Design and development of the u-vane compressor

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**DESIGN AND DEVELOPMENT OF THE U-VANE
COMPRESSOR**

LIM YEU DE

SCHOOL OF MECHANICAL AND AEROSPACE ENGINEERING

2020

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SCHOOL OF MECHANICAL AND AEROSPACE ENGINEERING

A thesis submitted to the Nanyang Technological University Singapore
in partial fulfilment of the requirements for the degree of
Doctor of Philosophy

2020

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research, is free of plagiarised materials, and has not been submitted for a higher degree to any other University or Institution.

31st July 2019

Date

Lim Yeu De

Supervisor Declaration Statement

I have reviewed the contents and presentation style of this thesis and declare it is free of plagiarism and of sufficient grammatical clarity to be examined. To the best of my knowledge, the research and writing are those of the candidate except as acknowledged in the Author Attribution Statement. I confirm that the investigations were conducted in accord with the ethics policies and integrity standards of Nanyang Technological University and that the research data are presented honestly and without prejudice.

3 1 JUL 2019

Date

Professor Ooi Kim Tiow

Authorship Attribution Statement

This thesis contains material from 1 paper published in the following peer-reviewed journal(s) / from papers accepted at conferences in which I am listed as an author.

Appendix C.1 is published as Y. D. Lim, W. C. Poh, and K. T. Ooi, "Leakage Simulation of A Lubricant-free Rotary Type Swing Compressor Rotor Endface Clearance," in *International Compressor Engineering Conference Proceedings*, Purdue University, 2018.

The contributions of the co-authors are as follows:

- I provided the initial direction of the project.
- I prepared the manuscript drafts. The manuscript was revised by W.C. Poh and Professor Ooi.
- W.C. Poh and I performed the simulation work and analyzed the results.

31 July 2019
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Abstract

This thesis presents the developmental work of a new positive displacement rotary compressor introduced by the author, namely U-Vane compressor (UVC), credited to the “U” like-shaped new vane bush component. The compressor is designed with the motivation of eliminating the vane tip friction of the rolling piston compressor and simultaneously reducing the unnecessarily large size of the cylinder and housing volume in the swing compressor. In swing compressor, the vane is rigidly attached to the rotor, which eliminates the vane tip friction existing in the rolling piston compressor. However, the vane must be sufficiently long to prevent the possible operation jam and results in unnecessarily large size of the cylinder and hence increasing the housing volume in order to house the long vane. Thus, the newly introduced vane bush in UVC is made with two extended shoulders so that the vane can be made shorter as compared to that in swing compressor and subsequently reduces the size of the cylinder and housing volumes. With reference to Appendix A, it is anticipated that the size of the cylinder and housing volumes of UVC will be 30 % smaller than that of the swing compressor.

Mathematical models have been formulated to better understand the operational and performance characteristics of the UVC. The models take into consideration of energy, flow and heat transfer, mass conservation and momentum aspects of the compressor. Additionally, the mechanical losses, the dynamic aspect of the journal bearing, the internal leakage flows were also accounted for.

The mathematical models were converted into a computational code using ForTran programming language and the 4th order Runge-Kutta method was used to integrate the simultaneous single order differential equations numerically. The working fluid was R-1234yf, the evaporating and condensing temperatures are 7.2 °C and 54.4 °C, respectively. The result shows that for a compressor with volumetric capacity of 32.5 cm³/rev, the mechanical efficiency is 75.71 % and the volumetric efficiency is 96.28 % when operating at 3000 rev/min. It is found that the largest frictional loss occurs at the vane sides, which contributes 47.8 %

to the total frictional loss. The results also show that the leakages through radial clearance and rotor end-face clearance account for 49.9 % and 37.0 % of the total internal leakage, respectively.

In order to further understand the operational behaviour of the compressor, the effects of several design and operational parameters on the compressor performance were also investigated. It is observed that lengthening the vane from 8.0 mm to 20.0 mm increases the mechanical efficiency by 8.7 %. By increasing the vane bush shoulder length from 4.0 mm to 12.0 mm, the mechanical efficiency improved by 3.3 %, mainly due to the reduction in the vane side friction. A Similar magnitude of improvement in the mechanical efficiency can be obtained by thickening the rotor wall from 9.0 mm to 17.0 mm. However, increasing the operating speed from 1000 rev/min to 5000 rev/min reduces the mechanical efficiency by 10.7 %, due to higher frictional loss.

For verification of the new design, a prototype machine with volumetric displacement of 33.5 cm³/rev has been designed, fabricated, instrumented and measured experimentally with air as the working fluid. The prototype was tested from 1200 rev/min to 2100 rev/min for discharge pressure of up to 5.27 bar (abs.), i.e. maximum pressure ratio of about 5.27. The maximum discrepancy between the predicted and measured power inputs was found to be 6.8 %. The measured mass flow rates are observed to be higher than the prediction when running at 1800 rev/min and 2100 rev/min and when the discharge pressure is below 5.0 bar, beyond which, the trend reverses. In addition, the volumetric efficiency of the prototype was measured to be from 43.4 % to 92.3 %. The variation of the instantaneous pressure in the working chamber was also measured, and the maximum and average mean discrepancies between the predicted and measured results are 8.3 % and 4.1 %, respectively.

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Table of Contents

Abstract	iv
Acknowledgement	vi
Table of Contents	viii
List of Figures	xiii
List of Tables	xxii
Nomenclature	xxiii
Subscript	xxviii
Chapter 1 Introduction.....	1
1.1 Background and Motivation	1
1.2 Objective and Scope	2
1.3 Purpose of the thesis	2
1.4 Organisation of the thesis.....	2
Chapter 2 Literature Review.....	4
2.1 Overview of Refrigeration and Energy Consumptions.....	4
2.2 Compressor Mechanisms and Designs	7
2.2.1 Sliding Vane Compressor	7
2.2.2 Rolling Piston Compressor	9
2.2.3 Screw Compressor	10
2.2.4 Scroll Compressor.....	12
2.2.5 Swing Compressor	14
2.2.6 Revolving Vane Compressor	15
2.3 Theoretical Model and Simulation Studies.....	18
2.3.1 Thermodynamic Model.....	18
2.3.2 Valve Dynamic Model.....	19
2.3.3 Leakage Model.....	21

2.3.4	Heat Transfer Model	23
2.3.5	Other Relevant Literatures	24
2.4	Concluding Remarks	25
Chapter 3	Introduction to U-Vane Compressor	27
3.1	Conceptual Design of U-Vane Compressor	27
3.2	The Main Components and the Assembly	33
3.3	The Working Principle	37
3.4	Comparison between Swing Compressor and UVC	38
3.5	Concluding Remarks	39
Chapter 4	Mathematical Model I: Geometrics, Kinematics Thermodynamics and Valve Reed Dynamics	41
4.1	Assumptions of the Mathematical Model	41
4.2	Geometrical Model	42
4.3	Kinematic Model	46
4.3.1	Sliding Velocity and Acceleration of Vane	46
4.3.2	Kinematics of Rotor	47
4.3.3	Kinematics of Vane Bush	47
4.4	Thermodynamic Model	48
4.5	Suction and Discharge Flows Model	51
4.5.1	Orifice Flow Areas	53
4.6	Valve Reed Dynamics	56
4.7	Concluding Remarks	61
Chapter 5	Mathematical Model II: Dynamics, Internal Leakage, Instantaneous Power and Efficiencies	62
5.1	Dynamic Model	62
5.1.1	Force Analysis of Vane Bush	62

5.1.2	Force Analysis of Rotor	66
5.1.3	Moment of Inertia of Vane Bush and Rotor	68
5.2	Mechanical Losses	71
5.2.1	Frictional Loss between Vane Side and Vane Slot.....	71
5.2.2	Frictional Loss between Vane Bush Side and Cylinder Wall.....	72
5.2.3	Vane Bush End-face Frictional Loss	72
5.2.4	Rotor End-face Frictional Loss	74
5.2.5	Eccentric End-face Frictional Loss	75
5.2.6	Frictional Loss between Eccentric and Inner Rotor Surface	76
5.3	Internal Leakage Model	76
5.3.1	Leakage through Radial Clearance	77
5.3.2	Leakage through Vane End-face Clearance.....	81
5.4	Power and Efficiencies	83
5.5	Concluding Remarks.....	87
Chapter 6	Theoretical Simulation Results and Study	88
6.1	Simulation Setup and Operational Parameters	88
6.2	Simulation Results and Discussions	92
6.3	Parametric Study	102
6.3.1	Effect of Operating Speed.....	102
6.3.2	Effect of Discharge Pressure.....	103
6.3.3	Effect of Vane Length.....	105
6.3.4	Effect of Vane Bush Shoulder Length.....	106
6.3.5	Effect of Rotor Wall Thickness	107
6.3.6	Effect of Rotor-to-Cylinder Radius Ratio.....	108
6.3.7	Effect of Slenderness Ratio of Journal Bearing.....	110

6.4	Comparison of the Performance between UVC and Existing Rotary Compressors.....	111
6.5	Concluding Remarks.....	113
Chapter 7	Design of Prototype	115
7.1	Design Specifications of Prototype.....	115
7.2	Design of Prototype	116
7.2.1	Shaft and Eccentric	117
7.2.2	Rotor	123
7.2.3	Vane Bush.....	127
7.2.4	Cylinder Assembly.....	131
7.2.5	Housing Assembly	136
7.2.6	Counterweight.....	142
7.2.7	Assembly of Prototype.....	144
7.3	Design of Lubrication System	146
7.3.1	Lubrication Network Model	148
7.4	UVC Prototype.....	151
7.5	Concluding Remarks.....	159
Chapter 8	Experimental Study.....	160
8.1	Experimental Setup.....	160
8.2	Experiment Result and Discussion	164
8.2.1	Experimental Procedure.....	164
8.2.2	Considerations for Validation of Mathematical Model	165
8.2.3	Comparison between Predicted and Measured Results	169
8.2.4	Post-Experiment Prototype Components Inspection	182
8.3	Concluding Remarks.....	185
Chapter 9	Conclusion and Recommendations for Future Work.....	186

9.1	Conclusion	186
9.1.1	Compressor Design.....	186
9.1.2	Mathematical Modelling.....	187
9.1.3	Experimental Study.....	188
9.2	Recommendation for Future Work	189
9.3	Final Remarks	190
	References.....	191
Appendix A	Comparison of Cylinder and Housing Volumes between Swing Compressor and U-Vane Compressor	I
Appendix B	Derivation of Angular acceleration	IV
Appendix C	Rotor End-face Clearance Leakage and Heat Transfer Model	VI
Appendix D	Algorithm Flow Chart of Simulation	XXVI
Appendix E	Uncertainties Analysis.....	XXVII
Appendix F	Specifications of Experimental Devices	XXX
Appendix G	Calibration Data of Measuring Devices	XLI
Appendix H	Specification of Lubricant.....	XLVIII
Appendix I	Journal Bearing Model.....	L

List of Figures

Figure 2.1 Vapour compression refrigeration system [39]	5
Figure 2.2 Vapour compression refrigeration system with an expander [44].....	6
Figure 2.3 Schematic model of rotary type sliding vane compressor [49]	7
Figure 2.4 Schematic model of a rolling piston compressor [70]	9
Figure 2.5 Schematic view of a screw compressor [85]	11
Figure 2.6 Physical working principle of scroll compressor [99]	13
Figure 2.7 Schematic view of a swing compressor [15]	15
Figure 2.8 Schematic model of revolving vane compressor [21]	16
Figure 2.9 Schematic diagram and sectional view of improved revolving vane compressor [113]	16
Figure 2.10 Top and cross sectional view of CVEC [22]	17
Figure 2.11 Illustration of internal leakage paths in a rolling piston compressor [83]	21
Figure 3.1 Schematic of the rolling piston compressor and its major components	27
Figure 3.2 Schematic of the swing compressor and its major components	29
Figure 3.3 Schematics of (a) swing bushes in swing compressor and (b) vane- spring assembly in rolling piston compressor	29
Figure 3.4 Schematic of swing compressor with (a) long vane and (b) short vane	30
Figure 3.5 Modifying (a) swing bushes into (b) one-piece vane bush	31
Figure 3.6 The shoulders of vane bush are extended	31
Figure 3.7 Two slots are cut for the extended shoulders	32
Figure 3.8 Conceptual design of U-Vane compressor (UVC)	32
Figure 3.9 Comparison between swing compressor [108] and UVC	33
Figure 3.10 Main components of UVC: (a) cylinder, (b) rotor, (c) vane bush, (d) shaft and eccentric	34
Figure 3.11 Top sectional view of the assembly of UVC main components	34
Figure 3.12 Side sectional view of UVC conceptual design	36
Figure 3.13 Snapshots of the operation of U-Vane compressor	37

Figure 3.14 Size comparison between (a) swing compressor [108] and (b) UVC	38
Figure 4.1 Schematic diagram of U-Vane compressor (top view)	42
Figure 4.2 Schematic diagram of geometric of U-Vane compressor.....	44
Figure 4.3 Schematic diagram of the volume at the intersection of vane and cylinder	45
Figure 4.4 Schematic diagram of flow through orifice	51
Figure 4.5 Variation of suction port opening area (orifice flow area)	53
Figure 4.6 Variation of discharge port opening area	55
Figure 4.7 Fluid flow area when the valve lifts	56
Figure 4.8 Schematic diagram of geometry of the valve reed	57
Figure 4.9 Valve mode shape function	60
Figure 5.1 Geometrical dimensions of bush and vane	63
Figure 5.2 Vane tilting and the resultant force diagram of vane bush	63
Figure 5.3 Geometrical relations and force diagram of rotor	66
Figure 5.4 Partitions of the geometry of vane bush	69
Figure 5.5 Partitions of the geometry of rotor	69
Figure 5.6 Assumption of the speed of Section II	73
Figure 5.7 Internal leakage paths of U-Vane compressor.....	77
Figure 5.8 Close-up view of the radial clearance geometry	78
Figure 5.9 Fanno Flow model of leakage flow through radial clearance [150].	78
Figure 5.10 Assumed channel length for choked flow to occur	80
Figure 5.11 Leakage through the vane end-face clearance when (a) shoulders in the slots and (b) shoulders out of the slots.....	82
Figure 5.12 Fanno Flow model of leakage through the vane end-face clearance	82
Figure 5.13 Pressure-volume diagram (PV diagram) of U-Vane compressor...	83
Figure 5.14 Theoretical pressure-volume diagram of a compressor.....	86
Figure 6.1 Result of step size independence test	91
Figure 6.2 Variation of pressure with different step sizes	92

Figure 6.3 Basic simulation results: variations of (a) working chamber volume, (b) working fluid pressure, (c) working fluid mass inside the working chamber and (d) valve displacement	93
Figure 6.4 Variation of (a) vane exposure length and rate of change of vane exposure length, (b) γ_I and rate of change of γ_I	95
Figure 6.5 Variation of working fluid (a) pressure and (b) mass with and without internal leakage: solid lines – without leakage, dashed lines – with leakage	96
Figure 6.6 Leakage mass flow rate through various leakage paths	97
Figure 6.7 Journal bearing analysis: (a) minimum oil film thickness and compressor resultant load, (b) maximum pressure and required oil flow rate...	98
Figure 6.8 Variation of various frictional power losses.....	99
Figure 6.9 Variation of components of total instantaneous power of UVC	100
Figure 6.10 Components of (a) average frictional power loss and (b) power input	101
Figure 6.11 Effect of operating speed on the performance of UVC: (a) efficiencies and (b) variation of working fluid pressure	103
Figure 6.12 Effect of discharge pressure on the performance of UVC: (a) efficiencies and (b) total leakage mass flow rate	104
Figure 6.13 Variations of pressure and valve deflection when the discharge pressure is (a) 2.0 bar (abs.) and (b) 10.0 bar (abs.)	105
Figure 6.14 Effect of vane length on the performance of UVC.....	105
Figure 6.15 Variations of (a) vane side normal force and (b) friction between vane side and vane slot ($L_{f,rvs}$) with vane length	106
Figure 6.16 Effect of the vane bush shoulder length on the performance of UVC: (a) efficiencies and (b) vane side frictional loss ($L_{f,rvs}$)	107
Figure 6.17 Variations of (a) rotor end-face friction ($L_{f,ref}$) and (b) frictional loss between eccentric and inner rotor surface ($L_{f,ecc}$) with rotor wall thickness..	108
Figure 6.18 Effect of rotor wall thickness on the performance of UVC.....	108
Figure 6.19 Effect of rotor-to-cylinder radius ratio on the performance of UVC: (a) efficiencies and (b) various frictional losses	109

Figure 6.20 Variations of (a) rate of change of vane length and (b) rate of change of angle γ_l with rotor-to-cylinder radius ratio	110
Figure 6.21 Effect of slenderness ratio on the performance of UVC	111
Figure 6.22 Effect of slenderness ratio on (a) minimum oil film thickness and (b) variations of oil film thickness.....	111
Figure 7.1 Shaft and eccentric: (a) isometric views (b) detailed side view and bottom view	117
Figure 7.2 Free body diagram of forces acting on shaft and eccentric	118
Figure 7.3 Shear force diagram and bending moment diagram of shaft and eccentric	119
Figure 7.4 Endurance strength versus tensile strength for wrought steel [223]	121
Figure 7.5 Size factors for various shaft diameters [223].....	121
Figure 7.6 Isometric views of rotor: (a) view from top-front and (b) view from bottom-back	123
Figure 7.7 Detailed design of rotor	124
Figure 7.8 Free body diagram of vane	124
Figure 7.9 Shear force diagram and bending moment diagram of vane	125
Figure 7.10 (a) Stress analysis of vane and (b) deflection of vane	127
Figure 7.11 Isometric views of vane bush: (a) view from top-front and (b) view from bottom-back.....	127
Figure 7.12 Detailed design of vane bush.....	128
Figure 7.13 (a) Forces acting on vane bush; (b) free body diagram of left shoulder and (c) free body diagram of right shoulder	129
Figure 7.14 Shear force and bending moment diagrams of (a) left shoulder and (b) right shoulder.....	129
Figure 7.15 (a) Stress analysis of vane bush and (b) deflection of vane bush under Scenario 1 (7.0 bar).....	131
Figure 7.16 (a) Stress analysis of vane bush and (b) deflection of vane bush under Scenario 2 (1.0 bar).....	131
Figure 7.17 Cylinder assembly: (a) isometric view and (b) exploded view	132

Figure 7.18 Discharge reed valve of UVC prototype (a) discharge reed valve components; (b) assembly of reed valve on cylinder.....	132
Figure 7.19 Detailed design of cylinder.....	133
Figure 7.20 Detailed design of lower bearing.....	134
Figure 7.21 (a) Stress analysis of cylinder and (b) deflection of cylinder	136
Figure 7.22 Housing assembly: (a) isometric view and (b) exploded view.....	137
Figure 7.23 Details design of base plate	137
Figure 7.24 Housing: (a) isometric view and (b) detailed design.....	138
Figure 7.25 Cover: (a) isometric view and (b) detailed design.....	138
Figure 7.26 Assembly of lip seal roof.....	139
Figure 7.27 (a) Stress analysis and (b) deflection of cover.....	142
Figure 7.28 Counterweight: (a) isometric view and (b) top view.....	143
Figure 7.29 (a) Assembly of counterweight on shaft; (b) original position of centre of gravity; (c) shifted centre of gravity after the counterweight is mounted	143
Figure 7.30 (a) Isometric view of UVC prototype and (b) top view of the prototype without cover	144
Figure 7.31 Cross-sectional side view of UVC prototype assembly	145
Figure 7.32 Cross-sectional top view of UVC prototype assembly.....	145
Figure 7.33 Shaft rotation of UVC prototype	146
Figure 7.34 Schematics of lubrication system in U-Vane compressor prototype	148
Figure 7.35 Network circuit of lubrication system	150
Figure 7.36 Housing assembly is mounted on the test rig	155
Figure 7.37 Cylinder assembly sit on the base plate.....	156
Figure 7.38 Vane bush is slotted into the cylinder wall.....	156
Figure 7.39 Rotor is assembled on the cylinder.....	157
Figure 7.40 Suction pipe is assembled onto the cylinder.....	157
Figure 7.41 Shaft is inserted into the centre of rotor	158
Figure 7.42 Cover is assembled to the housing	158
Figure 7.43 Final assembly of UVC prototype.....	159
Figure 8.1 Schematic of the experiment setup.....	160

Figure 8.2 Actual experiment setup: front view and motor control panel	161
Figure 8.3 Actual experiment setup: suction and discharge	161
Figure 8.4 Actual experiment setup: back view.....	162
Figure 8.5 Friction of various lip seals as a function of pressure [240].....	166
Figure 8.6 Lubricant flow through the eccentric (a) without spiral groove; (b) with spiral groove	167
Figure 8.7 Schematic view of the changes in radial clearances.....	168
Figure 8.8 Measured power input at various operational conditions.....	169
Figure 8.9 Comparison between predicted and measured power inputs	171
Figure 8.10 Overall discrepancy between predicted and measured power inputs	172
Figure 8.11 Measured mass flow rate at various operational conditions.....	173
Figure 8.12 Volumetric efficiency of the compressor prototype at various operational conditions	173
Figure 8.13 Comparison between predicted and measured mass flow rate	174
Figure 8.14 Overall discrepancy between predicted and measured mass flow rates	175
Figure 8.15 Pressure transducers blocked by rotor at (a) 0°, (b) 70° and (c) 300°	176
Figure 8.16 Comparison between predicted and measured variations of instantaneous working chamber pressures at various operating speeds (average discharge pressure = 2.61 bar)	177
Figure 8.17 Comparison between predicted and measured variations of instantaneous working chamber pressures at various operating speeds (average discharge pressure = 3.53 bar)	177
Figure 8.18 Comparison between predicted and measured variations of instantaneous working chamber pressures at various operating speeds (average discharge pressure = 5.17 bar)	178
Figure 8.19 Misalignment between valve reed and discharge port	179
Figure 8.20 Comparison between predicted and measured results with 0.5 as the value of the effective force area coefficient (2.61 bar)	180

Figure 8.21 Comparison between predicted and measured results with 0.5 as the value of the effective force area coefficient (5.17 bar)	180
Figure 8.22 Noncircularity of the inner rotor surface diameter	181
Figure 8.23 Comparison between predicted and measured results after increased the instantaneous leakage flow through radial clearance (average discharge pressure = 5.17 bar, $C_{force} = 0.5$)	182
Figure 8.24 Shaft and eccentric after the experiment	183
Figure 8.25 Wear mark on the aluminium bronze bushing (rotor) after experiment	184
Figure 8.26 Wear mark on the vane after experiment	184
Figure A.1 Illustration of dimensions definition (a) swing compressor and (b) UVC	II
Figure C.1 Fluid domain of the leakage through rotor end-face clearance and boundary conditions (at 180°).....	VIII
Figure C.2 Result of CFD numerical simulation at 180°: (a) velocity profile (b) pressure distribution.....	VIII
Figure C.3 Leakage flow velocity curve fit	IX
Figure C.4 Heat transfer paths in horizontal direction.....	XI
Figure C.5 Heat transfer paths in vertical direction.....	XII
Figure C.6 Heat transfers due to oil flow.....	XII
Figure C.7 Expected working fluid flow field in (a) revolving vane compressor [170] and (b) U-Vane compressor [248].....	XVII
Figure C.8 Schematic diagram of (a) flow circulating in the housing chamber and (b) rectangular duct model	XVIII
Figure C.9 Discrete elements from the main components.....	XXIV
Figure D.1 Algorithm flow chart of simulation	XXVI
Figure F.1 Technical specification of ABB motor (model: 3GAA 131 213-BSE)	XXX
Figure F.2 Technical specification of Futek torque meter (model: TRS605)	XXXI
Figure F.3 Technical specifications of WIKA pressure transducer (model: A-10): page 1/3.....	XXXII

Figure F.4 Technical specifications of WIKA pressure transducer (model: A-10): page 2/3.....	XXXIII
Figure F.5 Technical specifications of WIKA pressure transducer (model: A-10): page 3/3.....	XXXIV
Figure F.6 Technical specifications of Endevco pressure transducer (model: 8530B-500): page 1/2	XXXV
Figure F.7 Technical specifications of Endevco pressure transducer (model: 8530B-500): page 2/2	XXXVI
Figure F.8 Technical specifications of Brooks MR3000 series variable area flow meter (model: MR-3A17-BLBN): page 1/2	XXXVII
Figure F.9 Technical specifications of Brooks MR3000 series variable area flow meter (model: MR-3A17-BLBN): page 2/2	XXXVIII
Figure F.10 Technical specification of TEF-MET/P bushing (Page 1/2).....	XXXIX
Figure F.11 Technical specification of TEF-MET/P bushing (Page 2/2).....	XL
Figure G.1 Calibration data for Futek torque meter (model: TRS 605)	XLI
Figure G.2 Calibration data for thermocouple (TCT-01)	XLI
Figure G.3 Calibration data for thermocouple (TCT-02)	XLII
Figure G.4 Calibration data for thermocouple (TCT-04)	XLII
Figure G.5 Calibration data for thermocouple (TCT-05)	XLIII
Figure G.6 Calibration data for thermocouple (TCT-08)	XLIII
Figure G.7 Calibration data for Endevco pressure transducer (MPT-01 (29408))	XLIV
Figure G.8 Calibration data for Endevco pressure transducer (MPT-02 (29407))	XLIV
Figure G.9 Calibration data for Endevco pressure transducer (MPT-03 (29410))	XLV
Figure G.10 Calibration data for WIKA pressure transducer (PT-02)	XLV
Figure G.11 Calibration data for WIKA pressure transducer (PT-03)	XLVI
Figure G.12 Calibration data for WIKA pressure transducer (PT-04)	XLVI
Figure G.13 Calibration data for WIKA pressure transducer (PT-05)	XLVI
Figure G.14 Calibration data for WIKA pressure transducer (PT-06)	XLVII
Figure G.15 Calibration data for WIKA pressure transducer (PT-07)	XLVII

Figure H.1 Technical data sheet of lubricant	XLVIII
Figure H.2 Viscosity – temperature chart of the lubricant.....	XLIX
Figure I.1 Schematic diagram of a journal bearing.....	L

List of Tables

Table 3.1 Comparison between swing compressor and U-Vane compressor....	39
Table 4.1 Geometrical representative of U-Vane compressor.....	43
Table 6.1 Operational conditions of simulation.....	89
Table 6.2 Predicted overall performance of UVC (R1234-yf)	101
Table 6.3 Comparison of predicted performance with various refrigerants	102
Table 6.4 Comparison of mechanical efficiency between UVC and existing rotary compressors.....	112
Table 6.5 Comparison of volumetric efficiency between UVC and existing rotary compressors.....	112
Table 7.1 Specifications of prototype	115
Table 7.2 Dimensions of major components	116
Table 7.3 Mechanical properties of 17-4 PH stainless steel [224]	120
Table 7.4 Approximate reliability factors [223]	120
Table 7.5 Mechanical properties of AISI 4130 steel [229].....	134
Table 7.6 Materials use in UVC prototype	151
Table 7.7 Physical dimensions of actual prototype	152
Table 7.8 Clearances between components of prototype.....	154
Table 8.1 Summary of the uncertainties of measuring devices	163
Table 8.2 Estimation of lubricant temperature in the prototype	170
Table A.1 Comparison of dimensions between swing compressor and UVC.....	I
Table C.1 Dimensions and operational conditions for numerical simulation..	VII
Table C.2 Summary of heat transfer paths.....	XII
Table C.3 Summary of the effective thermal resistance	XXII
Table E.1 Uncertainties of measuring devices.....	XXVII
Table E.2 Calculated uncertainties of power input.....	XXVII
Table E.3 Calculated uncertainties of volumetric efficiency.....	XXVIII

Nomenclature

A	Area (m ²); Cross sectional area (m ²)
A_{flow}	Flow area (m ²)
A_{force}	Force area (m ²)
A_{proj}	Projected area (m ²); Reference area (m ²)
$A_{tensile}$	Tensile stress area (m ²)
C	Centre axis of shaft (-); damping coefficient
C_d	Coefficient of discharge (-)
C_{dr}	Drag coefficient (-)
$C_{dv,eff}$	Effective force area coefficient (-)
C_R	Reliability factor (-)
C_s	Size factor (-)
COP	Coefficient of performance (-)
D_b	Bearing diameter (m)
D_{sg}	Width of the spiral groove (m)
D_e	Equivalent hydraulic diameter (m)
E	Energy (J); Young's modulus (Pa)
E_{dv}	Young's modulus of discharge valve (Pa)
$\dot{E}$	Rate of change of energy (J/s or W)
F	Force (N); Load (N)
F_1	Force acting on the vane slot side at the vane tip (N)
F_2	Force acting on the tip of extended shoulder by the vane (N)
F_{AB}	Force acting on projected line AB (N)
F_D	Drag force (N)
F_{Rx}	Resultant force acting on the rotor in x-direction (N)
F_{Ry}	Resultant force acting on the rotor in y-direction (N)
$F_{slot,vb}$	Force acting on the vane slot in the vane bush (N)
F_{vboc}	Force acting on vane bush by cylinder (N)
F_{vbsd}	Force acting on the bush shoulder (N)
F_x	Resultant load projected in the x-direction (N)
F_y	Resultant load projected in the y-direction (N)
H_{sg}	Depth of the spiral groove (m)
I	Sectional moment of inertia (m ⁴)
I_{vb}	Moment of inertia of vane bush (kgm ²)
I_{ro}	Moment of inertia of rotor (kgm ²)
K	Flow coefficient (-)

K_t	Stress concentration factor (-)
L	Length (m)
L_b	Length of bearing (m)
$L_{f,b}$	Frictional loss of the lubricated journal bearing (W)
$L_{f,ecc}$	Frictional loss between eccentric and the inner rotor surface (W)
$L_{f,ecef}$	Eccentric end-face frictional loss (W)
$L_{f,ls}$	Lip seal friction (W)
$L_{f,ref}$	Resultant rotor end-face frictional loss (W)
$L_{f,rvs}$	Vane side frictional loss (W)
$L_{f,vbs}$	Vane bush side frictional loss (W)
$L_{f,vbef}$	Resultant vane bush end-face frictional loss (W)
M	Mach number (-); Bending Moment (Nm)
M_t	Mach number at the throat (-)
M^*	Unity Mach number (= 1)
M_t^*	Assumed Mach number at the throat (-)
N	Safety factor (-); Number of data points (-)
Nu	Nusselt number (-)
P	Pressure (Pa); Power (W)
P^*	Pressure of fluid when the flow is choked (Pa)
P_{com}	Pressure in compression chamber (Pa)
P_e	Pressure at the exit (Pa)
P_O	Orvirk's solution (Pa)
P_S	Sommerfeld's solution (Pa)
P_{suc}	Pressure in suction chamber (Pa)
P_t	Pressure at the throat (Pa)
Pr	Prandtl number
Q	Heat transfer (J); Flow rate (m ³ /s)
$\dot{Q}$	Heat transfer rate (J/s or W)
R	Centre axis of eccentric/rotor (-); Specific gas constant (J/kg·K); Thermal resistance (K/W)
R_b	Radius of bearing (m)
Re	Reynold's number (-)
T	Temperature (K); Torsion (Nm)
$\bar{T}$	Average measured torque (Nm)
T^*	Temperature of fluid when the flow is choked (K)
T_{com}	Temperature in compression chamber (K)
T_e	Temperature at the exit (K)
U	Velocity (m/s)
U_b	Velocity of bearing (m/s)

U_j	Velocity of journal (m/s)
V	Volume (m ³); Shear force (N); Velocity (m/s)
$\bar{V}$	Relative velocity (m/s)
V_{suc}	Volume of suction chamber (m ³)
V_{com}	Volume of compression chamber (m ³)
V_{total}	Volumetric displacement (cm ³ /rev)
W	Work done (J)
$\dot{W}$	Rate of work done/power (J/s or W)
$\dot{W}_{com}$	Compression power (W)
$\bar{\dot{W}}_{comp,actual}$	Actual average compression power (W)
$\bar{\dot{W}}_{comp,ideal}$	Ideal average compression power (W)
$\dot{W}_{f,loss}$	Total frictional power loss (W)
$\bar{\dot{W}}_{f,loss}$	Average frictional power loss (W)
$\dot{W}_{input}$	Power input (W)
$\dot{W}_{input,m}$	Measured power input (W)
$\dot{W}_{inst}$	Total instantaneous power (W)
$Y_{measured}$	Value of measured data point (-)
$Y_{predicted}$	Value of predicted data point (-)
d_{dv}	Diameter of discharge valve reed (m)
e	Eccentricity of the rotor with respect to the cylinder (m)
g	Acceleration of gravity (m/s ²)
$g_n(t)$	Mode participation factor (m)
h	Specified enthalpy (J/kg)
h_t	Heat transfer coefficient (W/m ² •K)
h_{min}	Minimum oil film thickness (m)
k	Thermal conductivity (W/m•K)
l	Length (m)
l_{com}	Axial length of compressor (m)
l_{dv}	Total length of valve reed (m)
$l_{dv,eff}$	Effective length of valve reed (m)
l_{ecc}	Axial length of eccentric (m)
l_f	Frictional channel length (m)
$l_{f,vef}$	Frictional channel length of the vane end-face clearance (m)
l_f^*	Assumed frictional channel length (m)
l_v	Vane exposure length in the working chamber (m)
$l_{v,max}$	Maximum vane exposure length in the working chamber (m)

l_{vbsd}	Length of the bush shoulder (m)
l_{vro}	Length of vane attached on the rotor (m)
m	Mass (kg)
$\dot{m}$	Mass flow rate (kg/s)
$\frac{dm}{dt}$	Mass flow rate (kg/s)
$\dot{m}_{actual}$	Actual mass flow rate (kg/s)
$\dot{m}_{ideal}$	Ideal mass flow rate (kg/s)
$\left(\frac{dm}{dt}\right)_{leak,vef}$	Leakage mass flow rate through the vane end-face clearance (kg/s)
$\left(\frac{dm}{dt}\right)_{leak,rad}$	Leakage mass flow rate through the radial clearance (kg/s)
n_{sg}	Pitch of the spiral groove (-)
r	Radius (m)
r_{cyl}	Radius of cylinder (m)
r_{ecc}	Radius of eccentric (m)
r_{hs}	Inner radius of housing (m)
r_{ro}	Radius of rotor (m)
r_{roi}	Inner radius of rotor (m)
r_s	Radius of shaft (m)
r_{vb}	Radius of vane bush (m)
s_n	Endurance strength (Pa)
s_y	Yield strength (Pa)
t	Time (s)
t_{dv}	Thickness of discharge valve reed (m)
t_{ro}	Thickness of rotor wall (m)
u	Specified internal energy (J/kg)
v	Velocity (m/s)
v_{centre}	Velocity of centre (m/s)
v_e	Velocity at the exit (m/s)
v_{leak}	Velocity of leakage flow (m/s)
w	Width (m)
w_{dv}	Width of discharge valve reed (m)
w_e	Width of the chamber at the exit of the channel (m)
$w_{e,vef}$	Effective width at the exit of the vane end-face clearance (m)
w_{vbsd}	Width of the bush shoulder (m)
w_{vro}	Width of the vane (m)
x	Horizontal axis

y	Valve displacement (m); vertical axis
y_{stop}	Height of valve stop (m)
z	Vertical elevation (m); height (m)
ΔP	Differential pressure (Pa); Pressure across flow element (Pa)
ΔT	Temperature difference (°C or K)
Δh_{evap}	Difference in specific enthalpy between the fluids at evaporator inlet and outlet (J/kg)
θ_1	Rotational angle of shaft (rad)
θ_2	Rotational angle of radial contact line w.r.t. eccentric centre (rad)
θ_a	Start of the oil film pressure of journal bearing (rad)
θ_b	End of the oil film pressure of journal bearing (rad)
ω_1	Angular velocity of shaft (rad/s)
ω_n	Natural frequency (rad/s)
γ_1	Subtended angle at centre of vane bush (rad)
ρ	Density (kg/m ³)
ρ_{air}	Density of air (kg/m ³)
ρ_{ro}	Density of rotor material (kg/m ³)
ρ_{suc}	Density of working fluid at suction condition (kg/m ³)
ρ_{vb}	Density of vane bush material (kg/m ³)
β	Spiral groove angle (rad)
σ_{all}	Allowable stress (Pa)
σ_{dis}	Discrepancy (-) or (%)
ε	$= \frac{\epsilon}{\delta_b}$, eccentricity ratio (-)
ε_b	Eccentricity ratio of the journal bearing (-)
ε_{ecc}	Eccentricity ratio of the eccentric (-)
$\dot{\varepsilon}$	Rate of change of eccentricity ratio w.r.t. time (s ⁻¹)
ϵ	Eccentricity (m)
κ	Specific heat ratio (-)
λ	Friction factor (-)
ϕ	Attitude angle (rad)
$\dot{\phi}$	Rate of change of attitude angle w.r.t. time (rad/s)
$\phi_n(x)$	Mode shape function (-)
φ	Tangential friction per unit circumference (N/m)
ξ	Damping ratio (-)
δ	Clearance (m); Gap distance (m)
δ_b	Clearance between journal and bearing (m)
$\delta_{ecc,rad}$	Radial clearance between the eccentric and the inner rotor surface (m)
δ_{ecf}	Eccentric end-face clearance (m)

δ_{rad}	Radial clearance (m)
$\delta_{rad,i}$	Instantaneous radial clearance (m)
δ_{vbef}	Vane bush end-face clearance (m)
δ_{vef}	Vane end-face clearance (m)
μ	Dynamic viscosity (Pa•s); Coefficient of friction (-)
μ_{ecro}	Coefficient of friction between the eccentric and the inner rotor surface (-)
μ_{lub}	Dynamic viscosity of lubricant (Pa•s)
μ_{vs}	Coefficient of friction at vane side (-)
μ_{vbs}	Coefficient of friction at vane bushing side (-)
η_{comp}	Compression efficiency
η_{mech}	Mechanical efficiency (-)
$\eta_{overall}$	Overall efficiency (-)
η_s	Isentropic efficiency (-)
η_{vol}	Volumetric efficiency (-)

Subscript

<i>Inst</i>	Instantaneous
<i>R</i>	Resultant
<i>all</i>	Allowable
<i>b</i>	Bearing
<i>ch</i>	Characteristic
<i>com</i>	Compression chamber; Compressor
<i>comp</i>	Compression
<i>cv</i>	Control volume
<i>cyl</i>	Cylinder
<i>dr</i>	Drag
<i>disc</i>	Discharge
<i>dv</i>	Discharge valve
<i>e</i>	Exit
<i>ecc</i>	Eccentric
<i>ecef</i>	Eccentric end-face
<i>ef</i>	End-face
<i>eff</i>	Effective
<i>evap</i>	Evaporator
<i>f</i>	Friction; Fluid
<i>i</i>	Inner
<i>inertia</i>	Inertia
<i>j</i>	Journal
<i>lub</i>	Lubricant
<i>loss</i>	Loss
<i>max</i>	Maximum
<i>mat</i>	Material

<i>mech</i>	Mechanical
<i>min</i>	Minimum
<i>n</i>	Normal
<i>o</i>	Outer
<i>press</i>	Pressure
<i>proj</i>	Projected
<i>r</i>	Rotational
<i>rad</i>	Radial
<i>ref</i>	Rotor end-face
<i>ro</i>	Rotor
<i>rvs</i>	Vane side
<i>s</i>	Shaft
<i>sdro</i>	Shoulder slot in rotor
<i>sg</i>	Spiral groove
<i>suc</i>	Suction
<i>t</i>	Tangential; Translational; Throat
<i>UVC</i>	U-Vane compressor
<i>v</i>	Vane
<i>vb</i>	Vane bush
<i>vbef</i>	Vane bush end-face
<i>vboc</i>	Vane bush on cylinder
<i>vbs</i>	Vane bush side
<i>vbsd</i>	Vane bush shoulder
<i>vol</i>	Volumetric
<i>vro</i>	Vane (attached to rotor)
<i>vs</i>	Vane side
<i>w</i>	Wall

Chapter 1 Introduction

In this chapter, background, motivation and thesis overview are presented.

1.1 Background and Motivation

A compressor is a mechanical device that draws in compressible working fluid such as gases, increases the working fluid pressure, and discharges it at a higher nominal pressure [1]. It is one of the main components in air-conditioning system, refrigeration system and gas compression systems which are commonly used in both household and industrial applications [2].

There are various types and sizes of compressors to suit different applications. In general, compressors can be classified into two main categories, namely positive displacement and dynamic types [3]. For positive displacement type compressor, the increase and subsequently decrease in the working chamber volume cause the working fluid to be drawn in, compressed and discharged, respectively. Reciprocating compressor [4], and rotary compressors such as sliding vane compressor [5], rotary screw compressor [6], rolling piston compressor [7] and scroll compressor [8] are the typical examples of positive displacement type. Instead of capturing a volume of working fluid, a dynamic type compressor speeds up the velocity of working fluid by pushing the working fluid with the rotation of impeller or blades and increases its pressure by converting the kinetic energy into pressure rise. Both axial compressor [9] and centrifugal compressor [10] are of the dynamic type compressor.

Compressors have been widely used and its demand is growing. In recent market research for room air-conditioners applications, rotary compressor contributes 90% of the compressor market, which is around 198 million units [11]. A research showed that rotary compressor will drive global market toward US\$30 billion in 2020 [12]. Furthermore, the efficiency of compressor is dramatically improving due to the persistent design and development effort by research centres, universities and manufacturers.

New innovative compressor mechanisms are introduced in recent years. For example, swing compressor [13-15], piezo compressor [16], twin-plate

compressor [17], revolving vane compressor and its variants [18-21], cross-vane mechanism [22], swing vane compressor [23] and double swing-vane compressor [24]. In this project, a novel compressor, namely U-Vane compressor which aims to improve the swing compressor and rolling piston compressor will be introduced and examined.

1.2 Objective and Scope

The objective of this research project is to develop and study a material-saving compressor theoretically and experimentally. In order to accomplish the project objective, the work scope is presented in sequential order as follows:

1. To review existing compressor designs and recognise their pros and cons.
2. To conceptually design a new compressor.
3. To develop the mathematical model and carry out the theoretical analysis and parametric studies of the compressor.
4. To design, fabricate and instrument the prototype of the compressor.
5. To conduct experimental study on the compressor performance.
6. To verify the mathematical model with measured data.

1.3 Purpose of the thesis

This thesis is submitted to Nanyang Technological University Singapore to fulfil the partial requirement for its research students' Ph.D. candidature. This thesis consists of the research study that the author has completed during candidature period.

1.4 Organisation of the thesis

This thesis comprises 9 chapters and it starts with an introduction to the research project motivation, objective and scope in Chapter 1. Chapter 2 reviews the relevant literatures including compressor mechanisms and existing simulation studies. It is followed by Chapter 3, where the conceptual design and the embodiment design of the compressor is presented based on the operating requirements. The mathematical formulation is shown in Chapters 4 and 5. The

governing equations of geometrical model, thermodynamic model and valve response model are shown in Chapter 4 while dynamic model, in which force analysis and various mechanical losses are identified, and internal leakage model are presented in Chapter 5. The above models are useful in predicting the theoretical performance. In Chapter 6, the result from the mathematical model is presented and parametric studies are carried out. After that, Chapter 7 shows the detailed design of the entire prototype. The experimental study and the validation of mathematical model are shown in Chapter 8. Lastly, Chapter 9 summarises the findings in this research project, along with the recommended future works.

Chapter 2 Literature Review

In Chapter 2, the design and development of a compressor by various research studies are reviewed and discussed. It comprises four sections: (1) refrigeration systems and energy consumption; (2) the existing compressor mechanisms and designs; (3) theoretical model and simulation studies; (4) concluding remarks.

2.1 Overview of Refrigeration and Energy Consumptions

Over the past decades, the world overall energy consumption has been growing drastically. This is especially so in China, the world's largest consumer of energy, where it saw an increment in energy consumption by 50 % since 2008 [25]. The energy consumption is mainly driven by the rapid urbanization and industrialization. According to the review by Dong *et al* [26], electricity contributes around 40 % of the overall energy consumption and it is essential to the industry and building (residential) sectors [27]. U.S. Energy Information Administrative [28] reported that cooling, air-conditioning and refrigeration (AC&R) account for the largest share of electricity consumption, which are 26 % and 21.2 % respectively in both households and commercial sectors in the United States. In Singapore, around 50.5 % of electricity consumption was attributed to household and commercial-related sectors in 2017 [29], which had increased by 17 % from 2007 level [30]. A study by Singapore National Environment Agency [31] showed that AC&R accounted for 41 % of household electricity consumption, meaning that about US\$560 million was consumed alone by these appliances in Singapore (with population of 5.6 million) [32], while it was more than US\$25 billion in United States annually [33].

In fact, AC&R systems do not only consume energy, but also indirectly result in large amount of greenhouse gases emissions due to the electricity required for their operation [27, 34], especially the electricity generated from burning fossil fuels - petroleum, natural gas and coal. The climate change caused by these greenhouse gases would again increase the energy consumed by AC&R, which led to the international treaties, namely Kyoto Protocol [35], Copenhagen

Accord [36] and the Paris Agreement [37] to curb the greenhouse gasses emissions and climate change. Based on these issues, high energy efficiency AC&R system is required. Currently, the most widely used refrigeration systems is the vapour compression refrigeration system [38] because of its simplicity. Figure 2.1 shows the basic components of a single-stage vapour compression refrigeration system, which are a condenser, an expansion valve, an evaporator and a compressor [39].

Figure 2.1 Vapour compression refrigeration system [39]

There are few ways to improve the energy efficiency of a refrigeration system. Firstly, the evolution from natural gases to synthetic gases as refrigerant, and subsequently the improvements in the component and system designs over the past century, have successfully reduced the energy consumption by the conventional refrigeration system [40]. Although some refrigerants are causing environmental issues when they leak out from the system to the atmosphere, they play a crucial role in an energy-saving refrigeration system [41-43]. For instance, replacing the CO₂ with R134a may effectively reduce the throttling loss during the expansion process in a CO₂ refrigeration system.

Figure 2.2 Vapour compression refrigeration system with an expander [44]

Another way of reducing energy consumption of a refrigeration system is to recover the work done from the expansion process by replacing the traditional expansion valve with an expander, as shown in Figure 2.2 [44, 45]. The device was initially introduced to recover the large throttling loss during the expansion process in a CO₂ refrigeration system [40], where the recovered work was used to partially drive the compressor and increase the energy efficiency of the system. Several researchers combined two refrigeration systems to improve the performance of the combined system, such as an absorption-compression system, which was introduced to reduce energy consumption by utilising the waste heat [46]. The COP of such system is expected to be higher than a single system due to a lower differential pressure across the compressor. Chen *et al.* [47] proposed an absorption/absorption-compression refrigeration system for heat sources with large temperature change and the cooling energy output was increased by 28.21 % as compared to the conventional system. Takahiko *et al.* [48] revealed that the combined system in cascading type has achieved a 40 % of COP improvement while the sub-cooling type had smaller size and lower cost. However, compared to a single system, a combined system has redundant components which is costlier and bulkier in size.

From the literature, it can be seen that the demand of compressor is increasing, and researchers are working hard on achieving a more energy-saving, as well as a cost-saving compressor technology.

2.2 Compressor Mechanisms and Designs

In this section, designs and limitations of several existing positive displacement type compressors are presented.

2.2.1 Sliding Vane Compressor

A schematic for rotary type sliding vane compressor is shown in Figure 2.3 [49].

Figure 2.3 Schematic model of rotary type sliding vane compressor [49]

The basic components of a sliding vane compressor are a cylinder, an eccentrically positioned rotor with several directed radially or non-radially vane slots and vanes which slide in the vane slots [50]. During operation, the vane tips slide against the inner cylinder wall and the working chamber volume varies due to the off-set of the rotor to the cylinder. It is noticed that to achieve compression of the working fluid, the vane tip must be kept in contact with the inner cylinder wall and this is accomplished by the centrifugal or spring force from the rotor rotation. The suction, compression and discharge processes of sliding vane compressor occur continuously and simultaneously, and the suction process predetermines the discharge pressure due to the intrinsic volume compression

ratio. The discharge process of the compressed working fluid begins when the discharge port is exposed to the working chamber.

Fundamentally, for a traditional sliding vane compressor, discharge valves are not required and thus reduces the total valve loss. However, to operate at wider pressure ratios, a discharge valve may be required [51]. Studying the effect on vibration, Yee and Soedel [52] reported that no imbalance rotating was observed while positioning four identical vanes at equal spacing in the compressor. Moreover, dynamic analysis of the case of tilted vane that having an inclination of -30° to $+30^\circ$ were examined by Tramschek and Mkumbwa [53]. They commented that the inclination angle increased specific capacity, but the corresponding manufacturing cost would be higher.

