國立交通大學

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碩士論文

往復式壓縮機之最佳化設計
Design Optimization of Reciprocating Compressor

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摘要

本研究的目的為撰寫一最佳化模組於往復式壓縮機效能模擬軟體。此模擬軟體整合熱流、機構與整體系統動態並以數值方法進行分析與模擬。以此軟體為基礎，本研究發展一最佳化模組，主要包含兩個部份：使用者介面與最佳化解題工具。使用者介面能幫助快速且方便的規劃與設定不同的最佳化問題，讓使用者或工程師簡單且容易的選擇設計變數、限制條件與目標函數，免除連結分析程式和最佳化程式的困難，並更輕易地處理大量設計變數的最佳化問題。最佳化解題工具則提供方法來解決不同由使用者介面所定義的不同類型的最佳化問題。閥片最佳化設計將是本文中主要探討的議題。此最佳化模組可幫助進行電腦輔助往復式壓縮機的設計，並減少繁瑣且耗時的實驗測試次數，而閥片最佳化的結果將可做為實體壓縮機模型變更設計以改善效能的參考與依據。

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ABSTRACT

The goal of this study is to add an optimization module into one simulation software used in predicting the performance of the reciprocating compressor. This simulation software integrates the analysis of thermal and mechanism modules and simulates by using the computer numerical method. Based on the software program, the study incorporates the optimization module, including user interface and optimization solvers. The user interface can help defining, formulating and setting the optimization problem quickly. With this interface, the user can select design variables, cost function, and constraints freely and conveniently to reduce difficulties in linking between analysis and optimization software. The analysis module in the study is the comprehensive simulation software of the reciprocating compressor. The optimization solvers can contain certain ways providing the user to select. The valve optimization is a main theme for discussion in the study. The optimization module constructed in the simulation software as a useful tool help to aid optimum design of the reciprocating compressor.

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Notation List

Symbols of the dynamic analysis of reciprocating compressor:

c.g.	Center of gravity
d_s	Distance of the valve passage
$\bar{d}$	Crankshaft rotation ditection vector, clockwise = -1
D_{CB}	Distance between the crankshaft and the main bearing
D_{BB}	Distance between the secondary bearing and the main bearing
D_{u1}	Distance between the center of mass of the first counterweight and the crankshaft measured parallel to the centerline of the crankshaft
D_{u2}	Distance between the center of mass of the second counterweight and the crankshaft measured parallel to the centerline of the crankshaft
e_U	Eccentricity of the counterweight with respect to the centerline of the crankshaft
f	Coefficient of friction
F_{23X}	Force exerted on the connecting rod in the x-direction by the crankshaft
F_{23Y}	Force exerted on the connecting rod in the y-direction by the crankshaft
F_{3PX}	Force exerted on the piston in the x-direction by the crankshaft
F_{3PY}	Force exerted on the piston in the y-direction by the crankshaft
$\vec{F}_C$	Resultant force exerted on the equivalent mass, M_C
$\vec{F}_{b1}$	Resultant force exerted by the counterweight, M_{u1}
$\vec{F}_{b2}$	Resultant force exerted by the counterweight, M_{u2}
F_{MUR}	Radial component of certain counterweight force
F_{MUT}	Tangential component of certain counterweight force
$\vec{F}_P$	Resultant force exerted on the equivalent mass, M_P

F_{PRESS} Pressure force exerted on the piston

$F_{X\_Bearing1}$ X-direction component of the force exerted on the crankshaft by the main bearing

$F_{Y\_Bearing1}$ Y-direction component of the force exerted on the crankshaft by the main bearing

$F_{X\_Bearing2}$ X-direction component of the force exerted on the crankshaft by the secondary bearing

$F_{Y\_Bearing2}$ X-direction component of the force exerted on the crankshaft by the secondary bearing

F_{XU} X-direction component of certain counterweight force

F_{YU} Y-direction component of certain counterweight force

F_{WP} Force exerted on the piston normal to the wall

F_{WF} Friction force exerted on the cylinder wall

H Eccentricity of the crankshaft

M Distance from crankshaft bearing center of gravity

M_{CR} Mass of the connecting rod

M_C Equivalent mass on the point, c, at the join between crankshaft and connecting rod

M_{CS} Mass of the crankshaft

M_P Equivalent mass on the piston

M_{Piston} Mass on the piston

M_{u1} Mass of the first counterweight

M_{u2} Mass of the secondary counterweight

M_u Mass of the certain counterweight

N	Revolutions per second of crankshaft rotating
I_{CSR}	Moment of inertia of the crankshaft and rotor about their axis of rotation
I_{CR}	Moment of inertia of the connecting rod about its center of gravity
I_{CS}	Crankshaft and rotor moment about their axis of rotation
I_{XXCC}	Moment of inertia of crankcase about an axis through its cg parallel to the x-direction
I_{YYCC}	Moment of inertia of crankcase about an axis through its cg parallel to the y-direction
I_{ZZCC}	Moment of inertia of crankcase about an axis through its cg parallel to the z-direction
L	Length between the connecting rod bearing centers
$L_{Journal}$	Journal length
l_1	Distance of the applied force related to the coordinate
P	Force exerted on the piston due to gas pressure
$P_{Bearing}$	The pressure exerted on the projected area of the bearing
R	Crankshaft throw
r	Journal radius
t_{valve}	Thickness of the valve
T	Torque exerted on the rotor by the motor
Tcc	Torque exerted between the connecting rod and the crankshaft
Tcp	Torque exerted between the piston and the connecting rod
V_{CGX}	Velocity of the connecting rod center of gravity in the x-direction
V_{CGY}	Velocity of the connecting rod center of gravity in the y-direction
V_P	Velocity of the piston in the x-direction
W	Force exerted on the bearing
w_2	Crankshaft angular velocity

w_3 Connecting rod angular velocity

X_B X-distance of crankshaft from crankcase center of gravity

X_{CY} Instantaneous X-distance of piston from crankcase center of gravity

X_{CG} Displacement of the connecting rod center related to the original position in the x-direction

X_P Displacement of the piston

Y_{CG} Displacement of the connecting rod center related to the original position in the x-direction

Z_{CB} Z-distance of secondary bearing form crankcase c.g.

Z_{CY} Z-distance of piston form crankcase c.g.

Z_{MB} Z-distance of main bearing form crankcase c.g.

θ Crankshaft rotation angle

Φ Connecting rod rotation angle

μ Absolute viscosity

CHAPTER 1

INTRODUCTION

This study is an extension of simulation software developed to describe the performance inside reciprocating compressor. The Energy and Resources Laboratories (ERL) of the Industrial Technology Research Institute (ITRI) entrust to extend to construct optimization module in the software. By appending the module, aiming at different expected requirements or objectives like variety of efficiencies in all respects of reciprocating compressor, the parameters using in the software can freely select to form the design variables and proceeding optimum methods to meet the predicted objective. This computer aided analysis software make the process developing design optimization of reciprocating compressor more efficient in time and cost.

1.1 Brief Statement of Reciprocating Compressors

The hermetic reciprocating compressors were developed for many years of reliable and they were also the first type of compressor to put in use for industrial air and gas compression and still hold multipurpose until now. Reciprocating compressors are applied from deep vacuum range to 450,000 kPa range, and even higher for the special and research services including mechanical, metallurgy, and chemical industries. They use in low flow rate diaphragm compressor and utility air units to high-volume vacuum pumps, and gas pipeline compressors. Besides, they are used in refrigeration, air conditioners and cooling devices [1].

Reciprocating compressors have simple action principle, its compressing and expanding element is a piston, performing a reciprocating motion in a cylinder. The motion start when the crank locates at the upper dead point, the piston expands increasingly. When the pressure of refrigerant in the cylinder is greater than the pressure in suction chamber, the suction valve opens and imports the refrigerant. After the crank moving to the lower dead point, the capacity in cylinder decreases and makes the pressure rising in cylinder, the suction valve closes. After the pressure in cylinder overcomes the pressure in the discharge chamber, the discharge valve opens to make refrigerant with high pressure export from cylinder. The process described above is called a cycle of compressing refrigerant.

In general, the pressure difference deciding the automatic spring loaded valves open or close. Many conditions for cylinder valves will seriously influence the performance efficiency of the suction and discharge of the refrigerant.

1.2 **Reference Documents**

This study is based on the simulation software of reciprocating compressor developed by ITRI. The software considers some mathematic models related with thermodynamic theory and kinematics and dynamic of the mechanism. Among, Huang [1] studied to stylize the thermodynamic simulation module and the kinematics and dynamic simulation module of mechanism is programmed by Hsiung [3]. Besides, most other references are adopted from the International Compressor Engineering Conferences holding by Purdue University every two years after 1970 and many outstanding papers are collected.

1.3 **Contents of the Study**

The contents of the study are to combine the simulation program of the hermitic reciprocating compressor and the optimization module to calculate and predict various efficiencies for assisting optimum design and reducing development time.

Utilizing the mathematical simulation during designing stage can reduce the times of experiments and just develop the necessary events. If it has the special requirements for the experimental conditions and environments and may cost enormous time and cash, the simulation can help saving those large quantities.

In general consideration, the affecting factors of the reciprocating compressor are too much to complete the optimization by experimentation. The program combines mathematical simulation and optimization module to find various optimum parameters sets for different design requirements and provides the foundation for arising performance of the reciprocating compressor in practice.

The main subjects of the optimization module attaching to the software will study and discuss the following performances of the compressor and the detailed statements of those will describe later.

1. Volumetric efficiency
2. Compressor efficiency
3. Mechanical efficiency
4. Motor efficiency
5. Flow rate of refrigerant
6. Capacity of refrigeration
7. Energy efficiency ratio(EER)

1.4 **Outlines**

This study continues to develop the software for the performance simulation of hermetic reciprocating compressors above stated. There are some objectives expected in the study be stated following the paragraph.

1. Establishing the optimization module in this simulation software (including user interface

and optimization solver).

2. Arbitrarily choosing parameters using in the software to form the design variables, and assign any needed objectives such as Energy Efficiency Ratio (EER) as an objective function, and define the constraint conditions. The module hopes to reduce the difficulties to link between analysis software and optimization solvers. By utilizing an optimum solver to find the best design variables group in different constraints and cost functions.
3. By applying the module to expand the parameters design within reciprocating compressors.
4. Integrating an independent optimization module can utilize in other kinds of simulation programs with friendly and easy user interface for developing optimization in various kind of analysis.

The brief descriptions for each chapter are as below:

Chapter 1 is the simple introduction of the background and objective of this study.

Chapter 2 is the statement of the simulation software using for the hermitic reciprocating compressors.

Chapter 3 illustrates the construction, combination and procedure of the optimization module.

Chapter 4 is the results and discussions about the valve optimization problem.

Chapter 5 includes conclusions and further works of the study.

CHAPTER 2

INTRODUCTION OF RECIPROCATING COMPRESSOR SIMULATION SOFTWARE

The software for simulating reciprocating compressor is introduced in this chapter. The software integrates analyses in thermodynamics, kinematics and dynamics of mechanism of the reciprocating compressor. Among, the critical component, reed valves using in reciprocating compressor is studied and analyzed in the program. Besides, it develops a complete dynamic force analysis of the crankcase and a journal-bearing model for predicting mechanical efficiency and performance in the system. The structure of the software is described in proper order below.

2.1 **Thermodynamics Analysis**

The thermodynamics module established is according to some principles including fluid dynamics, thermodynamics, and takes cylinder、suction chamber, discharge chamber, suction muffle, discharge muffle and suction plenum as independent six control volumes [1], Fig-2.1.1 shows the six divisions. Therefore, it considers mass conservation and energy conservation law, form the ordinary different equations. Then the process uses Roung-Kutta method [2] to solve the related equations and finally finds the pressure, temperature, mass flow and other respective effect in the compressor.

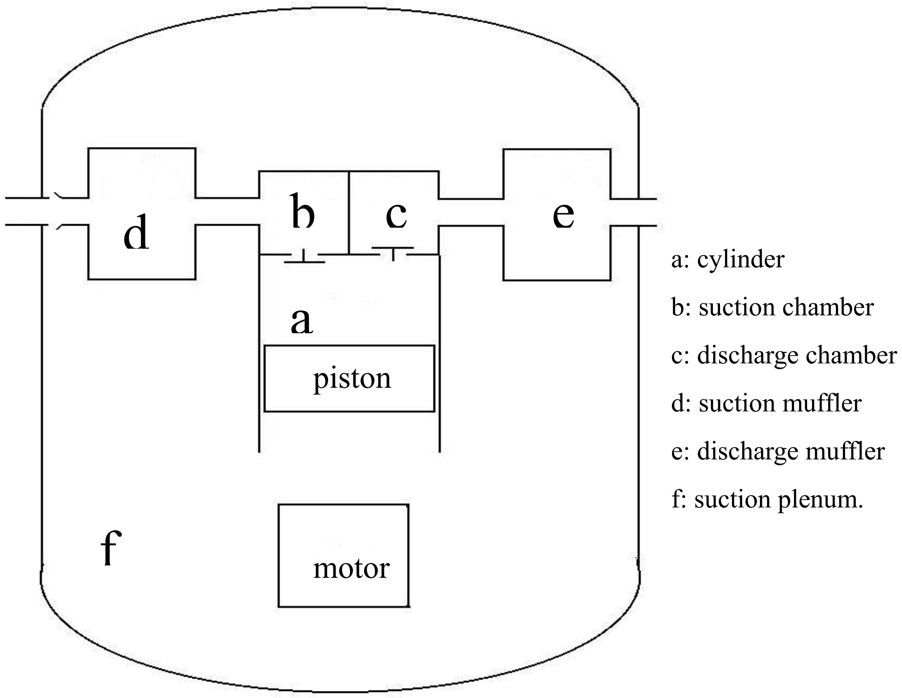


Fig- 2.1.1 The six control volume division [1]

2.2 **Dynamics Analysis**

The software contains five models, about slider-crank mechanism, balancing of crankshaft, crankshaft and journal bearing, mechanical power loss and crankcase vibration analysis models. There are simple descriptions of these five models [3].

2.2.1 Slider-Crank Mechanism

As show in Fig- 2.2.1, is the slider-crank mechanism. This model discusses kinematics and dynamics analyses. It consists of a piston, connecting rod and crankshaft. The force exerted on the piston and the torque applied to the rotor determines its motion. The kinematics is performed for the expressions developed for the inertia force for each of the moving members.

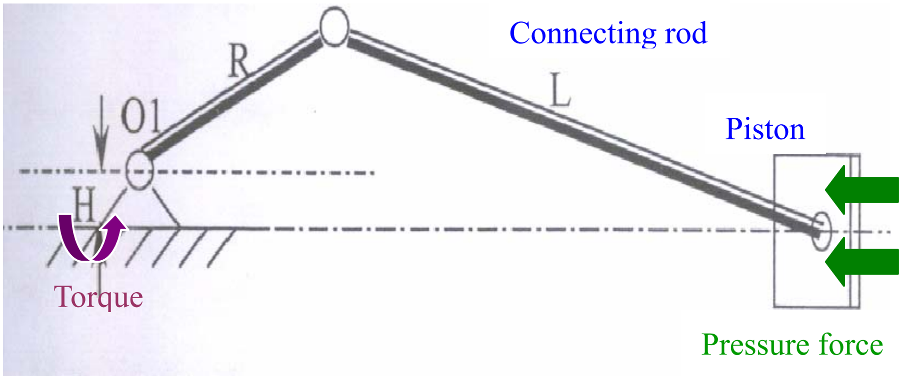


Fig- 2.2.1 The slider-crank mechanism

In dynamic forces analysis, beginning with drawing the free body diagram of the slider-crank mechanism, and assuming all the moving members are rigid bodies. Using Newton's second law and d'Alembert's principle can derives the equations.

2.2.2 Balancing of Crankshaft

In high speed rotating machinery, the dynamic force transmitted to the frame of the machine may cause some serious problems because those forces are time varying and may form vibratory motion to the frame. The vibration and the following noise may change the machine performance and can influence the structural integrity of the machine by resonance or natural-frequency vibration. Besides, the noise can make human discomfort.

A net unbalanced force acting on the frame of a machine is called shaking force. By adding or removing mass form different machine members, the redistribution of mass, to eliminate the shaking force and shaking moment. In Fig- 2.2.2, the slider-crank mechanism of reciprocating machines will produce imbalance while rotating. A rotating counterbalance is used to neutralize the rotating balance.

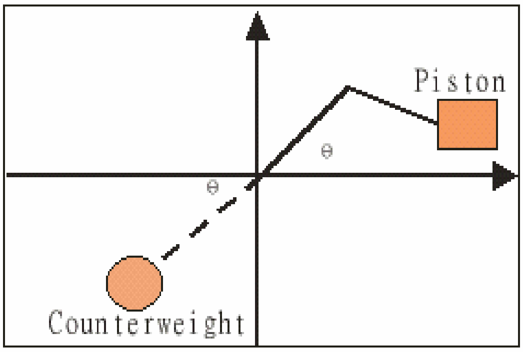


Fig- 2.2.2 Counterweight of the slider-crank mechanism [3]

Dynamical balancing of smaller reciprocating machines, is very difficult because of its inadequate volume. The counterweight can't be placed on opposite side of the crank, but can be shifted suitable distance parallel to the crankshaft below the slider-crank mechanism, and another reacting mass should be added to balance arising from the shifting counterweight. Fig- 2.2.3 illustrates vertical view of the counterweight of the slider-crank mechanism. Applying the equations derived from above analysis model, the minimization of the shaking force can be found.

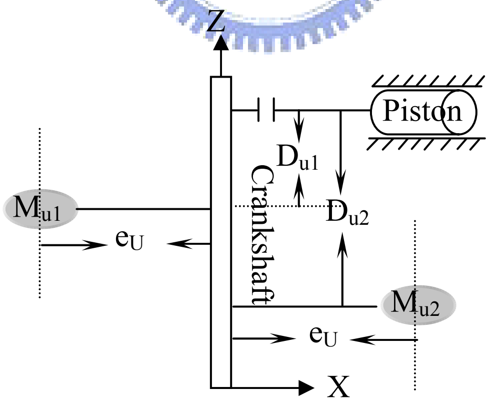


Fig- 2.2.3 The vertical view [3]

2.2.3 Crankshaft Journal Bearing

In order to get mechanical efficiency and performance of the reciprocating compressor, the system should find the frictional loss causing by dynamically loaded crankshaft journal bearing. A complete dynamic force analysis has been finished and the resultant forces acting on the journal bearings between the crankshaft and the slider-crank mechanism. The optimal counterweights have been evaluated to balance off the shaking force formed while the slider-crank mechanism rotating. Fig- 2.2.4 shows the applied counterweight and the main and second journal bearings of the crankshaft.

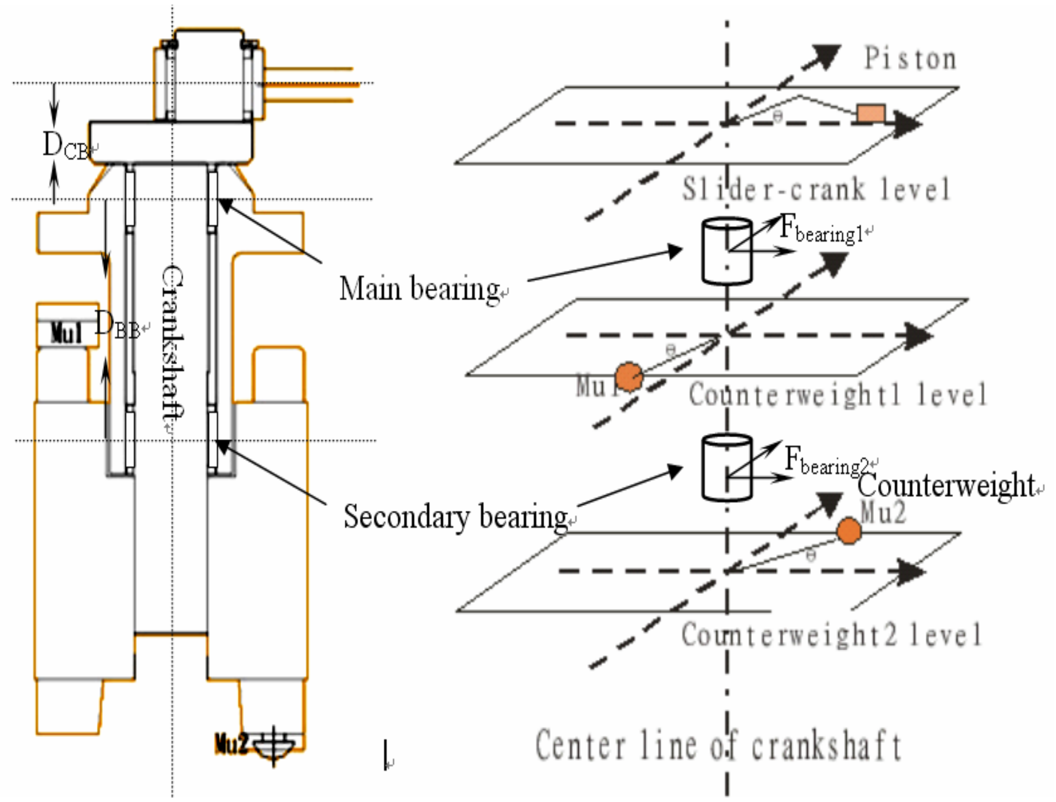


Fig- 2.2.4 Journal bearing of the crankshaft [3]

2.2.4 The Mechanical Power Loss

The correct evaluation to lower the mechanical power loss is very important in developing and designing high efficient compressor. Therefore, finding the equivalent friction coefficient between the moving members in the compressor is very important and can use these coefficients for the simulation. The coefficient can be found from experiment or theoretical calculation, but looking for friction loss is difficult in experimental measuring. Hence, the analysis uses theoretical calculation to get the mechanical frictional loss.

