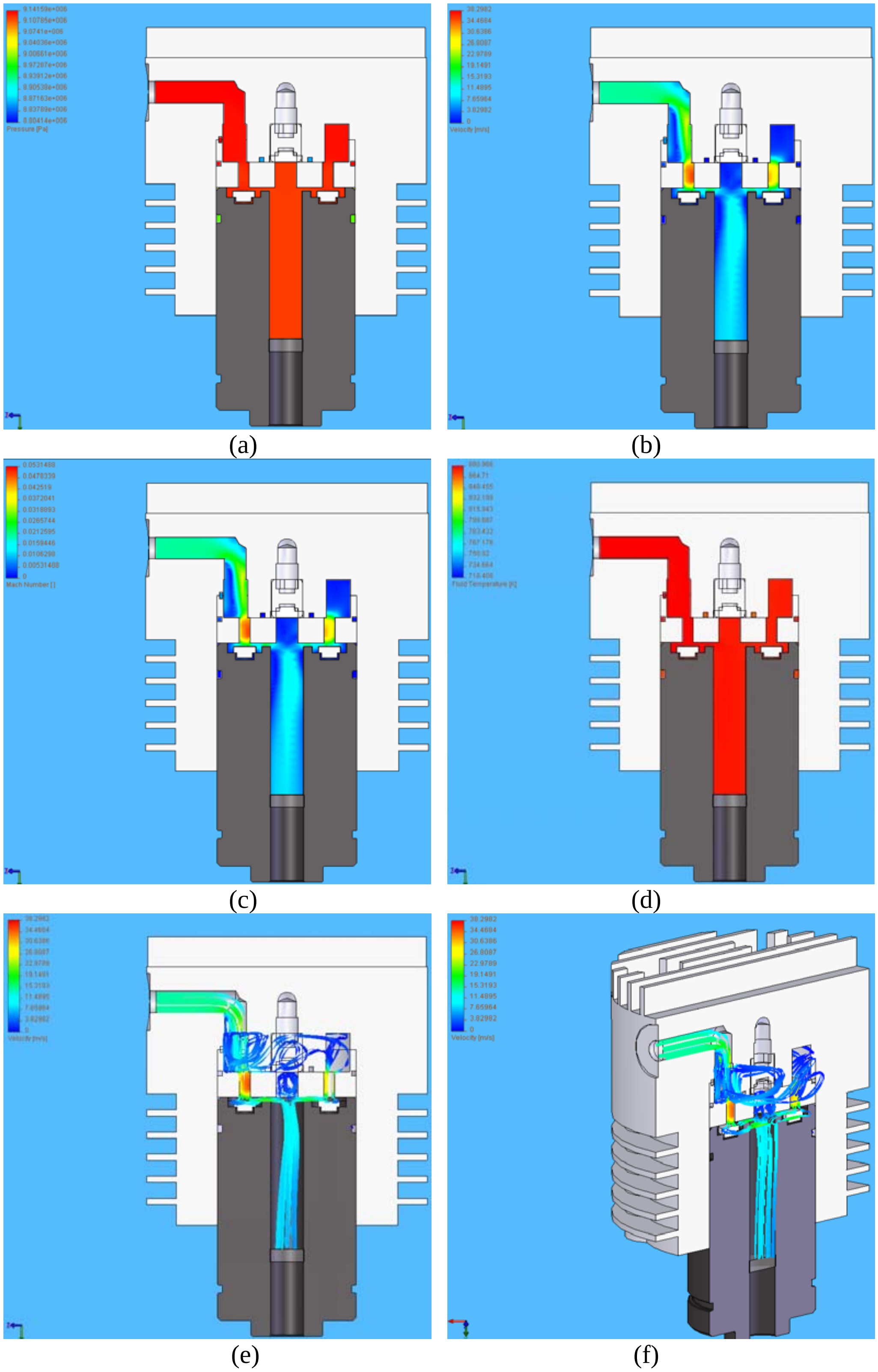
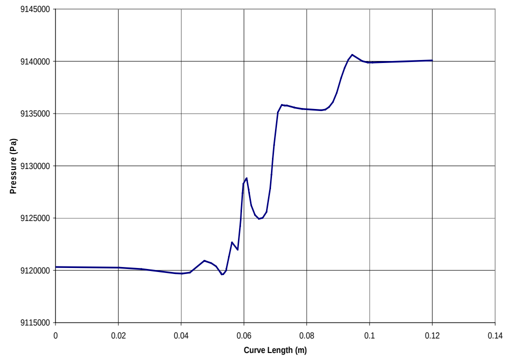
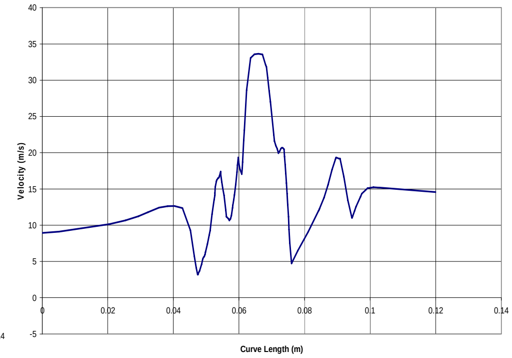


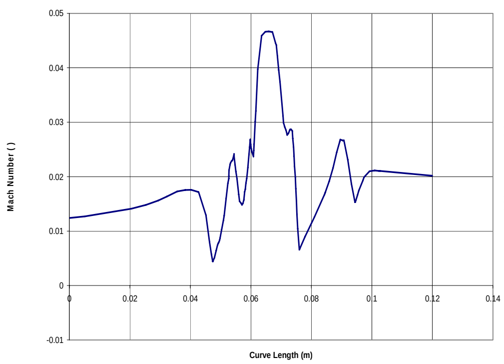
Figure 5.25 Flow analysis of cylinder 5 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Flow Trajectories



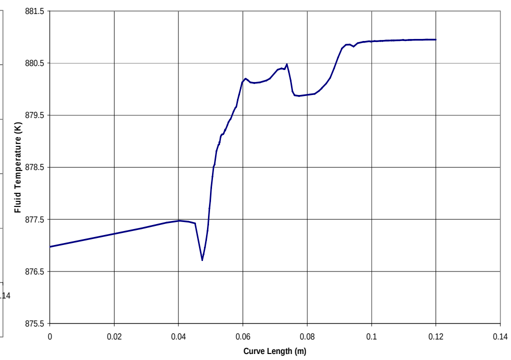
(a)



(b)

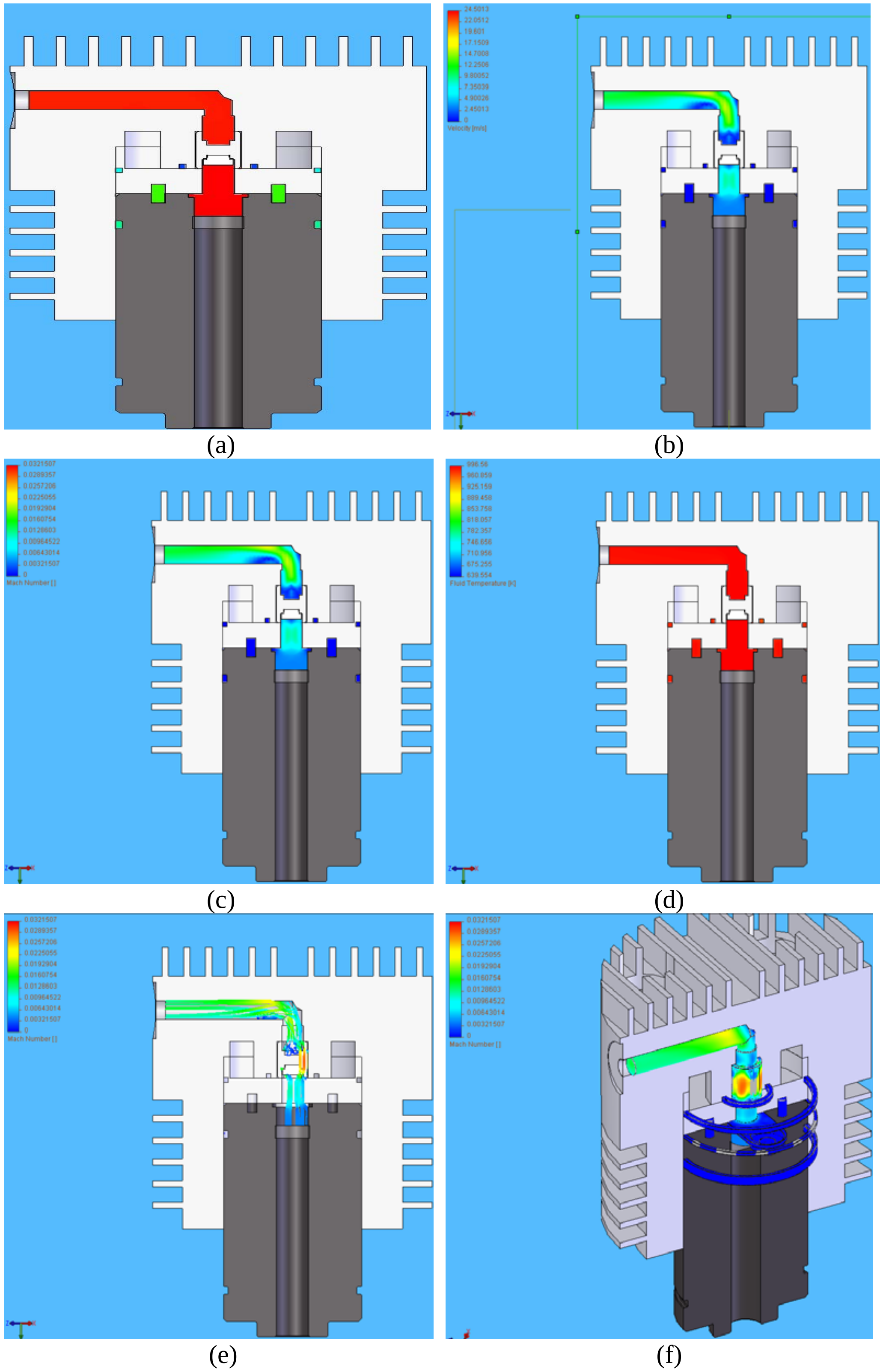


(c)



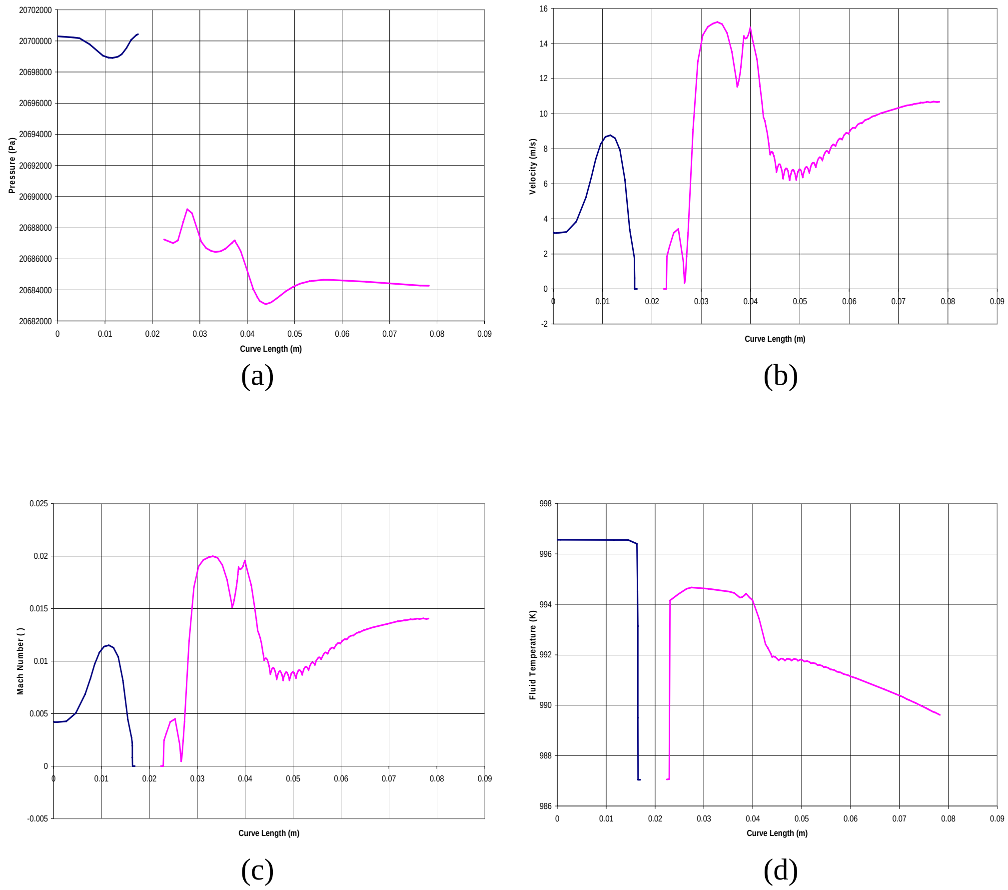
(d)

Figure 5.26 Graph flow analysis of cylinder 5 (suction)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature



(a) (b) (c) (d) (e) (f)

Figure 5.27 Flow analysis of cylinder 5 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature
(e). Flow Trajectories (f). Isometric view Surface Plot

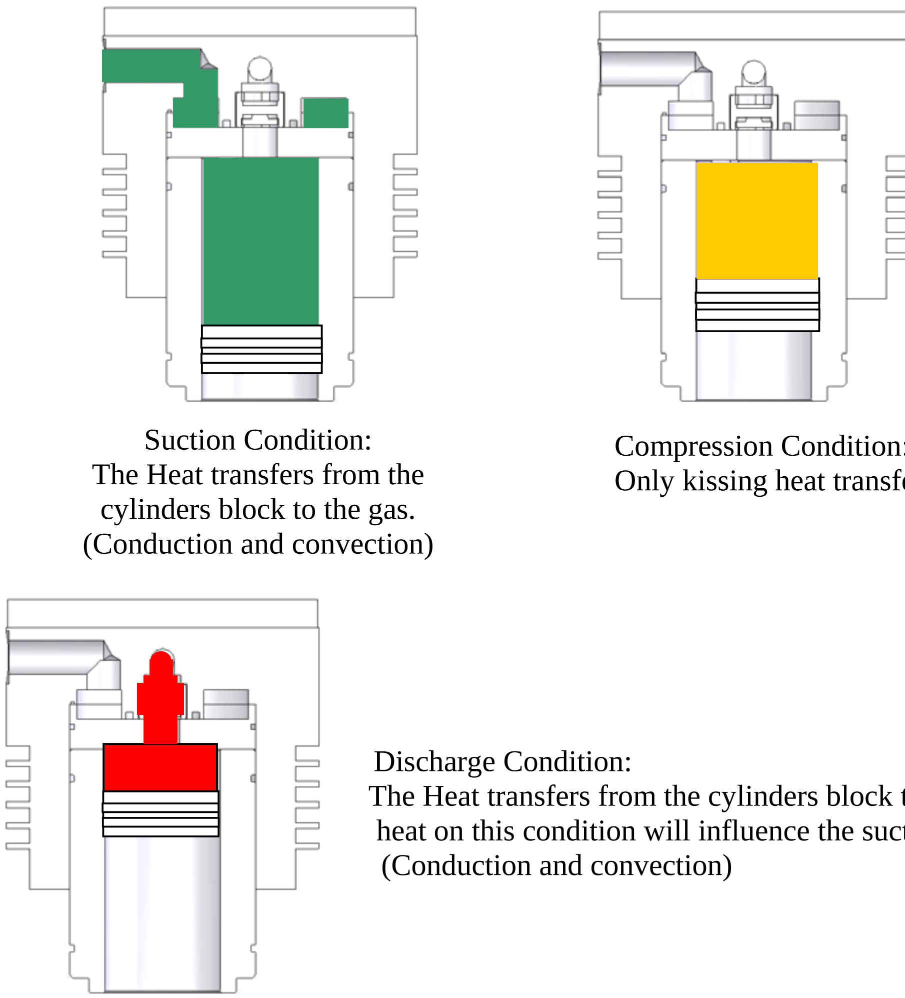


(a) (b) (c) (d)

Figure 5.28 Graph flow analysis of cylinder 5 (discharge)
(a). Pressure (b). Velocity (c). Mach number (d). Fluid temperature

5.3 Heat Transfer

In a low flow rate compressor the most important effect that reduces the flow capacity is the suction heat transfer, which in turn effects the performance of the compressor, Figure 5.29 shows the source of that heat. The suction heating in most compressors is pre-heating of the suction gas as it flows through the suction passage and the heating by the cylinder wall. This results in the increase of the suction temperature, which decreases the mass flow compressed, due to the decrease of gas density. The volumetric efficiency of the compressor is therefore reduced.



Suction Condition:
The Heat transfers from the
cylinders block to the gas.
(Conduction and convection)

Compression Condition:
Only kissing heat transfer

Discharge Condition:
The Heat transfers from the cylinders block to the gas. The heat on this condition will influence the suction condition (Conduction and convection)

Figure 5.29 Source of heat transfer

Heat from discharge to suction process will transfer from gas to surrounding through the wall of the cylinder block compressor. The amount of heat transfer to the suction pockets depends on the wall temperature of the compressor. However, the hottest part on the cylinder block is still lower than the hottest gas in the compressor and obviously the coldest part on the cylinder block will be higher than the coldest suction gas temperature. Heat started to transfer from gas to the cylinder wall when the compression begun.

5.3.1. Convection Heat Transfer

Convection occurs whenever a solid surface is in contact with a fluid at different temperature. The evaluation of the convection heat transfer rate can be estimated according to the Newton's law of cooling (cooling of a hot surface by cold fluid), which can be expressed by the following equations.

$$\dot{Q} = hA\left(t_s - t_f\right) \qquad 5.23$$

Where

A = surface area of heat transfer

t_s = mean surface temperature

t_f = average fluid temperature

h = convective heat transfer coefficient

The convection coefficient of "*h*" depends on the geometry of the surface, the fluid flow characteristic, and the fluid properties. It is assumed that the flow in the cylinder as in pipe and by using the standard pipe flow heat transfer correlations to determine the Nusselt number.

$$Nu = 0.23\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3} \qquad 5.24$$

and,

$$Nu = \frac{hD_h}{K_g} \qquad 5.25$$

Then,

$$h = \frac{NuK_g}{D_h} \qquad 5.26$$

Where:

R_e = Reynold's number

Pr = Prandtl number

K_g = gas thermal conductivity

D_h = hydraulic diameter

The flow condition between the gas and the cylinder block wall was assumed turbulent. The Reynold's number is given by:

$$\mathrm{Re} = \frac{\rho v D_h}{\mu} \qquad 5.27$$

Multiplying the above equation by the area of the channel "A_c"

$$\textit{The mass flow rate } \dot{m} = \textit{area} \times \textit{velocity} \times \textit{density}$$

$$\mathrm{Re} = \frac{A_c \rho v D_h}{A_c \mu} = \frac{\dot{m} D_h}{A_c \mu} \qquad 5.28$$

Rewriting the equation:

$$\mathrm{Re} = \frac{\dot{m} D_h}{A_c \mu} \qquad 5.29$$

The Prandtl number is given by:

$$\mathrm{Pr} = \frac{c_p \mu}{k} \qquad 5.30$$

Where:

c_p = specific heat at constant pressure

k = thermal conductivity of the gas

μ = absolute viscosity

The heat transfer into the gas in suction pocket occurs from the heated walls and can be evaluated using the following equations.

$$\dot{Q}_{suction\text{-}1} = \dot{m}\, cp(T_1 - T_L) \qquad 5.31$$

$$\dot{Q}_{suction\text{-}2} = A_{suction} h \left[\frac{(T_1 - T_L)}{\log \frac{T_{suc\text{-}wall} - T_L}{T_{suc\text{-}wall} - T_1}} \right] \tag{5.32}$$

As the gas enters the suction port, it will mix with the gas circulates over the compressor and the gas that enters the cylinder directly from the suction port of the compressor, the mixed mean state of gas occurs before the wall convection heat transfer in the suction process begin. This mixed mean state of the gas is at a temperature denoted as T_L.

The discharge convection heat transfer also can be evaluated by similar equation

$$\dot{Q}_{discharge\text{-}1} = \dot{m}\, cp_{dis} (T_{dis} - T_{out}) \tag{5.33}$$

$$\dot{Q}_{disc\text{-}2} = A_{discharge} h \left[\frac{(T_{dis} - T_{out})}{\log \frac{T_{disl} - T_{dis\text{-}wall}}{T_{out} - T_{dis\text{-}wall}}} \right] \tag{5.34}$$

5.3.2. The wall Heat Transfer

The wall heat transfer is the combination of the conduction and the kissing heat transfer, (which is described in 5.3.2.2)

The wall heat transfer = conduction heat transfer + kissing heat transfer

5.3.2.1 Conduction

Heat can be transferred through the large mass of flow inside and outside of the cylinder block wall. This is due to between hot and cold section of the cylinder block.

In this model, the mechanism of heat transfer is simply assumed as a radial conduction, whose lumped conductance could be estimated by assuming cylindrical block base. Conduction through a heavy metal of the cylinder base is modeled as radial conduction though a cylindrical thermal resistance. If an elemental ring was considered of radius r and thickness dr let the temperature of the inner surface of this ring be T and that of the other surface be *T+dT*. Apply Fourier's law of conduction to this element then,

$$\dot{Q} = -kA\frac{dT}{dr} \qquad 5.35$$

When substituting A in the above equation

A = area of heat transfer perpendicular to the direction of heat flow. *i.e.* surface area = $2\pi rL$, so:

$$\dot{Q} = -2kr\pi\frac{dT}{dr} \qquad 5.36$$

Rearrange the above equation and integrating it between the the limit:

$$\int_{r_1}^{r_2}\frac{dr}{r} = \frac{-2\pi Lk}{\dot{Q}}\int_{T_1}^{T_2} dT$$

$$\dot{Q} = \frac{2kr\pi L}{\ln\frac{r_2}{r_1}}\left(T_1 - T_2\right) \qquad 5.37$$

5.3.2.2 Kissing Heat Transfer

Figure 5.30 shows the contact point of the piston ring that rubs against the cylinder wall as the piston reciprocates.

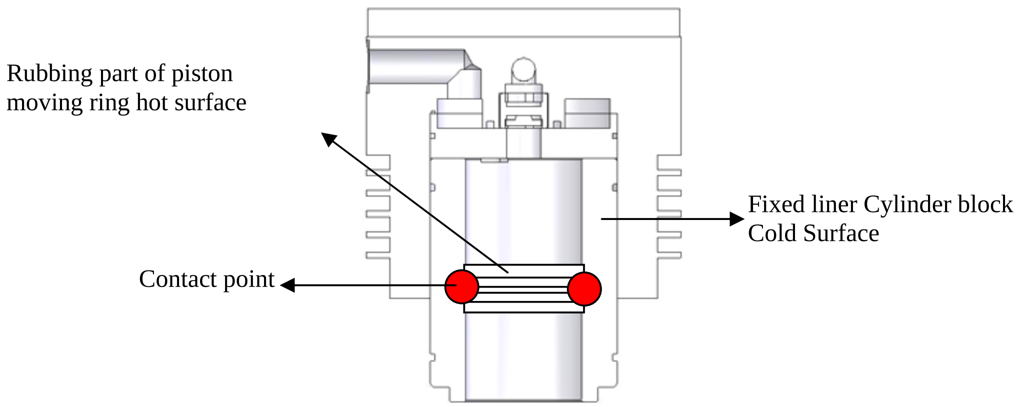


Figure 5.30 Contact "kissing" heat transfer

This transient touching contact of the cylinder wall with hot and cold piston ring as a mechanism of heat transfer within the reciprocating compressor is referred as the "kissing heat transfer". The estimated amount of kissing heat transfer depends on the time of contact between the piston ring and the cylinder wall. If the cylinder wall surface is perfectly smooth and there is no deformation on the geometry, the kissing heat transfer is very small or the value tends to become zero. Otherwise, the instantaneous heat flux can be expressed (Hamdy, 2005) by the following equation.

$$\dot{q} = \frac{k dT}{\sqrt{\pi \alpha t}} \qquad 5.38$$

Where:

k = thermal conductivity

dT = instantaneous temperature difference

α = thermal diffusivity of the cylinder material

t = time of contact

If the temperature of the hot part of the cylinder wall is T_h and the cold part of the piston ring is T_c, then the temperature of the piston will be that of an intermediate between the hot and cold part, $\frac{T_h + T_c}{2}$, then the value of the dT can be estimated as:

$$dT = T_h - T_m = \frac{T_h + T_c}{2} = \Delta T/2 \qquad 5.39$$

where:

$$\Delta T = T_h - T_c \qquad 5.40$$

The total energy transferred when the wall and the piston ring is in contact during the short time can be evaluated by integrating the instantaneous heat flux equation.

$$q = \int_0^t \dot{q}\,dt = \int_0^t \frac{kdT}{\sqrt{\pi\alpha t}}dt$$

$$q = \int_0^t \frac{kdT}{\sqrt{\pi\alpha t}} = \int_0^t \frac{2k\Delta T}{2\sqrt{\pi\alpha}}t^{-0.5}$$

$$q = \frac{k\Delta T}{\sqrt{\pi\alpha}}t^{1/2} \qquad 5.41$$

The average rate of kissing heat transfer between ring piston and cylinder wall is:

$$\dot{q} = \frac{1}{\tau}\frac{k\Delta T}{\sqrt{\pi\alpha}}t^{1/2} \qquad 5.42$$

Where

τ = time of the one cycle or revolution of the compressor = $\frac{1}{rps}$

rps = revolution/second

If it is assumed that the contact for $\frac{\theta}{360}$ of the time period of the crank rotation, then the contact time can be: $t = \frac{\tau\theta}{360}$

$$\dot{q} = \frac{1}{\tau}\frac{k\Delta T}{\sqrt{\pi\alpha}}\left[\frac{\tau\theta}{360}\right]^{1/2} \qquad 5.43$$

k is the thermal conductivity for gas and α is the thermal diffusivity for aluminum.

5.3.3. Temperature Estimation

The previous section discussed the wall heat transfer and the convection heat transfer at the suction and the discharge inside symmetrical wobble plate compressor. In the following section, five different equations had been developed and solved by iteration method in order to estimate the suction wall temperature, the discharge temperature, the discharge wall temperature, and the temperatures at beginning and at the end of the suction process.

5.3.3.1 The Suction Start Temperature

The gas from the suction port T_{in} mixed with the leaked gas circulating over the compressor until, it reaches the final temperature T_L as shown in Figure 5.31. The assumed proportion of gas flow into the inlet port of the compressor can be expressed by the following formula (G.H. Lee 2002), and x was chosen as 0.5.

$$T_L = xT_{in} + (1 + x)T_1 \qquad 5.44$$

5.3.3.2 The Compression Inlet Temperature

As described in section 5.3.1 heat transfer to the gas in the suction pocket can be evaluated by two methods as follows:

$$\dot{Q}_{\text{suction-1}} = \dot{m}\,cp\left(T_1 - T_L\right) \qquad 5.45$$

And

$$\dot{Q}_{\text{suction-2}} = A_{\text{suction}} h \left[\frac{\left(T_1 - T_L\right)}{\log \dfrac{T_{\text{suc-wall}} - T_L}{T_{\text{suc-wall}} - T_1}} \right] \qquad 5.46$$

When $\dot{Q}_{\text{suction-1}} = \dot{Q}_{\text{suction-2}}$, then,

$$\dot{m}\,cp\left(T_1 - T_L\right) = A_{\text{suction}} h \left[\frac{\left(T_1 - T_L\right)}{\log \dfrac{T_{\text{suc-wall}} - T_L}{T_{\text{suc-wall}} - T_1}} \right]$$

$$T_1 = \left[\frac{A_{\text{suction}} h T_1 - A_{\text{suction}} h T_L}{\log \dfrac{T_{\text{suc-wall}} - T_L}{T_{\text{suc-wall}} - T_1}} + \dot{m}\,cp T_L \right] \times \frac{1}{\dot{m}\,cp} \qquad 5.47$$

5.3.3.3 The Suction Wall Temperature

The model assumes that the heat transferred into suction chamber through the wall is as discussed in section 5.4.2, and can be expressed by the following equations:

$$\dot{Q}_{\text{suction-1}} = \dot{Q}_{\text{wall}} \qquad 5.48$$

$$\dot{Q} = \frac{2k\pi L}{\ln \frac{r_{suc}}{r_{dis}}} \left(T_{dis\text{-}wall} - T_{suc\text{-}wall} \right)$$

$$\dot{q}_{kiss} = \frac{1}{\tau} \frac{k \Delta T}{\sqrt{\pi \alpha}} \left[\frac{\tau \theta}{360} \right]^{1/2}$$

$$\dot{Q}_{kiss} = A_{kiss} \frac{1}{\tau} \frac{k \left(T_{dis\text{-}wall} - T_{suc\text{-}wall} \right)}{\sqrt{\pi \alpha}} \left[\frac{\tau \theta}{360} \right]^{1/2}$$

$$\dot{Q}_{wal1} = \dot{Q}_{con} + \dot{Q}_{kiss}$$

$$\dot{Q}_{wall} = \frac{2k\pi L}{\ln \frac{r_{suc}}{r_{dis}}} \left(T_{dis\text{-}wall} - T_{suc\text{-}wall} \right) + A_{kiss} \frac{1}{\tau} \frac{k}{\sqrt{\pi \alpha}} \left[\frac{\tau \theta}{360} \right]^{1/2} \left(T_{dis\text{-}wall} - T_{suc\text{-}wall} \right)$$

$$\dot{Q}_{wall} = \left(T_{dis\text{-}wall} - T_{suc\text{-}wall} \right) \left[\frac{2k\pi L}{\ln \frac{r_{suc}}{r_{dis}}} + A_{kiss} \frac{1}{\tau} \frac{k}{\sqrt{\pi \alpha}} \left[\frac{\tau \theta}{360} \right]^{1/2} \right]$$

$$\dot{Q}_{suction\text{-}1} = \dot{m}\, cp_{suc} \left(T_1 - T_L \right)$$

$$\dot{Q}_{suction\text{-}1} = \dot{Q}_{wall}$$

$$\dot{m}\, cp_{suc} \left(T_1 - T_L \right) = \left(T_{dis\text{-}wall} - T_{suc\text{-}wall} \right) \left[\frac{2k\pi L}{\ln \frac{r_{suc}}{r_{dis}}} + A_{kiss} \frac{1}{\tau} \frac{k}{\sqrt{\pi \alpha}} \left[\frac{\tau \theta}{360} \right]^{1/2} \right]$$

$$\left(T_{dis\text{-}wall} - T_{suc\text{-}wall} \right) = \frac{\dot{m}\, cp_{suc} \left(T_1 - T_L \right)}{\left[\frac{2k\pi L}{\ln \frac{r_{suc}}{r_{dis}}} + A_{kiss} \frac{1}{\tau} \frac{k}{\sqrt{\pi \alpha}} \left[\frac{\tau \theta}{360} \right]^{1/2} \right]}$$

$$T_{suc\text{-}wall} = \frac{\dot{m}\, cp_{suc} \left(T_1 - T_L \right)}{\left[\frac{2k\pi L}{\ln \frac{r_{suc}}{r_{dis}}} + A_{kiss} \frac{1}{\tau} \frac{k}{\sqrt{\pi \alpha}} \left[\frac{\tau \theta}{360} \right]^{1/2} \right]} + T_{dis\text{-}wall} \qquad 5.49$$

5.3.3.4 The Wall Temperature after Discharge

The model assumes that the heat transferred from the heated wall (at the end of discharge) to the freshly induced gas is as discussed in section 5.4.1. The temperature after discharge can be estimated by the following equations:

$$\dot{Q}_{dis} = \dot{Q}_{wall}$$

$$\left(T_{dis\text{-}wall} - T_{suc\text{-}wall}\right)\left[\frac{2k\pi L}{\ln\frac{r_{suc}}{r_{dis}}} + A_{kiss}\frac{1}{\tau}\frac{k}{\sqrt{\pi\alpha}}\left[\frac{\tau\theta}{360}\right]^{1/2}\right] = \dot{m}\,cp_{dis}\left(T_{dis} - T_{out}\right)$$

$$T_{dis\text{-}wall} = \frac{\dot{m}\,cp_{dis}\left(T_{dis} - T_{out}\right)}{\left[\frac{2k\pi L}{\ln\frac{r_{suc}}{r_{dis}}} + A_{kiss}\frac{1}{\tau}\frac{k}{\sqrt{\pi\alpha}}\left[\frac{\tau\theta}{360}\right]^{1/2}\right]} + T_{suc\text{-}wall} \qquad 5.50$$

5.3.3.5 The Discharge Gas Temperature

The model assumes that the heat transferred from discharge chamber equal to the suction heat transfer and can be expressed by the following equations:

$$\dot{Q}_{suc} = \dot{Q}_{dis}$$

$$\dot{m}_{dis}\,cp_{dis}\left(T_{dis} - T_{out}\right) = \dot{m}_{suc}\,cp_{suc}\left(T_1 - T_L\right)$$

$$\dot{m}_{dis}\,cp_{dis}T_{dis} - \dot{m}_{dis}\,cp_{dis}T_{out} = \dot{m}_{suc}\,cp_{suc}\left(T_1 - T_L\right)$$

$$T_{out} = \frac{\dot{m}_{dis}\,cp_{dis}T_{dis} - \dot{m}_{suc}\,cp_{suc}\left(T_1 - T_L\right)}{\dot{m}_{dis}\,cp_{dis}} \qquad 5.51$$

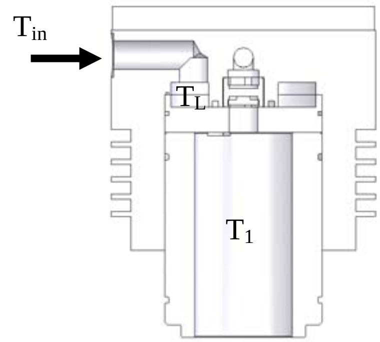


Figure 5.31 Mixing area

The equations 5.19 through 5.23 are solved simultaneously to estimate the suction wall temperature, the discharge gas temperature, the discharge wall temperature, the temperature of gas entering the compression chamber at the end of suction, and the temperature of the gas at the beginning of the suction process T_L.

The thermodynamic analyses of the gas within the cylinder block begin by assuming initial suction conditions. The analyses were completed through one compressor cycle to determine the pressure, temperature, and the mass flow rate within the compressor pocket. The suction and discharge wall temperatures should be estimated to predict the compression starting temperature.

5.3.4. Discussion on Heat Transfer and Simulation

The suction temperature of gas for cylinder 5, however, is very high reaching 604^0C in the cylinder. It is hoped that fins made on the outside of cylinder block 5 will enhance dissipation of heat to the atmosphere. The aftercooler installed should be able to decrease the temperature of the gas before it is storage.

Looking at the results of simulation when the gas is being discharged, it seems that the compressor is performing well and the final design of all cylinder and the valves acceptable. Apart from the heat generated that can cause the temperature of the cylinder 5 to be very high (about 663.5^{0}C) the flow through the discharge valve is still sub-sonic. Pressure difference required to open the discharge valve in this cylinder is about 206 bar.

Based on the acceptable simulated performance of cylinder 1 during suction and cylinder 5 during discharge, the design of the cylinder 2, 3, and 4 respectively are equally acceptable. The simulated results for these cylinders are evident to this conclusion.

Table 5.1 Material of cylinder accessories

No	Part Name	Material	Mass	Volume
1	Cylinder Block 1	Aluminum alloy 6061	1.27 kg	0.00047 m^3
2	Liner	Gray cast iron	1.08 kg	0.00015 m^3
3	Valve suction plate 1	Aluminum alloy 6061	0.00052 kg	1.91564e-007 m^3
4	Valve suction plate 2	Aluminum alloy 6061	0.00052 kg	1.91564e-007 m^3
5	Valve	Aluminum alloy 6061	0.085 kg	3.15131e-005 m^3
6	Valve plate	Aluminum alloy 6061	0.00069 kg	2.55442e-007 m^3
7	Valve spring sit	Aluminum alloy 6061	0.0029 kg	1.06523e-006 m^3

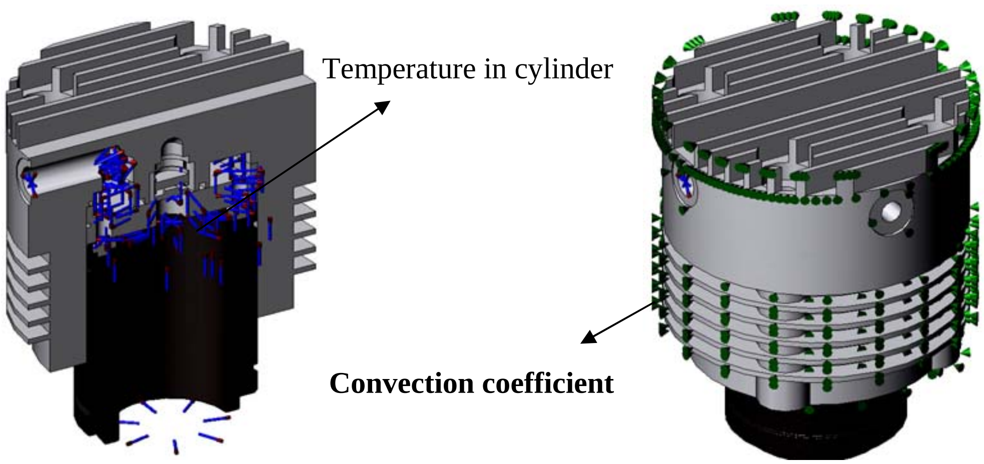


Figure 5.32 Boundary condition of simulation

Table 5.1 shows of list material of cylinder accessories and Figure 5.32 shows boundry conditions for heat transfer simulation from gas to cylinder block. Contact between parts with other in conditioning of touching faces - Bonded.