Ideally, the vane tips of a rotary vane compressor should keep in contact with the inner cylinder wall. However, Tojo *et al.* [54] found that under certain operational conditions, such as low speed and during starting-up, the chattering phenomena, where the vanes tips get separated from contacting the inner cylinder wall and spring back to the wall, caused noise and reduced the efficiency and durability of the compressor. To keep vane in touch with inner cylinder wall, one of the solutions suggested by Shu *et al.* [55] is to apply an optimal value of vane slot back-pressure. Ucer and Aksel [56] simulated the rotary sliding vane compressor at different conditions and found that the vane thickness would affect both mass flow and power requirement. Kruse [57] investigated the vane tip frictional loss and concluded that eccentricity has larger effect on vane tip friction than the vane tip radius. Aradau and Costiuc [58] calculated the relative compressor power loss with various friction coefficients of the blades in sliding vane compressor. The vane frictional power loss has been established as a function of rotor rotation angle and an inclined vane made by proper materials was suggested to reduce the friction coefficient and thus the vanes friction. Bianchi and Cipollone [50, 59] have investigated frictional power reduction in sliding vane rotary compressor using modelling and experimental approaches. Their experiment discovered that the frictional loss accounts for 10 % of the mechanical power input and the loss increases with the revolution speed. The frictional power loss has successfully been reduced by 56 % and the specific

friction power was lowered by 0.28 kW/(m³/min) at lower operating speed when the vane mass was 40 % lighter. Researchers also found that the major leakage source is the gap between rotor end-face and cylinder end-plate [60, 61]. The sliding vane mechanism were used for pump [60-63] and expander design [64-67].

It is found that the constant rubbing between the inner cylinder wall and the vane tip and is indeed one of the major frictional losses in sliding vane compressor. Hence, special surface treatment that comes with additional production cost is required to reduce the vane wear rate.

2.2.2 Rolling Piston Compressor

Rolling piston compressor is one of the most widely used compressor in room air-conditioners [68]. Figure 2.4 shows a schematic model of a rolling piston compressor [69].

Figure 2.4 Schematic model of a rolling piston compressor [70]

Unlike sliding vane compressor which contains multiple vanes, the main components of the compressor are a cylinder, a rolling piston (roller), a vane and a spring. When the shaft rotates, the rolling piston which is mounted on the shaft eccentrically, rotates in the cylinder and roll along the inner cylinder wall [71]. One of the main characteristics of the rolling piston compressor is the spring mechanism that is acting on the vane and kept in contact with the cylindrical roller. However, a study done by Ooi *et al.* [72] indicated that the magnitude of spring force could affect the compressor performance.

Matsuzaka and Nagatomo [73] examined the performance of rolling piston type rotary compressor theoretically and experimentally. Various loss factors have also been recognized and a maximum compressor efficiency of 74 % was obtained. Yanagisawa and Shimizu [71] analysed various frictional losses with changing loading force and sliding velocity in the rolling piston type compressor and found that vane tip frictional loss is greatly related to the changes in sliding velocity on the vane tip. Whereas vane side frictional loss is influenced mainly by the magnitude of the tangential force at the contact point. Also, the vane tip that kept in contact with the roller may cause halt of operation due to the wear rate as mentioned by Liu and Kosco [74]. They have investigated rolling piston compressor with tilted vane and found that at certain positive angle, the vane tip normal force has a minimum value, but the vane friction loss has no observable change. This indicated that a tilted vane structure may improve the reliability and durability of both roller and vane, whereas the mechanical efficiency remained unchanged. To reduce the wear rate at the vane tip, researchers [75, 76] recommended vane surface treatment or produces the vane with wear resistance material. Bae *et al.* [77] introduced techniques to prevent vane jumping and visualized the vane jumping using image processing technique. Light vane and long spring were suggested to prevent vane jumping. Vane jumping causes loud noise and most importantly, leakage between the suction and compression chambers. Frictional and leakage losses are the two most significant losses in rolling piston compressor. The possible leakage paths are found at the contact point between piston and the cylinder, gap due to vane jumping, upper and lower end-faces clearances and clearance between the vane and the vane slot [7, 78, 79]. Among them, leakage through clearances between the piston and bearing, and between the vane and the vane slot have the most significant effect on input power [80]. Further studies about the leakage characteristics were also carried out by Hui and Gao [69], Wu and Chen [81] and Cai *et al.* [82, 83].

2.2.3 Screw Compressor

A screw compressor (see Figure 2.5) comprises two intermeshing helical rotors, one lobe (male rotor) and flute (female rotor). It is mainly used in applications where high cooling load is required such as commercial buildings

[84]. Working chamber is formed when the lobe and the corresponding flute meshes.

Figure 2.5 Schematic view of a screw compressor [85]

A screw compressor operates in the same way as sliding vane compressor, in which operation is carried out in the absence of valve and with its built-in volume compression ratio. The entire working process of screw compressor can be described in four continuous phases [86]. First, the working fluid is drawn in from the suction port when the pair of flute and lobe start to un-mesh. The process lasts until the working fluid that trapped in the pocket is separated from both suction and discharge ports. The volume of the trapped pocket is then gradually reduced and the working fluid in the pocket is being compressed. Lastly, as the compressor is designed without a valve, the working fluid is discharged uninterruptedly through the discharge port. In the experiment conducted by Fujiwara *et al.* [87], they pointed out that as the inter-lobe clearance increases, torque requirement, volumetric and adiabatic efficiency decreases. During operation, both male rotor and female rotor are dynamically balanced as they are rotating about their own axis of rotation and thus, noise and vibration are relatively low compared to other rotary compressors [88].

The complicated helical profile presented in a screw compressor is always a challenge in fabrication as it requires well mating condition [89]. The state-of-the-art machining techniques allow clearance gap to be significantly reduced. However, the smaller clearance results in the increase of the flow viscous loss

between the pair of rotors, causing the mechanical efficiency to be lower [87]. Sangfors [90] modelled the screw compressor performance analytically to predict the performance of the screw compressors. He compared the performances under two different rotor profiles and found that the newly developed rotor profile performed better. Tang and Fleming [91] also introduced an optimization method to find the optimum geometrical parameters for a helical twin-screw compressor. Furthermore, a computer program was developed to analyse the leakage paths of a twin screw compressor, and the rotors contact line was found to be the major factor in the reduction of volumetric efficiency [92]. However, by improving the profile, the volumetric efficiency has no significant increment either. Stosic *et al.* [6, 89, 93] deliberated on the optimization issues of the screw rotor profile. They proposed a rotor profile that is able to achieve a higher mechanical efficiency. In addition, due to the advancement in computational power, more research studies related to screw compressor have also been done by using Computational Fluid Dynamic (CFD) analysis [94-97].

A screw compressor rotor profile is found to be one of the most critical factors in the performance. In short, a screw compressor is always employed with costly manufacturing to ensure its performance.

2.2.4 Scroll Compressor

The concept of scroll compressor was first introduced by Léon Creux in 1905 as stated in the review by Elson *et al.* [98]. The working process of a scroll compressor is shown in Figure 2.6 [99]. During operation, the volume of the working chamber is being reduced and this volume reduction corresponds to an increase in pressure – the compression of working fluid. After that, the compressed working fluid is discharged continuously through the discharge port and the whole suction-discharge cycle repeats. Similar to a screw compressor, a scroll compressor operates in the absence of valves.

Figure 2.6 Physical working principle of scroll compressor [99]

Scroll compressors are widely used in residential and automotive applications due to its light weight, low noise and smooth operation with little torque variation [100, 101]. It also has a high volumetric efficiency in the absence of re-expansion volume. With these features, scroll compressor is getting more popular in recent decades and its usage is just second to rolling piston compressor. Though scroll compressor looks superior among the compressors, it does not dominate the compressor market yet due to its complex spiral profile which precise fabrication is required in comparison with the cylindrical design of the rolling piston compressor [102]. This signifies that the clearances exist in the scroll compressor must be well-controlled else it may result in substantial internal leakage. Eternad and Nieter [103] conducted a parametric study of scroll compressor and identified that wrap height controls the compressor optimization strongly such as the clearance of tip leakage loss. The wrap thickness also plays an important role in the rigidity of the scrolls in withstanding the working fluid pressure force, sustaining the thermal distortion and minimizing the tip leakage loss. Ishii *et al.* [104] completed a theoretical evaluation for the gas leakage flows through the clearances between two scrolls. It was found that axial leakage

resulted by axial clearance influenced the compressor efficiency more dominantly than the radial leakage. To reduce the axial clearance leakage, a back-pressure chamber was suggested [105] to keep the orbiting scroll in contact with the fixed scroll. However, this increases the friction at scroll tips and lowered the mechanical efficiency of a scroll compressor [101].

From past studies, it is noticed that a compromise between the leakage and friction is essential for a scroll compressor. Also, controlling the complicated spiral profile and clearance means high production cost. These are the main intrinsic reasons that limit the best performance of a scroll compressor.

2.2.5 Swing Compressor

Swing compressor was first introduced by Yamamoto [106] in 1996. As shown in Figure 2.7, the design of a swing compressor is basically similar to that of the rolling piston type, but the rotary structure adopts that of a swing structure to improve the durability and the lubricating ability [14]. Masuda *et al.* [14] performed theoretical simulation and experimental investigation to compare rotary compressor and swing compressor. The total efficiency of latter was found to be 5 ~ 9 % higher than the rotary type in the middle to low speed region and 2 % higher in the high-speed range. Furusho *et al.* [107] conducted numerical and experimental studies on swing compressor characteristic and concluded that the total efficiency is higher than rotary type in capacity range of 15.5kW. The study also stated that different refrigerants should be used in different capacity range in order to maximize the efficiency of a swing compressor. Recently, swing mechanism has also been further developed into other variants. Hu *et al.* [23] introduced swing vane compressor (SVC) and reported that the mechanical efficiency in SVC is higher than that in the sliding vane compressor at a rotation speed below 4000 rev/min while the total frictional loss in SVC is 65 % less than sliding vane compressor. In 2017, Yang *et al.* [24] introduced a double swing vane compressor (DSVC), which the volumetric flow rate was reported to be 1.6 times more than that in the SVC. Recently, Pan *et al.* [108, 109] examined the leakage losses in a valve-less swing compressor and concluded that the major leakage losses are leakages through eccentric end-face clearance and radial

clearance, which took up 51.50 % and 26.72 % of the total leakage mass flow rate, respectively.

Figure 2.7 Schematic view of a swing compressor [15]

Although swing compressor eliminates the vane tip leakage and vane chattering effect that exist in a rolling piston compressor, a thick and long vane is required to prevent the swing bushes from collapsing at fully extended position, which increases the size of the cylinder to house the vane.

2.2.6 Revolving Vane Compressor

Revolving vane compressor (RV) was a novel mechanism introduced by Teh and Ooi in 2006 [110, 111]. It comprises three main components, a cylinder, a rotor and a vane [21], as shown in Figure 2.8. During operation, the rotor and vane revolve, and subsequently rotate the cylinder. The rotation of cylinder causes the chambers formed by the rotor, vane and the cylinder to vary and thus, the suction and compression of the working fluid occur. A theoretical study has shown that the total frictional loss of a RV compressor is around 20 % less than the frictional loss of a rolling piston compressor and the mechanical efficiency of a RV compressor has achieved 94.1 %, which is above the mechanical efficiency of rolling piston and scroll types compressor [21]. In addition, the leakage loss at the radial clearance in a RV compressor is reported to be 40 % less than rolling piston compressor [112].

Figure 2.8 Schematic model of revolving vane compressor [21]

Figure 2.9 Schematic diagram and sectional view of improved revolving vane compressor [113]

There is also an improved RV compressor design namely improved revolving vane compressor (RV-I) [113], where the overall frictional loss of RV-I is reduced by 20 % [114] and the leakage loss is reduced by 40 % [115]. One of the shortcomings of RV-I is the thickness of the rotating cylinder wall, which is required to lodge up the slot for the sliding vane, resulting in high rotating moment of inertia during operation. It is observed that large volume is underutilised and results in the wastage of material in the cylinder.

2.3 Theoretical Model and Simulation Studies

A theoretical simulation model is a useful tool to predict a compressor performance before having the prototype fabricated. In this section, development of simulation models including thermodynamic model, valve reed response model, heat transfer model and leakage model will be presented.

2.3.1 Thermodynamic Model

Thermodynamic model describes the process and reveals the states of the working fluid inside a compressor working chamber, such as mass, temperature and pressure during the operation. In 1972, Soedel [116] demonstrated the working fluid behaviour in a positive displacement reciprocating compressor as a polytropic process and some empirical coefficients were obtained from experiments. Squarer and Kothmann [117] carried out a simplified computer simulation analysis of the reciprocating compressor using ideal gas behaviour, and the performance of compressor was well predicted. In the same year, Karll [118] described the cylinder process in a compressor using the state equation of First Law of Thermodynamics in a computer simulation, while Prakash and Singh [119] modelled the process with the assumption of ideal gas behaviour. In the study by Rottger and Kruse [120], real gas equation of state was incorporated in their analytical model and the prediction was found more favourable than using the ideal gas equation of state. The real gas properties were adopted by Hiller and Glicksman [121] in three different computer simulation models. The results were verified with experimental measurements and the discrepancy is acceptable.

Ng *et al.* [122] compared the simulation results of using both ideal gas and real gas equations of states. He found that real gas equation of state improved the accuracy of various parameters only at low superheating, at the compressor suction inlet or high compression ratio. Moreover, Lee *et al.* [123] inspected the thermodynamics parameters of a compressor process using both polytropic process and first law analysis model. They concluded that although both approaches well-matched with the given working chamber pressure benchmark, first law analysis predicted the working chamber temperature better than polytropic model. Ooi and Wong [124] carried out an analytical studies of rolling

piston compressor and compared the performance with measured results. The state of the refrigerant was described by real gas equation of state and First Law of Thermodynamics. It was reported that the predicted performance of the simulation model was within a discrepancy of 10 % compared to the experimental result.

In conclusion, first law analysis and real gas equation of state provides better accuracy in evaluating the performance of compressor. Recently, with the development of refrigerant database such as NIST REFPROP [125], the First Law of Thermodynamics analysis and the real gas equation of state are commonly applied in compressor studies.

2.3.2 Valve Dynamic Model

Generally, a valve is installed at the suction or (and) discharge ports to ensure unidirectional flow of the working fluid. The valve behaviour is complex and it greatly affects the compressor performance [126] as some of the compressor failures are caused by the valve failure, specifically the fatigue fracture [127]. Thus, a valve dynamic model is another essential model in a compressor simulation.

Generally, valve dynamic is demonstrated by a non-linear differential equation. The first meaningful step in modelling valve dynamic for reciprocating compressor was presented by Costagliola [128] since 1950 as stated in the review by MacLaren [129]. The model was declared to be fundamentally correct even though the results were not in good agreement with experimental results. Most of the valve models were established based on the analysis initiated by Costagliola [128] because an analytical model for valve performance could deliver more rapidly and less costly qualitative results as compared to an experimental approach [130-132]. However, the error existed between analytical and experimental results was examined and it was primarily caused by the variation of the empirical coefficients, i.e. the coefficients of discharge and damping. In fact, the aforementioned models are semi-analytical as those coefficients employed were evaluated from experiments.

In 1972, Gatecliff and Lady [133] conducted a mathematical analysis on a non-uniform width cantilever reed valve with forced vibration. Rayleigh-Ritz

method and Galerkin's method were adopted in their model and the outcome was in good agreement with the experimental result. Gatecliff *et al.* [134] also established an outline of the procedure that is used to design cantilever reed valves and provide approximate indication for the valve performance. Ooi *et al.* [135] studied a simple reed valve model on a small rolling piston compressor. A standard polynomial function was used to formulate the mode shape function and the calculated valve deflection was superimposed with the result from finite element analysis. As a conclusion from the authors, a thicker valve that results in small valve displacement may in turn causes high compression loss and indicated power due to high differential pressure across the valve. Prater *et al.* [136] measured the undamped natural frequency and damping ratio for a discharge valve by using optical measurement instrument and the measured data were successfully incorporated into computer simulation model. Lohn *et al.* [137] studied the effect of damping coefficient of a reed valve through experiment and they concluded that it has negligible effect on valve bending stress but it may affect compressor reliability.

On the other hand, Woolatt [138] calculated the lift and pressure drop with single degree of freedom approach for the self-actuating compressor valves numerically. Khalifa and Liu [139] presented an analytical model to study the dynamic behaviour of a round reed valve with the presence of oil, while Pizarro-Recabarren *et al.* [140] improved the model by taking the finite amount of oil volume between the valve and valve seat into consideration. Their models showed similar results when the valve was at static equilibrium condition, but significant difference was observed between the models when the oil volume decreased. With the rise of computational power, computational fluid dynamics (CFD) techniques were widely used as an approach to study valve dynamic performance. For example, Kim *et al.* [141] analysed the valve dynamic and optimised the valve geometry via CFD fluid-structure interaction (FSI) implementation. Tan *et al.* [142] also studied the effective flow and force area on the reed valve by FSI model. Furthermore, Ding and Gao [69] successfully conducted a full 3-D CFD model for a rolling piston compressor with reed valve.

Not limited to these, there are still many research studies with the application of CFD [143-145].

Obviously, many remarkable studies about analytical and numerical valve dynamic model were published, and more developments of valve modelling are progressing. In short, a complete valve dynamic model not only includes valve reed displacement behaviour, but also the damping effect, the effective area of force acting on the valve, fluid flow behaviour and others.

2.3.3 Leakage Model

Internal leakage is another hot topic for investigation as all positive displacement type compressors encountered unfavourable mass loss of working fluid during operation [146, 147]. The mass loss of working fluid decreases volumetric efficiency, which in turn reduces the compressor performance. Internal leakage occurs when there are clearances between two mating parts or contact points [148]. The characteristics of the leakage flow has been well studied in the past few decades.

Figure 2.11 Illustration of internal leakage paths in a rolling piston compressor [83]

Basically, the leakage flow was demonstrated as a pressure driven gas flow through a convergent-divergent nozzle. To simplify the complex gas flow phenomena, constant flow area, ideal gas properties and approximated upstream pressure are assumed when modelling the leakage flow [7, 149]. The calculated results were verified with the measured results on a hermetic rolling piston compressor and both results showed good agreement.

Reed and Hamilton [147] stated the unknown state of the leakage fluid and proposed a prediction method by analysing two extreme cases of the internal leakage to determine the upper and lower bounds of the leakage flow magnitude. They assumed that the fluid filling the clearance of the former was entirely compressible gaseous refrigerant while for the case of latter, the fluid was incompressible mixture of gaseous refrigerant and lubricating oil. As a result, the measured leakage mass flow rate was in between the upper and lower bounds and the mixture of lubricating oil and refrigerant reduced the leakage effectively. Rodgers and Nieter [149] also pointed out that the lubricating oil and refrigerant mixture leakage model using solubility data of refrigerant dissolved in lubricating oil is more appropriate to predict the leakage and since oil acts as an obstacle on the leakage path, adequate lubricating oil should be supplied in order to reduce leakage in the working chamber. Yanagisawa and Shimizu [79, 150] simulated the internal leakage model emphasised on radial clearance and end-face clearance leakage flows in a rolling piston compressor by considering the dynamic change of the radial clearance and fluid viscosity. Cai *et al.* [82, 83] compared experimental results with different mathematical models, i.e. single-phase model, two-phase isothermal model and two-phase non isothermal model. They concluded that an actual leakage process could not be fully explained by a single leakage flow model, the single-phase model predicted the result under low length-diameter ratio and high-pressure difference. On the other hand, the result of two-phase model has a better prediction under a converse working condition. In brief, though the leakage model appeared to have been well-established, many researchers are still working on the leakage model [151, 152].

2.3.4 Heat Transfer Model

When the working fluid is being compressed in the compressor, its pressure and temperature build up and cause heat transfer to occur. This heat transfer may affect the pressure of the working chamber and subsequently, the compressor performance [153, 154]. In a review by Prasad [155], the presence of high temperature in a compressor may affect the material properties, dimensional integrity and stability of the components, and ultimately reduce the reliability of a compressor. Chong and Watson [156] examined the heat and mass transfers of a low speed reciprocating compressor compression process and they showed that significant convective heat transfer was presented due to the vortex motion of the working fluid. Recktenwald *et al.* [157] investigated the heat transfer in a reciprocating compressor cylinder with numerical models and their prediction result was in good agreement with the published experimental result. In fact, several numerical techniques were used for studying heat transfer in a compressor [158, 159] since 1980s, but the methods were highly iterative which may be inefficient in terms of computational time.

Another approach for heat transfer modelling is to establish the instantaneous heat transfer coefficients of the heat transfer occurs between the compressor chambers. Several correlations regarding the compressor in-chamber convective heat transfer have been done and most of them were for reciprocating machineries. Annand *et al.* [160] studied the heat transfer in a reciprocating internal combustion engine, and they concluded that a convective heat transfer coefficient can be expressed by a function of Reynold Number only. Adair *et al.* [161] studied and examined the correlations of in-chamber convective heat transfer coefficient for both reciprocating compressor and engine. They came out with a new and more appropriate expression of correlation for the convective heat transfer coefficient which is within 20 % of the experimental result. Liu and Zhou [162] investigated the temperature distribution of reciprocating compressor and they applied different empirical constants in the developed correlation of the convective heat transfer coefficient, which is similar to Adair's correlation. In addition, the correlations were widely applied on future heat transfer modelling on different compressors. Todescat *et al.* [163] and Fagotti *et al.* [164] compared

the available correlations under different operating conditions in reciprocating compressor and they found that predicted result from Annand's correlation was well-matched with the experimental data. Ooi [165] studied the heat transfer in a hermetic reciprocating refrigeration compressor through lumped thermal conductance approach and the discrepancy between the simulated and measured results was less than 10 % by adopting the correlation by Adair *et al.*. In addition, Diniz *et al.* [166] performed the lumped-parameter thermal model to predict the temperature in different chambers and components in a scroll compressor. They found that the suction gas temperature was closely related with the oil temperature. In addition, it was concluded that the temperature profile of the wraps could not be obtained by a model which considered only the convective heat transfer in the working chamber. Tuhovcak *et al.* [167] compared the heat transfer effect on the compressor efficiency by using various correlations developed in the literature. They concluded that the heat transfer models resulted in significant difference in the isentropic efficiency, but the effect on the volumetric efficiency was found more consistent.

The heat transfer studies of rotary compressor were carried out by several researchers too. Liu and Soedel [168] presented a heat transfer model for a rolling piston compressor while Padhy and Dwivedi [169] extended the use of Adair's correlation on a hermetic rolling piston compressor. The theoretical result was in good agreement with the experimental measurement. In the development of a novel revolving vane compressor by Tan and Ooi [170], a discrepancy of the pressure-angle profiles of less than 2 % was obtained between the predicted and measured results with the consideration of convective heat transfer. In short, heat transfer model becomes an essential as to improve the accuracy of thermodynamic process of a compressor simulation [18, 171-174].

2.3.5 Other Relevant Literatures

The development of the U-Vane compressor includes not only the theoretical models reviewed in the previous sections, but also other aspects such as journal bearing and lubrication.

As the shaft of U-Vane compressor is rotating at high speed, to ensure a smooth operation, it must be supported by bearings to function normally. Journal

bearing, which includes only a journal (shaft), a bearing (plain bearing) and the lubricant, are the most widely used bearing type in compressors [175]. They are well-known for the features of low wear, low noise and good damping. Hirani *et al.* [176-179] presented a step-by-step analytical solution for designing the journal bearing under both static and dynamic operational conditions with the assumptions of infinitely short and infinitely long bearings. The oil film pressure is estimated from the method proposed by Reason and Narang [180] while the lubricant feed is approximated based on the method proposed by Martin and Lee [181]. The procedure proposed has shortened the computational time from 4090s to 19s. Furthermore, the work by Sharma and Awasthi [182] can be an additional information for the journal bearing design.

When the U-Vane compressor is operating, the rubbing components may generate undesirable frictional and heat losses which may affect the performance of the compressor. Thus, it must be well-lubricated by lubricating fluid (lubricant). The lubrication system design in the U-Vane compressor can be done by adopting the method proposed by Kim *et al.* [183, 184]. The lubricant circulation system was modelled as an equivalent electrical circuit diagram where the electrical current, resistance and voltage source represent lubricant flow rate, flow friction and lubricant pump respectively. Additionally, several studies related to compressor lubrication have also been carried out [185-189].

2.4 Concluding Remarks

In this chapter, some remarkable points are as follows:

- The energy consumption by AC&R system is increasing globally due to the growing number of AC&R systems in operation.
- The demand of compressor is growing together with the AC&R system and hence, researchers are focusing on a more energy-saving or cost-saving compressor technology.
- A compressor should contain fewer components, as well as simpler in geometries, which also means that a compressor should be light-weight, small-size and low production cost.

Currently, rotary compressors are the most widely used compressors due to its advantages in size, geometry, noise and vibration. The existing compressors can be further improved and optimized with more researches and investigations. In addition, a new compressor mechanism with better performance can be designed and studied. In this project, a new rotary compressor which is anticipated to be more cost-saving due to the lesser amount of material used in the compressor is introduced by the author. The detailed design will be presented in the Chapter 3.

Chapter 3 Introduction to U-Vane Compressor

Compressor

In this chapter, the conceptual design and the working principle of the U-Vane compressor will be presented.

3.1 Conceptual Design of U-Vane Compressor

From the literature review, the energy consumed by refrigeration system is increasing dramatically, which implicitly indicates the growing demand of compressor as it is one of the most important components in a refrigeration system [11]. It can be foreseen that substantial amount of materials will be used to produce compressors in the future. Hence, it is essential to have a compressor mechanism that is more material-saving. To achieve the objectives, a new compressor, which is called U-Vane compressor (or UVC), is conceptualised and designed by the author. It is designed with the aim to improve on the weaknesses of the existing compressors while retaining their strengths, i.e. reduces the housing size while retains the mechanical efficiency. The design process of UVC was taken its lead from the analysis of the rolling piston compressor.

Figure 3.1 Schematic of the rolling piston compressor and its major components

Let's first look at the existing rolling piston compressor, one of the most widely used rotary compressors, especially in household air-conditioning [11]. Figure 3.1 shows the rolling piston compressor with its major components. The use of vane-spring assembly ensures the vane tip to be always in contact with the roller to reduce the leakage across the chambers during operation. In fact, a trade-off between the spring force and the leakage is always presented in the vane-spring assembly. Having a high spring force magnitude results in high frictional loss and material wear at the vane tip and the outer roller surface, while low spring force magnitude leads to insufficient contact and leakage between the vane tip and the roller [72, 190] and ultimately, reduces the compressor performance. The vane-spring assembly is undeniably a critical player in the performance of rolling piston compressor. Henceforth, swing compressor is introduced [14, 106, 107], which replaces the vane-spring assembly by a pair of swing bushes, as shown in Figures 3.2 and 3.3.

In a swing compressor, the vane is rigidly attached to the rotor in order to eliminate the vane tip leakage and frictional loss existing in a rolling piston compressor. The working principle of a swing compressor is similar to that of the rolling piston compressor. The working chamber is formed by the space enclosed by the vane, the contact line between rotor and cylinder, the outer rotor surface and the inner cylinder surface. When the shaft rotates, the size of the volume of working chamber varies and thus results in suction, compression and discharge of the working fluid. The centre of the eccentric orbits about the shaft centre, however, the rotor in this case unlike that in the rolling piston compressor, its movement is constrained and guided by the swing bushes since the vane is attached rigidly to the rotor on one end and crosses over the swing bushes on the other end. As a result, the rotor has a combined movements of swivelling (about its owned centre-axis) and translating (up and down) during operation.

Figure 3.2 Schematic of the swing compressor and its major components

Figure 3.3 Schematics of (a) swing bushes in swing compressor and (b) vane-spring assembly in rolling piston compressor

Figure 3.4 Schematic of swing compressor with (a) long vane and (b) short vane

As revealed in Section 2.2.5, the vane of a swing compressor is required to be sufficiently long to prevent the swing bushes from collapsing as shown in Figure 3.4. This will close the moving path of the vane and the operation will eventually cease. Moreover, it increases the size of a swing compressor where the compressor cylinder and housing are required to be large to accommodate the long vane. Although the swing compressor eliminates the wear of vane tip in the rolling piston compressor, its unnecessarily large size of the cylinder and housing to accommodate the required length of the vane makes the design unattractive.

In order to overcome the aforesaid issues while maintaining the advantages of the compressor, a few modifications on the vane and the rotor are proposed, these include:

- Making the swing bushes (normally two pieces) into one single piece (which is also called vane bush) to prevent potential jam of the swing bushes (Figure 3.5).

Figure 3.5 Modifying (a) swing bushes into (b) one-piece vane bush

- The shoulders of the new bush are extended to house the vane (attached to the rotor) when the vane is at the fully extracted position. This allows a shorter vane to be employed (Figure 3.6).

Figure 3.6 The shoulders of vane bush are extended

- Two slots are cut at the sides of the vane to accommodate the extended shoulders from the vane bush as shown in Figure 3.7.

Figure 3.7 Two slots are cut for the extended shoulders

The finalised conceptual design of the compressor is shown in Figure 3.8, which is named “U-Vane Compressor” (UVC), credited to the use of one of its components – the new vane bush which is “U” like-shaped. With these modifications, UVC can be made more compact as a shorter vane can be used and a compartment to accommodate the long vane is not required as illustrated in Figure 3.9. Thus, it will be lighter in overall weight. The detail design of the UVC, its working principle and the comparison between UVC and swing compressor [108] are presented in the following sections.

Figure 3.8 Conceptual design of U-Vane compressor (UVC)

Figure 3.9 Comparison between swing compressor [108] and UVC

3.2 The Main Components and the Assembly

Figure 3.10 shows the main components of UVC, which are: 1) cylinder, 2) rotor, 3) vane bush, 4) shaft and eccentric. A bush slot is designed on the cylinder wall to hold the vane bush, which acts as a hinge joint, in position while the bush is swivelling during the operation. The vane is one end fixedly attached to the rotor while the other end inserted into the vane bush. The shoulders of the bush will go into the two slots designed at the sides of the vane when the vane is at the retracted position.

Figure 3.10 Main components of UVC: (a) cylinder, (b) rotor, (c) vane bush, (d) shaft and eccentric

Figure 3.11 Top sectional view of the assembly of UVC main components

Figure 3.11 depicts the assembly of the main components of UVC. The shaft and the cylinder are assembled concentrically. The centre of the shaft and cylinder is located at “C”, while “R” is the centre of the eccentric and the rotor. The distance between “C” and “R” is termed eccentricity, which is denoted by “e”. A contact line represented by “A” is formed between the inner cylinder surface and outer rotor surface after the components are assembled. The region

enclosed by this contact line “A”, the inner cylinder surface, the outer rotor surface, the vane and the vane bush form the working chamber. The working chamber that is communicated with the suction port is referred as the suction chamber, while the one which is being compressed is termed the compression chamber. The compression chamber will communicate with the discharge port once the pressure of the working fluid in compression chamber rises and sufficiently high to open the discharge valve. During the operation, a prime mover, which is an electric motor, rotates the shaft, which in turn causes the eccentric to roll along the inner rotor surface. As a result, the rotor swings and causes the variation of working chamber volume. Subsequently, the changes in the size of the chamber volume causes the working fluid to be induced into the working chamber, compressed and discharged out from the working chamber through the discharge valve, and completes the compressor cycle. Also, a reed is mounted at the discharge port, in such a way that it only opens when the pressure in the working chamber is sufficiently high to open it. In addition, the reed prevents the flow of the discharged fluid into the working chamber, when the working chamber pressure is at the pressure lower than the discharge plenum pressure. The valve reed response model is presented in Chapter 4.

A few configurations about the suction and discharge ports’ positions are available by considering their operation and nearness to the respective working chamber. Although the positions of suction and discharge ports in current design are located at the cylinder wall, they can be located at the upper end-face cover too. One of the disadvantages of locating the port on the end-face is that it allows the communication between the ports with the working chamber and leads to the significant ‘internal leakage’, which is undesirable. Figure 3.12 illustrates the conceptual design of U-Vane compressor, where the upper bearing acts as the cover of the working chamber and the lower bearing acts as the base of the working chamber.

Figure 3.12 Side sectional view of UVC conceptual design

3.3 The Working Principle

The working principle of U-Vane compressor is described in Figure 3.13.

Figure 3.13 Snapshots of the operation of U-Vane compressor

The compressor cycle starts from Figure 3.13a, which indicates the end of the suction process and the commencement of the compression process, where the maximum amount of working fluid is trapped in the working chamber. When the shaft rotates, the rotor swings, causing the volume of the working chamber to reduce and the working fluid pressure to rise. Meanwhile, the low pressure working fluid starts to be drawn into the working chamber through the suction port when the suction chamber volume increases, as shown in Figures 3.13b and c. Both suction and compression processes occur concurrently in separate chambers during the rotation. The reed valve remains closed during the compression process, only until the fluid pressure in the compression chamber overcomes the pressure on the other side of the valve reed, as well as the valve reed stiffness, the reed starts to open and this marks the beginning of the

discharge process, as illustrated in Figures 3.13c and d. At the same time, the suction process occurs at the suction chamber continues uninterruptedly, until the rotor reaches the position as shown in Figure 3.13a where the suction process ends and the cycle repeats.

During operation, the vane slides in its slot in the bush, while the shoulders are accommodated in the corresponding slots in the rotor. Furthermore, as the bush body is cylindrical and housed in the cylinder wall, it swivels about the virtual centre “O” only. Thus, the bush guides the motion of vane and hence the rotor.

3.4 Comparison between Swing Compressor and UVC

In a swing compressor [106], the two-piece split bushes may collapse if the vane is short. With the proposed new vane bush, a short vane can be used, which in turn reduces the overall weight and size of the compressor.

Figure 3.14 Size comparison between (a) swing compressor [108] and (b) UVC

Figure 3.14 demonstrates the size comparison between U-Vane compressor and an actual swing compressor [108]. Both compressors are designed with the same sizes of rotors, cylinders and eccentrics. It is noticeable that by replacing the swing bushes in a swing compressor with a new single-piece vane bush, the size of the cylinder and housing can be shrunk, and the overall mass of the compressor can be reduced. An estimation on the volumes of the cylinder and housing of the compressors is presented in Appendix A. It is anticipated that the

size of the cylinder and housing of UVC will be 30 % smaller than that of the swing compressor.

The comparison between the two compressors is shown in Table 3.1.

Table 3.1 Comparison between swing compressor and U-Vane compressor

	Swing Compressor [106]	U-Vane Compressor
Size and weight	Cylinder must contain a compartment to allow the fully protruded vane.	Cylinder is modified to hold the vane bush. This reduces the cylinder thickness and the weight saving of up to 30% is expected due to the reduced size of cylinder and housing.
Number of major components	4; Cylinder, rotor, shaft and eccentric, swing bushes	4; Cylinder, rotor, shaft and eccentric, vane bush
Component geometry	The fully retracted vane is housed in the cylinder wall only.	The fully retracted vane is partly housed in the cylinder wall and rotor.
Rotor	A vane is attached to the rotor.	A vane is attached to the rotor and provision is made on the rotor to partly house the shoulders of the bush.
Bush	A split bush made in a pair and may not be able to accommodate shorter vane.	A single-piece vane bush which accommodates only part of the fully retracted vane.
Vane thickness	Thicker is needed as longer vane is required.	Shorter vane is made possible and hence thinner vane can be used.

3.5 Concluding Remarks

The conceptual design of U-Vane compressor is presented in this chapter. The details of the major components and the working principle are shown and described. A comparison between swing compressor and U-Vane compressor is also summarised. Though the U-Vane compressor eliminates some of the problems in the existing compressor mechanism, it is not without its downsides.

More investigations about the compressor will be made and discussed. In next chapter, theoretical model regarding the U-Vane compressor is formulated.

Chapter 4 Mathematical Model I: Geometrics,

Kinematics Thermodynamics and Valve Reed

Dynamics

The performance of a compressor can be predicted theoretically by solving a series of differential equations which describe the operational characteristics of the compressor. These equations are known as the mathematical model of the compressor. With the increase in computational power, theoretical analysis of compressor in predicting compressor performance can be carried out more accurately, and the time and cost of developing a new compressor can be reduced accordingly.

In this chapter, the mathematical model of U-Vane compressor (UVC), including: (1) geometrical model, which deals with the variation of the working chamber volume; (2) kinematics model, which describes the velocity and acceleration behaviours of the compressor components; (3) thermodynamic model, which demonstrates the variation of the working fluid thermodynamics properties due to the varying working chamber volume, the inflow and outflow of working fluid, as well as the interactions with work and heat; (4) suction and discharge flows model, which expresses the mass flow rate through the suction and discharge ports, and (5) valve reed dynamics, which simulates the valve reed response to the pressure forces, will be presented.

4.1 Assumptions of the Mathematical Model

Prior to formulating the mathematical model of UVC, the following assumptions are adopted:

- The working fluid properties are homogeneous.
- The operation of the compressor is in steady state.
- The processes in the working chamber control volume are in quasi-steady state and the working fluid properties propagate instantaneously.

- One-directional flows through suction and discharge ports.
- Shaft angular velocity is constant during the operation although it may vary slightly in actual compressor operation.
- The lubricant properties, coefficient of friction and coefficient of discharge remain constant throughout the operational cycle.

4.2 Geometrical Model

The geometrical model expresses the variation of working chamber volume during the compressor operation. In this section, the mathematical expression of the working chamber volume with respect to the shaft rotational angle and time is formulated.

Figure 4.1 shows the schematic of U-Vane compressor, where the descriptions of the symbols used are listed in Table 4.1. A schematic diagram of the geometric relations of U-Vane compressor is presented in Figure 4.2.

Figure 4.1 Schematic diagram of U-Vane compressor (top view)

Table 4.1 Geometrical representative of U-Vane compressor

Nomenclature	Description
Point O	Centre axis of vane bush
Point C	Shared centre axis of shaft and cylinder
Point R	Shared centre axis of eccentric and rotor
Point A	Contact point (line) between outer rotor surface and inner cylinder surface
Point B	Intersection point between the circumference of the outer rotor surface and the longitudinal centre axis of the vane
l_v	Vane exposure length in the working chamber (m)
r_{vb}	Radius of the vane bush (m)
r_{ro}	Radius of the rotor (m)
r_{cyl}	Radius of the cylinder (m)
r_s	Radius of the shaft (m)
e	$= r_{cyl} - r_{ro}$, eccentricity of the rotor with respect to the cylinder (m)
θ_1	Rotational angular position (angle) of the shaft at centre axis C (rad), with respect to line OC and line CA
θ_2	Rotational angular position (angle) of the eccentric at centre axis R (rad), with respect to line OR and line RA
γ_1	Angle subtended by the centre axis C and the centre axis R at point O (rad), which is also the rotational angular position of vane bush

$$l_v = -(r_{ro} - r_{vb}) + \sqrt{e^2 + (r_{cyl} + r_{vb})^2 - 2e(r_{cyl} + r_{vb})\cos\theta_1} \quad (4.1)$$

$$\gamma_1 = \cos^{-1} \left[\frac{(r_{cyl} + r_{vb})^2 - e^2 + (r_{vb} + l_v + r_{ro})^2}{2(r_{cyl} + r_{vb})(r_{vb} + l_v + r_{ro})} \right] \quad (4.3)$$

$$V_{suc} = \frac{1}{2} l_{com} \left[r_{cyl}^2 \theta_1 - r_{ro}^2 \theta_2 - e(r_{cyl} + r_{vb}) \sin \theta_1 - l_v w_{vro} + r_{vb}^2 \gamma_1 + \frac{1}{4} w_{vro}^2 \tan \gamma_1 \right] \quad (4.4)$$

$$V_{com} = \pi l_{com} (r_{cyl}^2 - r_{ro}^2) - \frac{1}{2} l_{com} \left[r_{cyl}^2 \theta_1 - r_{ro}^2 \theta_2 - e(r_{cyl} + r_{vb}) \sin \theta_1 - l_v w_{vro} + r_{vb}^2 \gamma_1 + \frac{1}{4} w_{vro}^2 \tan \gamma_1 \right] \quad (4.5)$$

where,

V_{suc} is the volume of the suction chamber (m³)

V_{com} is the volume of the compression chamber (m³)

l_{com} is the axial length of the compressor (m)

w_{vro} is the width of the vane (m)

The volumes of the shoulder slots at the rotor connected to the working chamber are not formulated into Equations (4.4) and (4.5) as they are neutralised with the volume occupied by the bush shoulders. To ensure the comprehensiveness of the geometrical model and consider the vane width, the last terms of the right-hand side of both Equations (4.4) and (4.5) are added, which represent the volume intercepted by the cylinder when the vane is tilted from line OC . With reference to Figure 4.3, the volume is derived by approximating the intercepted area as a triangle.

Figure 4.3 Schematic diagram of the volume at the intersection of vane and cylinder

Finally, due to the dead volume caused by the presence of gaps and clearances when the vane is fully retracted in the slot, constant terms are added into Equations (4.4) and (4.5) to account for the dead volume.

4.3 Kinematic Model

The kinematic model includes the rate of change of the vane exposure length ($\frac{dl_v}{dt}$), the rate of change of the rotor angular position (angle θ_2) and the rate of change of the angular position of vane bush (angle γ_1). It is noted that the model is developed with the assumption of negligible effect of the working fluid resistance. The geometrical shapes are seen as rigid body which the deformation of the part can be neglected.

4.3.1 Sliding Velocity and Acceleration of Vane

By differentiating Equation (4.1) with respect to time, t , the rate of change of vane exposure length in the working chamber, which is also known as the sliding velocity of the vane, can be obtained as Equation (4.6):

$$\begin{aligned} \frac{dl_v}{dt} &= \frac{dl_v}{d\theta_1} \cdot \frac{d\theta_1}{dt} \\ &= \frac{e(r_{vb} + r_{cyl}) \sin \theta_1}{\sqrt{e^2 + (r_{vb} + r_{cyl}) - 2e(r_{vb} + r_{cyl}) \cos \theta_1}} \omega_1 \end{aligned} \quad (4.6)$$

and the sliding acceleration of the exposed vane is derived below:

$$\begin{aligned} \frac{d^2 l_v}{dt^2} &= \frac{\alpha_1 e(r_{vb} + r_{cyl}) \sin \theta_1}{\sqrt{e^2 + (r_{vb} + r_{cyl}) - 2e(r_{vb} + r_{cyl}) \cos \theta_1}} + \omega_1^2 e(r_{vb} + r_{cyl}) \\ &\quad \cdot \left\{ \frac{\cos \theta_1}{\sqrt{e^2 + (r_{vb} + r_{cyl}) - 2e(r_{vb} + r_{cyl}) \cos \theta_1}} \right. \\ &\quad \left. - \frac{e(r_{vb} + r_{cyl}) \sin^2 \theta_1}{[e^2 + (r_{vb} + r_{cyl}) - 2e(r_{vb} + r_{cyl}) \cos \theta_1]^{1.5}} \right\} \end{aligned} \quad (4.7)$$

where,

t is time (s)

$\frac{dl_v}{dt}$ is the rate of change of the vane exposure length or sliding velocity of the

vane (m/s)

$\frac{d^2l_v}{dt^2}$ is the sliding acceleration of the vane (m/s²)

$\omega_1 = \frac{d\theta_1}{dt}$ is the angular velocity of the shaft, or angular velocity of the rotor

centre axis “R” with respect to shaft centre axis “C” (rad/s)

$\alpha_1 = \frac{d\omega_1}{dt}$ is the angular acceleration of the shaft, or angular acceleration of the

centre axis “R” with respect to shaft centre axis “C” (rad/s²)

4.3.2 Kinematics of Rotor

The angular velocity of the rotor about its own axis can be found by differentiating Equation (4.2) with respect to time and combining with Equation (4.6), the following equation is formulated:

$$\begin{aligned} \frac{d\theta_2}{dt} &= \frac{d\theta_2}{dl_v} \cdot \frac{dl_v}{d\theta_1} \cdot \frac{d\theta_1}{dt} \\ &= \frac{1}{\sqrt{1 - \left[\frac{(r_{vb} + r_{cyl})^2 - (r_{vb} + l_v + r_{ro})^2 - e^2}{2e(r_{vb} + l_v + r_{ro})} \right]^2}} \\ &\quad \cdot \left[\frac{1}{e} + \frac{(r_{vb} + r_{cyl})^2 - (r_{vb} + l_v + r_{ro})^2 - e^2}{2e(r_{vb} + l_v + r_{ro})^2} \right] \\ &\quad \cdot \frac{dl_v}{d\theta_1} \cdot \omega_1 \end{aligned} \quad (4.8)$$

4.3.3 Kinematics of Vane Bush

The angular velocity of vane bush is the rate of change of the angle γ_1 (Figure 4.2). Similarly, by differentiating Equation (4.3) with respect to time and combining with Equation (4.6), the rate of change of the angle γ_1 can be obtained as shown in Equation (4.9):

$$\begin{aligned}
\frac{d\gamma_1}{dt} &= \frac{d\gamma_1}{dl_v} \cdot \frac{dl_v}{d\theta_1} \cdot \frac{d\theta_1}{dt} \\
&= \frac{1}{\sqrt{1 - \left[\frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})} \right]^2}} \\
&\quad \cdot \left[\frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})^2} - \frac{1}{r_{vb} + r_{cyl}} \right] \\
&\quad \cdot \frac{dl_v}{d\theta_1} \cdot \omega_1
\end{aligned} \tag{4.9}$$

Moreover, the angular acceleration ($\frac{d^2\gamma_1}{dt^2}$) of vane bush is derived and shown in Appendix B.

4.4 Thermodynamic Model

During operation, the compressor experiences suction, compression and discharge processes. These processes are driven dominantly by the variation of the working chamber volume. Consequently, the thermodynamics properties of the working fluid in the chamber vary and these happenings can be described mathematically in thermodynamic model.

Based on the First Law of Thermodynamics, with the assumption that the working fluid properties are disseminated instantaneously, or in other words, uniform properties and no spatial variation throughout the working chamber, Moran and Shapiro [191] presented the following equation for an energy rate balance of a control volume:

$$\begin{aligned}
\frac{dE_{cv}}{dt} &= \dot{Q}_{cv} - \dot{W}_{cv} + \sum_i \dot{m}_i \left(h_i + \frac{v_i^2}{2} + gz_i \right) \\
&\quad - \sum_o \dot{m}_o \left(h_o + \frac{v_o^2}{2} + gz_o \right)
\end{aligned} \tag{4.10}$$

where,

E_{cv} is the total energy of the control volume (J)

$\dot{Q}_{cv}$ is the rate of net heat transfer into the control volume (J)

$\dot{W}_{cv}$ is the rate of net work done by the control volume (J)

$\dot{m}_i$ is the mass flow rate of working fluid flowing into the control volume (kg)

$\dot{m}_o$ is the mass flow rate of working fluid flowing out from the control volume (kg)

h_i is the specific enthalpy of the working fluid flowing into the control volume (J/kg)

h_o is the specific enthalpy of the working fluid flowing out from the control volume (J/kg)

v_i is the velocity of the working fluid flowing into the control volume (m/s)

v_o is the velocity of the working fluid flowing out from the control volume (m/s)

z_i is the vertical height of the working fluid flowing into the control volume (m/s)

z_o is the vertical height of the working fluid flowing out from the control volume (m/s)

g is the acceleration of gravity (m/s^2)

The second term of the right-hand side of Equation (4.10) represents the rate of change of work done due to the change in the size of working chamber volume, and they can be rewritten as follows:

$$\dot{W}_{cv} = p_{cv} \frac{dV_{cv}}{dt} \quad (4.11)$$

where,

p_{cv} is the pressure of the control volume (Pa)

V_{cv} is the volume of the control volume (m^3)

Also, the total control volume energy E_{cv} is the summation of internal energy, kinetic energy and potential energy, which can be presented by:

$$E_{cv} = m_{cv} \left(u_{cv} + \frac{v_{cv}^2}{2} + gz_{cv} \right) \quad (4.12)$$

where,

m_{cv} is the mass of the control volume (kg)

u_{cv} is the specific internal energy of the control volume (J/kg)

v_{cv} is the velocity of the control volume (m/s)

z_{cv} is the vertical height of the control volume (m)

Combining Equation (4.10), Equations (4.11) and (4.12), the following equation can be obtained

$$\begin{aligned} & \frac{d \left[m_{cv} \left(u_{cv} + \frac{v_{cv}^2}{2} + gz_{cv} \right) \right]}{dt} \\ &= \dot{Q}_{cv} - p_{cv} \frac{dV_{cv}}{dt} \\ &+ \sum_i \dot{m}_i \left(h_i + \frac{v_i^2}{2} + gz_i \right) - \sum_o \dot{m}_o \left(h_o + \frac{v_o^2}{2} + gz_o \right) \end{aligned} \quad (4.13)$$

The kinetic and potential energies are relatively small compared to the enthalpy. For example, consider a compressor with volumetric displacement of 32.5 cc, operates at 3000 RPM with air as the working fluid. The suction and discharge port diameters are 15 mm and 10 mm respectively. By compressing the air to discharge pressure of 2 bar, the change in the specific kinetic energies (KE) between inlet and outlet is less than 1% of the change in the enthalpy [125]. In addition, the change in the elevation of the control volume inlets and outlets are also small compared to the enthalpy. Hence, both kinetic and potential terms can be neglected. Rewriting Equation (4.13) and applying the product rule, it can be simplified as below:

$$m_{cv} \frac{du_{cv}}{dt} + u_{cv} \frac{dm_{cv}}{dt} = \dot{Q}_{cv} - p_{cv} \frac{dV_{cv}}{dt} + \sum_i \dot{m}_i h_i - \sum_o \dot{m}_o h_o \quad (4.14)$$

Applying the Law of Conservation of Mass [192], the net rate of change of mass in the control volume is expressed as:

$$\frac{dm_{cv}}{dt} = \sum_i \dot{m}_i - \sum_o \dot{m}_o \quad (4.15)$$

In Equations (4.14) and (4.15), the possible leakages and re-expansion mass flow due to the presences of clearances and dead volume in the working chamber are included and indicated by subscripts i and o . As two working fluid properties are required to know the state of the working fluid (gas), another equation related to the specific property, says the working fluid density, is derived as below:

$$\rho = \frac{m_{cv}}{V_{cv}} \quad (4.16)$$

$$\therefore \frac{d\rho}{dt} = \frac{m_{cv}}{V_{cv}^2} \frac{dV_{cv}}{dt} + \frac{1}{V_{cv}} \frac{dm_{cv}}{dt}$$

In U-Vane compressor, Equations (4.14) - (4.16) are applied to both suction and compression chambers, respectively. During operation, the suction chamber volume would increase and gain energy from the inlet flow and leakage flow from the high pressure working fluid at the compression chamber, while the compression chamber volume would decrease and lose energy through leakage and outlet flow. By solving Equations (4.14) - (4.16) using 4th Runge-Kutta numerical integration method [193], the instantaneous internal energy and density of the working fluid in each of the chambers can be obtained.