Considering the friction bearing part of the Reciprocating Compressor, all the join of the moving members are journal bearings that are often employed as the main bearing. Those bearings support or transmit loads and operate over a long period of time under the conditions of oil-film pressure. The compressor applies the journal bearings showing as Fig- 2.2.5 and four journal bearings can be found respectively set on the crankshaft marked by A and B, between the connecting rod and the crank marked by C, and between the piston and the connecting rod marked by D.

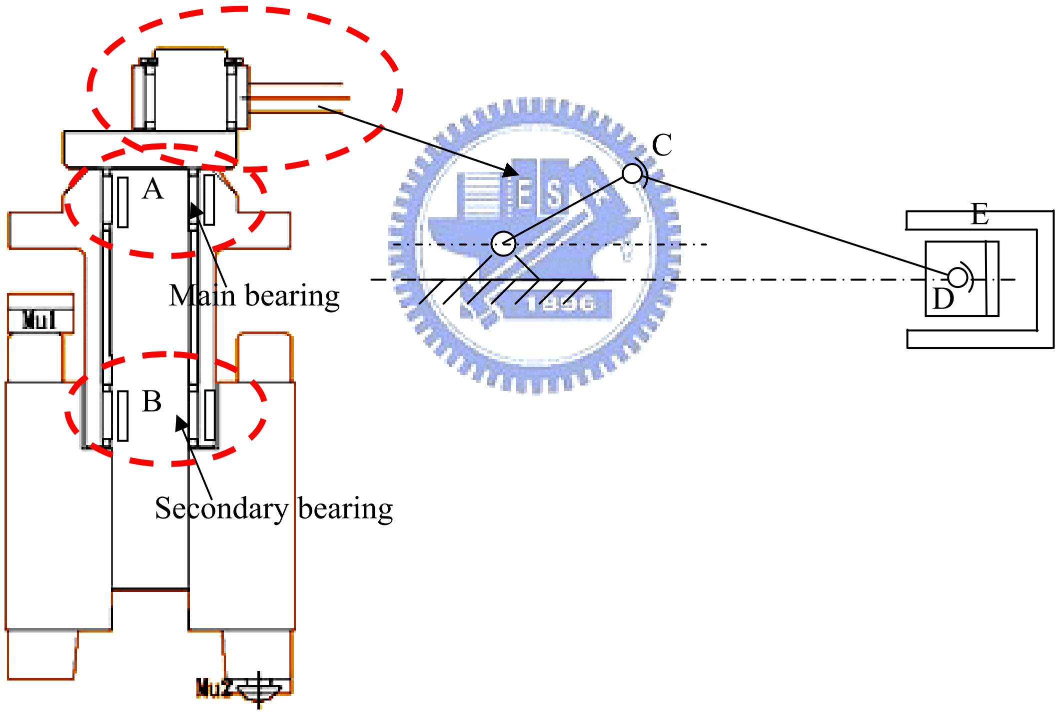


Fig- 2.2.5 Bearing part of the compressor [3]

Hydrodynamic lubrication means that the load-carrying surfaces of the bearings are separated by a relatively thick film of lubricant and can prevent metal-to-metal contact. While operating, the bearings will take large dynamic loading and be in such a severe requirement of high speed and surface temperature. The bearings may fail due to wear, fatigue, corrosion and fluid erosion if the operating parameters exceed certain limits. Wear causing by friction is the most important form of failure, so lubrication is the efficient way of reducing friction wear.

To maintain the hydrodynamic lubrication, it should notice the difference between the boundary and full-fill lubrication. In liquid friction theory, when lubricating oil film forms continuously, the expression of the load-carrying capacity of the journal bearing can be got and can use Sommerfeld number related to the friction coefficient to solve the friction loss of the journal bearing.

2.2.5 Crankcase Vibration Analysis

The suspension system of a reciprocating hermetic compressor must be designed for long life and quiet operation. It must have enough stiffness to accept thousands of cycles and the stresses induced during compressor operation. Besides, it must be soft for low noise transmission [4]. The inertia forces, unbalance forces, and the forces exerted on the slider-crank mechanism and on the crankshaft showing in the previous sections are integrated to form a mathematical analysis of compressor motion within a housing.

As described in [5], all the equations for the resultant forces exerted on the slider-crank are solved assuming constant angular acceleration during a small angular displacement.

All the forces that act as applied forces to the six-degree-of-freedom crankcase are shown in Fig- 2.2.6. The inputs of the crankcase vibration model are the pressure in the cylinder and the torque applied on the rotor by the motor. The motor stator reaction torque is generated by the “Motor-Speed-Torque curves” of induction motors and the cylinder pressure is calculated by the six-control-volume simulation module.

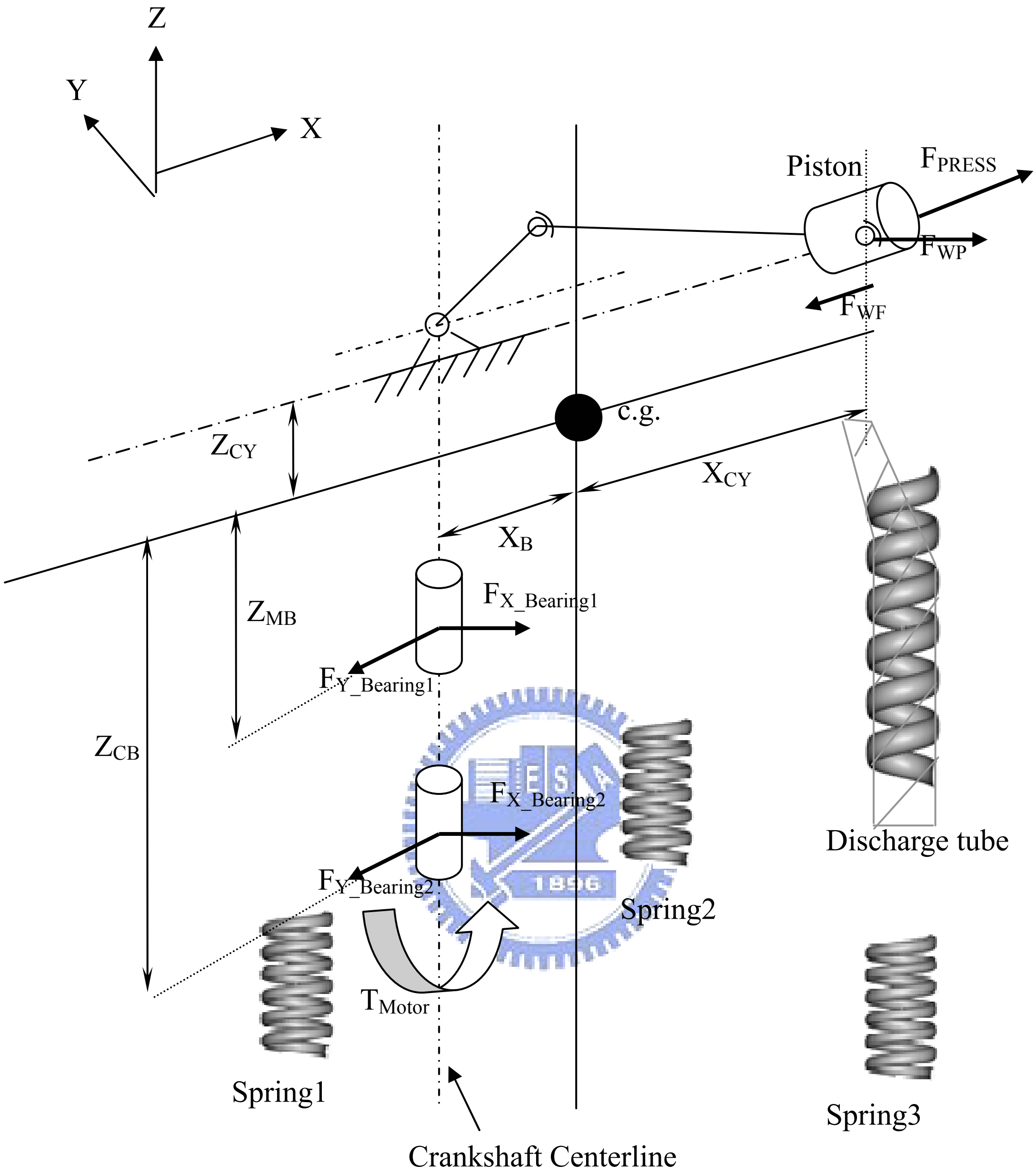


Fig- 2.2.6 Forces on crankcase arising from mechanism and pressure force [3]

2.2.6 The Dynamic Simulation Method

Fig- 2.2.7 expresses the whole flow chart of the reciprocating compressor dynamic analysis. The mathematic models using in the simulation have been brief illustrated in the preceding sections. At first, the unbalanced force analysis of the crankcase should be performed to get the counterweights for the consequent power loss analysis. The procedure for solving the compressor efficiency includes calculating the crank-case mechanism analysis, the journal bearing analysis, and the power of the crankcase while operating. After getting mechanical power loss, the input power of the motor is acquired by experiment data. The updated parameters return to the slider-crank mechanism analysis to perform the next step kinematics and dynamic forces analysis. This iteration calculating converges after the percentage error of total power loss of motor is smaller than a minimum. The software uses the Runge-Kutta method [2] for calculating the governing equation of the crankcase.

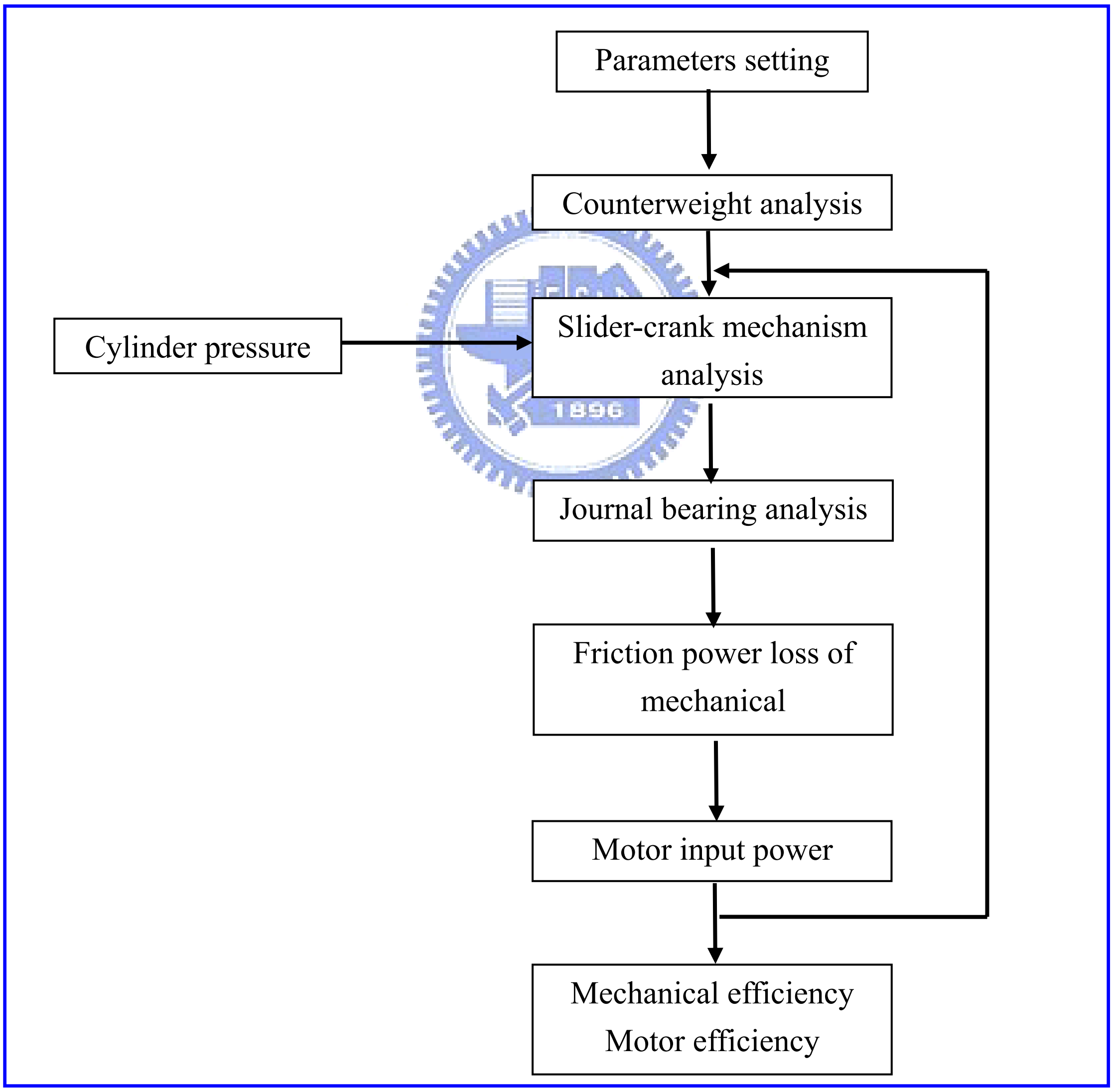


Fig- 2.2.7 The flow chart of the dynamic analysis

2.3 **Reed Valve Vibration Analysis**

The typical used reed valve will be introduced and a mathematical model of valve vibration will be constructed Appendix [A] . The profile and parameters of the valve will be analyzed and applied in the simulation program.

2.3.1 The Reed Valve Design

When gas pressure overcomes the spring force of the reed and it bends to allow gas flow. The reed springs backs to seal the port after the pressure dropping. If the valve motion is abnormal, the compressor efficiency will obviously descend. Hence it is the key element to control suction and discharge gas flow in the reciprocating compressor. Fig- 2.3.1 shows the geometry of the valve plate.

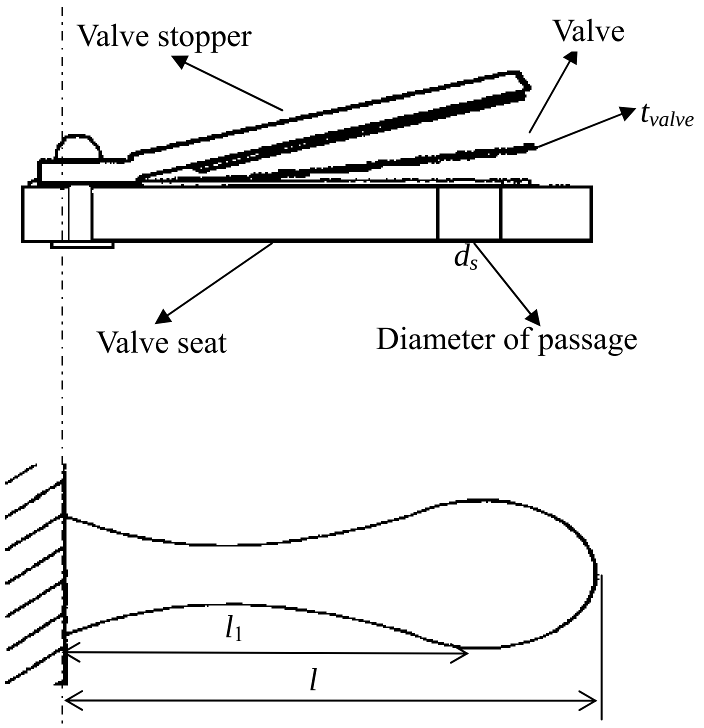


Fig- 2.3.1 Geometry of the reed valve [3]

In reed valve design, reducing the impacting velocity to the seat to protect the valve from crack or other dangerous situations is very important. The valves are activated automatically by the pressure difference through valves. The valves' geometry is simple but enormously affects the performance. If the valves form unsteady motion, the compression and capacity efficiency could vary severely. In general, valve lift and valve area affects gas velocity and must design carefully. It causes over-compression if the valve plate is too stiff, and generates unnecessary fluctuation if too soft.

2.3.2 Dynamics of Valve

The mathematical model usually to describe the reed valve motion is considered as a linear model and applied the beam equation to describe the motion of the valve plate. If the vibration of a reed valve involves amplitude that is small enough, taking the valve motion as a linear model [6] is suitable. But since the displacement of the reed valve may several times to the thickness of the valve, it has enormous amplitude in the actual valve vibration and must use nonlinear model [7] to find the more accurate results.

The effective flow and forces areas of valves are important parameters for the numerical simulation of reciprocating hermetic compressor. Various valve, port and seat geometries influence the effective flow area and the effective force area.

The effective area is defined a multiplication of the flow coefficient and the valve passage area. It will change with the different valve lifting. The effective force area is found by measuring the thrust force on the reed valve and the pressure difference across the valve assembly. The detailed description about valve dynamics is showed in Appendix [A] .

The mathematical formulation in the software uses the Lagrangian method to derive the governing equation for a cantilever type valve. The kinetic energy, potential energy, and the work of external load could be obtained by applying the Lagrangian's equations. The assumed-modes method [8] is used to simplify the governing equations of the reed valve vibration and it's useful for simplifying the complicated valve vibration and generates a nonlinear strain term in the motion equation to represent the dynamic displacement of valve and finally, using the Runge-Kutta method to perform a numerical simulation for solving those differential equations.

2.4 **Software Process**

The software is programmed by using the standard window Graphical User Inter-face and can make users familiar with all the bottoms and pages and they will also find the resulting tables or graphics of a simulation showing just select a simple item in the list-box. Fig- 2.4.1 illustrates the software process and it contains five modules and each one has a specific calculating kernel that couldn't work independently. The brief statement of the five modules is as following:

1. Parameter setting: setting the slider-crank parameters, journal bearing parameters, refrigerant and lubricating oil parameters, simulation step size, and the thermal simulation initial conditions, etc.
2. Valve simulation: Calculating the displacement of the suction and discharge valves under the known valve profile and other characters.
3. Thermal simulation: the module is used to find the solutions of the pressure, mass flow or other respective effects on the compressor capacity and cylinder.
4. Mechanism simulation: in order to reduce the shaking force while operating, the module calculates the counterweights of the internal structure and the resultant forces acting on the join of the moving frames and the slider-crank motion analysis. Then the mechanical friction loss will be obtained by the journal bearing analysis.

5. The simulation results: the module will show the complete outcome including the efficiencies and EER of the compressor when the mechanism and thermal simulation have been done.