Cylinder wall temperature equal to gas temperature that is 303 K and convection coefficient is 25 W/(m2K).

Table 52 shows the simulation results, characteristic temperature with minimum of 3029 K at node 3219 and maximum 303 K at node 161. Maximum and minimum location can show in table 5.2. Table 5.3 and 5.4 showns properties of materials aluminum and gray cast iron used part of cylinder block.

Table 5.2 Thermal result of cylinder block

Name	Type	Min	Location	Max	Location
Plot1	TEMP: Nodal temperature	302.9 K Node: 3219	(-74.7 mm, -124.9 mm, -3.2 mm)	303 Kelvin Node: 161	(-21.9 mm, -74.4 mm, -31.3 mm)

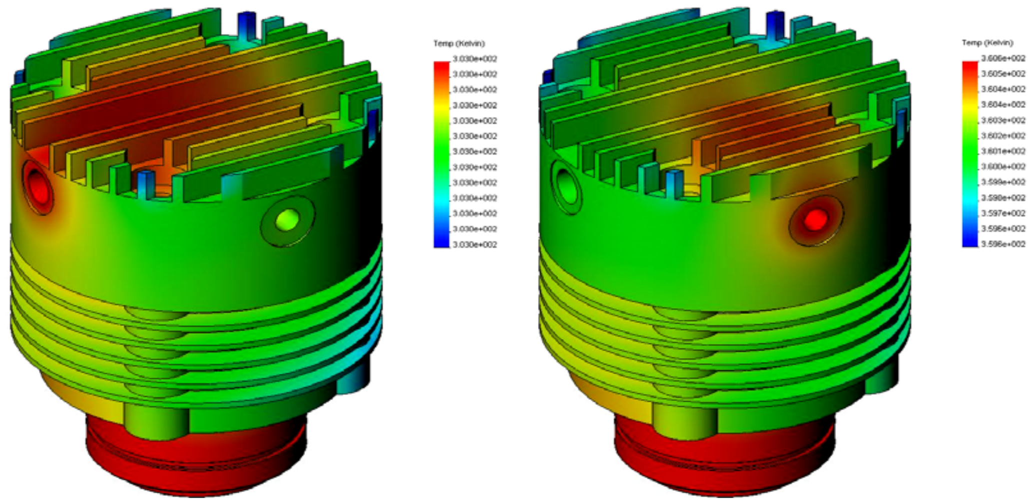
Table 5.3 Properties of aluminum alloy 6061

Property Name	**Value**
Elastic modulus	6.9e+010 N/m^2
Poisson's ratio	0.33
Shear modulus	2.6e+010 N/m^2
Thermal expansion coefficient	2.4e-005 /K
Mass density	2700 kg/m^3
Thermal conductivity	170 W/(m.K)
Specific heat	1300 J/(kg.K)
Tensile strength	1.2408e+008 N/m^2
Yield strength	5.5149e+007 N/m^2

Table 5.4 Properties of gray cast iron

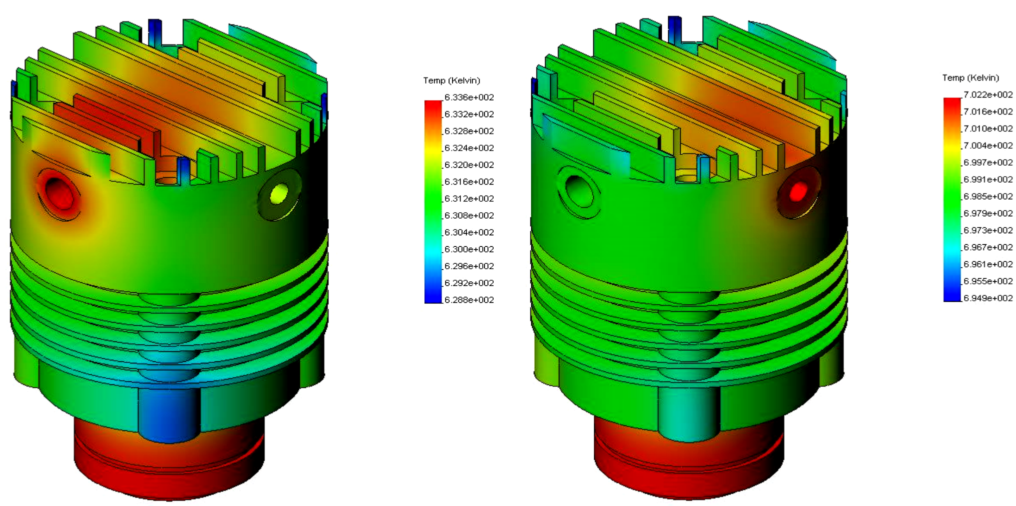
Property Name	**Value**
Elastic modulus	6.6178e+010 N/m^2
Poisson's ratio	0.27
Shear modulus	5e+010 N/m^2
Thermal expansion coefficient	1.2e-005 /K
Mass density	7200 kg/m^3
Thermal conductivity	45 W/(m.K)
Specific heat	510 J/(kg.K)
Tensile strength	1.5166e+008 N/m^2
Compressive strength	5.7217e+008 N/m^2

Result of simulations:



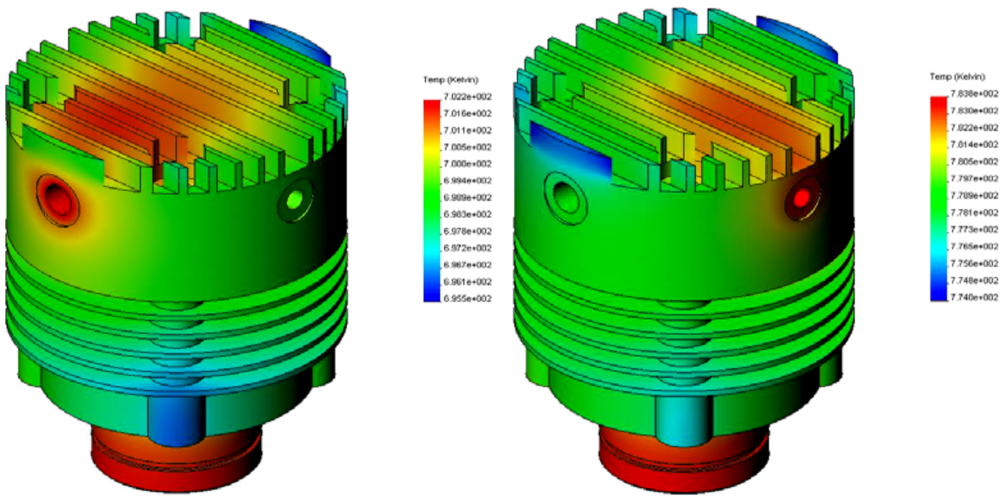
(a) (b)

Figure 5.33 Heat transfer analysis of cylinder 1
(a). Suction (b). Discharge



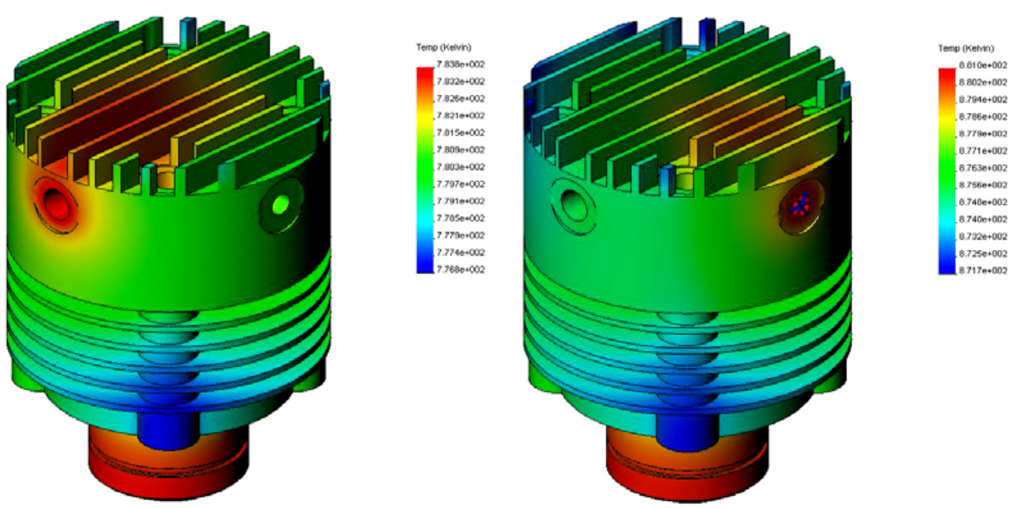
(a) (b)

Figure 5.34 Heat transfer analysis of cylinder 2
(a). Suction (b). Discharge



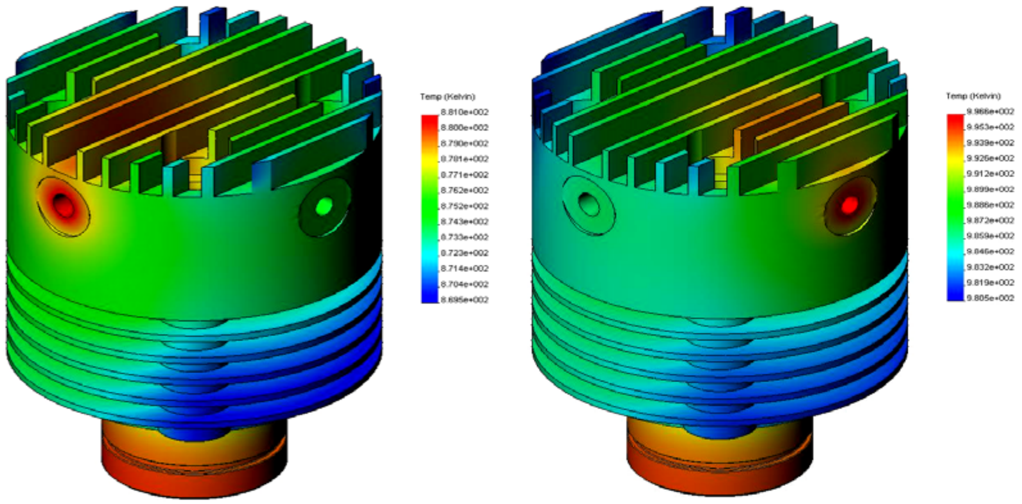
(a) (b)

Figure 5.35 Heat transfer analysis of cylinder 3
(a). Suction (b). Discharge



(a) (b)

Figure 5.36 Heat transfer analysis of cylinder 4
(a). Suction (b). Discharge



(a) (b)

Figure 5.37 Heat transfer analysis of cylinder 5
(a). Suction (b). Discharge

5.4 Discussion of Thermodynamic Analysis

This part discusses the analytical results of the thermodynamic calculation that had been done. The results are as shown in Figures 5.38 and 5.39. Suction, compression and discharge stroke of all five stages can be seen clearly based on the variation of pressures as shown in Figures 5.38 and 5.39. With a discharge pressure of a bout 206 bar and a suction pressure of a bout 3.45 bar given an optimum pressure ratio $\left(r = \left(\frac{p_6}{p_1} \right)^{\frac{1}{5}} \right)$ of about 2.689. if n is number of stages and for each stage the discharge pressure is P_{n+1} x 2.689. Therefore P_2=7.818 bar, P_3=17.732 bar, P_4=40.214 bar, P_5=91.203 bar, P_6=206.843 bar. The values from the graphs seem to agree with these calculated values, respectively.

Figure 5.38 shows the relationship between the shaft angle of rotation with the pressure in the cylinder block. This shaft angle of rotation determines the position of the piston during the suction, compression and discharge of the gas. In stages 5 for example the compression process starts at the angle of shaft rotation of 120^0 and discharge at 200^0. While in the stages 1 the compression process starts at 200^0 and discharge at 260^0.

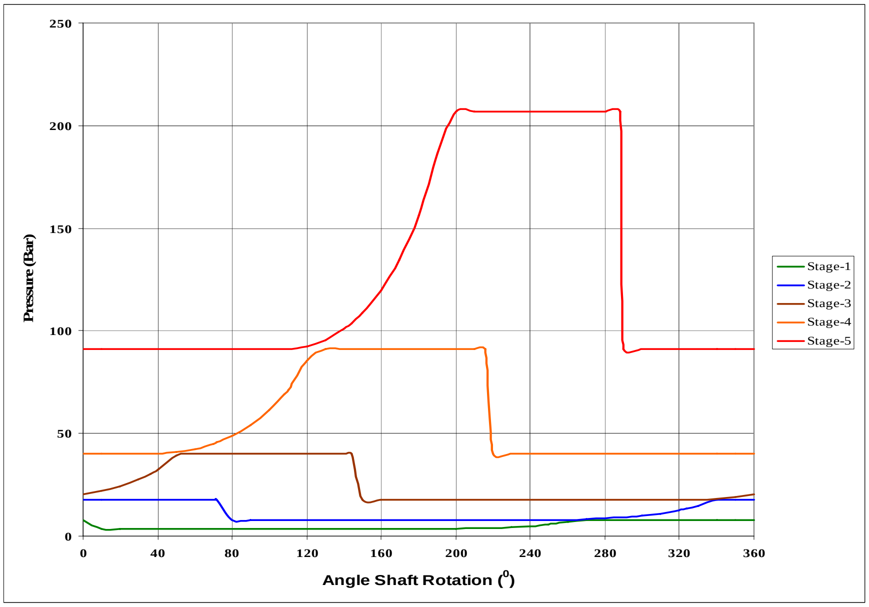


Figure 5.38 The variation pressure with every angle shaft rotation

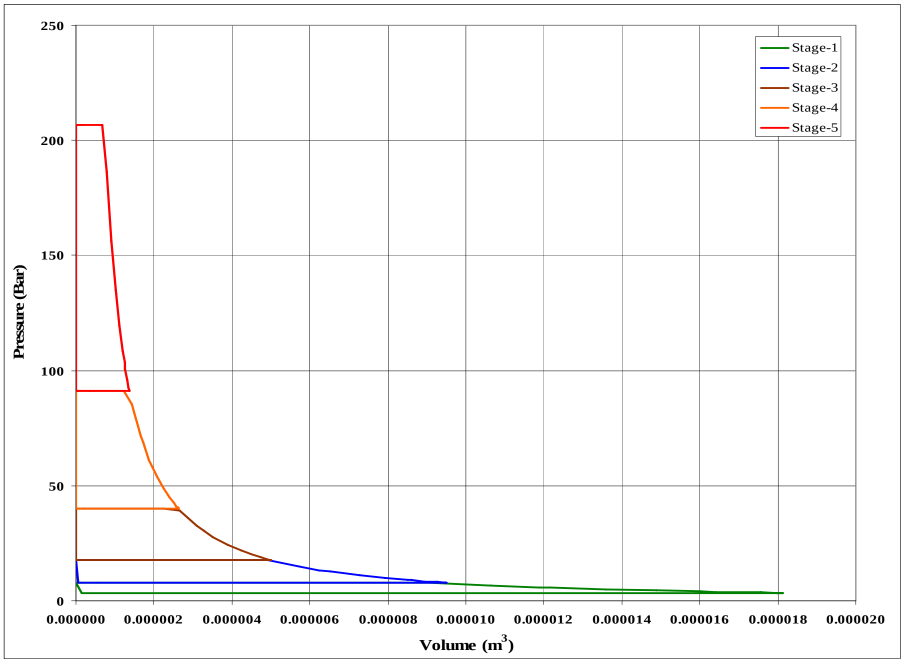


Figure 5.39 P-V diagram of compressor

CHAPTER 6

EXPERIMENTAL, RESULT AND DISCUSSION

6.1 Introduction

This chapter discusses the set-up of an experimental rig, the experimental procedure and the test results. The rig is specially design to test our new symmetrical wobble plate compressor prototype. The data acquisition “DAQ” system was incorporated in the rig to record all measurements.

6.2 Experimental Set Up

The schematic diagram of the apparatus is shown in Figure 6.1 and the complete experimental rig is shown in Figure 6.2. Figure 6.3 to Figure 6.14 show the respective parts of the rig. The compressor was driven by a motor of 50 Hz, 37 Kw. An inverter was used to control the motor speed. The pressure of air as it flow through the compressor was measured by 20 pressure sensors installed at different appropriate locations at which were also installed so temperature sensors (thermocouple). The pressure and temperature were measured across each stage of compression:

- 10 points at the suction pressure side
- 10 points at the discharge pressure side.

Both pressure and the temperature of the air were measured at a common point on the discharge and suction ports respectively. The suction pressure of the air that entered the compressor was 3 bar. A standard air compressor was used to supply air at 14 and regulated to 1-3 bar. A flow meter was used to measure the flow rate of air. The data acquisition (DAQ) system recorded all measured pressure and temperature readings.

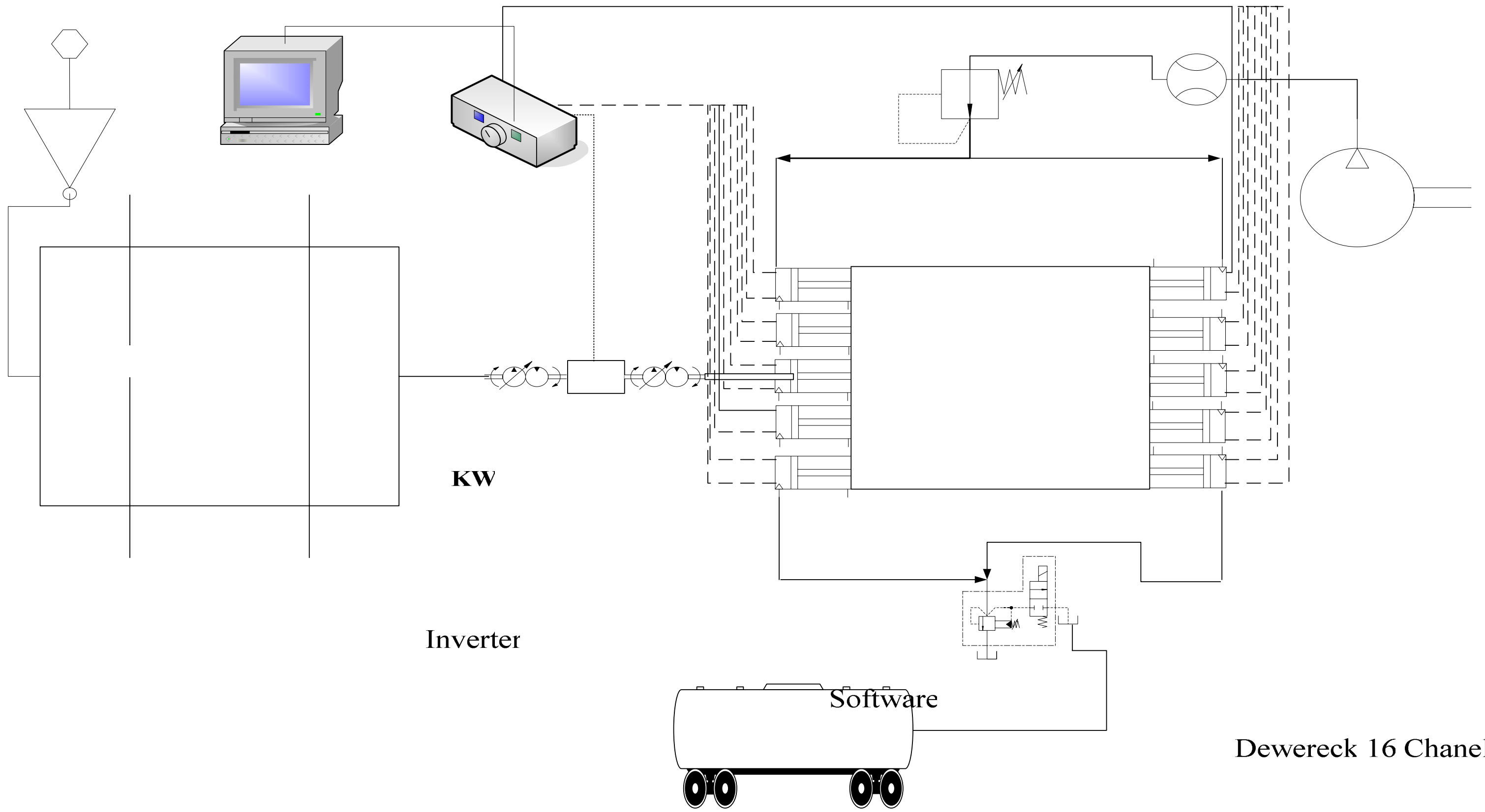


Figure 6.1 The experimental set-up

Electric Motor

Torque
Meter

Direct Coupling Direct Coupling

Figure 6.2 General rig assembly

Figure 6.3 Inverter

Figure 6.4 Electric motor

Figure 6.5 Rubber coupling (direct coupling)

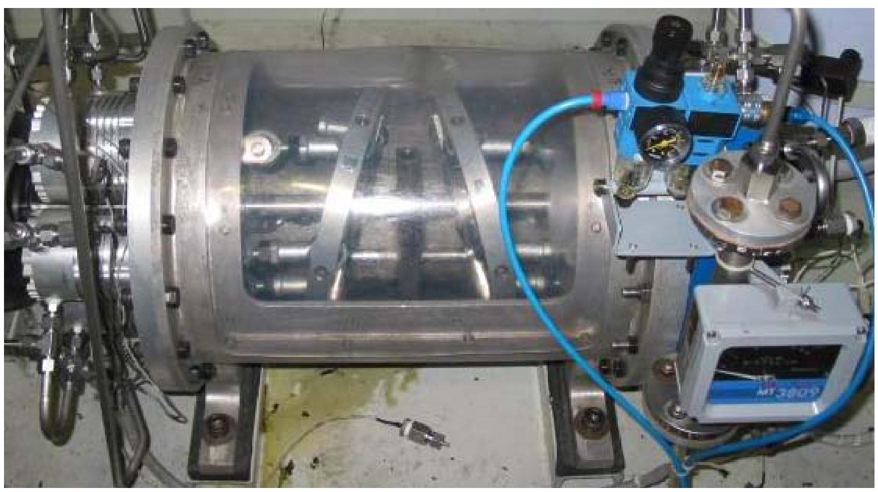


Figure 6.6 Symmetrical wobble plate mechanism

Figure 6.7 Data acquisition system

Figure 6.8 Air compressor

Figure 6.9 Flow meter

Figure 6.10 Pressure regulator

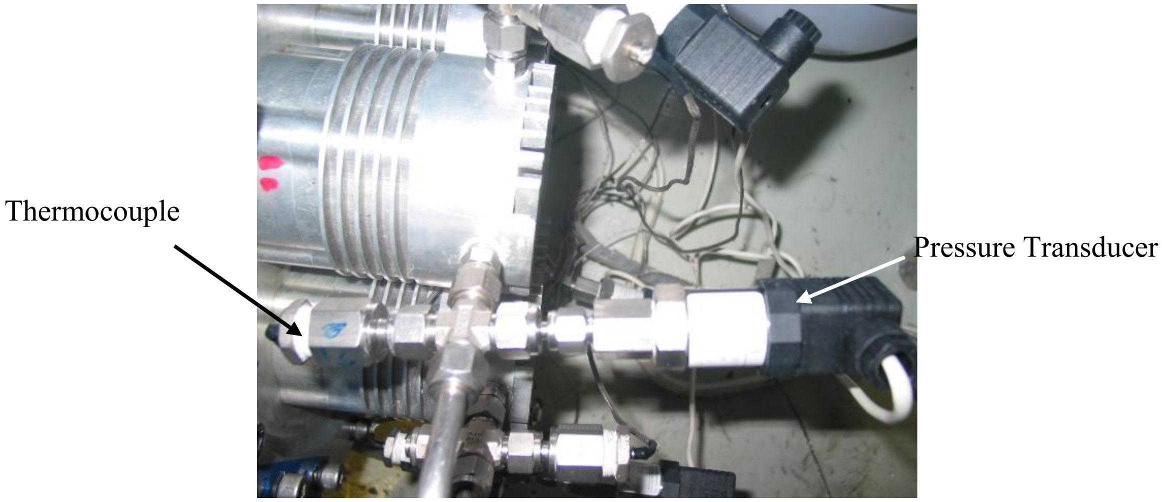


Figure 6.11 Pressure transducer & thermocouple

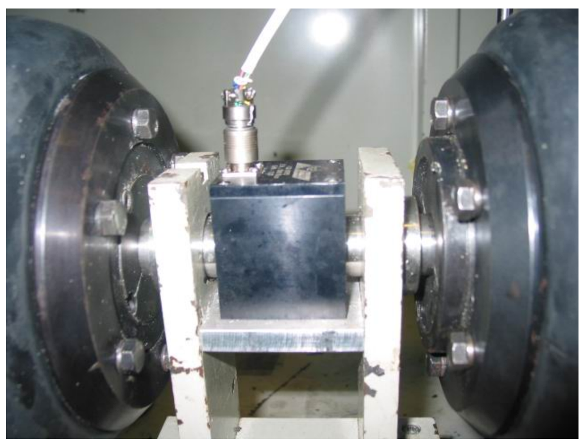


Figure 6.12 torque transducer

Figure 6.13 Relief valve

Figure 6.14 Storage tank

6.2.1. Data Acquisition "DAQ" System

The DAQ system setting as shown in Figure 6.15 consists of transducers, signal conditioner or signal amplifier, DAQ hardware, and software.

Transducers sense change of condition and convert the changes into electric signals to the DAQ system. Such sensors are the thermocouples, pressure transducers and torque transducers. In each case, the electric signal is proportional to the change in physical parameter. The DAQ received the signals from the 21 thermocouples to give temperature reading, and 5 piezoelectric pressure transducers, to give pressure reading.

The electric signals generated by the transducers (thermocouple or the pressure sensor) must be optimized for input range of the DAQ board. Signal conditioning accessories can amplify low-level signals, and than isolate and filter them for more accurate measurements. The low-level signals should be amplified to increase the resolution and reduce noise interference. The temperature thermocouples are connected to 3 CAL-PAD-CB8-K-P modules. Each module is able to handle 8 input channels by using a connector block with 21 k type thermocouple connectors. The pressure signals can be measured by using the piezoelectric pressure transducer, which acts on the diaphragm, and converts the pressure into proportional force.

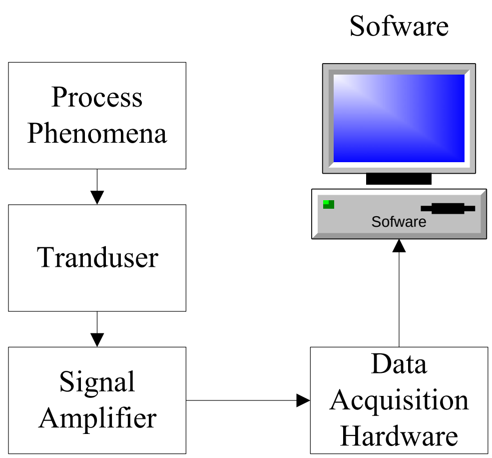


Figure 6.15 Data acquisition system "DAQ"

This force is conveyed onto the quartz, which under loading condition will yield an electrostatic charge. An electrode picks up this negative charge and passes it to a plug, after which the connected charge amplifier converts and optimizes it into a positive voltage. The five DEWETRON change amplifier modules were used to amplify the pressure signal in the compressor. Another similar type had been used in the experiments is a DAQP-BRIDGE-B. The whole modules were assembled in the DEWERACK-16 Channel rack housing.

The data acquisition hardware comes in many physical formats. A common type is the plug-in card, which fits into a free expansion slot in the computer. The analog input specifications can give information on both the capabilities and accuracy of the DAQ product. Basic specifications, which are available on most DAQ products, indicate the number of Channels, sampling rate, resolution, and input range. The DAQ card used in the experiment is national instrument DAQ hardware PCI-6023E type 267. The sampling rate gets more points in a given time and can therefore offer a better representation of the original signal. The resolution is the number of bits that the analog digital converter uses to represent the analog signal. The higher the resolution, the higher the number of division the range broken into, and therefore, the smaller the detectable voltages change from the modules. The DAQ card used up to 16 analog input channel, 200000 sample/sec, and 12-bit resolution.

The driver software transforms the DAQ and PC into a complete DAQ, analysis, and display system. The DAQ hardware without software is useless. The majority of DAQ application used driver software. The software manages the DAQ operation and its integration with the computer resources. The driver software for a DAQ board will translate the binary code value of the analog digital converter to voltage by multiplying it by a constant. The software used in this system was DEWESOFT version 6.2.9. The selection of the software and DAQ hardware should be handled together. This is because that hardware developed by one company may sometime not match with the software developed and supplied by another.

First, the software should be set to our system type and the hardware requirement. The software was featuring a general set up display, sound, and sample rate selection. The input scaling, calibration and temperature modules setting range are shown in Figure 6.16 and the example of pressure setting modules is shown in Figure 6.18.

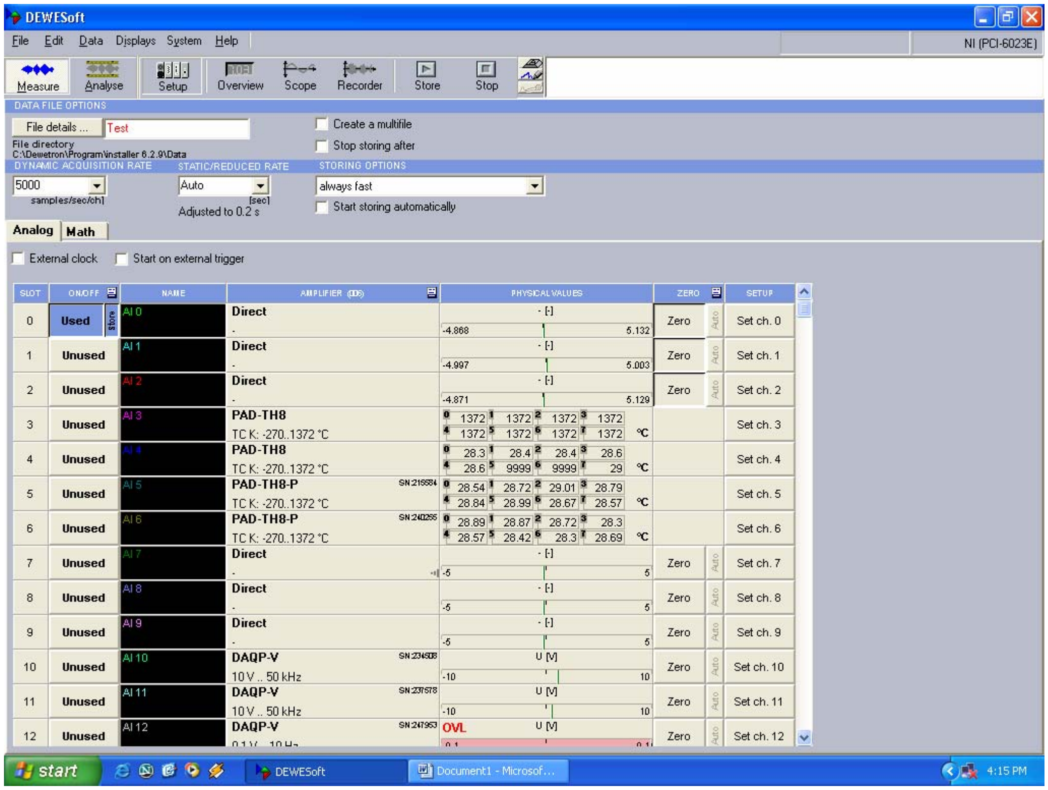


Figure 6.16 Scan of the pressure and temperature modules setting

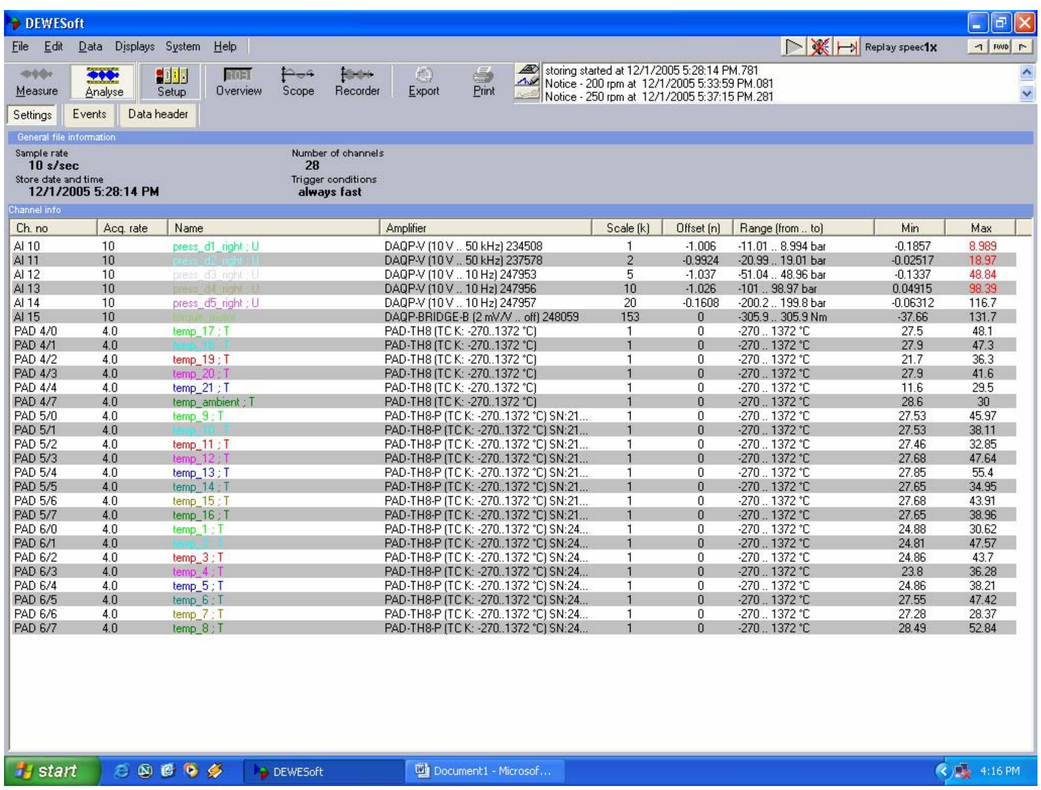


Figure 6.17 Sample of the pressure module setting sensor

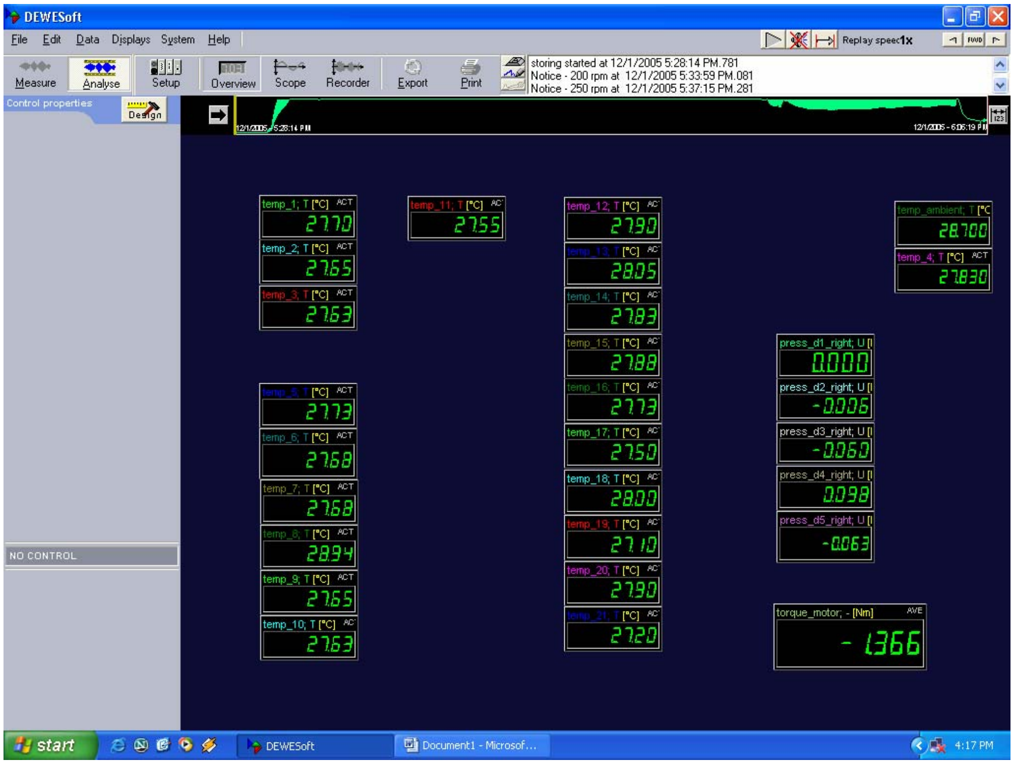


Figure 6.18 Sample of display desired meter

6.2.2. Components of Experimental Rig

6.2.2.1 Compressor

The compressor is the product or prototype to be studied. It is well secured to the rig by four bolts. It has a window though which the wobbling and anti rotating mechanism could be observed. The compressor is designed to operate up to 1500 rpm to deliver 10 Nm^3/hr air or natural gas up to a maximum design pressure 206 bar.

6.2.2.2 Electric Motor

The compressor is driven by a three-pass induction-type motor, with a frequency of 50 Hz, 37 Kw, and maximum speed of 3000 rev/min.