It is noted that the above equations are limited in its application on simple compressible system where the inlet and outlet flows are uniform, and the work done by other forces such as surface tension, shear and viscous friction of the working fluid are treated as insignificant when compared to the other terms in the energy balance of the control volume.

4.5 Suction and Discharge Flows Model

The mass flow stated in Equation (4.15) can be modelled as one-dimensional flow through orifice [194].

Figure 4.4 Schematic diagram of flow through orifice

As depicted in Figure 4.4, a volume of fluid is flowing through the orifice under steady and compressible conditions, as expressed by the streamlines. The one-dimensional energy equation involves in this situation can be written as:

$$Q + h_1 + \frac{v_1^2}{2} + gz_1 = h_2 + \frac{v_2^2}{2} + gz_2 \quad (4.17)$$

where,

Q is the specific heat transfer between the working fluid and surroundings (J/kg)

v_1 is the velocity of the fluid at upstream of orifice (m/s)

v_2 is the velocity of the fluid at downstream of orifice (m/s)

h_1 is the specific enthalpy of the fluid at upstream of orifice (J/kg)

h_2 is the specific enthalpy of the fluid at downstream of orifice (J/kg)

z_1 is the elevation of the fluid at upstream of the orifice (m)

z_2 is the elevation of the fluid at downstream of the orifice (m)

By assuming steady and adiabatic flow, and neglecting the initial velocity, as well as the change in elevation, Equation (4.17) can be simplified and expressed by:

$$\begin{aligned} h_1 &= h_2 + \frac{v_2^2}{2} \\ v_2 &= \sqrt{2(h_1 - h_2)} \end{aligned} \quad (4.18)$$

To account for flow loss, a flow coefficient (K) is introduced and the mass flow rate can be formulated:

$$\frac{dm}{dt} = K \rho_1 A_{flow} \sqrt{2(h_1 - h_2)} \quad (4.19)$$

Based on the conservation of mass, the flow may transform a portion of its enthalpy to increase the kinetic energy when passing through the opening of orifice which is smaller than the pipe size. Ideally, under the conditions of adiabatic and reversible, the fluid flow will back to its initial state after passing through the opening. This type of flow is isentropic flow, which can never be achieved as dissipative effects and friction are always presented within the flow. Therefore, coefficient of discharge C_d is introduced, which accounts for the flow coefficient (K) and isentropic efficiency (η_s) [170]. The definition of isentropic efficiency and the expression of actual mass flow rate are shown below:

$$\eta_s = \frac{h_1 - h_2}{h_1 - h_{2s}} \quad (4.20)$$

$$\frac{dm}{dt} = C_d \rho_1 A_{flow} \sqrt{2(h_1 - h_{2s})} \quad (4.21)$$

where,

$\frac{dm}{dt}$ is the mass flow rate through suction or discharge ports (kg/s)

C_d is the coefficient of discharge (-)

ρ_1 is the density of the fluid at the upstream of orifice (kg/m³)

A_{flow} is the opening area of suction and discharge port (m²)

η_s is the isentropic efficiency (-)

h_{2s} is the isentropic specific enthalpy of the fluid at downstream of orifice (J/kg)

It is noted that for high Reynold Number and sharp-edged orifices, the coefficient of discharge C_d is ranged from 0.5 to 0.6 [195, 196].

4.5.1 Orifice Flow Areas

In U-Vane compressor, the port opening areas (A_{flow}) for both suction and discharge ports are not always fully opened during operation. As the ports are radially positioned on the cylinder wall, the port opening is varying as a result of the moving radial contact line between the rotor and the cylinder, as shown in Figure 4.5.

Figure 4.5 Variation of suction port opening area (orifice flow area)

In the beginning of the suction cycle, the suction port is communicating with the compression chamber, as the radial contact line between the rotor and the cylinder has not entered the suction port opening yet. The suction port is partially opened when the contact line is within the suction port, which is within the angles $\theta_{suc,start}$ and $\theta_{suc,end}$. Consequently, it is fully opened after the radial contact line passes through the suction port (after $\theta_{suc,end}$) and the suction port is completely connected to the suction chamber.

As depicted in Figure 4.5, the mathematical expression of the suction port opening area is shown in the equations below:

$$A_{flow,suc} = 0, \quad 0 \leq \theta_1 < \theta_{suc,start} \quad (4.22)$$

$$A_{flow,suc} = r_{suc,port}^2 \left[\theta_{suc,port} - \frac{1}{2} \sin(2\theta_{suc,port}) \right], \theta_{suc,start} \leq \theta_1 < \theta_{suc,end} \quad (4.23)$$

$$A_{flow,suc} = \pi r_{suc,port}^2, \quad \theta_{suc,end} \leq \theta_1 \quad (4.24)$$

where,

$$\theta_{suc,port} = \cos^{-1} \left(\frac{r_{cyl} \sin \theta_1}{r_{suc,port}} \right) \quad (4.25)$$

$r_{suc,port}$ is the radius of the suction port (m)

Similarly, the discharge port opening area will also be affected by the transition of the radial contact line between the rotor and the cylinder, as shown in Figure 4.6. The mathematical expression is presented as following:

Figure 4.6 Variation of discharge port opening area

$$A_{flow,disc} = \pi r_{disc,port}^2, \quad 0 \leq \theta_1 < \theta_{disc,start} \quad (4.26)$$

$$A_{flow,disc} = r_{disc,port}^2 \left[\theta_{disc,port} - \frac{1}{2} \sin(2\theta_{disc,port}) \right], \quad \theta_{disc,start} \leq \theta_1 < \theta_{disc,end} \quad (4.27)$$

$$A_{flow,disc} = 0, \quad \theta_{disc,end} \leq \theta_1 \quad (4.28)$$

where,

$$\theta_{disc,port} = \cos^{-1} \left(\frac{r_{cyl} \sin \theta_1}{r_{disc,port}} \right) \quad (4.29)$$

$r_{disc,port}$ is the radius of the discharge port (m)

The offsets and curvatures of both suction and discharge ports are assumed negligible. In addition, the discharge port is covered by the valve reed, the lifting of valve reed must be considered to obtain the effective discharge flow area ($A_{flow,valve}$). As shown in Figure 4.7, assuming the curvature of the valve deflection is negligible, the fluid flow area can be formulated as:

$$A_{flow,valve} = 2\pi r y_{v,mid} \quad (4.30)$$

where,

$y_{v,mid}$ is the displacement of the valve at the centre of the discharge port (m)

Figure 4.7 Fluid flow area when the valve lifts

As a result, the effective discharge flow area is determined by comparing the values of the port opening area and the valve opening area. The smaller value of the two will be the effective discharge flow area in the orifice model.

4.6 Valve Reed Dynamics

Generally, the valve used in a compressor is an automatic reed valve, which consists of a valve reed and a valve stop plate. During the discharge process, the valve reed opens under sufficient pressure force, which is exerted by the working fluid to overcome the valve reed stiffness and the pre-set pressure at the other side of the valve reed. The valve stop plate, which is placed at a designated distance above the valve reed, constrains the maximum valve displacement to prevent the valve reed from over bending. The dynamic of valve reed is one of the crucial factors during the discharge process as it affects the flow area of the discharging working fluid. In U-Vane compressor, the reed valve is stationary and has the geometry as shown in Figure 4.8, thus it can be modelled with reference to the analysis carried out by Ooi *et al.* [135].

Figure 4.8 Schematic diagram of geometry of the valve reed

The geometry of the valve reed is simplified for ease of analysing, where the free end (x_1) is assumed as a circular disc and the other end is connected to a rectangular valve plate (x_2). Fundamentally, the valve dynamic is portrayed and modelled as the vibration of a beam with varying width, negligible shear and rotary motion. The equation describing the beam vibration is expressed below [197]:

$$\frac{\partial^2}{\partial x^2} \left[E_{dv} I_{dv}(x) \frac{\partial^2 y}{\partial x^2} + \rho_{dv} A(x) \frac{\partial^2 y}{\partial t^2} \right] = P(x, t) \quad (4.31)$$

The valve deflection (y) is given by:

$$y = \sum_{n=1}^{\infty} \phi_n(x) g_n(t) \quad (4.32)$$

where,

$\phi_n(x)$ is the mode shape function which is determined from the free vibration analysis (-)

$g_n(t)$ is the normalised coordinate or mode participation factor which can be obtained from the valve vibration governing equation (m)

There are infinite number of vibration mode shapes ($n \rightarrow \infty$), but due to the valve flow area, it is assumed that the valve will only experience the 1st mode of vibration. Hence, the governing equation of valve vibration used for solving the mode participation factor is expressed as following:

$$\ddot{g}_1 + 2\xi\omega_n\dot{g}_1 + \omega_n^2 g_1 = \frac{\int \phi_1(x) P dx}{\int \rho_v A_v(x) \phi_1^2(x) dx} \quad (4.33)$$

where,

$$\xi = \frac{C}{2\rho_{dv} A_{dv} \omega_n} \quad (4.34)$$

The valve reed natural frequency (ω_n) is approximated from Rayleigh's quotient [198]. With reference to Figure 4.8, for a valve reed with constant thickness (t_{dv}) and varying width (w_{dv}), the Rayleigh's quotient that describing the valve reed natural frequency are written as Equation (4.35):

$$\omega_n^2 = \frac{E_{dv} \left[\int_0^{x_1} I_{dv,x_1}(x) \left(\frac{d^2 \phi(x)}{dx^2} \right)^2 dx + I_{dv,x_2} \int_{x_1}^{x_2} \left(\frac{d^2 \phi(x)}{dx^2} \right)^2 dx \right]}{\rho_{dv} \left[\int_0^{x_1} A_{dv,x_1}(x) \phi^2(x) dx + A_{dv,x_2} \int_{x_1}^{x_2} \phi^2(x) dx \right]} \quad (4.35)$$

Meanwhile, the values of $I_{dv}(x)$ and $A_{dv}(x)$ are derived based on the geometry of the valve reed:

$$\left. \begin{aligned} A_{dv,x_1}(x) &= 2t_{dv} \sqrt{2rx - x^2} \\ I_{dv,x_1}(x) &= \frac{t_{dv}^3}{6} \sqrt{2rx - x^2} \end{aligned} \right\} 0 < x \leq x_1$$

$$\left. \begin{aligned} A_{dv,x_2}(x) &= w_{dv,x_2} t_{dv} \\ I_{dv,x_2}(x) &= \frac{w_{dv,x_2}^3 t_{dv}^3}{12} \end{aligned} \right\} x_1 < x \leq x_2$$
(4.36)

In Equation (4.33), the numerator of the term at right hand side is the working fluid pressure force acting on the valve reed. It can be simply expressed by the product of the pressure difference across the valve reed and the effective force area. However, in actual case, the effective force area is a function of the valve deflection, it changes when the valve is lifted from the valve seat. Thus, to

simplify the effective force area, the discharge port area is considered with a coefficient, which is expressed as:

$$\int \phi_1(x) P dx = C_{dv,eff} \phi_1(x) A_{force} P(t) \quad (4.37)$$

Hence, substituting Equation (4.37) into Equation (4.33) yields:

$$\ddot{g}_1 + 2\xi\omega_n\dot{g}_1 + \omega_n^2 g_1 = \frac{C_{dv,eff}\phi_1(x)A_{force}P(t)}{\int \rho_{dv}A_{dv}(x)\phi_1^2(x)dx} \quad (4.38)$$

where,

ξ is the damping ratio, which is the ratio of actual damping coefficient to critical damping coefficient (-)

C is the damping coefficient (-)

ρ_{dv} is the density of the valve material (kg/m³)

A_{dv} is the valve reed cross-sectional area (m²)

I_{dv} is the sectional moment of inertia (m⁴)

E_{dv} is the Young's modulus (Pa)

$C_{dv,eff}$ is the effective force area coefficient, which has to be obtained from experiment (-)

A_{force} is the force area on the valve reed (m²)

$P(t)$ is the pressure difference across the valve reed, which is a function of time (Pa)

Ooi *et al.* [135] has analysed the first mode of the valve deflection mode shape function ($\phi_1(x)$) by using finite element model (FEM). Two mode shapes of the valve reed with different boundary conditions are considered, which are the first case: when the valve just leaves the valve seat, and the second case: when the valve hit the valve stop plate. The mode shape functions can be expressed by the following polynomial functions.

For the first case,

$$\phi_1(x) = \left(\frac{x}{l_{dv,eff}}\right)^4 - 4\left(\frac{x}{l_{dv,eff}}\right) + 3 \quad (4.39)$$

and the corresponding satisfied boundary conditions are:

$$\frac{d^2\phi(0)}{dx^2} = 0, \frac{d^3\phi(0)}{dx^3} = 0 \quad \text{at } x=0, \text{ and } \phi(l_{dv,eff}) = 0, \frac{d\phi(l_{dv,eff})}{dx} = 0 \quad \text{at } x=l_{dv,eff}.$$

For the second case,

$$\phi_1(x) = \left(\frac{x}{l_{dv,eff}}\right)^4 - \frac{3}{2}\left(\frac{x}{l_{dv,eff}}\right)^3 + \frac{1}{2}\left(\frac{x}{l_{dv,eff}}\right) \quad (4.40)$$

with boundary conditions $\phi(0) = 0, \frac{d^2\phi(0)}{dx^2} = 0$ at $x=0$, and $\phi(l_{dv,eff}) = 0$ at $x=l_{dv,eff}$. Also, with reference to Figure 4.8, $l_{dv,eff} = x_1 + x_2$ is the effective length of the valve measured from the clamped end to the tip of the valve reed. Figure 4.9 plotted the mode shapes of both cases.

Figure 4.9 Valve mode shape function

In addition, Equation (4.38) can be separated into two first order differential equations by letting $h = \frac{dg}{dt} = \dot{g}$, which form the coupled equations below:

$$\frac{dh_1}{dt} = \frac{C_{dv,eff}\phi_1(x)A_{flow,disc}P(t)}{\int \rho_{dv}A_{dv}(x)\phi_1^2(x)dx} - 2\xi\omega_n h_1 - \omega_n^2 g_1 \quad (4.41)$$

$$\frac{dg_1}{dt} = h_1 \quad (4.42)$$

The subscript “1” in Equations (4.41) and (4.42) refer to the 1st vibration mode. By solving Equations (4.41) and (4.42) altogether with the thermodynamics model, the mode participation factor can be obtained, subsequently, the value of valve deflection can be found by multiplying it with the mode shape function.

4.7 Concluding Remarks

To this end, the basic mathematical model has been formulated for UVC. The geometrical model of the UVC, which describes the variations of the suction chamber and the compression chamber volumes, is shown. It is followed by the kinematics of several moving parts in UVC, mainly the rotor and the vane bush. The thermodynamic model that explains the variation of the working fluid thermodynamics properties during the compressor operation is presented. The suction and discharge flows model that portrays the flow of working fluid during the suction and discharge processes are formulated. Lastly, the valve reed dynamics which demonstrates the discharge valve response to the working fluid pressure is modelled mathematically.

The simulation results will be shown and discussed in Chapter 6.

Chapter 5 Mathematical Model II: Dynamics, Internal Leakage, Instantaneous Power and Efficiencies

Though the in-chamber working fluid properties can be described by the basic mathematical model formulated in Chapter 4, it is insufficient to evaluate the performance of a compressor without analysing the dynamic and internal leakage aspects of the compressor during operation.

In this chapter, (1) dynamics model, which analyses the forces acting on each of the compressor components; (2) mechanical losses, which derive the frictional losses in U-Vane compressor (UVC) and; (3) internal leakage model, which illustrates the internal leakage flow from high pressure chamber to low pressure chamber during operation, will be presented. Subsequently, the instantaneous power required to drive the compressor, as well as the definition of the efficiencies, will be expressed mathematically.

5.1 Dynamic Model

Basically, the dynamic aspect of the compressor involves all the forces acting on the compressor components that are in contact during the operation, especially the moving parts – vane bush, rotor, shaft and eccentric. As introduced in Chapter 3, the rotor, which is having a swaying motion guided by the vane bush, is subjected to forces exerted by the vane bush. Additionally, the extended shoulders of the vane bush may pose extra frictional forces on the vane or worse, leading to operation failures, if the force acting on it is not analysed properly. Thus, it is essential to investigate the dynamics of the components in order to determine the critical geometrical dimensions of the components.

5.1.1 Force Analysis of Vane Bush

Based on Figures 5.1 and 5.2, consider the leaning of the vane due to the sliding clearance between the vane and the slot, Equations (5.1) - (5.4) can be obtained from the equilibriums of forces and torques [199]:

Figure 5.1 Geometrical dimensions of bush and vane

Figure 5.2 Vane tilting and the resultant force diagram of vane bush

$$\sum F_x = 0;$$

$$\begin{aligned} & F_{vboc,n} \cos \gamma_1 + F_{vboc,t} \sin \gamma_1 - F_{1,n} \cos \gamma_1 - \\ & F_{1,t} \sin \gamma_1 + F_{2,n} \cos \gamma_1 - F_{2,t} \sin \gamma_1 - \\ & F_{vbsd,press} \cos \gamma_1 + F_{vbsd,1} \sin \gamma_1 + F_{vbsd,2} \sin \gamma_1 + F_{slot,vb} \sin \gamma_1 = 0 \end{aligned} \quad (5.1)$$

$$\begin{aligned}
\sum F_y &= 0; \\
&-F_{vboc,n} \sin \gamma_1 + F_{vboc,t} \cos \gamma_1 + F_{1,n} \sin \gamma_1 - \\
&F_{1,t} \cos \gamma_1 - F_{2,n} \sin \gamma_1 - F_{2,t} \cos \gamma_1 + \\
&F_{vbsd,press} \sin \gamma_1 + F_{vbsd,1} \cos \gamma_1 + F_{vbsd,2} \cos \gamma_1 + F_{slot,vb} \cos \gamma_1 = 0
\end{aligned} \tag{5.2}$$

$$\begin{aligned}
\sum T_o &= I_{vb,O} \frac{d^2 \gamma_1}{dt^2} \text{ about point } O \text{ for } l_v < l_{vbsd}; \\
&(F_{1,t} - F_{2,t}) \frac{w_{vro}}{2} + F_{2,n} (r_{vb} + l_{vbsd}) - \\
&F_{1,n} (r_{vb} + l_v - l_{vro}) - F_{vbsd,press} \left(r_{vb} + \frac{l_v}{2} \right) - \\
&F_{vbsd,1} \frac{w_{vro} + w_{vbsd}}{2} + F_{vbsd,2} \frac{w_{vro} + w_{vbsd}}{2} = I_{vb,O} \frac{d^2 \gamma_1}{dt^2}
\end{aligned} \tag{5.3}$$

$$\begin{aligned}
\sum T_o &= I_{vb,O} \frac{d^2 \gamma_1}{dt^2} \text{ about point } O \text{ for } l_v > l_{vbsd}; \\
&(F_{1,t} - F_{2,t}) \frac{w_{vro}}{2} + F_{2,n} (r_{vb} + l_{vbsd}) - \\
&F_{1,n} (r_{vb} + l_v - l_{vro}) - F_{vbsd,press} \left(r_{vb} + \frac{l_{vbsd}}{2} \right) - \\
&F_{vbsd,1} \frac{w_{vro} + w_{vbsd}}{2} + F_{vbsd,2} \frac{w_{vro} + w_{vbsd}}{2} = I_{vb,O} \frac{d^2 \gamma_1}{dt^2}
\end{aligned} \tag{5.4}$$

where,

$F_{1,n}$ is the force acting on the bush shoulder by the vane tip in normal direction

(N)

$F_{1,t}$ is the frictional force acting on the bush shoulder by the vane tip in tangential

direction (N)

$F_{2,n}$ is the force acting on the bush shoulder tip by the vane in normal direction

(N)

$F_{2,t}$ is the frictional force acting on the bush shoulder tip by the vane in tangential

direction (N)

$F_{vboc,n}$ is the force acting on the vane bush by the cylinder wall in the x-direction

(N)

$F_{vboc,t}$ is the force acting on the vane bush by the cylinder wall in the y-direction (N)

$F_{vbsd,press}$ is the resultant pressure force acting on the extended bush shoulder due to the pressure difference between compression and suction chambers (N)

$F_{vbsd,1}$ is the resultant pressure force acting on the shoulder at the suction chamber (N)

$F_{vbsd,2}$ is the resultant pressure force acting on the shoulder at the compression chamber (N)

$F_{slot,vb}$ is the resultant pressure force acting on the vane slot in the bush (N)

l_{vbsd} is the length of shoulder from the radius of the vane bush (m)

l_{vro} is the length of vane from the radius of the outer rotor surface (m)

w_{vro} is the width of the vane (m)

w_{vbsd} is the width of the shoulder, which is assumed to be the same value as the corresponding shoulder slot at rotor (m)

$I_{vb,O}$ is the moment of inertia of the vane bush about point O (kgm^2)

The resultant pressure force, vane side frictional forces and the bush-cylinder wall frictional force ($F_{vboc,f}$) are defined as follows:

$$F_{vbsd,press} = \begin{cases} (P_{com} - P_{suc}) l_v l_{com} & \text{for } l_v > l_{vbsd} \\ (P_{com} - P_{suc}) l_{vbsd} l_{com} & \text{for } l_v < l_{vbsd} \end{cases} \quad (5.5)$$

$$F_{slot,vb} = \frac{1}{2} (P_{com} - P_{suc}) w_{vro} l_{com} \quad (5.6)$$

$$F_{vbsd,1} = P_{suc} w_{vbsd} l_{com} \quad (5.7)$$

$$F_{vbsd,2} = P_{com} w_{vbsd} l_{com} \quad (5.8)$$

$$F_{1,t} = \mu_{vs} F_{1,n} \quad (5.9)$$

$$F_{2,t} = \mu_{vs} F_{2,n} \quad (5.10)$$

$$F_{vboc,f} = \mu_{vbs} \sqrt{F_{vboc,n}^2 + F_{vboc,t}^2} \quad (5.11)$$

where,

μ_{vs} is the coefficient of friction at vane side (-)

μ_{vbs} is the coefficient of friction at the side of the vane bush (-)

P_{com} is the instantaneous working fluid pressure in the compression chamber (Pa)

P_{suc} is the instantaneous working fluid pressure in the suction chamber (Pa)

For μ_{vs} and μ_{vbs} , constant values of 0.15 and 0.10 are used, respectively, as proposed by Yanagisawa *et al.* [200] and Teh and Ooi [21].

5.1.2 Force Analysis of Rotor

Figure 5.3 shows the geometrical relations and force diagram of rotor.

Figure 5.3 Geometrical relations and force diagram of rotor

As the rotor is exposed to the working fluid in the working chamber, it encounters with working fluid pressure forces from both suction and compression chambers. The pressure force is divided into two parts, one acts on the vane while another acts on the rotor. The magnitudes of the pressure forces are defined as:

$$F_{vro,press} = \begin{cases} (P_{com} - P_{suc})(l_v - l_{vbsd})l_{com} & \text{for } l_v > l_{vbsd} \\ 0 & \text{for } l_v < l_{vbsd} \end{cases} \quad (5.12)$$

$$F_{AB,press} = (P_{com} - P_{suc})l_{AB}l_{com} \quad (5.13)$$

where,

$F_{vro,press}$ is the resultant pressure force acting on the exposed vane in the working chamber (N)

$F_{AB,press}$ is the resultant pressure force acting on the rotor (N)

The length l_{AB} shown in Figure 5.3 refers to the projected length that the working fluid pressure is acting on, which is expressed by:

$$l_{AB} = 2r_{ro} \sin \frac{\theta_2}{2} \quad (5.14)$$

By taking the reaction forces from the vane bush into the force analysis, and applying the force and torque equilibriums, the equations are brought out as:

$$\sum F_x = 0;$$

$$\begin{aligned} & F_{1,n} \cos \gamma_1 + F_{1,t} \sin \gamma_1 - F_{2,n} \cos \gamma_1 + \\ & F_{2,t} \sin \gamma_1 - F_{slot,vb} \sin \gamma_1 - \\ & F_{vro,press} \cos \gamma_1 + F_{Rx} - F_{AB,press} \cos \left(\frac{\pi}{2} + \gamma_1 - \frac{\theta_2}{2} \right) = 0 \end{aligned} \quad (5.15)$$

$$\sum F_y = 0;$$

$$\begin{aligned} & -F_{1,n} \sin \gamma_1 + F_{1,t} \cos \gamma_1 + F_{2,n} \sin \gamma_1 + \\ & F_{2,t} \cos \gamma_1 - F_{slot,vb} \cos \gamma_1 + \\ & F_{vro,press} \sin \gamma_1 + F_{Ry} + F_{AB,press} \sin \left(\frac{\pi}{2} + \gamma_1 - \frac{\theta_2}{2} \right) = 0 \end{aligned} \quad (5.16)$$

Since the rotor is oscillating about point O , the torque equilibrium can be taken about point O , which yields:

$$\sum T_o = I_{ro,O} \frac{d^2 \gamma_1}{dt^2} \text{ about point } O \text{ for } l_v < l_{vbsd};$$

$$\begin{aligned} & F_{1,n} (r_{vb} + l_v - l_{vro}) + F_{Rx} (l_v + r_{vb} + r_{ro}) \cos \gamma_1 - \\ & F_{1,t} \frac{w_{vro}}{2} + F_{2,t} \frac{w_{vro}}{2} - F_{2,n} (r_{vb} + l_{vbsd}) - \\ & F_{Ry} (l_v + r_{vb} + r_{ro}) \sin \gamma_1 - \\ & F_{AB,press} (r_{vb} + l_v + r_{ro}) \sin \frac{\theta_2}{2} = I_{ro,O} \frac{d^2 \gamma_1}{dt^2} \end{aligned} \quad (5.17)$$

$$\sum T_o = I_{ro,O} \frac{d^2 \gamma_1}{dt^2} \text{ about point } O \text{ for } l_v > l_{vbsd};$$

$$\begin{aligned}
& F_{1,n} (r_{vb} + l_v - l_{vro}) + F_{Rx} (l_v + r_{vb} + r_{ro}) \cos \gamma_1 - \\
& F_{1,t} \frac{W_{vro}}{2} + F_{2,t} \frac{W_{vro}}{2} - F_{2,n} (r_{vb} + l_{vbsd}) - \\
& F_{Ry} (l_v + r_{vb} + r_{ro}) \sin \gamma_1 - F_{vro,press} \left(r_{vb} + l_{vbsd} + \frac{l_v - l_{vbsd}}{2} \right) - \quad (5.18) \\
& F_{AB,press} (r_{vb} + l_v + r_{ro}) \sin \frac{\theta_2}{2} = I_{ro,O} \frac{d^2 \gamma_1}{dt^2}
\end{aligned}$$

where,

F_{Rx} is the resultant force acting on the rotor in x-direction (N)

F_{Ry} is the resultant force acting on the rotor in y-direction (N)

$I_{ro,O}$ is the moment of inertia of the rotor about point O (kgm²)

5.1.3 Moment of Inertia of Vane Bush and Rotor

The moment of inertia of rotating parts is important as it affects the magnitude of the contact forces between the components and the magnitude of torque input to drive the compressor. Although both the vane bush and the rotor have unusual geometries, they can be partitioned into sections and simplified into regular shapes. Parallel axis theorem is applied to the section if the centre of mass of that section is not coincided with the axis of rotation. Figures 5.4 and 5.5 show the original geometries of the vane bush and the rotor and their partitioned sections. The shapes in dotted-lined in both figures are the virtual masses representing the vane slot and the shoulder slots, respectively.

Figure 5.4 Partitions of the geometry of vane bush

Figure 5.5 Partitions of the geometry of rotor

For simplicity, the detailed features such as the fillet and the offset of the bush shoulders are neglected. Approximating that the vane slot has the same length as the vane, the moment of inertia of vane bush (Figure 5.4) about point O can be expressed as:

$$I_{vb,O} = \frac{m_{I,vb}}{2} r_{vb}^2 + m_{II,vb} \left\{ \frac{1}{12} \left[(2w_{vbsd} + w_{vro})^2 + l_{vbsd}^2 \right] + \left(r_{vb} + \frac{l_{vbsd}}{2} \right)^2 \right\} - m_{III,vb} \left\{ \frac{1}{12} \left[w_{vro}^2 + (l_{vro} + l_{vbsd})^2 \right] + \left[\frac{1}{2} (l_{vro} + l_{vbsd}) - r_{vb} \right]^2 \right\} \quad (5.19)$$

Similarly, the fillets at the vane tip and at the edges of the shoulder slots (Figure 5.5) are ignored and hence, the moment of inertia of the rotor about point O can be defined:

$$I_{ro,O} = m_{I,ro} \left[\frac{(w_{vro}^2 + l_{vro}^2)}{12} + \left(r_{vb} + l_v - \frac{l_{vro}}{2} \right)^2 \right] + m_{II,ro} \left[\frac{(r_{ro}^2 - r_{roi}^2)}{2} + (r_{vb} + l_v + r_{ro})^2 \right] - 2m_{III,ro} \left\{ \frac{(w_{vbsd}^2 + l_{vbsd}^2)}{12} + \left(r_{vb} + l_v + \frac{l_{vbsd}}{2} \right)^2 + \left(\frac{w_{vbsd}}{2} + \frac{w_{vro}}{2} \right)^2 \right\} \quad (5.20)$$

with reference to the figures above, where,

$I_{vb,O}$ is the mass moment of inertia of vane bush about point O (kgm²)

$I_{ro,O}$ is the mass moment of inertia of rotor about point O (kgm²)

$m_{I,vb}$ is the mass of the vane bush Section I (kg)

$m_{II,vb}$ is the mass of the vane bush Section II (kg)

$m_{III,vb}$ is the virtual mass of the vane bush Section III (kg)

$m_{I,ro}$ is the mass of the rotor Section I (kg)

$m_{II,ro}$ is the mass of the rotor Section II (kg)

$m_{III,ro}$ is the virtual mass of the rotor Section III (kg)

l_{vbsd} is the length of the shoulder slot at the rotor (m)

In addition, the mass of each sections is defined as below:

$$m_{I,vb} = \rho_{vb} \pi r_{vb}^2 l_{com}$$

$$m_{II,vb} = \rho_{vb} l_{vb} (2w_{vbsd} + w_{vro}) l_{com}$$

$$m_{III,vb} = \rho_{vb} (l_{vbsd} + l_{vro}) w_{vro} l_{com}$$

$$m_{I,ro} = \rho_{ro} l_{vro} w_{vro} l_{com}$$

$$m_{II,ro} = \rho_{ro} \pi (r_{ro}^2 - r_{roi}^2) l_{com}$$

$$m_{III,ro} = \rho_{ro} l_{vbsd} w_{vbsd} l_{com}$$

ρ_{vb} and ρ_{ro} are the densities of the materials used to produce vane bush and rotor respectively.

5.2 Mechanical Losses

In this section, several frictional losses are identified from the contact regions between each of the compressor components. These frictional losses comprise:

- (i) Vane side frictional loss due to the sliding of the vane along the contact surface in the vane slot at the bush
- (ii) Vane bush side frictional loss due to the swivelling motion of the vane bush in the slot located at the cylinder wall
- (iii) End-face frictional losses due to the relative motion between two contact surfaces, which are found in: between the lower/upper end-faces of vane bush and compressor base/cover; between the lower/upper end-faces of rotor and compressor base/cover, between the lower/upper end-faces of eccentric and the compressor base/cover

The mathematical expressions of each of the frictional losses are presented in this section.

5.2.1 Frictional Loss between Vane Side and Vane Slot

When the vane slides along the surface of the vane slot in the bush, friction occurs at the two contact surfaces, as referred to Figure 5.2. The vane side frictional loss can be formulated as:

$$L_{f,rvs} = (F_{1,t} + F_{2,t}) \frac{dl_v}{dt} = \mu_{vs} (F_{1,n} + F_{2,n}) \frac{dl_v}{dt} \quad (5.21)$$

Where,

$L_{f,rvs}$ is the vane side frictional loss (W)

5.2.2 Frictional Loss between Vane Bush Side and Cylinder Wall

Vane bush side friction is incurred when the vane bush swivels in the slot at the cylinder wall during operation. With reference to Figure 5.2, the frictional loss is derived as below:

$$L_{f,vbs} = \frac{d\gamma_1}{dt} r_{vb} \mu_{vbs} \sqrt{F_{vbc,n}^2 + F_{vbc,t}^2} = \frac{d\gamma_1}{dt} r_{vb} F_{vbc,f} \quad (5.22)$$

where,

$L_{f,vbs}$ is the vane bush side frictional loss (W)

5.2.3 Vane Bush End-face Frictional Loss

Generally, the end-face friction is caused by the shear stress ($\bar{\tau}$) in the lubricant between two rubbing surfaces, which is given as:

$$\bar{\tau} = \mu \frac{dv}{dy} = \frac{\mu_{lub}}{\delta} \bar{V} \quad (5.23)$$

where,

μ_{lub} is the absolute viscosity of the lubricant (Pa•s)

δ is the distance between the two rubbing surfaces (m)

$\bar{V}$ is the relative velocity (m/s)

With reference to Figure 5.4, the vane bush end-face frictional loss is separated into sections and analysed accordingly. The shear stress (τ_{vbef}) incurred in the end-face is expressed as:

$$\tau_{vbef} = \frac{\mu_{lub}}{\delta_{vbef}} r \frac{d\gamma_1}{dt} \quad (5.24)$$

where,

r is the distance variable from the centre axis O (m)

δ_{vbef} is the vane bush end-face clearance (m)

From Figure 5.4, the vane bush end-face frictional loss of Section I can be derived from the integration of the shear stress, as shown in Equation (5.25):

$$\begin{aligned}
L_{f,vbef,I} &= 2 \int_{A_{I,vb}} \tau_{vbef} d\vec{A} \cdot d\vec{v} \\
\rightarrow L_{f,vbef,I} &= \frac{2\mu_{lub}}{\delta_{vbef}} \int_0^{r_{vb}} r \frac{d\gamma_1}{dt} 2\pi r d\vec{r} \cdot \vec{r} \frac{d\gamma_1}{dt} = \frac{4\pi\mu_{lub}}{\delta_{vbef}} \left(\frac{d\gamma_1}{dt} \right)^2 \int_0^{r_{vb}} r^3 dr \quad (5.25) \\
\rightarrow L_{f,vbef,I} &= \frac{\pi\mu_{lub} r_{vb}^4}{\delta_{vbef}} \left(\frac{d\gamma_1}{dt} \right)^2
\end{aligned}$$

For Section II, it is assumed that the whole Section II is moving at the same relative speed as its centre with respect to the centre axis O , as illustrated in Figure 5.6.

Figure 5.6 Assumption of the speed of Section II

The end-face frictional loss of Section II can be expressed as:

$$\begin{aligned}
L_{f,vbef,II} &= 2 \int_{A_{II,vb}} \tau_{vbef} d\vec{A} \cdot d\vec{v} \\
\rightarrow L_{f,vbef,II} &= \frac{2\mu_{lub}}{\delta_{vbef}} (2w_{vbsd} + w_{vbsd}) l_{vbsd} v_{centre}^2 \quad (5.26) \\
\rightarrow L_{f,vbef,II} &= \frac{2\pi\mu_{lub} l_{vbsd}}{\delta_{vbef}} (2w_{vbsd} + w_{vro}) \left(r_{vb} + \frac{l_{vbsd}}{2} \right)^2 \left(\frac{d\gamma_1}{dt} \right)^2
\end{aligned}$$

To account for the virtual ‘Section III’ (Figure 5.4), a ratio is defined based on the sectional areas of the partitions. This ratio is multiplied with the total vane bush end-face frictional loss to obtain a more accurate approximation of the end-face frictional loss. The area ratio is defined as:

$$\frac{A_{III,vb}}{A_{I,vb} + A_{II,vb}} = \frac{(l_{vro} + l_{vbsd}) w_{vro}}{\pi r_{vb}^2 + (2w_{vbsd} + w_{vro}) l_{vbsd}} \quad (5.27)$$

Lastly, the resultant vane bush end-face frictional loss is obtained:

$$L_{f,vbef} = \left(1 - \frac{(l_{vro} + l_{vbsd}) w_{vro}}{\pi r_{vb}^2 + (2w_{vbsd} + w_{vro}) l_{vbsd}} \right) (L_{f,vbef,I} + L_{f,vbef,II}) \quad (5.28)$$

where,

$L_{f,vbef}$ is the resultant vane bush end-face frictional loss (W)

$A_{I,vb}$ is the sectional area of Section I (m²)

$A_{II,vb}$ is the sectional area of Section II (m²)

$A_{III,vb}$ is the sectional area of Section III (m²)

5.2.4 Rotor End-face Frictional Loss

The total rotor end-face frictional loss comprises two portions – vane and rotor (hollow cylinder portion), as shown in Figure 5.5. Each portion consists of the rubbings due to the linear translational motion and the rotational motion. For rotor end-face, the shear stress in between the rubbing surfaces caused by the rotational motion is the same as Equation (5.24) ($\tau_{r,ref} = \tau_{vbef}$). In the case of the translational motion, the shear stress is expressed by:

$$\tau_{t,ref} = \frac{\mu_{lub}}{\delta_{ref}} \cdot \frac{dl_v}{dt} \quad (5.29)$$

Thus, the frictional loss of the vane end-face can be derived below:

$$\begin{aligned} L_{f,ref,vro} &= 2(L_{t,vro} + L_{r,vro}) \\ \rightarrow L_{f,ref,vro} &= 2 \left[\int_{A_{vro}} \tau_{t,ref} d\vec{A} \cdot d\vec{v} + \int_{A_{vro}} \tau_{r,ref} d\vec{A} \cdot d\vec{v} \right] \\ \rightarrow L_{f,ref,vro} &= 2 \left[l_{vro} w_{vro} \frac{\mu_{lub}}{\delta_{ref}} \left(\frac{dl_v}{dt} \right)^2 + \frac{\mu_{lub}}{\delta_{ref}} (l_{vro} w_{vro}) \left(r_{vb} + l_v - \frac{l_{vro}}{2} \right)^2 \left(\frac{d\gamma_1}{dt} \right)^2 \right] \end{aligned} \quad (5.30)$$

where,

$L_{f,ref,vro}$ is the end-face frictional loss of the vane (W)

$A_{vro} = l_{vro} w_{vro}$ is the vane end-face sectional area (m²)

Therefore, the end-face frictional loss of the rotor is expressed as below:

$$L_{f,ref,ro} = \frac{2\pi\mu_{lub}}{\delta_{ref}}(r_{ro}^2 - r_{roi}^2)\left(\frac{dl_v}{dt}\right)^2 + \frac{2\pi\mu_{lub}}{\delta_{ref}}\left[\frac{1}{2}(r_{ro}^4 - r_{roi}^4) + (r_{vb} + l_v + r_{ro})^2(r_{ro}^2 - r_{roi}^2)\right]\left(\frac{d\gamma_1}{dt}\right)^2 \quad (5.31)$$

where,

$L_{f,ref,ro}$ is the end-face frictional loss of the rotor (hollow cylinder portion) (W)

It is noticeable that the shoulder slots at the rotor are neglected because of their comparatively small rubbing surface area. As a result, the resultant rotor end-face frictional loss ($L_{f,ref}$) is obtained by adding Equation (5.30) and (5.31) together:

$$\begin{aligned} L_{f,ref} &= L_{f,ref,vro} + L_{f,ref,ro} \\ &= 2\frac{\mu_{lub}}{\delta_{ref}}\left\{l_{vro}w_{vro}\left(\frac{dl_v}{dt}\right)^2 + \left(l_{vro}w_{vro}\right)\left(r_{vb} + l_v - \frac{l_{vro}}{2}\right)^2\left(\frac{d\gamma_1}{dt}\right)^2 + \right. \\ &\quad \left.\pi(r_{ro}^2 - r_{roi}^2)\left(\frac{dl_v}{dt}\right)^2 + \right. \\ &\quad \left.\pi\left[\frac{1}{2}(r_{ro}^4 - r_{roi}^4) + (r_{vb} + l_v + r_{ro})^2(r_{ro}^2 - r_{roi}^2)\right]\left(\frac{d\gamma_1}{dt}\right)^2\right\} \end{aligned} \quad (5.32)$$

5.2.5 Eccentric End-face Frictional Loss

Also, the eccentric is rotating with respect to the top and bottom surfaces, the end-face frictional loss between the two surfaces is derived below:

$$L_{f,ecef} = \frac{\pi\mu_{lub}}{\delta_{ecef}}(2r_{ecc}^4 - r_s^4)\omega_1^2 \quad (5.33)$$

where,

$L_{f,ecef}$ is the eccentric end-face frictional loss (W)

δ_{ecef} is the eccentric end-face clearance (m)

r_{ecc} is the radius of the eccentric (m)

5.2.6 Frictional Loss between Eccentric and Inner Rotor Surface

During the compressor operation, the eccentric rotates and its surface slides against the inner rotor surface, which causes frictional loss between the two surfaces. This frictional loss is formulated as:

$$L_{f, ecc} = \frac{2\pi\mu_{\text{lub}}}{\delta_{ecc, rad}} r_{ecc}^3 l_{ecc} \left(\omega_1 - \frac{d\gamma_1}{dt} \right)^2 \quad (5.34)$$

where,

$L_{f, ecc}$ is the frictional loss between the eccentric and the inner rotor surface (W)

$\delta_{ecc, rad}$ is the radial clearance between the eccentric and the inner rotor surface (m)

In addition to the aforementioned frictional losses, there is journal bearing frictional loss, which is presented in Appendix I.

5.3 Internal Leakage Model

Most compressors experience ‘internal leakages’ from high pressure chamber to low pressure chamber during the operation due to the presence of clearance gaps between the mating surfaces of the compressor components. It is an undesirable phenomenon as the compressor performance will be affected. Thus, it is important to understand the internal leakage aspect in the U-Vane compressor. Figure 5.7 depicts three main internal leakage paths in UVC, as listed below:

- (i) Leakage through the radial clearance at the contact line between cylinder and rotor
- (ii) Leakage through the vane end-face clearance
- (iii) Leakage through the rotor end-face clearance (Appendix C.1)

Figure 5.7 Internal leakage paths of U-Vane compressor

The leakage through the rotor end-face clearance is incorporated into the mathematical model through CFD numerical approach [201], as presented in Appendix C.1. Additionally, the leakage through the vane side clearance is not listed above because of the vane tilting shown in Figure 5.2. The clearance between the vane side and the slot is assumed to be sealed and no leakage occurs at the vane side clearance. It is also assumed that the leaking fluid is a pure working fluid single-phase flow only in order to find the possible maximum leakage mass flow rate of the working fluid. However, in actual compressor operation, the leakage flow could be a mixture flow or a two-phase flow due to the presence of oil. In this case, a flow coefficient (C_{leak}) can be added to the maximum leakage mass flow rate to account for the leakage flow conditions.

5.3.1 Leakage through Radial Clearance

A close-up view of the leakage through the radial clearance between cylinder and rotor is shown in Figure 5.8. This leakage flow is modelled as Fanno Flow by taking the frictional loss and the viscous effect into consideration [150], as shown in Figure 5.9.

Figure 5.8 Close-up view of the radial clearance geometry

Figure 5.9 Fanno Flow model of leakage flow through radial clearance [150]

As proposed by Yanagisawa and Shimizu [150], choked flow is first assumed to occur at the exit of the channel and the governing equations derived below can be applied to find the leakage mass flow rate:

$$\lambda \frac{l_f}{2\delta_{rad}} = \frac{1 - M_t^2}{\kappa M_t^2} + \frac{\kappa + 1}{2\kappa} \ln \frac{M_t^2 (\kappa + 1)}{2 + M_t^2 (\kappa - 1)} \quad (5.35)$$

$$\frac{P_t}{P_e} = \frac{1}{M_t} \sqrt{\frac{(\kappa + 1)}{2 + M_t^2 (\kappa - 1)}} \quad (5.36)$$

$$\frac{P_{com}}{P_t} = \left(1 + \frac{\kappa - 1}{2} M_t^2 \right)^{\frac{\kappa}{\kappa - 1}} \quad (5.37)$$

$$\frac{P_{com}}{P_e} = \frac{P_{com}}{P_t} \cdot \frac{P_t}{P_e} \quad (5.38)$$

$$T_e = \frac{T_{com}}{1 + \frac{M_t^2(\kappa-1)}{2}} \quad (5.39)$$

$$v_e = M_e \sqrt{\kappa R T_e} \quad (5.40)$$

$$\left(\frac{dm}{dt} \right)_{leak,rad} = \delta_{rad} w_e v_e \left(\frac{P_e}{R T_e} \right) \quad (5.41)$$

where

$$l_f = \frac{2\pi\delta_{rad}r_{cyl}}{e\sqrt{1-\left(1-\frac{\delta_{rad}}{e}\right)^2}} \text{ is the length of channel (m)} \quad (5.42)$$

$$\lambda = \begin{cases} \frac{96}{Re} & \text{for } Re \leq 3560 \\ \frac{0.3164}{Re^{0.25}} & \text{for } Re > 3560 \end{cases} \text{ is the friction factor (-)} \quad (5.43)$$

$$Re = \frac{2}{\mu l_e} \cdot \left(\frac{dm}{dt} \right)_{leak,rad} \text{ is the Reynolds Number (-)} \quad (5.44)$$

δ_{rad} is the radial clearance between rotor and cylinder (m)

M_t is the Mach number at the throat (-)

κ is the specific heat ratio (-)

R is the specific gas constant (J/kg•K)

P_t is the pressure at the throat (Pa)

P_e is the pressure of the working fluid at the exit of the channel (Pa)

T_{com} is the temperature at the compression chamber (K)

T_e is the temperature at the exit of the channel (K)

v_e is the working fluid velocity at the exit of the channel (m/s)

w_e is the width of the chamber at the exit of the channel (m)

μ is the dynamic viscosity of the working fluid (Pa•s)

$\left(\frac{dm}{dt}\right)_{leak,rad}$ is the leakage mass flow rate through the radial clearance (kg/s)

To have the flow at the channel exit to be choked, the calculated pressure ratio from Equation (5.38) should be equal to or less than the actual pressure ratio. On the other hand, if the calculated pressure ratio is higher than the actual pressure ratio, the flow will not be choked at the channel exit and a longer channel length is required to have choked flow to occur, as illustrated in Figure 5.10.

Figure 5.10 Assumed channel length for choked flow to occur

For such case, the Mach number at the throat (M_t^*) can be iterated by using Equations (5.45) - (5.50):

$$\lambda \frac{l_f^*}{2\delta_{rad}} = \frac{1-M_t^{*2}}{\kappa M_t^{*2}} + \frac{\kappa+1}{2\kappa} \ln \frac{M_t^{*2}(\kappa+1)}{2+M_t^{*2}(\kappa-1)} \quad (5.45)$$

$$\lambda \frac{l_f^* - l_f}{2\delta_{rad}} = \frac{1-M_e^2}{\kappa M_e^2} + \frac{\kappa+1}{2\kappa} \ln \frac{M_e^2(\kappa+1)}{2+M_e^2(\kappa-1)} \quad (5.46)$$

$$\frac{P_t}{P^*} = \frac{1}{M_t^*} \sqrt{\frac{(\kappa+1)}{2+M_t^{*2}(\kappa-1)}} \quad (5.47)$$

$$\frac{P_{com}}{P_t} = \left(1 + \frac{\kappa-1}{2} M_t^{*2}\right)^{\frac{\kappa}{\kappa-1}} \quad (5.48)$$

$$\frac{P_e}{P^*} = \frac{1}{M_e^*} \sqrt{\frac{(\kappa+1)}{2+M_e^{*2}(\kappa-1)}} \quad (5.49)$$

$$\frac{P_{com}}{P_e} = \frac{P_{com}}{P_t} \cdot \frac{P_t}{P^*} \cdot \frac{P^*}{P_e} \quad (5.50)$$

where

l_f^* is the assumed frictional channel length (m)

M_t^* is the assumed Mach number at the throat (-)

P^* is the pressure of the fluid when the flow is choked (Pa)

T^* is the temperature of the fluid when the flow is choked (K)

M^* is the unity Mach number ($M^* = 1$) (-)

The calculated pressure ratio from Equation (5.50) is examined against the actual pressure ratio for each iterations and would stop only if the difference between the pressure ratios falls within the pre-set convergence criterion of $\pm 1\%$. After that, the leakage mass flow rate is evaluated. As the equations are non-linear, direct root search method and bisection method [202] are employed to solve the equations.

5.3.2 Leakage through Vane End-face Clearance

For the leakage through the vane end-face clearance, it can be analysed as Fanno Flow model without a convergent section. Equations (5.45) - (5.50) are adapted and the equivalent frictional channel length is replaced with the effective vane width ($l_{f,vef}$) for this case. During the compressor operation, the overall vane width that exposed to the working chamber varies, as illustrated in Figure 5.11, which in turn causes the frictional channel length of the Fanno Flow to vary.