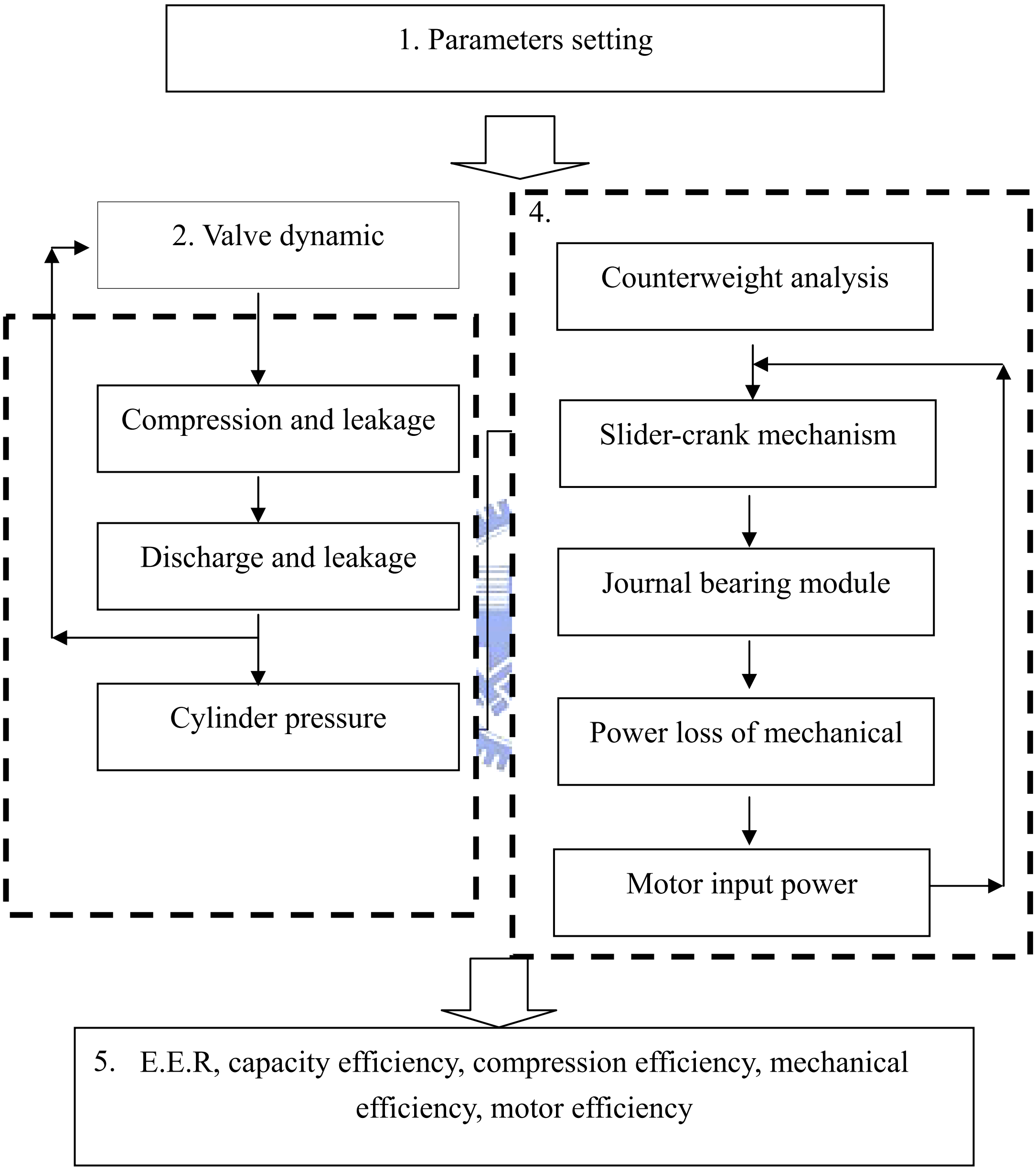


Fig- 2.4.1 The program procedure [3]

2.5 **Simulation Results and Remarks**

After the forces exerted on the slider-crank frames and the crankshaft has been solved, the mechanical power loss and the compressor efficiencies can then be done.

2.5.1 Dynamic Forces Analysis of the Slider-Crank

In designing the reciprocating compressor, the dynamic forces are very important consideration. It must notice the magnitude of the forces on the slider-crank mechanism and the torques to act on the frames. The kinematics analysis of the slider-crank mechanism has been proved and dynamic forces may be approximate true if the mechanical efficiency is confirmed.

2.5.2 Mechanical Power Loss and Compressor Efficiencies

The sum of the friction power and the actual compression power is the total mechanical power loss of a compressor. Using the total mechanical power loss, the mechanical efficiency can be obtained to evaluate the compressor energy efficiency ratio (E.E.R). The following equations illustrate the calculation of the compressor efficiencies and E.E.R.

$$\text{Mechanical efficiency} = \frac{\text{Actual compression power}}{\text{Actual comression power} + \text{Mechanical power loss}} \tag{2.5.1}$$

$$\text{Motor efficiency} = \frac{\text{Actual compression power} + \text{Mechanical power loss}}{\text{Motor input power}} \tag{2.5.2}$$

$$\text{E.E.R} = \frac{\text{Capacity}}{\text{Motor input power}} \tag{2.5.3}$$

There is a good agreement obtained on E.E.R from both the simulation software results and the experiment. But the software still has two insufficient considerations, oil-refrigerant mixture and the Sommerfeld number related to coefficient of friction.

2.5.3 Results of the Vibration Analysis of the Reed Valve

The accuracy of natural frequency of the suction and discharge valves has checked with finite element method in "ANSYS". The assumed-modes method for simplifying the valve vibration equation can be proved usability [3], [7].

The flow of refrigerant and the over compression loss in the cylinder could be changed slightly due to the oscillatory motion of the valves as the damping coefficient changing and it can also affect the efficiency of the capacity and the compression. In general, the small valve damping coefficients results in bigger amplitude of vibration as valve lifting and it may reduce the reliability of the valve working life, while higher damping coefficient results in a smoother valve motion.

In practice, the damping effects on the reed valve are caused by cushion of the viscous oil or the refrigerant flow that relax the impacts. Hence there exists a value of damping coefficient agreeing with the experiment measuring.

From simulation results, though the thinner valve causes no restriction to the flow in the cylinder and therefore enlarges the effect force area, damage may happen and shorten the valve reliability. So there exist an optimum thickness that can compromises between working life and the valve flow area.

2.5.4 The Crankcase Vibration Analysis Results

The steady-state conditions will never be reached if the crankcase vibration system is without damping. Thus, Coulomb damping is imported into the crankcase vibration equations to obtain the steady-state crankcase movements and it is sufficient to dampen transients, but it may also affect the maximum displacement and rotation of the crankcase. Therefore, the damping coefficients have to be trial and error selected for the springs so that the maximum crankcase vibration can change less than five percent.

Model of the six-degree-of-freedom crankcase can calculate the displacement and rotation of any points on the crankcase body and what elements affect the vibration of the crankcase critically in the compressor body can also be derived to enable good reliability for the compressor operating life.

2.5.5 Remarks

The simulation results were initially verified by the experiments in a calorimeter with ASHRAE test conditions (the calorimeter accuracy is 96% for energy efficiency ratio E.E.R). Table- 2.5.1 reveals both the results for the same testing conditions. The E.E.R. result simulated from the software is 0.9395, which is close to the experiment (E.E.R. = 0.9690).

	Unit	Simulation	Experiment
Inhaled pressure	kPa	120	117
Exhaled pressure	kPa	1470	1468
Refrigerant mass flow rate	kg/hr	5.1349	5.2290
capacity of refrigeration	kcal/hr	194.6221	189.7000
E.E.R	kcal/hr-W	0.9395	0.9690

Table- 2.5.1 Simulated and experimental results

The simulation software is developed for small reciprocating compressor. The dynamic, thermodynamic, fluid dynamics, bearing analysis and valve vibration are included in the software. By using the software, the efficiencies and E.E.R of the reciprocating compressor can be predicted while operating.

The software outputs include the thermal and dynamic forces properties and the forces exerted on the slider-crank mechanism or other respective effect. When applying a new refrigerant type or carries out a novel element design such as valve or crankshaft, the operating characteristic become a useful reference.

The study debugs and improves the simulation software to meet conformity with the experiments provided by the Energy and Resources Laboratories (ERL) of Industrial Technology Research Institute (ITRI).

By using the software, it can be found that the main factor affecting the compressor efficiencies is the vibration of the valves. Besides, a good lifting of the suction and discharge valves brings a long valve working life and higher compressor efficiencies with good refrigerant condition.

CHAPTER 3

OPITMIZATION MODULE

By extending the simulation software described in the previous chapter, it is expected that an optimization module can construct and append to the software. The module uses the Borland C++ Builder accompanying with OOP (Object-Oriented Programming) concept to construct the Windows GUI (Graphical User Interface), which provides "What you see is what you get" interface that is friendly and simple operating [9]. Users could select various types of design conditions and proceed to simulate different optimization problems in the reciprocating hermetic compressor software. This chapter will make a detail statement about the optimization module.

3.1 Structure of Optimization Module

The section describes the full structure of the optimization module. First of all, the simulation software described in Chapter 2 is considered as an independent simulation module. The optimization module that constructs to link the simulation module, mainly comprises two sub-modules, user interface and optimization solver. The optimization module then must test and verify the linking feasibility between it and the simulation module, furthermore, check and modify errors for developing optimization. After proving the applicability, it is expected the optimization module may be constructed as an independent and easy separating module to apply the software and even be used for other simulation software.

Fig- 3.1.1 shows the flow chart to integrate the simulation and optimization modules making use of the study. The flow chart mainly includes three parts: user interface, simulation module, and optimization solver. Users initially define and set optimization problems in the user interface sub-module, then make the related arguments as input data and pass it to the simulation module. The simulation module executes the analysis and produces the results called output data. The output data will be transferred to the optimization solver. Through various optimization methods in the optimization solver, new arguments can be generated as new input data, then pass back to the simulation module for next iteration. The details about the procedure presented as following sub-sections.

3.1.1 User Interface

Before commencing optimization, it is necessary to define and formulate the optimization problems. As presented in Fig- 3.1.1, the user interface sub-module help finish this stage. Users just choose what kinds and numbers of design variables, and set the cost function, constraints or bounds in the user interface. The sub-module is friendly and performs easily without coding the software themselves. Besides, it can reduce the some incompatible and time-consuming problems occur in a general manner linking between simulation and optimization. Fig- 3.1.2 to Fig- 3.1.5 shows the schemes of user interface about choosing and setting design variables, constraints, cost function, and involved parameters used in optimization (like convergent criteria). Furthermore, the user interface also can select different optimization method for various optimization problems in this reciprocating compressor simulation software.

3.1.2 Simulation Module

After completing establishment of optimization problems from the user interface, the input data can be formed and then transfers to the simulation module. The simulation module is described in Chapter 2, but the prior simulation software provides only interactive form to set or adjust parameters. So the simulation software is recomposed as a batch type mode in this study. That is to say, the modified simulation software can be executed only imports one input data file [10], [11]. As soon as the software has finished, it generates an output data including many results such as efficiencies or performance in the reciprocating compressor. The output data then sends to optimization solver to proceed with next stage.

3.1.3 Optimization Solver

Referring Fig- 3.1.1 again, from the optimization solver sub-module, the output data is checked whether satisfy the constraint conditions or limitations of defined optimization problem at first. Then it identifies coincidence with the convergent criteria or not. The sub-module procedure will stop if the output data meet the convergent criteria, otherwise using various optimization methods to solve the problem and form a new design variables group as current input data. The current input data then passes back to the simulation module to continue iterations until the final results satisfying the convergent conditions.

This optimization solver “Most” [12] is a multifunctional optimization system tool and provides many methods for different optimization problems. A Sequential Quadratic Programming (SQP) method is selected to deal with single objective optimization problems for continuous variables for accuracy, reliability and efficiency in the optimization module and described below:

SQP method uses Karush-Kuhn-Tucker (KKT) conditions as a basis and can divided three stages [13]. The first is using BFGS (Broyden-Fletcher-Goldfarb-Shanno) method to update the approximate second information for the Lagrange function (like Hessian matrix for keeping the Hessian approximation positive definite). After calculating Hessian of the

Lagrange function, QP(Quadratic Programming) sub-problem can be solved at each iteration to get the solution as a search direction for next stage. The third are the determination of step size and descent function calculation. Using the search direction generated before, a step size can be resolved to minimize the descent function where is defined by Pshenichy and Danilin [14].

Besides SQP method, the modified branch-and-bound algorithm which converts discontinuous design space into a continuous one by dropping discontinuous restrictions is used to solve discrete optimization problems [15]. For multi-objective optimization, the solver also provides Compromise Programming, Goal Programming and the Surrogate Worth Trade-off method for designer to find the best compromise solution [12], [16], [17].

3.1.4 Considerations

Consulting Fig- 3.1.1 and previous description, it is found that some advantages in the optimization module. These advantages state bellow:

1. If the design consideration of the optimization problem changes, the user interface can easily set and complete the formulation, moreover, establish required setting conditions in the problem for the convenience of developing simulation and optimization.
2. The optimization solver provides several algorithms for dealing with different types of optimization problems. The related algorithmic parameters used in different optimization methods are decided appropriately and automatically in the module.
3. Due to the optimization are separated distinctly, the optimization module can even utilize in other analysis simulation programs readily.

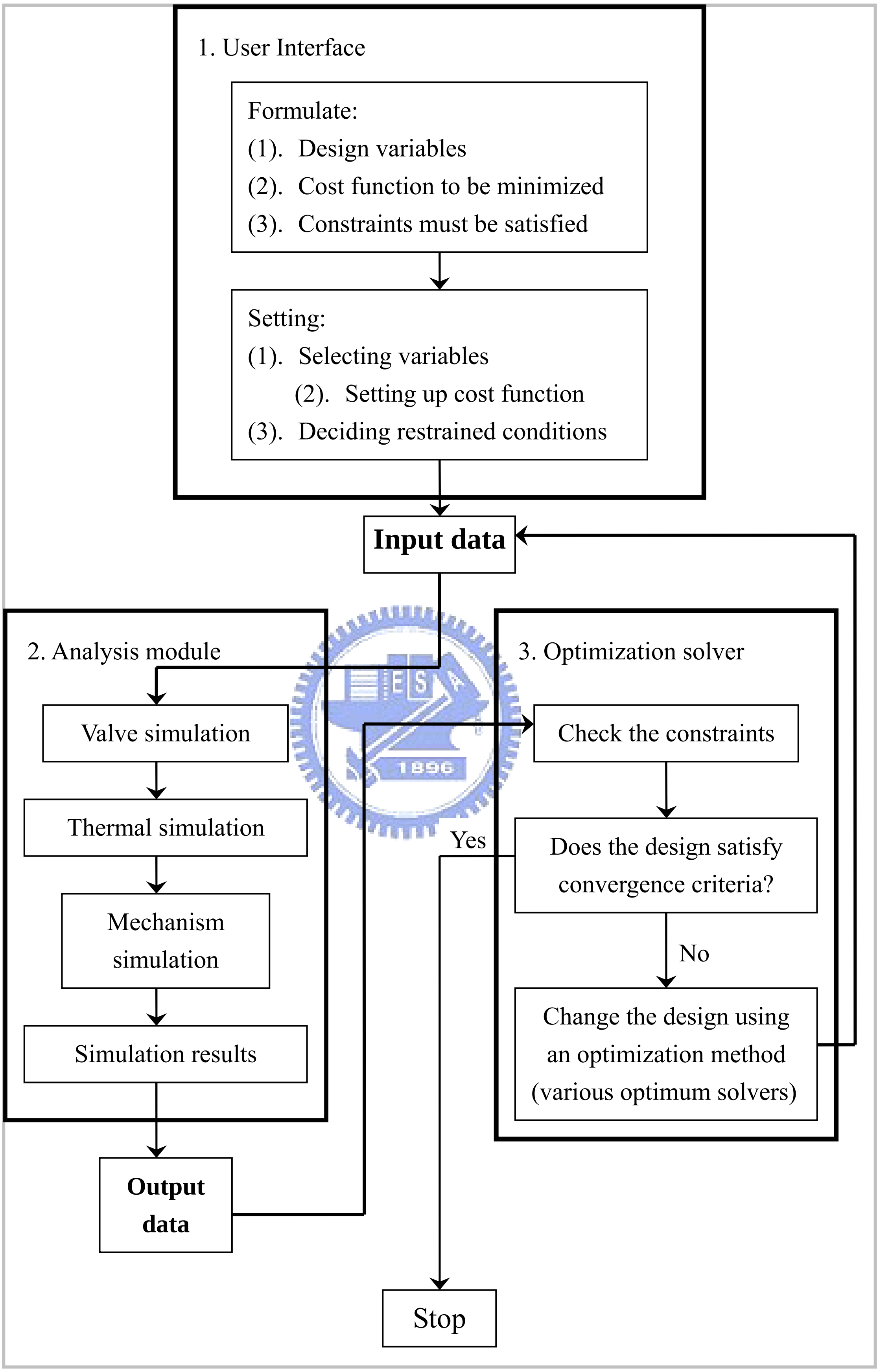


Fig- 3.1.1 Optimization procedure

Fig- 3.1.2 Design variables setting page

Fig- 3.1.3 Constraints setting page

Fig- 3.1.4 Cost function setting page

Fig- 3.1.5 Related parameters setting page

3.2 **Design Variables Setting In User Interface**

For an optimization problem, how to set and decide the parameters as design variables is a major consideration. It involves many requirements or conditions in various fields. For the hermitic reciprocating compressor, the simulation explained in Chapter 2 combines valve dynamic, thermal simulation, mechanism simulation, all have input parameters in respective parts. By user interface, all these parameters can be selected arbitrarily when formulating optimization problems. All the parameters from different parts are explained as following sub-sections, and the discussion about those in the optimization problems are illustrated in Chapter 4.

3.2.1 Numerical Simulation Parameters

The setting page for numerical simulation parameters using in the optimization module are presented in Fig- 3.2.1. Users can easily mark or cancel parameters they need in defined optimization problems. It includes such as refrigerant properties of inlet and outlet of the compressor, motor rotational speed, numerical step-size, etc. Table- 3.2.1 lists the 11 parameters and its default values applying in numerical simulation pages of the optimization module.

3.2.2 Slider-Crank Parameters Setting

The parameters of slider-crank mechanism are setting here and the related geometries and sizes are constructed in this page showed in Fig- 3.2.2. The decisions in this part may affect the behaviors in the slider-crank mechanism results. If users want to focus optimization problems in this mechanism design, the setting page is a beginning. Table- 3.2.2 describes the

11 parameters can be defined or marked in this page and their initial values.

3.2.3 Eccentric Shaft and Mass Redistribution Setting

The Fig- 3.2.3 shows setting page about mass redistribution and shaft in user interface, it has 7 parameters including positions of the shifting counterweight. It is also a consideration if users have requirement for optimization problems concerning eccentric shaft distance or position, or the positions of the shifting counterweight. The 7 parameters and its default values in the setting pages are described in Table- 3.2.3.

3.2.4 Bearing Parameters Setting

The four bearing properties and dimensions and initial values are listed Table- 3.2.4 Influences in bearing properties and lubrication conditions are important factors for mechanical power loss. In the user interface, bearing properties can be chose and defined as design variables for optimization. Fig- 3.2.4 shows the setting pages for the four bearings between slider-crank and crankshaft.

3.2.5 The Control Volume Parameters Setting

In the previous simulation software of the reciprocating compressor it divides six control volumes for simulation. All related parameters are introduced in the optimization module. The user interface could make them as design variables in accordance with various optimization problems. Descriptions about individual control volume as follows:

Chambers size setting

As show in Fig- 3.2.5, this setting page defines the related size or thermal parameters of suction and discharge chamber. In addition, the cylinder parameters could be set in it. Table- 3.2.5 expresses the parameters and its initial value for simulation. Though the related parameters in chambers and cylinder are universal for industrial use, the optimization module also provides them the feasibility as design variables for developing optimization in the reciprocating compressor.

Mufflers parameters setting

The suction and discharge mufflers are dimensioned and selected in this setting page showed in Fig- 3.2.6. The parameters and its default values are listed in Table- 3.2.6.

Suction plenum and passages conditions parameters setting

The parameters about suction plenum and channels are displayed in Fig- 3.2.7. This setting page provides various parameters in suction plenum and passages conditions for users to formulate optimization problems. Table- 3.2.7 displays the parameters and its initial values.

3.2.6 Suction and Discharge Reed Valves Setting

The sizes, geometry, and physical properties of the suction and discharge valves are indicated and chose in this setting page. Due to [6], [7] and previous studies, it is known that the reed valve behaviors take a key role because of requirements for durability, stiffness, and operating life in the reciprocating compressor. Therefore, this study takes great efforts to proceeding design optimization of the valve characteristics and these results and discussions are illustrated in Chapter 4.

The suction and discharge reed valve setting page are showed in Fig- 3.2.8, Fig- 3.2.9.The related parameters and default values are listed in the Table- 3.2.8.