6.2.2.3 Flow Meter

Flow meter of type BROOKS-MT 3809, with a range of flow rates of 0-35 m^3/hr at T=70^{0}C and Pe=10 bar was used in the experiment. The flow meter is installed before the suction port. This flow meter was designed to measure low pressure the air flow that comes into the compressor.

6.2.2.4 Pressure Regulator

A design suction pressure for the compressor is around 3-7 bar. FESTO-FRC-1/8-S-b type pressure regulator is used to set the suction pressure. This regulator was installed after the flow meter but before the suction port, the outlet pressure of the regulator is 0 – 16 bars.

6.2.2.5 Inverter

A variable speed driver was used to change the speed of the compressor. The best way to change the speed of all AC motor is by changing the frequency of the power supply. The ABB drive inverter, type AC550-01-072A, operating power of 37 kW, and 50/60Hz 3-phase were used. The ACS 401000932 is a microprocessor based Pulse Width Modulated (PWM) adjustable frequency AC drive and it was used in controlling the motor speed of the compressor. The ACS 400 drive is equipped with a library of pre-programmed application, which allows the configuration of inputs, outputs, and the performance parameter for specific applications.

6.2.2.6 Pressure Measurement

The measurements of pressure were taken before and after each stage for both sets of right and left cylinder. There are two different sets of pressure devices used. One is the pressure transducer and the other is the pressure gauge.

6.2.2.6.1 Pressure Gauge

For the left side 6 pressure gauges were used to measure the suction pressure, interstage pressure and the discharge pressure respectively. The pressure gauge of a SKON model with different ranges was used. These ranges are:

- Skon : 0 – 10 (Bar) → Suction
- Skon : 0 – 25 (Bar) → interstage 1 and 2
- Skon : 0 – 25 (Bar) → interstage 2 and 3
- Skon : 0 – 70 (Bar) → interstage 3 and 4
- Skon : 0 – 250 (Bar) → interstage 4 and 5
- Skon : 0 – 400 (Bar) → discharge

6.2.2.6.2 Piezo-Electric Pressure Transducers

To generate a useful output signal from a set of cylinders of the right side, piezo-electric pressure transducers were mounted at location before and after each stage. In piezoelectric pressure sensors, the pressure acts on the surface of diaphragm, which converts it into a proportional force. This force is transmitted to a crystal, giving rise to an electric change on the opposing surfaces. Corresponding to five numbers of stages five piezo-electric pressure transducers are mounted as shown in Figure 6.11. They are installed with respective ranges of pressure measurement of ascending order as follows:

i. Model: XPM5-10G-LC4
   - Range: 0 ...10 bar abs.
   - Over-range: Without damage: 2 x FS, Without destruction: 5 x FS
   - Linearity: +/- 0.35 % FS
   - Repeatability: +/- 0.2% FS
   - Operating temperature range: -40 to 120 deg C
   - Shielded cable with 4 Teflon wires with cable length 4 m
   - Body and flush diaphragm in titanium

ii. Model: XPM5-20G-LC4
   - Range: 0 ...20 bar abs.
   - Over-range: Without damage: 2 x FS, Without destruction: 5 x FS
   - Linearity: +/- 0.35 % FS
   - Repeatability: +/- 0.2% FS
   - Operating temperature range: -40 to 120 deg C
   - Shielded cable with 4 Teflon wires with cable length 4 m
   - Body and flush diaphragm in titanium

iii. Model: XPM5-50G-LC4
   - Range: 0 ...50 bar abs.
   - Over-range: Without damage: 2 x FS, Without destruction: 5 x FS
   - Linearity: +/- 0.35 % FS
   - Repeatability: +/- 0.2% FS
   - Operating temperature range: -40 to 120 deg C
   - Shielded cable with 4 Teflon wires with cable length 4 m
   - Body and flush diaphragm in titanium

iv. Model: XPM5-100G-LC4
   - Range: 0 ...100 bar abs.
   - Over-range: Without damage: 2 x FS, Without destruction: 5 x FS
   - Linearity: +/- 0.35 % FS
   - Repeatability: +/- 0.2% FS

- Operating temperature range: -40 to 120 deg C
- Shielded cable with 4 Teflon wires with cable length 4 m
- Body and flush diaphragm in titanium

v. Model: XPM5-350G-LC4

- Range: 0 ...350 bar abs.
- Over-range: Without damage: 2 x FS, Without destruction: 5 x FS
- Linearity: +/- 0.35 % FS
- Repeatability: +/- 0.2% FS
- Operating temperature range: -40 to 120 deg C
- Shielded cable with 4 Teflon wires with cable length 4 m
- Body and flush diaphragm in titanium

The sensors are products of KISTLER.

6.2.2.6.3 Mounting of Pressure Sensor

The accuracy of the pressure measurement depends very much on the method used to install of the pressure sensors. The mounting must be appropriate with the fitting that was used or appropriate with the standard fitting that have to be used. There are two types of mounting, first direct mounting; by direct drilling on the sensor installed location without using fitting and seal. Using this direct mounting has more risks on leakage. Second mounting was by using SWAGELOK fitting. The advantages of the mounted fitting which followed the standard (the NPT standard) were that it offered lesser risk of leakage.

6.2.2.7 Temperature

All temperatures were measured using thermocouples. Thermocouple is based on the principle that when two dissimilar metals are joined a predictable voltage will be generated that relates to the difference in temperature between the junction and the reference junction. The thermocouples used in this experiment are of "K" type. Range of thermocouple is from minus 20^0C to 400^0C. 21 of thermocouples were installed in the experimental rig. These thermocouples were installed in two locations, 10 at suction, 10 at discharge, of each stage and the last one was installed before the storage tank.

6.3 Experimental Procedure

The experiment carried out was actually very simple using air as substance to be compressed. The objective was to compress the air up to an operating pressure of 206 bar, at operating speed between 0 rpm to 1500 rpm to achieve a flow rate of 10 Nm^3/hr. The following procedure was used to carry out each test.

i. Switch on the supply air compressor and regulate the pressure of the air to about 3 bar.
ii. Set of the pressure relief valve about 206 bar.
iii. Switch on the inverter for the setting speed of the compressor to about 300 rpm to 1500 rpm.
iv. Switch on data acquisition system and setting of the pressure sensors and temperature sensors.
v. Click record at data acquisition system.
vi. Running of the compressor
vii. Increase speed of compressor if the compressor pressures can not build-up.
viii. Finally, to shut down, by using inverter also, reduce speed of the compressor gradually to 0 rpm.

6.4 Experimental Result and Discussion

The experiment carried out was more of test and commissioning and the experimental work were conducted at three different speeds, namely 250 rpm, 400 rpm, and 600 rpm. The results of the experimental test are shown graphically in figure 6.19 to figure 6.31 one set of graph shown the variation of pressure of each stage with time. The second set of graphs show the variation of operating torque also with time. While the third set show the variation the compressed air of temperatures with time.

6.4.1. Experiment Result

First test

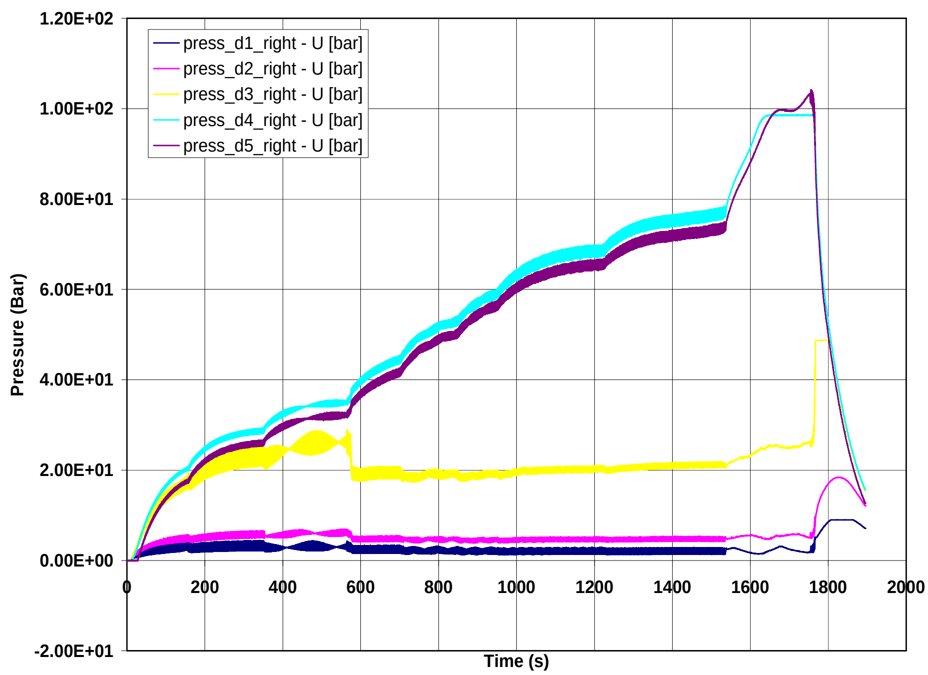


Figure 6.19 Graph pressure vs time at
(Suction pressure 1 bar and at speed 600 rpm)

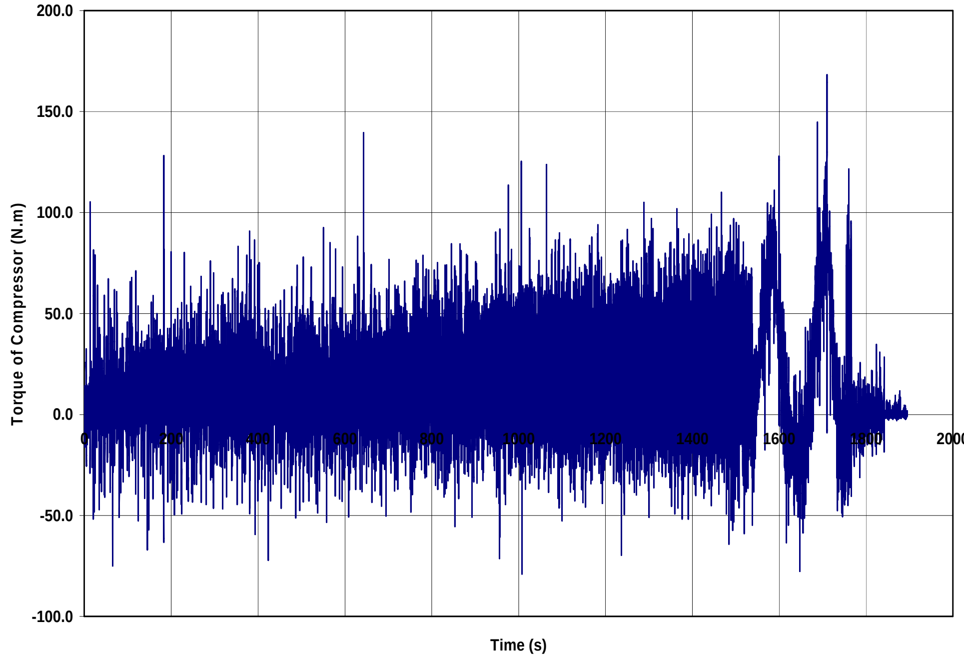


Figure 6.20 Graph torque of compressor with variation speed at (Suction pressure 1 bar and at speed 600 rpm)

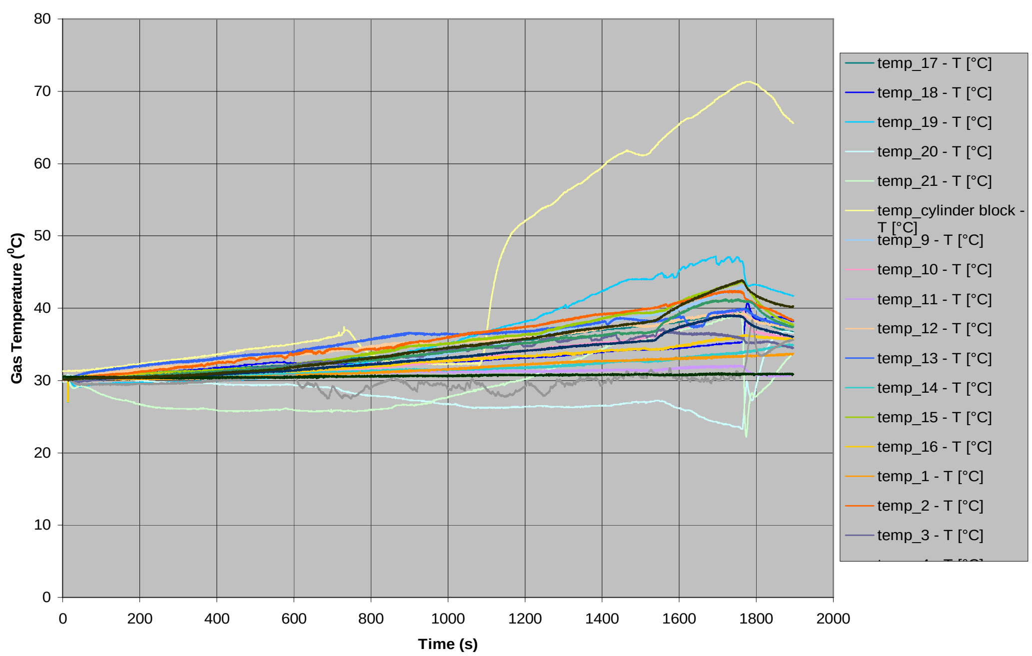


Figure 6.21 Graph gas temperature of compressor with variation speed at (Suction pressure 1 bar and at speed 600 rpm)

Second test

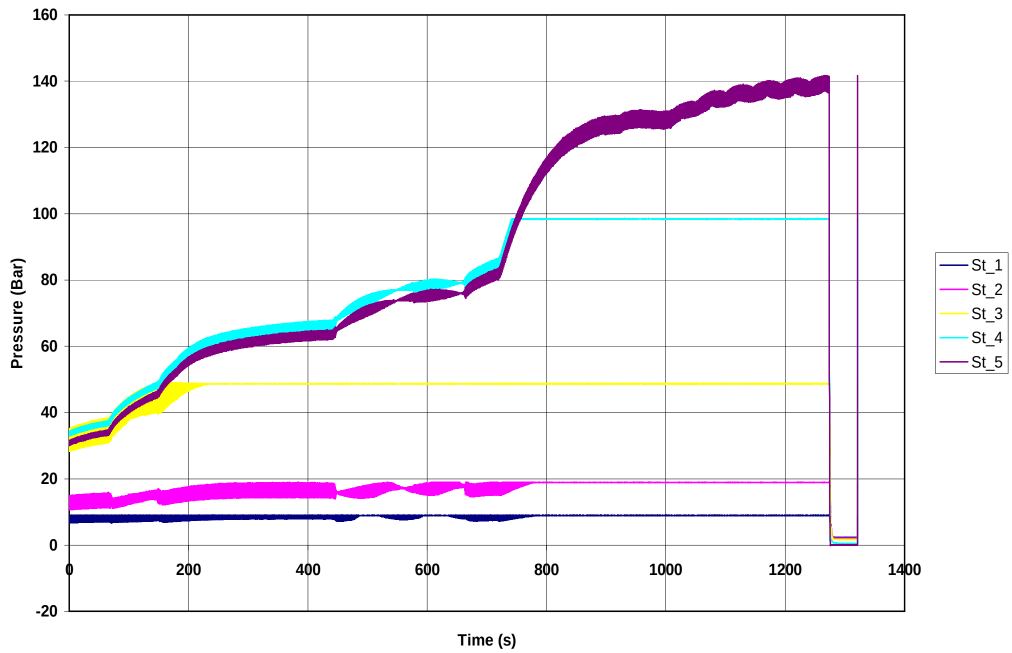


Figure 6.22 Graph pressure vs time at
(Suction pressure 3 bars and at speed 400 rpm)

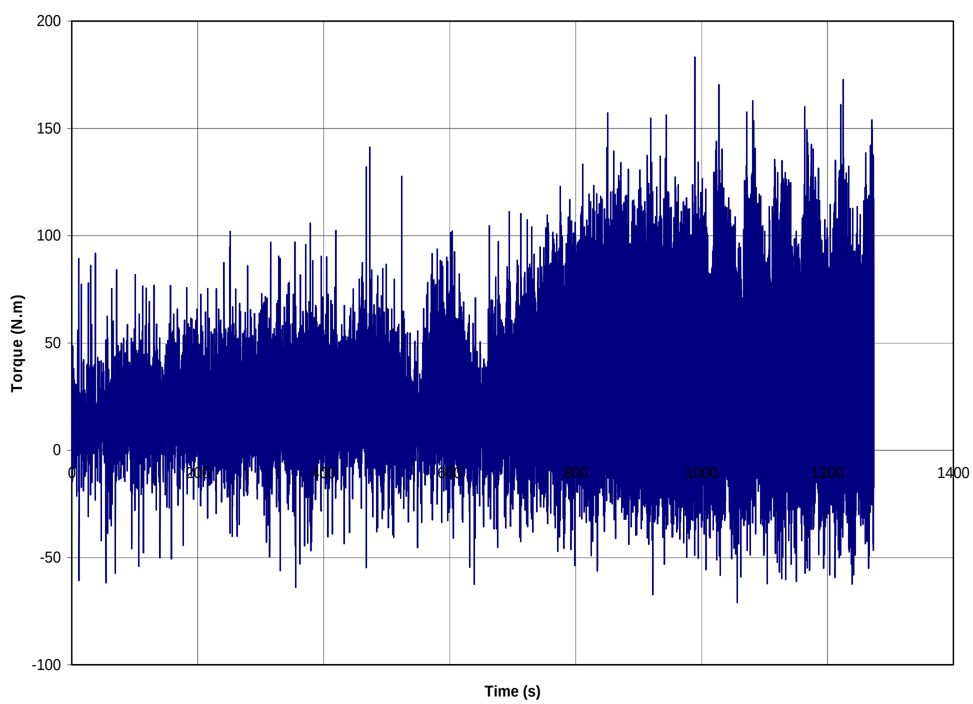


Figure 6.23 Graph torque of compressor with variation speed at
(Suction pressure 3 bars and at speed 400 rpm)

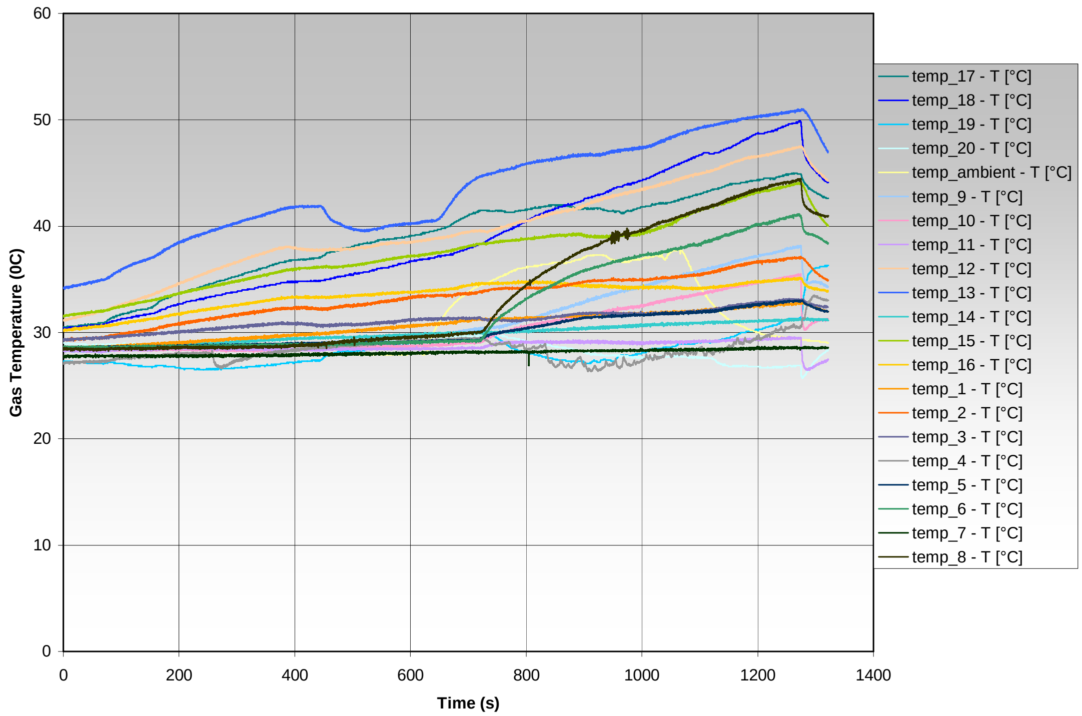


Figure 6.24 Graph gas temperature of compressor with variation speed at (Suction pressure 3 bars and at speed 400 rpm)

<u>Third Test</u>

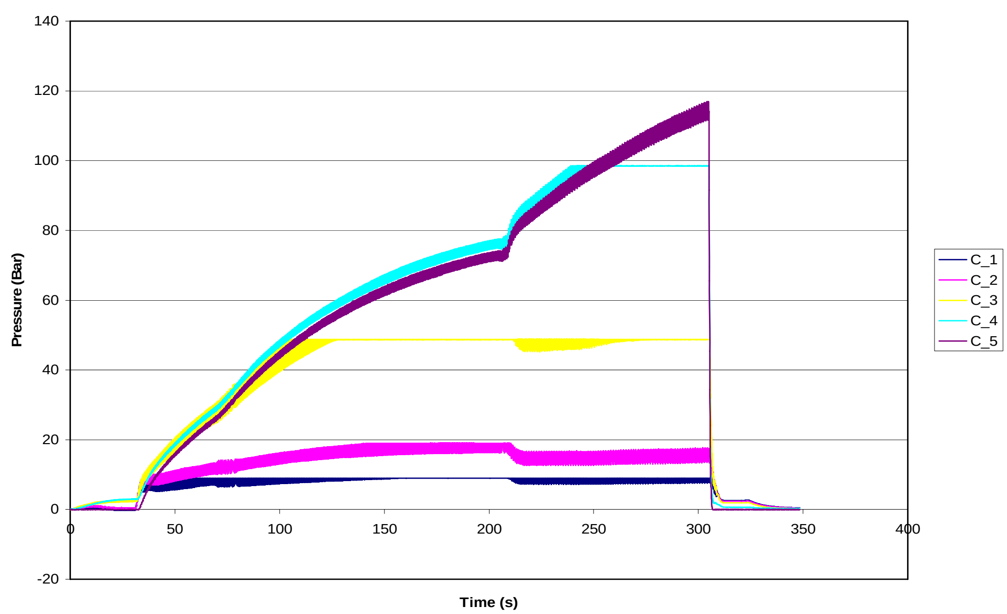


Figure 6.25 Graph pressure vs time at (Suction pressure 3 bars and at speed 250 rpm)

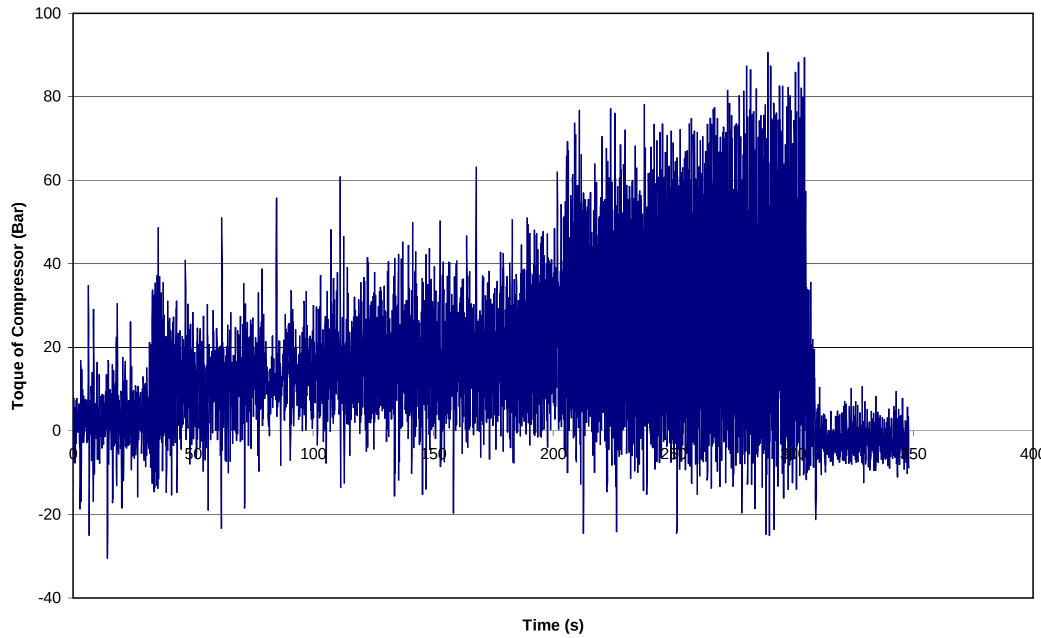


Figure 6.26 Graph torque of compressor with variation speed at (Suction pressure 3 bars and at speed 250 rpm)

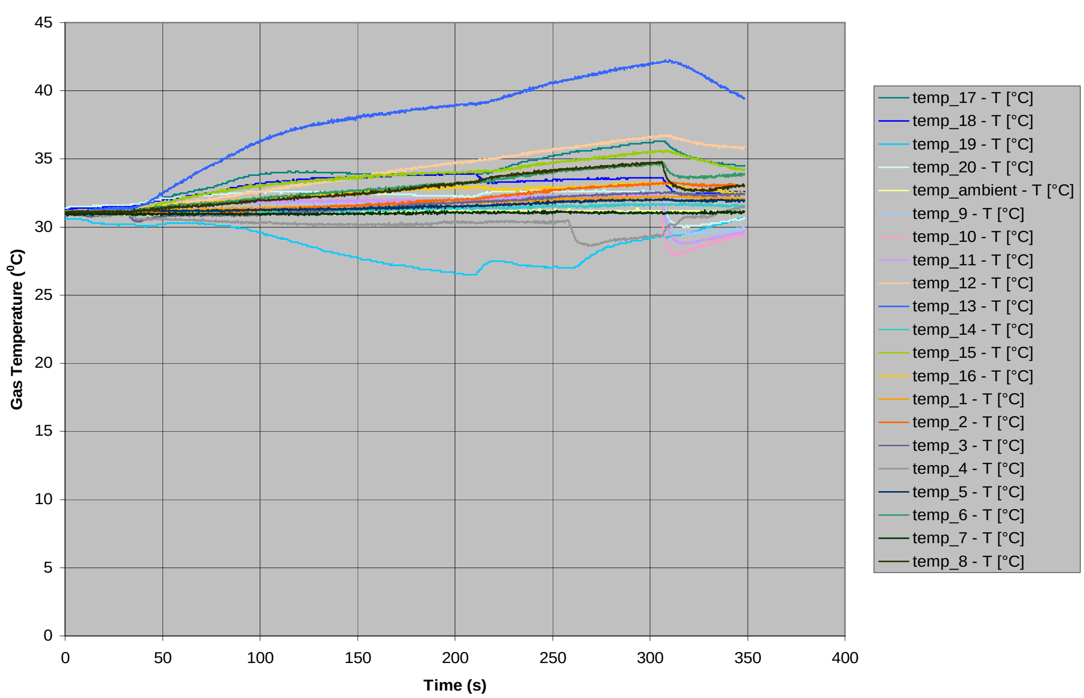


Figure 6.27 Graph gas temperature of compressor with variation speed at (Suction pressure 3 bars and at speed 250 rpm)

Four Test

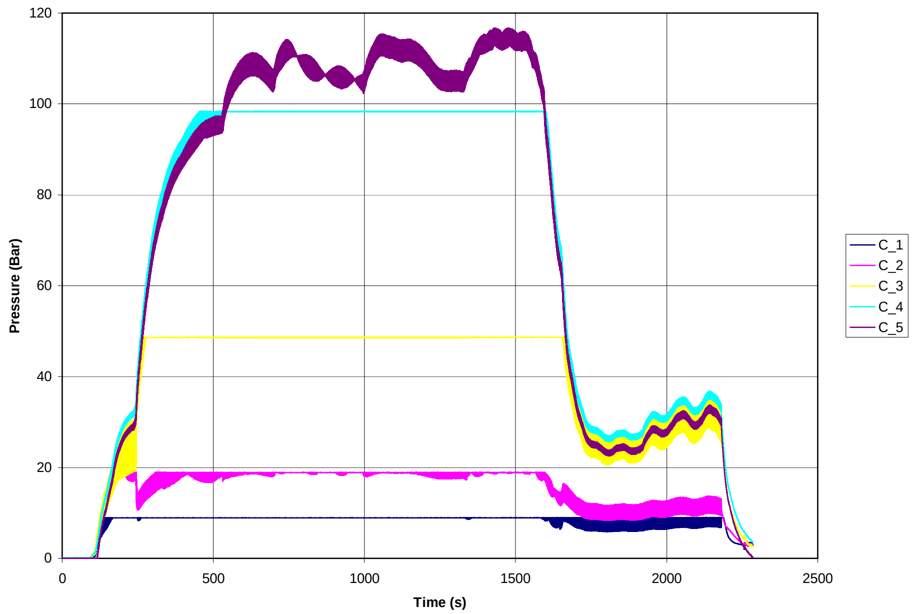


Figure 6.28 Graph pressure vs time at
(Suction pressure 3 bars and at speed 400 rpm)

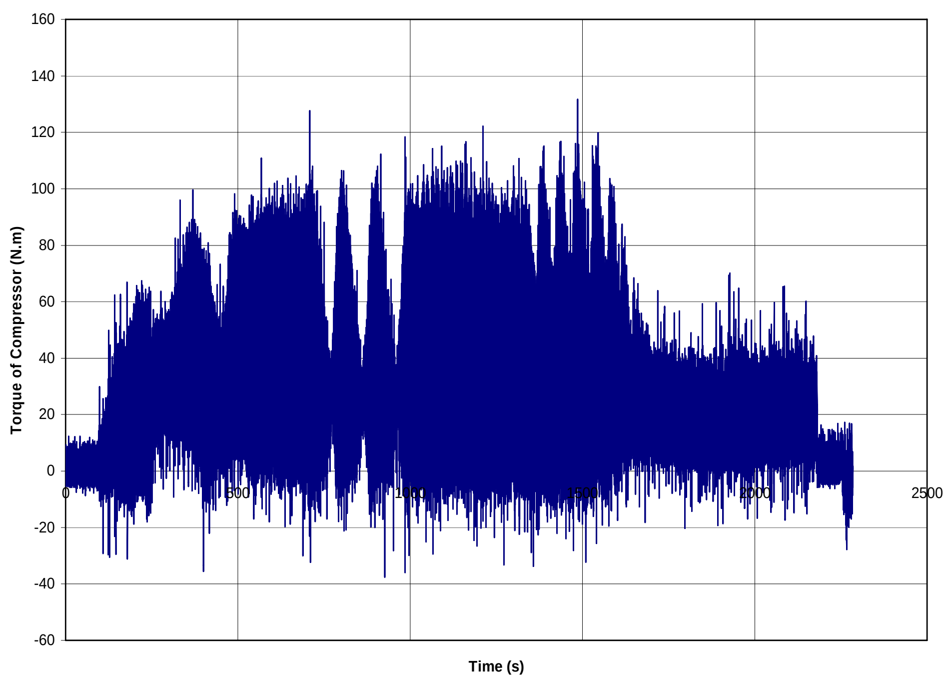


Figure 6.29 Graph torque of compressor with variation speed at
(Suction pressure 3 bars and at speed 400 rpm)

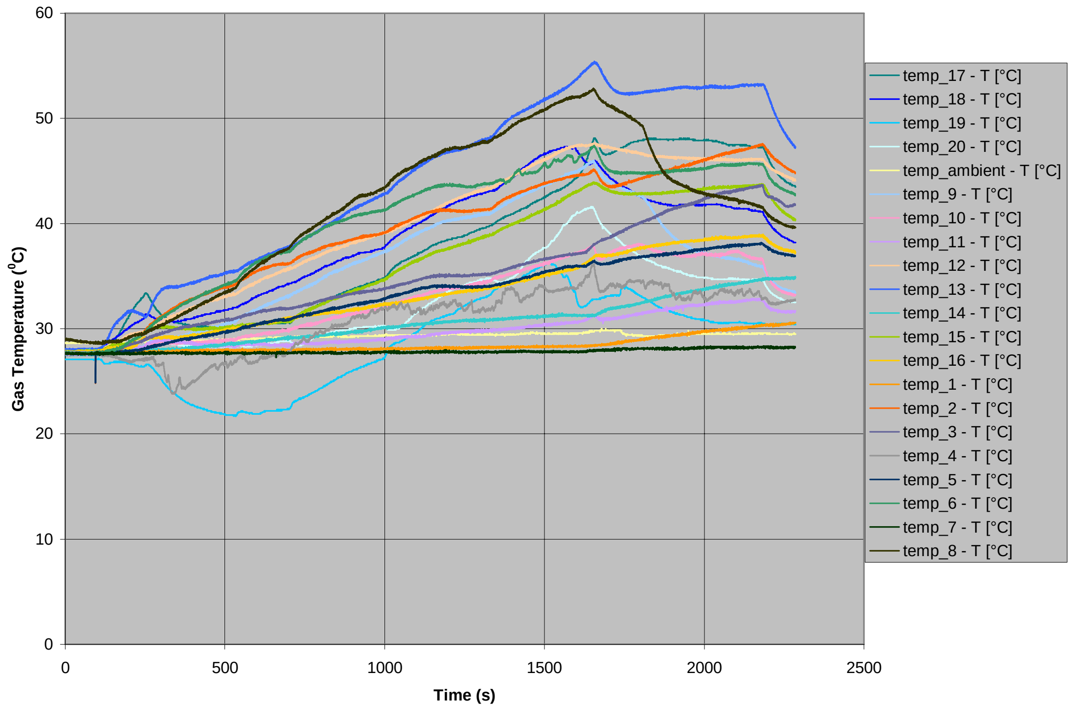


Figure 6.30 Graph gas temperature of compressor with variation speed at (Suction pressure 3 bars and at speed 400 rpm)

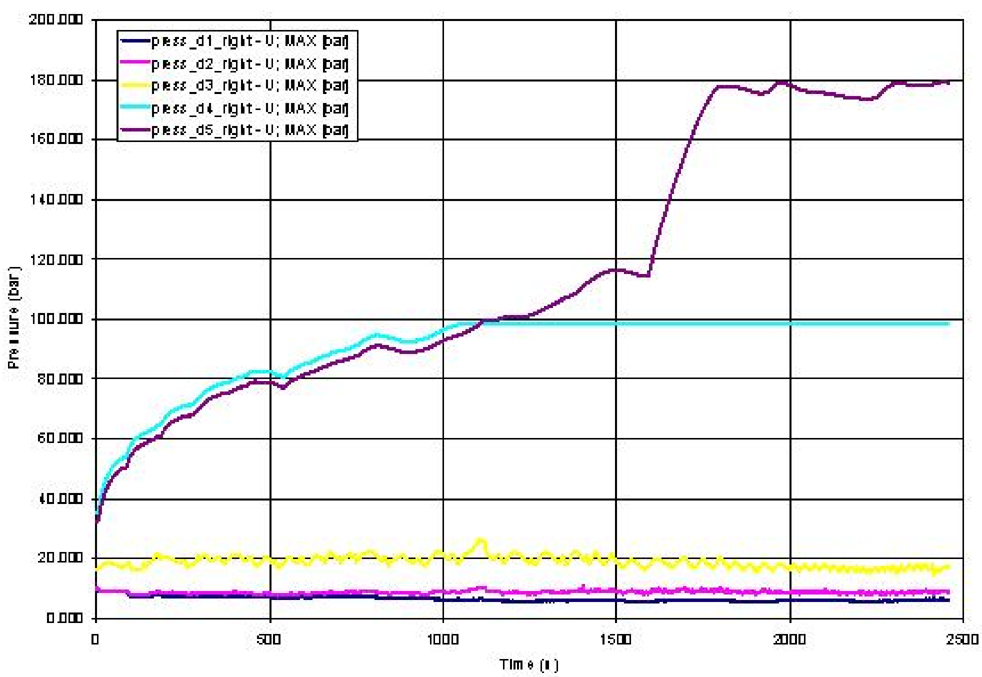


Figure 6.31 Graph pressure vs time at (Suction pressure 3 bars and at speed 400 rpm)

6.4.2. Discussion

The experiments carried out were actually a series of test and commissioning of the prototype. These tests and commissioning was carried out for about 1,5 years. Every failure experienced, the research went back to the drawing board and to the machinist for any modification or rectifications job. The actual elaborate test on performance and its comparison with other compressor of same category will have to be done later when there is time and funding. It is reminded that the objective of the project is to prove that the principle works and the set general specification are met.

The first trial run test was conducted with all suction ports of all cylinders (left and right) exposed to atmospheric pressure of about 1.013 bar. Maximum speed for the first test was 700 rpm.

Results of the first stage could be seen in Figure 6.19 which shows the discharge pressure, Figure 6.20 which shows the torque imposed and Figure 6.21 which shows the temperature, all against the duration of the test.

The maximum pressure that has produced from the data is 104.187 bar and the maximum torque is 164.45 Nm, and the temperature maximum of the gas is 47.2^{0}C. This test was stopped when the compressor speed in 600 rpm. Before the actual test and commissioning were carried out an air compressor and regulated were connected to the suction port of the first cylinder. The objective is to maintain a suction pressure of 3 bar to simulate the actual suction condition of the compressor when the compressing natural gas. In addition flow meter was installed to record the flow rate of the gas.