Figure 5.11 Leakage through the vane end-face clearance when (a) shoulders in the slots and (b) shoulders out of the slots

With reference to Figure 5.1, the effective vane width that exposed in the chamber and the frictional channel ‘width’ are defined as below:

$$l_{f,vef} = \begin{cases} \left(1 - \frac{l_{vbsd}}{l_v + l_{vbsd}}\right) w_{vro} + \left(\frac{l_{vbsd}}{l_v + l_{vbsd}}\right) (w_{vro} + 2w_{vbsd}) & \text{when } l_v > l_{vbsd} \\ w_{vro} + 2w_{vbsd} & \text{when } l_v < l_{vbsd} \end{cases} \quad (5.51)$$

$$w_{e,vef} = l_v + l_{vbsd} \quad (5.52)$$

Figure 5.12 Fanno Flow model of leakage through the vane end-face clearance

Therefore, together with Equations (5.45) - (5.50), the mass flow rate of the leakage through the vane end-face clearance can be found from Equation (5.53) below:

$$\left(\frac{dm}{dt} \right)_{leak,vef} = \delta_{vef} w_{e,vef} v_e \left(\frac{P_e}{RT_e} \right) \quad (5.53)$$

where,

δ_{vef} is the vane end-face clearance (m)

$w_{e,vef}$ is the width of the chamber at the exit of the channel (m)

$\left(\frac{dm}{dt} \right)_{leak,vef}$ is the leakage mass flow rate through the vane end-face clearance (kg/s)

5.4 Power and Efficiencies

After introducing various mathematical models of U-Vane compressor, the total instantaneous power required to drive the compressor can be derived.

Figure 5.13 Pressure-volume diagram (PV diagram) of U-Vane compressor

Fundamentally, the total instantaneous power consumed by UVC can be divided into the compression power and the total frictional power loss. The mathematical expression of compression power ($\dot{W}_{com}$) is given as:

$$\dot{W}_{com} = \int_0^{4\pi} \left(P(\theta) \frac{dV(\theta)}{d\theta} \right) d\theta \cdot \frac{\omega_1}{2\pi} \quad (5.54)$$

where,

ω_1 is the shaft angular speed (rad/s)

$P(\theta)$ is the working fluid pressure in the working chamber (Pa)

$V(\theta)$ is the working chamber volume (m³)

Both $P(\theta)$ and $V(\theta)$ are functions of the shaft rotational angle (θ_1). The compression power is defined as the area bounded by the whole compressor operation cycle as illustrated in Figure 5.13, which comprises the total work done to compress the working fluid from the nominal suction pressure to nominal discharge pressure, suction and discharge losses. Both suction and discharge losses are related to the sizes of the corresponding ports and the valve reed geometry design. The discharge loss is the area above the nominal discharge pressure, while the suction loss is represented by the area below the nominal suction pressure.

The total frictional power loss ($\dot{W}_{f,loss}$) is the summation of all the frictional losses established in Sections 5.2 and Appendix I. By summing up Equations (5.21) - (5.22), (5.28), (5.32) - (5.34) and the journal bearing loss (Equation (II3)), the total frictional power loss can be found:

$$\dot{W}_{f,loss} = L_{f,rvs} + L_{f,vhs} + L_{f,vef} + L_{f,ef} + L_{f,ecef} + L_{f,ecc} + L_{f,b} \quad (5.55)$$

To this end, the total instantaneous power required by U-Vane compressor can be obtained in below:

$$\dot{W}_{Inst} = \dot{W}_{com} + \dot{W}_{f,loss} \quad (5.56)$$

where,

$\dot{W}_{Inst}$ is the total instantaneous power (W)

$\dot{W}_{com}$ is the compression power (W)

$\dot{W}_{f,loss}$ is the total frictional power loss (W)

Since the total instantaneous power represents the instant power required by UVC at each shaft rotational angle, to obtain the power input to UVC, the mean value of the total instantaneous power is taken, which can be mathematically expressed as:

$$\dot{W}_{input} = \frac{1}{2\pi} \sum_{i=1}^n \dot{W}_{Inst} \Delta\theta_{1,i} \quad (5.57)$$

In general, the performance of a compressor is evaluated in terms of efficiencies and hence, various efficiencies are defined for UVC as below [203, 204]:

$$\eta_{comp} = \frac{\bar{\dot{W}}_{comp,ideal}}{\bar{\dot{W}}_{comp,actual}} \quad (5.58)$$

$$\eta_{vol} = \frac{\dot{m}_{actual}}{\dot{m}_{ideal}} \quad (5.59)$$

$$\eta_{mech} = \frac{\bar{\dot{W}}_{comp,actual}}{\bar{\dot{W}}_{comp,actual} + \bar{\dot{W}}_{f,loss}} = \frac{\bar{\dot{W}}_{comp,actual}}{\dot{W}_{input}} \quad (5.60)$$

where,

$\bar{\dot{W}}_{comp,ideal}$ is the average compression power of compressor in ideal condition (W)

$\bar{\dot{W}}_{comp,actual}$ is the average compression power of compressor in actual condition (W)

$\dot{m}_{ideal}$ is the mass flow rate of compressor in ideal condition (kg/s)

$\dot{m}_{actual}$ is the mass flow rate of compressor in actual condition (kg/s)

$\dot{W}_{input}$ is the power input to the compressor (W)

$\bar{\dot{W}}_{f,loss}$ is the average frictional power loss of compressor (W)

η_{comp} is the compression efficiency (-)

η_{vol} is the volumetric efficiency (-)

η_{mech} is the mechanical efficiency (-)

In Equation (5.58), $\bar{\dot{W}}_{comp,ideal}$ is the required compression power based on the theoretical cyclic process of a compressor [203]. It is defined as the area bounded by $S1$, $S2$, $D1$ and $D2$ of the theoretical pressure-volume diagram, as shown in Figure 5.14.

Figure 5.14 Theoretical pressure-volume diagram of a compressor

Hence, the ideal compression power ($\bar{W}_{comp,ideal}$), and the ideal mass flow rate ($\dot{m}_{ideal}$, Equation (5.59)) can be expressed mathematically as below:

$$\bar{W}_{comp,ideal} = \int_0^{4\pi} \left(P_{ideal}(\theta) \frac{dV(\theta)}{d\theta} \right) d\theta \cdot \frac{\omega_1}{2\pi} \quad (5.61)$$

$$\dot{m}_{ideal} = \rho_{suc} V_{total} \frac{\omega_1}{2\pi} \quad (5.62)$$

where,

$P_{ideal}(\theta)$ is the ideal working fluid pressure (Figure 5.15) in the working chamber (kPa)

ρ_{suc} is the density of working fluid at suction condition (kg/m^3)

V_{total} is the volumetric displacement of the compressor (m^3)

Lastly, in the case of the refrigeration system, coefficient of performance (COP) [205] is applied:

$$COP = \frac{\dot{m}_{actual} \Delta h_{evap}}{\dot{W}_{input}} \quad (5.63)$$

where,

Δh_{evap} is the difference in the specific enthalpy between the fluids at evaporator inlet and outlet (J/kg)

5.5 Concluding Remarks

Several mathematical models that describe the operational behaviour of the compressor have been developed scientifically. The dynamic model of the U-Vane compressor portrays the details of the forces acting on each of the compressor components. Several contact regions between the compressor components are recognised and the mechanical losses encountered in these regions have been established. The dynamical instantaneous load of the compressor can be supported by the journal bearing as explained in Appendix I. Leakage flow paths have been identified, including leakages through the radial clearance between the rotor and the cylinder, the vane end-face clearance, and the rotor end-face clearance. Last but not least, the total instantaneous power and several efficiencies to evaluate the compressor performance are presented.

In next chapter, the simulation results will be presented and discussed.

Chapter 6 Theoretical Simulation Results and Study

The mathematical models presented in Chapters 4 and 5 are converted into computational prediction model using ForTran 90 programming language [206] and solved numerically using the 4th order of Runge-Kutta numerical integration method [207]. REFPROP database developed by NIST [125] is also integrated and simulated concurrently to obtain the instant thermodynamics properties of the working fluid in each integration step. In this chapter, the predictions from the simulation model are presented and analysed. Several design and operational parameters are varied to examine their effects on the compressor performance.

6.1 Simulation Setup and Operational Parameters

In this simulation studies, various operational parameters, the dimensions of major components and the clearances of UVC are tabulated in Table 6.1. R-1234yf is used as the working fluid due to its increasing use especially in the vehicle air-conditioning system [208]. R-1234yf has similar performance characteristics as R-134a and it is developed as a replacement of R-134a. Moreover, it is more environmentally friendly in terms of a lower global warming potential [209, 210]. Hence, R-1234yf is selected to be the working fluid for the simulation. The degree of suction superheat, evaporating and condensing temperatures are selected based on the AHRI [211] refrigerant compressor performance testing conditions. The properties of lubricant are given by the supplier and the dynamic viscosity is chosen by assuming steady state operation at constant temperature of 80°C [212]. The volumetric displacement is 32.5 cm³/rev as it is a typical compressor volumetric displacement for room air-conditioning [213, 214]. The operating speed is set at 3000 rev/min, which is a common operating speed for household refrigeration or air-conditioning compressor.

Table 6.1 Operational conditions of simulation

Operational Conditions and Specifications	
Working fluid [209, 210]	R-1234yf
Evaporating temperature [211]	7.2 °C
Condensing temperature [211]	54.4 °C
Degree of suction superheat [211]	27.8 °C
Dynamic viscosity of lubricant [212], μ_{lub}	0.0129 Pa·s
Volumetric displacement [213, 214], V_{total}	32.5 cm ³ /rev
Operating speed, ω_1	3000 rev/min (RPM)
Major Dimensions of UVC Components	
Shaft radius, r_s	12.5 mm
Eccentric radius, r_{ecc}	17.0 mm
Eccentric axial length, l_{ecc}	25.0 mm
Cylinder radius, r_{cyl}	36.0 mm
Rotor radius, r_{ro}	31.5 mm
Rotor wall thickness, t_{ro}	14.5 mm
Vane length, l_{vro}	43.0 mm from the centre of rotor
Vane width, w_{vro}	6.0 mm
Coefficient of friction of vane side [200], μ_{vs}	0.15
Length of shoulder slot on rotor, l_{vbsd}	6.34 mm
Vane bush radius, r_{vb}	9.0 mm
Coefficient of friction of the side of vane bush [21], μ_{vbs}	0.10
Shoulder length, l_{vbsd}	6.34 mm measured from outer vane bush surface
Compressor axial length, l_{com}	35.0 mm
Suction port diameter, $d_{\text{suc, port}}$	16.5 mm
Coefficient of discharge [215], $C_{\text{d, suc}}$	0.61
Discharge port diameter, $d_{\text{disc, port}}$	9.0 mm

Coefficient of discharge [215], $C_{d, disc}$	0.61
Material density, ρ_{mat}	7750 kg/m ³
Journal bearing radius, R_b	12.5 mm
Journal bearing length, L_b	25.0 mm
Valve Dimensions and Physical Properties	
Total length of valve reed, l_{dv}	45.0 mm
Diameter of valve reed circular section, d_{dv, x_1}	12.5 mm
Length of valve reed rectangular section, l_{dv, x_2}	26.0 mm
Width of valve reed rectangular section, w_{dv, x_2}	6.0 mm
Thickness of valve reed, t_{dv}	0.3 mm
Height of valve stop, y_{stop}	5.0 mm
Damping ratio, ξ	0.2
Density of valve reed material, ρ_{dv}	8000 kg/m ³
Young's modulus of valve reed material, E_{dv}	197.0 GPa
Clearances	
Radial clearance between rotor and cylinder, δ_{rad}	20.0 μ m
Radial clearance between eccentric and inner rotor surface, $\delta_{ecc, rad}$	25.0 μ m
Clearance between rotor end-face and cylinder endplate, δ_{ref}	20.0 μ m
Clearance between eccentric end-face and cylinder endplate, δ_{ecef}	5.0 mm
Clearance between vane bush end-face and cylinder endplate, δ_{vbef}	20.0 μ m
Clearance between journal and bearing, δ_b	25.0 μ m

The vane length and shoulder length are determined based on the geometry shown in Figure 5.1, the geometrical relation can be expressed by:

$$l_{vro} + l_{vbsd} > l_{v, max} \quad (6.1)$$

$$l_{v,\max} = 2r_{oc} - 2r_{ro} \quad (6.2)$$

Generally, both vane and shoulder lengths can be determined arbitrarily as long as Equation (6.1) is satisfied. However, in the practical design of U-Vane compressor, increasing the vane length would require a larger vane bush while increasing the shoulder length would require a thicker rotor wall. Hence, to account for the practicality and to ensure a smooth compressor operation, it is advised that the vane length (l_{vro}) should have a minimum length of about 90% of the maximum vane exposure length ($l_{v,\max}$) whereas the shoulder length (l_{vbsd}) should have a minimum length of about 25% of the maximum vane exposure length.

A step size independence test has been carried out before the final converged simulation results were accepted here for discussion. The power input to UVC is used as an assessing parameter. Figure 6.1 shows the result of the step size independence test where the power input and the computational time are normalized to the result obtained from the smallest step size. The result indicates that the power input starts to converge when the angular step sizes are below 0.009° and the discrepancies are less than 0.01 %. Hence, any step sizes that are below or equal to 0.009° are acceptable, as shown in Figure 6.2, where the discrepancies between the pressure profiles of each step size are imperceptible. To meet a compromise between converged result and the computational time, 0.009° is selected as the integration angular step size for the simulation study.

Figure 6.1 Result of step size independence test

Figure 6.2 Variation of pressure with different step sizes

6.2 Simulation Results and Discussions

The simulation is conducted under adiabatic condition as the effect of heat transfer on compressor performance at high operating speed (i.e. 3000 rev/min) is insignificant [170]. Basically, one operational cycle of U-Vane compressor takes two full revolutions to complete, and it includes suction process, compression process and discharge process. The variations of working chamber volume, working fluid pressure, mass of the working fluid inside the working chamber and the valve reed displacement over one operational cycle are shown in Figure 6.3. It is noted that the results shown in Figures 6.3 and 6.4 are simulated in perfect sealing condition, i.e. no internal leakage.

Figure 6.3 Basic simulation results: variations of (a) working chamber volume, (b) working fluid pressure, (c) working fluid mass inside the working chamber and (d) valve displacement

The variation of UVC working chamber volume over the entire operational cycle is shown in Figure 6.3a. During suction process, the volume increases until it reaches the maximum at 360° (one revolution). The angle where the suction process ends, and the compression process begins as the working chamber volume starts to decrease. The variations of working fluid pressure and mass of the processes are described in Figures 6.3b and c. During suction process in the first revolution, the working fluid pressure is maintained at the suction pressure while the working fluid mass increases gradually. It is observed that the working fluid pressure is dropping in the beginning until the angle reaches 25°, this is because of the radially positioned suction port on the cylinder wall, where the opening area is influenced by the radial contact between cylinder and rotor, as described in Section 4.5.1. Before the angle of about 25°, suction port is yet

uncovered and no fluid is being drawn into the suction chamber. Since the suction chamber volume continues to increase without the working fluid flowing into the chamber, the working fluid left in the dead volume of the compressor expands and results in the pressure drop. Subsequently, the suction chamber back to the suction pressure once the suction port is uncovered after 25° and the working fluid starts to flow into the suction chamber. After the angle of 360° , the working fluid undergoes the compression process. It is noted that the working fluid mass in the chamber remains unchanged as there is no more working fluid flowing into the working chamber from the suction port. However, at about 360° , a sudden increase of mass is noticed, which corresponds to the slight increase of pressure as shown in Figure 6.3b. Due to the dead volume and clearance in the compressor working chamber, an amount of pressurized working fluid is kept in the dead volume during the discharge process. Once the compressor starts a new cycle, the amount of high pressure working fluid re-expands into the compression chamber. However, as the suction port is still communicating with the compression chamber, the re-expanded working fluid flows out from the compressor through the suction port as the suction pressure is lower than the chamber pressure. This phenomenon is termed 'reversed flow'. Eventually, the working fluid pressure in the chamber reaches an equilibrium with the suction pressure, where the equilibriums in pressure is spotted in Figure 6.3b and the decrease in mass is shown in Figure 6.3c. After that, the compression process continues until the working fluid pressure reaches the desired discharge pressure at about 580° . However, the discharge process does not initiate instantaneously. Due to the valve reed stiffness and the flow resistance, the working fluid is compressed to a higher pressure in order to trigger the valve opening. Hence, a pressure rise is observed at about 600° , where the valve starts to open. The discharge process starts, and the working fluid begins to flow out from the compressor through the discharge port, as shown in the decreasing trend of the working fluid mass depicted in Figure 6.3c. The process lasts whenever the force is large enough to open the valve. Figure 6.3d shows the variation of valve displacement, which increases rapidly due to the extraordinary high differential pressure across the valve in the beginning of the discharge process. Then, it drops

as the pressure force acting on the valve is getting lower. It is noted that as soon as the radial contact line between cylinder and rotor passes through the discharge port, the discharge process may end too. Once the rotor moved passed the discharge port, the remaining working fluid in the chamber is further compressed since the size of the working chamber volume is still decreasing. This can be observed in Figures 6.3b and d, the pressure surges up and the valve reed displacement decreases after 700° . The compressor operational cycle ends at 720° .

Figure 6.4 Variation of (a) vane exposure length and rate of change of vane exposure length, (b) γ_1 and rate of change of γ_1

The kinematics of UVC over one rotation is shown in Figure 6.4. Initially, the vane exposure length increases until the maximum length is reached at 180° , then it decreases, as shown in Figure 6.4a. In the same figure, the rate of change of vane exposure length is also presented and it displays a sinusoidal form over one rotation. The sinusoidal form is again observed in the variation of the angle γ_1 shown in Figure 6.4b, while the rate of change of angle γ_1 increases until it reaches the maximum at 180° and decreases after that.

Figure 6.5 Variation of working fluid (a) pressure and (b) mass with and without internal leakage: solid lines – without leakage, dashed lines – with leakage

Figure 6.5 shows the instantaneous working fluid pressures and masses in the working chamber with and without the internal leakage. In Figure 6.5a, it is significant that the working fluid pressure after 700° is lower compared to that without internal leakage. It is also found that when the leakage occurs, both working fluid pressure and mass in the working chamber are slightly lower during the compression process (360° - 600°), as shown in Figures 6.5a and b.

The leakage mass flow rates of each of the leakage paths over one rotation are shown in Figure 6.6. It is observed that the leakages through radial clearance and rotor end-face clearance account for 49.9 % and 37.0 % of the total internal leakage, respectively. Figure 6.6 also shows that the internal leakages reach their peaks at about 240°, which is mainly due to the high pressure ratio between the compression and suction chambers. In addition, it is the moment when the working fluid is being compressed to overcome the valve reed stiffness and the

discharge pressure force acting on the back side of the valve. After that, the leakage mass flow rates through the end-face clearances decrease as a result of the decreasing flow channel ‘width’ of the end-face leakage paths. Unlike the end-face leakages, the leakage through the radial clearance follows the pattern of the pressure profile shown in Figure 6.3b because the radial clearance (flow channel width) does not vary during the operation. In short, higher pressure ratio and wider leakage path will result in more internal leakage flow.

Figure 6.6 Leakage mass flow rate through various leakage paths

Figure 6.7a shows the oil film thickness and the resultant load of journal bearing during the compressor operation. It is seen that the oil film thickness hit the minimum at about 240° , coincides with the maximum load in the same figure and the maximum pressure shown in Figure 6.7b. This indicates that at this instant, the journal bearing experiences maximum load from the compressor, and the corresponding amount of pressure to support the load is generated from squeezing the oil film. Furthermore, it is important to maintain the oil flow rate at this region in order to produce the adequate supporting pressure force. Otherwise, metal-to-metal contact may occur, which further causes undesirable high frictional loss or worse, leads to seizure of operation. From Figure 6.7b, the required oil flow rate is $1.6 \text{ cm}^3/\text{s}$ when the journal bearing is experiencing the maximum load at 240° , which suggests that the supplied oil flow rate must be more than $1.6 \text{ cm}^3/\text{s}$ in order to have a smooth operation. Hence, the lubricant

network of the compressor must be well-designed to sufficiently supply the lubricant into the compressor.

Figure 6.7 Journal bearing analysis: (a) minimum oil film thickness and compressor resultant load, (b) maximum pressure and required oil flow rate

Various frictional power losses are shown in Figure 6.8. Obviously, the largest frictional loss occurs at the vane sides ($L_{f,rvs}$). As seen from Figure 6.8a, it varies and reaches the peak of 500.0 W at 240° during the operation. It is because of the high pressure force from the working fluid acting on the vane, which in turn creates large normal contact forces between the vane sides and the vane slot, and results in high frictional loss. In the same figure, the frictional losses at the rotor end-face ($L_{f,ref}$) and the rubbing between the eccentric and the rotor ($L_{f,ecc}$) show comparatively steady trend at about 80.0 W and 50.0 W, respectively. The journal bearing friction ($L_{f,b}$) is shown in Figure 6.8b, with an average loss of about 30.0 W, whereas the vane bush side frictional loss varies in

a range from 0 W to 15 W ($L_{f,vbs}$) over the operational cycle. Figure 6.8c shows that the vane bush end-face friction ($L_{f,vbef}$) is below 0.35 W due to the small rubbing end-face area and the magnitude of relative motion with respect to the cylinder endplates (cover and base). As the eccentric axial length is made shorter than that of the compressor, the clearance gap distances between the eccentric end-faces and the cylinder endplates are large, i.e. 5.0 mm, causing the frictional loss at the eccentric end-face ($L_{f,ecef}$) negligible compared with the other frictional losses, which is about 0.05 W averagely as shown in Figure 6.8c.

Figure 6.8 Variation of various frictional power losses

Figure 6.9 Variation of components of total instantaneous power of UVC

Figure 6.9 presents the variations of various instantaneous power components over one revolution. It is observed that the total instantaneous power ($\dot{W}_{Inst}$) increases initially, reaches the maximum of about 3500.0 W at 240°, and decreases after that. The compression power ($\dot{W}_{com}$) follows the same profile pattern as the total instantaneous power. Thus, it is suggested that the total instantaneous power is mainly affected by the compression power. The total frictional power loss ($\dot{W}_{f,loss}$) shows a steady trend of about 200.0 W at the first half and increases at the second half of the rotation. With reference to Figure 6.8, it is seen that the rotor end-face friction, the friction between the eccentric and the inner rotor surface, and the journal bearing friction lead at the first half, while the vane side friction dominates at the second half of the rotation.

Figure 6.10 exhibits the compositions of the total frictional power loss and power input. In Figure 6.10a, the vane side friction is the largest frictional loss, which accounts for 47.8 % of the total frictional loss. It is followed by the frictional losses at the rotor end-face, the rubbing between the eccentric and the inner rotor surface, and the journal bearing, which contribute 23.7 %, 14.4 % and 11.3 %, respectively, to the total frictional loss. Overall, the total frictional power loss and the compression power devote about 24.3 % and 75.7 % to the power input as shown in Figure 6.10b.

Figure 6.10 Components of (a) average frictional power loss and (b) power input

Table 6.2 Predicted overall performance of UVC (R1234-yf)

Overall Performance (Predicted)	
Compression efficiency, η_{comp}	94.46 %
Volumetric efficiency, η_{vol}	96.28 %
Mechanical efficiency, η_{mech}	75.71 %
Coefficient of performance, COP	3.27
Total mass flow rate, $\dot{m}_{actual}$	31.32 g/s
Power input, $\dot{W}_{input}$	1129.24 W

The predicted overall performance of UVC is tabulated in Table 6.2. The compression and volumetric efficiencies of the compressor are 94.46 % and 96.28 %, respectively. The high compression efficiency denotes that the dimensions of the reed valve, the suction and discharge ports are appropriate. Furthermore, due to the tight clearances configuration in the simulation, the internal leakage flow is relatively insignificant with respect to the total mass flow rate and thus, the volumetric efficiency is high. The mechanical efficiency of the compressor is 75.71 %, which indicates that the frictional losses are the main source of energy loss in the compressor. The predicted performances of the compressor with various refrigerants under same configurations as listed in Table 6.1, are compared in Table 6.3.

Table 6.3 Comparison of predicted performance with various refrigerants

	R-1234yf	R-32	R-134a
Compression efficiency, η_{comp}	94.46 %	96.63 %	95.18 %
Volumetric efficiency, η_{vol}	96.28 %	95.15 %	96.18 %
Mechanical efficiency, η_{mech}	75.71 %	81.91 %	75.73 %
Coefficient of performance, COP	3.27	3.35	3.41
Power input, $\dot{W}_{input}$	1129.24 W	2691.53 W	1144.87 W

The compression and volumetric efficiencies are found similar in the results presented in Table 6.3, which are around 95 %. Compressor with R-32 as the working fluid has the highest mechanical efficiency, which is 81.91 %, whereas both R-1234yf compressor and R-134a compressor have identical mechanical efficiencies of 75.71 % and 75.73 %, respectively. Although the mechanical efficiency of the compressor with R-32 is higher than the one with R-1234yf and R-134a, the power input of 2691.53 W is required to drive a R-32 compressor under the same operational conditions. It is more than twice the power input to the other two compressors, as shown in Table 6.3. Hence, it is suggested that in the applications that power consumption is not the primary concern, R32 compressor has the best overall performance among the compressors listed in Table 6.3.

6.3 Parametric Study

After analysed the simulation result, several design and operational parameters are selected to investigate their effects on the compressor performance. Unless otherwise stated, the operational conditions and compressor dimensions follow that in Table 6.1.

6.3.1 Effect of Operating Speed

The effect of operating speed on compressor performance is investigated by varying the speed from 1000 rev/min to 5000 rev/min. As shown in Figure 6.11a, the compression efficiency is reduced from 99.1 % to 88.5 %, which is

mainly caused by the increase of valve loss as shown in Figure 6.11b, where higher suction and discharge losses are noticed at the operating speed of 5000 rev/min, compared to that of 1000 rev/min. Furthermore, higher operating speed also increases the vane sliding velocity and the rate of change of angle γ_1 , causing the friction to increase dramatically and reduces the mechanical efficiency by 16.8 %. However, the volumetric efficiency improved by 8.6 % when the operating speed increases from 1000 rev/min to 5000 rev/min, which is due to a shorter cycle time given to the leakages to occur.

Figure 6.11 Effect of operating speed on the performance of UVC: (a) efficiencies and (b) variation of working fluid pressure

6.3.2 Effect of Discharge Pressure

The effect of discharge pressure on compressor performance is inspected while the working fluid is air. The main reason that air is selected as the working fluid is to isolate the effect of discharge pressure from the effects caused by other operational parameters, and to examine the effect of the desired inspection range of discharge-to-suction pressure ratios (i.e. desired pressure ratio: 2.0 to 10.0) on the compressor performance. Since for R-1234yf, the suction and discharge pressures are determined from the dew point temperatures, which is constrained by the range of the temperatures given in the REFPROP database [125]. In this case, both suction and discharge pressures may need to be varied in order to achieve the desired inspection range of pressure ratio, which may result in unjustifiable effect on the compressor performance as the change in suction condition may alter the mass flows into the compressor. By using air as the

working fluid, the suction pressure is remained constant at 1.0 bar (abs.) and the discharge pressure is varied from 2.0 bar (abs.) to 10.0 bar (abs.), i.e. pressure ratio of 2.0 – 10.0. Figure 6.12 shows the effect of the discharge pressure on the compressor performance. It is found that the mechanical efficiency increased by 38.0 % when the discharge pressure varies from 2.0 bar to 10.0 bar. From the definition in Equation (5.60), the increase in the discharge pressure requires more compression power to the compressor and thus, results in the increase of mechanical efficiency. In addition, although frictional loss also increases because of the larger pressure force acting on the components, its increment is smaller than that in the compression power. The volumetric efficiency decreased by 5.0 % since the internal leakage of a compressor is driven by the pressure ratio. The increment in the discharge pressure causes larger pressure difference and subsequently, results in more leakages from the compression chamber to the suction chamber. As shown in Figure 6.12b, the leakage mass flow rate is increasing with discharge pressure. An improvement of 7.0 % is observed in the compression efficiency when the discharge pressure increases to 10.0 bar. As shown in Figure 6.13, the over-compression rate is about 20.0 % of the total pressure rise when it is 2.0 bar, while it is only 12.0 % when the discharge pressure is 10.0 bar. At low discharge pressure, the discharge valve stiffness becomes the primary resistance during the discharge process. Thus, the working fluid is compressed to a higher pressure in order to overcome the valve stiffness, which incurs higher discharge loss relative to the total compression power.

Figure 6.12 Effect of discharge pressure on the performance of UVC: (a) efficiencies and (b) total leakage mass flow rate

Figure 6.13 Variations of pressure and valve deflection when the discharge pressure is (a) 2.0 bar (abs.) and (b) 10.0 bar (abs.)

6.3.3 Effect of Vane Length

In Section 6.2, the vane side frictional loss is realised to be the largest friction component. Thus, it is essential to investigate the effect of vane length on the compressor performance. Figure 6.14 shows the compressor performance when the vane length increases from 8.0 mm to 20.0 mm.

Figure 6.14 Effect of vane length on the performance of UVC

It is found that in Figure 6.14, varying the vane length has no effect on the volumetric and compression efficiencies. However, the mechanical efficiency increased by 8.7 % when the vane is lengthened from 8.0 mm to 20.0 mm. It is suggested that a longer vane effectively reduces the normal forces acting on the vane slot from the vane sides and the vane tip, which lower the corresponding

frictional loss, as presented in Figure 6.15. Figure 6.15a compares the normal forces when the vane lengths are 8.0 mm and 20.0 mm, while Figure 6.15b shows the variations of vane side frictions. In practical design, a longer vane requires the bush to be bigger as it houses the vane during operation, which may lead to larger compressor cylinder and housing.

Figure 6.15 Variations of (a) vane side normal force and (b) friction between vane side and vane slot ($L_{f,rvs}$) with vane length

6.3.4 Effect of Vane Bush Shoulder Length

The effect of the vane bush shoulder length on compressor performance is also examined and presented in Figure 6.16. It is found that the shoulder length poses similar effect as the vane length, where it has insignificant effect on the volumetric and compression efficiencies. Conversely, the increment of the shoulder length from 4.0 mm to 12.0 mm results in an improvement on the mechanical efficiency from 75.6 % to 78.1 %.

Figure 6.16 Effect of the vane bush shoulder length on the performance of UVC:
(a) efficiencies and (b) vane side frictional loss ($L_{f,rvs}$)

As seen from Figure 6.16b, the friction between the vane sides and the vane slot decreases when the shoulder is getting longer. This is because of the reduction in the normal force as discussed in Section 6.3.3. However, a long shoulder implies that a thick rotor wall is required in order to accommodate the long bush shoulder during operation, which may result in additional frictional losses such as the end-face friction. Therefore, the effect of the rotor wall thickness must be inspected.

6.3.5 Effect of Rotor Wall Thickness

In general, to maintain the compressor swept volume, the outer rotor diameter is kept constant and thus, varying the rotor wall thickness will result in the change in the inner rotor diameter and the eccentric diameter. This results in double-sided effect on the frictional losses as depicted in Figure 6.17. By increasing the rotor wall thickness, the rotor end-face friction increases due to the larger rubbing end-face area, but the frictional loss between the eccentric and the inner rotor surface decreases thanks to the smaller rubbing surface area. Ultimately, the decrease of the latter leads, and results in overall reduction of the total frictional loss and the improvement on the mechanical efficiency. As shown in Figure 6.18, by thickening the rotor wall from 9.0 mm to 17.0 mm, the mechanical efficiency increased by 3.6 %. The wall thickness has no influence on the compression and volumetric efficiencies. In short, by thickening the rotor wall, the effect on the reduction of the frictional loss between the eccentric and

inner rotor surface is more significant than the increase of the rotor end-face friction, given the outer rotor diameter unchanged.

Figure 6.17 Variations of (a) rotor end-face friction ($L_{f,ref}$) and (b) frictional loss between eccentric and inner rotor surface ($L_{f,ecc}$) with rotor wall thickness

Figure 6.18 Effect of rotor wall thickness on the performance of UVC

6.3.6 Effect of Rotor-to-Cylinder Radius Ratio

Figure 6.19 shows the effect of the rotor-to-cylinder radius ratio on the performance of UVC. It is observed that the mechanical efficiency is drastically increased by 22.3 % when the ratio varies from 0.78 to 0.94, while the effect on the compression efficiency is negligible, as seen in Figure 6.19a. It is also noted that the volumetric efficiency is lower at high rotor-to-cylinder radius ratio.

Basically, a low rotor-to-cylinder radius ratio will increase the compressor volumetric displacement. Thus, to keep the volumetric displacement same, the axial length of the compressor will be reduced. Therefore, the compressor configuration can be in thin and long or fat and short under the condition of constant volumetric displacement. As a result, for high rotor-to-cylinder radius ratio, the compressor axial length will be long, which leads to the increase of the leakage through the radial clearance and eventually reduces the volumetric efficiency. The improvement of the mechanical efficiency is found to be caused by the reductions of various frictional losses. As portrayed in Figure 6.19b, the vane side friction and the rotor end-face friction dropped by 70.8 % and 88.9 %, respectively, when the rotor-to-cylinder radius ratio varies from 0.78 to 0.94. In fact, it is the result of the changes in the compressor kinematics as shown in Figure 6.20. By having high rotor-to-cylinder radius ratio, both rates of changes of vane exposure length and angle γ_1 , i.e. $\frac{dl_v}{dt}$ and $\frac{d\gamma_1}{dt}$, decrease significantly, and subsequently reduce the frictional losses.

Figure 6.19 Effect of rotor-to-cylinder radius ratio on the performance of UVC:
(a) efficiencies and (b) various frictional losses

Figure 6.20 Variations of (a) rate of change of vane length and (b) rate of change of angle γ_l with rotor-to-cylinder radius ratio

6.3.7 Effect of Slenderness Ratio of Journal Bearing

Journal bearing is one of the crucial components in the compressor operation. Especially the minimum oil film thickness $h_{\min}$, which determines the allowable surface roughness of the shaft. Thus, the effect of the design parameter of journal bearing, particularly the slenderness ratio (L_b/D_b), is studied. It is noted that the slenderness ratio affects the frictional loss only in the present mathematical model and hence, the effect on the compression and volumetric efficiencies are not considered. Figure 6.21 shows the changes of mechanical efficiency and journal bearing friction when the slenderness ratio varies from 0.5 to 1.5. Obviously, the journal bearing friction increases from 15.9 W to 46.0 W, and results in the decrement of the mechanical efficiency from 76.7 % to 74.7 %. Principally, a high slenderness ratio enlarges the lubricated surface area at the journal bearing, which in turn increases the journal bearing friction. The effect of slenderness ratio on the minimum oil film thickness is shown in Figure 6.22. It is found that the smallest minimum oil film thickness increases with the slenderness ratio because of the bigger lubricated surface area to undertake the load from the rotating shaft. It is concluded that in current compressor configuration, for every 0.25 increment in the slenderness ratio, the minimum oil film thickened by about 0.002 μm and the journal bearing friction increased by 7.5 W.

Figure 6.21 Effect of slenderness ratio on the performance of UVC

Figure 6.22 Effect of slenderness ratio on (a) minimum oil film thickness and (b) variations of oil film thickness

6.4 Comparison of the Performance between UVC and Existing Rotary Compressors

The mechanical efficiency of U-Vane compressor is compared with that of existing rotary compressor mechanisms as shown in Table 6.4. The comparison of volumetric efficiency is shown in Table 6.5.

Table 6.4 Comparison of mechanical efficiency between UVC and existing rotary compressors

Mechanism	Mechanical Efficiency (%)	Investigator
Scroll	80.0	Ishii <i>et al.</i> [101]
Scroll	92.3 – 93.4	Ishii <i>et al.</i> [216-218]
Rolling piston	85.0 – 90.0	Ozu and Itami [203]
Rolling piston	94.0	Matsuzaka and Nagatomo [73]
Swing	74.0	Masuda <i>et al.</i> [14]
Swing vane	92.0	Hu <i>et al.</i> [23]
Revolving vane	94.1	Teh and Ooi [219]
UVC	75.7 – 81.9	Present work

Table 6.5 Comparison of volumetric efficiency between UVC and existing rotary compressors

Mechanism	Volumetric Efficiency (%)	Investigator
Scroll	80.0	Ishii <i>et al.</i> [220]
Rolling piston	94.0	Matsuzaka and Nagatomo [73]
Rolling piston	90.0	Yanagisawa and Shimizu [79]
Swing	80.0 – 94.0	Masuda <i>et al.</i> [14]
Revolving vane	95.0	Teh and Ooi [112]
UVC	90.0 – 95.0	Present work

The comparison above indicates that the performance of UVC is akin to the performance of other existing rotary compressor mechanisms. Since a vane compartment is not required in UVC, the internal leakage flow through the vane side to the vane compartment is eliminated and hence, results in higher volumetric efficiency compared with other compressors. However, the mechanical efficiency of UVC is slightly lower due to the larger contact forces between the vane sides and the vane bush shoulders. It is believed that through optimizing the design parameters of the UVC, higher mechanical efficiency of the compressor can be achieved.

6.5 Concluding Remarks

The simulation operational parameters and compressor configuration have been presented. A step size independent test has also been conducted and 0.009° is selected as the angular step size for the simulation study. The simulation results, including the variation of working chamber volume, working fluid thermodynamic properties, kinematics of the compressor and internal leakage have been presented and analysed. Various mechanical losses, instantaneous power components and the compressor performance have also been shown and discussed. The effects of several parameters have been investigated:

- Higher operating speed increases the volumetric efficiency but reduces the compression and mechanical efficiencies. By increasing the operating speed from 1000 rev/min to 5000 rev/min, the volumetric efficiency increased by 8.6 % while the compression and mechanical efficiencies dropped by 10.7 % and 16.8 %, respectively.
- Using air as the working fluid and maintaining the suction pressure at 1.0 bar (abs.), the increment of discharge pressure from 2.0 bar (abs.) to 10.0 bar (abs.) increases the mechanical and compression efficiencies by 38.0 % and 7.0 %, respectively. However, the volumetric efficiency decreased by 5.0 %.
- Lengthening the vane from 8.0 mm to 20.0 mm has no effect on compression and volumetric efficiencies. However, mechanical efficiency improved by 8.7 %, which is mainly due to the reduction in friction occurs at the vane side.
- Like the effect of vane length, lengthening the vane bush shoulder from 4.0 mm to 12.0 mm increases the mechanical efficiency by 3.3 %.
- To keep the volumetric displacement constant, the outer rotor diameter remains unchanged and hence, thickening the rotor wall changes the inner rotor diameter and the eccentric diameter. It is found that thickening the rotor wall from 9.0 mm to 17.0 mm does

not affect compression and volumetric efficiencies but improves the mechanical efficiency by 3.6 %.

- Varying the rotor-to-cylinder radius ratio from 0.78 to 0.94 noticeably increases the mechanical efficiency by 22.3 %. However, the effect on the compression and volumetric efficiencies is insignificant.
- The increment of slenderness ratio (L_b/D_b) of the journal bearing from 0.5 to 1.5 reduces the mechanical efficiency by 2.6 %. This is predominantly caused by the change in the journal bearing friction, which increases from 15.9 W to 45.9 W.

In next chapter, the design of the UVC prototype will be presented.

Chapter 7 Design of Prototype

The performance of the U-Vane compressor will be investigated experimentally thus it is necessary to fabricate a prototype of the compressor. In this chapter, the detailed design of UVC prototype and the lubrication system will be presented.

7.1 Design Specifications of Prototype

The specifications of the prototype are tabulated in Table 7.1:

Table 7.1 Specifications of prototype

Specifications of Prototype	
Working fluid	Air
Suction pressure	1.0 bar* (abs.)
Discharge pressure	7.0 bar (abs.)
Suction temperature	Room temperature**
Dynamic viscosity of lubricant, μ_{lub} [212]	0.0129 Pa·s
Volumetric displacement, V_{total} [213, 214]	33.5 cm ³ /rev
Operating speed, ω_1	3000 rev/min (RPM)

* The suction pressure is the atmospheric pressure in the laboratory, which is 1.0 bar (abs.) averagely.

** Room temperature refers to the temperature in the laboratory.

Theoretically, a compressor can operate with any working fluids, thus, air is chosen to be the working fluid in the experiment as it is available abundantly. A discharge pressure of 7.0 bar (abs.) is designated as it is a typical pressure rating for air compressor, i.e. 7.0 abs. bar [221]. Although the volumetric displacement of the compressor can be varied over a wide range depends on the actual application, 33.5 cm³/rev is selected as the volumetric displacement for the prototype to test its feasibility in application such as air-conditioning. The designated dimensions of major components in UVC prototype are summarised in Table 7.2.

Table 7.2 Dimensions of major components

Dimensions of Major Components	
Shaft radius, r_s	12.5 mm
Eccentric radius, r_{ecc}	17.0 mm
Eccentric axial length, l_{ecc}	25.0 mm
Cylinder radius, r_{cyl}	36.0 mm
Rotor radius, r_{ro}	31.5 mm
Rotor wall thickness, t_{ro}	14.5 mm
Vane length, l_{vro}	43.0 mm from the rotor centre
Vane width, w_{vro}	6.0 mm
Vane bush radius, r_{vb}	9.0 mm
Vane bush shoulder length, l_{vbsd}	6.34 mm measured from vane bush surface
Compressor axial length, l_{com}	35.0 mm
Valve Dimensions	
Total length of valve reed, l_{dv}	45.25 mm
Diameter of valve reed circular section, d_{dv,x_1}	12.5 mm
Length of valve reed rectangular section, l_{dv,x_2}	25.45 mm
Width of valve reed rectangular section, w_{dv,x_2}	6.0 mm
Thickness of valve reed, t_{dv}	0.3 mm

7.2 Design of Prototype

The UVC prototype consists of 10 components, including shaft, eccentric (attached to shaft as one component), rotor, vane bush, cylinder assembly (3 components - cylinder, reed valve, lower bearing), housing assembly (3 components - base plate, housing and cover) and counterweight. The detailed designs of the components are presented in this section.

7.2.1 Shaft and Eccentric

The shaft and eccentric are the driving components of the UVC prototype, where the shaft is coupled to an electric motor during operation. The eccentric is kept at an offset distance (eccentricity, e) from the shaft. They are designed in one body to reduce the transmission loss of torque from the motor to the eccentric through the shaft, as shown in Figure 7.1. Spiral oil groove is cut on the surfaces of both the eccentric and the lower part of the shaft (lower journal). The lower shaft is submerged in the oil sump, when the shaft rotates, the spiral groove drags up the lubricating oil from the oil sump for the rubbing components. A vertical oil feeding hole is designed at the centre of the shaft with an opening at the bottom of the shaft. It connects with the radial oil feeding hole at the upper part of shaft (upper journal) so that the upper journal can be lubricated.

Figure 7.1 Shaft and eccentric: (a) isometric views (b) detailed side view and bottom view

During operation, the shaft is subjected to high pressure force and torsion since it is exposed to the high pressure working fluid, which may lead to material failure if the dimension is rashly determined. Thus, it is essential to examine the minimum allowable shaft diameter to prevent the shaft from material failure.

Figure 7.2 Free body diagram of forces acting on shaft and eccentric

Figure 7.2 shows the free body diagram of forces acting on the shaft and eccentric, where $F_{l,b}$ and $F_{u,b}$ represent lower and upper bearing support forces, which are unknowns, while F_r represents the resultant load from the compressor. When the working fluid pressure difference between the compression and suction chambers is the highest, the shaft is subjected to the highest resultant load, which is 1244.8 N as obtained from the theoretical simulation. By applying balance of moment at points A and C and substituting the values, where $l_{l,b} = 25.0$ mm, $\delta_{l,b} = 5.0$ mm, $l_{ecc} = 25.0$ mm, $\delta_{u,b} = 5.0$ mm, $l_{u,b} = 25.0$ mm and $l_1 = 114.5$ mm, $F_r = 1244.8$ N, $F_{l,b}$ and $F_{u,b}$ can be calculated as follows:

$$\begin{aligned}
 F_{u,b} &= \frac{\frac{1}{2}l_{ecc} + \delta_{l,b} + \frac{1}{2}l_{l,b}}{\frac{1}{2}l_{u,b} + \delta_{u,b} + \frac{1}{2}l_{ecc} + \delta_{l,b} + \frac{1}{2}l_{l,b}} \times F_r \\
 &= \frac{30.0}{60.0} \times 1244.8 \\
 &= 622.4 \text{ N}
 \end{aligned}$$

$$\begin{aligned}
F_{l,b} &= \frac{\frac{1}{2}l_{ecc} + \delta_{u,b} + \frac{1}{2}l_{u,b}}{\frac{1}{2}l_{l,b} + \delta_{l,b} + \frac{1}{2}l_{ecc} + \delta_{u,b} + \frac{1}{2}l_{u,b}} \times F_r \\
&= \frac{30.0}{60.0} \times 1244.8 \\
&= 622.4 \text{ N}
\end{aligned}$$

Once the forces are obtained, the shear force and bending moment diagrams can be determined as shown in Figure 7.3.

Figure 7.3 Shear force diagram and bending moment diagram of shaft and eccentric

From the diagram, the maximum magnitude of shear force and bending moment are 622.4 N and 18.7 Nm, respectively. In addition, the maximum torsion that the shaft experiences corresponds to the highest resultant load is found to be 6.1 Nm as calculated from simulation. With these critical loads, the following equations can be used for determining the minimum diameter of the shaft [222]:

$$d = \sqrt{\frac{2.94K_t V N}{s_n}} \quad (7.1)$$

$$d = \left[\frac{32N}{\pi} \sqrt{\left(\frac{K_t M}{s_n'} \right)^2 + \frac{3}{4} \left(\frac{T}{s_y} \right)^2} \right]^{\frac{1}{3}} \quad (7.2)$$

$$s_n' = s_n C_s C_R \quad (7.3)$$

where,

K_t is the stress concentration factor (-)

V is the shear force experienced by the shaft (N)

N is the safety factor, SF

M is the bending moment experienced by the shaft (Nm)

T is the torsion (Nm)

s_n' is the modified endurance strength (Pa)

s_y is the yield strength (Pa)

s_n is the estimated endurance strength (Pa)

C_s is the size factor (-)

C_R is the reliability factor (-)

Since the shaft will be made of the material, 17-4 PH stainless steel [223], the mechanical properties of the material tabulated in Table 7.3 is applied.

Table 7.3 Mechanical properties of 17-4 PH stainless steel [223]

Properties		Conditions	
		T (°C)	Treatment
Density (kg/m ³)	7750	25	Solution annealed
Poisson's Ratio	0.3		
Elastic Modulus (GPa)	197.0		
Tensile Strength (MPa)	1069.31		
Yield Strength (MPa)	1020.41		
Elongation (%)	14.78		
Reduction in Area (%)	36.91		
Hardness (Rockwell C)	31		

Table 7.4 Approximate reliability factors [222]

Desired Reliability	C_R
0.50	1.0
0.90	0.9
0.99	0.81

Figure 7.4 Endurance strength versus tensile strength for wrought steel [222]

Figure 7.5 Size factors for various shaft diameters [222]

The calculation of the diameters at point A and point C can be facilitated by assuming a sharp fillet at the edge of shaft and eccentric, so that a stress concentration factor (K_t) of 2.5 is used [222]. For steel with machined surface condition, the endurance strength (s_n) is 360 MPa as indicated in Figure 7.4. From Figure 7.5, the design size factor (C_s) is initially assumed to be 0.8 as the actual shaft diameter is yet to be decided. Due to a required reliability of 0.99, the reliability factor (C_R) is 0.81 from Table 7.4, while the safety factor (N) is set at 3 since the prototype will be operating under typical industrial conditions [222]. Hence, from Equation (7.3), the modified endurance strength (s_n') can be calculated as follows:

$$\begin{aligned} s_n' &= 360 \times 0.8 \times 0.81 \\ &= 233.28 \text{ MPa} \end{aligned}$$

The known values are substituted into Equation (7.1), and the first estimated shaft diameter at point A can be calculated:

$$\begin{aligned} d_A &= \sqrt{\frac{2.94 \times 2.5 \times 622.4 \times 3}{233.28 \times 10^6}} \\ &= 7.67 \text{ mm} \end{aligned}$$

The above calculations are reiterated with the new size factor (C_s) of 1.0 based on Figure 7.5 and the updated modified endurance strength of 291.6 MPa. Thus, the final shaft diameter at point A is:

$$\begin{aligned} d_A &= \sqrt{\frac{2.94 \times 2.5 \times 622.4 \times 3}{291.6 \times 10^6}} \\ &= 6.86 \text{ mm} \end{aligned}$$

The diameter at point A is determined again with Equation (7.2). by substituting the value in with the maximum torsion of 6.1 Nm and the yield strength (s_y) from Table 7.3, the diameter is obtained as:

$$\begin{aligned} d_A &= \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.5(0)}{291.6 \times 10^6} \right)^2 + \frac{3}{4} \left(\frac{6.1}{1020.41 \times 10^6} \right)^2} \right]^{\frac{1}{3}} \\ &= 5.41 \text{ mm} \end{aligned}$$

The diameters obtained from both equations are compared and the bigger diameter is selected as the minimum allowable diameter. Hence, 6.86 mm is the minimum allowable diameter for point A. The procedure is repeated for point B, C and D, and the minimum allowable at these points are: $d_B = 17.28$ mm, $d_C = 6.86$ mm, $d_D = 5.41$ mm. Overall, the minimum required shaft diameter is 17.28 mm. In the final prototype design, the shaft diameter is 25.0 mm while the eccentric (point B) diameter is 34.0 mm in order to accommodate the lubrication channel such as the vertical oil feeding hole and the spiral oil groove.