Fig- 3.2.1 Numerical simulation parameters page

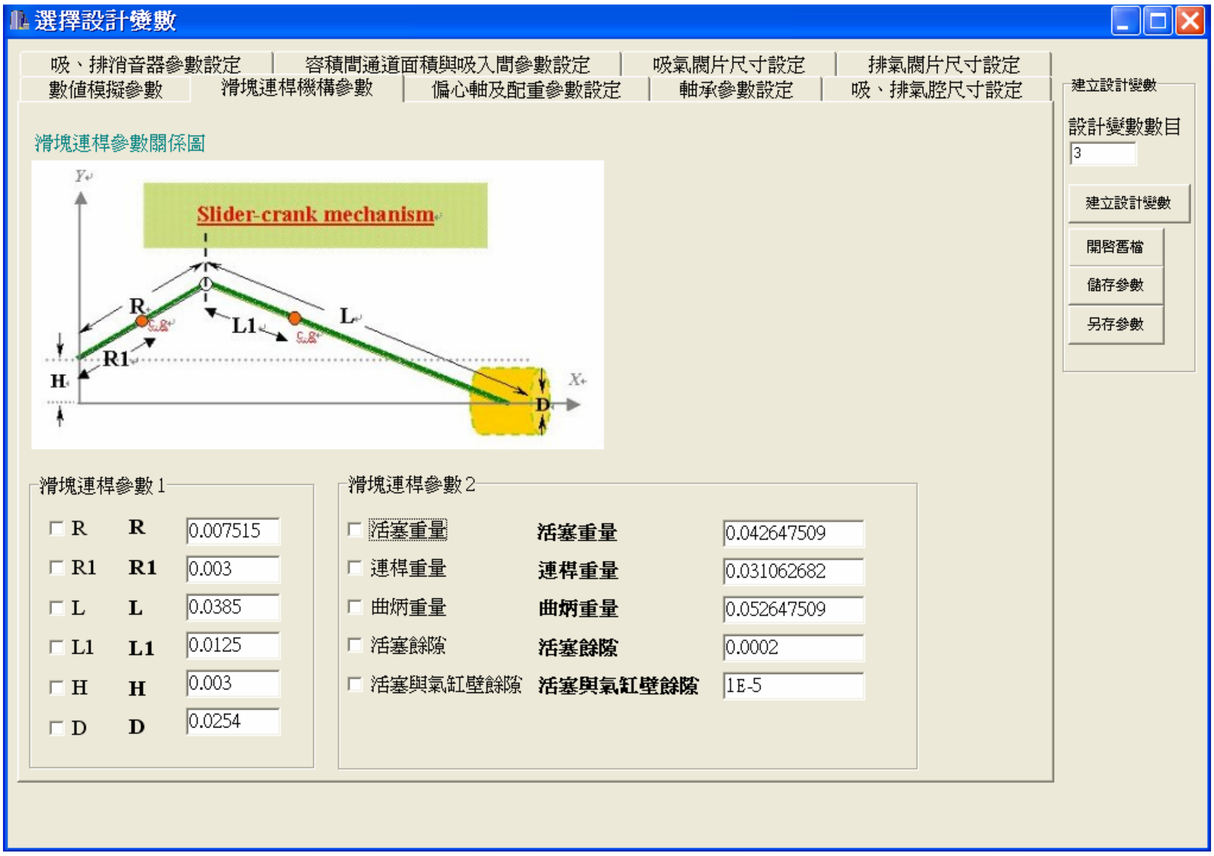


Fig- 3.2.2 Slider-crank parameters page

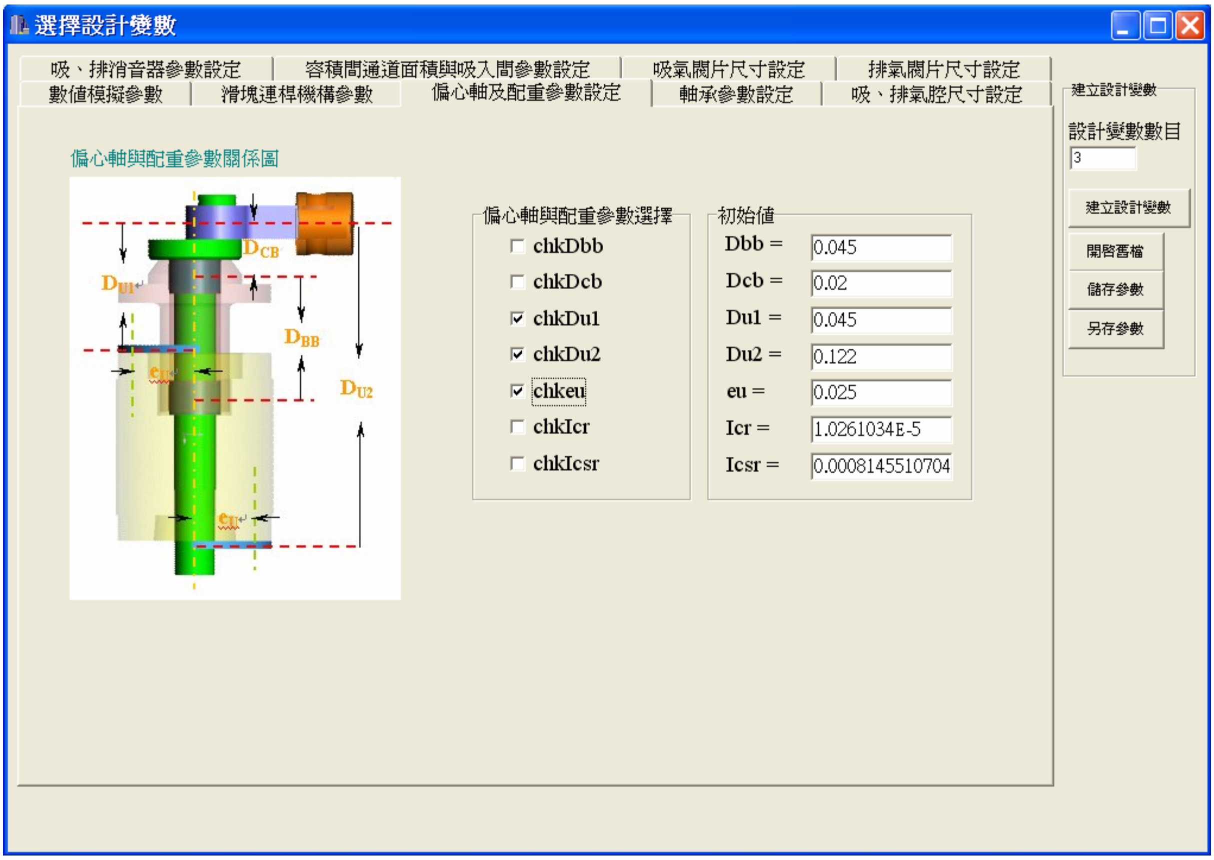


Fig- 3.2.3 Eccentric shaft and mass redistribution page

Fig- 3.2.4 Bearing parameters setting page

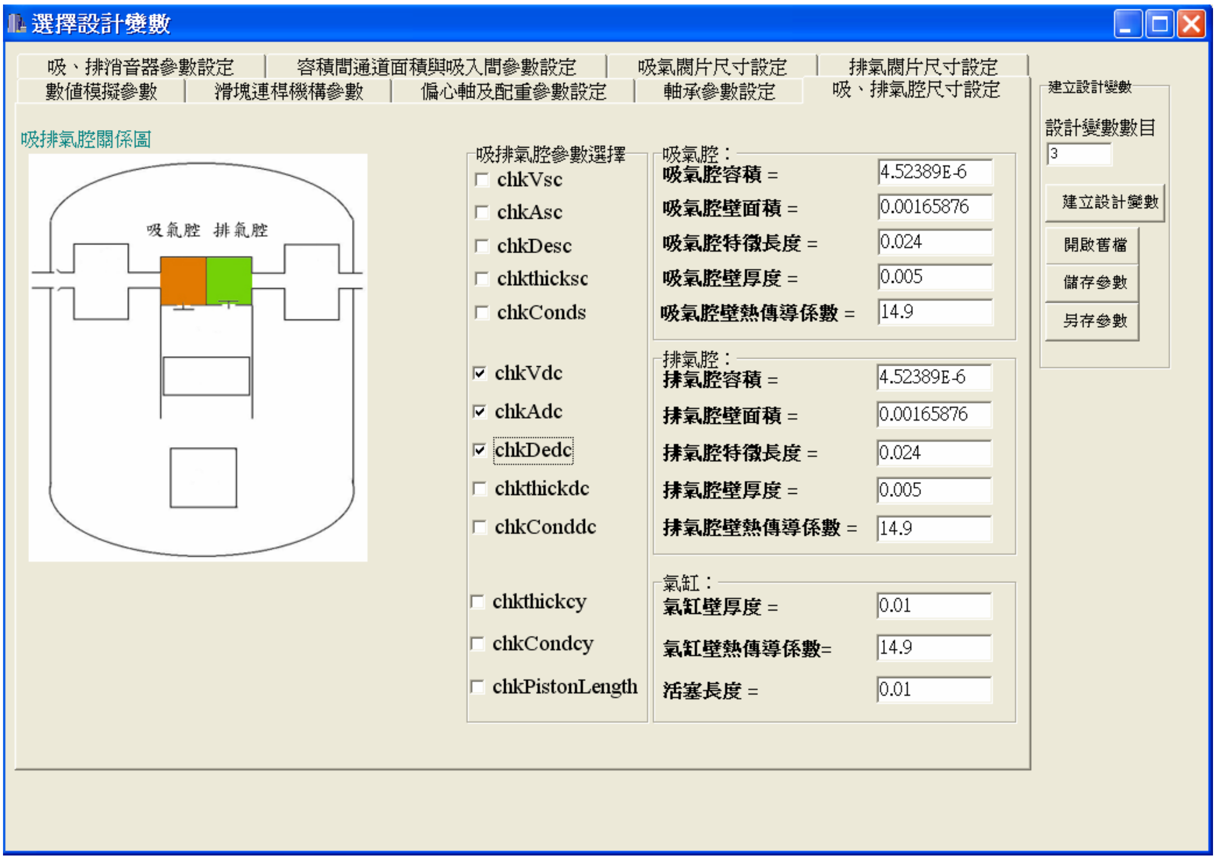


Fig- 3.2.5 Chambers & cylinder parameters setting page

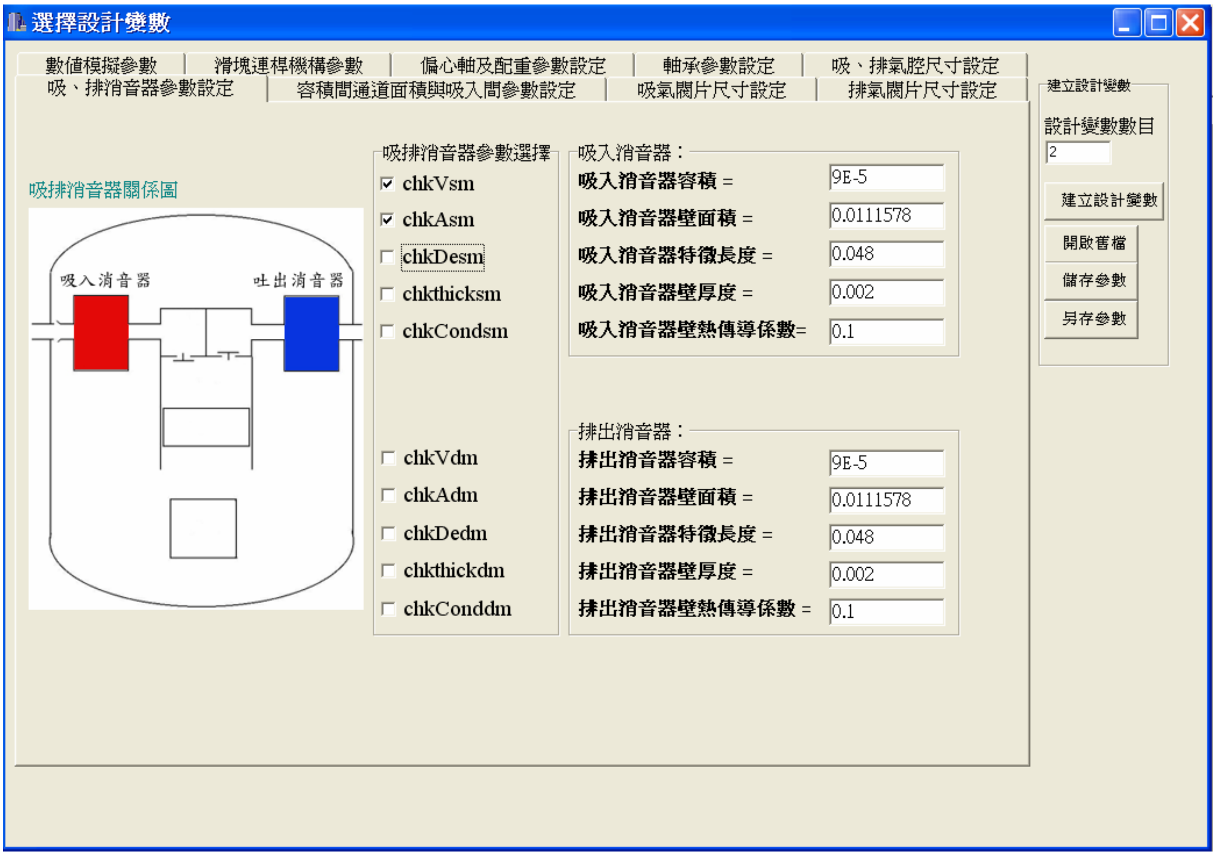


Fig- 3.2.6　Mufflers parameters setting page

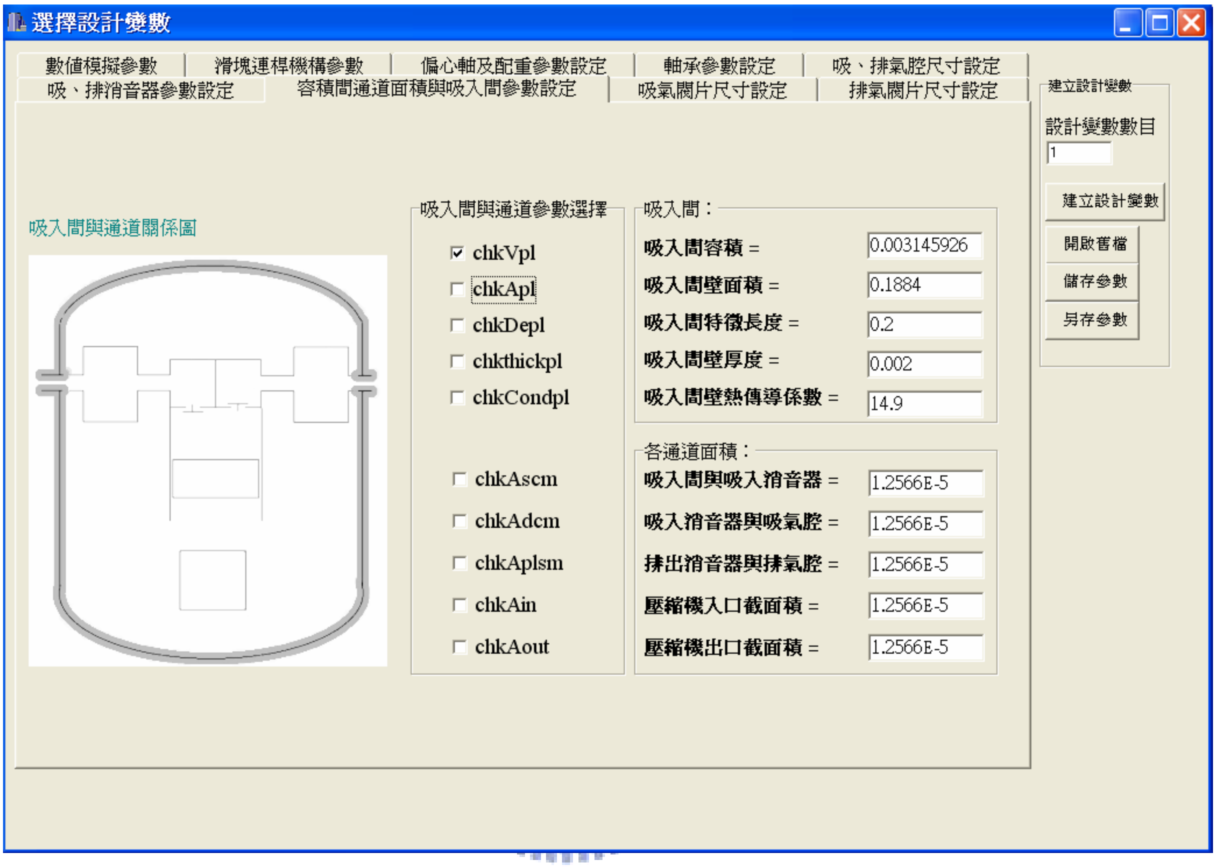


Fig- 3.2.7 Suspend plenum & passages setting page

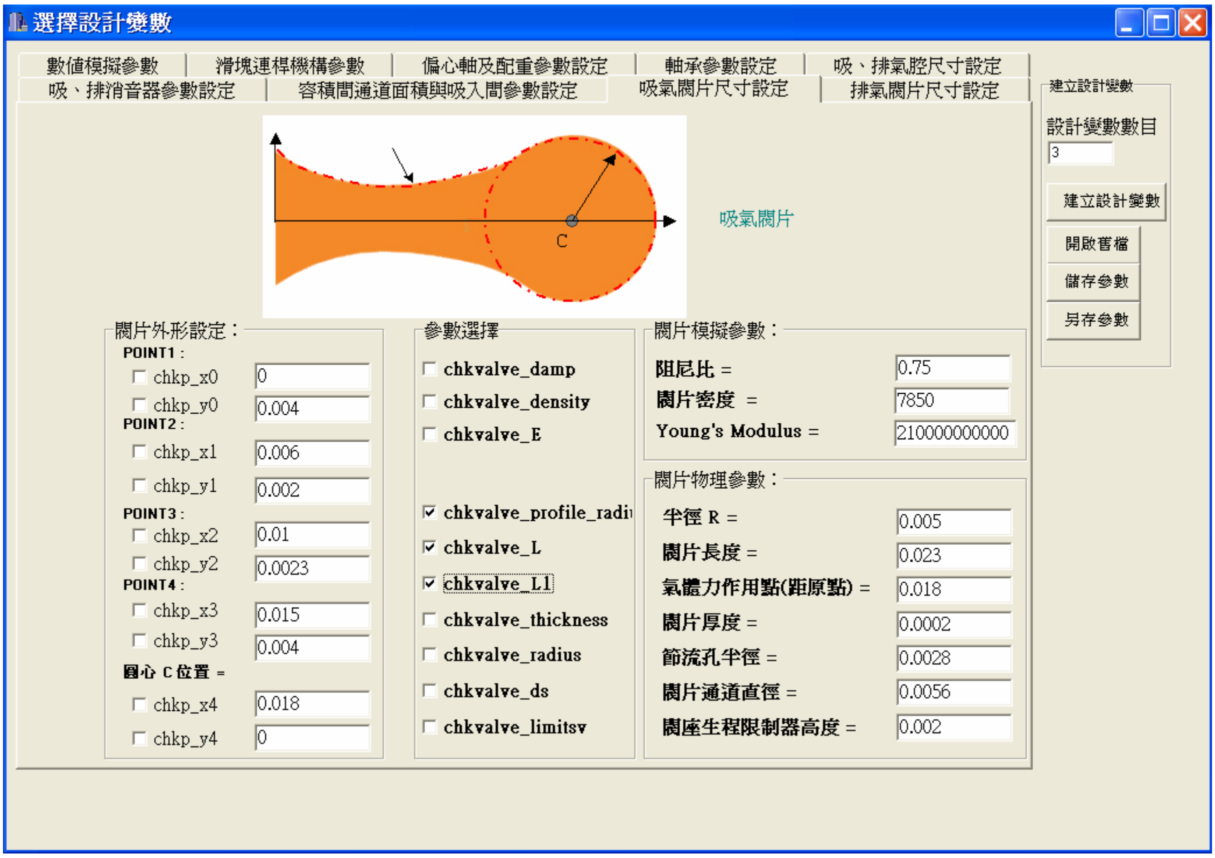


Fig- 3.2.8　Suspection valve setting page

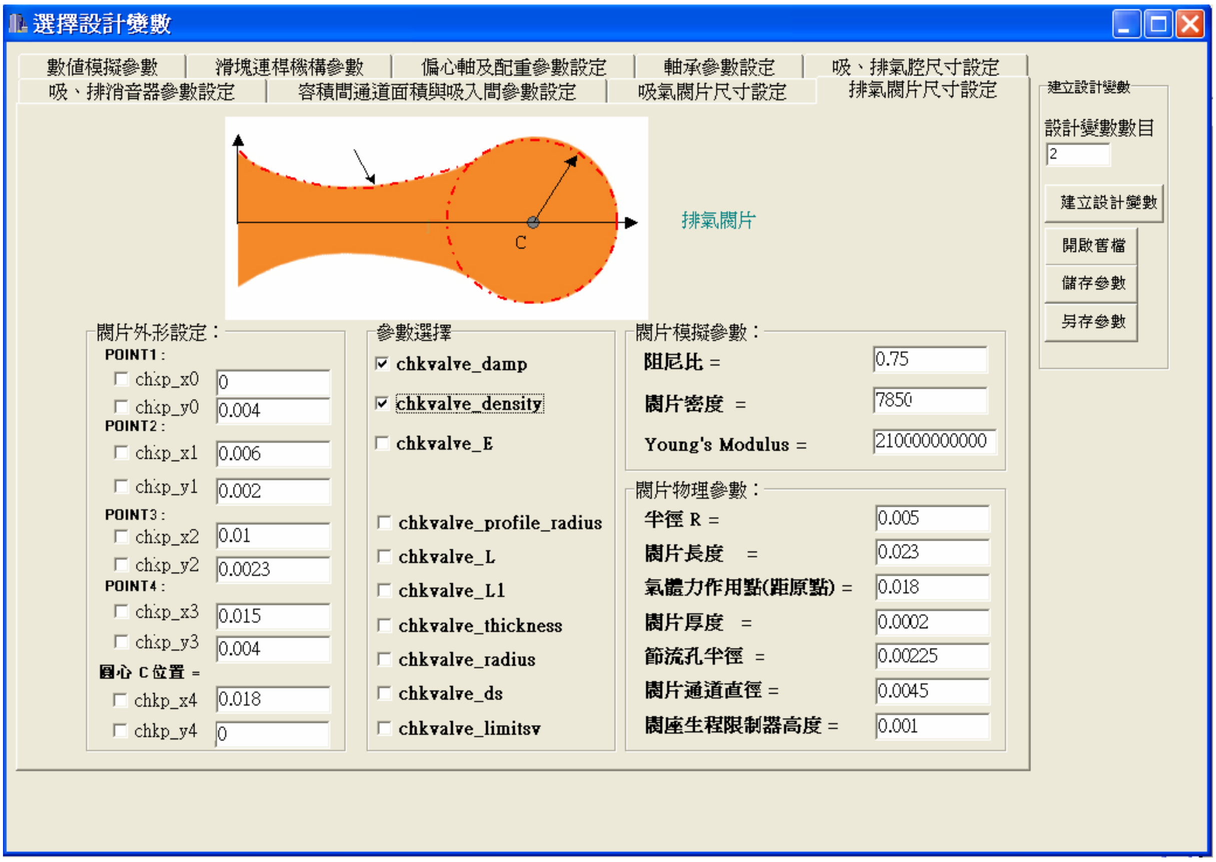


Fig- 3.2.9 Discharge valve setting page

		Default value	Unit
RPM (rpm)		3600	rpm
Step Size (degree)		1	degree
Temp in crankcase (°K)		313	°K
Environment thermal convection coefficient		2	(kJ/hr m^2)
Environment temp		300	°K
Refrigerant properties			
Inlet	Pressure	120	kPa
	Density	4.7999	kg/m^3
	Temp	313	°K
	Enthalpy	437.16	kJ/kg
Outlet	Pressure	1470	kPa
	Enthalpy	278.5	kJ/kg