When everything was set the test was continued. In the second test the suction pressure was 3 bar, using the air compressor and the regulator this suction pressure was controlled to be at all time constant. The second test produced a very good result with the maximum pressure of 141.71 bar and torque of 183.23 Nm. This result can be seen in Figures 6.22 to 6.24. These result are closed to the design specification where the discharge pressure to be obtained was 206 bar.

The discharge gas temperature was surprisingly low when in the pressure was high. The highest gas temperature out of cylinder 5 was around 72^0C. This test was stopped when there was a part failure. The bolt on the piston and coupler were broken which was found later due to the existence of high side force on the piston.

In the third test, the piston and the coupler were joint together and become one piece. Results of this test are given in Figures 6.25 to 6.27. Unfortunately, the results were not as good as expected. In this third test the maximum pressure of only 116 bar and the torque of 90.57 Nm had been obtained. In this third test, there was a a new problem when the piston on the stage 5 was bent and caused the same problem where high side force still existed. Another improvement was therefore needed.

Next, to reduce the side force on the piston, a new guide system consisting of cross head was introduced to a guide the piston movement. The test was run again and the results are given in Figures 6.28 to 6.30. The side force was reduced successfully after been taken away by the cross head. In this test the maximum pressure was 116.69 bar and the maximum torque was about 131.67 N.m. The temperature of the gas was reduced but a new problem existing in the form of leak which existed between the cylinder liner and the piston ring. This source of leak was discovered due to improver installation of the piston ring and the two raider rings. This however was the scope of another researcher who was responsible to develop an effective combination of piston and raider rings assembly.

After the leakage problem was overcome a fourth test was conducted. The results are given in Figure 6.31. In this test the maximum pressure obtained was 180 bar, the maximum torque was about 170 N.m, flow capacity of the gas was 5 Nm^3/hr, running at a speed of 650 rpm.

Listed here with are summary of problems encountered and improvement made:

- Problems
  - High side force on 1st, 2nd, 3rd, and 4th stages.

- Due to high friction, piston ring on 1st and 2nd stages worn out and burnt. Piston on the 3rd stage failed due to improper machining of groove on which piston ring was installed.
- Due to machining problem, shaft was not centric with the housing while wobble plate tilting angle was not the same with that originally designed due to low quality of machining of the rotor creating variation in rotor angle.
- Excessive heat that caused piston ring of stage 5 to weaken and come out of the groove.

• Improvements
  - Use crosshead concept for 1st, 2nd, 3rd, and 4th stages.
  - One piece piston and coupler design for 1st, 2nd, 3rd, and 4th stages.
  - Fabricate new shaft and rotor to correct the wobble plate tilting angle.
  - Change to new bearings.
  - Modify piston groove to achieve the correct tolerance for piston rings.

Overall, the test was quite successful except for cylinder 5, shown in Table 6.1, where the discharge pressure obtained was 180 bar which was lower than our expectation of 206.84 bar.

Table 6.1 Comparison of design pressures with that of test results.

Stages	Design pressure (bar)	Testing pressure (bar)	Design pressure ratio	Testing Pressure ratio
1	7.82	8.99	2.27	2.99
2	17.7	19	2.27	2.11
3	40.2	48.9	2.27	2.57
4	91.20	98.6	2.27	2.02
5	206.84	180	2.27	1.83

The table also shown the difference between the design pressure ratio and that obtained in the test. This difference happened because of the cylinder liner were machined not according to the dimensions specified. These were found when accurate measurements were carried out to check the quality of the machining. Table 6.2 gives the difference for each cylinder.

Table 6.2 The comparison of dimension on the design and the results of the cylinder block machining.

No of cylinder	Design (mm)	Machining results (mm)
1	39	39
2	28.25	28
3	20.47	20.5
4	14.83	14.5
5	10.74	10

CHAPTER 7

CONCLUSION

7.1 Conclusions

An ambitious effort was made to carry out an extremely risky project of designing, fabricating and testing of a new very high pressure compressor. The compressor in meant to compress natural gas from 3 bar to 206 bar. The design was supposed to be fairly small, compact and stable. A single wobble plate concept is known to be acceptable as a refrigerant gas compressor for the automotive air-conditioning system where the working pressure was relatively low at about 22 bar only. At this pressure, leaking and friction are not significant and can be neglected. The compressor was unstable, higher noise and vibration. During testing of the new compressor natural gas was replaced by air and this has created fairly safe working environment. Nevertheless the commissioning work to be done on the new compressor in the future will be on natural gas. The following conclusions are derived from the present work.

i. A new symmetrical wobble plate reciprocating compressor model has been developed to compress gas up to 206 bar from an inlet condition of 3 bar. Two prototypes made were based on gas flow rates of $10m^3$/hr and 1 m^3/hr respectively. Both of compressor using air as working fluid.

ii. A complete engineering analysis on material, force, thermodynamic, kinematics, fluid flow, heat transfer were carried out during the development of the new compressor model but only last four analysis are reported in this thesis. The first two analysis are reported by a co-worker from the same project.

iii. The tilting angle of the wobble plate is 16^0 and this is the maximum possible allowed by the standard universal end joints that are available in the market. With this limitation and for the compressor to operate with minimum possible operating torque and optimum pressure ratio, the combined analysis described in (ii) gives an optimum number of stages of five.

iv. Temperature rise due to compression of the air for both prototypes was found to be not significant. As such the inter-cooler and after-cooler provided were found unnecessary and were not used.

v. Both prototypes operated with good stability at all speeds and noise generated was acceptably low. The $1m^3/hr$ prototype compressor was run at 1100 rpm producing a discharge pressure of 260 bar.

vi. The piston rings have been through an exhaustive development in term of concept, design and material selection. The final concept, shape and size of the piston rings and of the material selected passed all tests at all pressures. This scope of work was carried out and reported by a second co-worker in the project.

vii. One of the objectives of the project was to develop an oil free lubrication system. As such only grease was applied to bearings, end joints, anti rotating mechanism and other rubbing surfaces. However this method was not very successful. The heat generated by friction appeared to cause the grease to vaporize. Regular greasing was performed during the test to minimize friction which could give detrimental affect on the overall performance of the new compressor. The aspect on lubrication will continue to be studied also by the second co-worker.

7.2 Recommendation for Future Research Work

This research has carried out work on the development of concept and design of a symmetrical wobble plate compressor. However there are still several aspects could be done or continued to further develop the new compressor in term of performance:

i. Perform a test establish the durability of the symmetrical wobble plate concept.

ii. Perform dynamic test to the compressor to evaluate its stability at various speed.

iii. Conduct detail study on the flow of gas through the compressor.

iv. Conduct detailed performance test on the suction and discharge valves.

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APPENDIX A

Distribution torque analysis symmetrical wobble plate compressor for 3 stages

Distribution torque analysis symmetrical wobble plate compressor for 3 stages

Appendix A.1 Torque analysis of symmetrical wobble plate compressor for piston 1

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0000	13.49603	5962.364	0.0000
10	4.924	0.0001	3.44738	1523.005	1.7289
20	4.700	0.0004	3.44738	1523.005	3.4065
30	4.333	0.0009	3.44738	1523.005	4.9824
40	3.834	0.0015	3.44738	1523.005	6.4093
50	3.219	0.0023	3.44738	1523.005	7.6433
60	2.505	0.0033	3.44738	1523.005	8.6463
70	1.714	0.0043	3.44738	1523.005	9.3865
80	0.870	0.0054	3.44738	1523.005	9.8405
90	0.000	0.0065	3.44738	1523.005	9.9934
100	0.870	0.0077	3.44738	1523.005	9.8405
110	1.714	0.0088	3.44738	1523.005	9.3865
120	2.505	0.0098	3.44738	1523.005	8.6463
130	3.219	0.0107	3.44738	1523.005	7.6433
140	3.834	0.0116	3.44738	1523.005	6.4093
150	4.333	0.0122	3.44738	1523.005	4.9824
160	4.700	0.0127	3.44738	1523.005	3.4065
170	4.924	0.0130	3.44738	1523.005	1.7289
180	5.000	0.0131	3.44738	1523.005	0.0000
190	4.924	0.0130	3.48067	1537.714	-1.7456
200	4.700	0.0127	3.58309	1582.959	-3.5406
210	4.333	0.0122	3.76259	1662.263	-5.4380
220	3.834	0.0116	4.03391	1782.125	-7.4997
230	3.219	0.0107	4.42099	1953.134	-9.8020
240	2.505	0.0098	4.96162	2191.976	-12.4442
250	1.714	0.0088	5.71546	2525.011	-15.5622
260	0.870	0.0077	6.77885	2994.804	-19.3502
270	0.000	0.0065	8.31314	3672.631	-24.0988
280	0.870	0.0054	10.60206	4683.848	-30.2637
290	1.714	0.0043	13.49603	5962.364	-36.7471
300	2.505	0.0033	13.49603	5962.364	-33.8491
310	3.219	0.0023	13.49603	5962.364	-29.9226
320	3.834	0.0015	13.49603	5962.364	-25.0915
330	4.333	0.0009	13.49603	5962.364	-19.5056
340	4.700	0.0004	13.49603	5962.364	-13.3358
350	4.924	0.0001	13.49603	5962.364	-6.7686
360	5.000	0.0000	13.49603	5962.364	0.0000

Appendix A.2 Torque analysis of symmetrical wobble plate compressor for piston 2

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 2 (Nm)
0	5.000	0.0098	19.45346	2934.289	-16.4584
10	4.924	0.0088	22.40382	3379.311	-20.7457
20	4.700	0.0077	26.55457	4005.394	-25.9803
30	4.333	0.0065	32.52774	4906.366	-32.5563
40	3.834	0.0054	41.41729	6247.233	-41.0893
50	3.219	0.0043	52.83517	7969.465	-50.277
60	2.505	0.0033	52.83517	7969.465	-46.5049
70	1.714	0.0023	52.83517	7969.465	-41.2035
80	0.870	0.0015	52.83517	7969.465	-34.5387
90	0.000	0.0009	52.83517	7969.465	-26.733
100	0.870	0.0004	52.83517	7969.465	-18.0562
110	1.714	0.0001	52.83517	7969.465	-8.8124
120	2.505	0.0000	52.83517	7969.465	0.674653
130	3.219	0.0001	13.49603	2035.692	2.574244
140	3.834	0.0004	13.49603	2035.692	4.874156
150	4.333	0.0009	13.49603	2035.692	6.997907
160	4.700	0.0015	13.49603	2035.692	8.880454
170	4.924	0.0023	13.49603	2035.692	10.46775
180	5.000	0.0033	13.49603	2035.692	11.71776
190	4.924	0.0043	13.49603	2035.692	12.60083
200	4.700	0.0054	13.49603	2035.692	13.09937
210	4.333	0.0065	13.49603	2035.692	13.20722
220	3.834	0.0077	13.49603	2035.692	12.92886
230	3.219	0.0088	13.49603	2035.692	12.27843
240	2.505	0.0098	13.49603	2035.692	11.27887
250	1.714	0.0107	13.49603	2035.692	9.96106
260	0.870	0.0115	13.49603	2035.692	8.362994
270	0.000	0.0122	13.49603	2035.692	6.52892
280	0.870	0.0127	13.49603	2035.692	4.508469
290	1.714	0.0130	13.49603	2035.692	2.355715
300	2.505	0.0131	13.49603	2035.692	0.128186
310	3.219	0.0130	13.62711	2055.464	-2.1347
320	3.834	0.0127	14.03153	2116.465	-4.48103
330	4.333	0.0122	14.74005	2223.336	-6.98748
340	4.700	0.0116	15.80940	2384.633	-9.74225
350	4.924	0.0107	17.33200	2614.295	-12.853
360	5.000	0.0098	19.45346	2934.289	-16.4584

Appendix A.3 Torque analysis of symmetrical wobble plate compressor for piston 3

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 3 (Nm)
0	5.000	0.0098	52.83517	2720.964	15.2618
10	4.924	0.0107	52.83517	2720.964	13.3774
20	4.700	0.0116	52.83517	2720.964	11.1163
30	4.333	0.0122	52.83517	2720.964	8.5514
40	3.834	0.0127	52.83517	2720.964	5.7609
50	3.219	0.0130	52.83517	2720.964	2.8259
60	2.505	0.0131	52.83517	2720.964	-0.1713
70	1.714	0.0130	53.35395	2747.68	-3.1796
80	0.870	0.0127	54.94291	2829.511	-6.2665
90	0.000	0.0122	57.72272	2972.669	-9.5340
100	0.870	0.0115	61.91589	3188.613	-13.0994
110	1.714	0.0107	67.88510	3496.022	-17.1068
120	2.505	0.0098	76.20199	3924.334	-21.7432
130	3.219	0.0088	87.76964	4520.058	-27.2633
140	3.834	0.0077	104.04669	5358.311	-34.0315
150	4.333	0.0065	127.47570	6564.884	-42.5923
160	4.700	0.0054	162.35256	8361.011	-53.8024
170	4.924	0.0043	206.84271	10652.21	-65.9367
180	5.000	0.0033	206.84271	10652.21	-61.3158
190	4.924	0.0023	206.84271	10652.21	-54.7748
200	4.700	0.0015	206.84271	10652.21	-46.4690
210	4.333	0.0009	206.84271	10652.21	-36.6181
220	3.834	0.0004	206.84271	10652.21	-25.5051
230	3.219	0.0001	206.84271	10652.21	-13.4703
240	2.505	0.0000	206.84271	10652.21	-0.9018
250	1.714	0.0001	52.83517	2720.964	3.0088
260	0.870	0.0004	52.83517	2720.964	6.1648
270	0.000	0.0009	52.83517	2720.964	9.1273
280	0.870	0.0015	52.83517	2720.964	11.7923
290	1.714	0.0023	52.83517	2720.964	14.0678
300	2.505	0.0033	52.83517	2720.964	15.8779
310	3.219	0.0043	52.83517	2720.964	17.1658
320	3.834	0.0054	52.83517	2720.964	17.8960
330	4.333	0.0065	52.83517	2720.964	18.0548
340	4.700	0.0077	52.83517	2720.964	17.6489
350	4.924	0.0088	52.83517	2720.964	16.7040
360	5.000	0.0098	52.83517	2720.964	15.2618

APPENDIX B

Distribution torque analysis symmetrical wobble plate compressor for 4 stages

Distribution torque analysis symmetrical wobble plate compressor for 4 stages

Appendix B.1 Torque analysis of symmetrical wobble plate compressor for piston 1

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0000	9.595	3,692.434	0.000
10	4.924	0.0001	3.447	1,326.707	1.767
20	4.700	0.0005	3.447	1,326.707	3.482
30	4.333	0.0010	3.447	1,326.707	5.093
40	3.834	0.0018	3.447	1,326.707	6.551
50	3.219	0.0027	3.447	1,326.707	7.812
60	2.505	0.0038	3.447	1,326.707	8.837
70	1.714	0.0050	3.447	1,326.707	9.594
80	0.870	0.0063	3.447	1,326.707	10.058
90	0.000	0.0077	3.447	1,326.707	10.214
100	0.870	0.0090	3.447	1,326.707	10.058
110	1.714	0.0103	3.447	1,326.707	9.594
120	2.505	0.0115	3.447	1,326.707	8.837
130	3.219	0.0126	3.447	1,326.707	7.812
140	3.834	0.0136	3.447	1,326.707	6.551
150	4.333	0.0143	3.447	1,326.707	5.093
160	4.700	0.0149	3.447	1,326.707	3.482
170	4.924	0.0152	3.447	1,326.707	1.767
180	5.000	0.0153	3.447	1,326.707	0.000
190	4.924	0.0152	3.481	1,339.520	-1.784
200	4.700	0.0149	3.583	1,378.933	-3.619
210	4.333	0.0143	3.763	1,448.016	-5.558
220	3.834	0.0136	4.034	1,552.429	-7.665
230	3.219	0.0126	4.421	1,701.396	-10.019
240	2.505	0.0115	4.962	1,909.455	-12.719
250	1.714	0.0103	5.715	2,199.565	-15.906
260	0.870	0.0090	6.779	2,608.807	-19.778
270	0.000	0.0077	8.313	3,199.270	-24.631
280	0.870	0.0063	9.595	3,692.434	-27.993
290	1.714	0.0050	9.595	3,692.434	-26.702
300	2.505	0.0038	9.595	3,692.434	-24.596
310	3.219	0.0027	9.595	3,692.434	-21.743
320	3.834	0.0018	9.595	3,692.434	-18.232
330	4.333	0.0010	9.595	3,692.434	-14.173
340	4.700	0.0005	9.595	3,692.434	-9.690
350	4.924	0.0001	9.595	3,692.434	-4.918
360	5.000	0.0000	9.595	3,692.434	0.000

Appendix B.2 Torque analysis of symmetrical wobble plate compressor for piston 2

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0077	23.137	3,977.031	-30.619
10	4.924	0.0063	26.703	4,590.086	-35.174
20	4.700	0.0050	26.703	4,590.086	-33.914
30	4.333	0.0038	26.703	4,590.086	-31.566
40	3.834	0.0027	26.703	4,590.086	-28.174
50	3.219	0.0018	26.703	4,590.086	-23.826
60	2.505	0.0010	26.703	4,590.086	-18.652
70	1.714	0.0005	26.703	4,590.086	-12.819
80	0.870	0.0001	26.703	4,590.086	-6.527
90	0.000	0.0000	26.703	4,590.086	0.000
100	0.870	0.0001	9.595	1,649.237	2.345
110	1.714	0.0005	9.595	1,649.237	4.606
120	2.505	0.0010	9.595	1,649.237	6.702
130	3.219	0.0018	9.595	1,649.237	8.561
140	3.834	0.0027	9.595	1,649.237	10.123
150	4.333	0.0038	9.595	1,649.237	11.342
160	4.700	0.0050	9.595	1,649.237	12.186
170	4.924	0.0063	9.595	1,649.237	12.638
180	5.000	0.0077	9.595	1,649.237	12.697
190	4.924	0.0090	9.595	1,649.237	12.374
200	4.700	0.0103	9.595	1,649.237	11.688
210	4.333	0.0115	9.595	1,649.237	10.672
220	3.834	0.0126	9.595	1,649.237	9.361
230	3.219	0.0135	9.595	1,649.237	7.799
240	2.505	0.0143	9.595	1,649.237	6.032
250	1.714	0.0149	9.595	1,649.237	4.109
260	0.870	0.0152	9.595	1,649.237	2.081
270	0.000	0.0153	9.595	1,649.237	0.000
280	0.870	0.0152	9.688	1,665.279	-2.101
290	1.714	0.0149	9.975	1,714.605	-4.272
300	2.505	0.0143	10.477	1,800.998	-6.587
310	3.219	0.0135	11.236	1,931.434	-9.134
320	3.834	0.0126	12.317	2,117.281	-12.018
330	4.333	0.0115	13.825	2,376.446	-15.378
340	4.700	0.0103	15.924	2,737.215	-19.399
350	4.924	0.0090	18.880	3,245.256	-24.348
360	5.000	0.0077	23.137	3,977.031	-30.619

Appendix B.3 Torque analysis of symmetrical wobble plate compressor for piston 3

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0153	26.703	2,050.175	0.000
10	4.924	0.0152	26.961	2,069.976	-2.757
20	4.700	0.0149	27.754	2,130.881	-5.592
30	4.333	0.0143	29.145	2,237.636	-8.589
40	3.834	0.0136	31.246	2,398.986	-11.846
50	3.219	0.0126	34.245	2,629.188	-15.482
60	2.505	0.0115	38.433	2,950.703	-19.655
70	1.714	0.0103	44.272	3,399.013	-24.580
80	0.870	0.0090	52.509	4,031.420	-30.563
90	0.000	0.0077	64.393	4,943.869	-38.063
100	0.870	0.0063	74.319	5,705.961	-43.258
110	1.714	0.0050	74.319	5,705.961	-41.262
120	2.505	0.0038	74.319	5,705.961	-38.008
130	3.219	0.0027	74.319	5,705.961	-33.599
140	3.834	0.0018	74.319	5,705.961	-28.175
150	4.333	0.0010	74.319	5,705.961	-21.902
160	4.700	0.0005	74.319	5,705.961	-14.974
170	4.924	0.0001	74.319	5,705.961	-7.600
180	5.000	0.0000	74.319	5,705.961	0.000
190	4.924	0.0001	26.703	2,050.175	2.731
200	4.700	0.0005	26.703	2,050.175	5.380
210	4.333	0.0010	26.703	2,050.175	7.870
220	3.834	0.0018	26.703	2,050.175	10.123
230	3.219	0.0027	26.703	2,050.175	12.072
240	2.505	0.0038	26.703	2,050.175	13.657
250	1.714	0.0050	26.703	2,050.175	14.826
260	0.870	0.0063	26.703	2,050.175	15.543
270	0.000	0.0077	26.703	2,050.175	15.784
280	0.870	0.0090	26.703	2,050.175	15.543
290	1.714	0.0103	26.703	2,050.175	14.826
300	2.505	0.0115	26.703	2,050.175	13.657
310	3.219	0.0126	26.703	2,050.175	12.072
320	3.834	0.0136	26.703	2,050.175	10.123
330	4.333	0.0143	26.703	2,050.175	7.870
340	4.700	0.0149	26.703	2,050.175	5.380
350	4.924	0.0152	26.703	2,050.175	2.731
360	5.000	0.0153	26.703	2,050.175	0.000

Appendix B.4 Torque analysis of symmetrical wobble plate compressor for piston 4

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0077	74.319	2,548.585	19.622
10	4.924	0.0090	74.319	2,548.585	19.121
20	4.700	0.0103	74.319	2,548.585	18.062
30	4.333	0.0115	74.319	2,548.585	16.491
40	3.834	0.0126	74.319	2,548.585	14.466
50	3.219	0.0135	74.319	2,548.585	12.052
60	2.505	0.0143	74.319	2,548.585	9.321
70	1.714	0.0149	74.319	2,548.585	6.350
80	0.870	0.0152	74.319	2,548.585	3.216
90	0.000	0.0153	74.319	2,548.585	0.000
100	0.870	0.0152	75.042	2,573.375	-3.247
110	1.714	0.0149	77.265	2,649.598	-6.601
120	2.505	0.0143	81.158	2,783.103	-10.179
130	3.219	0.0135	87.036	2,984.667	-14.114
140	3.834	0.0126	95.411	3,271.858	-18.572
150	4.333	0.0115	107.090	3,672.350	-23.763
160	4.700	0.0103	123.347	4,229.850	-29.978
170	4.924	0.0090	146.241	5,014.932	-37.626
180	5.000	0.0077	179.217	6,145.752	-47.316
190	4.924	0.0063	206.843	7,093.113	-54.355
200	4.700	0.0050	206.843	7,093.113	-52.408
210	4.333	0.0038	206.843	7,093.113	-48.779
220	3.834	0.0027	206.843	7,093.113	-43.537
230	3.219	0.0018	206.843	7,093.113	-36.819
240	2.505	0.0010	206.843	7,093.113	-28.823
250	1.714	0.0005	206.843	7,093.113	-19.810
260	0.870	0.0001	206.843	7,093.113	-10.087
270	0.000	0.0000	206.843	7,093.113	0.000
280	0.870	0.0001	74.319	2,548.585	3.624
290	1.714	0.0005	74.319	2,548.585	7.118
300	2.505	0.0010	74.319	2,548.585	10.356
310	3.219	0.0018	74.319	2,548.585	13.229
320	3.834	0.0027	74.319	2,548.585	15.643
330	4.333	0.0038	74.319	2,548.585	17.526
340	4.700	0.0050	74.319	2,548.585	18.831
350	4.924	0.0063	74.319	2,548.585	19.530
360	5.000	0.0077	74.319	2,548.585	19.622

APPENDIX C

Distribution torque analysis symmetrical wobble plate compressor for 5 stages

Distribution torque analysis symmetrical wobble plate compressor for 5 stages

Appendix C.1 Torque analysis of symmetrical wobble plate compressor for piston 1

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0000	7.818	933.982	0.000
10	4.924	0.0001	3.447	411.821	0.542
20	4.700	0.0005	3.447	411.821	1.068
30	4.333	0.0010	3.447	411.821	1.563
36	4.049	0.0014	3.447	411.821	1.838
40	3.834	0.0018	3.447	411.821	2.010
50	3.219	0.0027	3.447	411.821	2.397
60	2.505	0.0038	3.447	411.821	2.712
70	1.714	0.0050	3.447	411.821	2.944
72	1.549	0.0052	3.447	411.821	2.980
80	0.870	0.0063	3.447	411.821	3.087
90	0.000	0.0076	3.447	411.821	3.135
100	0.870	0.0089	3.447	411.821	3.087
108	1.549	0.0099	3.447	411.821	2.980
110	1.714	0.0102	3.447	411.821	2.944
120	2.505	0.0114	3.447	411.821	2.712
130	3.219	0.0125	3.447	411.821	2.397
140	3.834	0.0134	3.447	411.821	2.010
144	4.049	0.0137	3.447	411.821	1.838
150	4.333	0.0142	3.447	411.821	1.563
160	4.700	0.0147	3.447	411.821	1.068
170	4.924	0.0151	3.447	411.821	0.542
180	5.000	0.0152	3.447	411.821	0.000
190	4.924	0.0151	3.481	415.798	-0.548
200	4.700	0.0147	3.583	428.032	-1.111
210	4.333	0.0142	3.763	449.476	-1.706
216	4.049	0.0137	3.913	467.453	-2.086
220	3.834	0.0134	4.034	481.887	-2.352
230	3.219	0.0125	4.421	528.127	-3.075
240	2.505	0.0114	4.962	592.711	-3.903
250	1.714	0.0102	5.715	682.763	-4.881
252	1.549	0.0099	5.899	704.736	-5.100
260	0.870	0.0089	6.779	809.796	-6.069
270	0.000	0.0076	7.818	933.982	-7.109
280	0.870	0.0063	7.818	933.982	-7.000
288	1.549	0.0052	7.818	933.982	-6.759
290	1.714	0.0050	7.818	933.982	-6.677

300	2.505	0.0038	7.818	933.982	-6.151
310	3.219	0.0027	7.818	933.982	-5.437
320	3.834	0.0018	7.818	933.982	-4.559
324	4.049	0.0014	7.818	933.982	-4.168
330	4.333	0.0010	7.818	933.982	-3.544
340	4.700	0.0005	7.818	933.982	-2.423
350	4.924	0.0001	7.818	933.982	-1.230
360	5.000	0.0000	7.818	933.982	0.000

Appendix C.2 Torque analysis of symmetrical wobble plate compressor for piston 2

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0052	17.732	1,111.590	-8.121
10	4.924	0.0040	17.732	1,111.590	-7.622
20	4.700	0.0029	17.732	1,111.590	-6.879
30	4.333	0.0019	17.732	1,111.590	-5.909
36	4.049	0.0014	17.732	1,111.590	-5.228
40	3.834	0.0012	17.732	1,111.590	-4.737
50	3.219	0.0006	17.732	1,111.590	-3.398
60	2.505	0.0002	17.732	1,111.590	-1.935
70	1.714	0.0000	17.732	1,111.590	-0.397
72	1.549	0.0000	17.732	1,111.590	-0.084
80	0.870	0.0001	7.818	490.134	0.513
90	0.000	0.0004	7.818	490.134	1.186
100	0.870	0.0009	7.818	490.134	1.819
108	1.549	0.0015	7.818	490.134	2.283
110	1.714	0.0016	7.818	490.134	2.391
120	2.505	0.0025	7.818	490.134	2.881
130	3.219	0.0036	7.818	490.134	3.273
140	3.834	0.0047	7.818	490.134	3.556
144	4.049	0.0052	7.818	490.134	3.637
150	4.333	0.0060	7.818	490.134	3.722
160	4.700	0.0073	7.818	490.134	3.769
170	4.924	0.0086	7.818	490.134	3.698
180	5.000	0.0099	7.818	490.134	3.515
190	4.924	0.0111	7.818	490.134	3.229
200	4.700	0.0123	7.818	490.134	2.852
210	4.333	0.0132	7.818	490.134	2.396
216	4.049	0.0137	7.818	490.134	2.091
220	3.834	0.0140	7.818	490.134	1.877

230	3.219	0.0146	7.818	490.134	1.310
240	2.505	0.0150	7.818	490.134	0.710
250	1.714	0.0152	7.818	490.134	0.095
252	1.549	0.0152	7.818	490.134	-0.029
260	0.870	0.0151	7.868	493.221	-0.524
270	0.000	0.0148	8.070	505.904	-1.156
280	0.870	0.0143	8.443	529.317	-1.823
288	1.549	0.0137	8.885	556.982	-2.398
290	1.714	0.0136	9.018	565.315	-2.549
300	2.505	0.0127	9.842	616.963	-3.364
310	3.219	0.0116	10.992	689.113	-4.305
320	3.834	0.0104	12.592	789.414	-5.423
324	4.049	0.0099	13.399	839.948	-5.936
330	4.333	0.0092	14.837	930.152	-6.792
340	4.700	0.0078	17.732	1,111.590	-8.365
350	4.924	0.0065	17.732	1,111.590	-8.369
360	5.000	0.0052	17.732	1,111.590	-8.121

Appendix C.3 Torque analysis of symmetrical wobble plate compressor for piston 3

Angle Shaft Rotation (Deg)	**Tilting Angle Wobble Plate (Deg)**	**Stroke of Compressor (m)**	**Pressure Distribution in Cylinder (Bar)**	**Force Distribution of Piston (N)**	**Distribution of Total Torque Piston 1 (Nm)**
0	5.000	0.0137	20.152	662.958	-2.894
10	4.924	0.0129	21.892	720.205	-3.887
20	4.700	0.0118	24.329	800.370	-5.020
30	4.333	0.0107	27.714	911.724	-6.348
36	4.049	0.0099	30.358	998.716	-7.267
40	3.834	0.0094	32.449	1,067.503	-7.943
50	3.219	0.0081	39.198	1,289.554	-9.915
60	2.505	0.0068	40.214	1,322.974	-10.179
70	1.714	0.0055	40.214	1,322.974	-9.861
72	1.549	0.0053	40.214	1,322.974	-9.758
80	0.870	0.0043	40.214	1,322.974	-9.225
90	0.000	0.0031	40.214	1,322.974	-8.291
100	0.870	0.0021	40.214	1,322.974	-7.090
108	1.549	0.0014	40.214	1,322.974	-5.965
110	1.714	0.0013	40.214	1,322.974	-5.664
120	2.505	0.0007	40.214	1,322.974	-4.062
130	3.219	0.0002	40.214	1,322.974	-2.340
140	3.834	0.0000	40.214	1,322.974	-0.555
144	4.049	0.0000	40.214	1,322.974	0.162

Distribution torque analysis symmetrical wobble plate compressor for 5 stages

150	4.333	0.0000	17.732	583.339	0.543
160	4.700	0.0003	17.732	583.339	1.307
170	4.924	0.0008	17.732	583.339	2.024
180	5.000	0.0015	17.732	583.339	2.673
190	4.924	0.0023	17.732	583.339	3.236
200	4.700	0.0033	17.732	583.339	3.697
210	4.333	0.0045	17.732	583.339	4.045
216	4.049	0.0052	17.732	583.339	4.196
220	3.834	0.0057	17.732	583.339	4.272
230	3.219	0.0071	17.732	583.339	4.372
240	2.505	0.0084	17.732	583.339	4.345
250	1.714	0.0097	17.732	583.339	4.191
252	1.549	0.0099	17.732	583.339	4.146
260	0.870	0.0109	17.732	583.339	3.917
270	0.000	0.0120	17.732	583.339	3.529
280	0.870	0.0130	17.732	583.339	3.038
288	1.549	0.0137	17.732	583.339	2.582
290	1.714	0.0139	17.732	583.339	2.459
300	2.505	0.0145	17.732	583.339	1.808
310	3.219	0.0150	17.732	583.339	1.101
320	3.834	0.0151	17.732	583.339	0.358
324	4.049	0.0152	17.732	583.339	0.055
330	4.333	0.0151	17.796	585.443	-0.401
340	4.700	0.0149	18.183	598.197	-1.180
350	4.924	0.0144	18.949	623.387	-2.002
360	5.000	0.0137	20.152	662.958	-2.894

Appendix C.4 Torque analysis of symmetrical wobble plate compressor for piston 4

Angle Shaft Rotation (Deg)	**Tilting Angle Wobble Plate (Deg)**	**Stroke of Compressor (m)**	**Pressure Distribution in Cylinder (Bar)**	**Force Distribution of Piston (N)**	**Distribution of Total Torque Piston 1 (Nm)**
0	5.000	0.0137	40.214	694.268	3.031
10	4.924	0.0144	40.214	694.268	2.230
20	4.700	0.0149	40.214	694.268	1.370
30	4.333	0.0151	40.214	694.268	0.476
36	4.049	0.0152	40.214	694.268	-0.066
40	3.834	0.0151	40.274	695.294	-0.427
50	3.219	0.0150	40.977	707.443	-1.335
60	2.505	0.0145	42.525	734.170	-2.275
70	1.714	0.0139	45.040	777.586	-3.278
72	1.549	0.0137	45.676	788.558	-3.490

Distribution torque analysis symmetrical wobble plate compressor for 5 stages

80	0.870	0.0130	48.731	841.308	-4.382
90	0.000	0.0120	53.932	931.100	-5.632
100	0.870	0.0109	61.167	1,056.007	-7.090
108	1.549	0.0099	68.964	1,190.606	-8.462
110	1.714	0.0097	71.268	1,230.393	-8.840
120	2.505	0.0084	85.595	1,477.729	-11.006
130	3.219	0.0071	91.203	1,574.554	-11.801
140	3.834	0.0057	91.203	1,574.554	-11.530
144	4.049	0.0052	91.203	1,574.554	-11.326
150	4.333	0.0045	91.203	1,574.554	-10.918
160	4.700	0.0033	91.203	1,574.554	-9.979
170	4.924	0.0023	91.203	1,574.554	-8.734
180	5.000	0.0015	91.203	1,574.554	-7.216
190	4.924	0.0008	91.203	1,574.554	-5.464
200	4.700	0.0003	91.203	1,574.554	-3.529
210	4.333	0.0000	91.203	1,574.554	-1.466
216	4.049	0.0000	91.203	1,574.554	-0.193
220	3.834	0.0000	40.214	694.268	0.291
230	3.219	0.0002	40.214	694.268	1.228
240	2.505	0.0007	40.214	694.268	2.132
250	1.714	0.0013	40.214	694.268	2.972
252	1.549	0.0014	40.214	694.268	3.130
260	0.870	0.0021	40.214	694.268	3.721
270	0.000	0.0031	40.214	694.268	4.351
280	0.870	0.0043	40.214	694.268	4.841
288	1.549	0.0053	40.214	694.268	5.121
290	1.714	0.0055	40.214	694.268	5.175
300	2.505	0.0068	40.214	694.268	5.342
310	3.219	0.0081	40.214	694.268	5.338
320	3.834	0.0094	40.214	694.268	5.166
324	4.049	0.0099	40.214	694.268	5.052
330	4.333	0.0107	40.214	694.268	4.834
340	4.700	0.0118	40.214	694.268	4.355
350	4.924	0.0129	40.214	694.268	3.747
360	5.000	0.0137	40.214	694.268	3.031

Appendix C.5 Torque analysis of symmetrical wobble plate compressor for piston 5

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0052	91.203	826.292	6.037
10	4.924	0.0065	91.203	826.292	6.221
20	4.700	0.0078	91.203	826.292	6.218
30	4.333	0.0092	91.203	826.292	6.033
36	4.049	0.0099	91.203	826.292	5.839
40	3.834	0.0104	91.203	826.292	5.677
50	3.219	0.0116	91.203	826.292	5.162
60	2.505	0.0127	91.203	826.292	4.505
70	1.714	0.0136	91.203	826.292	3.726
72	1.549	0.0137	91.203	826.292	3.557
80	0.870	0.0143	91.203	826.292	2.846
90	0.000	0.0148	91.203	826.292	1.888
100	0.870	0.0151	91.203	826.292	0.878
108	1.549	0.0152	91.203	826.292	0.049
110	1.714	0.0151	91.237	826.597	-0.160
120	2.505	0.0150	92.477	837.829	-1.214
130	3.219	0.0146	95.591	866.044	-2.314
140	3.834	0.0140	100.821	913.427	-3.498
144	4.049	0.0137	103.593	938.545	-4.005
150	4.333	0.0132	108.594	983.848	-4.810
160	4.700	0.0123	119.597	1,083.536	-6.305
170	4.924	0.0111	134.912	1,222.287	-8.053
180	5.000	0.0099	156.250	1,415.612	-10.152
190	4.924	0.0086	186.393	1,688.704	-12.740
200	4.700	0.0073	206.843	1,873.976	-14.409
210	4.333	0.0060	206.843	1,873.976	-14.230
216	4.049	0.0052	206.843	1,873.976	-13.904
220	3.834	0.0047	206.843	1,873.976	-13.596
230	3.219	0.0036	206.843	1,873.976	-12.515
240	2.505	0.0025	206.843	1,873.976	-11.015
250	1.714	0.0016	206.843	1,873.976	-9.142
252	1.549	0.0015	206.843	1,873.976	-8.727
260	0.870	0.0009	206.843	1,873.976	-6.956
270	0.000	0.0004	206.843	1,873.976	-4.534
280	0.870	0.0001	206.843	1,873.976	-1.961
288	1.549	0.0000	206.843	1,873.976	0.142
290	1.714	0.0000	91.203	826.292	0.295