7.2.2 Rotor

The rotor and the vane are made as one component, as shown in Figure 7.6.

Figure 7.6 Isometric views of rotor: (a) view from top-front and (b) view from bottom-back

As the inner rotor surface is rubbing against the eccentric, a bushing made by aluminium bronze is press-fitted into the inner rotor surface to protect the rotor and the eccentric from the friction welding between the two rubbing surfaces. Aluminium bronze is used because it has excellent wear resistance and low coefficient of friction [224, 225]. The detail design of the rotor is described in Figure 7.7. Basically, the geometry of the rotor can be described by a cylindrical portion and a protruded vane. As the geometry is irregular, the centre of gravity (C.G.) is not aligned with the centre of the rotor “R”. Thus, two balancing holes are designed so that the C.G. of the rotor is aligned with the centre of the rotor. Furthermore, oil relief grooves are made at the shoulder slots so that the

lubricating oil will not get trapped during operation. This can avoid high pressure from oil squeezing action when the shoulder retracts into the slots. In fact, this oil pressure is undesirable as it causes excessive load to the shaft, which may lead to journal bearing seizure due to metal-to-metal contact under excessive load at the journal bearing.

Figure 7.7 Detailed design of rotor

During operation, as the vane is exposed to high pressure force from the working fluid, it is crucial to determine the vane width so that the material is strong enough to withstand the high pressure force. With reference to the force analysis in Figure 5.3, assuming cantilever beam for the vane and neglecting the radial vane tip, the free body diagram is shown below [226]:

Figure 7.8 Free body diagram of vane

The axial length of the compressor is 35.0 mm while the total length of the vane (length OB) is 17.0 mm. The vane width is determined by considering the most critical condition: when the vane is fully protruded, i.e. $OA = 9.5$ mm and $AB = 7.5$ mm and exposing to the maximum differential pressure on the vane, i.e.

7.0 bar discharge pressure and 1.0 bar suction pressure, the distributed pressure forces $F_{rv,press}$ can be calculated by:

$$\begin{aligned} F_{rv,press} &= (7.0 \times 10^5 - 1.0 \times 10^5) \times 0.035 \times (0.0095 + 0.0075) \\ &= 357.0 \text{ N} \end{aligned}$$

$F_{1,n}$ and $F_{2,n}$ are obtained from simulation, which are 569.8 N and 303.8 N, respectively. Taking force balance in vertical axis, F_{rv} can be calculated:

$$\begin{aligned} F_{rv} + F_{1,n} &= F_{rv,press} + F_{2,n} \\ F_{rv} &= 91.0 \text{ N} \end{aligned}$$

Also, taking the equilibrium of moments about O , the moment M_{rv} is obtained:

$$\begin{aligned} M_{rv} + (0.017) F_{1,n} &= \left(\frac{0.017}{2} \right) F_{rv,press} + (0.0095) F_{2,n} \\ M_{rv} &= -3.8 \text{ Nm} \end{aligned}$$

With these findings, the shear force and bending moment diagrams are found and presented in Figure 7.9, where the maximum bending moment (M_{max}) is calculated as 3.96 Nm.

Figure 7.9 Shear force diagram and bending moment diagram of vane

The maximum bending moment is used to determine the vane width using elastic flexure formula of a beam [226], as expressed in following equations:

$$\sigma_{max} = \frac{M_{max}}{S_{vro}} \quad (7.4)$$

$$S_{vro} = \frac{w_{vro} l_{vro}^2}{6} \quad (7.5)$$

where,

σ_{max} is the maximum stress on the vane (Pa)

S_{vro} is the elastic section modulus of the vane with rectangular cross-section (m³)

The maximum stress that the vane can tolerate is the yield strength as listed in Table 7.3, which is 1020.41 MPa. Introducing the reliability factor (C_R) and the safety factor (N) with the values of 0.81 and 3, respectively, the minimum vane width can be determined:

$$\begin{aligned} w_{vro} &= \frac{6M_{max}}{\sigma_{max} l_{vro}^2} \times \frac{N}{C_R} \\ w_{vro} &= \frac{6(3.96)}{1020.41 \times 10^6 \times 0.017^2} \times \frac{3}{0.81} \\ &= 0.298 \text{ mm} \end{aligned}$$

For easier fabrication and to reduce the deflection, the vane width in the prototype is designed with 6.0 mm. The stress and deflection analyses based on the 6.0 mm vane width are simulated in SolidWorks Simulation 2017 and displayed in Figure 7.10. The simulation assumes that the vane is experiencing the working fluid pressure force only without the support forces from the bush. Therefore, the vane is experiencing both maximum stress and deflection. The results show that the maximum stress, 11.36 MPa experienced by the vane is below the yield strength (1020.41 MPa), while the maximum deflection of 2.676 μm at the vane tip is within the designed tolerance of 10.0 μm .

Figure 7.10 (a) Stress analysis of vane and (b) deflection of vane

7.2.3 Vane Bush

The vane bush accommodates the vane and guides the swaying motion of the rotor during operation, as shown in Figure 7.11. The vane bush is made in one-piece so that the material itself can provide support to prevent the shoulders from collapsing.

Figure 7.11 Isometric views of vane bush: (a) view from top-front and (b) view from bottom-back

Figure 7.12 Detailed design of vane bush

The detail design of the vane bush is illustrated in Figure 7.12. The tips of the shoulders are chamfered by 0.6 mm to ease the sliding of the shoulders into the slot in the rotor, and to help the residual lubricant and working fluid to escape through the oil relief grooves on the rotor. An oil relief hole is made at the back of the vane bush so that the lubricant accumulated in the vane slot can discharge through the hole. Subsequently, the lubricant flows into the working chamber through the oil relief channel at the bottom end-face of the vane bush. It is noted that the shoulders are subjected to high pressure force from the working fluid in the working chamber, thus, the shoulder width must be analysed and determined.

With reference to the force analysis in Figure 5.2, the analyses of the left and right shoulders are done separately at fully protruded vanes position and are subjected to the distributed pressure forces shown in Figure 7.13. $F_{vbsd,suc}$ and $F_{vbslot,1}$ are forces caused by the suction pressure of 1.0 bar, $F_{vbsd,dis}$ is the force caused by the discharge pressure of 7.0 bar and $F_{vbslot,2}$ is assumed to be half of the difference between suction and discharge pressures, i.e. 3.0 bar. When the vane is at fully protruded position, Length AB and OB are 5.50 mm and 6.34 mm while $F_{n,1}$ and $F_{n,2}$ are obtained from simulation, which are 569.8 N and 303.8 N, respectively.

Figure 7.13 (a) Forces acting on vane bush; (b) free body diagram of left shoulder and (c) free body diagram of right shoulder

By taking balance of vertical forces and balance of moments about point O , $F_{vbsd,l}$, $M_{vbsd,l}$, $F_{vbsd,r}$ and $M_{vbsd,r}$ are calculated as -587.44 N, -0.49 Nm, 215.04 N and 1.64 Nm, respectively. The shear force and bending moment diagrams for left and right shoulders are shown in Figure 7.14.

Figure 7.14 Shear force and bending moment diagrams of (a) left shoulder and (b) right shoulder

The maximum bending moment ($M_{\max}$) acting on the right shoulder is found to be 1.64 Nm, as shown in Figure 7.14b. This is used for the determination of the shoulder widths at both sides. Applying Equations (7.4) and (7.5), with the reliability factor $C_R = 0.81$, safety factor $N = 3$, and the yield strength from Table 7.3, the shoulder width can be determined:

$$w_{vbsd} = \frac{6M_{\max}}{\sigma_{\max} l_{vbsd}^2} \times \frac{N}{C_R}$$

$$w_{vbsd} = \frac{6(1.64)}{1020.41 \times 10^6 \times 0.00634^2} \times \frac{3}{0.81}$$

$$= 0.89 \text{ mm}$$

where,

w_{vbsd} is the shoulder width (m)

The calculated vane bush shoulder width is deemed to be the width with enough strength to withstand the forces acting on it, but it would be too small for actual application as it may result in unfavourable deflection. In the prototype, the shoulder width is designed to be 4.0 mm to minimize the effect of deflection. To examine the deflection when the shoulder width is 4.0 mm, two extreme scenarios are considered and simulated in SolidWorks Simulation 2017. Scenario 1 postulates that the vane slot is filled by the working fluid with pressure of 7.0 bar with the normal forces acting on the shoulders by the vane. Scenario 2 keeps the normal forces too, but the working fluid pressure in the vane slot is 1.0 bar. Figure 7.15 shows the simulation result of Scenario 1, the maximum stress is 185.90 MPa at the oil relief hole, while the maximum deflection is 35.77 μm at the left shoulder tip. The result is similar to Scenario 2 demonstrated in Figure 7.16. The maximum stress is 141.70 MPa at the oil relief hole whereas the maximum deflection is 29.80 μm at the left shoulder tip. From the simulation results, the maximum stresses experienced by the vane bush in both scenarios are much lower than the yield strength of 1020.41 MPa, and the maximum deflections encountered are well below the chamfered shoulder width of 0.6 mm (600 μm). This warrants that the shoulders are able to slide into the corresponding slots unobstructed during operation.

Figure 7.15 (a) Stress analysis of vane bush and (b) deflection of vane bush under Scenario 1 (7.0 bar)

Figure 7.16 (a) Stress analysis of vane bush and (b) deflection of vane bush under Scenario 2 (1.0 bar)

7.2.4 Cylinder Assembly

The cylinder assembly in UVC prototype comprises a lower bearing, cylinder and reed valve, as shown in Figure 7.17. Basically, the cylinder and the lower bearing are designed in circular shape for ease of manufacturing. The cylinder sits on the lower bearing and two 6.0 mm dowel pins are used to align the concentricity between the cylinder and the lower bearing so that their centres coincide at the centre axis “C”. To protect the shaft, a shaft bushing of TEF-MET/P type produced by Singa Bearing Solutions Pte. Ltd. (Appendix F.3) [227]

is assembled onto the lower bearing. The bushing type is used as it has excellent wear behaviour, high load capacity (up to 140 MPa subjected to dynamical load) and high PV-value (up to 60 MPa•m/s under lubricated condition). On the other hand, the discharge port on the cylinder wall is covered by a valve reed, which is assembled in a valve compartment located at the outer cylinder wall. The configuration of valve reed is one end fixed by a M5 screw while the other end free, therefore the free end can be deflected under the working fluid pressure force during the discharge process. To prevent the valve reed from over deflection, a valve stop plate is fixed above the valve reed to restrict the magnitude of the valve reed deflection, as illustrated in Figure 7.18.

Figure 7.17 Cylinder assembly: (a) isometric view and (b) exploded view

Figure 7.18 Discharge reed valve of UVC prototype (a) discharge reed valve components; (b) assembly of reed valve on cylinder

Figure 7.19 presents the detailed design of cylinder. In UVC prototype, the centre of the cylinder is a through hole, which houses the shaft, eccentric and rotor while the vane bush is slotted into the designated location at the cylinder wall. The suction and discharge ports are positioned as close to the centre axis as possible so that more working fluid can be drawn in and discharged out from the compressor during the operation. Also, O-ring grooves are designed on the top surface of the cylinder and around the dowel pin holes to accommodate O-rings. In addition, the O-rings seal up the possible leakages through the clearance between the cover and cylinder, and through the dowel pin holes clearance. Likewise, O-ring groove is also cut on the top surface of the lower bearing where the cylinder sits on, as shown in Figure 7.20. A total of eight radial oil feeding straight holes are created from the outer lower bearing surface toward the centre of the lower bearing to guarantee smooth lubricant flow from the oil reservoir to the moving components during operation.

Figure 7.19 Detailed design of cylinder

Figure 7.20 Detailed design of lower bearing

Both the cylinder and the lower bearing will be made of AISI 4130 steel, where the mechanical properties of AISI 4130 steel are summarised in Table 7.5 [228].

Table 7.5 Mechanical properties of AISI 4130 steel [228]

Properties		Conditions	
		T (°C)	Treatment
Density (kg/m ³)	7850	25	Normalized at 870 °C
Poisson's Ratio	0.27-0.30		
Elastic Modulus (GPa)	205		
Tensile Strength (MPa)	670.0		
Yield Strength (MPa)	435.0		
Elongation (%)	25.5		
Reduction in Area (%)	60.0		
Hardness (HB)	197		

Prior to the final design of the cylinder shown above, the minimum cylinder wall thickness is obtained by thin-walled pressure vessel approximation method [226] as the cylinder radius is normally much bigger than its wall thickness. The equations related to longitudinal stress (σ_l) and hoop stress (σ_h) of a thin wall pressure vessel are shown below:

$$\sigma_l = \frac{\Delta P \times r}{2t} \quad (7.6)$$

$$\sigma_h = \frac{\Delta P \times r}{t} \quad (7.7)$$

where,

ΔP is the differential pressure across the cylinder wall (Pa)

r is the inner radius of the cylinder wall (m)

t is the cylinder wall thickness (m)

It is obvious that the thickness requirement under hoop stress is more stringent than the one by longitudinal stress. Since the cylinder is intended to have uniform wall thickness, Equation (7.7) alone is sufficient for the minimum cylinder wall thickness calculation. The maximum differential pressure that the cylinder wall may experience is when the housing chamber is at the pressure of 7.0 bar while the working chamber is at the pressure of 1.0 bar. With the yield strength of the material, 435.0 MPa, which is the maximum stress that the cylinder can withstand, and a safety factor (N) of 3, the minimum cylinder wall thickness can be obtained:

$$\begin{aligned} t_{cyl,min} &= \frac{\Delta P \times r}{\sigma_h} \times N \\ &= \frac{(7.0 - 1.0) \times 10^5 \times 0.036 \times 3}{435.0 \times 10^6} \\ &= 0.149 \text{ mm} \end{aligned}$$

As the radius of the cylinder (36.0 mm) is 232 times larger than the minimum cylinder wall thickness, the thin-walled pressure vessel assumption is valid. In final design, the cylinder wall thickness is set at 34.0 mm for ease of fabrication and assembly. Additionally, simulation is conducted to analyse the stress and deformation in SolidWorks Simulation 2017 with the conditions of maximum differential pressure across the cylinder wall, and the top and bottom surfaces of the cylinder fixed. As a result, the simulated maximum stress of 2.53 MPa is much lower than the yield strength of the material, while the maximum deflection of 18.84 μm is within the designed tolerance of 20 μm for the cylinder radius, as illustrated in Figure 7.21.

Figure 7.21 (a) Stress analysis of cylinder and (b) deflection of cylinder

7.2.5 Housing Assembly

Housing assembly is the outermost component in UVC prototype, which includes suction pipe, discharge pipe, base plate, housing and, as presented in Figure 7.22. Generally, the base plate is located at the bottom of the UVC prototype, any other components are assembled on top of it. Figure 7.23 shows the details of base plate. The centre of base plate is machined to have a lower bearing seat so that the lower bearing can be concentrically placed on the base plate. O-ring groove is cut on the base plate to accommodate O-ring which prevents high pressure working fluid from leaking out of the compressor. Also, there are two 4.0 mm dowel pin holes and twelve M5 screw clearance holes located at the outer side of the O-ring groove in order to align and secure the housing on the base plate, as well as to securely assemble the entire compressor on the test rig. The corresponding dowel pin holes and screw clearance holes can be found on the flange of the housing as shown in Figure 7.24. The discharge pipe is attached on the housing wall, whereas the suction pipe passes through the housing wall and connects the compressor working chamber with the surrounding atmosphere.

Figure 7.22 Housing assembly: (a) isometric view and (b) exploded view

Figure 7.23 Details design of base plate

Figure 7.24 Housing: (a) isometric view and (b) detailed design

Figure 7.25 Cover: (a) isometric view and (b) detailed design

Figure 7.26 Assembly of lip seal roof

The housing is roofed by a cover, which is also the upper bearing of the compressor, as displayed in Figure 7.25. There are two pressure transducer ports designed at the cover to accommodate the sensors for measuring the working chamber pressure. Since the shaft is protruded out from the upper bearing hole, a lip seal [229] will be employed at the lip seal groove to refrain the lubricant and the working fluid from leaking through the journal bearing clearance. Due to high differential pressure across lip seal, four M5 screw tap holes are designed beside the lip seal groove to tighten the lip seal roof that helps to prevent the lip seal from being pushed out of the groove, as illustrated in Figure 7.26. In addition, a shaft bushing of TEF-MET/P type [227] (Appendix F.3), which is the same type as the bushing assembled in the lower bearing, is employed at the upper bearing hole to protect the shaft. Two 6.0 mm dowel pin holes are used to align the cover with the cylinder assembly, while two 4.0 mm dowel pin holes and twelve screw clearance holes are made to assemble the cover on the housing precisely.

As the housing chamber is filled with pressurized working fluid, the pressure force from the working fluid tends to separate the housing from the base plate. Thus, the minimum number of screws required to resist the force must be determined. Consider the atmosphere pressure is 1.0 bar and the working fluid pressure in the housing chamber is at the discharge pressure, i.e. 7.0 bar, the pressure force acting on the housing flange can be calculated from the equation below:

$$F_{house,press} = \Delta P_{house} \times \frac{\pi}{4} (D_{baseplate,oring}^2 - D_{house,inner}^2) \quad (7.8)$$

where,

$F_{house,press}$ is the pressure force acting on the housing in vertical direction (N)

ΔP_{house} is the differential pressure across the housing flange (Pa)

$D_{baseplate,oring}$ is the diameter of the O-ring groove on the base plate (m)

$D_{house,inner}$ is the inner diameter of the housing (m)

The differential pressure across the housing flange is 6.0 bar, the diameter of the O-ring groove $D_{baseplate,oring}$ is 215.0 mm and the inner diameter of the housing $D_{house,inner}$ is 160.0 mm. By putting in the values into Equation (7.8), $F_{house,press}$ is found to be 9719.3 N.

The allowable stress of an individual screw, σ_{all} is given by:

$$\sigma_{all} = \frac{F_{house,press}}{n \times A_{tensile}} \quad (7.9)$$

where,

σ_{all} is the allowable stress of an individual screw, which is given as $0.75 \times \sigma_{proof}$ (Pa)

n is the number of screws (-)

$A_{tensile}$ is the tensile stress area of the screw (m²)

The proof strength of a M5 steel screws σ_{proof} is 310 MPa and the tensile stress area $A_{tensile}$ is 16.1×10^{-6} m² [222]. Introducing safety factor (N) of 3, the minimum number of screws can be determined by rearranging Equation (7.9):

$$\begin{aligned} n &= \frac{F_{house,press}}{0.75 \times \sigma_{proof} \times A_{tensile}} \times N \\ &= \frac{9719.3}{0.75 \times 310 \times 10^6 \times 16.1 \times 10^{-6}} \times 3 \\ &= 7.79 \\ &\simeq 8 \end{aligned}$$

The required minimum number of screws is found to be 8. In fact, to prevent the working fluid or lubricant from leaking through the clearance

between the base plate and housing flange, 12 screws are used in the prototype. On the other hand, the minimum housing wall thickness is determined by thin-walled pressure vessel approximation method [226]. Consider the housing is made in aluminium alloy 6061-T6 with a yield strength of 240.0 MPa [230], by applying Equation (7.7), the minimum wall thickness ($t_{housing}$) is found to be 0.6 mm. Instead, to accommodate the O-ring groove and screw clearance holes to secure the cover, the thickness is designed as 20.0 mm.

Besides, the minimum number of screws for the cover is also determined by considers only the area exposed to pressure is subjected to pressure force. With the O-ring groove diameter at the housing $D_{house,oring}$ of 175.8 mm, the pressure force acting on the cover $F_{cover,press}$ is found to be 14563.9 N with the condition of 6.0 bar differential pressure across the cover. Since M5 steel screws are used to secure the cover, the proof strength of 310 MPa and tensile stress area of $16.1 \times 10^{-6} \text{ m}^2$ are applied. Subsequently, by introducing a safety factor of 3, the minimum number of screws can be found:

$$\begin{aligned} n &= \frac{F_{cover,press}}{0.75 \times \sigma_{proof} \times A_{tensile}} \times N \\ &= \frac{14563.9}{0.75 \times 310 \times 10^6 \times 16.1 \times 10^{-6}} \times 3 \\ &= 11.67 \\ &\approx 12 \end{aligned}$$

Hence, a total of 12 screws are sufficient to assemble the cover on the housing tightly. In addition, the stress and deflection of the cover are analysed through SolidWorks Simulation 2017. The mechanical properties of AISI 4130 tabulated in Table 7.5 are applied for the cover since it is going to be made by the material. The result shown in Figure 7.27 depicts that the maximum stress of 45.18 MPa at the screw clearance hole is well below the yield strength of 435.0 MPa. Although the maximum deflection of 342.90 μm at the middle region is conspicuous, leakages are unlikely to occur. Once the cover is assembled securely onto the housing, the O-ring with 3.0 mm cross-sectional diameter used in between the cylinder end-face and the cover will be compressed by 25.0 %, i.e.

750 μm [231, 232]. Thus, the clearance is sealed up despite the maximum deflection of 342.90 μm .

Figure 7.27 (a) Stress analysis and (b) deflection of cover

7.2.6 Counterweight

In UVC prototype, the offset design of the eccentric causes the centre of gravity (C.G.) of the moving components assembly, i.e., shaft-eccentric-rotor assembly rotates eccentrically about the centre of rotation (shaft centre axis) during operation. Therefore, a counterweight is required to balance the centrifugal force acting on the shaft while it is rotating, which is displayed in Figure 7.28. The counterweight is produced in a cubic shape for ease of the dimensions control and it is mounted on the upper part of the shaft with a M5 screw, as shown in Figure 7.29a. Fundamentally, the purpose of the counterweight is to shift the C.G. of the shaft-eccentric-rotor assembly as close as possible to the centre axis of rotation in order to minimize the centrifugal effect, as illustrated in Figures 7.29b and c. By using the counterweight, the distance between C.G. and centre axis of rotation is reduced from 2.65 mm to 0.003 mm, as measured from the CAD model.

Figure 7.28 Counterweight: (a) isometric view and (b) top view

Figure 7.29 (a) Assembly of counterweight on shaft; (b) original position of centre of gravity; (c) shifted centre of gravity after the counterweight is mounted

Although the shape of counterweight makes its dimensions easy to control, it may induce friction drag force since it is rotating at high angular speed. The effect of the friction drag force can be calculated by the following equation:

$$F_D = C_{dr} \times \frac{1}{2} \rho_{air} V^2 \times A_{proj} \quad (7.10)$$

where,

F_D is the drag force (N)

C_{dr} is the drag coefficient which depends on the reference area A_{proj} (-)

ρ_{air} is the density of air (kg/m³)

V is the velocity of the moving body (m/s)

A_{proj} is the reference area, for a cube, the reference area is the projected area in the moving direction (m^2)

Since the counterweight is rotating in an atmospheric environment, density of air $\rho_{air} = 1.165 \text{ kg/m}^3$ is applied by considering the room temperature of about 30°C at standard atmospheric pressure [233]. The velocity V of the counterweight at a shaft speed of 3000 rev/min is calculated to be 5.34 m/s. The projected area of counterweight A_{proj} is simplified as a square-shaped area of 0.000915 m^2 . Meanwhile, for a square-shaped reference area, the drag coefficient C_{dr} is 1.05 [234]. Input these values into Equation (7.10), the drag force F_d is found to be 0.016 N, which is negligible as it is only 0.0013 % of the resultant load on the shaft F_r of 1244.8 N obtained from simulation.

7.2.7 Assembly of Prototype

To this end, the major components in UVC prototype are assembled and the assembly is presented in Figure 7.30. The components are shown in the cross-sectional views in Figures 7.31 and 7.32. The operation of the prototype is illustrated in Figure 7.33.

Figure 7.30 (a) Isometric view of UVC prototype and (b) top view of the prototype without cover

Figure 7.31 Cross-sectional side view of UVC prototype assembly

Figure 7.32 Cross-sectional top view of UVC prototype assembly

Figure 7.33 Shaft rotation of UVC prototype

7.3 Design of Lubrication System

In general, the role of lubricant in a compressor is essential as it reduces the friction and wear by lubricating the sliding surfaces and improves the reliability of the compressor. Furthermore, the lubricant also seals up the clearances between the mating components which minimizes the internal leakage occurs between the compression and suction chambers. The lubricant acts as a cooling agent that cool down the compressor sliding parts and prevent them from overheating which may cause seizure in operation. Hence, it is important to have an efficient lubrication system to ensure a smooth compressor operation. In this section, the design of lubrication system is unveiled and analysed in this section.

Figure 7.34 presents the lubrication system in U-Vane compressor prototype, where the arrows indicate the path and direction the lubricant is

flowing through. These flow paths comprise different flow elements such as straight holes (paths 2-3 and 3-4), radial hole (path 4-5), journal bearing clearances (paths 3-9 and 5-6), spiral grooves (paths 3-9 and 8-7) and rotor end-face clearances (paths 6-s, 6-c, 9-s and 9-c). The system runs mechanically without using any electrical pump.

With reference to Figure 7.34, point 1 is the surface level of oil sump in the housing chamber and point 2 is the bottom of the UVC prototype. As the housing chamber is always in high discharge pressure, the pressure is acting continually on the lubricant reservoir (oil sump) surface at point 1, which pushes the lubricant from points 1 and 2 to point 3 through the radial straight holes in the lower bearing. At point 3, the bottom of the shaft, a partial amount of the lubricant is driven up by the lower bearing spiral groove and reaches the junction point (point 9), while the remaining lubricant climbs up to point 4. At point 4, the lubricant is expelled through the rotating radial holes and reach point 5. Then, it flows through the upper journal bearing clearance, reaches point 6 and exit to the suction chamber (point s) or compression chamber (point c) through the rotor end-face clearance. Let's get back to the partial amount of lubricant that reached point 9. Since it is a junction point, the lubricant either overcomes the height difference and reach point 8 or flows into the suction chamber (point s) or compression chamber (point c) through the rotor end-face clearance. In addition, the lubricant that reaches point 8 is dragged up to point 7 with the assistance of the eccentric spiral groove. It meets with the lubricant at point 6, and eventually exits radially to the working chamber through the rotor end-face clearance. In the end, the lubricant in the working chamber is discharged out to the housing chamber and flows back to oil sump.

It is crucial to ensure that lubricant is supplied adequately into the compressor and circulated throughout the oil flow paths. If the lubricant becomes sluggish or being trapped at certain region, the lubricant temperature will increase, and its viscosity will decrease as it is being heated by the friction. Eventually, the lubrication in the compressor is ineffective due to a poor lubricant viscosity.

Figure 7.34 Schematics of lubrication system in U-Vane compressor prototype

7.3.1 Lubrication Network Model

From the method proposed by Kim *et al.* [183, 184], the interaction between flow elements and the flow rates at each point is represented by an electrical network circuit as demonstrated in Figure 7.35. In his method, the flow elements, pressure differences and lubricant flow rates are represented by electrical resistances (R), voltages sources (ρgh and ΔP) and electrical currents (Q), respectively. According to Kim *et al.* [185], the mathematical expressions of flow resistances are modelled by Hagen-Poiseuille flow and Couette flow, as presented below:

Flow through straight hole:

$$\frac{\Delta P}{Q} = \frac{8\mu L}{\pi r_{hole}^4} \quad (7.11)$$

Flow through radial hole, presented as voltage source:

$$\Delta P = \frac{\rho \omega^2}{2} (r_o^2 - r_i^2) \quad (7.12)$$

Flow through bearing clearance:

$$\frac{\Delta P}{Q} = \frac{12\mu L}{\delta_b^3 \pi r (1 + 1.5\varepsilon^2)} \quad (7.13)$$

The total flow through the spiral groove consists of Hagen-Poiseuille flow and Couette flow:

$$\frac{\Delta P}{Q} = \frac{12\mu L}{H_{sg}^3 D_{sg} \cos \beta} \quad (7.14)$$

$$Q = \frac{1}{2} A_{sg} r \omega \sin \beta \quad (7.15)$$

Flow through end-face clearance:

$$\frac{\Delta P}{Q} = \frac{6\mu \ln(\frac{r_2}{r_1})}{\pi \delta_{ef}^3} \quad (7.16)$$

where,

ΔP is the pressure across the flow element (Pa)

Q is the lubricant flow rate (m³/s)

μ is the dynamic viscosity of lubricant (Pa•s)

L is the length of the flow element (m)

r_{hole} is the radius of the hole (m)

ρ is the density of lubricant (kg/s)

ω is the shaft rotational speed (rad/s)

r_o is the outer radius (m)

r_i is the inner radius (m)

δ_b is the journal bearing clearance (m)

r is the radius of the journal (shaft) (m)

ε is the eccentricity ratio of journal bearing (-)

H_{sg} is the depth of the spiral groove (m)

D_{sg} is the width of the spiral groove (m)

β is the spiral groove angle, which is defined by $\tan^{-1} \left(\frac{\text{circumference}}{n_{sg}} \right)$ (rad)

n_{sg} is the pitch of the spiral groove (-)

A_{sg} is the cross sectional area of spiral groove (m^2)

$\frac{r_2}{r_1}$ is the ratio of outer radius to inner radius (-)

δ_{ef} is the end-face clearance (m)

Figure 7.35 Network circuit of lubrication system

The subscript sg denotes spiral groove. By applying Kirchhoff's Voltage Law and Kirchhoff's Current Law in the circuit [235], thirteen independent equations can be formed:

$$Q_1 + Q_8 + Q_{9a} - Q_{9b} - Q_2 = 0 \quad (7.17)$$

$$Q_2 - Q_{3c} - Q_{3s} - Q_4 - Q_{5a} + Q_{5b} = 0 \quad (7.18)$$

$$Q_4 + Q_{5a} - Q_{5b} - Q_{6c} - Q_{6s} + Q_7 = 0 \quad (7.19)$$

$$-Q_7 - Q_8 - Q_{9a} + Q_{9b} = 0 \quad (7.20)$$

$$R_{7s}Q_{3s} - R_{7c}Q_{3c} = P_c - P_s \quad (7.21)$$

$$R_{8s}Q_{6s} - R_{8c}Q_{6c} = P_c - P_s \quad (7.22)$$

$$R_9Q_8 - R_{9sg}Q_{9a} = \rho gh_9 - \rho gh_{9sg} \quad (7.23)$$

$$R_7 Q_4 - R_{7sg} Q_{5a} = \rho g h_7 - \rho g h_{7sg} \quad (7.24)$$

$$R_2 Q_1 + (R_3 + R_4 + R_5 + R_6) Q_2 + R_{7c} Q_{3c} = P_1 - P_c + \rho g (h_1 - h_3 + h_5) + \Delta P_4 + \Delta P_6 \quad (7.25)$$

$$R_2 Q_1 + (R_3 + R_4 + R_5 + R_6) Q_2 + R_7 Q_4 + R_{8c} Q_{6c} = P_1 - P_c + \rho g (h_1 - h_3 + h_5 + h_7) + \Delta P_4 + \Delta P_6 \quad (7.26)$$

$$(R_3 + R_4 + R_5 + R_6) Q_2 + R_7 Q_4 - R_8 Q_7 + R_9 Q_8 = \rho g (-h_3 + h_5 + h_7 + h_9) + \Delta P_4 + \Delta P_6 - \Delta P_8 \quad (7.27)$$

$$Q_{5b} = \frac{1}{2} A_{sg, ecc} r_{ecc} \omega \sin \beta_{ecc} \quad (7.28)$$

$$Q_{9b} = \frac{1}{2} A_{sg, lb} r_s \omega \sin \beta_{lb} \quad (7.29)$$

In UVC prototype, the diameters of the straight holes in the lower bearing and in the shaft (paths 2-3 and 3-4) are set at 10.0 mm, and the diameter of the radial hole (path 4-5) is 3.0 mm. The width and depth of the spiral grooves on both the eccentric and the lower shaft are 1.5 mm and 0.4 mm (paths 8-7 and 3-9). In addition, the clearances of the physical prototype presented in Table 7.8 are applied. The above independent equations are arranged in matrix form and solved by Gauss Elimination Method [236]. The result shows that, at the discharge pressure of 7.0 bar and when operates at 3000 rev/min, the supplied lubricant flow rate into the compressor is 2.36 cm³/s averagely, which is more than the average required lubricant flow rate of 1.88 cm³/s at the journal bearing.

7.4 UVC Prototype

The compressor components were manufactured by Time Engineering Sdn. Bhd. and the materials used is tabulated in Table 7.6:

Table 7.6 Materials use in UVC prototype

Material Used in UVC Prototype		
1	Shaft and eccentric	AISI 630 (17-4 PH Stainless Steel) [223]
2	Rotor	AISI 630 (17-4 PH Stainless Steel)
3	Vane bush	AISI 630 (17-4 PH Stainless Steel)

4	Cylinder	AISI 4130 [228]
5	Lower bearing	AISI 4130
6	Reed valve	AISI 304L [237]
7	Cover (upper bearing)	AISI 4130
8	Housing	Aluminium Alloy 6061-T6 [230]
9	Base Plate	AISI 4130
10	Suction pipe and discharge pipe	AISI 304L
11	Counterweight	AISI 304L

The physical dimensions of the components were inspected for the consistency to the designed tolerances so that the exact clearance dimensions between the components can be determined. Several measuring tools such as coordinate measuring machine, digital vernier caliper and micro-meter screw gauge with accuracies of ± 0.001 mm and ± 0.01 mm were used. Each of the dimensions was taken from the mean of multiple measurements. The measured dimensions of main components are summarised in Table 7.7. The asterisks (*) specify that the parts have been reworked to get better fitting and smoother operation.

Table 7.7 Physical dimensions of actual prototype

Component/Dimension	Designed Dimension (mm)	Measured Dimension (mm)
Cylinder Assembly		
Axial length	$35.00 + 0.02$	35.02
Diameter	72.00 ± 0.02	72.002
Vane bush slot diameter	$18.00 + 0.02 \sim 0.04$	18.000
Lower bearing shaft bushing length	25.00	25.00
Lower bearing shaft bushing diameter	25.00	25.065
Shaft		

Upper shaft diameter (upper journal)	25.00 – 0.02 ~ 0.04	24.99
Eccentric diameter*	34.00 – 0.03	33.83
Eccentric length	25.00 ± 0.02	25.01
Eccentric spiral groove depth	0.40 ± 0.10	0.38
Eccentric spiral groove width	1.50 ± 0.10	1.62
Eccentricity (<i>e</i>)	4.50 ± 0.01	4.374
Lower shaft diameter (lower journal)	25.00 – 0.02 ~ 0.04	24.97
Lower shaft length	30.50 ± 0.10	30.66
Lower shaft spiral groove depth	0.40 ± 0.10	0.48
Lower shaft spiral groove width	1.50 ± 0.10	1.65
Total Length	200.00 ± 0.10	200.15
Rotor		
Axial length	35.00 ± 0.01	34.94
Outer diameter	63.00 ± 0.05	63.230
Inner diameter*	34.00 + 0.03 ~ 0.06	33.899
Vane length (from centre of rotor)	43.00 - 0.05	42.945
Vane width	6.00 – 0.01	5.98
Vane Bush		
Diameter	18.00 – 0.02	17.95
Axial length	35.00 + 0.02	34.92
Vane slot width	6.00 + 0.01 ~ 0.02	6.03
Left shoulder width	4.00 – 0.02	4.02
Left shoulder length	6.34 + 0.10	6.35
Right shoulder width	4.00 – 0.02	3.98
Right shoulder length	6.34 + 0.10	6.35
Suction Pipe		

Diameter	16.00 ± 0.10	16.50
Discharge Port		
Diameter	9.00 ± 0.10	9.10
Valve Reed		
Thickness	0.30 ± 0.01	0.30
Diameter	12.50 ± 0.10	12.50
Length of neck section	-	25.45
Width of neck section	6.00 ± 0.10	6.15
Total length	45.25 ± 0.10	45.25
Cover (Upper Bearing)		
Upper bearing shaft bushing length	25.00	25.00
Upper bearing shaft bushing diameter	25.00	25.089

Based on the measured dimensions, the clearances in the prototype can be determined as listed in Table 7.8.

Table 7.8 Clearances between components of prototype

Clearances	
Radial clearance between rotor and cylinder, δ_{rad}	12.0 μm
Radial clearance between eccentric and inner rotor surface, $\delta_{ecc,rad}$	32.5 μm
Radial clearance between upper journal and bearing, $\delta_{bear,up}$	49.5 μm
Radial clearance between lower journal and bearing, $\delta_{bear,low}$	47.5 μm
Radial clearance between vane bush and slot, $\delta_{rad,vh}$	25.0 μm
Clearance between rotor end-face and cover, δ_{ref}	80.0 μm

In addition, several measurements were made on the surface roughness at the bearing region with a profilometer, the average of the measured arithmetical mean deviations R_a is 1.101 μm . The moving parts are fabricated with sliding fit tolerance to ensure smooth operation of the prototype. Although some of the disparities between the designed and measured dimensions are noteworthy, the fittings between each component are satisfied. The prototype components and the assembly procedure are portrayed below:

1. The housing assembly is tightened on the test rig with M5 screws.

Figure 7.36 Housing assembly is mounted on the test rig

2. Cylinder assembly is placed onto the designated lower bearing seat at the base plate.

Figure 7.37 Cylinder assembly sit on the base plate

3. Vane bush is slotted into the cylinder wall.

Figure 7.38 Vane bush is slotted into the cylinder wall

4. Rotor is assembled on the cylinder and the free end of the vane is inserted into the slot at the bush.

Figure 7.39 Rotor is assembled on the cylinder

5. Suction pipe is assembled onto the cylinder through the suction hole on the housing.

Figure 7.40 Suction pipe is assembled onto the cylinder

6. Shaft is inserted into the lower bearing and the centre of rotor.

Figure 7.41 Shaft is inserted into the centre of rotor

7. The cover is aligned with the dowel pins and assembled to the housing with M5 screws. Then, the lip seal roof is assembled on the cover.

Figure 7.42 Cover is assembled to the housing

8. Counterweight is assembled onto the shaft and tightened with a M5 screw. The assembly of the UVC prototype is complete.

Figure 7.43 Final assembly of UVC prototype

The performance of the UVC prototype will be tested experimentally and the result will be discussed in the next chapter.

7.5 Concluding Remarks

The design specification of U-Vane compressor prototype has been stated. The working fluid is air and the volumetric displacement of the prototype is $33.5 \text{ cm}^3/\text{rev}$. Based on the specifications, the design of UVC prototype has been presented. The prototype consists of 10 components, which includes shaft, eccentric, rotor, vane bush, cylinder, reed valve, lower bearing, base plate, housing, cover and counterweight. For major components, the critical dimensions have been determined. Stress and deflection analyses have also been conducted. In addition, the lubrication system for UVC prototype has been designed and it has been modelled according to the method proposed by Kim *et al.* [185]. The result shows that the lubricant is sufficiently supplied to the compressor. Lastly, the fabricated prototype components and the assembly procedure have been shown and described.

Chapter 8 Experimental Study

In this chapter, the U-Vane compressor (UVC) prototype is instrumented and investigated experimentally. The observations from the measurements of the prototype are presented and the results are compared with the prediction from UVC mathematical model. Lastly, the analysis of the post-experiment prototype components is presented.

8.1 Experimental Setup

An experiment setup is designed to measure the performance and operational characteristics of U-Vane compressor prototype. Figure 8.1 shows the schematic of the experiment setup. The measured parameters are motor operating speed (ω), torque input to the compressor (TM), temperature (T), pressure (P), and volumetric flow rate (Q). Figures 8.2 to 8.4 show the actual experiment setup.

Figure 8.1 Schematic of the experiment setup

Figure 8.2 Actual experiment setup: front view and motor control panel

Figure 8.3 Actual experiment setup: suction and discharge

Figure 8.4 Actual experiment setup: back view

In the experiment, the prototype is driven by an ABB 7.5 kW two-pole induction motor with a control panel to adjust the operating speed. The motor is selected as it is capable in providing a maximum torque output of 50 Nm at the maximum operating speed of 2920 rev/min. The operating speed is recorded from the control panel as shown in Figure 8.3c. A torque meter of model Futek TRS605, which is able to record the instantaneous torque up to 50 Nm at high sampling rate (i.e. 25,000 data measured per second), is mounted along the shaft axis to measure the torque input to the prototype. It is one end coupled with the motor shaft and the other end coupled with the compressor so that the torque transmitted from the motor to the compressor can be measured. During the operation, the atmospheric air is drawn into the prototype and the working chamber pressure is measured by two Endevco piezo resistive pressure transducers (model: 8530B-500) which are inserted into the prototype. The pressure transducers have high sensitivity (0.6 ± 0.2 mV/psi) and are able to capture the pressure at high sampling rate, i.e. 25,000 data measured per second. The compressed working fluid, in this case the air, is subsequently discharged into a Swagelok stainless steel air receiver (model: 304L-HDF4-500) with an internal volume of 0.5 Litre, which the discharge pressure is measured by WIKA pressure transducer (model:

A-10) and the discharge temperature is measured by type-T thermocouple with gauge size of 4.8 mm. The internal volume of the air receiver is determined such that the compressor requires at least 10 revolutions to fill the air receiver with compressed air in order to reduce the discharge pulsation of the compressor. The WIKA pressure transducer and the type-T thermocouple are employed as they are able to measure the pressure and temperature stably with high accuracy (Table 8.1). The pressure and the temperature of the compressed air flowing out to the atmosphere are again measured as the pressure and temperature are required to calculate the air density. Meanwhile, the discharge volumetric flow rate is measured by a Brooks Instrument rotameter (Model: MR-3A17-BLBN) with a measurement range of 4 – 50 L/min. A ball valve is installed after the rotameter to control the discharge flow rate. In addition, regulating the compressed air from the air receiver reduces the discharge pulsation of the compressor, which allows steady measurement readings from the float in the rotameter. The signals from the sensors are captured by National Instrument data acquisition system (DAQ) and the measured readings are taken at the maximum sampling rate of 25,000 Hz. The specifications of the measuring devices and the calibration data are shown in Appendices F and G. The uncertainties of the measuring devices are summarised in Table 8.1.

Table 8.1 Summary of the uncertainties of measuring devices

Uncertainties of Measuring Devices	
Operating speed (from ABB control panel)	± 5.0 rev/min
Futek torque meter (model: TRS605)	± 0.25 Nm
Type T thermocouple [238]	± 1.0 °C
WIKA Pressure transducer (model: A-10)	± 0.125 bar
Endevco piezo resistive pressure transducers (model: 8530B-500)	± 0.15 bar
Brooks Instrument stainless steel float rotameter (Model: MR-3A17-BLBN)	± 2 L/min

As mentioned in Section 7.1, atmospheric air is used as the working fluid because it is available abundantly. Furthermore, the pressurized air could be directly discharged out to the atmosphere because it is more environmentally

friendly (it causes almost zero harm to the environment) when compared with R-1234yf that was used in the simulation presented in Chapter 6. This allows the use of an open-loop test circuit and subsequently reduces the cost and complexity of the experiment setup. Also, using air as the working fluid in the experiment is sufficient for the validation of the developed mathematical model of the compressor since the compression ratio of a typical positive displacement type compressor is primarily driven by the change of the working chamber volume and it is independent from the working fluid used in the compressor.

8.2 Experiment Result and Discussion

In this section, the experimental procedures for examining the prototype performance is presented. The comparison between the measured and predicted results are analysed and discussed. The prototype was operated from 1200 rev/min to 2100 rev/min and the highest attained discharge pressure was 5.27 bar (abs.). Beyond this, vibration and leakage were observed, and the prototype was prone to seizure before reaching the steady state. The prototype was then disassembled for inspection on its components.

8.2.1 Experimental Procedure

Prior to starting the experiment, the pipes and fittings were checked to prevent possible leakage. The operating speed for the motor was controlled by a control panel as displayed in Figure 8.2. The experiment was conducted by adjusting the operating speed from 1200 rev/min to 2100 rev/min with an interval of about 200 rev/min. For each speed, the mass flow rate was regulated, and the corresponding discharge pressure was recorded. All the measurements were taken accordingly at steady state, i.e. after 15 minutes of operation. The experimental procedures for prototype performance measurements are described below:

1. The motor was switched on and the speed was gradually increased from rest to the desired operating speed to warm up the compressor prototype.

2. The ball valve was regulated to build up the pressure in the air receiver.
3. The conditions were maintained for 15 minutes so that the discharge pressure can be stabilized (steady state).
4. Measurements were then taken through DAQ system. For a constant operating speed, steps 2 to 4 were repeated to obtain various measurements based on different flow rates. Also, the temperatures of the housing and cover were monitored and recorded throughout the process which were later used for estimating the lubricant temperature and viscosity in the prototype.
5. After the data collection, the motor was switched off to vent the pressurized air in the receiver tank. The compressor was put to cool down for about 40 minutes.
6. The experimental procedure was restarted from step 1 when the desired operating speed was increased by the stated interval.

When the operating speed was more than 2100 rev/min, vibration was observed, which was believed to be caused by the imperfect dynamic balance of the rotating parts, including the shaft, the eccentric, the rotor, the counterweight, the aluminium coupling and the universal coupling. Since all the dimensions of the parts are designed with tolerances, the deviations between the physical and nominal designed dimensions may result in undesirable dynamic balance of the rotating parts. The vibration has caused the compressor operation to seize before the discharge pressure was stabilized. Thus, no measurements were taken after that. The analysis of the collected measurements is then carried out.

8.2.2 Considerations for Validation of Mathematical Model

In U-Vane compressor prototype, a lip seal is used to seal the clearance between the protruded shaft and the cover. Although it prevents the possible leakage of pressurized air from the prototype, additional friction from rubbing against the lip seal must be considered and implemented into the mathematical model. The lip seal friction can be calculated from the following equation proposed by Heinz *et al.* [239]:

$$L_{f,ls} = 2\pi\phi r_s \omega_1 \quad (8.1)$$

where,

$L_{f,ls}$ is the frictional loss caused by lip seal (W)

ϕ is the tangential friction per unit of circumference (N/m)

The tangential friction per unit of circumference can be read from Figure 8.5 where the type of the lip seal used in the experiment is spring-loaded elastomer. The elastomer type is employed as it is low in cost compared to the other types of lip seals, and it is capable to withstand the designated operating pressure of up to 7.0 bar (abs).

Figure 8.5 Friction of various lip seals as a function of pressure [239]

As mentioned in Section 7.3, the lubricant is used to lubricate the rubbing surfaces and to prevent the metal-to-metal contact. However, as illustrated in Figure 8.6b, the lubricant tends to flow in the spiral groove as the flow resistance of the spiral groove is smaller than the flow resistance of the clearance. As a result, the protruded portion of the eccentric surface may rub against the inner rotor surface in the lack of lubricant when the rotor is under pressure force from

the working fluid. This produces excessive frictional loss due to the contact between the eccentric and the inner rotor surface.

Figure 8.6 Lubricant flow through the eccentric (a) without spiral groove; (b) with spiral groove

In such circumstance, dry friction is applied to Equation (5.34) and the resultant frictional loss between eccentric and inner rotor surface becomes a combination of viscous and dry frictions:

$$L_{f, ecc} = \left[\frac{n_{sg} D_{sg}}{l_{ecc}} \frac{2\pi\mu_{lub}}{\delta_{ecc, rad}} r_{ecc}^3 l_{ecc} \left(\omega_1 - \frac{d\gamma_1}{dt} \right) + \mu_{ecro} F_R r_{ecc} \right] \left(\omega_1 - \frac{d\gamma_1}{dt} \right) \quad (8.2)$$

where,

n_{sg} is the pitch of the spiral groove (-)

D_{sg} is the width of the spiral groove (m)

μ_{ecro} is the coefficient of friction between the eccentric and the inner rotor surface (-)

F_R is the resultant force acting on the eccentric (N)

The term $\frac{n_{sg} D_{sg}}{l_{ecc}}$ is added into the equation by assuming only the spiral

groove region is experiencing the viscous friction. Since an aluminium bronze bushing is press-fitted into the inner rotor, the value of the coefficient of friction is taken as 0.1 [240], which is a value obtained experimentally for the sliding between the aluminium bronze and the stainless steel. In addition, the rubbing

mark due to the dry friction is presented in Section 8.2.4, which verifies the metal-to-metal contact at this region.

The clearances measured from the prototype are much larger than the desired and thus, the effect of the instantaneous radial clearance between the outer rotor surface and the inner cylinder surface may become obvious when the clearance between the eccentric and the inner rotor surface varies, as shown in Figure 8.7.