Table- 3.2.1 Numerical simulation parameters list

slider-crank mechanism					
parameters	default	Unit	parameters	default	Unit
R	0.007515	m	piston weight	4.2648E-02	kg
R1	0.003	m	link weight	3.1063E-02	kg
L	0.0385	m	crank weight	5.2648E-02	kg
L1	0.0125	m	piston clearance	2.0000E-04	m
H	0.003	m	clearance between piston and cylinder	1.0000E-05	m
D	0.0254	m			

Table- 3.2.2 Slider-crank mechanism parameters list

shaft and mass redistribution parameters		
parameters	default	Unit
Dbb	0.045	m
Dcb	0.025	m
Du1	0.045	m
Du2	0.122	m
eu	0.025	m
lcr	1.026E-05	m^4
lcsr	0.000814551	m^4

Table- 3.2.3 Shaft and mass redistribution parameters list

bearing parameters		Unit
clearance ratio	700	
slider-crank related bearing parameters		
A length	0.014	m
A radius	0.008	m
B length	0.014	m
B radius	0.008	m
eccentric shaft related bearing parameters		
C length	0.014	m
C radius	0.008	m
D length	0.014	m
D radius	0.008	m

Table- 3.2.4 Bearing parameters list

suction chamber			discharge chamber			cylinder		
parameters	default	Unit	parameters	default	Unit	parameters	default	Unit
volume	4.53E-06	m^3	volume	4.53E-06	m^3	thickness	0.01	m
area	0.0016588	m^2	area	0.0016588	m^2	thermal conductivity coefficient	14.9	$J/m^2\cdot$℃
characteristic size	0.024	m	characteristic size	0.024	m	length	0.01	m
thickness	0.005	m	thickness	0.005	m			
thermal conductivity coefficient	14.9	$J/m^2\cdot$℃	thermal conductivity coefficient	14.9	J/m2·℃			

Table- 3.2.5 Chamber and cylinder parameters list

suction muffler			discharge muffler		
parameters	default	Unit	parameters	default	Unit
volume	9.00E-05	m^3	volume	9.00E-05	m^3
area	0.0111578	m^2	area	0.0111578	m^2
characteristic length	0.048	m	characteristic size	0.048	m
thickness	0.002	m	thickness	0.002	m
thermal conductivity coefficient	0.1	$J/m^2\cdot$℃	thermal conductivity coefficient	0.1	$J/m^2\cdot$℃

Table- 3.2.6 The mufflers parameters list

suction plenum			passages areas		
parameters	default	Unit	parameters	default	Unit
volume	0.003145926	m^3	suction plunem and suction muffler	1.2566E-05	m^2
area	0.1884	m^2	suction muffler and suction chamber	1.2566E-05	m^2
characteristic length	0.2	m	discharge muffler and discharge chamber	1.2566E-05	m^2
thickness	0.002	m	compressor inlet section area	1.2566E-05	m^2
thermal conductivity coefficient	14.9	$J/m^2 \cdot ℃$	compressor outlet section area	1.2566E-05	m^2

Table- 3.2.7 Plenum and passages parameters list

suction valve			discharge valve		
parameters	default	Unit	parameters	default	Unit
shape			shape		
point1	0		point1	0	
	0.004			0.004	
point2	0.006		point2	0.006	
	0.002			0.002	
point3	0.01		point3	0.01	
	0.0023			0.0023	
Point 4	0.015		point 4	0.015	
	0.004			0.004	
center of circle	0.018		center of circle	0.018	
	0			0	
simulating parameters			simulating parameters		
damping ratio	0.75		damping ratio	0.75	
density	7850	kg/m^3	density	7850	kg/m^3
Young's module	2.10E+11	Pa	Young's module	2.10E+11	Pa
physical parameters			physical parameters		
radius	0.005	m	radius	0.005	m
length	0.023	m	length	0.023	m
action point	0.018	m	action point	0.018	m
thickness	0.0002	m	thickness	0.0002	m
reducing orifice radius	0.0028	m	reducing orifice radius	0.0028	m
valve passage diameter	0.004	m	valve passage diameter	0.004	m
Limiting lift	0.002	m	limiting lift	0.001	m

Table- 3.2.8 Vvalve parameters setting list

3.3 **Cost Function and Constraints Setting**

In consideration of optimization problems, the construction of the mathematic formulation is a key point for solving the problem well. Correct formulation can express the design problem accurately. For this reason, the formulation should exactly choose design parameters as variables, cost function and constraints, furthermore, combine those to image the substance of the formulation in the optimization problem. Section 3.2 has explained the parameters decision, the cost function and constraints setting constructed in user interface will interpret in this section.

3.3.1 Cost Function Setting

The final results (performances or efficiencies) show in the reciprocating compressor simulation software will be stated as the selections of the cost or objective function in optimization problems. The related figure, Fig- 3.3.1 shows the wasted work of various parts of the reciprocating compressor system from simulation, and refrigerant mass flow rate, E.E.R. other performance indices. The users can select different kinds of results or indices as cost function in user interface.

The statement of the performance indices are illustrated as below:

1. Volumetric efficiency:

   $\left[\text{real flow rate}/\text{ideal flow rate}\right]$ (%)

2. Compressor efficiency:

   $\left[\text{ideal compression work}/\text{real compression work}\right]$ (%)

3. Mechanical Efficiency:

   $\left[\text{real compression work}/\text{total mechanical work}\right]$ (%)

4. Motor efficiency:

$\left[\text{bearing output work}/\text{motor output work}\right]$ (%)

5. Flow rate of refrigerant:

   $\left[\text{volumetric efficiency} \times \text{ideal maximum flow rate}\right]$ (kg/hr)

6. Capacity of refrigeration:

   $\left[\text{flow rate of refrigerant} \times \text{enthalpy difference}\right]$ (kcal/hr)

7. Energy efficiency ratio(EER):

   $\left[\text{capacity of refrigeration}/\text{motor input work}\right]$

The users initially plan to decide what the cost function for its problem is and then selects it in the user interface module to do the consequent simulation and optimization. At present, the optimization module adopts the single objective function to reduce the complexity of the problem, and the module also can accomplish discrete and the multi-objective functions optimization problem.

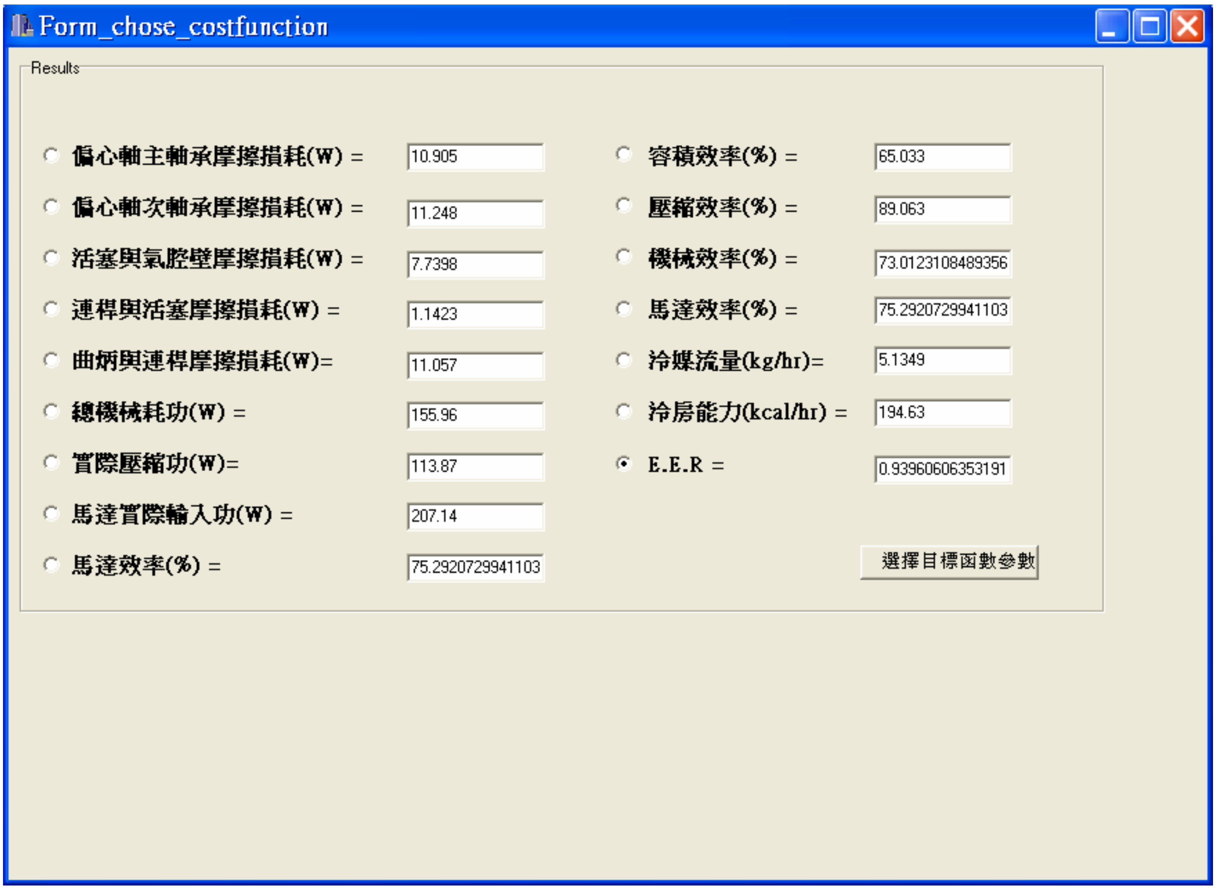


Fig- 3.3.1 Cost function selecting page

3.3.2 Constraints Setting

Design constraints are used to limit the range of design variables and can divide into explicit and implicit constraints, boundary and characteristic constraints, and equal or unequal constraints. In the user interface module, when deciding the design variables for an optimization problem, the users also can set their boundary constraints (lower bound and upper bound) at the same time. As show in Fig- 3.3.2, an example, the numerical simulation variables setting page can select the design variables and identify its ranges.

The constraint conditions just set the boundary constraints in the optimization module presently, but in fact the accurate problem formulation should consider many other kinds of constraints like characteristic or implicit constraints to get the highly believable results in simulation and optimization. Thus, how to define the related constraints from various design variables in an optimization problem is an important issue to discuss. The statements for this are presented in the next Chapter.

Fig- 3.3.2 Constraints selecting parameters list

CHAPTER 4

OPTIMIZATION OF THE REED VALVES

From the description in previous Chapters, it is found that the valve characteristics play the critical role for efficiencies and performance of the hermetic reciprocating compressors. So these characteristics can be used to produce some design problems for optimization. This Chapter will take detailed statement for results and discussions about the optimization problems.

4.1 **Optimization Problem for Suction-valve**

The paper [18] considers the affections of E.E.R and refrigerant mass flow rate for different suction-valve thickness. The reason for choosing suction-valve thickness as a variable is that there are actual sizes can carry out experiments for verification. So in the section the parameters that can be modified in practical manufacture to do experiments are initially selected as design variables in the optimization problem.

4.1.1 Formulation of Problem

The mathematic model of the suction-valve constructed in the simulated software applying the assumed method and deriving by Lagrangian approach. Chapter 2 has a simple introduction. The details could refer the Appendix [A] .

The physical characteristics in the suction-valve are considered in the problem. The reasons for selecting these design variables, constraint conditions and cost function are described below:

Design variables:

1. valve thickness (t_{valve})

   Due to the mathematic model, it is known that different valve-thickness will affect the cross-section and moment of inertia of the valve, more over to mass, stiffness. So if too stiff the valve plate causes over-compression and delay of closing. However the inessential fluctuation of the valve plate produced because of too flexible valve [7]. Besides, the valve thickness is considered as a design variable for the problem.

2. diameter of valve passage (d_s)

   The influence of effective force area and effective flow area of valve is upon the change for diameter of valve passage. However, the experimental coefficient β using for correcting pushing force (Appendix [A]) is also affected by diameter of valve passage. So it is also a design variable in the problem.

3. distance of valve passage (l_1)

   Let reviewing Fig- 2.3.1 again, the l_1 is defined for the parameter. It represents the position that the center of the diameter of passage located to and due to symmetry of valve, the length l_1 is considering only one direction.

Therefore, these three variables are highly nonlinear and have coupled influence to the pushing force, and respectively have effect for other calculations. So how to decide the sizes from their relations to promote the efficiency is the target of the optimization problem.

Constraints:

1. compression efficiency

   The compression efficiency changes with the effective force area of valve [19]. It is expected to arise from the increasing effective force area to get the better efficiency, and forms a constraint in the problem.

2. volume efficiency

   If the designer hopes to obtain high volume efficiency, he may limit the size and number of valves, however, may tend to lower area and decrease the compression efficiency [19]. As a general rule, high volume efficiency and high compression efficiency (low power requirement) do not go together. It must get a compromise between the two constraints.

3. capacity of refrigeration

   This result depends on the testing conditions of the import and export refrigerant. The experimental results show that the capacity of refrigeration approximates to 190 (kcal/hr) at the same initial conditions. It is expected to get the result more to promote cooling ability. So this result is also a considering constraint in the problem.

Cost function

There are many results can be the cost function. The E.E.R is a usual index for determining the whole efficiency of the reciprocating compressor. It defines capacity of refrigeration dividing motor input power. The motor stator reaction torque is generated by Motor-Speed-Torque curves of induction motors that show the average torque. The higher E.E.R generally represents the better performance. Due to reducing the complication of optimization, the single objective optimization is used in the problem and the cost function is maximization of E.E.R.

The completed definitions, formulation and setting for the design variables, constraints, cost function are listed in the Table- 4.1.1. The optimum results can be acquired after executing the optimization model stated in Chapter 3. The results and discussions show in the next sub-section.

4.1.2 Results and Discussions

Table-4.1.2 shows the results from the optimization problem. It is found that the suction-valve thickness decreases to lower bound, the diameter of passage and the distance of passage increase to the upper bound could get maximum E.E.R. The decreasing suction-valve thickness for better E.E.R is proved by experiments [18] if considering only this one variable. In the problem the thinner valve thickness still has better result. The other two design variables are proved when the design bounds are limited in the problem, the large diameter of the passage and distance of diameter arise the cost function E.E.R. The trends for those are reasonable because when increasing these, the pushing force could lift up the valve more for importing more refrigerant.

One constraint could not be satisfied in the problem, the compression efficiency. It illustrates that the needed solution could not be found in the design space for the problem. The problem runs 11 iterations and the related history for these iterations are showed in Fig- 4.1.1 to Fig- 4.1.4. It is known that the designs satisfied the constraints except for the compression efficiency and get the maximum E.E.R. Fig- 4.1.5 shows the constraints sensitivity to design variables in the problem. It can observe the thickness of suction valve influence the volumetric efficiency and capacity of refrigeration more. The distance of passage has greater sensitive effect to compression efficiency. The results are the consultation to manufacture and how to select changes to get best effects without increasing difficulties in practice is depend on experiences and cost to manufacture.

design variables	No.	name	unit	initial value	lower bound	upper bound
	1	valve thickness	m	0.0002	0.0001	0.0004
	2	diameter of valve passage	m	0.0056	0.00504	0.00616
	3	distance of valve passage	m	0.018	0.015	0.019
constraints	No.	name		initial value	comparison	condition
	1	volumetric eff.	%	65.3	≧	70
	2	compression eff.	%	68.49	≧	75
	3	capacity of refrigeration	kcal/hr	194.6221	≧	200
cost function	No.	name		initial value		requirement
	1	EER	kcal/hr · W	0.9395		max

Table- 4.1.1 Formulation of the suction-valve problem

design variables			constraints			cost function		
name	unit	results	name	unit	results	name	unit	results
valve thickness	m	0.0001	volumetric eff.	%	77.07	EER	kcal/hr · W	0.9906
diameter of valve passage	m	0.00616	compression eff.	%	72.13			
distance of valve passage	m	0.019	capacity of refrigeration	kcal/hr	230.6542			