Distribution torque analysis symmetrical wobble plate compressor for 5 stages

300	2.505	0.0002	91.203	826.292	1.438
310	3.219	0.0006	91.203	826.292	2.526
320	3.834	0.0012	91.203	826.292	3.521
324	4.049	0.0014	91.203	826.292	3.886
330	4.333	0.0019	91.203	826.292	4.392
340	4.700	0.0029	91.203	826.292	5.113
350	4.924	0.0040	91.203	826.292	5.666
360	5.000	0.0052	91.203	826.292	6.037

APPENDIX D

Distribution torque analysis symmetrical wobble plate compressor for 6 stages

Distribution torque analysis symmetrical wobble plate compressor for 6 stages

Appendix D.1 Torque analysis of symmetrical wobble plate compressor for piston 1

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0000	6.821	1,928.590	0.000
10	4.924	0.0002	3.447	974.723	1.770
20	4.700	0.0006	3.447	974.723	3.488
30	4.333	0.0014	3.447	974.723	5.102
40	3.834	0.0024	3.447	974.723	6.563
50	3.219	0.0037	3.447	974.723	7.827
60	2.505	0.0052	3.447	974.723	8.854
70	1.714	0.0069	3.447	974.723	9.612
80	0.870	0.0086	3.447	974.723	10.077
90	0.000	0.0105	3.447	974.723	10.233
100	0.870	0.0123	3.447	974.723	10.077
110	1.714	0.0140	3.447	974.723	9.612
120	2.505	0.0157	3.447	974.723	8.854
130	3.219	0.0172	3.447	974.723	7.827
140	3.834	0.0185	3.447	974.723	6.563
150	4.333	0.0195	3.447	974.723	5.102
160	4.700	0.0203	3.447	974.723	3.488
170	4.924	0.0208	3.447	974.723	1.770
180	5.000	0.0209	3.447	974.723	0.000
190	4.924	0.0208	3.481	984.137	-1.788
200	4.700	0.0203	3.583	1,013.094	-3.626
210	4.333	0.0195	3.763	1,063.847	-5.569
220	3.834	0.0185	4.034	1,140.557	-7.680
230	3.219	0.0172	4.421	1,249.999	-10.037
240	2.505	0.0157	4.962	1,402.851	-12.743
250	1.714	0.0140	5.715	1,615.982	-15.935
260	0.870	0.0123	6.779	1,916.633	-19.814
270	0.000	0.0105	6.821	1,928.590	-20.248
280	0.870	0.0086	6.821	1,928.590	-19.938
290	1.714	0.0069	6.821	1,928.590	-19.018
300	2.505	0.0052	6.821	1,928.590	-17.518
310	3.219	0.0037	6.821	1,928.590	-15.486
320	3.834	0.0024	6.821	1,928.590	-12.986
330	4.333	0.0014	6.821	1,928.590	-10.095
340	4.700	0.0006	6.821	1,928.590	-6.902
350	4.924	0.0002	6.821	1,928.590	-3.503
360	5.000	0.0000	6.821	1,928.590	0.000

Appendix D.2 Torque analysis of symmetrical wobble plate compressor for piston 2

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0052	13.496	2,229.694	-20.693
10	4.924	0.0037	13.496	2,229.694	-18.586
20	4.700	0.0025	13.496	2,229.694	-15.859
30	4.333	0.0014	13.496	2,229.694	-12.579
40	3.834	0.0006	13.496	2,229.694	-8.838
50	3.219	0.0002	13.496	2,229.694	-4.753
60	2.505	0.0000	13.496	2,229.694	-0.460
70	1.714	0.0002	6.821	1,126.903	1.966
80	0.870	0.0006	6.821	1,126.903	4.113
90	0.000	0.0014	6.821	1,126.903	6.128
100	0.870	0.0025	6.821	1,126.903	7.936
110	1.714	0.0038	6.821	1,126.903	9.472
120	2.505	0.0052	6.821	1,126.903	10.681
130	3.219	0.0069	6.821	1,126.903	11.525
140	3.834	0.0087	6.821	1,126.903	11.981
150	4.333	0.0105	6.821	1,126.903	12.044
160	4.700	0.0123	6.821	1,126.903	11.723
170	4.924	0.0140	6.821	1,126.903	11.041
180	5.000	0.0157	6.821	1,126.903	10.034
190	4.924	0.0172	6.821	1,126.903	8.742
200	4.700	0.0185	6.821	1,126.903	7.216
210	4.333	0.0195	6.821	1,126.903	5.507
220	3.834	0.0203	6.821	1,126.903	3.668
230	3.219	0.0208	6.821	1,126.903	1.750
240	2.505	0.0209	6.821	1,126.903	-0.193
250	1.714	0.0208	6.888	1,137.971	-2.135
260	0.870	0.0203	7.093	1,171.861	-4.124
270	0.000	0.0195	7.452	1,231.147	-6.231
280	0.870	0.0185	7.993	1,320.575	-8.537
290	1.714	0.0172	8.764	1,447.879	-11.143
300	2.505	0.0157	9.837	1,625.253	-14.178
310	3.219	0.0140	11.331	1,871.962	-17.816
320	3.834	0.0123	13.432	2,219.116	-22.309
330	4.333	0.0105	13.496	2,229.694	-22.987
340	4.700	0.0086	13.496	2,229.694	-22.901
350	4.924	0.0069	13.496	2,229.694	-22.137
360	5.000	0.0052	13.496	2,229.694	-20.693

Appendix D.3 Torque analysis of symmetrical wobble plate compressor for piston 3

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0157	19.453	1,877.926	-16.720
10	4.924	0.0140	22.404	2,162.733	-21.190
20	4.700	0.0123	26.554	2,563.407	-26.666
30	4.333	0.0105	26.703	2,577.808	-27.551
40	3.834	0.0087	26.703	2,577.808	-27.407
50	3.219	0.0069	26.703	2,577.808	-26.363
60	2.505	0.0052	26.703	2,577.808	-24.433
70	1.714	0.0038	26.703	2,577.808	-21.667
80	0.870	0.0025	26.703	2,577.808	-18.154
90	0.000	0.0014	26.703	2,577.808	-14.017
100	0.870	0.0006	26.703	2,577.808	-9.408
110	1.714	0.0002	26.703	2,577.808	-4.497
120	2.505	0.0000	26.703	2,577.808	0.532
130	3.219	0.0002	13.496	1,302.843	2.777
140	3.834	0.0006	13.496	1,302.843	5.164
150	4.333	0.0014	13.496	1,302.843	7.350
160	4.700	0.0025	13.496	1,302.843	9.267
170	4.924	0.0037	13.496	1,302.843	10.860
180	5.000	0.0052	13.496	1,302.843	12.091
190	4.924	0.0069	13.496	1,302.843	12.935
200	4.700	0.0086	13.496	1,302.843	13.382
210	4.333	0.0105	13.496	1,302.843	13.432
220	3.834	0.0123	13.496	1,302.843	13.098
230	3.219	0.0140	13.496	1,302.843	12.400
240	2.505	0.0157	13.496	1,302.843	11.365
250	1.714	0.0172	13.496	1,302.843	10.027
260	0.870	0.0185	13.496	1,302.843	8.423
270	0.000	0.0195	13.496	1,302.843	6.594
280	0.870	0.0203	13.496	1,302.843	4.585
290	1.714	0.0208	13.496	1,302.843	2.444
300	2.505	0.0209	13.496	1,302.843	0.224
310	3.219	0.0208	13.627	1,315.492	-2.043
320	3.834	0.0203	14.031	1,354.528	-4.408
330	4.333	0.0195	14.740	1,422.922	-6.954
340	4.700	0.0185	15.809	1,526.150	-9.773
350	4.924	0.0172	17.332	1,673.133	-12.980
360	5.000	0.0157	19.453	1,877.926	-16.720

Appendix D.4 Torque analysis of symmetrical wobble plate compressor for piston 4

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0209	26.703	1,506.251	0.000
10	4.924	0.0208	26.961	1,520.799	-2.762
20	4.700	0.0203	27.754	1,565.545	-5.603
30	4.333	0.0195	29.145	1,643.976	-8.605
40	3.834	0.0185	31.246	1,762.516	-11.868
50	3.219	0.0172	34.245	1,931.637	-15.511
60	2.505	0.0157	38.432	2,167.842	-19.691
70	1.714	0.0140	44.271	2,497.195	-24.625
80	0.870	0.0123	52.508	2,961.796	-30.619
90	0.000	0.0105	52.835	2,980.272	-31.289
100	0.870	0.0086	52.835	2,980.272	-30.810
110	1.714	0.0069	52.835	2,980.272	-29.389
120	2.505	0.0052	52.835	2,980.272	-27.071
130	3.219	0.0037	52.835	2,980.272	-23.931
140	3.834	0.0024	52.835	2,980.272	-20.067
150	4.333	0.0014	52.835	2,980.272	-15.600
160	4.700	0.0006	52.835	2,980.272	-10.665
170	4.924	0.0002	52.835	2,980.272	-5.413
180	5.000	0.0000	52.835	2,980.272	0.000
190	4.924	0.0002	26.703	1,506.251	2.736
200	4.700	0.0006	26.703	1,506.251	5.390
210	4.333	0.0014	26.703	1,506.251	7.884
220	3.834	0.0024	26.703	1,506.251	10.142
230	3.219	0.0037	26.703	1,506.251	12.095
240	2.505	0.0052	26.703	1,506.251	13.682
250	1.714	0.0069	26.703	1,506.251	14.853
260	0.870	0.0086	26.703	1,506.251	15.572
270	0.000	0.0105	26.703	1,506.251	15.814
280	0.870	0.0123	26.703	1,506.251	15.572
290	1.714	0.0140	26.703	1,506.251	14.853
300	2.505	0.0157	26.703	1,506.251	13.682
310	3.219	0.0172	26.703	1,506.251	12.095
320	3.834	0.0185	26.703	1,506.251	10.142
330	4.333	0.0195	26.703	1,506.251	7.884
340	4.700	0.0203	26.703	1,506.251	5.390
350	4.924	0.0208	26.703	1,506.251	2.736
360	5.000	0.0209	26.703	1,506.251	0.000

Appendix D.5 Torque analysis of symmetrical wobble plate compressor for piston 5

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0157	52.835	1,741.417	15.505
10	4.924	0.0172	52.835	1,741.417	13.510
20	4.700	0.0185	52.835	1,741.417	11.152
30	4.333	0.0195	52.835	1,741.417	8.510
40	3.834	0.0203	52.835	1,741.417	5.667
50	3.219	0.0208	52.835	1,741.417	2.705
60	2.505	0.0209	52.835	1,741.417	-0.299
70	1.714	0.0208	53.354	1,758.519	-3.299
80	0.870	0.0203	54.943	1,810.891	-6.373
90	0.000	0.0195	57.723	1,902.506	-9.628
100	0.870	0.0185	61.915	2,040.699	-13.193
110	1.714	0.0172	67.884	2,237.424	-17.220
120	2.505	0.0157	76.200	2,511.523	-21.909
130	3.219	0.0140	87.767	2,892.764	-27.532
140	3.834	0.0123	104.044	3,429.226	-34.475
150	4.333	0.0105	104.540	3,445.572	-35.523
160	4.700	0.0086	104.540	3,445.572	-35.390
170	4.924	0.0069	104.540	3,445.572	-34.209
180	5.000	0.0052	104.540	3,445.572	-31.977
190	4.924	0.0037	104.540	3,445.572	-28.721
200	4.700	0.0025	104.540	3,445.572	-24.507
210	4.333	0.0014	104.540	3,445.572	-19.439
220	3.834	0.0006	104.540	3,445.572	-13.658
230	3.219	0.0002	104.540	3,445.572	-7.345
240	2.505	0.0000	104.540	3,445.572	-0.711
250	1.714	0.0002	52.835	1,741.417	3.038
260	0.870	0.0006	52.835	1,741.417	6.355
270	0.000	0.0014	52.835	1,741.417	9.469
280	0.870	0.0025	52.835	1,741.417	12.264
290	1.714	0.0038	52.835	1,741.417	14.637
300	2.505	0.0052	52.835	1,741.417	16.505
310	3.219	0.0069	52.835	1,741.417	17.809
320	3.834	0.0087	52.835	1,741.417	18.514
330	4.333	0.0105	52.835	1,741.417	18.612
340	4.700	0.0123	52.835	1,741.417	18.116
350	4.924	0.0140	52.835	1,741.417	17.062
360	5.000	0.0157	52.835	1,741.417	15.505

Appendix D.6 Torque analysis of symmetrical wobble plate compressor for piston 6

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0052	104.540	2,013.298	18.684
10	4.924	0.0069	104.540	2,013.298	19.989
20	4.700	0.0086	104.540	2,013.298	20.679
30	4.333	0.0105	104.540	2,013.298	20.756
40	3.834	0.0123	104.540	2,013.298	20.240
50	3.219	0.0140	104.540	2,013.298	19.162
60	2.505	0.0157	104.540	2,013.298	17.563
70	1.714	0.0172	104.540	2,013.298	15.495
80	0.870	0.0185	104.540	2,013.298	13.015
90	0.000	0.0195	104.540	2,013.298	10.189
100	0.870	0.0203	104.540	2,013.298	7.085
110	1.714	0.0208	104.540	2,013.298	3.777
120	2.505	0.0209	104.540	2,013.298	0.346
130	3.219	0.0208	105.555	2,032.846	-3.157
140	3.834	0.0203	108.687	2,093.169	-6.812
150	4.333	0.0195	114.175	2,198.858	-10.746
160	4.700	0.0185	122.458	2,358.377	-15.102
170	4.924	0.0172	134.252	2,585.511	-20.058
180	5.000	0.0157	150.684	2,901.981	-25.838
190	4.924	0.0140	173.537	3,342.096	-32.746
200	4.700	0.0123	205.687	3,961.263	-41.208
210	4.333	0.0105	206.843	3,983.517	-42.574
220	3.834	0.0087	206.843	3,983.517	-42.352
230	3.219	0.0069	206.843	3,983.517	-40.739
240	2.505	0.0052	206.843	3,983.517	-37.756
250	1.714	0.0038	206.843	3,983.517	-33.482
260	0.870	0.0025	206.843	3,983.517	-28.054
270	0.000	0.0014	206.843	3,983.517	-21.661
280	0.870	0.0006	206.843	3,983.517	-14.538
290	1.714	0.0002	206.843	3,983.517	-6.949
300	2.505	0.0000	206.843	3,983.517	0.822
310	3.219	0.0002	104.540	2,013.298	4.292
320	3.834	0.0006	104.540	2,013.298	7.981
330	4.333	0.0014	104.540	2,013.298	11.358
340	4.700	0.0025	104.540	2,013.298	14.320
350	4.924	0.0037	104.540	2,013.298	16.782
360	5.000	0.0052	104.540	2,013.298	18.684

APPENDIX E

Distribution torque analysis symmetrical wobble plate compressor for 7 stages

Distribution torque analysis symmetrical wobble plate compressor for 7 stages

Appendix E.1 Torque analysis of symmetrical wobble plate compressor for piston 1

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0000	6.187	1,470.029	0.000
10	4.924	0.0002	3.447	819.038	1.711
20	4.700	0.0007	3.447	819.038	3.371
25.7143	4.507	0.0012	3.447	819.038	4.277
30	4.333	0.0016	3.447	819.038	4.930
40	3.834	0.0028	3.447	819.038	6.342
50	3.219	0.0043	3.447	819.038	7.563
51.4286	3.122	0.0045	3.447	819.038	7.720
60	2.505	0.0060	3.447	819.038	8.556
70	1.714	0.0079	3.447	819.038	9.288
77.1429	1.115	0.0093	3.447	819.038	9.639
80	0.870	0.0099	3.447	819.038	9.737
90	0.000	0.0120	3.447	819.038	9.889
100	0.870	0.0141	3.447	819.038	9.737
102.857	1.115	0.0147	3.447	819.038	9.639
110	1.714	0.0162	3.447	819.038	9.288
120	2.505	0.0181	3.447	819.038	8.556
128.571	3.122	0.0195	3.447	819.038	7.720
130	3.219	0.0198	3.447	819.038	7.563
140	3.834	0.0213	3.447	819.038	6.342
150	4.333	0.0225	3.447	819.038	4.930
154.286	4.507	0.0229	3.447	819.038	4.277
160	4.700	0.0233	3.447	819.038	3.371
170	4.924	0.0239	3.447	819.038	1.711
180	5.000	0.0241	3.447	819.038	0.000
190	4.924	0.0239	3.481	826.949	-1.727
200	4.700	0.0233	3.583	851.280	-3.503
205.714	4.507	0.0229	3.675	873.225	-4.560
210	4.333	0.0225	3.763	893.927	-5.381
220	3.834	0.0213	4.034	958.383	-7.421
230	3.219	0.0198	4.421	1,050.343	-9.699
231.429	3.122	0.0195	4.488	1,066.195	-10.049
240	2.505	0.0181	4.962	1,178.778	-12.313
250	1.714	0.0162	5.715	1,357.862	-15.398
257.143	1.115	0.0147	6.187	1,470.029	-17.300
260	0.870	0.0141	6.187	1,470.029	-17.477
270	0.000	0.0120	6.187	1,470.029	-17.748

Distribution torque analysis symmetrical wobble plate compressor for 7 stages

280	0.870	0.0099	6.187	1,470.029	-17.477
282.857	1.115	0.0093	6.187	1,470.029	-17.300
290	1.714	0.0079	6.187	1,470.029	-16.670
300	2.505	0.0060	6.187	1,470.029	-15.356
308.571	3.122	0.0045	6.187	1,470.029	-13.856
310	3.219	0.0043	6.187	1,470.029	-13.575
320	3.834	0.0028	6.187	1,470.029	-11.383
330	4.333	0.0016	6.187	1,470.029	-8.849
334.286	4.507	0.0012	6.187	1,470.029	-7.677
340	4.700	0.0007	6.187	1,470.029	-6.050
350	4.924	0.0002	6.187	1,470.029	-3.071
360	5.000	0.0000	6.187	1,470.029	0.000

Appendix E.2 Torque analysis of symmetrical wobble plate compressor for piston 2

Angle Shaft Rotation (Deg)	**Tilting Angle Wobble Plate (Deg)**	**Stroke of Compressor (m)**	**Pressure Distribution in Cylinder (Bar)**	**Force Distribution of Piston (N)**	**Distribution of Total Torque Piston 1 (Nm)**
0	5.000	0.0045	11.105	1,664.679	-16.181
10	4.924	0.0030	11.105	1,664.679	-13.948
20	4.700	0.0018	11.105	1,664.679	-11.236
25.7143	4.507	0.0012	11.105	1,664.679	-9.497
30	4.333	0.0008	11.105	1,664.679	-8.115
40	3.834	0.0002	11.105	1,664.679	-4.678
50	3.219	0.0000	11.105	1,664.679	-1.035
51.4286	3.122	0.0000	11.105	1,664.679	-0.505
60	2.505	0.0001	6.187	927.489	1.498
70	1.714	0.0006	6.187	927.489	3.543
77.1429	1.115	0.0012	6.187	927.489	4.943
80	0.870	0.0015	6.187	927.489	5.482
90	0.000	0.0026	6.187	927.489	7.242
100	0.870	0.0041	6.187	927.489	8.757
102.857	1.115	0.0045	6.187	927.489	9.137
110	1.714	0.0058	6.187	927.489	9.972
120	2.505	0.0077	6.187	927.489	10.844
128.571	3.122	0.0094	6.187	927.489	11.298
130	3.219	0.0097	6.187	927.489	11.347
140	3.834	0.0117	6.187	927.489	11.472
150	4.333	0.0138	6.187	927.489	11.223
154.286	4.507	0.0147	6.187	927.489	11.007
160	4.700	0.0159	6.187	927.489	10.621
170	4.924	0.0178	6.187	927.489	9.698

180	5.000	0.0195	6.187	927.489	8.495
190	4.924	0.0211	6.187	927.489	7.059
200	4.700	0.0223	6.187	927.489	5.441
205.714	4.507	0.0229	6.187	927.489	4.456
210	4.333	0.0232	6.187	927.489	3.695
220	3.834	0.0238	6.187	927.489	1.872
230	3.219	0.0239	6.187	927.489	0.023
231.429	3.122	0.0241	6.187	927.489	-0.240
240	2.505	0.0239	6.232	934.120	-1.817
250	1.714	0.0234	6.399	959.190	-3.686
257.143	1.115	0.0229	6.601	989.494	-5.076
260	0.870	0.0226	6.703	1,004.793	-5.651
270	0.000	0.0214	7.169	1,074.558	-7.787
280	0.870	0.0200	7.835	1,174.451	-10.187
282.857	1.115	0.0195	8.070	1,209.730	-10.936
290	1.714	0.0183	8.765	1,313.885	-12.966
300	2.505	0.0164	10.058	1,507.698	-16.284
308.571	3.122	0.0147	11.105	1,664.679	-18.927
310	3.219	0.0144	11.105	1,664.679	-19.048
320	3.834	0.0123	11.105	1,664.679	-19.596
330	4.333	0.0102	11.105	1,664.679	-19.594
334.286	4.507	0.0094	11.105	1,664.679	-19.420
340	4.700	0.0082	11.105	1,664.679	-19.025
350	4.924	0.0063	11.105	1,664.679	-17.883
360	5.000	0.0045	11.105	1,664.679	-16.181

Appendix E.3 Torque analysis of symmetrical wobble plate compressor for piston 3

Angle Shaft Rotation (Deg)	**Tilting Angle Wobble Plate (Deg)**	**Stroke of Compressor (m)**	**Pressure Distribution in Cylinder (Bar)**	**Force Distribution of Piston (N)**	**Distribution of Total Torque Piston 1 (Nm)**
0	5.000	0.0147	19.932	1,885.103	-21.954
10	4.924	0.0126	19.932	1,885.103	-22.863
20	4.700	0.0105	19.932	1,885.103	-23.070
25.7143	4.507	0.0094	19.932	1,885.103	-22.858
30	4.333	0.0085	19.932	1,885.103	-22.537
40	3.834	0.0065	19.932	1,885.103	-21.249
50	3.219	0.0048	19.932	1,885.103	-19.226
51.4286	3.122	0.0045	19.932	1,885.103	-18.880
60	2.505	0.0032	19.932	1,885.103	-16.520
70	1.714	0.0019	19.932	1,885.103	-13.216
77.1429	1.115	0.0012	19.932	1,885.103	-10.552

Distribution torque analysis symmetrical wobble plate compressor for 7 stages

80	0.870	0.0009	19.932	1,885.103	-9.429
90	0.000	0.0003	19.932	1,885.103	-5.300
100	0.870	0.0000	19.932	1,885.103	-0.987
102.857	1.115	0.0000	19.932	1,885.103	0.255
110	1.714	0.0001	11.105	1,050.300	1.862
120	2.505	0.0005	11.105	1,050.300	4.191
128.571	3.122	0.0012	11.105	1,050.300	6.055
130	3.219	0.0013	11.105	1,050.300	6.350
140	3.834	0.0024	11.105	1,050.300	8.265
150	4.333	0.0038	11.105	1,050.300	9.874
154.286	4.507	0.0045	11.105	1,050.300	10.458
160	4.700	0.0055	11.105	1,050.300	11.131
170	4.924	0.0074	11.105	1,050.300	12.008
180	5.000	0.0094	11.105	1,050.300	12.494
190	4.924	0.0114	11.105	1,050.300	12.591
200	4.700	0.0135	11.105	1,050.300	12.315
205.714	4.507	0.0147	11.105	1,050.300	11.999
210	4.333	0.0156	11.105	1,050.300	11.691
220	3.834	0.0175	11.105	1,050.300	10.751
230	3.219	0.0193	11.105	1,050.300	9.533
231.429	3.122	0.0195	11.105	1,050.300	9.339
240	2.505	0.0208	11.105	1,050.300	8.076
250	1.714	0.0221	11.105	1,050.300	6.423
257.143	1.115	0.0229	11.105	1,050.300	5.144
260	0.870	0.0231	11.105	1,050.300	4.613
270	0.000	0.0238	11.105	1,050.300	2.691
280	0.870	0.0240	11.105	1,050.300	0.697
282.857	1.115	0.0241	11.105	1,050.300	0.121
290	1.714	0.0240	11.160	1,055.438	-1.329
300	2.505	0.0235	11.425	1,080.565	-3.421
308.571	3.122	0.0229	11.843	1,120.102	-5.323
310	3.219	0.0227	11.932	1,128.458	-5.654
320	3.834	0.0216	12.719	1,202.925	-8.116
330	4.333	0.0202	13.854	1,310.296	-10.911
334.286	4.507	0.0195	14.472	1,368.693	-12.242
340	4.700	0.0186	15.443	1,460.583	-14.171
350	4.924	0.0167	17.653	1,669.541	-18.071
360	5.000	0.0147	19.932	1,885.103	-21.954

Appendix E.4 Torque analysis of symmetrical wobble plate compressor for piston 4

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0229	21.267	1,269.049	-6.362
10	4.924	0.0218	22.601	1,348.637	-9.261
20	4.700	0.0204	24.534	1,463.974	-12.488
25.7143	4.507	0.0195	25.972	1,549.792	-14.526
30	4.333	0.0188	27.244	1,625.655	-16.168
40	3.834	0.0170	31.010	1,850.391	-20.472
50	3.219	0.0150	35.775	2,134.714	-25.276
51.4286	3.122	0.0147	35.775	2,134.714	-25.449
60	2.505	0.0129	35.775	2,134.714	-26.139
70	1.714	0.0109	35.775	2,134.714	-26.165
77.1429	1.115	0.0094	35.775	2,134.714	-25.665
80	0.870	0.0088	35.775	2,134.714	-25.345
90	0.000	0.0068	35.775	2,134.714	-23.701
100	0.870	0.0050	35.775	2,134.714	-21.289
102.857	1.115	0.0045	35.775	2,134.714	-20.469
110	1.714	0.0034	35.775	2,134.714	-18.192
120	2.505	0.0021	35.775	2,134.714	-14.523
128.571	3.122	0.0012	35.775	2,134.714	-11.020
130	3.219	0.0011	35.775	2,134.714	-10.411
140	3.834	0.0004	35.775	2,134.714	-6.001
150	4.333	0.0000	35.775	2,134.714	-1.443
154.286	4.507	0.0000	35.775	2,134.714	0.519
160	4.700	0.0001	19.932	1,189.373	1.735
170	4.924	0.0005	19.932	1,189.373	4.194
180	5.000	0.0012	19.932	1,189.373	6.498
190	4.924	0.0023	19.932	1,189.373	8.582
200	4.700	0.0036	19.932	1,189.373	10.389
205.714	4.507	0.0045	19.932	1,189.373	11.280
210	4.333	0.0053	19.932	1,189.373	11.873
220	3.834	0.0071	19.932	1,189.373	12.997
230	3.219	0.0091	19.932	1,189.373	13.735
231.429	3.122	0.0093	19.932	1,189.373	13.808
240	2.505	0.0111	19.932	1,189.373	14.072
250	1.714	0.0132	19.932	1,189.373	14.002
257.143	1.115	0.0147	19.932	1,189.373	13.705
260	0.870	0.0153	19.932	1,189.373	13.530
270	0.000	0.0173	19.932	1,189.373	12.670
280	0.870	0.0191	19.932	1,189.373	11.447

282.857	1.115	0.0195	19.932	1,189.373	11.034
290	1.714	0.0207	19.932	1,189.373	9.892
300	2.505	0.0220	19.932	1,189.373	8.048
308.571	3.122	0.0229	19.932	1,189.373	6.273
310	3.219	0.0230	19.932	1,189.373	5.962
320	3.834	0.0237	19.932	1,189.373	3.691
330	4.333	0.0240	19.932	1,189.373	1.295
334.286	4.507	0.0241	19.932	1,189.373	0.246
340	4.700	0.0240	19.998	1,193.288	-1.163
350	4.924	0.0236	20.421	1,218.569	-3.691
360	5.000	0.0229	21.267	1,269.049	-6.362

Appendix E.5 Torque analysis of symmetrical wobble plate compressor for piston 5

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	5.000	0.0229	35.775	1,346.860	6.752
10	4.924	0.0236	35.775	1,346.860	4.080
20	4.700	0.0240	35.775	1,346.860	1.313
25.7143	4.507	0.0241	35.775	1,346.860	-0.279
30	4.333	0.0240	35.835	1,349.119	-1.469
40	3.834	0.0237	36.478	1,373.325	-4.262
50	3.219	0.0230	37.872	1,425.825	-7.147
51.4286	3.122	0.0229	38.138	1,435.847	-7.573
60	2.505	0.0220	40.129	1,510.812	-10.223
70	1.714	0.0207	43.440	1,635.435	-13.602
77.1429	1.115	0.0195	46.610	1,754.796	-16.280
80	0.870	0.0191	48.105	1,811.070	-17.430
90	0.000	0.0173	54.599	2,055.567	-21.898
100	0.870	0.0153	63.675	2,397.284	-27.271
102.857	1.115	0.0147	64.209	2,417.376	-27.856
110	1.714	0.0132	64.209	2,417.376	-28.459
120	2.505	0.0111	64.209	2,417.376	-28.602
128.571	3.122	0.0093	64.209	2,417.376	-28.065
130	3.219	0.0091	64.209	2,417.376	-27.917
140	3.834	0.0071	64.209	2,417.376	-26.417
150	4.333	0.0053	64.209	2,417.376	-24.132
154.286	4.507	0.0045	64.209	2,417.376	-22.925
160	4.700	0.0036	64.209	2,417.376	-21.116
170	4.924	0.0023	64.209	2,417.376	-17.443
180	5.000	0.0012	64.209	2,417.376	-13.207

190	4.924	0.0005	64.209	2,417.376	-8.524
200	4.700	0.0001	64.209	2,417.376	-3.527
205.714	4.507	0.0000	64.209	2,417.376	-0.588
210	4.333	0.0000	35.775	1,346.860	0.910
220	3.834	0.0004	35.775	1,346.860	3.786
230	3.219	0.0011	35.775	1,346.860	6.569
231.429	3.122	0.0012	35.775	1,346.860	6.953
240	2.505	0.0021	35.775	1,346.860	9.163
250	1.714	0.0034	35.775	1,346.860	11.478
257.143	1.115	0.0045	35.775	1,346.860	12.915
260	0.870	0.0050	35.775	1,346.860	13.432
270	0.000	0.0068	35.775	1,346.860	14.954
280	0.870	0.0088	35.775	1,346.860	15.991
282.857	1.115	0.0094	35.775	1,346.860	16.193
290	1.714	0.0109	35.775	1,346.860	16.508
300	2.505	0.0129	35.775	1,346.860	16.492
308.571	3.122	0.0147	35.775	1,346.860	16.057
310	3.219	0.0150	35.775	1,346.860	15.947
320	3.834	0.0170	35.775	1,346.860	14.901
330	4.333	0.0188	35.775	1,346.860	13.396
334.286	4.507	0.0195	35.775	1,346.860	12.624
340	4.700	0.0204	35.775	1,346.860	11.489
350	4.924	0.0218	35.775	1,346.860	9.249
360	5.000	0.0229	35.775	1,346.860	6.752

Appendix E.6 Torque analysis of symmetrical wobble plate compressor for piston 6

Angle Shaft Rotation (Deg)	**Tilting Angle Wobble Plate (Deg)**	**Stroke of Compressor (m)**	**Pressure Distribution in Cylinder (Bar)**	**Force Distribution of Piston (N)**	**Distribution of Total Torque Piston 1 (Nm)**
0	5.000	0.0147	64.209	1,525.201	17.762
10	4.924	0.0167	64.209	1,525.201	16.508
20	4.700	0.0186	64.209	1,525.201	14.798
25.7143	4.507	0.0195	64.209	1,525.201	13.642
30	4.333	0.0202	64.209	1,525.201	12.700
40	3.834	0.0216	64.209	1,525.201	10.290
50	3.219	0.0227	64.209	1,525.201	7.642
51.4286	3.122	0.0229	64.209	1,525.201	7.248
60	2.505	0.0235	64.209	1,525.201	4.828
70	1.714	0.0240	64.209	1,525.201	1.921
77.1429	1.115	0.0241	64.209	1,525.201	-0.176
80	0.870	0.0240	64.261	1,526.444	-1.014

Distribution torque analysis symmetrical wobble plate compressor for 7 stages

90	0.000	0.0238	65.253	1,550.007	-3.971
100	0.870	0.0231	67.573	1,605.095	-7.050
102.857	1.115	0.0229	68.501	1,627.147	-7.968
110	1.714	0.0221	71.398	1,695.953	-10.371
120	2.505	0.0208	77.042	1,830.033	-14.072
128.571	3.122	0.0195	83.704	1,988.264	-17.678
130	3.219	0.0193	85.010	2,019.300	-18.328
140	3.834	0.0175	96.094	2,282.583	-23.365
150	4.333	0.0156	111.552	2,649.769	-29.494
154.286	4.507	0.0147	115.244	2,737.466	-31.273
160	4.700	0.0135	115.244	2,737.466	-32.097
170	4.924	0.0114	115.244	2,737.466	-32.817
180	5.000	0.0094	115.244	2,737.466	-32.564
190	4.924	0.0074	115.244	2,737.466	-31.298
200	4.700	0.0055	115.244	2,737.466	-29.011
205.714	4.507	0.0045	115.244	2,737.466	-27.257
210	4.333	0.0038	115.244	2,737.466	-25.735
220	3.834	0.0024	115.244	2,737.466	-21.542
230	3.219	0.0013	115.244	2,737.466	-16.551
231.429	3.122	0.0012	115.244	2,737.466	-15.782
240	2.505	0.0005	115.244	2,737.466	-10.922
250	1.714	0.0001	115.244	2,737.466	-4.852
257.143	1.115	0.0000	115.244	2,737.466	-0.370
260	0.870	0.0000	64.209	1,525.201	0.799
270	0.000	0.0003	64.209	1,525.201	4.288
280	0.870	0.0009	64.209	1,525.201	7.629
282.857	1.115	0.0012	64.209	1,525.201	8.538
290	1.714	0.0019	64.209	1,525.201	10.693
300	2.505	0.0032	64.209	1,525.201	13.366
308.571	3.122	0.0045	64.209	1,525.201	15.276
310	3.219	0.0048	64.209	1,525.201	15.556
320	3.834	0.0065	64.209	1,525.201	17.192
330	4.333	0.0085	64.209	1,525.201	18.234
334.286	4.507	0.0094	64.209	1,525.201	18.494
340	4.700	0.0105	64.209	1,525.201	18.666
350	4.924	0.0126	64.209	1,525.201	18.498
360	5.000	0.0147	64.209	1,525.201	17.762

Appendix E.7 Torque analysis of symmetrical wobble plate compressor for piston 7

Angle Shaft Rotation (Deg)	Tilting Angle Wobble Plate (Deg)	Stroke of Compressor (m)	Pressure Distribution in Cylinder (Bar)	Force Distribution of Piston (N)	Distribution of Total Torque Piston 1 (Nm)
0	4.924	0.0045	115.244	1,727.157	18.554
10	4.700	0.0063	115.244	1,727.157	19.739
20	4.507	0.0082	115.244	1,727.157	20.149
25.7143	4.333	0.0094	115.244	1,727.157	20.330
30	3.834	0.0102	115.244	1,727.157	20.332
40	3.219	0.0123	115.244	1,727.157	19.763
50	3.122	0.0144	115.244	1,727.157	19.637
51.4286	2.505	0.0147	115.244	1,727.157	18.654
60	1.714	0.0164	115.244	1,727.157	17.044
70	1.115	0.0183	115.244	1,727.157	15.613
77.1429	0.870	0.0195	115.244	1,727.157	14.981
80	0.000	0.0200	115.244	1,727.157	12.517
90	0.870	0.0214	115.244	1,727.157	9.714
100	1.115	0.0226	115.244	1,727.157	8.860
102.857	1.714	0.0229	115.244	1,727.157	6.637
110	2.505	0.0234	115.244	1,727.157	3.359
120	3.122	0.0239	115.244	1,727.157	0.447
128.571	3.219	0.0241	115.244	1,727.157	-0.044
130	3.834	0.0239	115.267	1,727.505	-3.531
140	4.333	0.0238	116.720	1,749.278	-7.197
150	4.507	0.0232	120.532	1,806.409	-8.853
154.286	4.700	0.0229	122.952	1,842.681	-11.166
160	4.924	0.0223	126.993	1,903.231	-15.583
170	5.000	0.0211	136.621	2,047.528	-16.288
180	4.924	0.0195	150.257	2,251.895	-20.943
190	4.700	0.0178	169.225	2,536.173	-26.514
200	4.507	0.0159	195.623	2,931.789	-30.237
205.714	4.333	0.0147	206.843	3,099.941	-33.348
210	3.834	0.0138	206.843	3,099.941	-38.342
220	3.219	0.0117	206.843	3,099.941	-37.926
230	3.122	0.0097	206.843	3,099.941	-37.763
231.429	2.505	0.0094	206.843	3,099.941	-36.243
240	1.714	0.0077	206.843	3,099.941	-33.328
250	1.115	0.0058	206.843	3,099.941	-30.538
257.143	0.870	0.0045	206.843	3,099.941	-29.269
260	0.000	0.0041	206.843	3,099.941	-24.205
270	0.870	0.0026	206.843	3,099.941	-18.322
280	1.115	0.0015	206.843	3,099.941	-16.520

Distribution torque analysis symmetrical wobble plate compressor for 7 stages

282.857	1.714	0.0012	206.843	3,099.941	-11.840
290	2.505	0.0006	206.843	3,099.941	-5.006
300	3.122	0.0001	206.843	3,099.941	0.941
308.571	3.219	0.0000	206.843	3,099.941	1.074
310	3.834	0.0000	115.244	1,727.157	4.854
320	4.333	0.0002	115.244	1,727.157	8.420
330	4.507	0.0008	115.244	1,727.157	9.854
334.286	4.700	0.0012	115.244	1,727.157	11.658
340	4.924	0.0018	115.244	1,727.157	14.472
350	5.000	0.0030	115.244	1,727.157	16.788
360	5.000	0.0045	115.244	1,727.157	16.788