Figure 8.7 Schematic view of the changes in radial clearances

Since the main components of the rolling piston compressor and the U-Vane compressor, which are the piston (rotor), the eccentric and the cylinder, are similar in geometry, the relationship given by Yanagisawa *et al.* [150] is applied for the instantaneous radial clearance between the outer rotor surface and the inner cylinder surface, as expressed below:

$$\delta_{rad,i} = \delta_{rad} - \epsilon_{ecc}\delta_{ecc,rad} \cos(\theta_1 - \theta_F - \phi_{ecc}) - \epsilon_b\delta_b \cos(\theta_1 - \theta_F - \phi_b) \quad (8.3)$$

where,

$\delta_{rad,i}$ is the instantaneous radial clearance between outer rotor surface and inner cylinder surface (m)

ε_{ecc} is the eccentricity ratio of the eccentric, which can be found from journal bearing analysis (-)

ε_b is the eccentricity ratio of the journal bearing (shaft) (-)

8.2.3 Comparison between Predicted and Measured Results

The power input to the compressor prototype is obtained by multiplying the measured average torque with the operating speed:

$$\dot{W}_{input,m} = \bar{T} \omega \quad (8.4)$$

where,

$\dot{W}_{input,m}$ is the measured power input to the prototype (W)

$\bar{T}$ is the measured average torque (Nm)

ω is the operating speed (rad/s)

The measured power input at various conditions is shown in Figure 8.8. Obviously, the power input increases with the operating speed, which is mainly caused by the increase in frictional losses at higher operating speeds. Furthermore, as the discharge pressure increases, the power input increases because more compression power is required to compress the working fluid to a higher pressure. In short, the higher the discharge pressure, the higher power input required at a given operating speed. Likewise, the higher the operating speed, the higher power input required when the discharge pressure is the same.

Figure 8.8 Measured power input at various operational conditions

It is analysed that at higher operating speed, the increase in power input is mainly caused by the increase in the working fluid compression rate and the frictional loss. Basically, the viscosity of the lubricant influences the compressor frictional loss. This viscosity is extremely reliant on the lubricant temperature. Since the temperatures of the housing and cover were recorded during the experiment, the lubricant temperature in the prototype, as well as the oil sump temperature can be estimated.

Table 8.2 Estimation of lubricant temperature in the prototype

Operating Speed (rev/min)	Housing Temp. (°C)	Cover Temp. (°C)	Estimated Oil Sump Temp. (°C)	Estimated Lubricant Temp. (°C)
1216	35.0	45.0	38.0	48.0
1417	40.0	52.0	43.0	55.0
1639	42.0	56.0	45.0	60.0
1839	45.0	60.0	48.0	64.0
2112	47.0	65.0	50.0	70.0

The steady state housing temperature and the cover temperature of each operating speed are listed in Table 8.2. The oil sump temperature is estimated from the housing temperature as the oil sump is deposited inside the housing chamber. It is estimated by an additional 3.0 °C to the housing temperature for each operating speed. The cover temperature was used to estimate the localized lubricant temperature and the bearing lubricant temperature as the cover is where the upper bearing is located at. Hence, the lubricant temperature is approximated by an additional temperature range from 3.0 °C to 5.0 °C to the cover temperature according to the operating speed. The viscosity of lubricant is then read from the viscosity-temperature chart provided by the supplier (Appendix H) using the estimated lubricant temperature. In addition, the estimated temperatures are input to the heat transfer model (Appendix C.2), which is simulated concurrently with the mathematical model presented in Chapters 4 and 5 to give a more comprehensive predicted result.

Figure 8.9 Comparison between predicted and measured power inputs

Figure 8.9 shows the comparisons between the predicted and measured power inputs to the prototype, where the solid lines represent the predicted result whereas the dots represent the measured result. Good agreement is observed between the predicted and the measured power inputs with maximum and average discrepancies of 6.8 % and 3.8 %, respectively. The result also implies that the estimation of the lubricant temperatures is reasonable. The discrepancy between the predicted and measured power inputs is shown in Figure 8.10 with ± 10 % error bands, where the error band lines are displayed as the dashed-lines. It is noticed that all the data points fall within the error bands, which clearly signifies the capability of the prediction model. In fact, the lubricant viscosity in

the prototype is related to the localized lubricant temperature. Since the estimated lubricant temperature was treated as constant at each operating speed, the discrepancy may lie in the approximation between the actual localized lubricant temperature and the estimated lubricant temperature.

Figure 8.10 Overall discrepancy between predicted and measured power inputs

The measured mass flow rate at various operational conditions are plotted in Figure 8.11. As expected, the mass flow rate rises with the operating speed. However, the mass flow rate decreases when the discharge pressure increases at a given operating speed. As the suction pressure is maintained at the atmospheric pressure of about 1.0 bar (abs.), higher discharge pressure indicates more leakages from the compression chamber to the suction chamber at a given operating speed since the leakages are pressure-driven. In short, the least measured mass flow rate is observed when the operating speed is lowest (1216 rev/min in this case) for each of the discharge pressures, and the measured mass flow rate is lower at higher discharge pressure for a given operating speed. Figure 8.12 shows the volumetric efficiencies of the prototype at various operational conditions, which were measured to be from 43.4 % to 92.3 %. The lowest efficiency is found when the operating speed is 1216 rev/min at the discharge pressure of 4.26 bar, whereas the highest efficiency is spotted at a discharge

pressure of 2.54 bar when operating at 2112 rev/min. As described above, the poor efficiencies at low operating speed and high discharge pressure are mainly resulted by the internal leakage. Hence, it is recommended that the operating speed must be sufficiently high in order to maintain the high volumetric efficiency of the compressor.

Figure 8.11 Measured mass flow rate at various operational conditions

Figure 8.12 Volumetric efficiency of the compressor prototype at various operational conditions

Figure 8.13 Comparison between predicted and measured mass flow rate

The comparison between the predicted and measured mass flow rates is presented in Figure 8.13. The solid lines represent the predicted mass flow rate while the dots represent the measured mass flow rate. The predicted mass flow rate is calculated by assuming a constant leakage blocking rate of 55.0 % (i.e. leakage flow coefficient $C_{leak} = 0.45$, Equation (C3)) since the lubricant in the compressor may seal the possible leakage paths. In Figure 8.13, it is observed that at the operating speeds of 1839 rev/min and 2112 rev/min, when the discharge pressure is below 5.0 bar, the measured mass flow rates are higher than the prediction. The discrepancy is mainly lied in the assumption of the constant leakage blocking rate. The leakage blocking rate in reality may vary with the

operating speed, localized lubricant viscosity, solubility of working fluid in the lubricant, clearance dimensions and pressure ratio. When the measured flow rate is more than the prediction, the blocking effect of the lubricant in the compressor may be more significant than the prediction. Additionally, it indirectly implies the success of the lubrication network system design. Other than the above findings, at the operating speeds of 1216 rev/min, 1417 rev/min and 1639 rev/min, the predicted and measured mass flow rates are in good agreement, as shown in Figures 8.13a-c.

Figure 8.14 Overall discrepancy between predicted and measured mass flow rates

The discrepancy between the predicted and measured mass flow rates is exhibited in Figure 8.14, where the error bands of $\pm 15\%$ are shown as dashed-lines. It is noticed that the mass flow rate tends to be under-predicted by the mathematical model when the predicted mass flow rate is above 0.7 g/s. Below that, the agreement between the predicted and measured mass flow rates is good and the average discrepancy is 12.7 %.

The leakage flow in actual compressor operation involves not only single phase gas flow, but also single phase dissolved gas-oil flow and two phase oil-gas mixture flow. Thus, a single phase leakage flow model may not govern the leakage model of a compressor at all operational conditions. In short, leakage is an unfavourable phenomenon, making the clearances in the compressor tight or

increases the operating speed of the compressor may effectively reduce the leakage flow.

The variations of the instantaneous pressure in the working chamber during the compressor operation were also measured by the pressure transducers inserted in the cover of the prototype. The positions of the sensors are selected to be as close to the suction and discharge ports as possible. Also, they are kept in a distance to avoid the communication between each other. However, the pressure transducers are covered by the rotor end-face during operation, as illustrated in Figure 8.15. Thus, the measurement angular range of the instantaneous pressure in the working chamber is from 70° to 300° only.

Figure 8.15 Pressure transducers blocked by rotor at (a) 0° , (b) 70° and (c) 300°

The comparisons between predicted and measured variations of instantaneous pressure in the working chamber at various operational conditions are presented in Figures 8.16 to 8.18. The effective force area coefficient ($C_{dv,eff}$, Equation 4.39) with a constant value of 0.2 is employed in the prediction model. Moreover, the value of valve damping ratio (ξ) is kept at a constant of 0.5 to account for the viscous damping effect as a result of the oil stiction on the valve [139, 241].

Figure 8.16 Comparison between predicted and measured variations of instantaneous working chamber pressures at various operating speeds (average discharge pressure = 2.61 bar)

Figure 8.17 Comparison between predicted and measured variations of instantaneous working chamber pressures at various operating speeds (average discharge pressure = 3.53 bar)

Figure 8.18 Comparison between predicted and measured variations of instantaneous working chamber pressures at various operating speeds (average discharge pressure = 5.17 bar)

It is observed that the predicted and measured variations of instantaneous pressure in the working chamber are in good agreement with the maximum and average mean discrepancies of 8.3 % and 4.1 %, respectively. The mean discrepancy is calculated based on the following formula:

$$\sigma_{dis} = \frac{\sum \sqrt{\frac{(Y_{measured} - Y_{predicted})^2}{Y_{measured}^2}}}{N} \quad (8.5)$$

where,

σ_{dis} is the average discrepancy (-)

N is the total number of data points (-)

$Y_{measured}$ is the value of the measured data point (-)

$Y_{predicted}$ is the value of the predicted data point (-)

In the results shown above, the flow losses during the suction and discharge processes are recognisable. It is noted that the measured valve vibration during the discharge process exhibits higher frequency and magnitude compared to the predicted valve vibration. There are few factors that could cause the discrepancy between the valve vibrations. In Section 4.6, it is mentioned that the effective force area coefficient is a function of the valve lift, thus the difference between

the valve vibration could be affected by value of the effective force area coefficient during the valve lifting in the prediction model since it is kept as a constant for all operational conditions. In addition, the discrepancy between the valve vibrations could also be affected by the assumption of only the first vibration mode in the one dimensional (1-D) valve reed vibration analysis. In actual operation, the valve reed may vibrate in the valve lateral direction (2-D vibration), which is likely to happen when the longitudinal centre axis of the valve reed is not perfectly aligned with the centre axis of the discharge port, as demonstrated in Figure 8.19. It is noted that if the valve stresses are to be studied, higher modes of valve reed vibrations may need to be incorporated [242]. This discrepancy, however, contributes relatively insignificant difference in the flow losses in the compressor overall energy loss consideration.

Figure 8.19 Misalignment between valve reed and discharge port

At the operating speed of 2112 rev/min, the measured magnitude of the pressure at the angle of about 240° is lower than the prediction as shown in Figures 8.16e, 8.17e and 8.18d. The similar deviation of the pressure is also found at the operating speed of 1839 rev/min when the discharge pressures are 2.61 bar and 5.17 bar. As discussed earlier, this is because of the value of effective force area coefficient as it is kept as a constant for all operational conditions in the mathematical model. In reality, the effective force area coefficient may be affected by the valve lift, the operating speed, the discharge pressure and the oil stiction effect on the valve reed. Figures 8.20 and 8.21 demonstrate the attempts of applying a higher effective force area coefficient, i.e. $C_{dv,eff} = 0.5$, at the operating speeds of 1839 rev/min and 2112 rev/min when the discharge pressures are 2.61 bar and 5.17 bar. Apparently, the agreement between the predicted and

measured pressure ‘humps’ at 240° is better with a higher effective force area coefficient. In the attempt, the reductions of the mean discrepancy between the instantaneous pressures for 1839 rev/min and 2112 rev/min are 23.3 % and 22.3 %, respectively.

Figure 8.20 Comparison between predicted and measured results with 0.5 as the value of the effective force area coefficient (2.61 bar)

Figure 8.21 Comparison between predicted and measured results with 0.5 as the value of the effective force area coefficient (5.17 bar)

Although the magnitude of the pressure is corrected, deviation is found between the predicted and measured instantaneous pressure increasing rates at angles of 200° to 250°, as shown in Figures 8.21b and d. The identical

observation is also spotted in Figures 8.16a-b, 8.17b-c and 8.18a-b, where the measured pressure has a slower increasing rate than that of the prediction. After carefully investigation, it is believed that the reason may lie in the noncircularity of the inner rotor diameter as depicted in Figure 8.22.

Figure 8.22 Noncircularity of the inner rotor surface diameter

Figure 8.22a shows the measured circularity using coordinate measuring machine (CMM) while Figure 8.22b shows the corresponding circularity on the rotor. A total of six points have been measured to inspect the circularity of the inner rotor diameter. It is visible that the distance between the centre and the circumference at the angles of about 30° and 200°, i.e. points 1 and 4, are about 10 μm more than the nominal diameter (displayed as red solid line). Since the points are positioned diagonally, the diametral clearance between the eccentric and the inner rotor surface in this diagonal direction may increase about 20 μm . Additionally, as discussed in Section 8.2.2, it is possible to further widen the dynamical instantaneous radial clearance between the outer rotor surface and the inner cylinder surface due to the accumulation of the clearances. As a result, the instantaneous radial clearance leakage flow increases and subsequently reduces the instantaneous pressure in the working chamber. Based on Figures 8.21b and d, Figures 8.23b and d present the effect when the instantaneous leakage flow increases at the angles from 180° to 240°, while keeping the value of the effective

force coefficient to be 0.5. As a result, the agreement between the predicted and measured instantaneous working chamber pressure is better.

In general, the noncircularity of the inner rotor surface would make the journal bearing analysis at this region more complicated and results in unpredictable increase in leakage flow. Hence, it will be essential to ensure the roundness of the components which are preferably to be circular.

Figure 8.23 Comparison between predicted and measured results after increased the instantaneous leakage flow through radial clearance (average discharge pressure = 5.17 bar, $C_{force} = 0.5$)

8.2.4 Post-Experiment Prototype Components Inspection

After completed the experiment, the UVC prototype was disassembled and its components were inspected.

Shaft

Figure 8.24 Shaft and eccentric after the experiment

Figure 8.24 shows the shaft and the eccentric after the experiment. The rubbing against the lip seal has yielded an obvious “polished ring” on the shaft. It is also observed that there are rubbing marks on the protruded surface of the eccentric, which corresponds to the wear marks on the aluminium bronze bushing at the inner rotor surface as shown in Figure 8.25. These findings imply that the additional considerations in Section 8.2.2 are reasonable. Meanwhile, designing the spiral groove on the eccentric surface is inappropriate as it may cause additional frictional loss or damage the compressor components.

Figure 8.25 Wear mark on the aluminium bronze bushing (rotor) after experiment

Furthermore, since the vane sides are rubbing against the vane bush during the operation, wear marks are also found on the vane sides, as shown in Figure 8.26. Hence, these regions must be designed carefully as they may cause wear issue to the extent of operation seizure.

Vane side (right)

Vane side (left)

Figure 8.26 Wear mark on the vane after experiment

8.3 Concluding Remarks

An open test circuit has been setup to examine the operational and performance characteristics of the U-Vane compressor prototype. Several measurements have been taken in the experiment and the mathematical model of UVC has been validated with the collected data. The discrepancy between the predicted and measure power inputs has been found in good agreement with maximum and average discrepancies of 6.8 % and 3.8 %, respectively. However, the average discrepancy between the predicted and measured flow rates are larger, which is 12.7 %. Furthermore, it has been observed that the measured mass flow rates are higher than the prediction at the operating speeds of 1800 rev/min and 2100 rev/min, when the discharge pressure is below 5.0 bar. This implicitly points out the lubricant has successfully blocked substantial amount of leakage flow during the operation. In addition, the volumetric efficiencies at various operational conditions have been evaluated to be from 43.4 % to 92.3 %. The predicted and measured variations of instantaneous pressure in the working chamber at various operational conditions have also been compared and good agreement has been obtained with maximum and average mean discrepancies of 8.3 % and 4.1 %, respectively. In conclusion, the mathematical model of UVC is able to predict the performance and the variation of instantaneous pressure in the working chamber.

Chapter 9 Conclusion and

Recommendations for Future Work

In this research project, a novel U-Vane compressor (UVC) has been studied and discussed. The conceptual design has been presented, mathematical models have been formulated and verifications with measurements were carried out. Performance prediction using R-1234yf as the working fluid has been conducted. A prototype has been designed, fabricated, instrumented and tested experimentally using air as the working fluid in an open test circuit. The measurements have been used to verify the predictions from the mathematical models. In this final chapter, the conclusion of this project and the recommendations for the future work are presented.

9.1 Conclusion

The conclusion of the research project is categorized into three sections, namely compressor design, mathematical modelling and experimental study.

9.1.1 Compressor Design

Various types of compressor mechanisms have been reviewed.

- It is noted that the vane in a swing compressor must be sufficiently long to prevent the swing bushes from collapsing. This design has resulted in an unnecessarily large cylinder and housing in order to accommodate the long vane. To reduce the size of the cylinder and the housing, the U-Vane compressor has been conceptualised and designed.
- A new vane bush is introduced. It is a single piece bush with two extended shoulders. These shoulders allow a shorter vane to be employed. Provision has also been made on the rotor to accommodate the extended shoulders.
- With this modification, the cylinder and housing volume of the UVC is estimated to be about 30 % smaller than that of the swing compressor, see Appendix A.

9.1.2 Mathematical Modelling

Several mathematical models have been formulated to describe the performance and operational characteristics of the UVC. The simulation results using R-1234yf as the working fluid have been presented and discussed. The following findings were obtained:

- The leakage flows through radial clearance and rotor end-face clearance contribute 49.9 % and 37.0 %, respectively, to the total leakage flow.
- The largest frictional loss occurs at the vane sides, which accounts for 47.8 % of the total frictional loss. The frictional losses at the rotor end-face, the rubbing between eccentric and inner rotor surface and the journal bearing, account for 23.7 %, 14.4 % and 11.3 %, respectively.
- When operating at 3000 rev/min, the predicted volumetric and mechanical efficiencies are 96.28 % and 75.71 %, respectively.

To better understand the operational behaviour of the UVC, parametric studies have been carried out, which are summarised below:

- The volumetric efficiency increased by 8.6 % when the operating speed increases from 1000 rev/min to 5000 rev/min. However, the mechanical efficiency reduced by 16.8 %.
- When keeping the suction pressure at 1.0 bar (abs.) and using air as the working fluid, increasing the discharge pressure from 2.0 bar (abs.) to 10.0 bar (abs.) will see the volumetric efficiency reduced by 5.0 %, while the mechanical efficiency improved by 38.0 %.
- Lengthening the vane from 8.0 mm to 20.0 mm increases the mechanical efficiency by 8.7 %, whereas the volumetric efficiency is unaffected. It is mainly caused by the reduction in the vane side friction.
- Lengthening the vane bush shoulder from 4.0 mm to 12.0 mm results in an improvement of 3.3 % in the mechanical efficiency, while volumetric efficiency remains the same.

- Thickening the rotor wall from 9.0 mm to 17.0 mm, while maintaining the volumetric displacement the same, causes the mechanical efficiency to increase by 3.6 % due to the reduction in the frictional loss between the eccentric and inner rotor surface. However, it has no effect on the volumetric efficiency.
- The mechanical efficiency increased by 22.3 % when the rotor-to-cylinder radius ratio varies from 0.78 to 0.94. The effect on the volumetric efficiency is insignificant.
- The increment of slenderness ratio of the journal bearing from 0.5 to 1.5, reduces the mechanical efficiency by 2.6 %.

9.1.3 Experimental Study

In this research project, an open test circuit with air as the working fluid has been setup to examine the performance of the UVC. The prototype has been successfully operated from 1200 rev/min to 2100 rev/min with the highest attained discharge pressure of 5.27 bar (abs.), i.e. maximum pressure ratio of about 5.27. The measurements have been compared with the predicted results from the mathematical model. The following observations have been made:

- The power input increases with the operating speed and discharge pressure. The maximum and average discrepancies are 6.8 % and 3.8 %, respectively.
- The mass flow rate increases with the operating speed as expected, while it decreases with the discharge pressure. The measured mass flow rates at high operating speeds (i.e. 1800 rev/min and 2100 rev/min) are higher than the prediction when the discharge pressure is below 5.0 bar. This implicitly indicates the success of the lubrication system design. The average discrepancy between the prediction and measured mass flow rates is 12.7 %.
- The volumetric efficiency has been measured to be from 43.4 % to 92.3 %. It was observed that low discharge pressure and high operating speed would result in high volumetric efficiency, and vice versa.

- The variation of the instantaneous pressure in the working chamber has also been measured. The maximum and average mean discrepancies between the predicted and measured results are 8.3 % and 4.1 %, respectively.

9.2 Recommendation for Future Work

The U-Vane compressor has been studied in this project. The followings are proposed for future work:

- For future prototype, a design optimisation approach can be used to search for a combination of design variables which produces the optimum compressor performance at a prescribed objective function [243], i.e. optimal values of vane length and bush shoulder length.
- In this project, the internal leakage through the rotor end-face has been modelled through numerical approach (Appendix C.1). A more comprehensive 3-D numerical simulation may be considered.
- It was observed that during the testing of the prototype, the noise and vibration level was rather high. Future study may focus on vibration and noise reduction. Noise sources such as those caused by flow induced noise at the discharge and the vibration of the valve reed can be considered for future study.
- To ensure a reliable compressor operation, the temperature of the components should be measured and assessed. The locations of the temperature sensors can take its cue from the heat transfer model presented in Appendix C.2. The measured results can then be used for the validation of the heat transfer model.
- In current prototype, the locations of the miniature pressure transducers measure a limited angular range of the instantaneous pressure in the working chamber between 70° and 300° only. To measure the full cycle instantaneous working chamber pressure, it is suggested the sensors to be positioned radially on the cylinder wall.
- After the experiment, wear marks were observed on the vane tips due to rubbing against the bush. Thus, the vane is suggested to be coated

with harder materials with low friction to minimise the potential wear issues.

9.3 Final Remarks

A lot of learnings, knowledges and practical experiences have been gained by the author throughout the research project. The U-Vane compressor has proven to be functional as intended. The author hopes that the insights gained in this thesis would assist in the future developmental work of the compressor and a substantial amount of materials can be saved from producing rotary compressors.

References

- [1] M. Y. Q. Lin, Y. D. Lim, and K. T. Ooi, "Working principle of new multi-chamber rotary compressor," in *IOP Conference Series: Materials Science and Engineering*, 2017, vol. 232, no. 1, p. 012059: IOP Publishing.
- [2] R. C. Arora, *Refrigeration and air conditioning*. PHI Learning Pvt. Ltd., 2012.
- [3] D. Japikse and N. Baines, *Introduction to turbomachinery*. Concepts ETI, 1994.
- [4] H. P. Bloch and J. J. Hoefner, *Reciprocating Compressors:: Operation and Maintenance*. Elsevier, 1996.
- [5] G. L. Kollberg and M. C. Shaw, "Sliding vane compressor," U.S. Patent 2,150,122, 1939.
- [6] N. Stosic, I. K. Smith, and A. Kovacevic, "Optimisation of screw compressors," *Applied Thermal Engineering*, vol. 23, no. 10, pp. 1177-1195, 2003.
- [7] P. N. Pandeya and W. Soedel, "Rolling piston type rotary compressors with special attention to friction and leakage," in *International Compressor Engineering Conference*, Purdue University, 1978.
- [8] L. Li and Y. Yu, "The Theoretical Analysis and Experimental Research for a Refrigerating Scroll Compressor," in *International Compressor Engineering Conference*, Purdue University, 1990.
- [9] A. Howell, "Fluid dynamics of axial compressors," *Proceedings of the Institution of Mechanical Engineers*, vol. 153, no. 1, pp. 441-452, 1945.
- [10] W. J. Kearton, "The influence of the number of impeller blades on the pressure generated in a centrifugal compressor and on its general performance," *Proceedings of the Institution of Mechanical Engineers*, vol. 124, no. 1, pp. 481-568, 1933.
- [11] "World Compressor Market," in *Japan Air Conditioning Heating and Refrigeration News (JARN)*, ed: JARN, Ltd., 2019.
- [12] "Rotary Air Compressors Drive Global Market Towards \$30BN," in *News*, ed: ACE Compressor, 2016.
- [13] M. Masuda, K. Kato, and Y. Shibamoto, "Rotary compressor," U.S. Patent 7,029,252, 2006.
- [14] M. Masuda, K. Sakitani, Y. Yamamoto, T. Uematsu, and A. Mutoh, "Development of swing compressor for alternative refrigerants," in *International Compressor Engineering Conference*, Purdue University, 1996.
- [15] T. Ohkawa *et al.*, "Development Of Hermetic Swing Compressors For CO2 Refrigerants," in *International Compressor Engineering Conference*, Purdue University, 2002.
- [16] K. T. Ooi, "Simulation of a piezo-compressor," *Applied thermal engineering*, vol. 24, no. 4, pp. 549-562, 2004.
- [17] K. T. Ooi and Y. L. Teh, "Twin-plate rotary compressor," U.S. Patent 7,726,960, 01 Jun, 2010.

- [18] K. T. Ooi and K. T. Aw, "Thermal modelling of a dry revolving vane compressor," in *IOP Conference Series: Materials Science and Engineering*, 2017, vol. 232, no. 1, p. 012055: IOP Publishing.
- [19] A. Subiantoro and K. T. Ooi, "Introduction of the Revolving Vane Expander," (in English), *HVAC&R Research*, vol. 15, no. 4, pp. 801-816, Jul 2009.
- [20] K. M. Tan and K. T. Ooi, "Journal bearings design for a novel revolving vane compressor," (in English), *International Journal of Refrigeration-Revue Internationale Du Froid*, vol. 34, no. 1, pp. 94-104, Jan 2011.
- [21] Y. L. Teh and K. T. Ooi, "Theoretical study of a novel refrigeration compressor - Part I: Design of the revolving vane (RV) compressor and its frictional losses," (in English), *International Journal of Refrigeration-Revue Internationale Du Froid*, vol. 32, no. 5, pp. 1092-1102, Aug 2009.
- [22] K. S. Yap, K. T. Ooi, and A. Chakraborty, "Analysis of the novel cross vane expander-compressor: Mathematical modelling and experimental study," *Energy*, 2017.
- [23] X. Hu, Z. Qu, X. Yang, and J. Sun, "Theoretical study on frictional losses of a novel automotive swing vane compressor," *International Journal of Refrigeration*, vol. 36, no. 3, pp. 758-767, 2013.
- [24] X. Yang, C. Dong, and Z. Qu, "Design and dynamic analysis of a novel double-swing vane compressor for electric vehicle air conditioning systems," *International Journal of Refrigeration*, vol. 76, pp. 52-62, 2017.
- [25] Global Energy Statistical Yearbook 2018 [Online]. Available: <https://yearbook.enerdata.net/>
- [26] K. Y. Dong, R. J. Sun, H. Li, and H. D. Jiang, "A review of China's energy consumption structure and outlook based on a long-range energy alternatives modeling tool," *Petroleum Science*, vol. 14, no. 1, pp. 214-227, 2017.
- [27] BP Statistical Review of World Energy [Online]. Available: <https://www.bp.com/content/dam/bp/en/corporate/pdf/energy-economics/statistical-review/bp-stats-review-2018-full-report.pdf>
- [28] Use of Electricity [Online]. Available: https://www.eia.gov/energyexplained/index.php?page=electricity_use
- [29] Singapore Energy Statistics 2018 [Online]. Available: https://www.ema.gov.sg/cmsmedia/Publications_and_Statistics/Publications/SES18/Publication_Singapore_Energy_Statistics_2018.pdf
- [30] Z. Abdullah, "Singapore's household electricity consumption up 17 per cent over past decade," in *The Straits Times*, ed: Singapore Press Holdings Ltd, 2018.
- [31] (2018, 30 November 2018). *Four In Five Households Motivated To Save Energy If They Can Save Money: NEA Study*. Available: <https://www.nea.gov.sg/media/releases/news/index/four-in-five-households-motivated-to-save-energy-if-they-can-save-money-nea-study>
- [32] Energy Prices [Online]. Available: https://www.ema.gov.sg/cmsmedia/publications_and_statistics/publications/ses/2017/energy-prices/index.html

- [33] Annual Energy Review 2009 [Online]. Available: https://www.eia.gov/totalenergy/data/annual/pdf/sec2_19.pdf
- [34] Energy Resources [Online]. Available: <https://www.worldenergy.org/data/resources/>
- [35] C. Breidenich, D. Magraw, A. Rowley, and J. W. Rubin, "The Kyoto protocol to the United Nations framework convention on climate change," *American Journal of International Law*, vol. 92, no. 2, pp. 315-331, 1998.
- [36] "Copenhagen Accord," in *2009 United Nations Climate Change Conference (a.k.a Copenhagen Summit)*, Copenhagen, Denmark, 2009.
- [37] R. Falkner, "The Paris Agreement and the new logic of international climate politics," *International Affairs*, vol. 92, no. 5, pp. 1107-1125, 2016.
- [38] P. K. Bansal and A. Martin, "Comparative study of vapour compression, thermoelectric and absorption refrigerators," *International Journal of Energy Research*, vol. 24, no. 2, pp. 93-107, 2000.
- [39] Y. A. Cengel and M. A. Boles, *Thermodynamics: An Engineering Approach Fifth Edition*. 2002.
- [40] G. Lorentzen, "Revival of Carbon-Dioxide as a Refrigerant," (in English), *International Journal of Refrigeration-Revue Internationale Du Froid*, vol. 17, no. 5, pp. 292-301, Jun 1994.
- [41] J. Basuk, "Freons and ozone in the stratosphere," *Bulletin of the American Meteorological Society*, vol. 56, no. 6, pp. 589-592, 1975.
- [42] M. J. Molina and F. S. Rowland, "Stratospheric sink for chlorofluoromethanes: chlorine atom-catalysed destruction of ozone," *Nature*, vol. 249, no. 28, pp. 810-812, 1974.
- [43] M. S. Reisch, "Honeywell licenses low-GWP refrigerant," in *Chemical & Engineering News* vol. 94, ed, 2016.
- [44] M. R. Heikal. (2011, 30 June 2018). *Refrigeration*. Available: <http://www.thermopedia.com/content/1086/>
- [45] J. S. Baek, E. A. Groll, and P. B. Lawless, "Development Of A Piston-Cylinder Expansion Device For The Transcritical Carbon Dioxide Cycle," *International Refrigeration and Air Conditioning Conference*, p. 584, 2002.
- [46] P. Srihirin, S. Aphornratana, and S. Chungpaibulpatana, "A review of absorption refrigeration technologies," *Renewable and sustainable energy reviews*, vol. 5, no. 4, pp. 343-372, 2001.
- [47] Y. Chen, W. Han, and H. G. Jin, "Analysis of an absorption/absorption-compression refrigeration system for heat sources with large temperature change," (in English), *Energy Conversion and Management*, vol. 113, pp. 153-164, Apr 1 2016.
- [48] T. Miyazaki, B. B. Saha, and S. Koyama, "Analytical Model of a Combined Adsorption Cooling and Mechanical Vapor Compression Refrigeration System," *Heat Transfer Engineering*, no. just-accepted, pp. 00-00, 2016.
- [49] L. Li, Y. Zhao, B. Guo, P. Shu, J. Shen, and S. He, "Wrap of cylinder and its effect on main features of rotary vane compressor for automobile air conditioning system," *International journal of refrigeration*, vol. 26, no. 5, pp. 566-574, 2003.

- [50] G. Bianchi and R. Cipollone, "Theoretical modeling and experimental investigations for the improvement of the mechanical efficiency in sliding vane rotary compressors," *Applied Energy*, vol. 142, pp. 95-107, 2015.
- [51] A. B. Tramschek and K. T. Ooi, "Effects of Port Geometry, Dimensions and Position on the Performance of a Rotary Compressor," in *Proceedings of the Purdue Compressor Technology Conference*, Purdue University, 1992, pp. 1177-1187.
- [52] V. Yee and W. Soedel, "Comments on blade excited rigid body vibrations of rotary vane compressors," in *Proceedings of the Purdue Compressor Technology Conference*, 1980, pp. 243-248.
- [53] A. B. Tramschek and M. H. Mkumbwa, "Mathematical modelling of radial and non-radial rotary sliding vane compressors," in *International Compressor Engineering Conference*, Purdue University, 1996.
- [54] X. Tojo, T. Kan, and A. Arai, "Dynamic behavior of sliding vane in small rotary compressors," in *International Compressor Engineering Conference*, Purdue University, 1978.
- [55] P. Shu, B. Guo, L. Li, and J. Hu, "Influence of Vane Slot Back-Pressure on the Characteristic of Vane Motion in Rotary Vane Compressor," in *International Compressor Engineering Conference*, Purdue University, 1998.
- [56] A. Ucer and H. Aksel, "Simulation Studies, on the Rotary Type Compressor Systems," in *International Compressor Engineering Conference*, Purdue University, 1980.
- [57] H. Kruse, "Experimental investigations on rotary vane compressors," in *International Compressor Engineering Conference*, Purdue University, 1980.
- [58] D. Aradau and L. Costiuc, "Friction power in sliding vane type rotary compressors," in *International Compressor Engineering Conference*, Purdue University, 1996.
- [59] G. Bianchi and R. Cipollone, "Friction power modeling and measurements in sliding vane rotary compressors," *Applied Thermal Engineering*, vol. 84, pp. 276-285, 2015.
- [60] G. Bianchi, F. Fatigati, S. Murgia, and R. Cipollone, "Design and analysis of a sliding vane pump for waste heat to power conversion systems using organic fluids," *Applied Thermal Engineering*, vol. 124, pp. 1038-1048, 2017.
- [61] G. Bianchi, F. Fatigati, S. Murgia, R. Cipollone, and G. Contaldi, "Modeling and experimental activities on a small-scale sliding vane pump for ORC-based waste heat recovery applications," in *International Compressor Engineering Conference*, Purdue University, 2016.
- [62] R. Cipollone and D. Di Battista, "Sliding vane rotary pump in engine cooling system for automotive sector," *Applied Thermal Engineering*, vol. 76, pp. 157-166, 2015.
- [63] R. Cipollone, D. Di Battista, G. Contaldi, S. Murgia, and M. Mauriello, "Development of a Sliding vane rotary pump for engine cooling," *Energy Procedia*, vol. 81, pp. 775-783, 2015.
- [64] G. Bianchi, S. Rane, A. Kovacevic, R. Cipollone, S. Murgia, and G. Contaldi, "Numerical CFD simulations on a small-scale ORC expander

- using a customized grid generation methodology," *Energy Procedia*, vol. 129, pp. 843-850, 2017.
- [65] M. Imran, M. Usman, B.-S. Park, and D.-H. Lee, "Volumetric expanders for low grade heat and waste heat recovery applications," *Renewable and Sustainable Energy Reviews*, vol. 57, pp. 1090-1109, 2016.
 - [66] P. Kolasiński, P. Błasiak, and J. Rak, "Experimental and numerical analyses on the rotary vane expander operating conditions in a micro organic Rankine cycle system," *Energies*, vol. 9, no. 8, p. 606, 2016.
 - [67] W. Suankramdee, T. Thongtip, and S. Aphornratana, "Development of a sliding vane expander in a micro-scale ORC system for utilizing low-grade heat," *Energy Procedia*, vol. 138, pp. 817-822, 2017.
 - [68] "Refrigerating Compressors Industry 2016 Market Research Report," in *Japan Air Conditioning, Heating and Refrigeration News (JARN)*, ed, 2018.
 - [69] H. Ding and H. Gao, "3-D Transient CFD Model For A Rolling Piston Compressor With A Dynamic Reed Valve," in *International Compressor Engineering Conference*, Purdue University, 2014.
 - [70] K. T. Ooi, "Design optimization of a rolling piston compressor for refrigerators," (in English), *Applied Thermal Engineering*, vol. 25, no. 5-6, pp. 813-829, Apr 2005.
 - [71] T. Yanagisawa and T. Shimizu, "Friction losses in rolling piston type rotary compressors. III," *International journal of refrigeration*, vol. 8, no. 3, pp. 159-165, 1985.
 - [72] K. T. Ooi, T. N. Wong, and E. C. Kwek, "Effects of vane spring stiffness on compressor performance," in *International Compressor Engineering Conference*, Purdue University, 1994.
 - [73] T. Matsuzaka and S. Nagatomo, "Rolling piston type rotary compressor performance analysis," in *Purdue Compressor Technology Conference*, Purdue University, 1982.
 - [74] Y. Liu and J. Kosco, "Vane Dynamics Analysis of a Tilted Vane Rotary Compressor," in *International Compressor Engineering Conference*, Purdue University, 1998.
 - [75] M. Takebayashi, H. Hata, T. Iizuka, A. Ishiyama, and H. Kohsokabe, "A Study on Wear Characteristics of a Rolling-Piston-Type Rotary Compressor," in *International Compressor Engineering Conference*, Purdue University, 2000.
 - [76] S. Tanaka, T. Nakahara, and K. Kyogoku, "Mixed Lubrication Analysis of Vane Tip in Rotary Compressor," in *International Compressor Engineering Conference*, Purdue University, 2000.
 - [77] J. Y. Bae, J. D. Kim, B. C. Lee, T. Y. Noh, and S. M. Byun, "Vane jumping in rotary compressor," in *International Compressor Engineering Conference*, Purdue University, 1998.
 - [78] C. M. N. F. Costa, R. T. S. Ferreira, and A. T. Prata, "Considerations about the leakage through the minimal clearance in a rolling piston compressor," in *International Compressor Engineering Conference*, Purdue University, 1990.
 - [79] T. Yanagisawa and T. Shimizu, "Leakage losses with a rolling piston type rotary compressor. II. Leakage losses through clearances on rolling piston

- faces," *International Journal of Refrigeration*, vol. 8, no. 3, pp. 152-158, 1985.
- [80] J. Lee, "Performance analysis of rolling piston type rotary compressor," in *International Compressor Engineering Conference*, Purdue University, 1988.
- [81] J. Wu and A. Chen, "A new structure and theoretical analysis on leakage and performance of an oil-free R290 rolling piston compressor," *International Journal of Refrigeration*, vol. 49, pp. 110-118, 2015.
- [82] D. Cai, G. He, T. Yokoyama, Q. Tian, X. Yang, and J. Pan, "Simulation and comparison of leakage characteristics of R290 in rolling piston type rotary compressor," *International Journal of Refrigeration*, vol. 53, pp. 42-54, 2015.
- [83] D. Cai *et al.*, "Leakage characteristics and an updated volumetric efficiency prediction model of rolling piston type rotary compressor for small capacity air-conditioner and heat pump applications," *Applied Thermal Engineering*, vol. 121, pp. 1080-1094, 2017.
- [84] Rotary Screw Compressor [Online]. Available: https://en.wikipedia.org/wiki/Rotary-screw_compressor
- [85] D. P. Roger Bywater. (2014, 30 November 2018). *Present-Day Supercharging - Expediency or Efficiency?* Available: http://www.jagweb.com/aj6eng/supercharging_article.php
- [86] A. Fujiwara and N. Sakurai, "Experimental analysis of screw compressor noise and vibration," in *International Compressor Engineering Conference*, Purdue University-, 1986.
- [87] M. Fujiwara, K. Kasuya, T. Matsunaga, and M. Watanabe, "Computer modeling for performance analysis of rotary screw compressor," in *International Compressor Engineering Conference*, Purdue University, 1984.
- [88] M. Wang. (2015, 30 November 2018). *Comparison of working principle and structure of screw compressor*. Available: <https://www.linkedin.com/pulse/comparison-working-principle-structure-screw-compressor-mark-wang>
- [89] N. Stošić, "On gearing of helical screw compressor rotors," *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, vol. 212, no. 7, pp. 587-594, 1998.
- [90] B. Sangfors, "Analytical modeling of helical screw machine for analysis and performance prediction," in *International Compressor Engineering Conference*, Purdue University, 1982.
- [91] Y. Tang and J. S. Fleming, "Obtaining the optimum geometrical parameters of a refrigeration helical screw compressor," in *International Compressor Engineering Conference*, Purdue University, 1992.
- [92] J. S. Fleming and Y. Tang, "The analysis of leakage in a twin screw compressor and its application to performance improvement," *Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering*, vol. 209, no. 2, pp. 125-136, 1995.
- [93] N. Stošić, I. Smith, A. Kovačević, and C. Aldis, "The design of a twin-screw compressor based on a new rotor profile," *Journal of Engineering Design*, vol. 8, no. 4, pp. 389-399, 1997.

- [94] A. Kovacevic, "Boundary adaptation in grid generation for CFD analysis of screw compressors," *International journal for numerical methods in engineering*, vol. 64, no. 3, pp. 401-426, 2005.
- [95] A. Kovacevic, S. Rane, N. Stosic, Y. Jiang, S. Lowry, and M. Furmanczyk, "Influence of approaches in CFD solvers on performance prediction in screw compressors," in *International Compressor Engineering Conference*, Purdue University, 2014.
- [96] S. R. Rane, A. Kovacevic, and M. Kethidi, "CFD Modelling in Screw Compressors with complex multi rotor configurations," in *International Compressor Engineering Conference*, Purdue University, 2012.
- [97] J. V. Voorde, J. Vierendeels, and E. Dick, "Development of a Laplacian-based mesh generator for ALE calculations in rotary volumetric pumps and compressors," *Computer methods in applied mechanics and engineering*, vol. 193, no. 39-41, pp. 4401-4415, 2004.
- [98] J. P. Elson, N. Kaemmer, S. Wang, and M. Perevozchikov, "Scroll Technology: An overview of past, present and future developments," in *International Compressor Engineering Conference*, Purdue University, 2008.
- [99] (2018, 04 December 2019). *Scroll Air Compressors*. Available: <http://oneki.biz/scroll.html>
- [100] T. Hirano, N. Matsumura, and K. Takeda, "Development of high efficiency scroll compressors, for air conditioners," in *International Compressor Engineering Conference*, Purdue University, 1988.
- [101] N. Ishii, M. Fuhushima, K. Sano, and K. Sawai, "A study on dynamic behavior of a scroll compressor," in *International Compressor Engineering Conference*, Purdue University, 1986.
- [102] Y. R. Lee and W. F. Wu, "On the profile design of a scroll compressor," *International journal of refrigeration*, vol. 18, no. 5, pp. 308-317, 1995.
- [103] S. Etemad and J. Nieter, "Computational parametric study of scroll compressor efficiency, design, and manufacturing issues," in *International Compressor Engineering Conference*, Purdue University, 1988.
- [104] N. Ishii, K. Bird, K. Sano, M. Oono, S. Iwamura, and T. Otokura, "Refrigerant leakage flow evaluation for scroll compressors," in *International Compressor Engineering Conference*, Purdue University, 1996.
- [105] K. Tojo, I. Hayase, and T. Tsuchiya, "Improvement of Self-Adjusting Back Pressure Mechanism for Scroll Compressors," in *International Compressor Engineering Conference*, Purdue University, 1998.
- [106] Y. Yamamoto, "Rotary compressor," U.S. Patent 5,577,903, 1996.
- [107] K. Furusho, T. Okawa, K. Saitoh, and K. Sakitani, "Numerical and Experimental Investigation of Swing Compressor Characteristics," in *International Compressor Engineering Conference*, Purdue University, 1998.
- [108] X. Pan, S. Sun, Z. Xing, and S. Pan, "Theoretical study and experimental verification of the leakage losses of R410A in the swing compressor with no valves," *International Journal of Refrigeration*, vol. 99, pp. 327-336, 2019.

- [109] X. Pan, M. Wang, W. Li, Z. Xing, and S. Pan, "Structural study on a swing compressor with no valves for air conditioning systems," *International Journal of Refrigeration*, vol. 88, pp. 300-306, 2018.
- [110] K. T. Ooi and Y. L. Teh, "Revolving vane compressor," U.S. Patent 8,206,140, 2012.
- [111] Y. L. Teh and K. T. Ooi, "Design and friction analysis of the revolving vane compressor," in *International Compressor Engineering Conference*, Purdue University, 2006.
- [112] Y. L. Teh and K. T. Ooi, "Theoretical study of a novel refrigeration compressor-Part III: Leakage loss of the revolving vane (RV) compressor and a comparison with that of the rolling piston type," *international journal of refrigeration*, vol. 32, no. 5, pp. 945-952, 2009.
- [113] K. T. Ooi and Y. L. Teh, "Design improvements of revolving vane compressors, RV-i," in *International Compressor Engineering Conference*, Purdue University, 2008.
- [114] A. Subiantoro and K. T. Ooi, "Design analysis of the novel revolving vane expander in a transcritical carbon dioxide refrigeration system," *international journal of refrigeration*, vol. 33, no. 4, pp. 675-685, 2010.
- [115] Y. L. Teh and K. T. Ooi, "Analysis of internal leakage across radial clearance in the improved revolving vane (RV-i) compressor," in *International Compressor Engineering Conference*, Purdue University, 2008.
- [116] W. Soedel, *Introduction to computer simulation of positive displacement type compressors*. Purdue University, School of Mechanical Engineering, 1972.
- [117] D. Squarer and R. Kothmann, "Digital Computer Simulation of a Reciprocating Compressor-A Simplified Analysis," in *International Compressor Engineering Conference*, Purdue University, 1972.
- [118] B. Karll, "Computer simulation of the cylinder process in a compressor based on the first law of thermodynamics," in *International Compressor Engineering Conference*, Purdue University, 1972.
- [119] R. Prakash and R. Singh, "Mathematical modeling and simulation of refrigerating compressors," in *International Compressor Engineering Conference*, Purdue University, 1974.
- [120] W. Rottger and H. Kruse, "Analysis of the Working-Cycle of Single Stage Refrigeration-Compressors Using Digital Computers," in *International Compressor Engineering Conference*, Purdue University, 1976.
- [121] C. C. Hiller and L. Glicksman, "Detailed modeling and computer simulation of reciprocating refrigeration compressors," in *International Compressor Engineering Conference*, Purdue University, 1976.
- [122] E. H. Ng, A. B. Tramschek, and J. F. T. MacLaren, "Computer simulation of a reciprocating compressor using a real gas equation of state," in *International Compressor Engineering Conference*, Purdue University, 1980.
- [123] S. Lee, R. Singh, and M. Moran, "First Law Analysis of a Compressor Using a Computer Simulation Model," in *International Compressor Engineering Conference*, Purdue University, 1982.

- [124] K. T. Ooi and T. N. Wong, "A computer simulation of a rotary compressor for household refrigerators," *Applied thermal engineering*, vol. 17, no. 1, pp. 65-78, 1997.
- [125] E. Lemmon, M. Huber, and M. McLinden, "NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg," <http://www.nist.gov/srd/nist23.cfm>, 2013.
- [126] A. Bredesen, "Influence of valve dynamics on compressor performance," *International Journal of Refrigeration*, vol. 2, no. 1, pp. 17-21, 1979.
- [127] W. Glaeser, "Failure mechanisms of reed valves in refrigeration compressors," *Wear*, vol. 225, pp. 918-924, 1999.
- [128] M. Costagliola, "The theory of spring-loaded valves for reciprocating compressors," *Journal of Applied Mechanics-Transactions of the ASME*, vol. 17, no. 4, pp. 415-420, 1950.
- [129] J. F. MacLaren, "Review of Simple Mathematical Models of Valves in Reciprocating Compressors," in *International Compressor Engineering Conference*, Purdue University, 1972.
- [130] J. MacLaren and S. Kerr, "Analysis of Valve Behavior in Reciprocating Compressors," in *Proceedings of the XII International Congress of Refrigeration*, 1967.
- [131] A. Traversari and P. Lacitignola, "Use and calculation of ring type valves for reciprocating compressors," *Quaderni Pignone*, no. 16, 1970.
- [132] M. Wambsganss Jr, D. Coates, and R. Cohen, "Simulation of reciprocating gas compressors with automatic reed valves," *Simulation*, vol. 8, no. 4, pp. 209-214, 1967.
- [133] G. Gatecliff and E. Lady, "Forced Vibration of a Cantilever Valve of Uniform Thickness and Non-Uniform Width," in *International Compressor Engineering Conference*, Purdue University, 1972.
- [134] G. Gatecliff, G. Griner, and H. Richardson, "A compressor valve model for use in daily design work," in *International Compressor Engineering Conference*, Purdue University, 1980.
- [135] K. T. Ooi, G. B. Chai, and E. C. Kwek, "A simple valve model to study the performance of a small compressor," in *International Compressor Engineering Conference*, Purdue University, 1992.
- [136] G. Prater and W. P. Hnat, "Optical measurement of discharge valve modal parameters for a rolling piston refrigeration compressor," *Measurement*, vol. 33, no. 1, pp. 75-84, 2003.
- [137] S. K. Lohn, E. L. L. Pereira, H. F. Camara, and C. J. Deschamps, "Experimental Investigation of Damping Coefficient for Compressor Reed Valves," in *International Compressor Engineering Conference*, Purdue University, 2016.
- [138] D. Woolatt, "A simple numerical solution for compressor valves with one degree of freedom," in *International Compressor Engineering Conference*, Purdue University, 1974.
- [139] H. E. Khalifa and X. Liu, "Analysis of stiction effect on the dynamics of compressor suction valve," in *International Compressor Engineering Conference*, Purdue University, 1998.