Table- 4.1.2 Results of the suction valve optimization

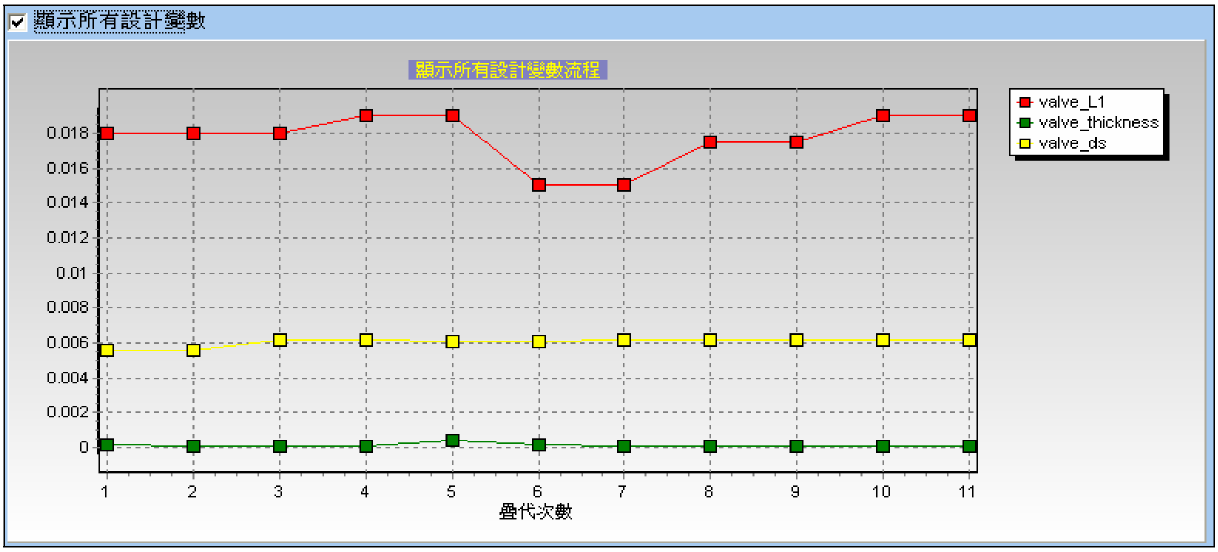


Fig- 4.1.1 History of design variables for the suction-valve problem

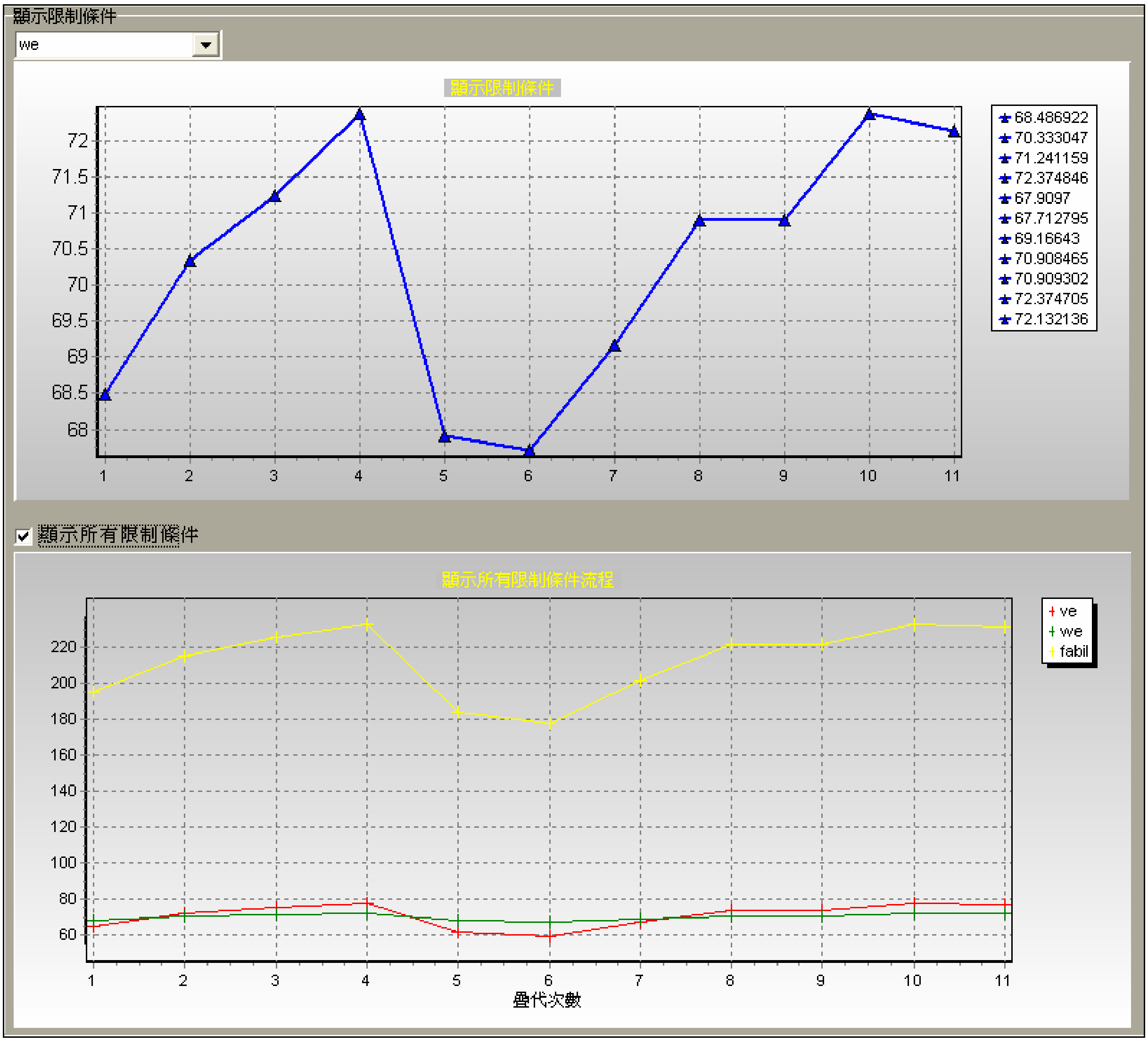


Fig- 4.1.2 History of constraints for the suction-valve problem

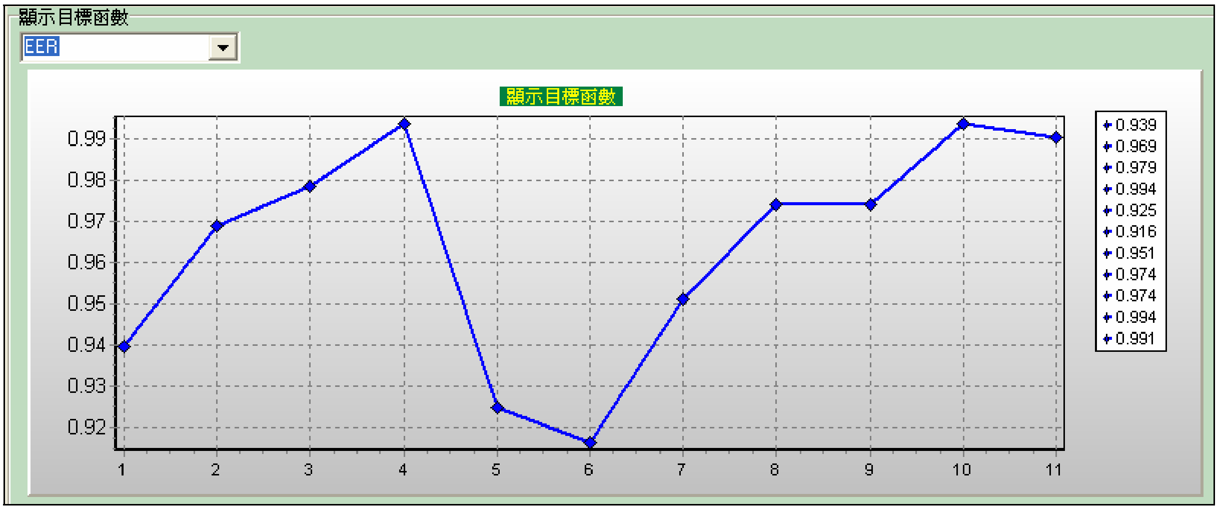


Fig- 4.1.3 History of cost function for the suction-valve problem

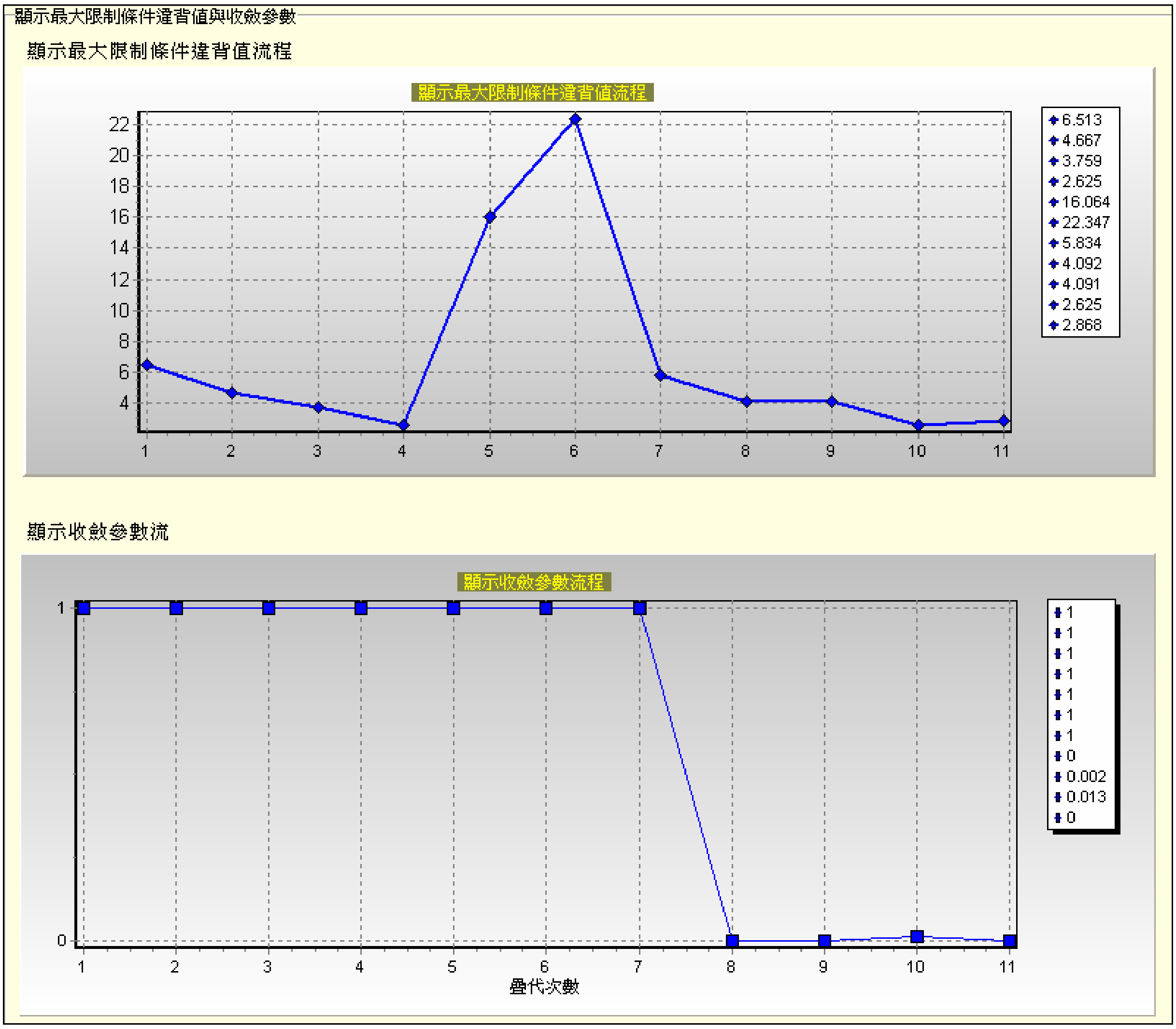
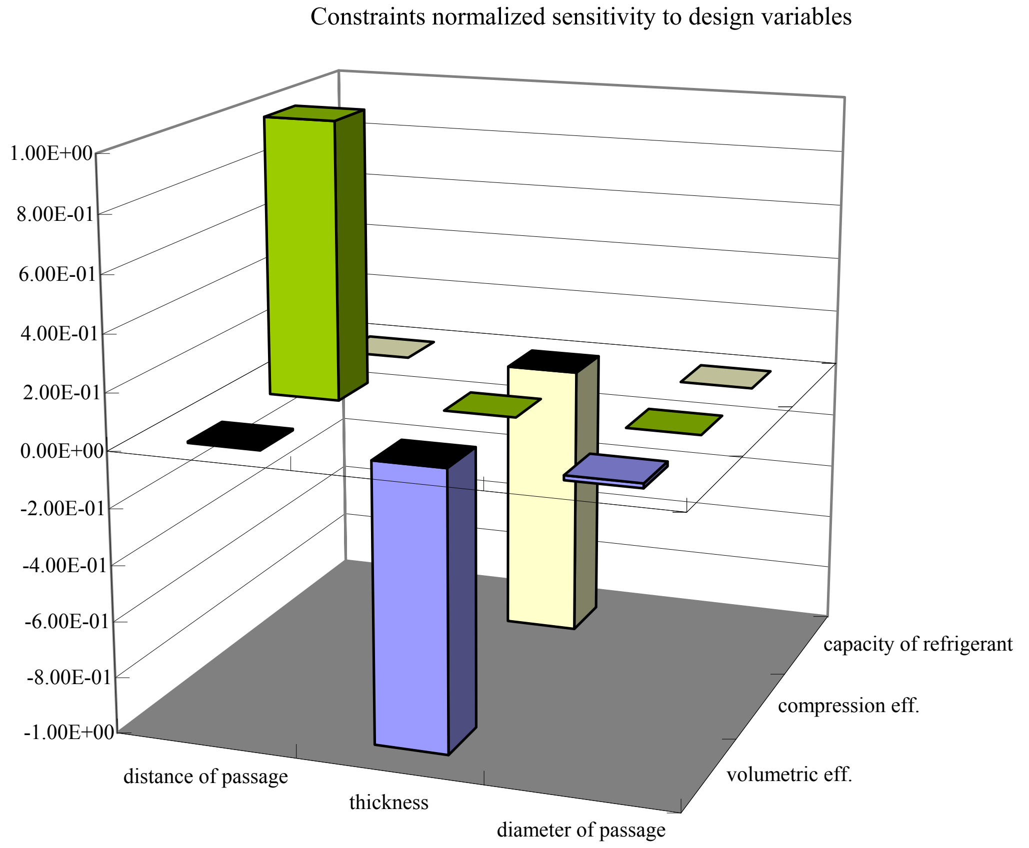


Fig- 4.1.4 History of constraint violation and convergence parameter



	distance of passage	thickness	diameter of passage
volumetric eff.	-5.26E-03	-1.00E+00	1.62E-02
compression eff.	1.00E+00	0.00E+00	0.00E+00
capacity of refrigerant	0.00E+00	-1.00E+00	0.00E+00

Fig- 4.1.5　Constraints normalized sensitivity to design variables

4.2 **Optimization Problem for Discharge-valve**

After the problem stated in 4.1, the characteristics of discharge-valve in the simulation software is another selection to improve the efficiency and performance. Because of higher difference between the cylinder and discharge chamber and shorter time for discharge step, how to design the suitable dimensions and sizes to get benefit is important.

4.2.1 Formulation of Problem

Design variables:

The three design variables, valve thickness, diameter of passage, distance of valve passage for the discharge-valve are using in the optimization problem. The bounds conditions are set by considering the practice in manufacture and consulting with ERL members.

Constraints and cost functions

The volume efficiency, compression efficiency, and capability of refrigeration are also considered as constraints in the problem. The E.E.R is also chosen as the cost function for maximization in the problem.

The definitions, formulation and setting for the design variables, constraints, cost function are listed in the Table- 4.2.1. The results of the optimization problem and discussions show in the next sub-section.

4.2.2 Results and Discussions

Table- 4.2.2 shows the results from the optimization problem. It is found that the discharge-valve thickness decreases and approximate to the lower bound, the diameter of the passage and distance of passage increase to the upper bound could get maximum E.E.R. In the problem the thinner valve thickness still has better result. The other two design variables are proved when the design bounds are limited in the problem, the large diameter of the passage and distance of passage arise the cost function E.E.R.

The E.E.R. seems can get better results comparing to the previous problem (1.0975 > 0.9906). However, volumetric efficiency and compression efficiency seems have different trends comparing to previous problem. The new design variables group in previous problem can inhale more refrigerant to get higher capacity of refrigeration (230.6542 kcal/hr·W), so the volumetric efficiency is higher (77.07%). However, the compression efficiency is just 72.13% and causing the lower E.E.R value (0.9903). Nevertheless, in this problem the capacity of refrigerant is just 220.46 kcal/hr·W, but can get higher E.E.R value (1.0975) due to its higher compression efficiency (80.58%). These conditions can also verified that a compromise between volumetric efficiency and compression efficiency.

All constraints could be satisfied in the problem. The problem also runs 6 iterations and the related history for these iterations are showed in Fig- 4.2.1 to Fig- 4.2.4. It is known that the designs after iteration 6 are feasible and designs before it had some violation of constraints.

The normalized sensitivity for 3 design variables to cost function shows in Fig- 4.2.5, found that the diameter of the passage is more sensitivity to E.E.R in this problem. It also can be a reference when deciding modification of dimension of the suction-valve by the designer. Fig- 4.2.6 shows the normalized constraints sensitivity to design variables. It is found that the diameter of discharge-valve is more sensitive to three constraints, the second is thickness of discharge-valve, the distance of distance of discharge-valve affects least to constraints in the problem. It could be a reference for inaccuracy of manufacture in practical.

design variables	No.	name	unit	initial value	lower bound	upper bound
	1	valve thickness	m	0.0002	0.0001	0.0003
	2	diameter of valve passage	m	0.0045	0.00405	0.00495
	3	distance of valve passage	m	0.018	0.015	0.02
constraints	No.	name		initial value	comparison	condition
	1	volumetric eff.	%	65.3	≧	70
	2	compression eff.	%	68.49	≧	75
	3	capacity of refrigeration	kcal/hr	194.6221	≧	200
cost function	No.	name		initial value		requirement
	1	EER	kcal/hr · W	0.9395		max

Table- 4.2.1 Formulation of the discharge-valve problem

design variables			constraints			cost fucntion		
name	unit	results	name	unit	results	name	unit	results
valve thickness	m	0.0001	volumetric eff.	%	73.66	EER	kcal/hr · W	1.0975
diameter of valve passage	m	0.00495	compression eff.	%	80.58			
distance of valve passage	m	0.02	capacity of refrigeration	kcal/hr	220.46			