APPENDIX F

Total Torque of Compressor with Variation of Tilting Angle

APPENDIX F

Total Torque of Compressor With Variation of Tilting Angle

Shaft Angle Rotation	Total Torque of Compressor With Variation of Tilting Angle										
	5	7	9	11	13	15	17	19	21	23	25
0	-6.149	-6.527	-6.407	-6.332	-6.456	-6.518	-6.569	-6.667	-6.691	-6.999	-7.246
10	-8.047	-8.752	-8.765	-8.807	-9.141	-9.409	-9.553	-9.780	-9.958	-10.549	-11.044
20	-10.435	-11.466	-11.577	-11.705	-12.230	-12.677	-12.903	-13.248	-13.551	-14.410	-15.133
30	-13.476	-14.820	-14.972	-15.139	-15.819	-16.396	-16.684	-17.128	-17.508	-18.602	-19.512
36	-15.710	-17.237	-17.378	-17.540	-18.291	-18.917	-19.231	-19.721	-20.120	-21.333	-22.332
40	-17.413	-19.062	-19.179	-19.323	-20.113	-20.758	-21.085	-21.601	-21.997	-23.283	-24.331
50	-22.689	-24.652	-24.645	-24.694	-25.548	-26.196	-26.536	-27.101	-27.445	-28.892	-30.033
60	-22.921	-24.883	-24.885	-24.971	-25.885	-26.609	-27.002	-27.626	-28.102	-29.745	-31.120
70	-21.881	-23.635	-23.539	-23.542	-24.318	-24.903	-25.236	-25.776	-26.150	-27.615	-28.835
72	-21.641	-23.342	-23.220	-23.200	-23.939	-24.487	-24.804	-25.322	-25.668	-27.087	-28.266
80	-22.699	-24.261	-23.944	-23.762	-24.337	-24.691	-24.929	-25.351	-25.523	-26.762	-27.758
90	-24.338	-25.794	-25.278	-24.944	-25.389	-25.583	-25.764	-26.120	-26.168	-27.320	-28.234
100	-26.372	-27.722	-26.980	-26.472	-26.774	-26.789	-26.908	-27.192	-27.099	-28.161	-28.987
108	-28.545	-29.843	-28.907	-28.252	-28.449	-28.325	-28.398	-28.631	-28.426	-29.440	-30.215
110	-29.196	-30.489	-29.504	-28.812	-28.986	-28.828	-28.891	-29.114	-28.881	-29.891	-30.657
120	-33.385	-34.728	-33.489	-32.608	-32.697	-32.396	-32.420	-32.611	-32.254	-33.292	-34.063
130	-33.636	-34.835	-33.435	-32.394	-32.291	-31.776	-31.691	-31.757	-31.168	-31.905	-32.360
140	-31.215	-32.255	-30.892	-29.866	-29.697	-29.136	-29.019	-29.034	-28.408	-28.987	-29.305
144	-30.203	-31.201	-29.872	-28.870	-28.695	-28.139	-28.019	-28.025	-27.405	-27.947	-28.235
150	-30.936	-32.168	-30.993	-30.135	-30.164	-29.825	-29.808	-29.944	-29.524	-30.362	-30.938
160	-31.758	-33.175	-32.094	-31.313	-31.468	-31.254	-31.291	-31.502	-31.173	-32.165	-32.875
170	-33.059	-34.740	-33.786	-33.117	-33.454	-33.423	-33.542	-33.864	-33.678	-34.913	-35.839
180	-35.238	-37.274	-36.463	-35.922	-36.494	-36.695	-36.921	-37.391	-37.386	-38.955	-40.181
190	-38.851	-41.354	-40.676	-40.261	-41.118	-41.586	-41.939	-42.589	-42.787	-44.780	-46.378
200	-39.649	-42.479	-42.028	-41.816	-42.957	-43.730	-44.223	-45.054	-45.529	-47.924	-49.914
210	-34.897	-37.669	-37.514	-37.539	-38.805	-39.773	-40.335	-41.226	-41.899	-44.340	-46.415

APPENDIX F
Total Torque of Compressor With Variation of Tilting Angle

216	-31.579	-34.244	-34.239	-34.380	-35.673	-36.712	-37.292	-38.189	-38.943	-41.343	-43.407
220	-30.314	-32.787	-32.697	-32.745	-33.876	-34.750	-35.244	-36.030	-36.618	-38.741	-40.529
230	-27.724	-30.092	-30.099	-30.218	-31.346	-32.248	-32.744	-33.519	-34.143	-36.199	-37.944
240	-24.711	-26.847	-26.875	-26.998	-28.025	-28.850	-29.301	-30.003	-30.577	-32.431	-34.007
250	-21.592	-23.399	-23.370	-23.431	-24.268	-24.923	-25.287	-25.863	-26.302	-27.840	-29.137
252	-20.991	-22.724	-22.675	-22.716	-23.507	-24.118	-24.461	-25.007	-25.411	-26.876	-28.106
260	-18.800	-20.224	-20.070	-20.011	-20.601	-21.017	-21.266	-21.681	-21.926	-23.083	-24.033
270	-15.545	-16.521	-16.228	-16.043	-16.361	-16.523	-16.652	-16.899	-16.955	-17.724	-18.336
280	-9.064	-9.378	-8.986	-8.685	-8.631	-8.459	-8.416	-8.412	-8.205	-8.337	-8.382
288	-3.938	-3.736	-3.273	-2.888	-2.546	-2.121	-1.947	-1.749	-1.344	-0.987	-0.599
290	-3.925	-3.813	-3.439	-3.142	-2.912	-2.613	-2.502	-2.380	-2.114	-1.942	-1.752
300	-2.693	-2.379	-1.924	-1.554	-1.191	-0.762	-0.592	-0.387	-0.020	0.338	0.692
310	-2.181	-1.754	-1.242	-0.823	-0.382	0.128	0.333	0.586	1.015	1.475	1.921
320	-2.674	-2.251	-1.708	-1.267	-0.813	-0.285	-0.074	0.184	0.630	1.090	1.539
324	-3.223	-2.841	-2.293	-1.849	-1.412	-0.895	-0.691	-0.444	-0.004	0.424	0.846
330	-4.503	-4.228	-3.679	-3.239	-2.851	-2.371	-2.188	-1.974	-1.558	-1.217	-0.865
340	-7.678	-7.703	-7.176	-6.767	-6.531	-6.181	-6.065	-5.955	-5.633	-5.556	-5.433
350	-6.797	-6.980	-6.649	-6.400	-6.331	-6.174	-6.136	-6.121	-5.961	-6.058	-6.102
360	-6.149	-6.527	-6.407	-6.332	-6.456	-6.518	-6.569	-6.667	-6.691	-6.999	-7.246
Total	-908.420	-962.791	-943.452	-930.820	-947.226	-954.212	-960.799	-973.884	-975.260	-1,017.665	-1,051.086

APPENDIX F
Total Torque of Compressor With Variation of Tilting Angle

Continued table 6.44

Shaft Angle Rotation	Total Torque of Compressor With Variation of Tilting Angle									
	27	**29**	**31**	**33**	**35**	**37**	**39**	**41**	**43**	**45**
0	-7.081	-7.575	-7.663	-7.704	-7.702	-8.217	-8.262	-8.272	-8.774	-9.375
10	-10.924	-11.750	-11.965	-12.092	-12.134	-12.974	-13.100	-13.153	-13.929	-14.876
20	-15.017	-16.165	-16.478	-16.658	-16.710	-17.850	-18.012	-18.061	-19.069	-20.238
30	-19.337	-20.776	-21.134	-21.314	-21.324	-22.709	-22.840	-22.819	-23.986	-25.208
36	-22.083	-23.679	-24.035	-24.186	-24.139	-25.643	-25.720	-25.621	-26.853	-28.032
40	-24.014	-25.709	-26.050	-26.167	-26.070	-27.643	-27.669	-27.505	-28.771	-29.898
50	-29.470	-31.409	-31.665	-31.648	-31.375	-33.105	-32.949	-32.567	-33.897	-34.822
60	-30.761	-33.080	-33.675	-34.025	-34.139	-36.503	-36.850	-36.994	-39.182	-41.193
70	-28.444	-30.565	-31.089	-31.397	-31.499	-33.689	-34.014	-34.166	-36.243	-38.838
72	-27.863	-29.931	-30.433	-30.726	-30.819	-32.959	-33.272	-33.418	-35.456	-38.006
80	-27.182	-29.067	-29.402	-29.540	-29.491	-31.397	-31.524	-31.494	-33.275	-35.295
90	-27.537	-29.408	-29.698	-29.806	-29.748	-31.683	-31.802	-31.784	-33.665	-35.865
100	-28.143	-30.009	-30.244	-30.316	-30.240	-32.213	-32.317	-32.304	-34.300	-36.694
108	-29.238	-31.141	-31.338	-31.385	-31.296	-33.348	-33.449	-33.448	-35.594	-38.238
110	-29.645	-31.566	-31.756	-31.797	-31.705	-33.786	-33.888	-33.892	-36.086	-38.810
120	-32.847	-34.938	-35.103	-35.119	-35.007	-37.314	-37.265	-36.907	-38.910	-40.887
130	-30.906	-32.569	-32.384	-32.042	-31.564	-33.220	-32.851	-32.375	-33.999	-35.394
140	-27.885	-29.296	-29.030	-28.624	-28.100	-29.472	-29.028	-28.493	-29.824	-30.812
144	-26.846	-28.185	-27.907	-27.494	-26.967	-28.259	-27.805	-27.265	-28.513	-29.395
150	-29.703	-31.425	-31.389	-31.198	-30.869	-32.632	-32.436	-32.132	-33.889	-35.644
160	-31.670	-33.559	-33.585	-33.430	-33.114	-35.027	-34.856	-34.554	-36.420	-38.281
170	-34.695	-36.870	-37.022	-36.960	-36.705	-38.912	-38.837	-38.603	-40.735	-42.978
180	-39.105	-41.694	-42.025	-42.101	-41.942	-44.588	-44.661	-44.539	-47.090	-49.950

APPENDIX F
Total Torque of Compressor With Variation of Tilting Angle

190	-45.336	-48.467	-49.000	-49.219	-49.145	-52.346	-52.556	-52.519	-55.570	-59.066
200	-49.098	-52.736	-53.601	-54.122	-54.318	-58.146	-58.724	-59.027	-62.753	-67.457
210	-45.904	-49.492	-50.511	-51.198	-51.564	-55.378	-56.134	-56.614	-60.327	-65.184
216	-43.066	-46.538	-47.615	-48.377	-48.832	-52.554	-53.398	-53.977	-57.621	-62.514
220	-40.059	-43.128	-43.953	-44.475	-44.705	-47.904	-48.456	-48.756	-51.800	-55.643
230	-37.582	-40.518	-41.359	-41.914	-42.193	-45.277	-45.878	-46.242	-49.204	-53.060
240	-33.697	-36.339	-37.107	-37.620	-37.888	-40.681	-41.253	-41.621	-44.336	-47.964
250	-28.812	-31.022	-31.623	-32.009	-32.190	-34.518	-34.954	-35.222	-37.492	-40.517
252	-27.770	-29.882	-30.441	-30.792	-30.947	-33.164	-33.560	-33.795	-35.954	-38.810
260	-23.630	-25.337	-25.707	-25.901	-25.931	-27.683	-27.887	-27.955	-29.628	-31.687
270	-17.905	-19.128	-19.324	-19.400	-19.363	-20.620	-20.701	-20.691	-21.904	-23.327
280	-7.922	-8.250	-8.090	-7.877	-7.619	-7.852	-7.574	-7.253	-7.378	-7.068
288	-0.130	0.226	0.646	1.063	1.470	1.999	2.520	3.039	3.695	5.181
290	-1.435	-1.354	-1.159	-0.974	-0.803	-0.695	-0.490	-0.299	-0.203	0.127
300	1.045	1.350	1.636	1.886	2.096	2.428	2.692	2.918	3.201	3.755
310	2.299	2.719	3.053	3.335	3.560	3.997	4.283	4.514	4.867	5.479
320	1.945	2.349	2.690	2.979	3.211	3.628	3.919	4.154	4.479	5.054
324	1.266	1.623	1.956	2.243	2.478	2.850	3.142	3.383	3.670	4.213
330	-0.420	-0.184	0.125	0.402	0.641	0.899	1.190	1.442	1.635	2.099
340	-4.967	-5.097	-4.905	-4.708	-4.513	-4.638	-4.436	-4.240	-4.395	-4.196
350	-5.781	-6.067	-6.002	-5.914	-5.806	-6.096	-6.004	-5.898	-6.198	-6.446
360	-7.081	-7.575	-7.663	-7.704	-7.702	-8.217	-8.262	-8.272	-8.774	-9.375
Total	-1,024.438	-1,093.211	-1,103.022	-1,106.025	-1,102.722	-1,173.109	-1,175.927	-1,173.300	-1,240.448	-1,315.136

APPENDIX G

Complete Engineering Drawing

Master Bill of Materials: First Prototype of 10m^3/hr Compressor
Date: 16-09-2004

Standard parts

System	Part Name	Part Number	Quantity	Supplier
Piston Assembly	Male end joint	SAKB 14 F	10	SKF
	Female end joint	SIKB 14 F	10	SKF
	End joint retainer	Truarc N5000-56-S0.562	30	
	Bolt	ISO 4762 M4 x 30 - 30N	10	
	Piston ring 1	ER-39-850-39-215	4	Eriks
	Piston ring 2	ER-39-850-28-209	6	Eriks
	Piston ring 3	ER-39-850-20-112	10	Eriks
	Piston ring 4	ER-39-850-15-012	12	Eriks
	Piston guide strip 1 & 2	ER-81-216-6.1-2.5	4	Eriks
	Piston guide strip 3	ER-81-214-4.0-2.5	2	Eriks
	Piston guide strip 4	ER-81-212-3.9-1.5	2	Eriks
Shaft Assembly	Shaft support bearing	22205/20E SKF Explorer bearing	2	SKF
	Rotor bearing	6208 SKF Explorer bearing	2	SKF
	Rotor bearing retainer	Truarc 5103-150	2	
	Wp bearing retainer	Truarc N5000-315 - S3.149	2	
	Shaft seal	211-8967 (20x35x7)	1	SKF
Anti-rotating Assembly	Female end joint (for balls of anti-rotating)	SILKB 14 F	2	NSK (Japan)
Valve	Spring		-	Tmn U
Reference	Standard of Compressor			

Customised parts

System	Part Name	Part/Drawing Number	Quantity	Material	Supplier
Piston Assembly	Bush Piston	Dwg. No.: 10P1-Bush Piston	20	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Joint Piston 1	Dwg. No.: 10P1-Joint Piston1	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Joint Piston 2	Dwg. No.: 10P1-Joint Piston2	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Joint Piston 3	Dwg. No.: 10P1-Joint Piston3	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Joint Piston 4	Dwg. No.: 10P1-Joint Piston4	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Joint Piston 5	Dwg. No.: 10P1-Joint Piston5	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Piston 1	Dwg. No.: 10P1-Piston1	2	Alloy Aluminum 6061	Robaco Sdn. Bhd.
	Piston 2	Dwg. No.: 10P1-Piston2	2	Alloy Aluminum 6061	Robaco Sdn. Bhd.
	Piston 3	Dwg. No.: 10P1-Piston3	2	Alloy Aluminum 6061	Robaco Sdn. Bhd.
	Piston 4	Dwg. No.: 10P1-Piston4	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Piston 5	Dwg. No.: 10P1-Piston5	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Pin Piston	Dwg. No.: 10P1-Pin Piston	10	Stainless Steel AISI 304	Robaco Sdn. Bhd.
Shaft Assembly	Shaft	Dwg. No.: 10P1-Shaft	1	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Rotor	Dwg. No.: 10P1-Rotor	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Pin Shaft	Dwg. No.: 10P1-Pin Shaft	2	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Wobble Plate	Dwg. No.: 10P1-Wobble Plate	2	Alloy Aluminum 5083	Robaco Sdn. Bhd.
	Pin Wobble Plate	Dwg. No.: 10P1-Pin WbPlate	10	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Bush Plate Bottom	Dwg. No.: 10P1-Bush Plate Bottom	10	Stainless Steel AISI 304	Robaco Sdn. Bhd.
	Bush Plate Top	Dwg. No.: 10P1-Bush Plate Top	10	Stainless Steel AISI 304	Robaco Sdn. Bhd.
Anti-rotating Assembly	Rod Anti-rotating	Dwg. No.: 10P1-Rod Antirotating	1	Stainless Steel AISI 304	Robaco Sdn. Bhd.
Housing Assembly	Housing	Dwg. No.: 10P1-Housing	1	Aluminum LM25	APP Sdn. Bhd.
	Transparent Cover	Dwg. No.: 10P1-Cover	1	Aluminum LM25	UTM
End Plate Assembly	End Plate Left	Dwg. No.: 10P1-End Plate Right	1	Aluminum LM25	APP Sdn. Bhd.
	End Plate Right	Dwg. No.: 10P1-End Plate Left	1	Aluminum LM25	APP Sdn. Bhd.
	Cylinder Block 1	Dwg. No.: 10P1-Cylinder Block 1	2	Aluminum 6061	Robaco Sdn. Bhd.
	Cylinder Block 2	Dwg. No.: 10P1-Cylinder Block 2	2	Aluminum 6061	Robaco Sdn. Bhd.
	Cylinder Block 3	Dwg. No.: 10P1-Cylinder Block 3	2	Aluminum 6061	Robaco Sdn. Bhd.
	Cylinder Block 4	Dwg. No.: 10P1-Cylinder Block 4	2	Aluminum 6061	Robaco Sdn. Bhd.
	Cylinder Block 5	Dwg. No.: 10P1-Cylinder Block 5	2	Aluminum 6061	Robaco Sdn. Bhd.
	Liner 1	Dwg. No.: 10P1-Liner 1	2	Cast Iron	Robaco Sdn. Bhd.
	Liner 2	Dwg. No.: 10P1-Liner 2	2	Cast Iron	Robaco Sdn. Bhd.
	Liner 3	Dwg. No.: 10P1-Liner 3	2	Cast Iron	Robaco Sdn. Bhd.
	Liner 4	Dwg. No.: 10P1-Liner 4	2	Cast Iron	Robaco Sdn. Bhd.
	Liner 5	Dwg. No.: 10P1-Liner 5	2	Teflon	Robaco Sdn. Bhd.
	Valve Retainer 1	Dwg. No.: 10P1-Valve 1	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Retainer 2	Dwg. No.: 10P1-Valve 2	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Retainer 3	Dwg. No.: 10P1-Valve 3	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Retainer 4	Dwg. No.: 10P1-Valve 4	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Retainer 5	Dwg. No.: 10P1-Valve 5	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 1	Dwg. No.: 10P1-Valve Plate 1	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 2	Dwg. No.: 10P1-Valve Plate 2	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 3	Dwg. No.: 10P1-Valve Plate 3	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 4	Dwg. No.: 10P1-Valve Plate 4	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 5	Dwg. No.: 10P1-Valve Plate 5	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 1 Suction	Dwg. No.: 10P1-Valve Plate S1	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 2 Suction	Dwg. No.: 10P1-Valve Plate S2	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 3 Suction	Dwg. No.: 10P1-Valve Plate S3	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 4 Suction	Dwg. No.: 10P1-Valve Plate S4	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Plate 5 Suction	Dwg. No.: 10P1-Valve Plate S5	2	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Seat 1	Dwg. No.: 10P1-Valve Seat 1	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Seat 2	Dwg. No.: 10P1-Valve Seat 2	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Seat 3	Dwg. No.: 10P1-Valve Seat 3	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Seat 4	Dwg. No.: 10P1-Valve Seat 4	1	Aluminum 6061	Robaco Sdn. Bhd.
	Valve Seat 5	Dwg. No.: 10P1-Valve Seat 5	1	Aluminum 6061	Robaco Sdn. Bhd.

Material Porperties:

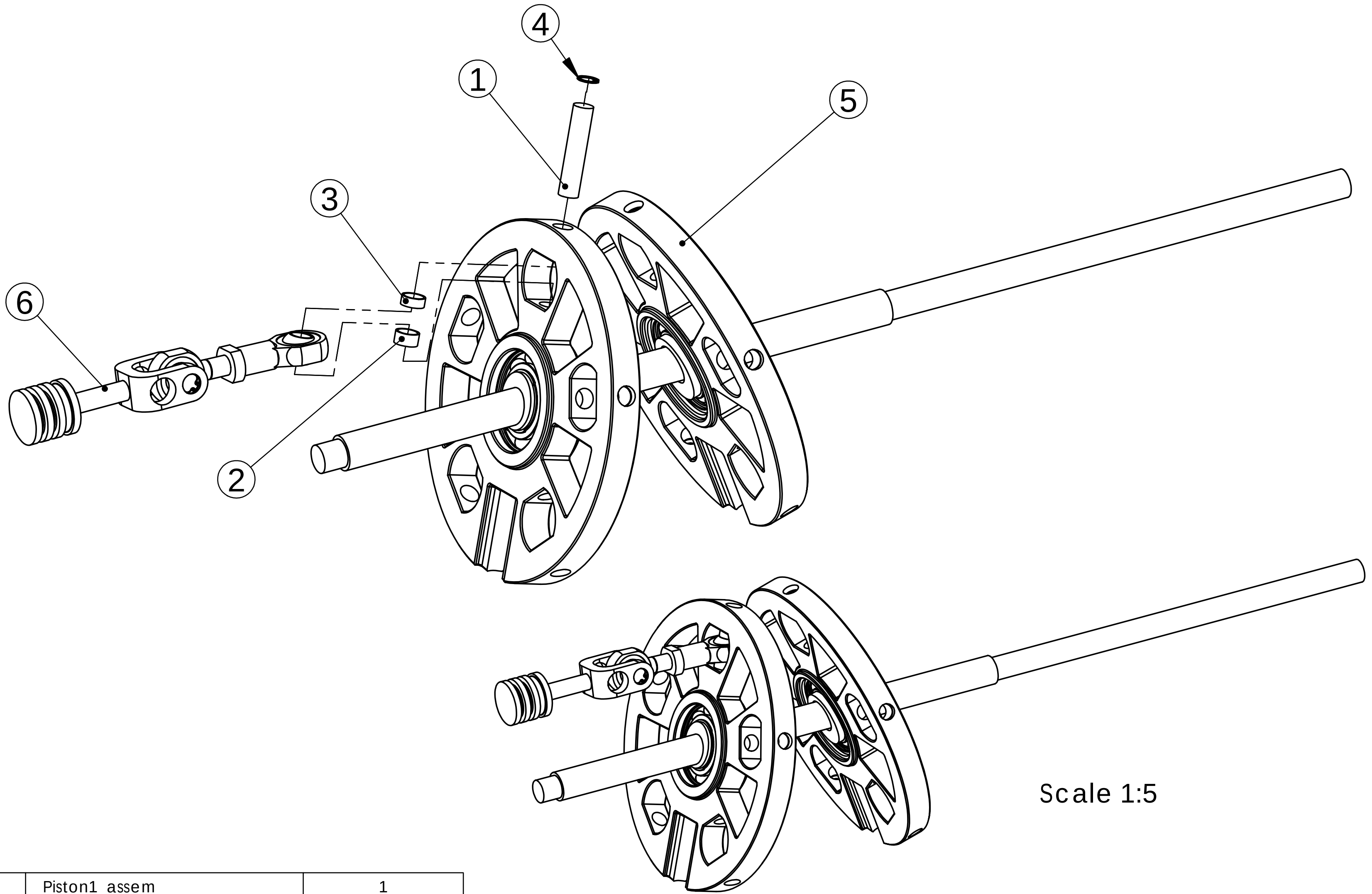
1. Alloy Aluminum 6061		2.Stainless Steel AISI 304	
E	6.9e10 Pa	E	1.9e11 Pa
Poisson ration	0.33	Poisson ration	0.29
Density	2700 kg/m3	Density	7700 kg/m3
Yield strength	5.515e7 Pa	Yield strength	2.068e8 Pa

Tolerance of Parts for 10Nm^3/hr First Prototype

No.	Dwg. No.	Rev.	Description of Tolerance	Date
1.	10P1-Wobble Plate	1	Hole diameter of anti-rotating sliding ball. Dominant dimension: **25.40mm** Upper limit: +0.04 Lower limit: 0 Standard: Standard running/sliding fit **RC6**.	24-05-04
	10P1-Wobble Plate	1	Housing diameter of bearing. Dominant dimension: **80.00mm** Upper limit: +0.010 Lower limit: -0.025 Standard: SKF Bearing application (housing) **K7** (ISO 286-2:1988).	24-05-04
2.	10P1-Rotor	1	Shaft diameter of bearing. Dominant dimension: **40.00mm** Upper limit: +0.013 Lower limit: +0.002 Standard: SKF Bearing application (shaft) **K5** (ISO 286-2:1988).	26-05-04
3.	10P1-Piston1	2	Width of groove for piston ring. Dominant dimension: **4.20mm** Upper limit: +0.20 Lower limit: -0.00 Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston1	2	Width of groove for guide strip. Dominant dimension: **6.30mm** Upper limit: +0.25 Lower limit: -0.00 Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston1	2	Fillet radius of internal corners for grooves of ring. Dominant dimension: **$0.80 \leq R \leq 1.20$mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston1	2	Fillet radius of internal corners for grooves of guide strip. Dominant dimension: **Max. R 0.30mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04
4.	10P1-Piston2	2	Width of groove for piston ring. Dominant dimension: **4.20mm** Upper limit: +0.20 Lower limit: -0.00	04-06-04

			Standard: Ericks (Fluid Sealing)	
	10P1-Piston2	2	Width of groove for guide strip. Dominant dimension: **6.30mm** Upper limit: +0.25 Lower limit: -0.00 Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston2	2	Fillet radius of internal corners for grooves of ring. Dominant dimension: **0.80 ≤ R ≤ 1.20mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston2	2	Fillet radius of internal corners for grooves of guide strip. Dominant dimension: **Max. R 0.30mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04
5.	10P1-Piston3	2	Width of groove for piston ring. Dominant dimension: **3.20mm** Upper limit: +0.20 Lower limit: -0.00 Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston3	2	Width of groove for guide strip. Dominant dimension: **4.20mm** Upper limit: +0.25 Lower limit: -0.00 Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston3	2	Fillet radius of internal corners for grooves of ring. Dominant dimension: **0.50 ≤ R ≤ 0.80mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston3	2	Fillet radius of internal corners for grooves of guide strip. Dominant dimension: **Max. R 0.30mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04
6.	10P1-Piston4	2	Width of groove for piston ring. Dominant dimension: **2.20mm** Upper limit: +0.20 Lower limit: -0.00 Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-	2	Width of groove for guide strip.	04-06-04

	Piston4		Dominant dimension: **4.10mm** Upper limit: +0.25 Lower limit: -0.00 Standard: Ericks (Fluid Sealing)	
	10P1-Piston4	2	Fillet radius of internal corners for grooves of ring. Dominant dimension: **0.30 ≤ R ≤ 0.50mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04
	10P1-Piston4	2	Fillet radius of internal corners for grooves of guide strip. Dominant dimension: **Max. R 0.30mm** Upper limit: - Lower limit: - Standard: Ericks (Fluid Sealing)	04-06-04



ITEM NO.	PART NAME	QTY.
6	Piston1_assem	1
5	Shaft_assem	1
4	Truarc N5000-56 - S0.562_end joint retainer	1
3	Bush_wplate_top	1
2	Bush_wplate_bottom	1
1	Pin_wplate	1/piston

BOM Table

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DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL
- -

FINISH
--

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	30-06-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

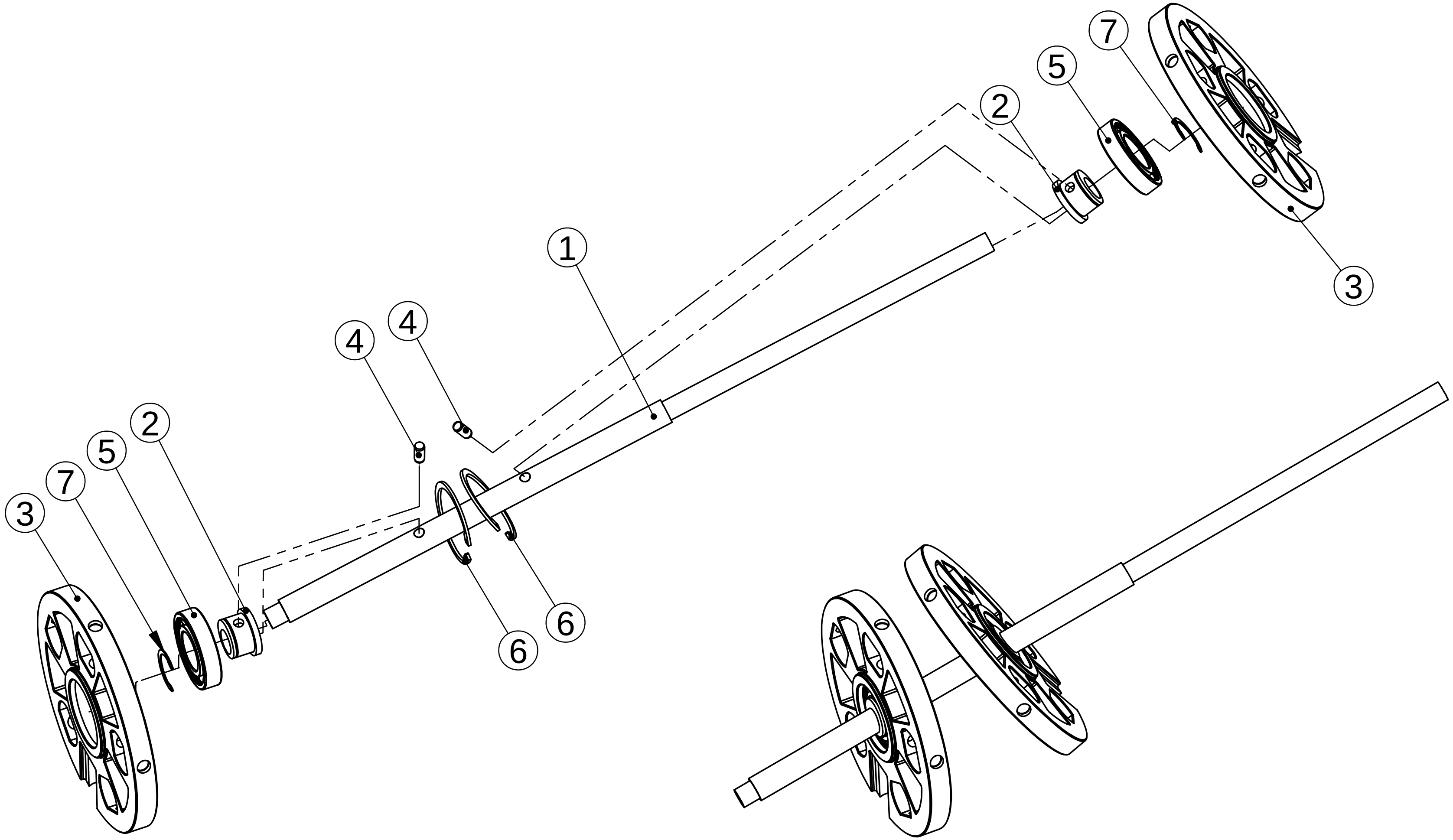
COMMENTS:
--

Universiti Teknologi Malaysia

NGV Refuelling Facilities & Equipment
Faculty of Mechanical Engineering
81310 UTM Skudai, Johor, Malaysia.
Telephone: +607-5534626
Telephone: +607-5566159

SIZE A | DWG. NO. **10P1-Wobble Assembly** | REV. 1

SCALE:1:4 | WEIGHT: | SHEET 1 OF 1



7	Truarc 5103-150_rotor bearing retainer	2
6	Truarc N5000-315 - S3.149_wp bearing retainer	2
5	Wobble_plate_bearing_6208	2
4	Shaft_pin_Parallel Pin ISO 8734 - 10 x 22 - A - St	2
3	Wobble Plate	2
2	Rotor	2
1	Shaft	1
ITEM NO.	PART NAME	QTY.

BOM Table

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TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL --

FINISH --

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	30-06-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS: --

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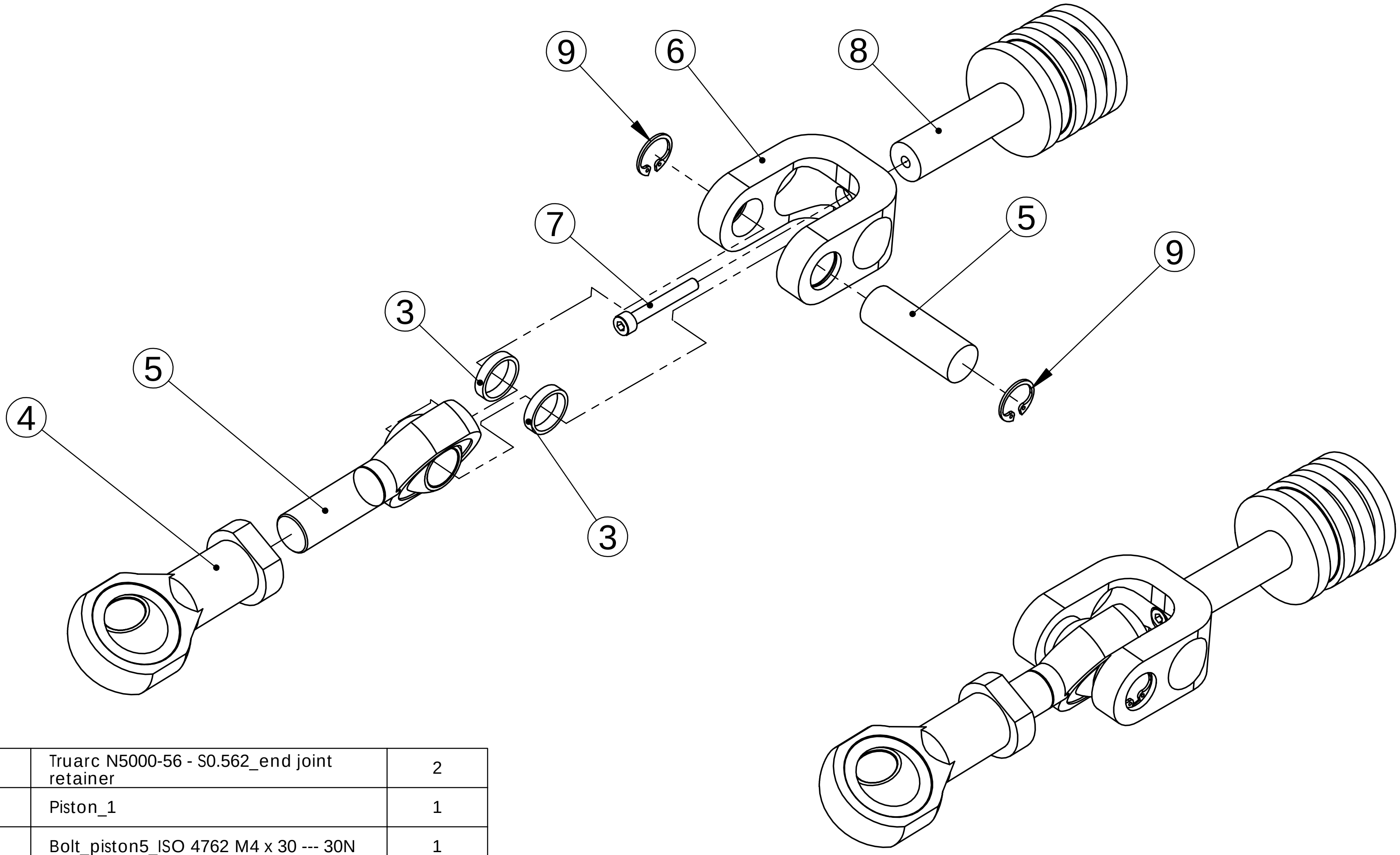
Faculty of Mechanical Engineering
81310 UTM Skudai, Johor, Malaysia.
Telephone: +607-5534626
Telephone: +607-5566159

SIZE A | DWG. NO. **10P1-Shaft Assembly** | REV. 1

SCALE:1:6 | WEIGHT: | SHEET 1 OF 1



9	Truarc N5000-56 - S0.562_end joint retainer	2
8	Piston_1	1
7	Bolt_piston5_ISO 4762 M4 x 30 --- 30N	1
6	Joint_Piston_1	1
5	Male_end_joint_SAKB_14_F	1
4	Female_end_joint_SIKB_14_F	1
5	Pin_piston	1
3	Bush_piston	2
ITEM NO.	PART NAME	QTY.