- [140] R. A. Pizarro-Recabarren, J. R. Barbosa Jr, and C. J. Deschamps, "Modeling the stiction effect in automatic compressor valves," *International Journal of Refrigeration*, vol. 36, no. 7, pp. 1916-1924, 2013.
- [141] J. Kim, S. Wang, S. Park, K. Ryu, and J. La, "Valve dynamic analysis of a hermetic reciprocating compressor," in *International Compressor Engineering Conference*, Purdue University, 2006.
- [142] Q. Tan, S. Pan, Z. Liu, X. Yu, and Q. Feng, "Numerical study on transient effective flow and force areas of reed valve in a rotary compressor," *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, vol. 228, no. 8, pp. 978-987, 2014.
- [143] K. Kinjo, A. Nakano, T. Hikichi, and K. Morinishi, "Study of CFD considering valve behavior in reciprocating compressor," in *International Compressor Engineering Conference*, Purdue University, 2010.
- [144] A. Mashayekh, T. Jacobs, M. Patterson, and J. Etcheverry, "Prediction of air-fuel ratio control of a large-bore natural gas engine using computational fluid dynamic modeling of reed valve dynamics," *International Journal of Engine Research*, vol. 18, no. 9, pp. 900-908, 2017.
- [145] B. Zhao, X. Jia, S. Sun, J. Wen, and X. Peng, "FSI model of valve motion and pressure pulsation for investigating thermodynamic process and internal flow inside a reciprocating compressor," *Applied Thermal Engineering*, vol. 131, pp. 998-1007, 2018.
- [146] R. Ferreira and D. E. B. Lilie, "Evaluation of the leakage through the clearance between piston and cylinder in hermetic compressors," in *International Compressor Engineering Conference*, Purdue University, 1984.
- [147] W. Reed and J. Hamilton, "Internal leakage effects in sliding vane, rotary compressors," in *International Compressor Engineering Conference*, Purdue University, 1980.
- [148] Y. Huang, "Leakage calculation through clearances," in *International Compressor Engineering Conference*, Purdue University, 1994.
- [149] R. Rodgers and J. Nieter, "Comprehensive analysis of leakage in rotary compressors," in *International Compressor Engineering Conference*, Purdue University, 1996.
- [150] T. Yanagisawa and T. Shimizu, "Leakage losses with a rolling piston type rotary compressor. I. Radical clearance on the rolling piston," *International journal of refrigeration*, vol. 8, no. 2, pp. 75-84, 1985.
- [151] B. Peng, L. Vincent, and W. Jin, "Heat Transfer and Leakage Analysis for R410A Refrigeration Scroll Compressor," in *International Conference on Mechanical Design*, 2017, pp. 1453-1469: Springer.
- [152] G. He, W. Sun, Y. Zhu, X. Yang, and D. Cai, "Experimental performance evaluation on leakage characteristics of R32 in rolling piston type rotary compressor," *International Journal of Refrigeration*, 2018.
- [153] R. Gerlach and R. Berry, "Effect of Heat Transfer and Related Variables on, Compressor Performance'," in *Proc. of the Fourth Annual Reciprocating Machinery Conf., San Antonio X*, 1989, vol. 7.

- [154] S. Brok, S. Touber, and J. Van der Meer, "Modeling of cylinder heat transfer-large effort, little effect?," in *International Compressor Engineering Conference*, Purdue University, 1980.
- [155] B. Prasad, "Heat transfer in reciprocating compressors-a review," in *International Compressor Engineering Conference*, Purdue University, 1998.
- [156] M. S. Chong and H. C. Watson, "Prediction of heat and mass transfer during compression in reciprocating compressors," in *International Compressor Engineering Conference*, Purdue University, 1976.
- [157] G. Recktenwald, J. Ramsey, and S. Patankar, "Predictions of heat transfer in compressor cylinders," in *International Compressor Engineering Conference*, Purdue University, 1986.
- [158] M. Singh and S. Rajvanshi, "Heat transfer between rotating eccentric cylinders with different radii," *International Journal of Heat and Mass Transfer*, vol. 25, no. 11, pp. 1719-1724, 1982.
- [159] R. Keribar and T. Morel, "Heat transfer and component temperature prediction in reciprocating compressors," in *International Compressor Engineering Conference*, Purdue University, 1988.
- [160] W. J. D. Annand, "Heat transfer in the cylinders of reciprocating internal combustion engines," *Proceedings of the Institution of Mechanical Engineers*, vol. 177, no. 1, pp. 973-996, 1963.
- [161] R. Adair, E. Qvale, and J. T. Pearson, "Instantaneous heat transfer to the cylinder wall in reciprocating compressors," in *International Compressor Engineering Conference*, Purdue University, 1972.
- [162] R. Liu and Z. Zhou, "Heat transfer between gas and cylinder wall of refrigerating reciprocating compressor," in *International Compressor Engineering Conference*, Purdue University, 1984.
- [163] M. L. Todescat, F. Fagotti, A. T. Prata, and R. T. d. S. Ferreira, "Thermal energy analysis in reciprocating hermetic compressors," in *International Compressor Engineering Conference*, Purdue University, 1992.
- [164] F. Fagotti, M. L. Todescat, R. T. S. Ferreira, and A. T. Prata, "Heat transfer modeling in a reciprocating compressor," in *International Compressor Engineering Conference*, Purdue University, 1994.
- [165] K. T. Ooi, "Heat transfer study of a hermetic refrigeration compressor," *Applied Thermal Engineering*, vol. 23, no. 15, pp. 1931-1945, 2003.
- [166] M. C. Diniz, E. L. L. Pereira, and C. J. Deschamps, "A lumped-parameter thermal model for scroll compressors including the solution for the temperature distribution along the scroll wraps," *International Journal of Refrigeration*, vol. 53, pp. 184-194, 2015.
- [167] J. Tuhovcak, J. Hejcik, and M. Jicha, "Comparison of heat transfer models for reciprocating compressor," *Applied Thermal Engineering*, vol. 103, pp. 607-615, 2016.
- [168] Z. Liu and W. Soedel, "Modeling Temperatures in High Speed Compressors for the Purpose of Gas Pulsation and Valve Loss Modelling," in *International Compressor Engineering Conference*, Purdue University, 1992.

- [169] S. K. Padhy and S. N. Dwivedi, "Heat transfer analysis of a rolling-piston rotary compressor," *International journal of refrigeration*, vol. 17, no. 6, pp. 400-410, 1994.
- [170] K. M. Tan and K. T. Ooi, "Heat transfer in compression chamber of a revolving vane (RV) compressor," *Applied Thermal Engineering*, vol. 31, no. 8-9, pp. 1519-1526, 2011.
- [171] K. T. Ooi and J. Zhu, "Convective heat transfer in a scroll compressor chamber: a 2-D simulation," *International Journal of Thermal Sciences*, vol. 43, no. 7, pp. 677-688, 2004.
- [172] K. Jang and S. Jeong, "Experimental investigation on convective heat transfer mechanism in a scroll compressor," *International Journal of Refrigeration*, vol. 29, no. 5, pp. 744-753, 2006.
- [173] S. Sun, Y. Zhao, L. Li, and P. Shu, "Simulation research on scroll refrigeration compressor with external cooling," *International journal of refrigeration*, vol. 33, no. 5, pp. 897-906, 2010.
- [174] W. Huagen, X. Ziwen, and S. Pengcheng, "Theoretical and experimental study on indicator diagram of twin screw refrigeration compressor," *International journal of refrigeration*, vol. 27, no. 4, pp. 331-338, 2004.
- [175] M. E. Leader, "Understanding Journal Bearings," in *Vibration Institute Annual Conference Proceedings*, 2012.
- [176] H. Hirani, T. Rao, K. Athre, and S. Biswas, "Rapid performance evaluation of journal bearings," *Tribology international*, vol. 30, no. 11, pp. 825-834, 1997.
- [177] H. Hirani, K. Athre, and S. Biswas, "Rapid and globally convergent method for dynamically loaded journal bearing design," *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, vol. 212, no. 3, pp. 207-214, 1998.
- [178] H. Hirani, K. Athre, and S. Biswas, "Dynamically loaded finite length journal bearings: analytical method of solution," *Journal of tribology*, vol. 121, no. 4, pp. 844-852, 1999.
- [179] H. Hirani, K. Athre, and S. Biswas, "Comprehensive design methodology for an engine journal bearing," *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, vol. 214, no. 4, pp. 401-412, 2000.
- [180] B. Reason and I. Narang, "Rapid Design and Performance Evaluation of Steady-State Journal Bearings—A Technique Amenable to Programmable Hand Calculators," *ASLE TRANSACTIONS*, vol. 25, no. 4, pp. 429-444, 1982.
- [181] F. Martin and C. Lee, "Feed-pressure flow in plain journal bearings," *ASLE TRANSACTIONS*, vol. 26, no. 3, pp. 381-392, 1983.
- [182] S. Sharma and R. Awasthi, "Effect of aspect ratio on the performance and stability of Hydrodynamic Journal Bearings," *International Journal of Advance Research and Innovation*, vol. 4, no. 1, pp. 96-105, 2016.
- [183] H. J. Kim and T. W. Lancey, "Numerical study on the lubrication oil distribution in a refrigeration rotary compressor," *International journal of refrigeration*, vol. 26, no. 7, pp. 800-808, 2003.

- [184] H. J. Kim, "Lubrication oil pumping by utilizing vane motion in a horizontal rotary compressor," *International journal of refrigeration*, vol. 28, no. 4, pp. 498-505, 2005.
- [185] K. M. Kim and K. Y. Cho, "A study on lubricating system of hermetic rotary compressor," in *International Compressor Engineering Conference*, Purdue University, 1988.
- [186] N. Stosic, A. Kovacevic, K. Hanjalic, and L. Milutinovic, "Mathematical modelling of the oil influence upon the working cycle of screw compressors," in *International Compressor Engineering Conference*, Purdue University, 1988.
- [187] T. Itoh, H. Kobayashi, M. Fujitani, and N. Murata, "Study on the oil supply system for rotary compressors," in *International Compressor Engineering Conference*, Purdue University, 1992.
- [188] A. J. Lückmann, M. V. C. Alves, and J. R. Barbosa, "Analysis of oil pumping in a reciprocating compressor," *Applied Thermal Engineering*, vol. 29, no. 14, pp. 3118-3123, 2009.
- [189] A. T. Prata and J. R. Barbosa Jr, "Role of the thermodynamics, heat transfer, and fluid mechanics of lubricant oil in hermetic reciprocating compressors," *Heat Transfer Engineering*, vol. 30, no. 7, pp. 533-548, 2009.
- [190] H. C. Sung, "Tribological characteristics of various surface coatings for rotary compressor vane," *Wear*, vol. 221, no. 2, pp. 77-85, 1998.
- [191] M. J. Moran and H. N. Shapiro, *Fundamentals of engineering thermodynamics*, 6th ed. Hoboken, N.J. : Wiley, c2008., 2008.
- [192] T. L. Bergman, F. P. Incropera, D. P. DeWitt, and A. S. Lavine, *Fundamentals of heat and mass transfer*. John Wiley & Sons, 2011.
- [193] S. C. Chapra and R. P. Canale, *Numerical methods for engineers* 6th ed. Boston : McGraw-Hill Higher Education, c2010., 2010.
- [194] M. Wiggert and D. Potter, "Mechanics of fluids," ed: Brookes/Cole Thomson Learning: Pacific Grove, CA, 2002.
- [195] H. Bean, "Discharge Coefficients of Square-edged Orifice for Measuring the Flow of Air," *Journal Research, NBS*, vol. 2, p. 561, 1929.
- [196] C. L. Hollingshead, M. C. Johnson, S. L. Barfuss, and R. E. Spall, "Discharge coefficient performance of Venturi, standard concentric orifice plate, V-cone and wedge flow meters at low Reynolds numbers," *Journal of Petroleum Science and Engineering*, vol. 78, no. 3-4, pp. 559-566, 2011.
- [197] W. Thomson, *Theory of vibration with applications*. CrC Press, 2018.
- [198] L. Meirovitch, *Fundamentals of vibrations*. Waveland Press, 2010.
- [199] C. E. Wilson, J. P. Sadler, and W. J. Michels, *Kinematics and Dynamics of Machinery, SI Units*, 3rd ed. Pearson Education, 2005.
- [200] T. Yanagisawa, T. Shimizu, I. Chu, and K. Ishijima, "Motion analysis of rolling piston in rotary compressor," in *International Compressor Engineering Conference*, Purdue University, 1982.
- [201] Y. D. Lim, W. C. Poh, and K. T. Ooi, "Leakage Simulation Of A Lubricant-free Rotary Type Swing Compressor Rotor Endface Clearance," in *International Compressor Engineering Conference*, Purdue University, 2018.

- [202] C. F. Gerald, *Applied numerical analysis*. Pearson Education India, 2004.
- [203] M. Ozu and T. Itami, "Efficiency analysis of power consumption in small hermetic refrigerant rotary compressors," *International Journal of Refrigeration*, vol. 4, no. 5, pp. 265-270, 1981.
- [204] R. Shaffer and W. Lee, "Energy Consumption in Hermetic Refrigerator Compressors," in *International Compressor Engineering Conference*, Purdue University, 1976.
- [205] H. J. Huff, D. Lindsay, and R. Radermacher, "Positive displacement compressor and expander simulation," in *International Compressor Engineering Conference*, Purdue University, 2002.
- [206] W. S. Brainerd, C. H. Goldberg, J. C. Adams, A. Garcia, S. McKay, and W. Christian, "Programmer's guide to Fortran 90," *Computers in Physics*, vol. 10, no. 2, pp. 135-136, 1996.
- [207] S. C. Chapra and R. P. Canale, *Numerical methods for engineers*. McGraw-Hill New York, 2012.
- [208] (2018, 20 Nov 2019). *Ask the Expert: How many light duty OEs use HFO-1234yf refrigerant?* Available: <https://www.vehicleservicepros.com/in-the-bay/tools-equipment/air-conditioning-a-c-repair-service/r-1234yf-tools-and-equipment/article/21005163/tech-tip-current-list-of-oes-that-use-hfo1234yf-refrigerant>
- [209] Y. Lee and D. Jung, "A brief performance comparison of R1234yf and R134a in a bench tester for automobile applications," *Applied Thermal Engineering*, vol. 35, pp. 240-242, 2012.
- [210] C. Zilio, J. S. Brown, G. Schiochet, and A. Cavallini, "The refrigerant R1234yf in air conditioning systems," *Energy*, vol. 36, no. 10, pp. 6110-6120, 2011.
- [211] AHRI, *Performance Rating of Positive Displacement Refrigerant Compressors and Compressor Units* (AHRI Standard 540). 2111 Wilson Boulevard, Suite 500, Arlington, VA 22201, USA: Air-Conditioning, Heating, and Refrigeration Institute (AHRI), 2015.
- [212] "Shell Refrigeration Oil S4 FR-F 68 - Technical Data Sheet," ed: Shell Singapore Pte. Ltd., 2014.
- [213] "EZi Series Brochure," ed: Daikin Industries Ltd., 2018.
- [214] T. Thiele. (2017, 27 June 2019). *Find the Right Size Window Air Conditioner for Every Room*. Available: <https://www.thespruce.com/window-air-conditioner-size-1152655>
- [215] T. Jankowski, E. Schmierer, F. Prenger, and S. Ashworth, "A series pressure drop representation for flow through orifice tubes," *Journal of Fluids Engineering*, vol. 130, no. 5, p. 051204, 2008.
- [216] N. Ishii, S. Yamamoto, S. Muramatsu, M. Yamamura, and M. Takahashi, "Optimum combination of parameters for high mechanical efficiency of a scroll compressor," in *International Compressor Engineering Conference*, Purdue University, 1992.
- [217] N. Ishii, M. Yamamura, S. Muramatsu, S. Yamamoto, and M. Sakai, "Mechanical efficiency of a variable speed scroll compressor," in *International Compressor Engineering Conference*, Purdue University, 1990.

- [218] N. Ishii, M. Yamarnura, S. Muramatsu, S. Yamada, and M. Takahashi, "A study on high mechanical efficiency of a scroll compressor with fixed cylinder diameter," in *International Compressor Engineering Conference*, Purdue University, 1994.
- [219] Y. Teh and K. Ooi, "Theoretical study of a novel refrigeration compressor–Part I: Design of the revolving vane (RV) compressor and its frictional losses," *international journal of refrigeration*, vol. 32, no. 5, pp. 1092-1102, 2009.
- [220] N. Ishii, M. Sakai, K. Sano, S. Yamamoto, and T. Otokura, "A fundamental optimum design for high mechanical and volumetric efficiency of compact scroll compressors," in *International Compressor Engineering Conference*, Purdue University, 1996.
- [221] J. Nelson. (2019, 27 June 2019). *Products - Air Compressor*. Available: <http://www.jpnelson.com.sg/index.php/en/products/air-compressor>
- [222] R. L. Mott and J. Tang, *Machine elements in mechanical design*. Pearson Prentice Hall New Jersey, 2004.
- [223] "Alloy 17-4 PH - Inspection Certificate," ed: Advance Materials Supplies Pte. Ltd., 2016.
- [224] E. van der Heide *et al.*, "Wear of aluminium bronze in sliding contact with lubricated stainless steel sheet material," *Wear*, vol. 261, no. 1, pp. 68-73, 2006.
- [225] Aluminium Bronze/Nickel Aluminium Bronze [Online]. Available: <https://www.teckleongmetals.com/products/bronze/aluminium-bronze-nickel-aluminium-bronze>
- [226] F. P. Beer, E. R. Johnston, and J. T. DeWolf, *Mechanics of Materials*. McGraw-Hill, New York, 2002.
- [227] TEF-MET/P [Online]. Available: <https://www.sbs-bearings.com.sg/products/tef-met-p-wrapped-composite-bearing/>
- [228] AISI 4130 [Online]. Available: <http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=m4130r>
- [229] "Precision Rotary Seal (Oil Seal)," ed: Advanced Sealing Devices Pte. Ltd., 2015.
- [230] Aluminium Alloys [Online]. Available: <https://www.efunda.com/materials/alloys/aluminum/properties.cfm>
- [231] "Parker O-ring Handbook," ed: Parker Hannifin Corporation, 2007.
- [232] "Technical O-ring Handbook," ed: ERIKS Group, 2019.
- [233] M. Cavcar, "The international standard atmosphere (isa)," *Anadolu University, Turkey*, vol. 30, p. 9, 2000.
- [234] P. J. Pritchard, J. W. Mitchell, and J. C. Leylegian, *Fox and McDonald's Introduction to Fluid Mechanics, Binder Ready Version*. John Wiley & Sons, 2016.
- [235] S. L. Herman, *Delmar's standard textbook of electricity*. Nelson Education, 2016.
- [236] S. C. Chapra and R. P. Canale, *Numerical methods for engineers*. McGraw-hill New York, 1998.
- [237] AISI 304L [Online]. Available: <http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MQ304L>

- [238] Type T Thermocouple [Online]. Available: <https://www.thermocoupleinfo.com/type-t-thermocouple.htm>
- [239] K. Heinz, B. Müller, and S. Nau, *Fluid sealing technology principles and applications*. Routledge, 1998.
- [240] J. Sullivan and L. Wong, "Wear of aluminium bronze on steel under conditions of boundary lubrication," *Tribology international*, vol. 18, no. 5, pp. 275-281, 1985.
- [241] F. Yoshizumi, Y. Kondoh, T. Moroi, S. Tamano, and Y. Morinishi, "Numerical simulation on the opening delay of a discharge reed valve in compressors," in *International Compressor Engineering Conference*, Purdue University, 2014.
- [242] I. Gonzalez, O. Lehmkuhl, A. Naseri, J. Rigola, and A. Oliva, "Fluid-structure interaction of a reed type valve," in *International Compressor Engineering Conference*, Purdue University, 2016.
- [243] Y. D. Lim, K. S. Yap, and K. T. Ooi, "Optimization of R134a cross vane expander compressor refrigeration system oriented to COP," *Journal of Mechanical Engineering Research*, vol. 9, no. 2, pp. 15-24, 2017.
- [244] H. H. Lee, *Finite Element Simulations with ANSYS Workbench 18*. SDC publications, 2018.
- [245] T. H. Shih, W. W. Liou, A. Shabbir, Z. Yang, and J. Zhu, "A new $k-\epsilon$ eddy viscosity model for high reynolds number turbulent flows," *Computers & Fluids*, vol. 24, no. 3, pp. 227-238, 1995.
- [246] K. T. Aw, "Development of a Lubricant-free Revolving Vane Compressor," School of Mechanical and Aerospace Engineering, Nanyang Technological University, Singapore, 2016.
- [247] W. X. Lim and Y. D. Lim, "Heat Transfer Modelling of a Novel Vane Compressor," School of Mechanical and Aerospace Engineering, Nanyang Technological University, Singapore, 2018.
- [248] F. P. Incropera, A. S. Lavine, T. L. Bergman, and D. P. DeWitt, *Fundamentals of heat and mass transfer*. Wiley, 2007.
- [249] W. M. Rohsenow, J. P. Hartnett, and Y. I. Cho, *Handbook of heat transfer*. McGraw-Hill New York, 1998.
- [250] S. Kakaç, R. K. Shah, and W. Aung, *Handbook of single-phase convective heat transfer*. 1987.
- [251] K. C. Cheng and R. S. Wu, "Viscous dissipation effects on convective instability and heat transfer in plane Poiseuille flow heated from below," *Applied Scientific Research*, vol. 32, no. 4, pp. 327-346, 1976.
- [252] M. M. Khonsari and E. R. Booser, *Applied tribology: bearing design and lubrication*. John Wiley & Sons, 2017.
- [253] J. F. Booker, "Dynamically loaded journal bearings: mobility method of solution," *Journal of Basic Engineering*, vol. 87, no. 3, pp. 537-546, 1965.
- [254] K. M. Tan and K. T. Ooi, "Journal bearings design for a novel revolving vane compressor," *International Journal of Refrigeration*, vol. 34, no. 1, pp. 94-104, 2011.

Appendix A Comparison of Cylinder and Housing Volumes between Swing Compressor and U-Vane Compressor

In U-Vane compressor, a new vane bush is introduced. It is expected that the size of the cylinder and housing volumes in a swing compressor can be reduced with the proposed new vane bush. An estimation on the size of the cylinder and housing volumes of both swing compressor and UVC is made in this Appendix.

With reference to Pan *et al.* [108], the dimensions of the swing compressor and U-Vane compressor are summarised in Table A.1. While the definitions of the dimensions are presented in Figure A.1.

Table A.1 Comparison of dimensions between swing compressor and UVC

	Swing Compressor [108]	U-Vane Compressor
Cylinder diameter (internal), $D_{i,cy}$	41.0 mm	41.0 mm
Cylinder diameter (external), $D_{o,cy}$	65.0 mm	56.0 mm
Rotor diameter (internal), $D_{i,ro}$	24.9 mm	24.9 mm
Rotor diameter (external), $D_{o,ro}$	32.9 mm	32.9 mm
Housing diameter (internal), $D_{i,hs}$	85.0 mm	69.0 mm
Housing diameter (external), $D_{o,hs}$	91.0 mm	75.0 mm
Flange angle, α_{flange}	80 °	80 °
Distance between bush slot and housing, h_{cyl}	5.0 mm	5.0 mm
Cylinder height, l_{cyl}	22.0 mm	22.0 mm

Figure A.1 Illustration of dimensions definition (a) swing compressor and (b) UVC

In this estimation, both compressors are designed with the same swept volume of $10.5 \text{ cm}^3/\text{rev}$, and the same major compressor dimensions such as the inner cylinder diameter, the outer rotor diameter and the compressor axial length. The thickness of the housing for both compressors are kept at a constant of 3.0 mm. The remaining dimensions of the UVC are determined reasonably while

keeping the distance h_{cyl} same as that of the swing compressor for a fair comparison. Neglecting the bush slot and the vane compartment, the volumes of the cylinder ($V_{cyl,swing}$) and the housing ($V_{hs,swing}$) in the swing compressor can be obtained below:

$$\begin{aligned}
 V_{swing} &= V_{cyl,swing} + V_{hs,swing} \\
 &= \left[\frac{\alpha_{flange}}{360} (D_{i,hs}^2 - D_{o,cy}^2) + (D_{o,cy}^2 - D_{i,cy}^2) + (D_{o,hs}^2 - D_{i,hs}^2) \right] \frac{\pi}{4} \times l_{cyl} \\
 &= \left[\frac{80}{360} (85^2 - 65^2) + (65^2 - 41^2) + (91^2 - 85^2) \right] \frac{\pi}{4} \times 22 \\
 &= 73722.7 \text{ mm}^3
 \end{aligned}$$

Similarly, for UVC, the volumes of the cylinder ($V_{cyl,UVC}$) and the housing ($V_{hs,UVC}$):

$$\begin{aligned}
 V_{UVC} &= V_{cyl,UVC} + V_{hs,UVC} \\
 &= \left[\frac{\alpha_{flange}}{360} (D_{i,hs}^2 - D_{o,cy}^2) + (D_{o,cy}^2 - D_{i,cy}^2) + (D_{o,hs}^2 - D_{i,hs}^2) \right] \frac{\pi}{4} \times l_{cyl} \\
 &= \left[\frac{80}{360} (69^2 - 56^2) + (56^2 - 41^2) + (75^2 - 69^2) \right] \frac{\pi}{4} \times 22 \\
 &= 46308.9 \text{ mm}^3
 \end{aligned}$$

Comparing the volumes of the cylinder and the housing of the two compressors:

$$\begin{aligned}
 \frac{V_{swing} - V_{UVC}}{V_{swing}} \times 100\% &= \frac{73722.7 - 46308.9}{73722.7} \times 100\% \\
 &= 37.18\% \\
 &\approx 37\%
 \end{aligned}$$

Hence, it is approximated that with the same volumetric displacement of $10.5 \text{ cm}^3/\text{rev}$, about 37 % of the cylinder and housing volumes can be reduced from the swing compressor by using the new vane bush in the UVC. It is noted that with the same volumetric displacement, operational condition, working fluid and lubricant, the power consumed by both swing compressor and UVC would be similar.

Appendix B Derivation of Angular acceleration

The derivation of angular acceleration ($\frac{d^2\gamma_1}{dt^2}$) of vane bush takes its lead

from:

$$\begin{aligned} \frac{d^2\gamma_1}{dt^2} &= \frac{d\gamma_1}{dt} = \frac{d}{dt} \left(\frac{d\gamma_1}{dl_v} \frac{dl_v}{d\theta_1} \frac{d\theta_1}{dt} \right) \\ \rightarrow \quad \frac{d^2\gamma_1}{dt^2} &= \frac{d}{dt} \left(\frac{d\gamma_1}{dl_v} \right) \frac{dl_v}{d\theta_1} \frac{d\theta_1}{dt} + \frac{d\gamma_1}{dl_v} \frac{d}{dt} \left(\frac{dl_v}{d\theta_1} \right) \frac{d\theta_1}{dt} + \frac{d\gamma_1}{dl_v} \frac{dl_v}{d\theta_1} \frac{d^2\theta_1}{dt^2} \end{aligned} \quad (B1)$$

Since the shaft rotational speed $\frac{d\theta_1}{dt}$ is assumed to be constant, the last term on the right hand side is zero. The following terms are derived and expressed below:

$$\begin{aligned} \frac{d\gamma_1}{dl_v} &= \frac{1}{\sqrt{1 - \left[\frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})} \right]^2}} \\ &\times \left[\frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})^2} - \frac{1}{r_{vb} + r_{cyl}} \right] \end{aligned} \quad (B2)$$

$$\frac{dl_v}{d\theta_1} = \frac{e(r_{vb} + r_{cyl})\sin\theta_1}{\sqrt{e^2 + (r_{vb} + r_{cyl})^2 - 2e(r_{vb} + r_{cyl})\cos\theta_1}} \quad (B3)$$

$$\begin{aligned}
\frac{d}{dt} \left(\frac{d\gamma_1}{dl_v} \right) = & \left\{ \frac{1}{\sqrt{1 - \left[\frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})} \right]^2}} \left[\frac{1}{(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})} \right. \right. \\
& \left. \left. - \frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})^3} \right] \right. \\
& \left. - \frac{1}{\left\{ 1 - \left[\frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})} \right]^2 \right\}^{1.5}} \right. \\
& \times \frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})} \\
& \left. \times \left[\frac{(r_{vb} + r_{cyl})^2 + (r_{vb} + l_v + r_{ro})^2 - e^2}{2(r_{vb} + r_{cyl})(r_{vb} + l_v + r_{ro})^2} - \frac{1}{r_{vb} + r_{cyl}} \right]^2 \right\} \frac{dl_v}{dt}
\end{aligned}
\tag{B4}$$

$$\begin{aligned}
\frac{d}{dt} \left(\frac{dl_v}{d\theta_1} \right) = & \left\{ \frac{e(r_{vb} + r_{cyl}) \cos \theta_1}{\sqrt{e^2 + (r_{vb} + r_{cyl})^2 - 2e(r_{vb} + r_{cyl}) \cos \theta_1}} - \right. \\
& \left. \frac{[e(r_{vb} + r_{cyl}) \sin \theta_1]^2}{[e^2 + (r_{vb} + r_{cyl})^2 - 2e(r_{vb} + r_{cyl}) \cos \theta_1]^{1.5}} \right\} \frac{d\theta_1}{dt}
\end{aligned}
\tag{B5}$$

Inputting Equations (B2) - (B5) into (B1), the angular acceleration ($\frac{d^2\gamma_1}{dt^2}$)

can be obtained.

Appendix C Rotor End-face Clearance Leakage and Heat Transfer Model

This appendix presents the formulation of the leakage through rotor end-face clearance and the heat transfer in the UVC.

C.1 Leakage through Rotor End-face Clearance

The leakage flows presented in Section 5.3 are modelled as Fanno Flow. However, the intricacy of the rotor geometry, on top of its swaying motion during operation, is causing the frictional channel length of Fanno Flow to vary and hence, the Fanno Flow model is not appropriate for the leakage flow through the rotor end-face clearance. The analyses carried out by former researchers on the leakage through the roller end-face clearance in a rolling piston compressor are not appropriate either because of the difference in the rotor motion between both compressors [78, 79]. Nevertheless, with the advancement of the Computational Fluid Dynamics (CFD), the leakage through the rotor end-face clearance can be incorporated by numerical approach [201] into the mathematical model.

Basically, many factors could affect the leakage flow such as the clearance dimension, the pressure ratio between the working chambers, the operating speed and the lubricant properties. In actual operation of a compressor, the leakage flow is a complex phenomenon because it can be a single-phase gas flow, single-phase oil flow with dissolved gas, or two-phase oil-gas mixture flow. Therefore, assumptions must be stated prior to the numerical study of the leakage flow.

In this section, the leakage flow through rotor end-face clearance of the UVC prototype with the experimental operational conditions is presented. The CFD simulation is conducted in the absence of lubricant so that the leakage flow is described as a single-phase gas flow. Also, ideal gas, constant temperature and no slip condition for the fluid flow during the entire process are assumed. The rotor geometry and operational parameters are presented in Table C.1.

Table C.1 Dimensions and operational conditions for numerical simulation

Components and Dimensions	
Rotor radius	31.50 mm
Vane width	6.00 mm
Vane length	17.00 mm
Shoulder slot width	4.00 mm
Shoulder slot length	4.70 mm
Shaft radius	12.50 mm
End-face clearance	0.08 mm
Operational Conditions	
Working fluid	Air
Suction chamber pressure	101.30 kPa (abs.)
Compression chamber pressure	700.00 kPa (abs.)
Temperature	40.00 °C

The fluid domain and the boundary conditions are shown in Figure C.1. Noting that the fluid domain and the boundary conditions shown are only for the instance where the shaft rotational angle is 180°. During operation, the centre of the rotor orbits about the shaft centre with an offset distance, which causes the shaft position with respect to the rotor, and the radial contact line between rotor and cylinder, to vary. Therefore, the boundary condition and the leakage flow path geometry at each rotational angle change.

The CFD simulation was carried out in ANSYS Workbench 18.1 [244] with the realizable k - ε turbulence flow model [245] as the solver since it gives an approximation with error less than 2.3 % in the compressor leakage flow, compared to the analytical approach [246]. The simulation was conducted over one revolution, with intervals of 45° and the corresponding fluid domain and boundary condition setups at each interval. The pressure ratio between the working chambers at each angle is obtained from the theoretical simulation of UVC. From the result, the relationship between the average flow velocity and pressure ratio throughout one revolution can be acquired. Ultimately, the leakage

flow through the rotor end-face clearance will be modelled as fluid flow through a rectangular duct.

Figure C.1 Fluid domain of the leakage through rotor end-face clearance and boundary conditions (at 180°)

Figure C.2 Result of CFD numerical simulation at 180°: (a) velocity profile (b) pressure distribution

Figure C.2 shows the result from the CFD simulation at 180°. It is observed that the flow across the shorter flow path has higher velocity when compared to the flow across the longer flow path, which are about 200 m/s and 50 m/s respectively. Due to the complicated rotor geometry, the lengths of every flow paths vary throughout the entire rotor section and longer flow path creates more resistance to the leakage. Hence, the leakage tends to flow through a shorter path

with less flow resistance. This also indicates that modelling the leakage flow through the rotor end-face clearance as Fanno Flow is inappropriate due to the varying ‘channel length’. The relationship between the average velocity of the flow out from the suction chamber and the pressure ratio is shown in Figure C.3.

Figure C.3 Leakage flow velocity curve fit

The logarithm used in curve fitting for the relationship between the average velocity and the pressure ratio is expressed in Equation (C1):

$$v_{leak,ref} = 71.921 \ln \left(\frac{P_{com}}{P_{suc}} \right) - 2.9277 \quad (C1)$$

This velocity is assumed to be the velocity of fluid flow across the projected length (l_{AB} , Equation (5.14)) which separates the suction and compression chambers as shown in Figure 5.3. Hence, the leakage mass flow rate can be obtained with the following equation:

$$\left(\frac{dm}{dt} \right)_{leak,ref} = \rho_{suc} \delta_{ref} l_{AB} v_{leak,ref} \quad (C2)$$

where,

$\left(\frac{dm}{dt} \right)_{leak,ref}$ is the leakage mass flow rate through the rotor end-face into the suction chamber (kg/s)

ρ_{suc} is the density of fluid at the suction chamber (kg/m³)

$v_{leak,ref}$ is the average leakage flow velocity to the suction chamber (m/s)

Note that the leakage flow model from the CFD study is done in the absence of the lubricant. In actual operation of the compressor, the lubricant could act as the sealant of leakage paths and the gas leakage flow could be reduced. To account for this, a flow coefficient ($C_{leak,ref}$) is added into Equation (C3) to obtain the actual leakage mass flow rate:

$$\left(\frac{dm}{dt} \right)_{leak,ref} = C_{leak,ref} \rho_{suc} \delta_{ref} l_{AB} v_{leak,ref} \quad (C3)$$

C.2 Heat Transfer Model

A simplified heat transfer model of U-Vane compressor has been formulated. It includes two parts: thermal resistance model and lumped thermal conductance model [247]. The following assumptions are applied in the model:

- Quasi-steady state condition, the properties of working fluid and material propagate instantaneously
- Elements are isothermal for lumped thermal conductance model
- Negligible viscous dissipation by lubricating oil to the cylinder wall, eccentric and rotor in lumped thermal conductance model
- Heat generation from the friction is assumed as a constant heat flux
- One-dimensional heat transfer between adjacent elements in either the radial or vertical directions
- The instantaneous temperatures, as well as the instantaneous properties of the working fluid in suction, compression and housing chambers remain constant at each rotational angle during heat transfer process
- The housing chamber temperature and atmosphere temperature remain constant at steady state
- Negligible thermal contact resistance between the compressor components

- Negligible radiation heat transfer between the compressor components

C.2.1 Heat Transfer Path

The heat transfer paths in UVC are identified. As the heat transfers are assumed to be one-dimensional, they are separated into horizontal direction heat transfer (radial direction) and vertical direction heat transfer (axial direction). The heat transfers in the horizontal direction are shown in Figure C.4 while the heat transfers in the vertical direction are shown in Figure C.5. Noting that the geometry of the compressor is simplified for ease in modelling the heat transfer.

Figure C.4 Heat transfer paths in horizontal direction

Figure C.5 Heat transfer paths in vertical direction

To account for the oil flow through the clearance between rotor end-face and cylinder end-plate, the heat transfer paths G and J can be divided into two sub-paths, namely G_o , G_i , J_o and J_i respectively, as illustrated in Figure C.6. The heat transfer paths in both horizontal direction and vertical direction are tabulated in Table C.2.

Figure C.6 Heat transfers due to oil flow

Table C.2 Summary of heat transfer paths

Path	Symbol	Description
------	--------	-------------

A	$\dot{Q}_{schc}$	Heat transfer rate between suction chamber and housing chamber
B	$\dot{Q}_{seccro}$	Heat transfer rate between suction chamber and eccentric-rotor assembly
C	$\dot{Q}_{dscv}$	Heat transfer rate between compression chamber and suction chamber through vane
D	$\dot{Q}_{dchc}$	Heat transfer rate between compression chamber and housing chamber
E	$\dot{Q}_{decro}$	Heat transfer rate between compression chamber and eccentric-rotor assembly
F	$\dot{Q}_{scatm}$	Heat transfer rate between suction chamber and atmosphere
G _i	$\dot{Q}_{oilecro,up}$	Heat transfer rate between lubricating oil and eccentric-rotor assembly through upper rotor end-face clearance
G _o	$\dot{Q}_{oilatm,up}$	Heat transfer rate between lubricating oil and atmosphere
H	$\dot{Q}_{dcatm}$	Heat transfer rate between compression chamber and atmosphere
I	$\dot{Q}_{scoilsmp}$	Heat transfer rate between suction chamber and oil sump
J _i	$\dot{Q}_{oilecro,low}$	Heat transfer rate between lubricating oil and eccentric-rotor assembly through lower rotor end-face clearance
J _o	$\dot{Q}_{oilsmp,low}$	Heat transfer rate between lubricating oil and oil sump
K	$\dot{Q}_{dcoilsmp}$	Heat transfer rate between compression chamber and oil sump

Recalls that in the heat transfer model of UVC, it is assumed that the heat transfer occurs instantaneously at each rotational angle. Thus, the heat transfer with respect to the shaft rotational angle is the instantaneous heat transfer at each rotational angle.

C.2.2 Heat Transfer Mechanisms

The mechanisms involve in the heat transfers in UVC are mainly conductive and convective heat transfers. Conductive heat transfer takes place when there is temperature difference in a solid material. The instantaneous conductive heat transfer rate is expressed below [248]:

$$\dot{Q} = -\frac{k_{mat}A}{L}\Delta T \quad (C4)$$

where

k_{mat} is the thermal conductivity of the material (for prototype, 17-4 PH stainless steel and AISI 4130) (W/m•K)

A is the cross-sectional (or “wetted”) area normal to the direction of heat transfer (m²)

L is the distance between the ends of the material (or thickness of the material) in the heat transfer direction (m)

ΔT is the temperature difference between the ends of the material in the heat transfer direction (°C or K)

Besides, as the components of UVC are mostly cylindrical, i.e. the cylinder and the rotor, the instantaneous conductive heat transfer of these components can be treated by heat transfer through a hollow cylinder, as presented below:

$$\dot{Q} = -\frac{2\pi k_{mat}l}{\ln \frac{r_o}{r_i}}\Delta T_{io} \quad (C5)$$

where,

l is the axial length of the hollow cylinder (m)

r_o is the outer radius of the hollow cylinder (m)

r_i is the inner radius of the hollow cylinder (m)

ΔT_{io} is the temperature difference between the outer surface and inner surface of the hollow cylinder (°C or K)

On the other hand, the convective heat transfers are found between the working fluid in the working chamber and the adjacent walls (cylinder, rotor, cover and lower bearing), between the oil sump and the lower bearing wall and between the housing chamber and the outer cylinder wall. The instantaneous convective heat transfer is given by:

$$\dot{Q} = h_i A_s (T_w - T_f) \quad (C6)$$

where,

h_i is the instantaneous convective heat transfer coefficient (W/m²•K)

A_s is the wetted surface area (m²)

T_w is the temperature of the wall (K or °C)

T_f is the temperature of the fluid (K or °C)

In general, the instantaneous convective heat transfer coefficient can be obtained from the following equations:

$$h_f = \frac{\text{Nu} \times k_f}{D_e} \quad (\text{C7})$$

$$\text{Nu} = C \times \text{Re}^m \times \text{Pr}^n \quad (\text{C8})$$

$$\text{Re} = \frac{\rho_f V_{ch} D_e}{\mu_f} \quad (\text{C9})$$

$$\text{Pr} = \frac{c_{p,f} \mu_f}{k_f} \quad (\text{C10})$$

where,

h_f is the convective heat transfer coefficient of the fluid (W/m²•K)

k_f is the thermal conductivity of the fluid (W/m•K)

D_e is the equivalent hydraulic diameter (m)

Nu is the Nusselt number (-)

Re is the Reynold number (-)

Pr is the Prandtl number (-)

ρ_f is the fluid density (kg/m³)

V_{ch} is the characteristic velocity of the fluid (m/s)

μ_f is the dynamic viscosity of the fluid (Pa•s)

$c_{p,f}$ is the specific heat capacity of fluid (J/kg•K)

The Nusselt number (Nu) correlation has to be obtained empirically, which is discussed in the following sections.

Convective Heat transfer Coefficient in Working Chamber

The instantaneous convective heat transfer coefficient has to be obtained from heat transfer correlations. For the convective heat transfer of the working

fluid in the working chamber in UVC, the instantaneous convective heat transfer coefficient is obtained by using the correlation developed by Liu and Zhou [162]:

$$\text{Nu}_l = 0.75 \text{Re}^{0.8} \text{Pr}^{0.6} \quad (\text{C11})$$

It is noticed that the original work by Liu and Zhou was done on a reciprocating compressor, which the definitions of the equivalent hydraulic diameter and the characteristic velocity are not applicable. However, Tan and Ooi [170] had modified the equivalent hydraulic diameter and the characteristic hydraulic diameter and applied on revolving vane compressor chamber. In their approach, the working fluid flow field is emphasised as it affects the two parameters mentioned above (Figure C.7a). The similar approach done by Tan and Ooi is carried out in determining the equivalent hydraulic diameter and the characteristic velocity of the working fluid in the UVC by Lim *et al.* [247], as illustrated in Figure C.7b. In the case of UVC, the flow in the suction chamber is assumed to be turbulent flow while the flow near the outer rotor wall is largely following the moving direction of the wall. Similarly, for the flow in compression chamber, the fluid near to the outer rotor wall is being pushed by the rotor and hence it must flow in the same direction of the moving rotor. Since there is no outlet in compression chamber before the valve opens, eddy will be generated near the inner cylinder wall, as shown in Figure C.7b. However, considering the short time taken by the compression process, the working fluid near the rotor wall may not able to reach the inner cylinder wall before the discharge begins. Thus, the flow in the compression chamber is assumed to follow the moving direction of the rotor.

Figure C.7 Expected working fluid flow field in (a) revolving vane compressor [170] and (b) U-Vane compressor [247]

From Section 4.3.2, the resultant motion of the rotor is analysed to be the combination of rotational motion and translational motion. Therefore, the velocity of the working fluid in the working chamber can be expressed as follows:

$$V_{ch,f} = \sqrt{\left(\frac{dl_v}{dt}\right)^2 + \left(\frac{d\gamma_1}{dt} r_{ro}\right)^2} \quad (C12)$$

Additionally, the equivalent hydraulic diameters of suction chamber and compression chamber are formulated respectively:

$$D_{e,sc} = \frac{4A_{sc}}{P_{sc}} = \frac{2(\theta_2 l_{com} r_{ro} + l_v l_{com})}{\theta_2 r_{ro} + l_v + l_{com}} \quad (C13)$$

$$D_{e,dc} = \frac{4A_{dc}}{P_{dc}} = \frac{2[(2\pi - \theta_2) l_{com} r_{ro} + l_v l_{com}]}{(2\pi - \theta_2) r_{ro} + l_v + l_{com}} \quad (C14)$$

where,

$V_{ch,f}$ is the characteristic velocity of the working fluid in the working chamber (m/s)

$D_{e,sc}$ is the equivalent hydraulic diameter of suction chamber (m)

$D_{e,dc}$ is the equivalent hydraulic diameter of compression chamber (m)

A_{sc} is the total wetted area from the view of working fluid in the suction chamber (m²)

A_{dc} is the total wetted area from the view of working fluid in the compression chamber (m^2)

P_{sc} is the perimeter of the total wetted area in the suction chamber (m)

P_{dc} is the perimeter of the total wetted area in the compression chamber (m)

Hence, combining the Equations (C7) – (C14), the instantaneous convective heat transfer coefficient of the working fluid in the working chamber can be obtained.

Convective Heat transfer Coefficient in Housing Chamber

When the working fluid is discharged out to the housing chamber from the working chamber, it is assumed that the flow in the housing chamber is separated in to two opposite directions and circulating temporarily in the housing chamber, as illustrated in Figure C.8a.

Figure C.8 Schematic diagram of (a) flow circulating in the housing chamber and (b) rectangular duct model

Moreover, due to the continuous discharge pulsation and high flow speed of the discharging working fluid, the flow in the housing chamber can be modelled as turbulent flow in a rectangular duct, as presented in Figure C.8b. Hence, the average velocity of the flow is taken as the characteristic velocity, which is defined as:

$$V_{ch,hc} = \frac{\dot{m}_{disc}}{2(r_{hs} - r_{ocyl})\rho_{hc}l_{com}} \quad (C15)$$

and the corresponding equivalent hydraulic diameter is modified into the equivalent hydraulic diameter in circular duct as expressed below [249]:

$$D_{e,hc} = \frac{2(r_{hs} - r_{ocyl})l_{com}}{(r_{hs} - r_{ocyl} + l_{com})} \quad (C16)$$

where,

$V_{ch,hc}$ is the characteristic velocity of the working fluid in housing chamber (m/s)

$D_{e,hc}$ is the equivalent hydraulic diameter of the circular duct in the housing chamber (m)

$\dot{m}_{disc}$ is the mass flow rate in the housing chamber (kg/s)

r_{hs} is the radius of the inner housing (m)

ρ_{hc} is the density of the working fluid in the housing chamber (kg/m³)

The obtained $V_{ch,hc}$ and $D_{e,hc}$ can be used to calculate the values of Reynold number and Prandtl number calculated from Equations (C9) and (C10). Subsequently, the convective heat transfer coefficient in the housing chamber can be calculated by Equation (C7), with the Gnielinski's correlation [250] given below:

$$\begin{aligned} \text{Nu} &= 0.0214(\text{Re}^{0.8} - 100)\text{Pr}^{0.4} & \text{for} & \quad 0.5 \leq \text{Pr} \leq 1.5 \ \& \ 10^4 \leq \text{Re} \leq 5 \times 10^6 \\ \text{Nu} &= 0.012(\text{Re}^{0.87} - 280)\text{Pr}^{0.4} & \text{for} & \quad 1.5 \leq \text{Pr} \leq 500 \ \& \ 3 \times 10^3 \leq \text{Re} \leq 10^6 \end{aligned} \quad (C17)$$

Natural Convection in Oil Sump

The oil sump is assumed to be a big reservoir, which the velocity of the lubricant in the oil sump is negligible. Hence, the convective heat transfer between the oil sump and the lower bearing (base of the cylinder) is a natural convective heat transfer. It is modelled as the case of horizontal plate with hot surface facing downward [248]. The following correlation is applied for this case:

$$\text{Nu}_L = 0.27\text{Ra}_L^{0.25} \quad \text{for} \quad 10^5 \leq \text{Ra}_L \leq 10^{10} \quad (C18)$$

$$\text{Ra}_L = \text{Gr}_L \times \text{Pr} \quad (C19)$$

$$\text{Gr}_L = \frac{g\beta|T_w - T_{oilsmp}|L_c^3}{\nu^2} \quad (C20)$$

Where,

Ra_L is the Rayleigh number (-)

Gr_L is the Grashof number (-)

g is the acceleration of gravity (m/s^2)

β is the thermal expansion coefficient of the fluid (K^{-1})

T_{oilump} is the temperature of the oil in oil sump (K)

ν is the kinematic viscosity of the oil (m^2/s)

L_c is the characteristic length (m)

Basically, L_c is defined as the ratio of heat transfer area to the perimeter of that area. Since the lower bearing (cylinder base) is a circular plate, the characteristic length is defined as:

$$L_c = \frac{A_{base}}{P_{base}} = \frac{\pi r_{ocyl}^2}{2\pi r_{ocyl}} = \frac{r_{ocyl}}{2} \quad (C21)$$

On the other hand, the natural convection also occurs at the cover, in this case, the hot horizontal plate is facing upward, and the following Nusselt number is applied [248]:

$$\begin{aligned} Nu_L &= 0.54 Ra_L^{0.25} & for & \quad 10^4 \leq Ra_L \leq 10^7 \\ Nu_L &= 0.15 Ra_L^{1/3} & for & \quad 10^7 \leq Ra_L \leq 10^{11} \end{aligned} \quad (C22)$$

The corresponding fluid properties at the cover will be the air properties in the atmosphere.

Viscous Dissipation of the Lubricating Oil

When the lubricant flows through the rotor end-face clearance, it reduces the friction between the rubbing surfaces during compressor operation. The high speed lubricant flow also induces heat transfer between itself and the adjacent walls. Furthermore, as the lubricant is viscous, the shear stress that acting on the lubricant causes its temperature to increase. This type of heat transfer is termed viscous dissipation. It is significant when the flow channel is micro-sized and the fluid is highly viscous. Since the clearance dimension is much smaller compared to the rotor diameter, the oil flow in the clearance is modelled as the laminar flow through parallel plate duct with significant viscous dissipation effect [251].