Table- 4.2.2 Results of the discharge valve optimization

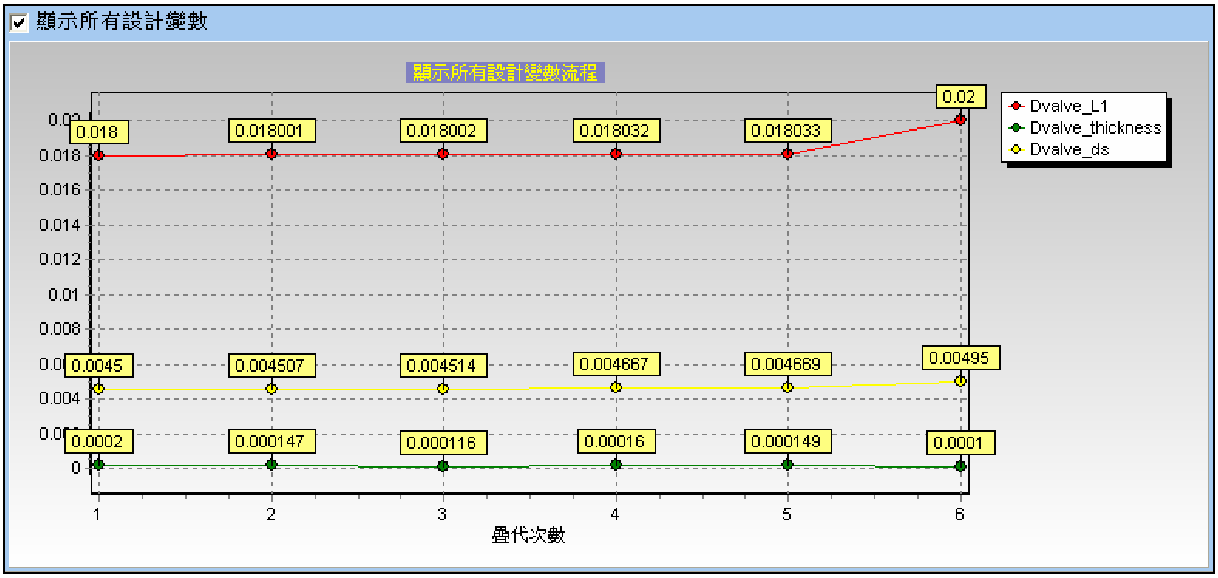


Fig- 4.2.1　History of design variables for the discharge-valve problem

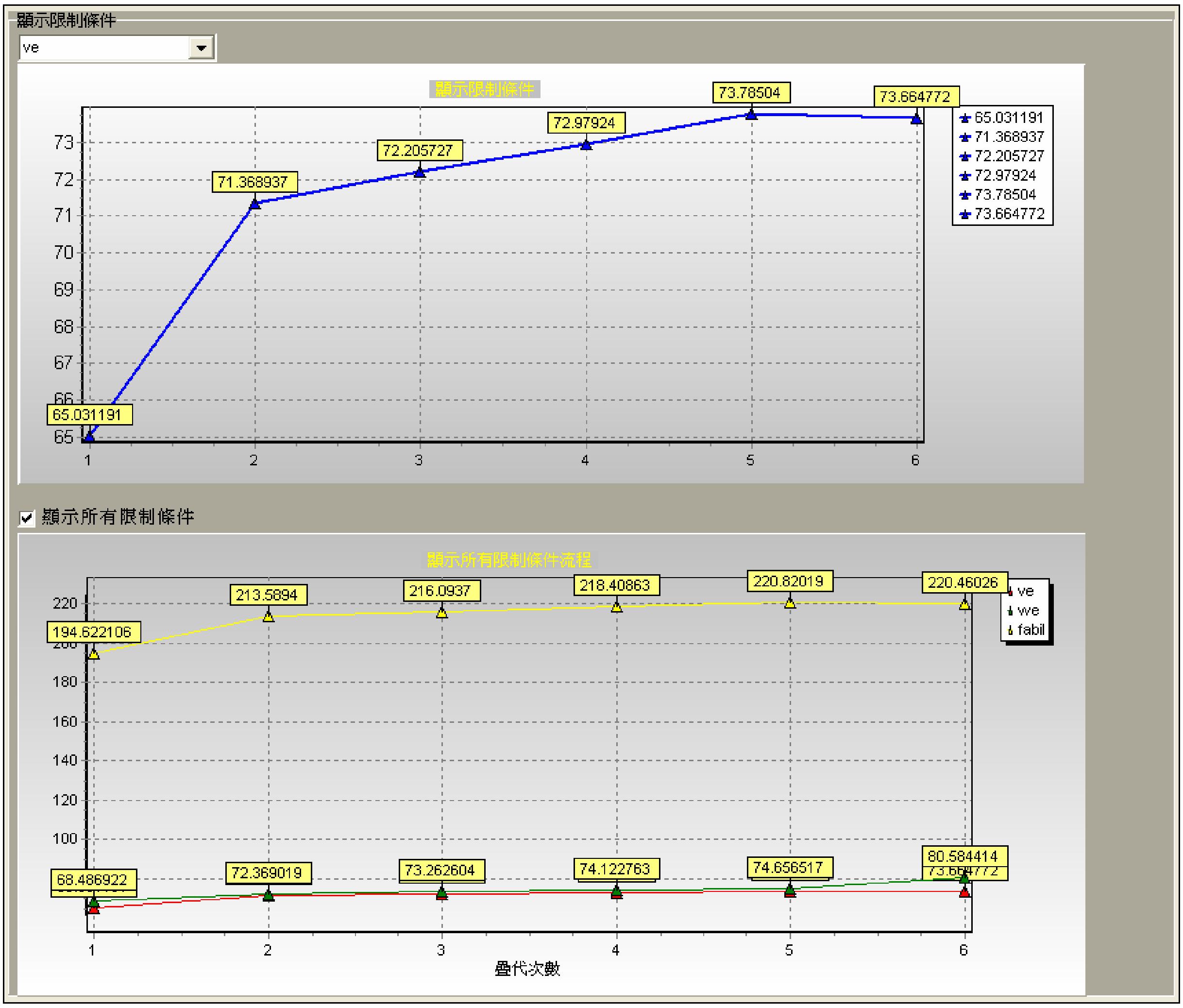


Fig- 4.2.2　History of constraints for the discharge-valve problem

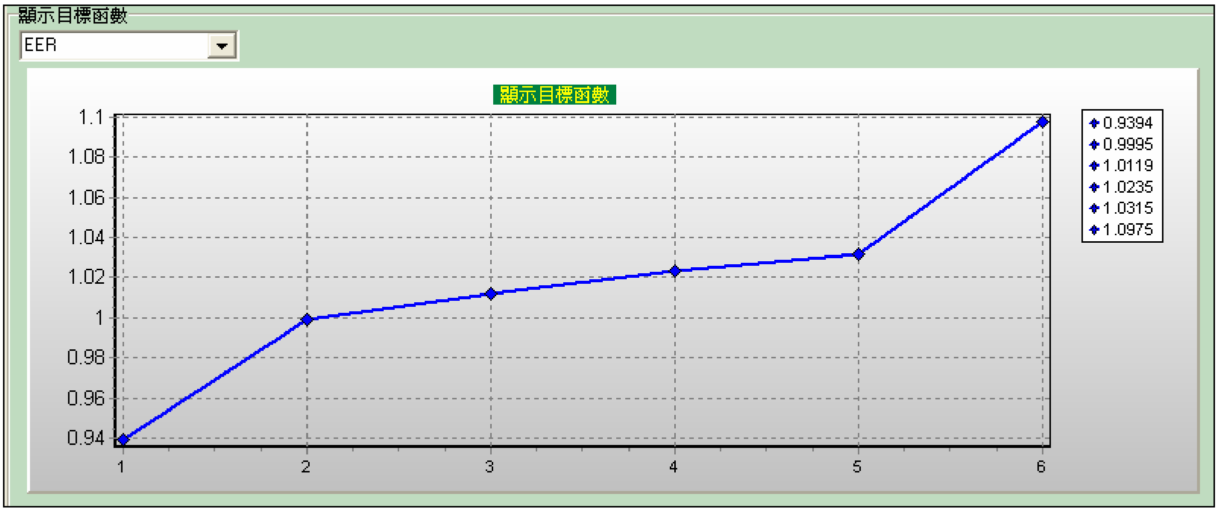


Fig- 4.2.3 History of cost function for the discharge-valve problem

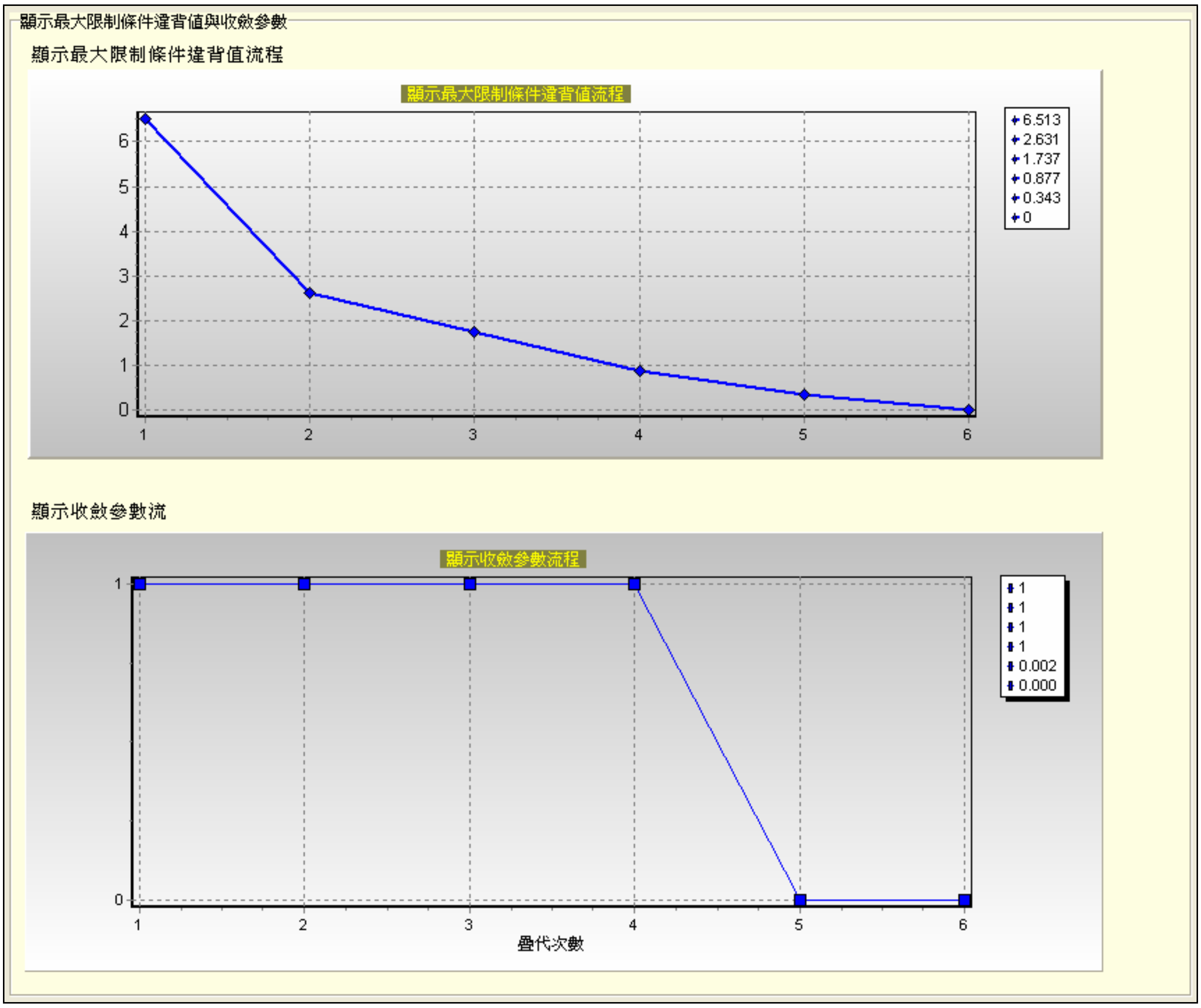


Fig- 4.2.4 History of constraint violation and convergence parameter

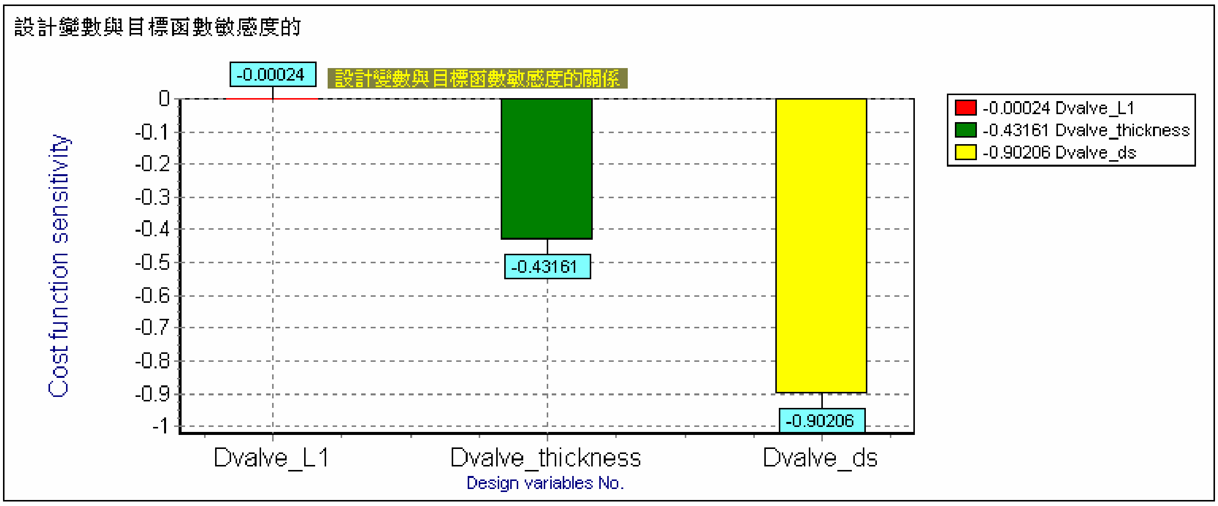
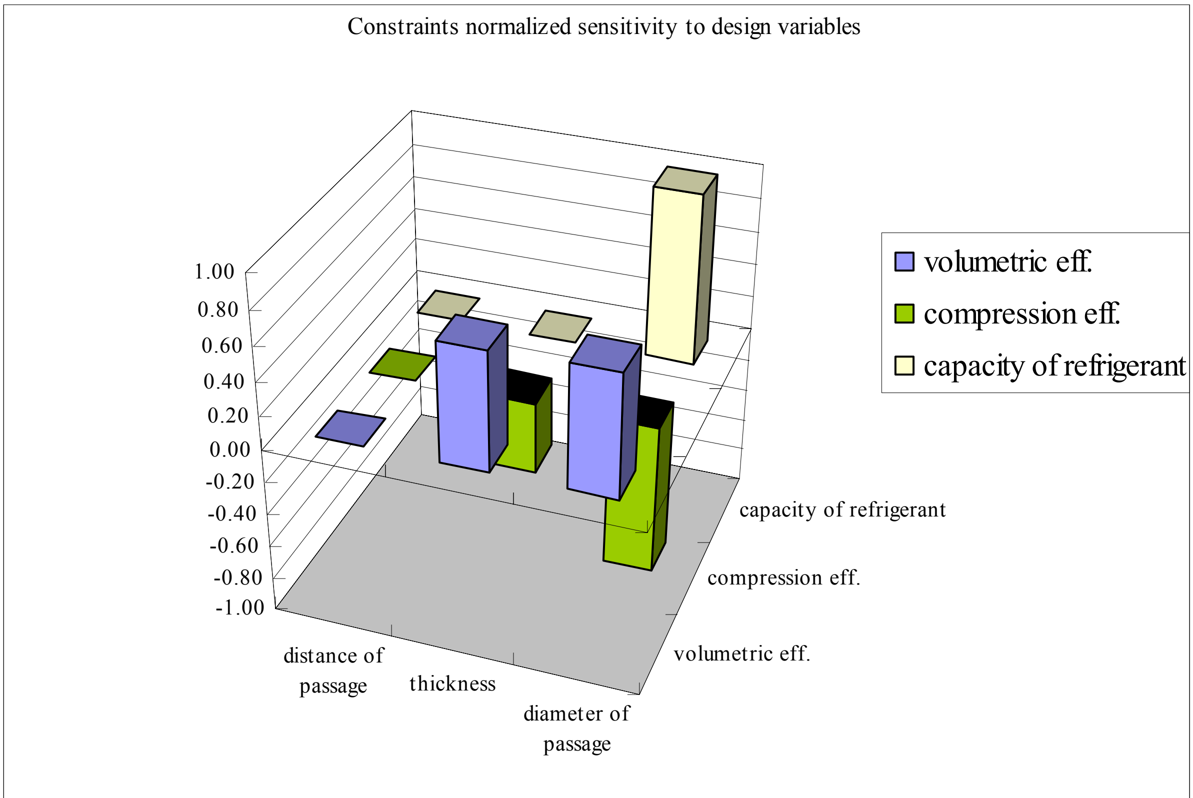


Fig- 4.2.5 Cost function normalized sensitivity to design variables



	distance of passage	thickness	diameter of passage
volumetric eff.	1.46E-03	6.95E-01	7.19E-01
compression eff.	0.00E+00	-4.26E-01	-9.05E-01
capacity of refrigerant	0.00E+00	0.00E+00	1.00E+00

Fig- 4.2.6 Constraints normalized sensitivity to design variables

4.3 **Synthetic Optimization of Suction and Discharge-valves**

After discussing the suction-valve and discharge-valve optimizations respectively, It is expected that considering both two valves at the same time, what's the relations and meaning of the results. The integrated optimization for these two components is developed in this section.

4.3.1 Formulation of Synthetic Optimization Problem

Design variables:

The design variables defined in the problem include the six parameters that discussing in the previous two optimization problems. In the mean time, all their bounds are identical using in it.

Constraints and Cost function:

In order to comparing the previous results in section 4.1 and section 4.2, the constraint conditions and cost function are use the same setting from the previous two optimization problems.

The detail definitions, formulations and related setting conditions of this problem are listed in the Table- 4.3.1.

4.3.2 Results and Discussions

Table- 4.3.2 shows the optimum results. The distances of suction-valve and discharge-valve passage and the diameter of suction-valve passage are unchanged and the thickness of suction-valve and discharge-valve decrease and the diameter of discharge-valve passage increase slightly can meet the convergent requirement, the E.E.R is 1.0165. It is deduced that this design variables group is a local maximum result for the cost function. Besides, all the constraint conditions are satisfied in the problem. So the design variables group is a feasible solution, Fig- 4.3.1 to Fig- 4.3.4 show some other history of the related arguments in the problem.

However, it may exists other local maximum results for the problem. Here, trying to take others as initial design variables group for the problem. After testing several times the best results shows in Table- 4.3.3. It is found that the capacity of refrigeration increase enormously and the volumetric efficiency and compression efficiency are also rise more. Besides, the E.E.R value promotes to 1.2448 and the design variables group could be a good reference if the design results can meet the manufacturing limit.

design variables	No.	name	unit	initial value	lower bound	upper bound
	1	suction-valve thickness	m	0.0002	0.0001	0.0004
	2	diameter of suction-valve passage	m	0.0056	0.00504	0.00616
	3	distance of suction-valve passage	m	0.018	0.015	0.19
	4	discharge-valve thickness	m	0.0002	0.0001	0.0003
	5	diameter of discharge-valve passage	m	0.0045	0.00405	0.00495
	6	distance of discharge-valve passage	m	0.018	0.015	0.02
constraints	No.	name		initial value	comparison	condition
	1	volumetric eff.	%	65.3	≧	70
	2	compression eff.	%	68.49	≧	75
	3	capacity of refrigeration	kcal/hr	194.6221	≧	200
cost fucntion	No.	name		initial value		requirement
	1	EER	kcal/hr · W	0.9395		max

Table- 4.3.1 Formulation of synthetic problem

design variables			constraints			cost fucntion		
name	unit	results	name	unit	results	name	unit	results
suction-valve thickness	m	0.00025	volumetric eff.	%	72.47	EER	kcal/hr · W	1.0165
diameter of suction-valve passage	m	0.0056	compression eff.	%	73.60			
distance of suction-valve passage	m	0.018	capacity of refrigeration	kcal/hr	216.8764			
discharge-valve thickness	m	0.0002						
diameter of discharge-valve passage	m	0.004553						
distance of discharge-valve passage	m	0.01801						

Table- 4.3.2 Results of the synthetic optimization

design variables			constraints			cost fucntion		
name	unit	results	name	unit	results	name	unit	results
suction-valve thickness	m	0.0001	volumetric eff.	%	94.02	EER	kcal/hr · W	1.2448
diameter of suction-valve passage	m	0.00616	compression eff.	%	90.55			
distance of suction-valve passage	m	0.019	capacity of refrigeration	kcal/hr	281.37			
discharge-valve thickness	m	0.0001						
diameter of discharge-valve passage	m	0.00495						
distance of discharge-valve passage	m	0.02						

Table- 4.3.3　Results of the best synthetic optimization

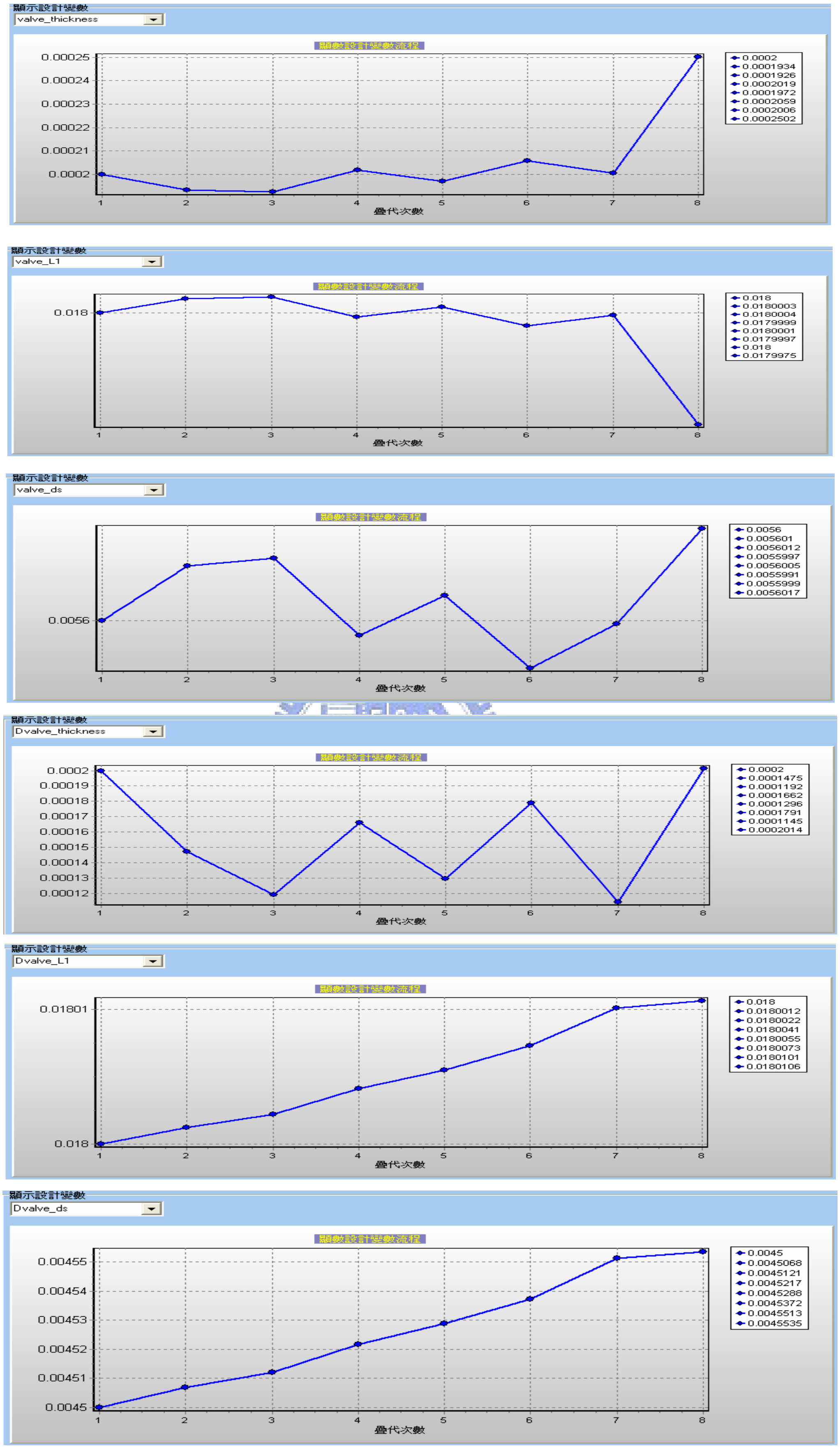
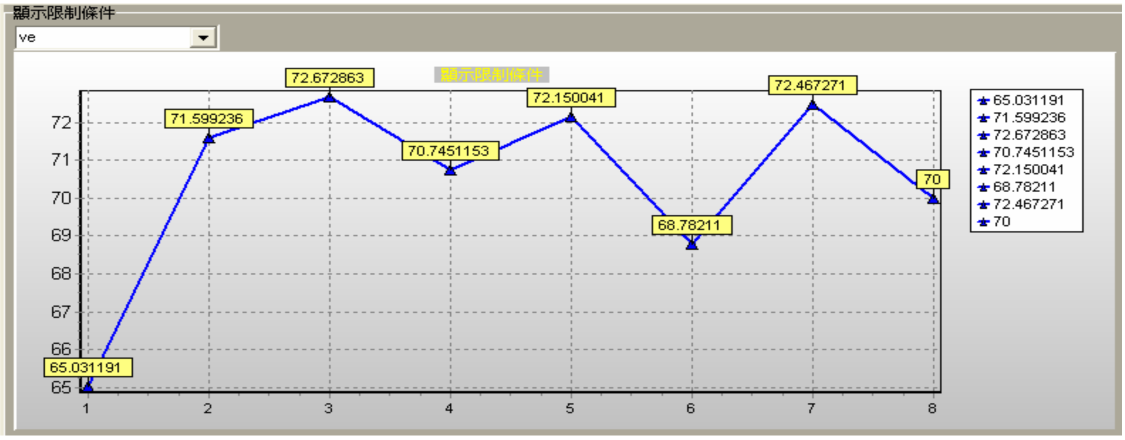
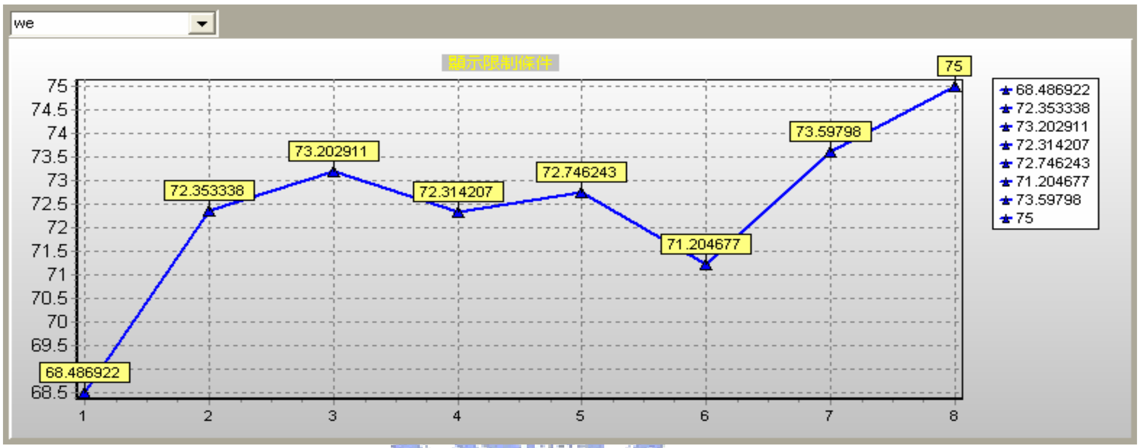