BOM Table

DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

MATERIAL --

FINISH --

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	29-06-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

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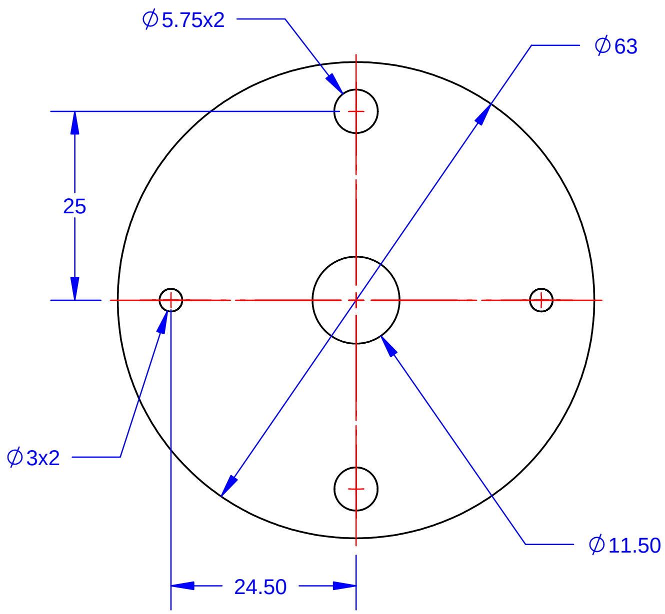
NGV Refuelling Facilities & Equipment

Faculty of Mechanical Engineering
81310 UTM Skudai, Johor, Malaysia.
Telephone: +607-5534626
Telephone: +607-5566159

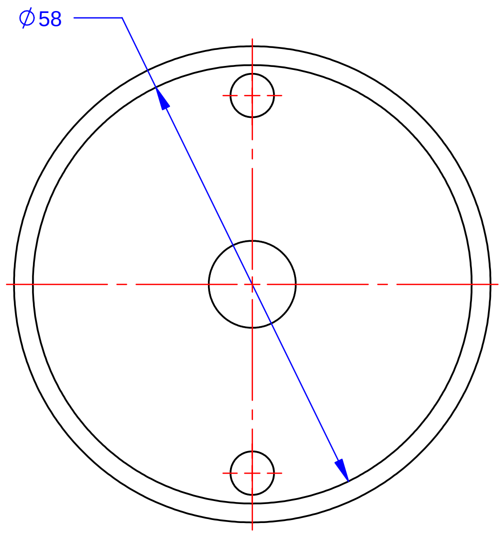
SIZE	DWG. NO.	REV.
A	**10P1-Piston1 Assy**	1

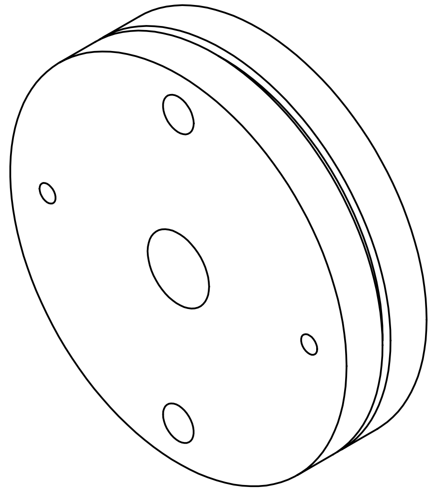
SCALE:1:2 | WEIGHT: | SHEET 1 OF 1

Top View

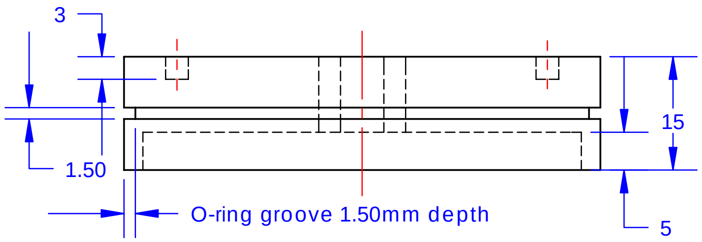


Bottom View





Isometric View



Front View

Chamfer of 0.5mm, 45deg at all edges

DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

	NAME	DATE
DRAWN	Ong KL	17-05-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

MATERIAL
Stainless Steel AISI 304

FINISH
--

DO NOT SCALE DRAWING

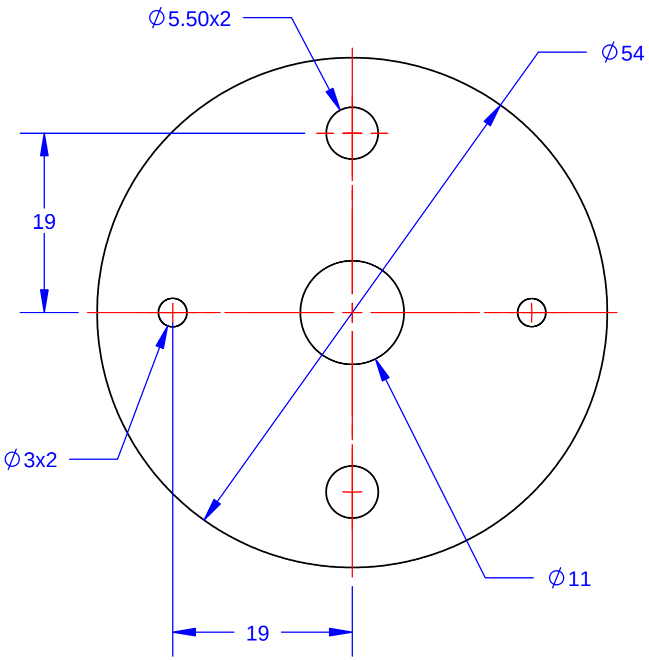
Universiti Teknologi Malaysia

NGV Refuelling Facilities & Equipment
Faculty of Mechanical Engineering
81310 UTM Skudai, Johor, Malaysia.
Telephone: +607-5534626
Telephone: +607-5566159

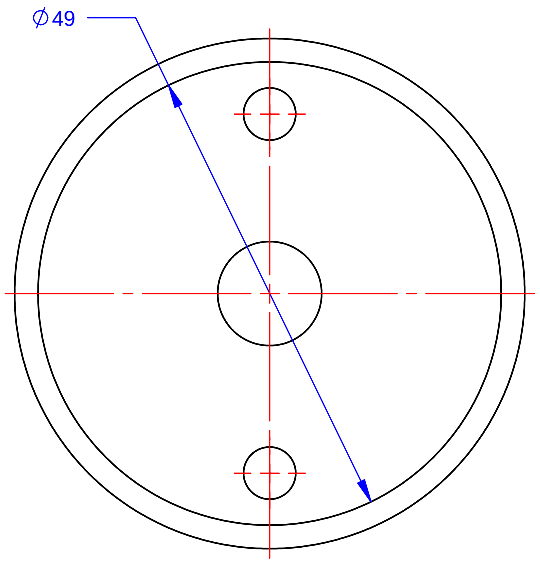
SIZE	DWG. NO.	REV.
A	10P1-Joint Piston1	1

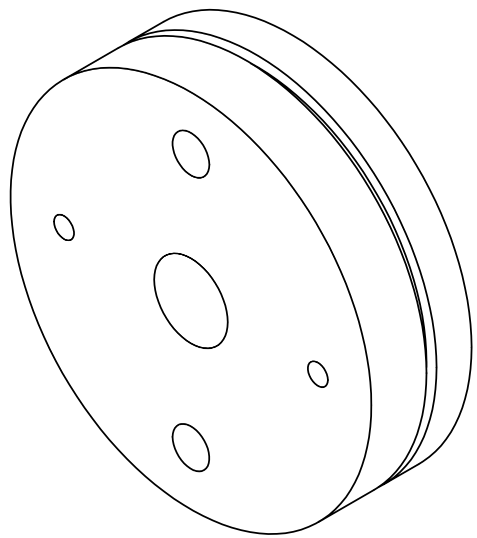
SCALE:1:1 | WEIGHT: | SHEET 1 OF 20

Top View

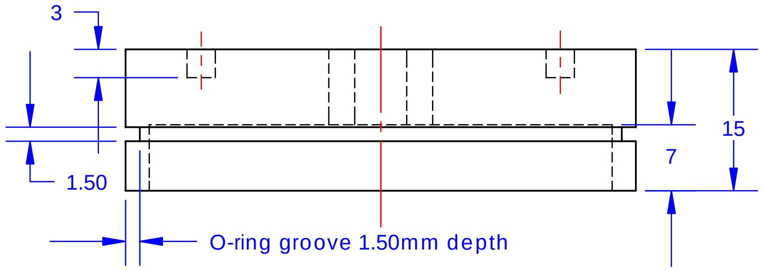


Bottom View





Isometric View



Front View

Chamfer of 0.5mm, 45deg at all edges

DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

MATERIAL
Stainless Steel AISI 304

FINISH
--

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	17-05-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

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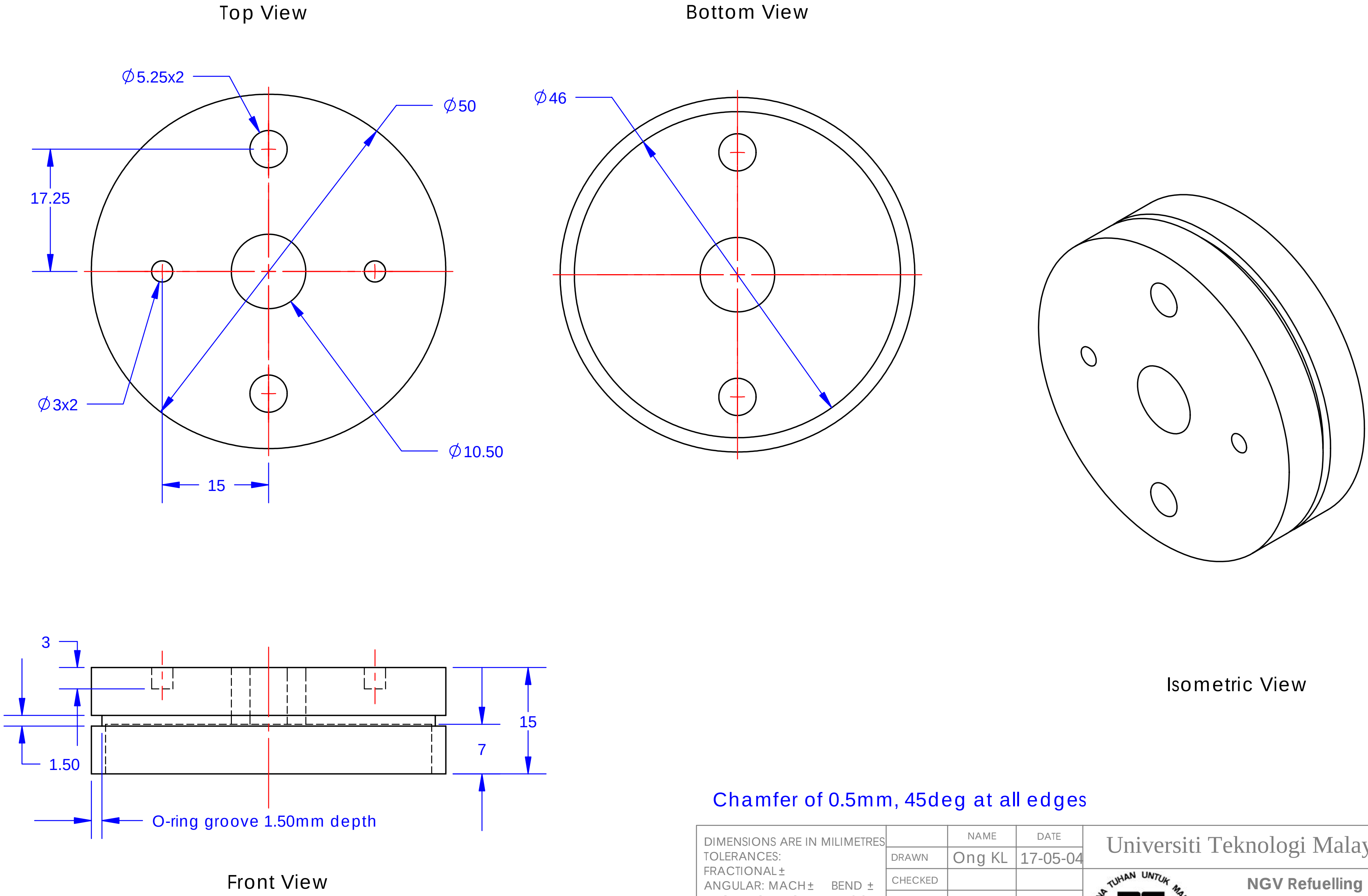
NGV Refuelling Facilities & Equipment

Faculty of Mechanical Engineering
81310 UTM Skudai, Johor, Malaysia.
Telephone: +607-5534626
Telephone: +607-5566159

SIZE	DWG. NO.	REV.
A	10P1-Joint Piston1	1

SCALE:1.25:1 WEIGHT: SHEET 2 OF 20



		NAME	DATE
DIMENSIONS ARE IN MILIMETRES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ±0.25 THREE PLACE DECIMAL ±	DRAWN	Ong KL	17-05-04
	CHECKED		
	ENG APPR.		
	MFG APPR.		
MATERIAL Stainless Steel AISI 304	Q.A.		
FINISH --	COMMENTS:		
DO NOT SCALE DRAWING			

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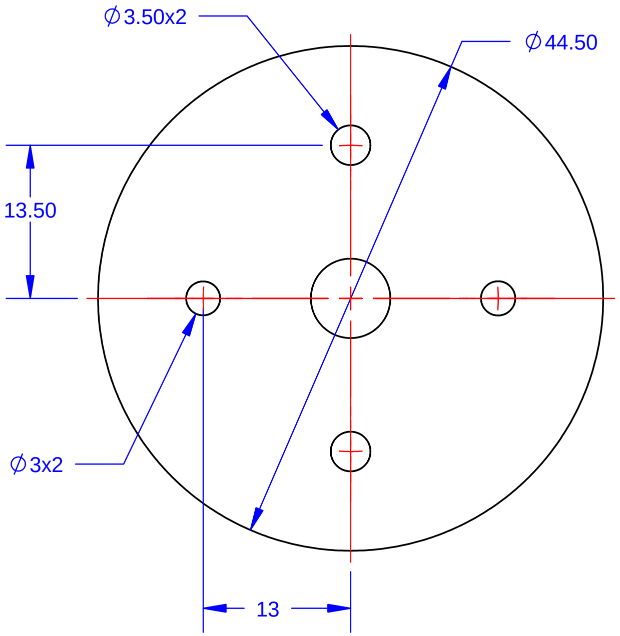
Telephone: +607-5566159

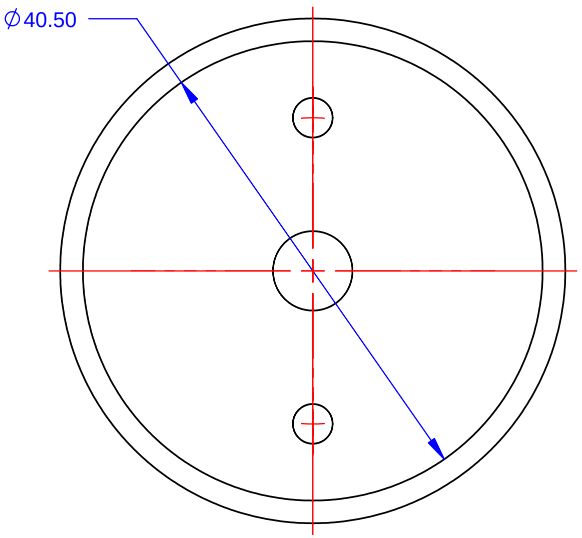
SIZE	DWG. NO.	REV.
A	10P1-Joint Piston1	1

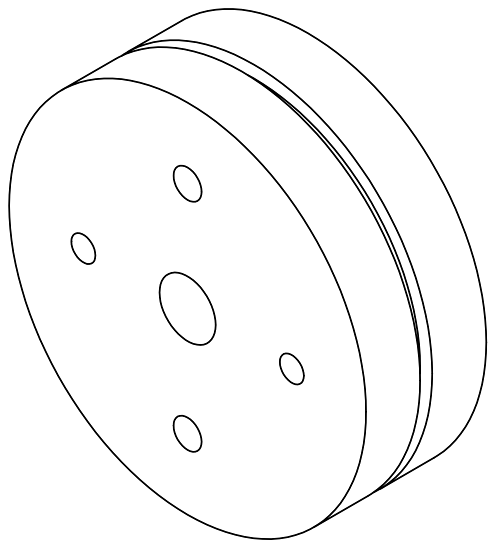
SCALE:1.25:1 WEIGHT: SHEET 3 OF 20

Top View

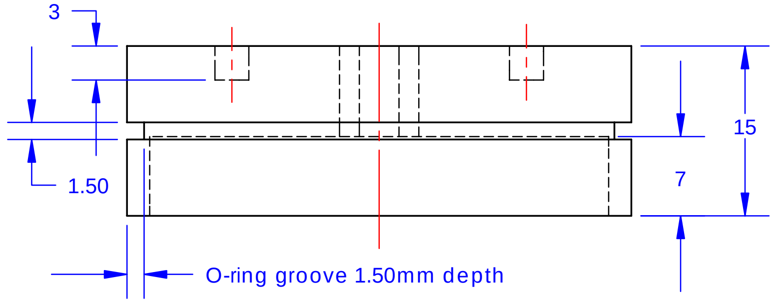


Bottom View





Isometric View



Front View

Chamfer of 0.5mm, 45deg at all edges

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DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

MATERIAL
Aluminum 2018

FINISH
--

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	09-09-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

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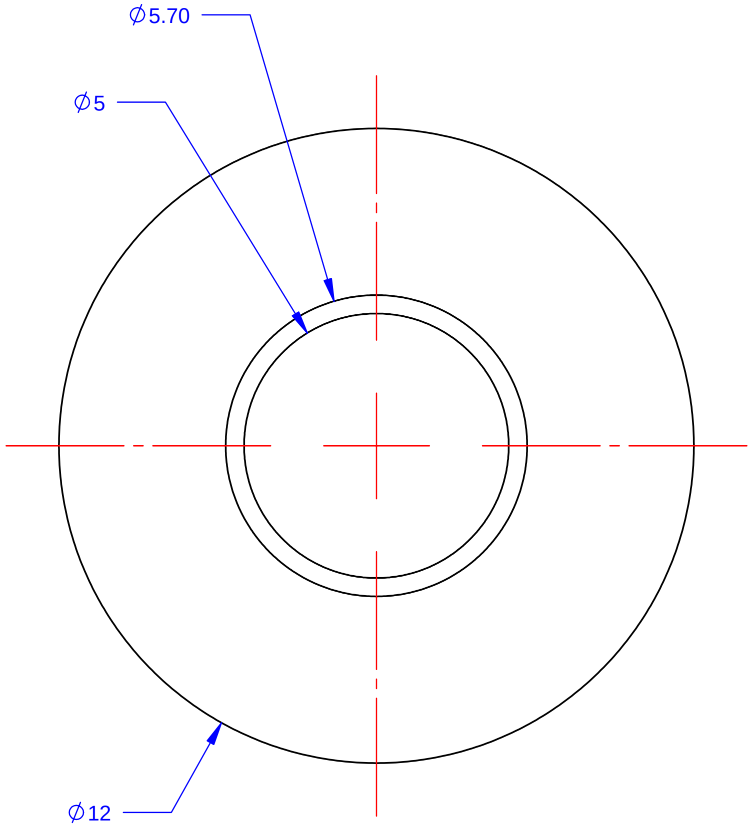
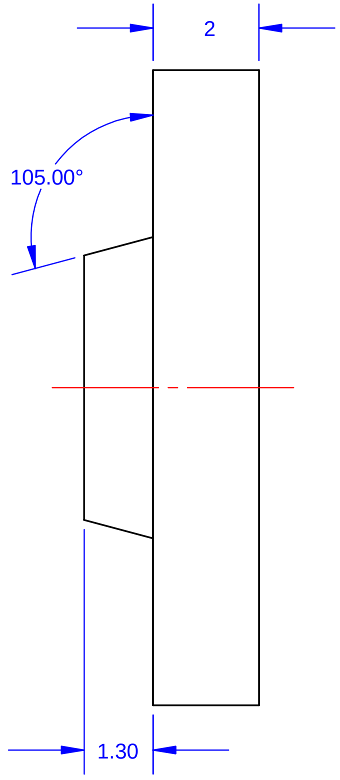
Telephone: +607-5566159

SIZE	DWG. NO.	REV.
A	**10P1-Valve 5**	2

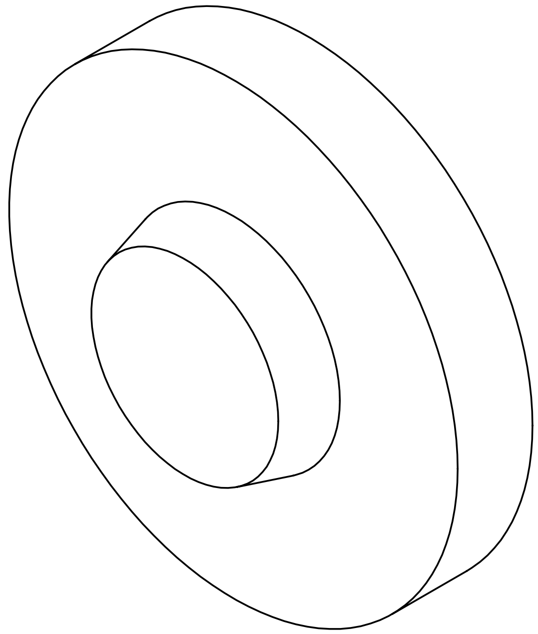
SCALE:1.5:1 | WEIGHT: | SHEET 5 OF 20



Front View



Side View



Isometric View

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DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

MATERIAL
Aluminum 2018

FINISH
--

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	10-09-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

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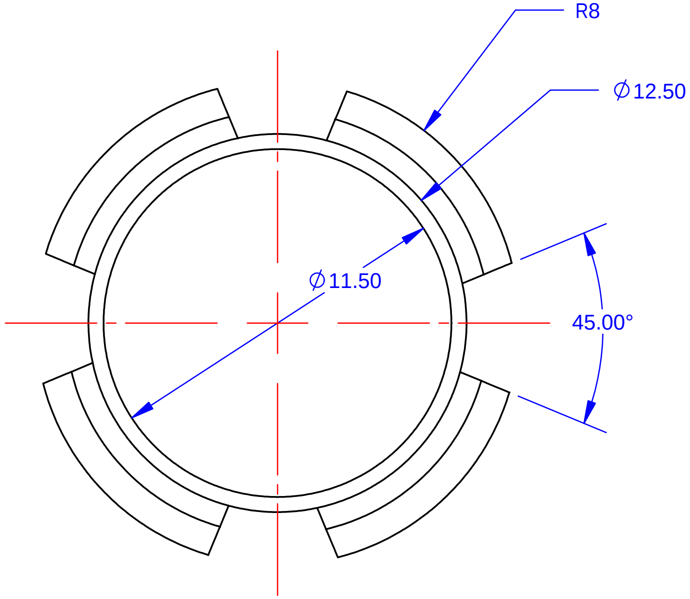
Telephone: +607-5534626
Telephone: +607-5566159

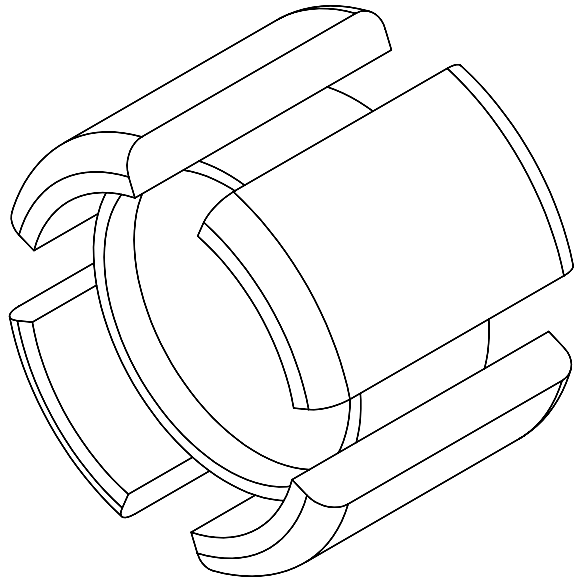
SIZE A | DWG. NO. 10P1-Valve Plate 1 | REV. 1

SCALE:7:1 | WEIGHT: | SHEET 6 OF 20

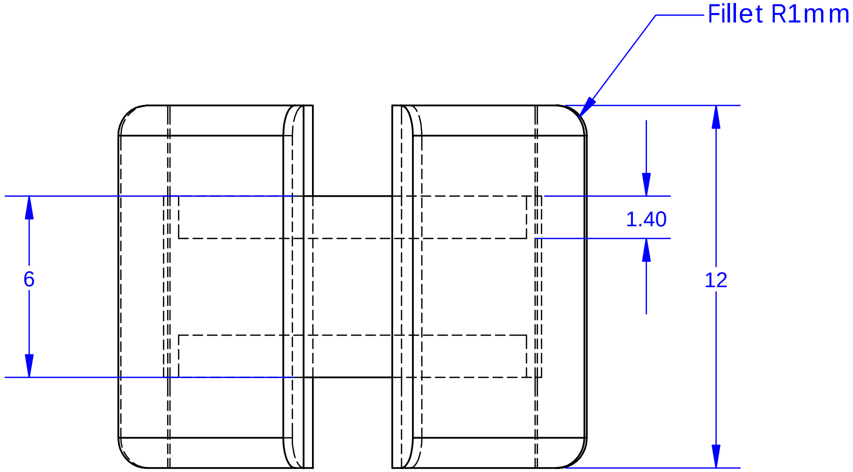
222



Top View



Isometric View



Front View

PROPRIETARY AND CONFIDENTIAL

DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

MATERIAL
Aluminum 2018

FINISH
--

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	10-09-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

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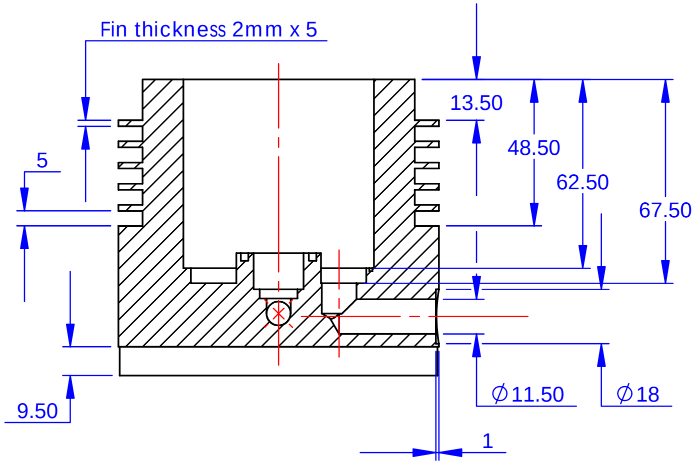
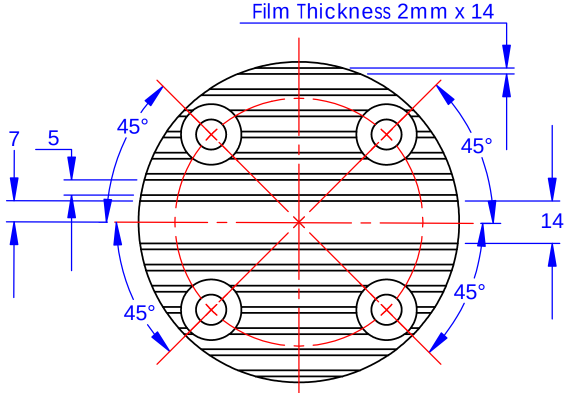
Telephone: +607-5566159

SIZE	DWG. NO.	REV.
A	10P1-Valve Seat 1	1

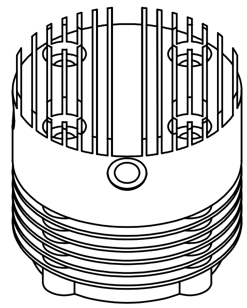
SCALE:4:1 | WEIGHT: | SHEET 16 OF 20



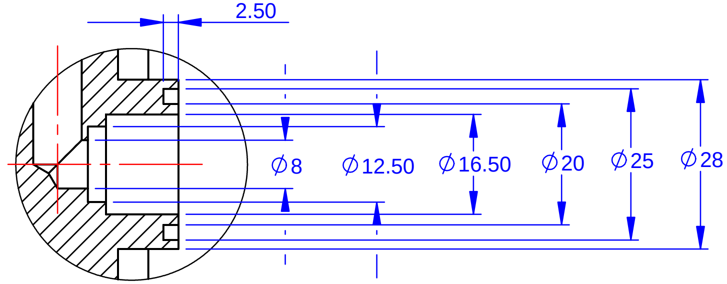
SECTION A-A



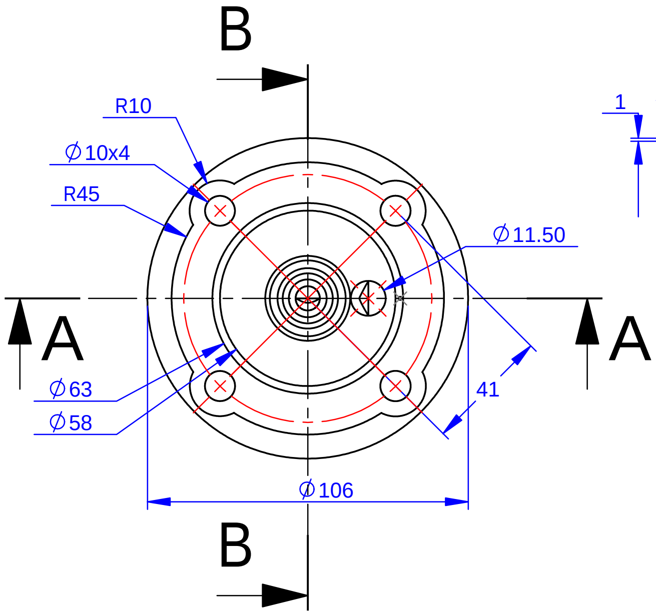
Top View



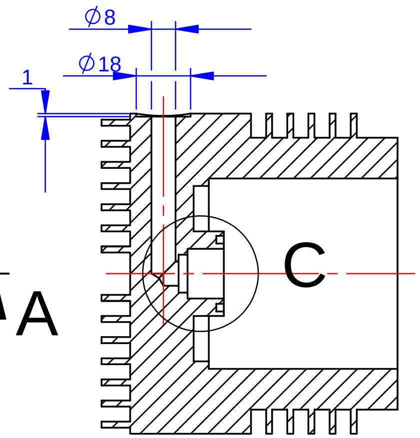
Isometric View



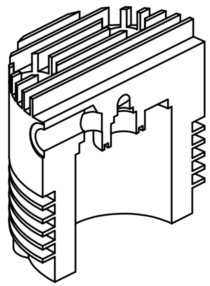
DETAIL C



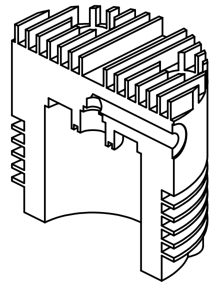
Bottom View



SECTION B-B



3D SECTION A-A



3D SECTION B-B

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TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

MATERIAL
Aluminum LM 25

FINISH
--

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ong KL	09-09-04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

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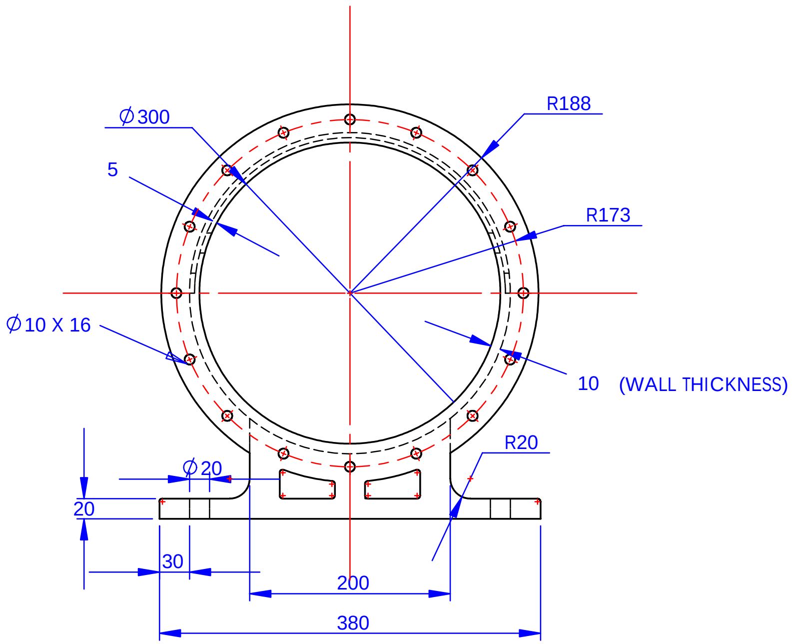
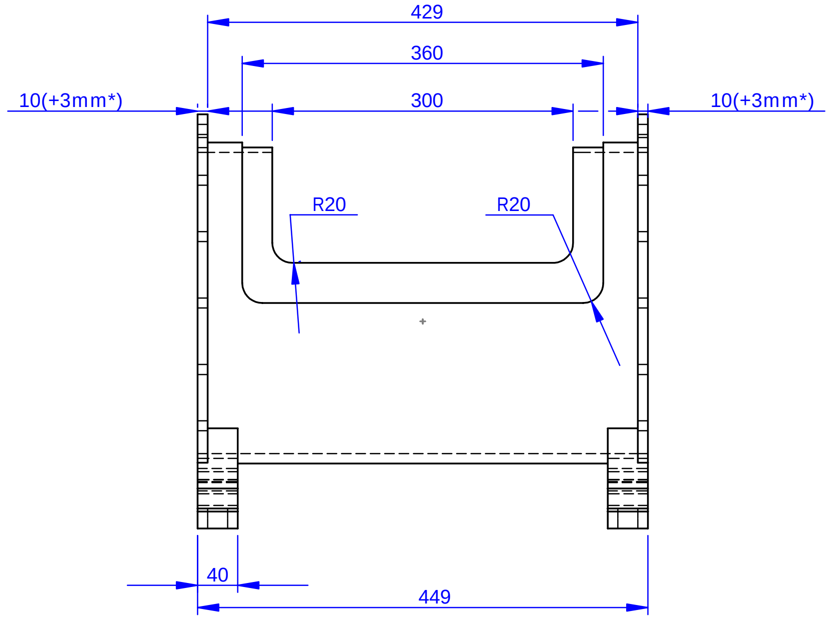
Telephone: +607-5566159

SIZE	DWG. NO.	REV.
A	10P1-Cylinder Block 1-left	6

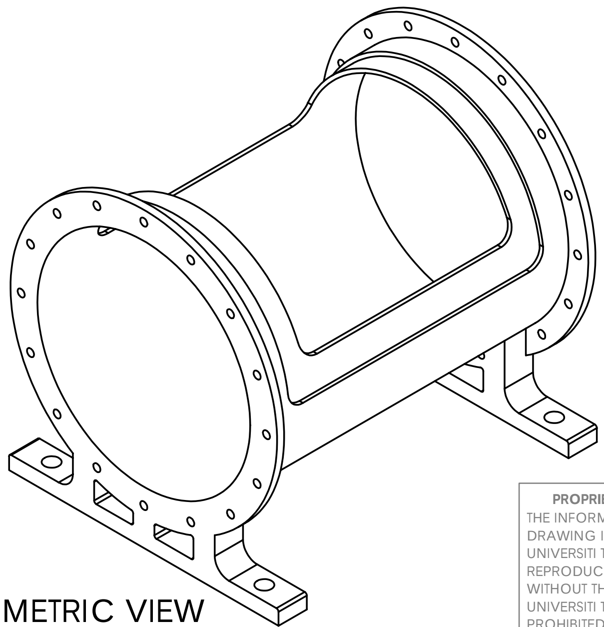
SCALE:1:2.5 WEIGHT: SHEET 1 OF 5



FRONT VIEW



SIDE VIEW



ISOMETRIC VIEW

*Machining stock is for casting process only

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		NAME	DATE
DIMENSIONS ARE IN MILIMETRES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ±0.25 THREE PLACE DECIMAL ±	DRAWN	Hasnol	8/9/04
	CHECKED		
	ENG APPR.		
	MFG APPR.		
MATERIAL Aluminium LM 25	Q.A.		
FINISH --	COMMENTS:		
DO NOT SCALE DRAWING			

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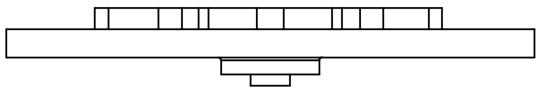
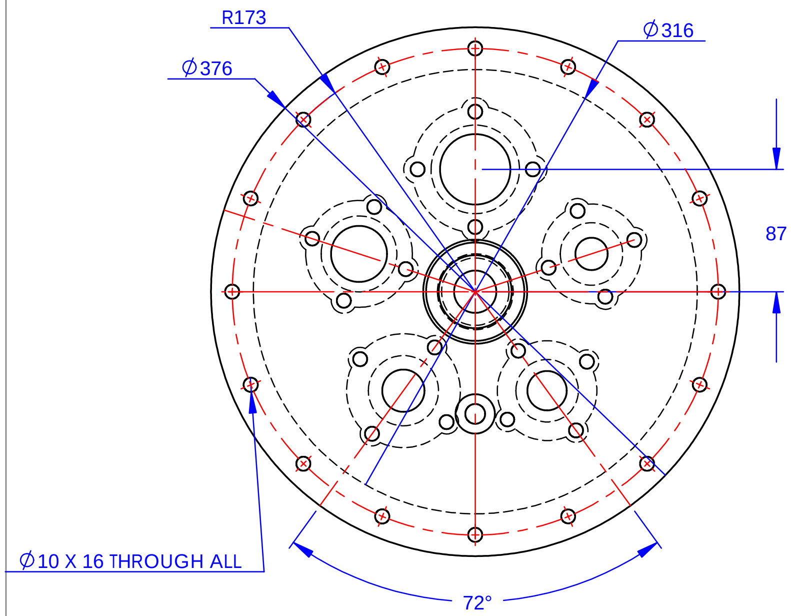
Faculty of Mechanical Engineering
81310 UTM Skudai, Johor, Malaysia.
Telephone: +607-5534626
Telephone: +607-5566159

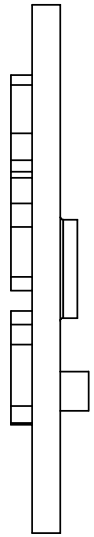
SIZE	DWG. NO.	REV.
A	Housing	1
SCALE: 1:7	WEIGHT:	SHEET 1 OF 1



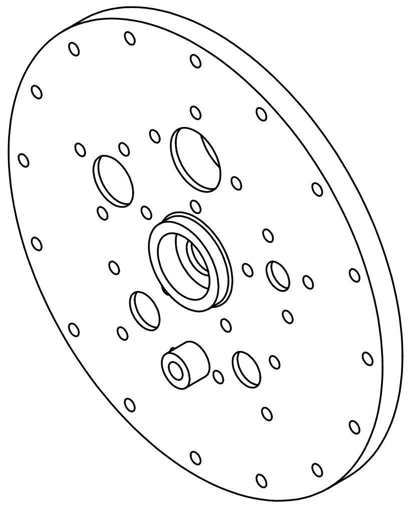
TOP VIEW



FRONT VIEW



SIDE VIEW



ISOMETRIC VIEW

PROPRIETARY AND CONFIDENTIAL

DIMENSIONS ARE IN MILIMETRES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±0.25
THREE PLACE DECIMAL ±

MATERIAL
Aluminium LM 25

FINISH
--

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Hasnol	8/9/04
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

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Facilities & Equipment
Faculty of Mechanical Engineering
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Telephone: +607-5534626
Telephone: +607-5566159

SIZE	DWG. NO.	REV.
A	Left End Plate - 1	1

SCALE: 1:5 WEIGHT: SHEET 1 OF 3

APPENDIX H

Patent Filing for New Multistage Symmetrical Wobble Plate Compressor

bustaman

Our Ref : 050556 MBA
Your Ref :
Date : 22 November 2005

The Registrar of Patents
(Intellectual Property Corporation of Malaysia)
Level 32, Menara Dayabumi
Jalan Sultan Hishamuddin
50623 Kuala Lumpur

BY HAND

Dear Sirs

NEW PATENT APPLICATION IN MALAYSIA
"WOBBLE PLATE COMPRESSOR"
IN THE NAME OF UNIVERSITI PUTRA MALAYSIA

I act for Universiti Putra Malaysia, (a University established under the laws of Malaysia) of 43400, UPM Serdang, Selangor, Malaysia, the applicant for the above-mentioned matter.