In this case, the hydraulic diameter is defined as $D_{e,oilcl} = 2\delta$, where δ is the dimension of the clearance gap. The correlation established by Cheng and Wu [251] is applied, which accounts for different temperatures between the two adjacent plates/walls. Assuming the temperature of the walls, $T_{w1} > T_{w2}$, the correlation is expressed as:

$$\begin{aligned} \text{Nu}_{\text{lub}} &= \frac{4(1 - 6\text{Br}_{\text{lub}})}{1 - \frac{48}{35}\text{Br}_{\text{lub}}} && \text{for higher temp., } (T_{w1}) \\ \text{Nu}_{\text{lub}} &= \frac{4(1 - 6\text{Br}_{\text{lub}})}{1 - \frac{48}{35}\text{Br}_{\text{lub}}} && \text{for lower temp., } (T_{w2}) \end{aligned} \quad (\text{C23})$$

$$\text{Br}_{\text{lub}} = \frac{\mu_{\text{lub}} V_{\text{lub}}^2}{k_{\text{lub}} (T_{w,\text{avg}} - T_{\text{lub}})} \quad (\text{C24})$$

where,

Br_{lub} is the Brinkman number of the lubricant, which is the ratio of the viscous heat dissipation to the heat transport by the molecular conduction (-)

V_{lub} is the lubricant flow velocity (m/s)

k_{lub} is the thermal conductivity of the lubricant (W/m•K)

$T_{w,\text{avg}}$ is the average temperature of the walls (K)

T_{lub} is the lubricant temperature (K)

C.2.3 Thermal Resistance Model

Upon developed the heat transfer mechanism, it is realised multiple mechanisms are required for the heat transfer paths in UVC. Hence, the heat transfer problems are solved by the thermal resistance approach. With this approach, the heat transfer rate with series and parallel of mechanisms can be solved by an equivalent thermal circuit, where the thermal resistances (R) of each of the mechanisms are defined by:

$$R_{\text{cond,plane}} = \frac{L}{k_{\text{mat}} A} \quad (\text{C25})$$

$$R_{cond,cyl} = \frac{\ln \frac{r_o}{r_i}}{2\pi k_{mat} l_{com}} \quad (C26)$$

$$R_{conv} = \frac{1}{h_t A_s} \quad (C27)$$

Then, the heat transfer rate can be expressed as follows:

$$\dot{Q} = \frac{T_1 - T_2}{R_{eff}} \quad (C28)$$

where T_1 and T_2 are the temperatures at two ends and R_{eff} is the effective thermal resistance between T_1 and T_2 . The effective thermal resistances of the heat transfer paths are summarised in Table C.3.

Table C.3 Summary of the effective thermal resistance

Heat Transfer Path	Effective Thermal Resistance
A, $\dot{Q}_{schc}$	$R_{schc} = \frac{1}{h_{t,sc} A_{sc cyl}} + \frac{\ln \left(\frac{r_{ocyl}}{r_{cyl}} \right)}{\theta_1 k_{mat} l_{com}} + \frac{1}{h_{t,hc} A_{sc cyl}}$
B, $\dot{Q}_{scecro}$	$R_{scecro} = \frac{1}{h_{t,sc} A_{scecro}} + \frac{\ln \left(\frac{r_{ro}}{r_{roi}} \right)}{\theta_2 k_{mat} l_{com}}$
C, $\dot{Q}_{dcscv}$	$R_{dcscv} = \frac{1}{h_{t,dc} A_{w,v}} + \frac{w_{vro}}{k_{mat} A_{w,v}} + \frac{1}{h_{t,sc} A_{w,v}}$
D, $\dot{Q}_{dchc}$	$R_{dchc} = \frac{1}{h_{t,dc} A_{dc cyl}} + \frac{\ln \left(\frac{r_{ocyl}}{r_{cyl}} \right)}{(2\pi - \theta_1) k_{mat} l_{com}} + \frac{1}{h_{t,hc} A_{dc cyl}}$
E, $\dot{Q}_{dcecro}$	$R_{dcecro} = \frac{1}{h_{t,dc} A_{dcecro}} + \frac{\ln \left(\frac{r_{ro}}{r_{roi}} \right)}{(2\pi - \theta_2) k_{mat} l_{com}}$
F, $\dot{Q}_{scatm}$	$R_{scatm} = \frac{1}{h_{t,sc} A_{sc,vert}} + \frac{t_{cover}}{k_{mat} A_{schc,vert}} + \frac{1}{h_{t,atm} A_{sc,vert}}$

$G_i, \dot{Q}_{oilecro,up}$	$R_{oilecro,up} = \frac{1}{h_{t,lub} A_{ecro,vert}}$
$G_o, \dot{Q}_{oilatm,up}$	$R_{oilatm,up} = \frac{1}{h_{t,lub} A_{ecro,vert}} + \frac{t_{cover}}{k_{mat} A_{ecro,vert}} + \frac{1}{h_{t,atm} A_{ecro,vert}}$
$H, \dot{Q}_{dcatm}$	$R_{dcatm} = \frac{1}{h_{t,dc} A_{dchc,vert}} + \frac{t_{cover}}{k_{mat} A_{dchc,vert}} + \frac{1}{h_{t,atm} A_{dchc,vert}}$
$I, \dot{Q}_{scoilsmp}$	$R_{scoilsmp} = \frac{1}{h_{t,oil} A_{scoilsmp,vert}} + \frac{t_{base}}{k_{mat} A_{scoilsmp,vert}} + \frac{1}{h_{t,oilsmp} A_{scoilsmp,vert}}$
$J_i, \dot{Q}_{oilecro,low}$	$R_{oilecro,low} = \frac{1}{h_{t,oil} A_{ecrooilsmp,vert}}$
$J_o, \dot{Q}_{oil oilsmp}$	$R_{oil oilsmp} = \frac{1}{h_{t,oil} A_{ecrooilsmp,vert}} + \frac{t_{base}}{k_{mat} A_{ecrooilsmp,vert}} + \frac{1}{h_{t,oilsmp} A_{ecrooilsmp,vert}}$
$K, \dot{Q}_{dcoilsmp}$	$R_{dcoilsmp} = \frac{1}{h_{t,dc} A_{dcoilsmp,vert}} + \frac{t_{base}}{k_{mat} A_{dcoilsmp,vert}} + \frac{1}{h_{t,oilsmp} A_{dcoilsmp,vert}}$

C.2.4 Lumped Thermal Conductance Model

The thermal resistances of the heat transfer paths have been developed and presented in previous sections. In this section, the temperatures of the components will be determined by lumped thermal conductance approach [170]. Consider only the main components in the UVC, they are split into a total of 19 discrete elements, as shown in Figure C.9.

Figure C.9 Discrete elements from the main components

With the assumptions of quasi-steady state, negligible volume change caused by thermal expansion, and uniform temperature for an individual element, applying First Law of Thermodynamics, the total heat transfer rate of i^{th} element can be represented by:

$$\dot{Q}_i = \sum_j^n H_{i,j} (T_i - T_j) = \dot{C}_i \quad (\text{C29})$$

where,

$\dot{C}_i$ is the heat generation in i^{th} element (W)

$H_{i,j}$ is the thermal conductance coefficient, which is related to the thermal properties, geometry of the element and the heat transfer mechanism (W/K)

An example is shown below for element 13:

$$H_{13,7} (T_{13} - T_7) + H_{13,10} (T_{13} - T_{10}) + H_{13,14} (T_{13} - T_{14}) = C_{13} \quad (\text{C30})$$

Rearranging equation (C30), the following equation is obtained:

$$U_{13,13} T_{13} + U_{13,7} T_7 + U_{13,10} T_{10} + U_{13,14} T_{14} = C_{13} \quad (\text{C31})$$

where,

$$U_{13,13} = H_{13,7} + H_{13,10} + H_{13,14}$$

$$U_{13,7} = -H_{13,7}$$

$$U_{13,10} = -H_{13,10}$$

$$U_{13,14} = -H_{13,14}$$

$$H_{13,7} = \frac{\theta_2 k_{mat} l_{com}}{3 \ln\left(\frac{r_{ro}}{r_{roi}}\right)}$$

$$H_{13,10} = \frac{(2\pi - \theta_2) k_{mat} l_{com}}{3 \ln\left(\frac{r_{ro}}{r_{roi}}\right)}$$

$$H_{13,14} = \frac{k_{mat} \pi r_{roi}^2}{\frac{l_{ecc}}{3}}$$

With the 19 discrete elements, a total of 19 simultaneous linear equations can be derived with the following form:

$$\sum_j^n U_{i,j} T_j = \dot{C}_i \quad (C32)$$

By solving the equations in thermal resistance model and lumped thermal conductance model together, the instantaneous heat transfer rates into suction chamber and compression chamber can be obtained.

Appendix D Algorithm Flow Chart of Simulation

The mathematical model presented in Chapters 4 and 5 are incorporated and simulated concurrently with ForTran programming language. The algorithm flow chart of the simulation is presented Figure D.1.

Figure D.1 Algorithm flow chart of simulation

Appendix E Uncertainties Analysis

Table E.1 summarises the uncertainties of the measuring devices, which are used for the uncertainties analysis.

Table E.1 Uncertainties of measuring devices

Uncertainties of Measuring Devices	
Operating speed, $\Delta\omega$	± 5.0 rev/min
Futek torque meter, ΔT	± 0.25 Nm
Brooks Instrument rotameter ΔQ	± 2 L/min

The uncertainty of the power input is calculated based on the following approximation:

$$\begin{aligned}
 \dot{W} &= T\omega \\
 \Delta\dot{W} &\approx \sqrt{\left(\frac{\partial\dot{W}}{\partial T}\Delta T\right)^2 + \left(\frac{\partial\dot{W}}{\partial\omega}\Delta\omega\right)^2} \\
 &= \sqrt{(\omega\Delta T)^2 + (T\Delta\omega)^2}
 \end{aligned} \tag{E1}$$

Table E.2 Calculated uncertainties of power input

Operating Speed (rev/min)	Discharge Pressure (bar)	Predicted Power Input (W)	Measured Power Input (W)	Discrepancy (%)	Uncertainties $\Delta\dot{W}$ (%)
1217	4.263	230.45	219.92	4.78	8.364
1214	3.749	210.71	201.06	4.80	9.126
1217	3.411	198.78	193.63	2.66	9.500
1215	3.004	181.73	176.90	2.73	10.382
1217	2.398	155.60	150.01	3.73	12.262
1427	5.241	303.82	312.41	-2.75	6.904
1411	4.822	290.40	277.24	4.75	7.693
1412	4.236	267.96	252.58	6.09	8.450
1415	3.662	242.86	233.85	3.85	9.146
1412	2.922	206.71	203.92	1.37	10.466
1423	2.670	195.71	199.08	-1.69	10.804
1638	5.268	363.10	350.36	3.64	7.067
1638	4.549	333.29	318.37	4.69	7.777
1639	3.685	289.96	289.30	0.23	8.563
1639	3.264	266.24	271.85	-2.06	9.113

1639	3.068	255.48	257.37	-0.74	9.626
1640	2.829	240.56	242.54	-0.82	10.220
1837	5.065	401.14	414.98	-3.33	6.691
1837	4.582	376.90	377.60	-0.19	7.353
1839	4.028	349.40	348.34	0.30	7.980
1839	3.653	325.64	344.12	-5.37	8.078
1839	3.073	292.14	310.51	-5.92	8.952
1840	2.640	259.47	272.32	-4.72	10.213
2107	5.140	475.80	496.99	-4.26	6.408
2112	4.435	436.91	455.15	-4.01	7.014
2114	4.004	410.89	433.13	1.65	7.377
2111	3.691	386.55	404.22	6.77	7.894
2112	3.342	361.58	411.39	5.36	7.760
2113	2.547	299.79	362.03	2.82	8.822

The uncertainty of the volumetric efficiency is calculated based on the following approximation:

$$\begin{aligned}
\eta_v &= \frac{\rho_m Q_m}{\rho_{suc} V_{cham} \frac{\omega}{60}} \\
\Delta \eta_v &\approx \sqrt{\left(\frac{\partial \eta_v}{\partial Q} \Delta Q \right)^2 + \left(\frac{\partial \eta_v}{\partial \omega} \Delta \omega \right)^2} \\
&= \sqrt{\left(\frac{\rho_m}{\rho_{suc} V_{cham} \frac{\omega}{60}} \Delta Q \right)^2 + \left(-\frac{\rho_m Q_m}{\rho_{suc} V_{cham} \frac{\omega^2}{60}} \Delta \omega \right)^2}
\end{aligned} \tag{E2}$$

Table E.3 Calculated uncertainties of volumetric efficiency

Operating Speed (rev/min)	Discharge Pressure (bar)	Predicted Ideal Mass Flow Rate (g/s)	Measured Mass Flow Rate (g/s)	Volumetric Efficiency (%)	Uncertainties $\Delta \eta_v$ (%)
1217	4.263	0.748	0.325	43.39	12.24
1214	3.749	0.748	0.357	47.72	10.78
1217	3.411	0.750	0.453	60.39	9.76
1215	3.004	0.749	0.455	60.78	8.60
1217	2.398	0.752	0.454	60.31	6.84
1427	5.241	0.876	0.499	57.04	12.85
1411	4.822	0.867	0.515	59.39	12.17

1412	4.236	0.868	0.482	55.58	10.45
1415	3.662	0.869	0.553	63.62	8.98
1412	2.922	0.870	0.551	63.31	7.17
1423	2.670	0.878	0.601	68.42	6.46
1638	5.268	1.004	0.551	54.86	11.23
1638	4.549	1.004	0.603	60.04	9.66
1639	3.685	1.008	0.697	69.12	7.80
1639	3.264	1.008	0.736	72.97	6.87
1639	3.068	1.009	0.803	79.60	6.43
1640	2.829	1.010	0.843	83.50	5.91
1837	5.065	1.121	0.805	71.85	9.49
1837	4.582	1.122	0.858	76.50	8.60
1839	4.028	1.125	0.905	80.46	7.55
1839	3.653	1.126	1.020	90.66	6.81
1839	3.073	1.127	1.020	90.52	5.67
1840	2.640	1.128	1.009	89.49	4.82
2107	5.140	1.282	0.869	67.77	8.47
2112	4.435	1.286	0.948	73.77	7.23
2114	4.004	1.289	1.043	80.90	6.89
2111	3.691	1.287	1.163	90.35	6.53
2112	3.342	1.291	1.166	90.32	6.28
2113	2.547	1.293	1.193	92.33	6.00

Appendix F Specifications of Experimental Devices

F.1 Motor Specifications

Figure F.1 Technical specification of ABB motor (model: 3GAA 131 213-BSE)

F.2 Technical Specifications of Measuring Devices

Figure F.2 Technical specification of Futek torque meter (model: TRS605)

Pressure transmitter for general industrial applications Model A-10

WIKA data sheet PE 81.60

Applications

- Machine building
- Machine tools
- Measurement and control technology
- Hydraulics and pneumatics
- Pumps and compressors

Special features

- Measuring ranges from 0 ... 1 to 0 ... 600 bar
- Non-linearity 0.25 % or 0.5 %
- Output 4 ... 20 mA, DC 0 ... 10 V, DC 0 ... 5 V and others
- Electrical connection: Angular connector form A and C, circular connector M12 x 1, cable outlet 2 m
- Process connection G 1/4 DIN 3852-E, 1/4 NPT and others

Pressure transmitter model A-10

Description

The model A-10 pressure transmitter for general industrial applications is not only notable for its compact design, but it also offers excellent quality at an extremely competitive price.

The user can choose between a non-linearity of 0.25 % and 0.5 %. A free test certificate provides information on the measuring points recorded during manufacture.

The model A-10 is set up for worldwide use through the international cULus and GOST certification. The various pressure units and process connections required for particular operating conditions are available at short notice.

Figure F.3 Technical specifications of WIKA pressure transducer (model: A-10):
page 1/3

Measuring ranges

Relative pressure									
bar	Measuring range	0 ... 1	0 ... 1.6	0 ... 2.5	0 ... 4	0 ... 6	0 ... 10 ¹⁾	0 ... 16 ¹⁾	0 ... 25 ¹⁾
	Overpressure limit	2	3.2	5	8	12	20	32	50
	Measuring range	0 ... 40	0 ... 60	0 ... 100	0 ... 160	0 ... 250	0 ... 400	0 ... 600	
psi	Overpressure limit	80	120	200	320	500	800	1,200	
	Measuring range	0 ... 15	0 ... 25	0 ... 30	0 ... 50	0 ... 100	0 ... 160 ¹⁾	0 ... 200 ¹⁾	0 ... 300 ¹⁾
	Overpressure limit	30	60	60	100	200	290	400	600
	Measuring range	0 ... 500	0 ... 1,000	0 ... 1,500	0 ... 2,000	0 ... 3,000	0 ... 5,000	0 ... 10,000	
	Overpressure limit	1,000	1,740	2,900	4,000	6,000	10,000	17,400	
Absolute pressure									
bar	Measuring range	0 ... 1	0 ... 1.6	0 ... 2.5	0 ... 4	0 ... 6	0 ... 10	0 ... 16	0 ... 25
	Overpressure limit	2	3.2	5	8	12	20	32	50
	Measuring range	0 ... 15	0 ... 25	0 ... 30	0 ... 50	0 ... 100	0 ... 150	0 ... 200	0 ... 300
psi	Overpressure limit	30	60	60	100	200	290	400	600
Vacuum and +/- measuring range									
bar	Measuring range	-1 ... 0	-1 ... +0.6	-1 ... +1.5	-1 ... +3	-1 ... +5			
	Overpressure limit	2	3.2	5	8	12			
	Measuring range	-1 ... +9	-1 ... +15	-1 ... +24					
psi	Overpressure limit	20	32	50					
	Measuring range	-30 inHG ... 0	-30 inHG ... +15	-30 inHG ... +30	-30 inHG ... +60	-30 inHG ... +100			
	Overpressure limit	30	60	60	150	250			
	Measuring range	-30 inHG ... +160	-30 inHG ... +200	-30 inHG ... +300					
	Overpressure limit	350	450	600					

¹⁾ If the medium water is measured, a higher overpressure limit is recommended.

The given measuring ranges are also available in kg/cm², MPa and kPa.

Other measuring ranges available on request

Vacuum resistance

Yes

Output signal

Signal type	Signal
Current (2-wire)	4 ... 20 mA
Voltage (3-wire)	DC 0 ... 10 V
	DC 0 ... 5 V
	DC 1 ... 5 V
	DC 0.5 ... 4.5 V
Ratiometric (3-wire)	DC 0.5 ... 4.5 V

Other output signals available on request

Depending on the signal type the following loads apply:

Signal type	Load in Ω
Current (2-wire)	$\leq (\text{power supply} - 8 \text{ V}) / 0.02 \text{ A}$
Voltage (3-wire)	$> \text{maximum output signal} / 1 \text{ mA}$
Ratiometric (3-wire)	$> 4.5 \text{ k}$

Figure F.4 Technical specifications of WIKA pressure transducer (model: A-10):
page 2/3

Voltage supply

Power supply

Output signal	Power supply	
	Standard	Option
4 ... 20 mA	DC 8 ... 30 V	DC 8 ... 35 V ¹⁾
DC 0 ... 10 V	DC 14 ... 30 V	DC 14 ... 35 V
DC 0 ... 5 V	DC 8 ... 30 V	DC 8 ... 35 V
DC 1 ... 5 V	DC 8 ... 30 V	DC 8 ... 35 V
DC 0.5 ... 4.5 V	DC 8 ... 30 V	DC 8 ... 35 V
DC 0.5 ... 4.5 V ratiometric	DC 5 V ± 10 %	-

¹⁾ Not possible with non-linearity 0.25 % BFSL

The power supply for the pressure transmitter must be made via an energy-limited electrical circuit in accordance with section 9.3 of UL/EN/IEC 61010-1, or an LPS to UL/EN/IEC 60950-1, or class 2 in accordance with UL1310/UL1585 (NEC or CEC). The power supply must be suitable for operation above 2,000 m should the pressure transmitter be used at this altitude.

Total current consumption

Signal type	Total current consumption
Current (2-wire)	Signal current, max. 25 mA
Voltage (3-wire)	8 mA
Ratiometric (3-wire)	8 mA

Accuracy

Optionally the model A-10 is available with an improved non-linearity. Depending on the selected non-linearity the following values result:

	Standard	Option
Non-linearity per BFSL (IEC 61298-2)	≤ ±0.5 % of span	≤ ±0.25 % of span
Measuring deviation of the zero signal	Typical: ≤ ±0.5% of span Maximum: ≤ ±0.8 % of span	Typical: ≤ ±0.15% of span Maximum: ≤ ±0.4 % of span
Accuracy at room temperature ¹⁾	≤ ±1 % of span	≤ ±0.5 % of span ≤ ±0.6 % of span (at DC 0 ... 5 V)

¹⁾ Including non-linearity, hysteresis, zero offset and end value deviation (corresponds to measured error per IEC 61298-2), calibrated in vertical mounting position with process connection facing downwards

Non-repeatability

≤ 0.1 % of span

Signal noise

≤ ±0.3 % of span

Temperature error at 0 ... 80 °C

- Typical: 1 % of span
- Maximum: 2.5 % of span

Long-term drift

≤ ±0.1 % of span

Figure F.5 Technical specifications of WIKA pressure transducer (model: A-10):
page 3/3

Model 8530B
Piezoresistive pressure transducer

Features

- 200 to 1000 psia
- Absolute reference

Description

The Endevco® model 8530B is a miniature, high sensitivity piezoresistive pressure transducer for measuring absolute pressure. The volume behind the diaphragm is evacuated and glass sealed to provide an absolute pressure reference. Full scale output is 300 mV with high overload capability and high frequency response. It is available in ranges from 200 psia to 1000 psia. See model 8530C for lower pressure ranges.

Endevco pressure transducers feature an active four-arm strain gage bridge diffused into a sculptured silicon diaphragm for maximum sensitivity and wideband frequency response. Self-contained hybrid temperature compensation provides stable performance over the temperature range of 0°F to 200°F (-18°C to +93°C). Endevco transducers also feature excellent linearity and provide stable performance over the temperature range of 0°F to 200°F (-18°C to +93°C). Endevco transducers also feature excellent linearity (even to 3X range), high shock resistance, and high stability during temperature transients.

The model 8530B is available with metric M5X0.8 mounting threads as 8530B-XXM5 on special orders. Also available with integral four-pin connector as model 8530B-XXM37. The standard cable length is 30 inches. Alternate lengths are available in quantity by using part number 8530B-XXX-ZZZ where "XXX" is the pressure range and "ZZZ" is the cable length in inches

Endevco model 136 three-channel system, model 4428A or 4430A signal conditioner, or OASIS 2000 computer-controlled system are recommended as signal conditioner and power supply.

Figure F.6 Technical specifications of Endevco pressure transducer (model: 8530B-500); page 1/2

Model 8530B Piezoresistive pressure transducer

Endevco

Specifications

	Units	-200	-500	-1000
Range	psia	0-200	0-500	0-1000
Sensitivity [1]	mV/psi	1.5 ±0.5	0.6 ±0.2	0.3 ±0.1
Combined: non-linearity, non repeatability, pressure hysteresis [2]	% FSO RSS max	0.50	0.50	1.0
Non-linearity, independent	% FSO typ	0.2	0.2	0.2
Non-repeatability	% FSO typ	0.1	0.1	0.1
Pressure hysteresis	% FSO typ	0.1	0.1	0.1
Zero measurand output [3]	mV max	±10	±10	±10
Zero shift after 3X over range	±% 3X FSO max (typ)	0.2 (0.05)	0.2 (0.05)	
after 2.5X over range	±% 2.5X FSO max (typ)			0.2 (0.1)
Thermal zero shift				
From 0 to 200°F (-18°C to +93°C)	±% FSO max	3	3	3
Thermal sensitivity shift				
From 0 to 200°F (-18°C to +93°C)	±% max	4	4	4
Resonant frequency	Hz	750 000	1 000 000	> 1 000 000
Non-linearity at 3X over range	% 3X FSO	1.5	2.0	2.0 @ % 2.5X FSO
Thermal transient response per	psi / °F	0.02	0.02	0.04
ISA-S37.10, PARA. 6.7, procedure I	psi / °C	0.04	0.04	0.07
Photoflash response [4]	equiv. psi	5	10	20
Warm-up time [5]	ms	1	1	1
Acceleration sensitivity	equiv. psi/g	0.0003	0.0002	0.0002
Burst pressure (diaphragm)	psia min	800	2000	4000
Case pressure [6]	psia min	1000	5000	5000

Electrical

Full scale output	300 ±100 mV at 10.0 Vdc
Supply voltage [7]	10.0 Vdc recommended, 18 Vdc maximum
Electrical configuration	Active four-arm piezoresistive bridge
Polarity	Positive output for increasing pressure
Resistance	
Input	2000 ±800 ohms
Output	1600 ±500 ohms
Isolation	100 megohms minimum at 50 Volts, leads to case, leads to shield, shield to case
Noise	5 microvolts rms typical, DC to 50 000 Hz; 50 microvolts rms maximum, DC to 50 000 Hz

Mechanical

Case, material	Stainless steel (17-4 PH CRES)
Cable, integral	Four conductor No. 32 AWG Teflon insulated leads, braided shield, silicone jacket
Dead volume port (+)	0.0003 cubic inches (0.005 cc)
Mounting/torque	10-32 UNF-2A threaded case 0.438 inch (11.12 mm) long / 15 ±5 lbf-in (1.7 ±0.6 Nm)
Weight	2.3 grams (cable weighs 9 grams/meter)

Environmental

Media [8] [9]	Clean dry gas. Contact factory for protective modifications for other media.
Temperature [10]	-65°F to +250°F (-54°C to +121°C)
Vibration	1000 g pk
Acceleration	1000 g
Shock	20 000 g, 100 microsecond haversine pulse
Humidity	Isolation resistance greater than 100 megohms at 50 V when tested per MIL-STD-202E, method 103B, test condition B

Calibration data [11]

Data supplied for all parameters in Certified Performance section. Optional calibrations available for all parameters in Typical Performance section

Included accessories

EHR93 O-ring, Viton

Optional accessories

EHR96 O-ring, Fluorosilicone
2432B-3 4 conductor shielded cable, white

Notes:

- 1 psi = 6.895 kPa = 0.069 bar.
- FSO (Full Scale Output) is defined as transducer output change from 0 to + full scale pressure.
- Zero Measurand Output (ZMO) is the transducer output with 0 psia applied.
- Per ISA-S37.10, Para. 6.7, Proc. II.
- Warm-up time is defined as elapsed time from excitation voltage "turn on" until the transducer output is within ±1% of reading accuracy.
- Case pressure identifies media containment pressure in the event of diaphragm rupture.

7. Use of excitation voltages other than 10.0 Vdc requires manufacture and calibration at that voltage since thermal errors increase with high excitation voltages.
8. Internal seals are epoxy and are compatible with clean dry gas media. Media in measurand port is exposed to CRES, Parylene C, epoxy and the Viton O-ring. Not suitable for use with high pH or low pH liquids, long term exposure to water, or exposure to solvents which may attack epoxies.
9. O-ring, Parker 5-125, compound V747-75 (Viton®) is supplied unless otherwise specified on purchase order. Fluorosilicone O-ring, for leak tight operation below 0°F is available on special order.
10. Units can be compensated over any 200°F (93°C) span from -65°F to +250°F (-54°C to +121°C) on special order.
11. Maintain high levels of precision and accuracy using Endevco's factory calibration services. Call Endevco's inside sales force at 800-982-6732 for recommended intervals, pricing and turn around time for these services as well as for quotations on our standard products.

Continued product improvement necessitates that Endevco reserve the right to modify these specifications without notice. Endevco maintains a program of constant surveillance over all products to ensure a high level of reliability. This program includes attention to reliability factors during product design, the support of stringent Quality Control requirements, and compulsory corrective action procedures. These measures, together with conservative specifications, have made the name Endevco synonymous with reliability.

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smart engineering for
extreme environments

Figure F.7 Technical specifications of Endevco pressure transducer (model: 8530B-500); page 2/2

3000 Series

Variable Area

Molded, Polycarbonate Purgemeters

Description

The Brooks® 3000 Series is molded of high-impact resistant polycarbonate. These purgemeters are available with 24 different air and water ranges and are supplied with scales in LPM Air, SCFH Air, CCM Water and GPH Water, all with 10:1 turndown ratios. The 3000 has been designed to maintain maximum pressures to 100 psig and temperatures to 150°F. These meters are particularly suited for purge applications. These economically engineered units have been designed to provide the highest quality with precision accuracy. These flowmeters are fitted with 1/8" FNPT inlet and outlet connections. An optional inlet or outlet control valve can be specified.

Product Features

- Flow tube and body molded of high-impact polycarbonate
- High quality construction
- Economy combined with accuracy
- Monitor or control air and water flows
- Flow ranges: Air 50 CCM to 100 LPM; Water 4 CCM to 40 GPH
- Supplied with easy-to-read 10:1 turn direct reading scales
- Ideally suited for OEM panel-mount applications
- Standard black body, custom colors available upon request

Product Specifications

Specifications	3000
Accuracy	± 4% Full Scale
Temperature	150°F/65°C Maximum
Pressure	100 PSI / 6.9 Bar Maximum
Materials of Construction	
Body & Tube	Polycarbonate
Floats	Black Glass, Carbide, Stainless Steel or Aluminum
Fittings	Brass or 303 Stainless Steel
Seal Material	Buna-N or Viton® Fluoroelastomer
Valve (Optional)	Cartridge Type: Brass or 303 Stainless Steel

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Figure F.8 Technical specifications of Brooks MR3000 series variable area flow meter (model: MR-3A17-BLBN): page 1/2

3000 Series Flow Ranges

SCFH of Air	Tube Code	LPM of Air	Tube Code
.1-1	3A00	0.5-5	3A12
.2-2.5	3A01	.1-1.2	3A13**
.4-5	3A02	4-5	3A14**
1-11	3A03	.2-2.5	3A23
2-22	3A04	1-10	3A15**
4-60	3A06	2-30	3A16**
10-110	3A07	4-50	3A17**
20-200	3A08	10-100	3A18**
CCM of Water	Tube Code	GPH of Water	Tube Code
5-110	3L10	.2-2.5	3L28
20-300	3L11**	4-5	3L19
		1-10	3L20
		2-25	3L21
		4-40	3L22

** Available for **QuickShip**.

Model Code

Code Description	Code Option	Option Description
I. Base Model Number	MR	Body/Tube Molded Polycarbonate
II. Tube Code	XXXX	From Table
III. Fittings	B	Brass
	S	Stainless Steel
IV. Valve	N	None
	V	Valve
	O	Outlet Valve (Vacuum Service Only)
V. Seals	BN	Buna-N
	VT	Viton
VI. Options	-NL	No Logo

Sample Standard Model Code	I	II	III	IV	V	VI
	MR	3A07	B	V	BN	-NL

Product Dimensions

Additional Options

- Certificate of Conformance
- ICC
- Paper Tag
- Stainless Steel Tag

Brooks Service and Support

Brooks is committed to assuring all of our customers receive the ideal flow solution for their application, along with outstanding service and support to back it up. We operate first class repair facilities located around the world to provide rapid response and support. Each location utilizes primary standard calibration equipment to ensure accuracy and reliability for repairs and recalibration and is certified by our local Weights and Measures Authorities and traceable to the relevant International Standards.

Visit www.BrooksInstrument.com to locate the service location nearest to you.

START-UP SERVICE AND IN-SITU CALIBRATION

Brooks Instrument can provide start-up service prior to operation when required. For some process applications, where ISO-9001 Quality Certification is important, it is mandatory to verify and/or (re)calibrate the products periodically. In many cases this service can be provided under in-situ conditions, and the results will be traceable to the relevant international quality standards.

CUSTOMER SEMINARS AND TRAINING

Brooks Instrument can provide customer seminars and dedicated training to engineers, end users, and maintenance persons.

Please contact your nearest sales representative for more details.

Due to Brooks Instrument's commitment to continuous improvement of our products, all specifications are subject to change without notice.

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Figure F.9 Technical specifications of Brooks MR3000 series variable area flow meter (model: MR-3A17-BLBN): page 2/2

F.3 Specification of TEF-MET/P bushing

🏠 > Products > TEF-MET®/P

TEF-MET®/P

Wrapped composite dry sliding bearing steel / special PTFE, lead-free, Maintenance-free

TECHNICAL DETAILS TEF-MET®/P

- Specific load capacity static to ≤ 250 [N/mm²]
- Specific load capacity dynamic to ≤ 140 [N/mm²] at slow sliding speed
- Sliding speed ≤ 2.0 [m/s] (≤ 5 [m/s] oil lubricated) [m/s]
- Friction value 0.02 [μ] to 0.20 [μ] (≤ 0.05 [μ] oil lubricated)
- Temperature strain -195 [°C] to 280 [°C]
- Max. PV-value 3.8 [N/mm² x m/s] (≤ 60 [N/mm² x m/s] oil lubricated)

MATERIALS

Support material Steel tin-plated

Intermediate layer Sintered bronze

Sliding layer PTFE

TOLERANCE DETAILS

Housing $\varnothing$ H7

Bushing inner- $\varnothing$ after mounting H9

Shaft tolerance f7 to h8

SHAFT MATERIAL

Steel, hardened or untempered, surface roughness $\leq R_z 2-3$

Figure F.10 Technical specification of TEF-MET/P bushing (Page 1/2)

PROPERTIES

Wrapped composite sliding bearing, steel

Special PTFE coating

Maintenance-free dry sliding bearing

DIN 1494 / ISO 3547

Also available: lead free!

All special designs are produced at short notice!

Cylindr. and flanged bushings, thrust washer, strips are stand. fabrication

Punchings and moldings are produced on customer requirement

RANGE OF APPLICATION

Machine and plant construction, textile processing and agricultural machines, hydraulic pumps, automotive industry

MAINTENANCE

TEF-MET®/P is a dry sliding bearing with a PTFE sliding layer. Lubrication is therefore not necessary. But it is suitable for lubricated applications (such as hydraulic oil lubrication).

MOUNTING ADVISE

Housing: Mounting bevel min. 1.5 mm x 15-45°

Shaft: Mounting bevel 5 mm x 15°, edges rounded

The application of an adequate force fitting mandrel is advisable. Grease lubrication of the outer surface may be necessary when mounting.

Stick-in: Adhesive should not contact the sliding surface

Figure F.11 Technical specification of TEF-MET/P bushing (Page 2/2)

Appendix G Calibration Data of Measuring Devices

G.1 Torque Meter

Figure G.1 Calibration data for Futek torque meter (model: TRS 605)

G.2 Thermocouple

Figure G.2 Calibration data for thermocouple (TCT-01)

Figure G.3 Calibration data for thermocouple (TCT-02)

Figure G.4 Calibration data for thermocouple (TCT-04)

Figure G.5 Calibration data for thermocouple (TCT-05)

Figure G.6 Calibration data for thermocouple (TCT-08)

G.3 Pressure Transducer

Figure G.7 Calibration data for Endevco pressure transducer (MPT-01 (29408))

Figure G.8 Calibration data for Endevco pressure transducer (MPT-02 (29407))

Figure G.9 Calibration data for Endevco pressure transducer (MPT-03 (29410))

Figure G.10 Calibration data for WIKA pressure transducer (PT-02)

Figure G.11 Calibration data for WIKA pressure transducer (PT-03)

Figure G.12 Calibration data for WIKA pressure transducer (PT-04)

Figure G.13 Calibration data for WIKA pressure transducer (PT-05)

Figure G.14 Calibration data for WIKA pressure transducer (PT-06)

Figure G.15 Calibration data for WIKA pressure transducer (PT-07)

Appendix H Specification of Lubricant

Technical Data Sheet

Previous Name: Shell Clavus Oil R

Shell Refrigeration Oil S4 FR-F 68

*Reliable Performance
For use with HFC Refrigerants*

Advanced Synthetic Refrigerator Compressor Lubricant

Shell Refrigeration Oil S4 FR-F is a synthetic refrigeration lubricant with a polyol ester base fluid. It has been developed for use with R134a and other HFC refrigerants.

DESIGNED TO MEET CHALLENGES

Performance, Features & Benefits

- Extended maintenance intervals**
Shell Refrigeration Oil S4 FR-F has excellent high temperature and oxidation stability providing long service life even where high compressor discharge temperatures are found.
- Wear protection**
Shell Refrigeration Oil S4 FR-F has been designed to minimise the wear in bearings and pistons – the most critical parts of a reciprocating compressor.

Main Applications

- Refrigerator Compressors**
Shell Refrigeration Oil S4 FR-F is recommended for use in open, semi-open and hermetic compressors in domestic, commercial and industrial refrigeration systems. It can be used in both rotary and reciprocating compressor types.
- Refrigerant Compatibility**
Shell Refrigeration Oil S4 FR-F is recommended for use with R134a and other types of HFC refrigerant.
- Seal Compatibility**
Shell Refrigeration Oil S4 FR-F is compatible with all sealing materials commonly used with HFC refrigerant systems.

Specifications, Approvals & Recommendations

- Shell Refrigeration Oil S4 FR-F meets the requirements of DIN 51503 KD.

For a full listing of equipment approvals and recommendations, please consult your local Shell Technical Helpdesk.

Typical Physical Characteristics

Properties	Method	Refrigeration Oil S4 FR-F 68
ISO Viscosity Grade	ISO 3448	68
Refrigerator Oil Group	DIN 51503	KD
Kinematic Viscosity @40°C	mm ² /s	66
Kinematic Viscosity @100°C	mm ² /s	8.8
Density @15°C	kg/m ³	991
Flash Point (COC)	°C	>230
Pour Point	°C	-42
Neutralisation Number	mg KOH/g	<0.06
Characteristics when used with R134a - Flocc-point	°C	<-30
Characteristics when used with R134a - Refrigerant Stability	@250°C	>96
Miscibility		Miscible over the whole range of typical refrigeration temperatures

These characteristics are typical of current production. Whilst future production will conform to Shell's specification, variations in these characteristics may occur.

Figure H.1 Technical data sheet of lubricant

The viscosity – temperature chart is provided by the supplier, who has confirmed that the relation is appropriate to be applied for the lubricant used in this project.

Figure H.2 Viscosity – temperature chart of the lubricant

Appendix I Journal Bearing Model

From the dynamics model of the U-Vane compressor presented in Section 5.1, the forces acting on each of the major compressor components are introduced and the instantaneous load on the compressor can be obtained. In general, this load is dynamical and is supported by a journal bearing, which is a hollow cylinder wrapping a rotating component (known as journal). Lubricant is filled within the space between the two components to prevent the components from contacting each other. As the load on compressor is varying in both direction and magnitude during the operation, dynamically loaded journal bearing model is implemented for U-Vane compressor mathematical model.

Figure I.1 Schematic diagram of a journal bearing

Figure I.1 shows the schematic diagram of a journal bearing. When the journal is loaded, its centre will shift away from the stationary bearing centre. Meanwhile, the rotation of the journal causes the lubricating oil wedge to be formed. Subsequently, the fluid pressure is established to provide support to the journal such that the distance (ϵ) is kept within a desired distance. Also, a minimum lubricating oil flow rate is required in order to ensure an adequately developed oil film pressure. Hence, it is important to have an appropriate design

of lubrication oil distribution system to make sure the lubricant is supplied at the required flow rate.

As mentioned earlier, the journal centre moves when loaded, and varies if the magnitude or direction of the load (F) change. This causes the magnitude of the lubricant film pressure and the minimum oil film thickness ($h_{\min}$) to fluctuate. As a result, the fatigue strength of the journal bearing material is a notable concern as it is exposed to fluctuating stress. In addition, the minimum oil film thickness determines the surface roughness of the journal and bearing surfaces. Hence, it is necessary to aware of the variations of the maximum film pressure and minimum oil film thickness for a reliable journal bearing operation.

By applying Reynold differential equation, the lubricant film pressure distribution in a journal bearing can be determined, with the assumptions of negligible gravity effect, Newtonian fluid, laminar, inertia-less and thin film flow [252], which is written as:

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\mu} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h^3}{12\mu} \frac{\partial P}{\partial y} \right) = \frac{1}{2} \frac{\partial}{\partial x} \left[\rho (U_b + U_j) h \right] - \rho U_j \frac{\partial h}{\partial x} + \rho (U_j - U_b) \quad (I1)$$

In Equation (I1), the terms on the left-hand side (LHS) represent the flow driven by the pressure gradients, as known as Poiseuille flow. The first term on the right-hand side (RHS) is the shear-driven flow, which is also known as Couette flow. The second term represents the flow caused by the wedge action. The flow due to the squeezing action where the journal and bearing surfaces move perpendicularly to each other is represented by the last term on the RHS.

Considers that rigid material is always used for the bearing so that no physical stretch is encountered in the bearing. Therefore, expanding the Couette flow term, neglecting the density wedge and the physical stretch in the bearing material, Equation (I1) can be simplified as below:

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\mu} \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h^3}{12\mu} \frac{\partial P}{\partial y} \right) = \frac{1}{2} \rho (U_b - U_j) \frac{\partial h}{\partial x} + \rho (U_j - U_b) \quad (I2)$$

Normally, $\frac{\partial h}{\partial x}$ is negative as fluid flow through converging gap is required to produce positive pressure force to support the load. This makes the first term on the RHS of Equation (I2) the most dominant term in generating the pressure to support the load in the journal bearing. In Equation (I2), the coordinate system that relative to one fixed point in the space is commonly used when there is only one rotating component. However, for the journal bearing of UVC, both journal and load are rotating. In this case, it would be convenient to make the coordinate system relative to the line of centres and hence Equation (I2) can be rewritten as [177]:

$$\begin{aligned} \frac{\partial}{\partial \theta} \left[(1 + \varepsilon \cos \theta)^3 \frac{\partial P}{\partial \theta} \right] + R_b^2 \frac{\partial}{\partial y} \left[(1 + \varepsilon \cos \theta)^3 \frac{\partial P}{\partial y} \right] \\ = 12\mu \left(\frac{R_b}{\delta_b} \right)^2 \left[\dot{\varepsilon} \cos \theta + \varepsilon (\dot{\phi} - \omega_{av}) \sin \theta \right] \end{aligned} \quad (I3)$$

In literature, two special conditions are considered to obtain closed form solutions for the pressure distribution in Equation (I3), which are the ‘infinitely short’ bearing approximation (Orvirk’s solution, P_o), which the pressure variation in the circumferential direction becomes relatively small compare to the pressure variation in the axial direction; and the ‘infinitely long’ bearing approximation (Sommerfeld solution, P_s) that neglects the pressure variation in the axial direction [253]. The expressions of the pressure distributions of both the approximations are shown in Equations (I4) and (I5).

$$P_o = \frac{-6\mu \left(\frac{L_b}{\delta_b} \right)^2 \left[\frac{1}{4} - \left(\frac{y}{L_b} \right)^2 \right] \left[\dot{\varepsilon} \cos \theta + \varepsilon (\dot{\phi} - \omega_{av}) \sin \theta \right]}{(1 + \varepsilon \cos \theta)^3} \quad (I4)$$

$$P_s = -6\mu \left(\frac{R_b}{\delta_b} \right)^2 \left[\dot{\varepsilon} \cos \theta + \left(\frac{2}{2 + \varepsilon^2} \right) \varepsilon (\dot{\phi} - \omega_{av}) \sin \theta \right] \frac{2 + \varepsilon \cos \theta}{(1 + \varepsilon \cos \theta)^2} \quad (I5)$$

However, the accuracy of the solutions proposed above is limited only to particular conditions. For infinitely short bearing assumption, the length to diameter ratio of the journal bearing, namely slenderness ratio (L_b/D_b) is required

to be less than 0.25 while the infinitely long bearing assumption requires a ratio of more than 2.0 to obtain an accurate prediction, respectively. These solutions have a very narrow range of application as the slenderness ratio of most of the practicable journal bearing design falls between 0.25 to 1.5 [176]. To make the solution more applicable for the journal bearing having finite length in reality, the solutions of infinitely short and infinitely long approximations were combined harmonically by Reason and Narang [180], as shown by the following relationship:

$$\frac{1}{P_{RN}} = \frac{1}{P_O} + \frac{1}{P_S} \quad (I6)$$

The combined solution is further simplified by Hirani *et al.* [178] and a pressure distribution was obtained as expressed below:

$$P(\theta, y) = \frac{-6\mu \left(\frac{L_b}{\delta_b}\right)^2 \left[\frac{1}{4} - \left(\frac{y}{L_b}\right)^2 \right] \left[\dot{\varepsilon} \cos \theta + \varepsilon (\dot{\phi} - \omega_{av}) \sin \theta \right]}{(1 + \varepsilon \cos \theta)^3 \left[1 + \frac{\frac{1}{4} - \left(\frac{y}{L_b}\right)^2}{B_1} \right]} \quad (I7)$$

where

$$B_1 = \frac{(1 + \varepsilon \cos \theta)(2 + \varepsilon \cos \theta)}{(L_b/\delta_b)^2} \quad (I8)$$

L_b is the length of the journal bearing (m)

R_b is the radius of the journal (m)

$\varepsilon = \frac{\epsilon}{\delta_b}$ is the eccentricity ratio (-)

$\dot{\varepsilon}$ is the rate of change of eccentricity ratio with respect to time (s^{-1})

ϕ is the attitude angle (rad)

$\dot{\phi}$ is the rate of change of attitude angle with respect to time (rad/s)

$\omega_{av} = \frac{1}{2}(\omega_j + \omega_b) - \omega_L$ is the average angular velocity of the journal and the bearing sleeve relative to the load (rad/s)

μ is the absolute viscosity of the lubricant (Pa•s)

δ_b is the radial clearance between journal (shaft) and bearing (m)

Hence, the pressure force produced by the oil film to support the load can be found by integrating the pressure distribution (Equation (I7)) over the surface area. By resolving the forces into the direction of the line of centre, and the direction perpendicular to it, the following force equations can be obtained:

$$F \cos \phi = - \int_{\theta_a - L_b/2}^{\theta_b} \int_{-L_b/2}^{L_b/2} P \cos \theta \, dz \, R \, d\theta \quad (I9)$$

$$F \sin \phi = \int_{\theta_a - L_b/2}^{\theta_b} \int_{-L_b/2}^{L_b/2} P \sin \theta \, dz \, R \, d\theta \quad (I10)$$

From which, θ_a and θ_b are the angles where the positive portion of the pressure lies in between (refer to Figure I.1), and they have the following expressions:

$$\theta_a = \tan^{-1} \left(\frac{\dot{\varepsilon}}{\varepsilon(\omega_{av} - \dot{\phi})} \right); \quad (I11)$$

$$\theta_b = \theta_a + K\pi, \quad K = \begin{cases} 1 & \text{for } \varepsilon(\omega_{av} - \dot{\phi}) > 0 \\ -1 & \text{for } \varepsilon(\omega_{av} - \dot{\phi}) < 0 \end{cases}$$

In Figure I.1, it is noted that two important parameters are introduced in the equations, which are the eccentricity ratio (ε) and the attitude angle (ϕ). The eccentricity ratio is the ratio of the distance between the centres of rotating journal (shaft) and bearing to the radial clearance, while the attitude angle is the angle between line of action and line of centres. In the theoretical model of UVC, the load (F) can be found from dynamic model and thus, Equations (I9) and (I10) are solved as a “reverse” problem to obtain the eccentricity ratio and the attitude angle, such as the mobility method proposed by Hirani *et al* [177, 178]. The method starts from making an initial guess for eccentricity ratio, attitude angle and the rates of change of both terms to evaluate the velocity of journal centre under the known load on the journal bearing until the last time step is reached. These steps are iterated until the convergence criteria is satisfied. Eventually, the

eccentricity ratio and attitude angle can be found by solving the following coupled equations:

$$\begin{Bmatrix} \dot{\varepsilon} \\ \varepsilon(\dot{\phi} - \omega_{av}) \end{Bmatrix} = \frac{F \left(\delta_b / R_b \right)^2}{6\mu L_b R_b} \begin{Bmatrix} M^\varepsilon \\ M^\phi \end{Bmatrix} \quad (\text{II2})$$

where,

$$M^\varepsilon = \frac{I_3 \cos \phi + I_2 \sin \phi}{I_1 I_3 - I_2^2} ;$$

$$M^\phi = \frac{I_1 \sin \phi + I_2 \cos \phi}{I_2^2 - I_1 I_3} ;$$

$$I_1 = \int_{\theta_a}^{\theta_b} B_2 \cos^2 \theta d\theta ;$$

$$I_2 = \int_{\theta_a}^{\theta_b} B_2 \cos \theta \sin \theta d\theta ;$$

$$I_3 = \int_{\theta_a}^{\theta_b} B_2 \sin^2 \theta d\theta ;$$

$$B_2 = \left[1 - \frac{B_1}{\sqrt{B_1 + 0.25}} \log \left(\frac{\sqrt{B_1 + 0.25} + 0.5}{\sqrt{B_1 + 0.25} - 0.5} \right) \right] \cdot \left[\frac{2 + \varepsilon \cos \theta}{(1 + \varepsilon \cos \theta)^2} \right]$$

The maximum film pressure and minimum oil film thickness can be determined after obtaining the eccentricity ratio and attitude angle. The frictional power loss incurred in the hydrodynamically lubricated journal bearing is obtained from the following equation [176, 254]:

$$L_{f,b} = R_b \cdot \omega_j \left[\frac{\mu \omega_j R_b^2 L_b \pi}{\delta_b \sqrt{1 - \varepsilon^2}} \left(\frac{2 + \varepsilon}{1 + \varepsilon} \right) + \frac{\delta_b \varepsilon}{2 R_b} \sqrt{F_x^2 + F_y^2} \sin \phi \right] \quad (\text{II3})$$

where,

$L_{f,b}$ is the frictional power loss of the lubricated journal bearing (W)

F_x is the resultant load on the journal bearing projected in the x-direction (N)

F_y is the resultant load on the journal bearing projected in the y-direction (N)

By linking the journal bearing model with the mathematical model of U-Vane compressor, a proper design of the bearing for the prototype can be finalised.