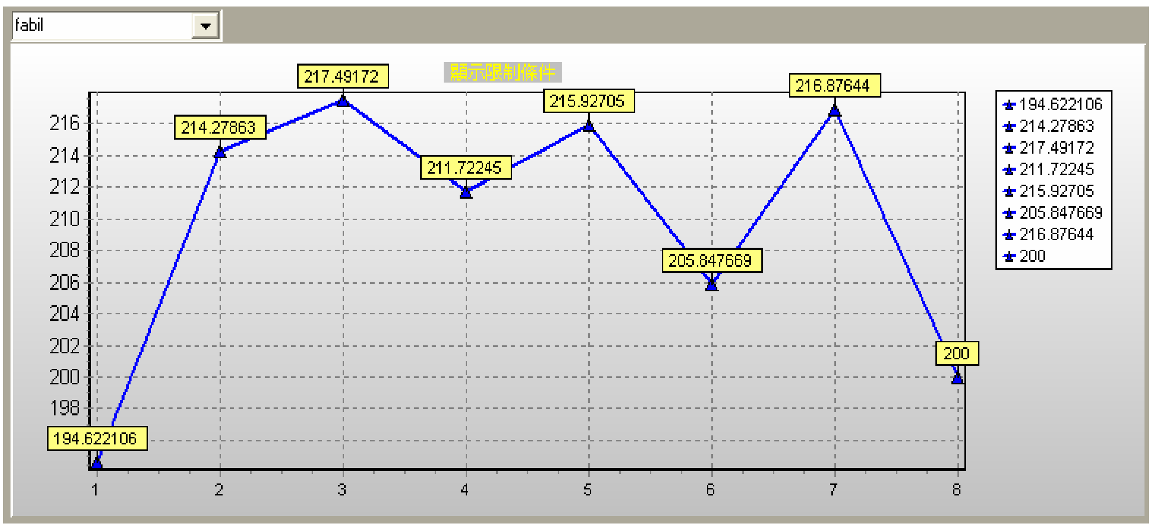
Fig- 4.3.1 History of design variables for the synthetic optimization problem



volumetric efficiency



compression efficiency



capacity of refrigeration

Fig- 4.3.2 History of constraints for synthetic optimization problem

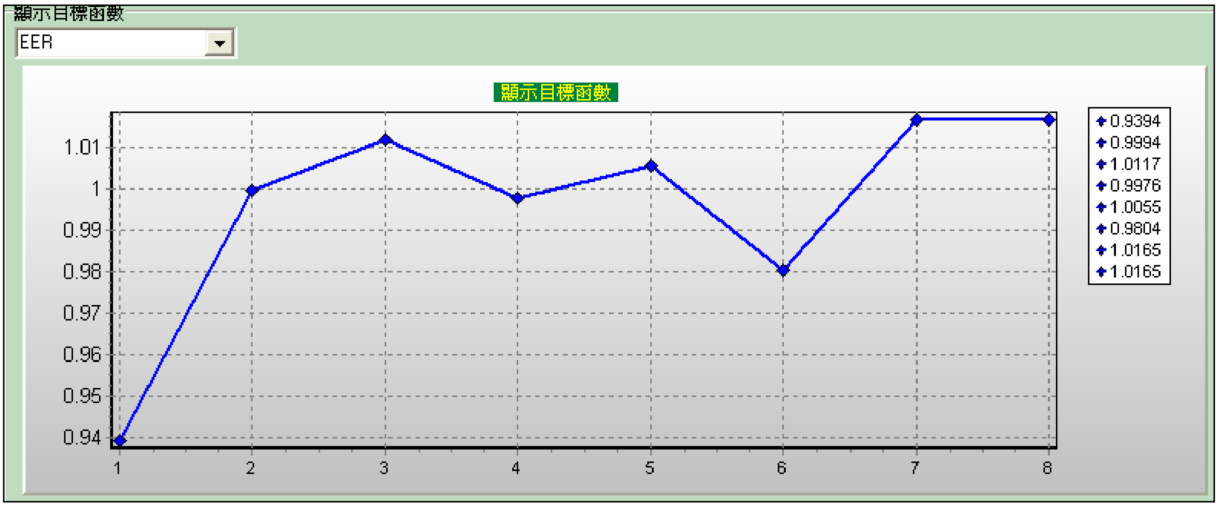


Fig- 4.3.3 History of cost function for the synthetic optimization problem

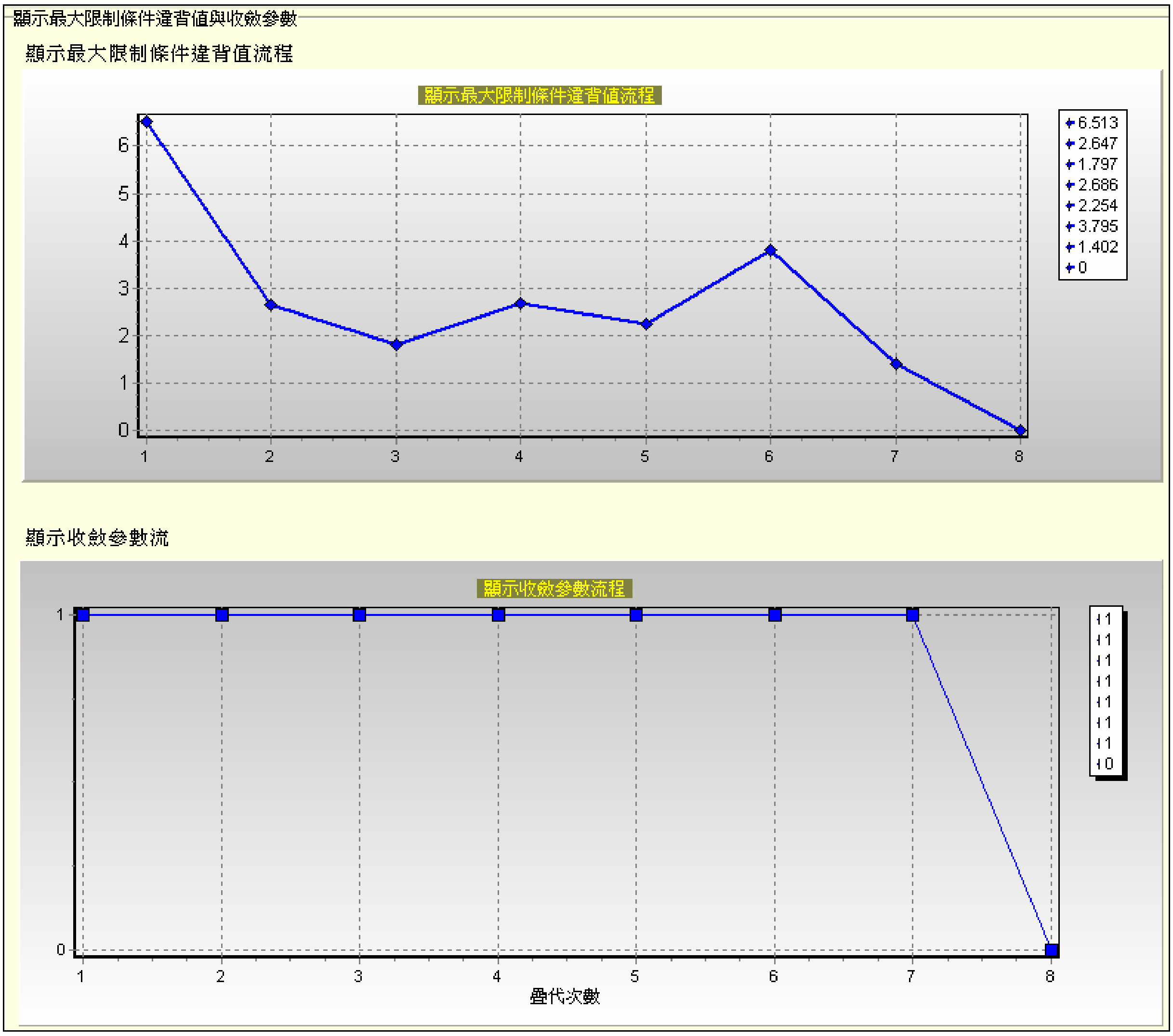


Fig- 4.3.4 History of constraint violation and convergence parameter

CHAPTER 5

CONCLUSIONS AND FURTHER WORKS

5.1 **Conclusions**

Debugging and improvement of the comprehensive performance simulation software, which the operating condition is below 500 W, developed before is carried out in the study. It has conformity with the experiments provided by the Energy and Resources Laboratories (ERL) of Industrial Technology Research Institute (ITRI).

The optimization module is constructed and developed in the simulation software. It is constructed by using Borland C++ Builder, and based on objective-oriented programming (OOP) concept of programming design with window-based GUI interface. It helps to reduce times for doing experiments of the actual reciprocating compressor are complicated and time-consuming. The conclusions of the thesis can be summarized as bellow:

1. The simulation software of the hermetic reciprocating compressor combines thermal dynamic, mechanism, valve dynamic, bearing analysis are improved, and recomposed for developing design optimization.
2. The optimization module constructed here includes two sub-modules, user interface and optimization solver. The user interface provides very friendly and easily operating interactive interface to help end user formulate and set up different optimization problems. When the problems change, the user can easily transfer it including cost function, design variables, and constraint conditions by the user interface and proceed to analyze and develop design optimization of the reciprocating compressor. Besides, it also contains post-process functions to represent the history or relation during the optimization process.

3. The optimization solver is applied in the module. The solver automatically provides suitable algorithmic parameters to match up its methods, help to lower complication for using it by the user. It also provides functions to deal with continuous, discrete and multi-objective optimization problems.
4. By the optimization module, end-user can handle a great quantity of optimization problems regardless of numbers or types of design variables, constraints and cost function.
5. The optimal results are provided in the study for discussing affections of suction-valve and discharge-valve to energy efficiency ration, E.E.R. It could be a consultation and provides a tendency to improve efficiency when proceeding actual experiments.

5.2 **Further Works**

By integrating the simulation software greatly to predict the behavior and performances of the hermetic reciprocating compressor, and linking the optimization module to the software, the designer can develop and study it more conveniently and quickly to save the manpower and cost. There are several considerations and issues should study and discuss in the future as follows:

(1). Verification of the optimization results with experiments:

The optimization results in the study should be verified by experiments to prove the feasibility and reliability.

(2). Continuing developing design optimization:

The study just discuss affections of some valve properties to efficiency of the reciprocating compressor in the software, the other design considerations, like material properties of valve, cylinder characteristics and so on, possibly have influence for simulation, should study from now on. Besides, the discrete and multi-objective optimization problems also can be investigated in later studies.

(3). Different refrigerant type

The software uses the HFC-134a as the refrigerant. With the trend of using natural refrigerants in the compressors, whether the simulation software can introduce other refrigerant such as CO_2, or introduce one without causing greenhouse effect such as R-600a, forms an issue in the further research.

(4). Debugging the software

The software uses many experimental formulas. Once the dimension or size changes, the formulas are suitable for use or not are very important, must correct and prove its applicability.

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Appendix [A]

Symbols of the reed valve analysis:

$A(x)$	Cross-section area of the reed valve
A_e	Effective force area
C_r	Reflecting coefficient of the valve
$C_1(x)$	The equation of the first valve line segment
$C_2(x)$	The equation of the secondary valve line segment
E	Young's modulus of the valve
f(t)	Generalized force
l	Length of the valve
l_1	Distance of the applied force related to the coordinate
I(x)	moment of inertia of the valve cross-sectional
r_s	Radius of the suction orifice
p_s	Pressure in the suction chamber
p	Pressure in the cylinder
q(t)	Time-dependent generalized coordinate
t_{valve}	Thickness of the valve
$u(x, t)$	Z-direction displacement of the valve
w_n	The natural frequency
$\phi(x)$	Shape function of the cantilever beam
β	Pushing force coefficient of the valve
ξ	Damping ratio
η	The nonlinear term of the valve governing equation
ρ	Density of the valve

Mathematical Formulation of Reed Valve [1]

The Lagrangian approach is used to derive the governing equation of motion for a cantilever type valve. The Kinetic energy (T), potential energy (U), and the work (W), of external load could be obtained with the application of the Lagrangian's equation.

The assumed-modes method [2] is used to simplify the governing equations of the reed valve vibration. Fig-A.1 shows the valve displacement relation. Since the natural boundary conditions will be automatically accounted in the kinetic and potential energy, its conditions are not particularly considered here. The valve displacement, $u(\mathrm{x,t})$, are as following:

$$u(\mathrm{x,t}) = \phi(\mathrm{x})\,\mathrm{q(t)} \tag{A-1}$$

Where,

$\phi(\mathrm{x})$: Valve admissible function (or shape function) that can be obtained form the free vibration analysis. The first mode consideration is sufficient for the function. The admissible function [2] satisfied the boundary conditions depicted as:

$$\phi(x) = x \;\; -4lx^3 + 6l^2x^2 \tag{A-2}$$

q(t) : The time-dependent generalized coordinate

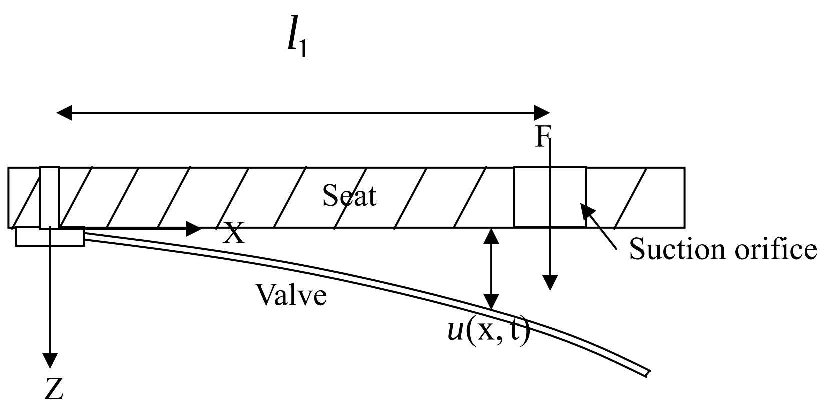

Fig-A.1 Suction valve displacement, $u(\mathrm{x,t})$

By the valve displacement, $u(\mathrm{x,t})$, the kinetic energy, potential energy, and the work of the extern load, are expressed as following:

$$T = \int_0^l \rho A(x)\ u'^2 dx \tag{A-3}$$

$$U = \frac{1}{2}\int_0^l EI(x)\ u''^2 dx + \frac{1}{8}\int_0^l EA(x)\ u'^4 dx \tag{A-4}$$

$$W = \int_0^l \beta A_e (Ps - P) u \delta (x - l_1) dx \tag{A-5}$$

Then, applying the Lagrange's equation in [2], $\frac{d}{dt}(\frac{\partial T}{\partial \dot{q}}) - \frac{\partial T}{\partial q} + \frac{\partial U}{\partial q} = W$, substituting the expression for the work of the extern load, the Kinetic energy, and potential energy into the energy expression, and adding the damping effect yields the governing equation of the valve vibration below.

$$m\ddot{q} + kq + hq^3 = F \tag{A-6}$$

Where, $$m = \int_0^l \rho A(x)\varphi^2 dx \tag{A-7}$$

$$k = \int_0^l EI(x)\varphi''^2 dx \tag{A-9}$$

$$h = \frac{1}{2}\int_0^l EA(x)\varphi''^4 dx \tag{A-9}$$

$$F = \pi\beta\phi(l_1)(p_s - p)\{r_s + [\phi(l_1)]^2 q^2\} \tag{A-10}$$

The pushing force coefficient, β [12][13], shown in Fig-A.2 is obtained from experiment and is related with the valve displacement, $u(x,t)$. Now, a damping effect, ξ is added to the equation (A-6), it becomes:

$$\ddot{q} + 2\xi w_n \dot{q} + w_n{}^2 q + \eta q^3 = F^{\#} \tag{A-11}$$

Where, $$w_n = \sqrt{\frac{k}{m}} \tag{A-12}$$

$$\eta = \frac{h}{m} \tag{A-13}$$

$$F^{\#} = \frac{F}{m} \tag{A-14}$$

The valve motion equation above is only suitable before the valve impact the seat. The valve velocity while vibration is affected by the collision and the velocity after impacting the seat can be simply obtained by multiplying a reflecting factor, Cr. The seat is considered as a rigid body and the stiffness is much higher than the valve. Therefore, the fully valve motion equation can be derived by the equation (A-11) and the following equation.

$$\dot{q}_{after} = -C_r \dot{q} \qquad \text{(A-15)}$$

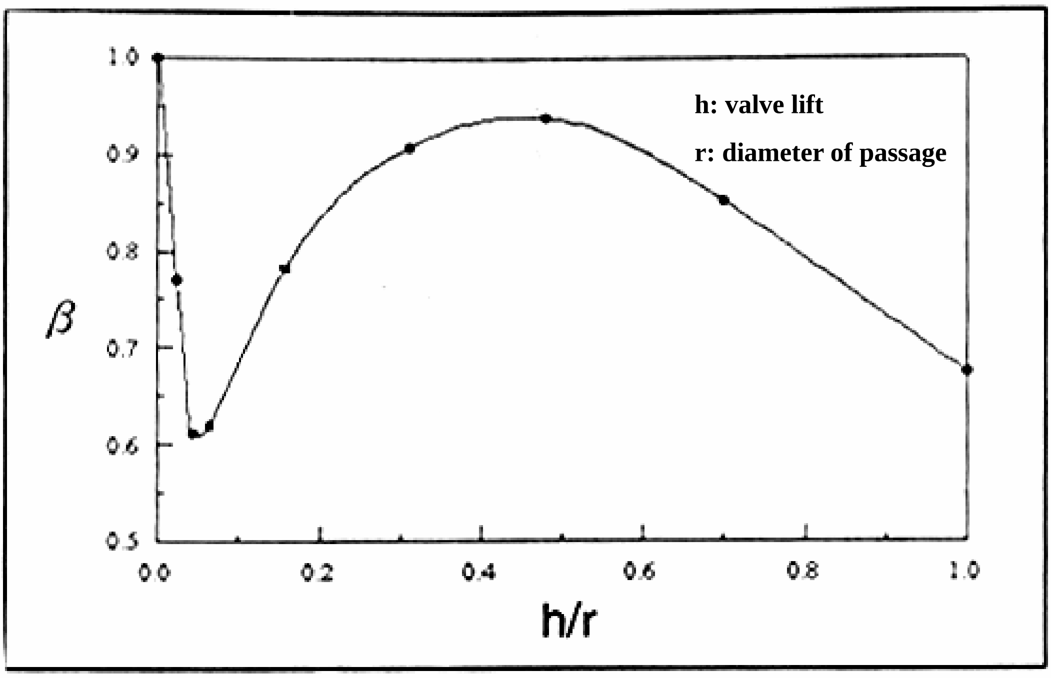

Fig-A.2 pushing force coefficient β

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