I enclose the following to effect the abovementioned application:

1. Patent Form No. 1 (2 copies)
2. Patent Form No. 17 (1 copy)
3. Statement Justifying the Applicant's Right to the Patent
4. Index cards (2 copies)
5. Specification (2 copies)
6. Filing fee of RM 200.00 (6 claims)

Kindly acknowledge receipt.

Yours faithfully
BUSTAMAN

MOHD BUSTAMAN HJ ABDULLAH
Encl.
AMJ/NAP/Rg

ACKNOWLEDGE RECEIPT

DITERIMA
23 NOV 2005

III. INVENTOR

Applicant is the inventor: Yes ☐ No ☑

If the applicant is not the inventor:

Name of inventor (1)	Md. Nor Bin Musa (a citizen of Malaysia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia
Name of inventor (2)	Ainullotfi Bin Abdul Latif (a citizen of Malaysia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia
Name of inventor (3)	Amran Bin Ayob (a citizen of Malaysia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia
Name of inventor (4)	Mohd Nasir Bin Tamin (a citizen of Malaysia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia
Name of inventor (5)	Hamdi Bin Mohd Nor (a citizen of Malaysia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia
Name of inventor (6)	Zair Azrar Bin Ahmad (a citizen of Malaysia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia
Name of inventor (7)	Nor Ilham Bin Mohd Ainon (a citizen of Malaysia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia

Name of inventor (8)	Ardiyansyah Syahrom (a citizen of Indonesia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia
Name of inventor (9)	Andril Arafat Suhasril (a citizen of Indonesia)
Address of inventor:	Faculty of Mechanical Engineering Universiti Teknologi Malaysia 81310 UTM Skudai, Johor Malaysia

A statement justifying the applicant's right to the patent accompanies this Form :
Yes ☑ No ☐

Additional information (if any)

IV. AGENT OR REPRESENTATIVE

Applicant has appointed a patent agent in accompanying Form No. 17: Yes ☑ No ☐

Agent's Registration No.: PA 92-0035

Applicants have appointed : Mohd Bustaman Hj Abdullah
to be their common representative

V. DIVISIONAL APPLICATION

This application is a divisional application: ☐

The benefit of the filing date ☐ priority date ☐

of the initial application is claimed in as much as the subject-matter of the present application is contained in the initial application identified below:

Initial Application No. : ....

Date of filing of initial application : ....

WOBBLE PLATE COMPRESSOR

The present invention relates to wobble plate compressor design. More particularly, the present invention relates to an improved wobble plate type compressor or swash plate compressor having a symmetrical and multistage configuration for use in a gas compression system.

BACKGROUND TO THE INVENTION

It is well known that gas compression systems are required to increase gas pressure. Gas pressure needs to be increased for gas transmission purposes and for storage purposes. Gas can only be distributed when pressure difference exist. For gas storage, gas pressure needs to be increased to reduce the amount of volume required to store the gas. High pressure requirement cannot be achieved using a single stage compression, thus it is necessary to provide for a multistage type compression. Reciprocating piston compressor is a natural choice for high pressure and small to medium flow rate requirement. Piston compressor has many variances according to the piston arrangement and chosen driver mechanism. Crankshaft drive compressor can be found with inline, V-shape, L-shape, vertical, horizontal and radial piston arrangement. Coaxial piston arrangement can also be achieved using swash plate and wobble plate mechanism.

Wobble plate compressor has long been used in the automotive air conditioning system with single stage compression. Example of fixed capacity wobble plate compressor is disclosed in U.S. Patent No. 4,784,045 while variable capacity wobble plate compressor is disclosed in U.S. Patent No. 4,428,718. Variable capacity wobble plate compressor has the ability to change its capacity by changing the piston stroke through varying the wobble plate tilting angle. The ratio between discharge and crankcase pressure is used to control wobble plate tilting angle. Connecting rod is used to connect wobble plate with piston with ball joint interface at both ends. Some of the inventions as in U.S. Patent No.5,079,996 omitted ball joint connection at piston side due to small piston depth or small wobble plate tilting angle as in U.S. Patent No. 4,138,203. Wobble plates slide on rotor either by using roller bearing and thrust bearing, thrust bearing and spherical bearing or roller bearing only as disclosed in U.S. Patent Nos. 4,867,649, 4,869,651 and 4,138,203 respectively. Wobble plate is prevented from rotating with rotor using anti rotation mechanism which is either thrust rider and slider plate (U.S. Patent No. 3,552,886), ball and slider plate (U.S. Patent No. 4,105,370), ball and guide rod mechanism (U.S. Patent No. 5,094,590), Rzeppa mechanism (U.S. Patent No. 5,079,996) and bevel gear (U.S. Patent No. 4,869,651). Slanted or fully supported drive shaft at both ends has been used in all previous inventions. Rotor shapes for fixed capacity wobble plate compressor tend to be simpler as disclosed in U.S. Patent No. 4,869,651, whereby for variable capacity wobble plate compressor, some arrangement is needed to change wobble plate tilting angle with the typical design as disclosed in U.S. Patent No. 4,428,718. Many improvements or design variations have been made on this mechanism alone. Housing design is normally split into two and three piece parts with cylinder block imbedded into the housing. End plate is used to house valve plate and lubrication pump.

SUMMARY OF THE INVENTION

Accordingly, it is an object of the current invention to provide a wobble plate type compressor aligned in symmetrical configuration, which results in significant reduction of vibration of the compressor. The pistons, cylinder block and wobble plate which mirror each other at the centre of drive shaft, reduces if not eliminates the horizontal force caused by gas reaction forces acting on the pistons.

Another object of the current invention is the introduction of multistage configuration that will allow higher gas compression than normally attained in single stage wobble plate compressors used in automotive refrigeration.

A further object of the current invention is to provide a wobble plate of the character described which is particularly designed to compress gas without oil contamination. This feature eliminates the inevitable blow-by of oil vapor passing into the gas being compressed, as the present feature is free from lubricating requirements on the part of the operator between periodic maintenance.

Yet another object of the current invention is to provide a compressor of the characters above which will involve a fewer number of parts with reduced machining requirements, and which may be easily and rapidly assembled to provide a unit at minimum cost.

Yet another object of the present invention is to provide a wobble plate compressor of the characters described which is composed of durable parts affording easy disassembly when required for maintenance and affording long, useful life.

Still another object of the present invention is to provide a structure of the character described which may be scaled up or down to readily provide units of different sizes and capacities and also be adopted to swash plate type compressors.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will now be described in greater detail, by way of an example, with reference to the accompanying drawings, in which:

Figure 1 is the isometric view of the compressor illustrating the housing, left and right end plate and acrylic cover;

Figure 2 is a see through drawing illustrating the assemblies of the internal component in the compressor;

Figure 3 is a section view showing parts of wobble plate and anti-rotation mechanism;

Figure 4 is the isometric view of piston assemblies illustrating different size and shapes of pistons and couplers;

Figure 5 is a cross section view of wobble plate assemblies illustrating the components involved;

Figure 6 illustrates the arrangement of cylinder block at the end plate, shown here with possible fittings arrangement at the suction and discharge port at each cylinder block;

Figure 7 is a cross section view of piston assemblies illustrating the components involved;

Figure 8 is a cross section view of cylinder block illustrating arrangement of liner, valve cover, suction valve plate and discharge valve plate; and

Figure 9 is cross section view of cylinder block illustrating the suction port and discharge port.

DETAILED DESCRIPTION OF THE DRAWINGS

Referring to Figure 1 through Figure 9, wherein like numerals indicate like corresponding parts throughout the nine views, a symmetrical multistage wobble plate compressor is generally shown.

The compressor has a housing, which includes crank case **1**, left end plate **2**, right end plate **3** and acrylic cover **4**. Left end plate **2** and right end plate **3** are clamped to crank case **1** using bolts.

The compressor of the present invention comprises of two sets of pistons in cylinder blocks **5**, wobble plate **6**, rotor **7** that mirror each other as shown in Figure 2. Drive shaft **8** is stepped at both ends to locate and fix the bearing at the end plate **2, 3** at both ends. Drive shaft **8** and rotor **7** are fixed together using pin. Rotor **7** and wobble plate **6** is connected together through deep groove bearing **9**. Wobble plate **6** have slots for wobble plate pins **10**, slot for anti-rotation ball **11** and flange at the front face around the periphery of bearing slot at its centre. Rotor **7** is provided with a slot for pin and flange at the back face.

Bearing **9** is tight fitted to both rotor **7** and wobble plate **6**. External c-clip is used to secure rotor **7** with bearing **9** while internal c-clip is used to secure wobble plate **6** with bearing **9**. This will prevent wobble plate **6** or bearing **9** from sliding to the front in case of tight fit failure. Flange at the front face of wobble plate **6** will press against the front face of bearing outer race **12** whereas flange at the back of rotor **7** will press against the back face of bearing inner race **13**. This will ensure bearing **9** or wobble plate **6** from sliding to the back in case of tight fit failure.

Rotation of drive shaft **8** with rotor **7** will induce wobbling motion in the wobble plate **6** through the bearing **9** interface. Wobble plate **6** is prevented from rotating with rotor **7** by the anti rotation mechanism which consist of a guide rod **14** and hollow spherical ball **11** that slide horizontally on the guide rod **14** and up and down in the slot for anti-rotation ball at wobble plate **6**. Wobble plate **6** wobbling motion will be transferred into piston **15** reciprocating motion through connecting rod **16**. End joint connection is used as the interface between the connecting rod **16** and wobble plate **6** and between the connecting rod **16** and piston **15**.

Different piston diameter size **15, 17, 18, 19, 20** and its corresponding cylinder block **5, 26, 27, 28, 29** are used for each stage. The largest piston **15** and cylinder block diameter size **5** is for the first stage. Piston and cylinder block diameter size will correspondingly reduce for higher number of stage. Each piston set has different shape of pistons **15, 17, 18, 19, 20** with corresponding number of groove, piston rings/rider rings and coupler **21, 22, 23, 24, 25**. The variations depend on stage pressure involved.

Pistons for the first stage to the third stage **15, 17, 18** are made from aluminum while the fourth and fifth stage **19, 20** is made from hard steel. Liner **39** is made from cast iron. The inner surface of the liner is hard-chromed to obtain mirror surface finishes. Piston ring and rider ring is made from self-lubricated PTFE material. Labyrinth groove is used for the forth stage and fifth stage piston omitting piston rings due to small piston diameter size. Teflon material is used for the liner at the last two stages with the clearance between piston and cylinder block is 5μm.

Coupler **21** is used to connect piston **15** to their respective connecting rod **16** using end-joint **30, 31**. Holes are made at the coupler **21** to ensure mass of each piston **15** with its corresponding coupler **21** is the same for all stages. Bolt **32** is used to fixed pistons **15** with coupler **21**. Couplers **21** are fitted to connecting rod end-joint ball **33** at the piston side using piston pin **34**, which is secured in the coupler **21** using two internal c-clips. End-joint ball centre is located on pin using piston bush **35**.

Connecting rod **16** is composed of female **31** and male end-joint **30** that are screwed into each other. A connecting rod **16** length is determined by length of thread engagement between both end-joint and fixed using nut and thread lock. Connecting rod end-joint ball at the wobble plate **36** is also fitted to wobble plate **6** using wobble plate pin **37**, which is secured in the wobble plate **6** using internal c-clip. End-joint ball centre is located on wobble plate pin **11** using wobble plate bush **38**.

All the bearings and end-joints used are lubricated using grease that needs no maintenance within the periodic maintenance interval. Sealing between cylinder block **5**, liner **39** and valve seat **40** is achieved using o-ring. O-ring is also used under piston ring to press piston ring against liner bore surface.

Each cylinder block **5** has two ports for suction **41** and discharge **42**. Suction valve plate **43** is positioned between liner **39** and valve cover **44** while discharge plate valve **45** is positioned between valve cover **44** and valve seat **40**. Fins on cylinder block **5** are used for cooling purpose.

CLAIMS

1. A wobble plate type compressor (6) for use in gaseous compression systems comprising of a compressor housing having a cylinder block provided with a plurality of cylinders (5, 26, 27, 28, 29) and a crank chamber (46) enclosed within each of said cylinders which is free from contamination, a drive shaft (8) rotatably supported in said housing, a rotor (7) around the drive shaft (8) which is arranged back-to-back and further connected to an inclined wobble plate (6), a coupling member of said wobble plate (6) with each having a plurality of pistons (15, 17, 18, 19, 20) arranged symmetrically, wherein said coupling member having one end which is coupled with said wobble plate (6) and another end which is coupled with each of said symmetrical pistons (15, 17, 18, 19, 20), and a rotation preventing means for preventing rotation of said wobble plate (6) with the rotor (7).

2. The wobble plate compressor (6) as claimed in claim 1, wherein said compressor (6) is multistage in design with symmetrical back-to-back arrangement that reduces vibration and sound.

3. The wobble plate compressor (6) as claimed in claim 2, wherein said multistage design provides for different pistons (15, 17, 18, 19, 20) and cylinder block sizes at different stages, thus allowing for higher gas pressure compression.

4. The wobble plate compressor (6) as claimed in claim 1, wherein said wobble plate coupling member comprises of a connecting rod (16) with end-joint connection at both ends to connect between pistons (15, 17, 18, 19, 20) and wobble plate (6).

5. The wobble plate compressor (6) as claimed in claim 1, wherein said piston rings and rider rings are self lubricated, while bearings (9) and end joints (30, 31) are equipped with grease for lubrication, thus providing a contamination free wobble plate (6).

6. The wobble plate compressor (6) as claimed in claim 1, wherein said rotation preventing means prevents rotation of wobble plate (6) with rotor (7) by the anti-rotation mechanism which consists of a guide rod (14) and hollow spherical ball (11) that slide horizontally on the guide rod (14) and up and down in the slot within the wobble plate (6).

ABSTRACT

WOBBLE PLATE COMPRESSOR

A wobble plate type compressor (6) for use in gaseous compression systems having a symmetrical and multistage configuration is disclosed, which includes two sets of different pistons (15, 17, 18, 19, 20) sizes being reciprocated within respective cylinders by two wobble plate members that mirror each other. Multistage configuration will have multiple piston (15, 17, 18, 19, 20) diameter sizes that allow for higher gas pressure compression.

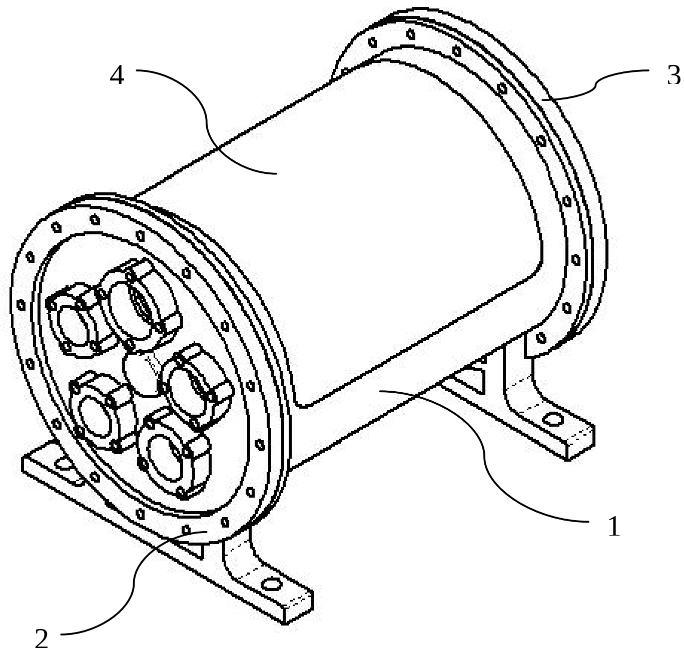

Figure 1

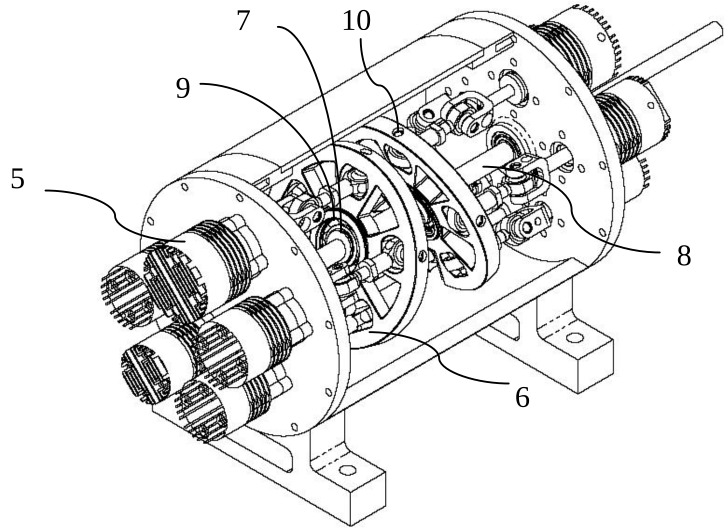

Figure 2

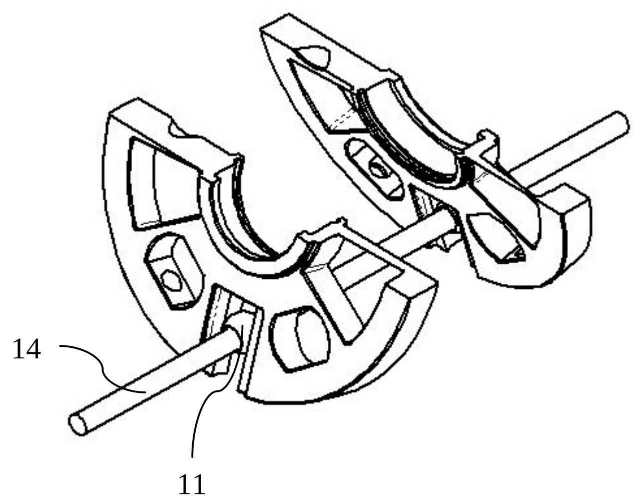

Figure 3

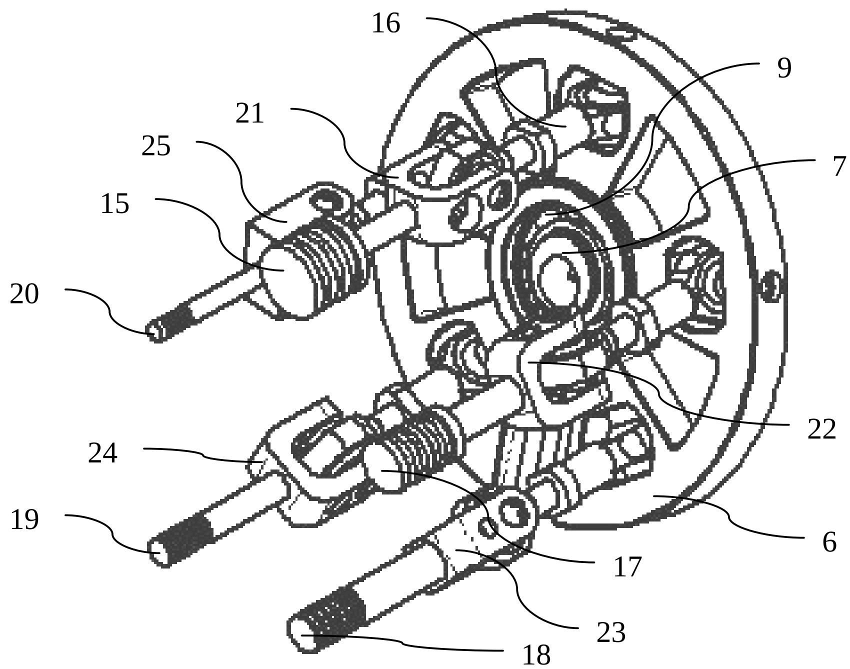

Figure 4

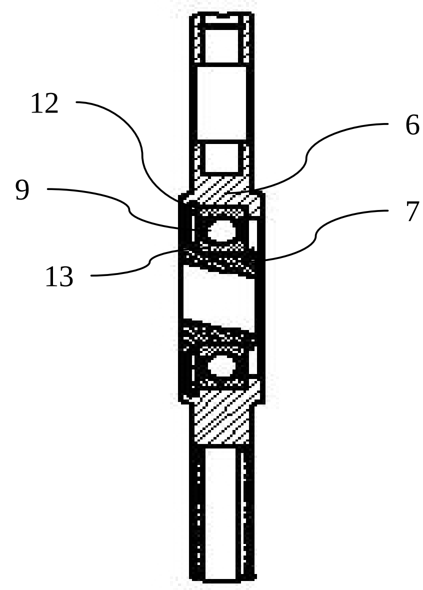

Figure 5

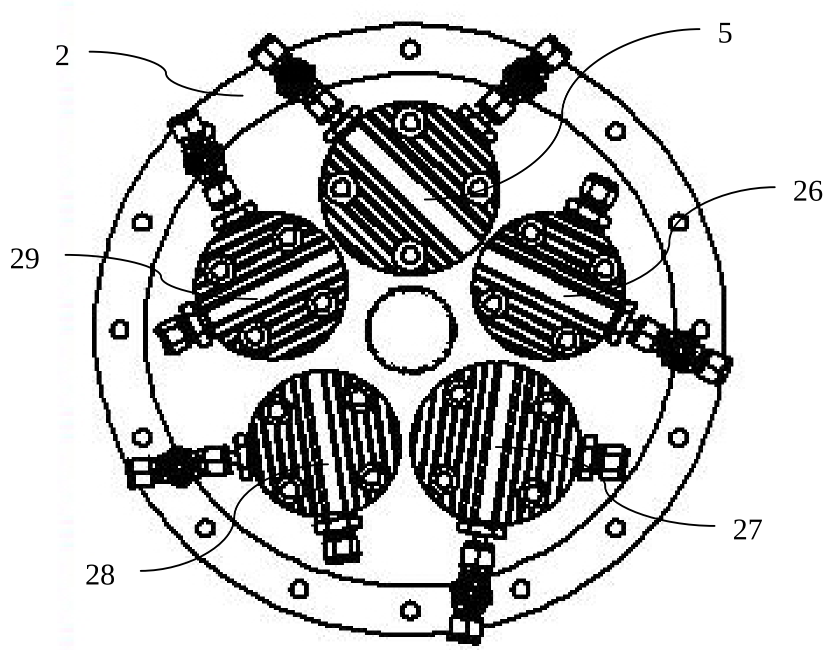

Figure 6

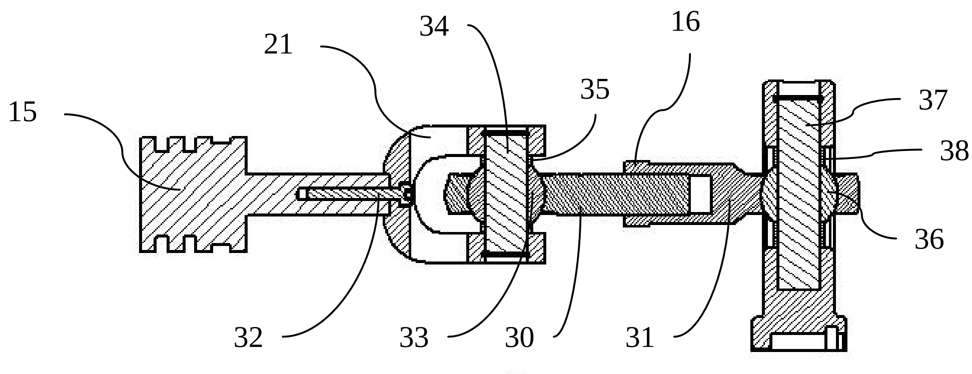

Figure 7

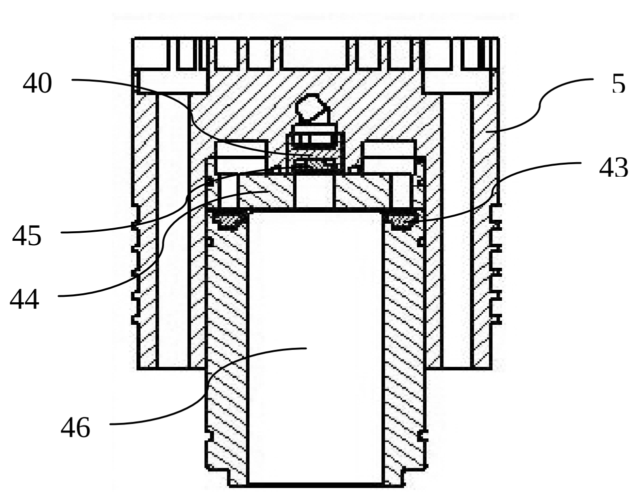

Figure 8

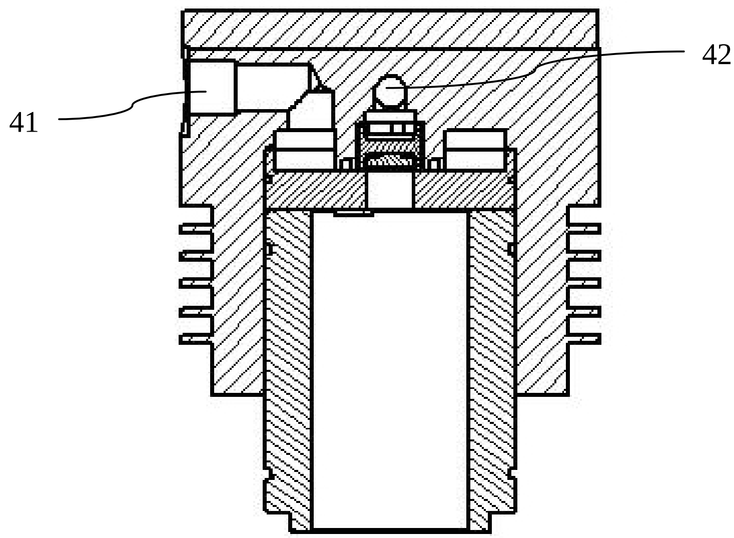

Figure 9

APPENDIX I

List of Patent Review

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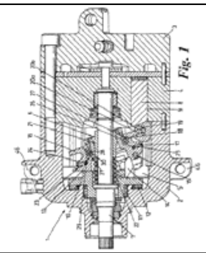
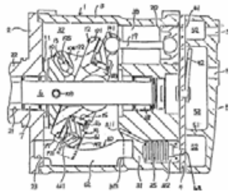
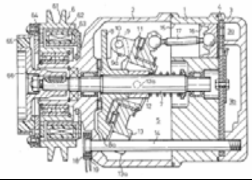
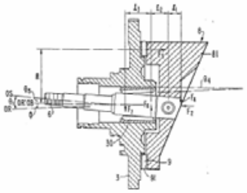
List of Patent Review

No	Title	Publication Number	Date	Inventor	Applicant	Drawing
1	**2**	**3**	**4**	**5**	**6**	**7**
1	A wobble plate arrangement for a compressor	EP1363022	2003	Schwarzkopf Otfried	ZEXEL VALEO COMPRESSOR EUROP G (DE)	
2	Wobble plate compressor	EP0280479	1998	Higuchi Teruo., Kikuchi Sei., Takai Kazuhiko., Kobayashi Hideto., Terauchi Kiyoshi	SANDEN CORP (JP)	
3	Compressor with rotation detecting mechanism	US5540560	1996	Kimura Kazuya., Takenaka Kenji., Fujisawa Yoshihiro., Kayukawa Hiroaki	TOYODA AUTOMATIC LOOM WORKS (JP)	
4	Wobble plate type compressor with a drive shaft attached to a cam rotor at an inclination angle	US4870894	1989	Toyoda Hiroshi., Shimizu Shigemi., Hatakeyama Hideharu., Kumagai Shuzo., Takahashi Hareo	SANDEN CORP (JP)	

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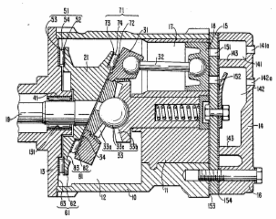
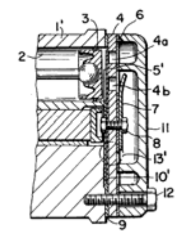
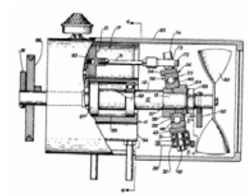
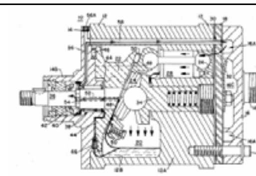
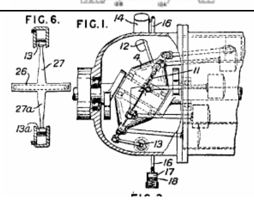
5	Wobble plate type compressor	US4869651	1989	Shimizu Shigemi., Shimizu Hidehiko., Terauchi Kiyoshi	SANDEN CORP (JP)	
6	Fluid suction and discharge apparatus	US4283166	1981	Masaharu Hiraga	SANKYO ELECTRIC CO	
7	Swash plate compressor	US4138203	1979	Slack Don S	SLACK DON S	
8	Refrigeration compressor	US3838942	1974	POKORNY F	MITCHELL J CO	
9	Improvements relating to reciprocating engines, pumps or compressors of the swash- or wobble-plate type	GB458360	1936	TURNER, K. K	KENNETH KESTELL TURNER	

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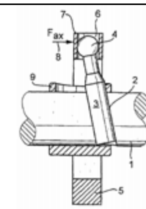
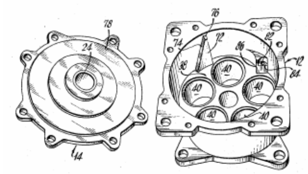
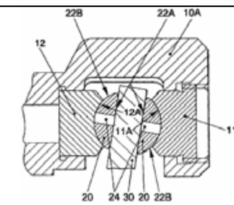
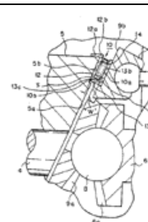
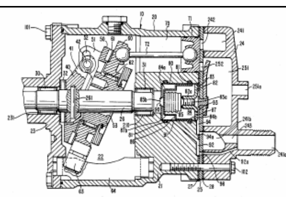
10	Wobble Plate Piston Mechanism	US2004007126	2004	Parsch willi	LUK FAHRZEUG HYDRAULIK (DE)	
11	Lubrication system for compressor unit	US4005948	1977	Hiraga Masaharu., Shimizu Shigemi	SANKYO ELECTRIC CO	
12	Plunger used in a wobble plate compressor in an air conditioner comprises jaws for receiving a sliding block	DE10231212	2003	LOY CHRISTOPH [DE]; DROESE HEIKO [DE]; GEBAUER KLAUS [DE]; RESKE THOMAS [DE]; NISSEN HARRY [DE]	VOLKSWAGENWERK AG [DE]	
13	Wobble plate type refrigerant compressor having a thrust bearing assembly for a wobble plate support	US4981419	1991	Kayukawa Hiroaki., Takenaka Kenji., Okamoto Takashi., Hyodo Akihiko	TOYODA AUTOMATIC LOOM WORKS (JP)	
14	Wobble plate type compressor with variable displacement	US4913626	1990	KiyoshiTerauchi	SANDEN CORP (JP)	

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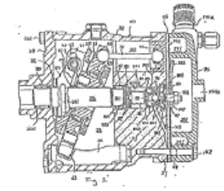
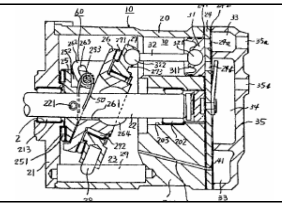
15	Wobble Plate Compressor with Suction-Discharge Differential Pressure Control of Displacement	US4913627	1990	KiyoshiTerauchi	SANDEN CORP (JP)	
16	Compressor with variable displacement mechanism	US4850811	1989	Takai Kazuhiko	SANDEN CORP (JP)	

APPENDIX J

List of Publications

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List of Publication

Andril Arafat, Zair Asrar Ahmad, Ardiyansyah Syahrom, Nor Ilham Mohd Ainon, Md. Nor Musa, Ainullotfi Abdul-Latif, and Wan Ali Wan Mat. Piston Ring Assembly for a New Natural Gas Vehicle Symmetrical Multistage Wobble-Plate Compressor *.1st Regional conference on vehicle engineering and technology 2006.* July 3-5, 2006. Kuala Lumpur: RIVET. 2006.

Ardiyansyah Syahrom., Md. Nor Musa., Wan Ali Wan Mat and Ainullotfi Adb Latif. Optimum Number of Stages of the New Multi-Stage Symmetrical Wobble Plate Compressor. *1st Regional Postgraduate Conference on Engineering and Science.* July 26-27, 2006. Johor Bahru: RPCES. 2006.

Ardiyansyah, Md Nor Musa, Ainullotfi Abdul-Latif, and Mohd Adlan Abdullah, "Development and Testing of a Compressor for Natural Gas Vehicle Refuelling", 2006th Purdue International Conference on Compressors, June 2006, Lafayette, USA.

Ardiyansyah, Md Nor Musa, Ainullotfi Abdul-Latif and Mohd Adlan Abdullah, "Discharge Flow analysis for new symmetrical wobble plate compressor", Regional Conference on Computational Mechanics & Numerical Analysis (CMNA), Syiah University, Aceh, June 2006, .

Ardiyansyah Syahrom., Md. Nor Musa., and Ainullotfi Adb Latif. Wobble *Design and Development of a Compressor for Natural Gas Vehicle Refuelling. International Conference Engineering Design 2005.* August 15-18, 2005. Melbourne: ICED. 2005.

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Ardiyansyah, Md Nor Musa, Ainullotfi Abdul-Latif, & Mohd Adlan Abdullah, “New Symmetrical Wobble-plate Compressor for Natural Gas Vehicle Refuelling”, 1st Australasian Natural Gas Vehicles Association Conference and Exhibition (ANGVA 2005), July 2005, Kuala Lumpur.

Ardiyansyah Syahrom., Md. Nor Musa., and Wan Ali Wan Mat. Pembangunan Pemampat Simetri Berperingkat Jenis Plat Wobal Untuk Gas Asli. *Malaysian Science and Technology Congress.* September 23-25, 2003. Kuala Lumpur: COSTAM. 2003. 237.

Zair Asrar Ahmad, Ardiyansyah Syahrom, Ainullotfi Abdul Latif and Md Nor Musa, “Study on the Stressing of a New Symmetrical Wobble Plate Compressor : Kinematic and Forces”, Malaysia S & T Congress, Kuala Lumpur, September 2003

Zair Asrar Ahmad, Ardiyansyah Syahrom, Ainullotfi Abdul-Latif and Md Nor Musa, “Analysis of anti-rotating mechanism in new symmetrical wobble plate compressor for NGV refulling appliance”, 2006th Purdue International Conference on Compressors, June 2006, Lafayette, USA

Zair Asrar Ahmad, Ardiyansyah Syahrom, Ainullotfi Abdul-Latif and Md Nor Musa, “Motion analysis for symmetrical wobble plate compressor”, Regional Conference on Computational Mechanics & Numerical Analysis (CMNA), Syiah University, Aceh, June 2006, .