STUDY OF EFFECT OF OIL INJECTION ON PERFORMANCE OF OIL FLOODED SCREW COMPRESSOR

By

Milan Patel

14MMET19

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2016

STUDY OF EFFECT OF OIL INJECTION ON PERFORMANCE OF OIL FLOODED SCREW COMPRESSOR

Major Project Report II

Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering (Thermal Engineering)

By

Milan R. Patel

(14MMET19)

Guided By

Prof. N K Shah

Mr. Hitesh H Patel

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2016

Declaration

This is to certify that

1. The thesis comprises my original work towards the degree of Master of Technology in Thermal Engineering at Nirma University and has not been submitted elsewhere for a degree or diploma.

2. Due acknowledgment has been made in the text to all other material used.

Milan R. Patel

14MMET19

Undertaking for Originality of the Work

I, **Milan R. Pate**l, Roll. No. **14MMET19**, give undertaking that the Major Project entitled "**STUDY OF EFFECT OF OIL INJECTION ON PERFORMANCE OF OIL FLOODED SCREW COMPRESSOR** " submitted by me, towards the partial fulfillment of the requirements for the degree of Master of Technology in **Mechanical Engineering (Thermal Engineering)** of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

Signature of Student

Date:

Place: Nirma University, Ahmedabad.

Endorsed by

(Signature of Guide)

Certificate

This is to certify that the Major Project Report entitled "**Study of Effect of Oil Injection on Performance of Oil Flooded Screw Compressor**" submitted by **Mr. Milan R. Patel (14MMET19)**, towards the partial fulfillment of the requirements for the award of Degree of **Master of Technology in Mechanical Engineering (Thermal Engineering)** of Institute of Technology, Nirma University, Ahmadabad is the record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination.The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institution for award of any degree.

Prof. N K Shah
Associate Professor,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmedabad.

Mr Hitesh H Patel
Engineering Leader-Rotary Engineering,
CASS-India,
Ingersoll Rand Limited,
Naroda GIDC Estate,
Ahmedabad.

Dr. R. N. Patel
Professor and Head,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmedabad.

Dr. P N Tekawani
(I/C) Director,
Institute of Technology,
Nirma University,
Ahmedabad.

Acknowledgments

I take this opportunity to express deep sense of gratitude and sincere thanks for the invaluable assistance that I have received from my honorable and learned guide Prof N K Shah (Associate Professor, Mechanical Engineering Department, IT, NU). He is the constant source of encouragement and momentum that any intricacy becomes simple. I gained a lot of invaluable guidance and prompt suggestions from him during my thesis work. I remain indebted of him forever and I take pride to work under him. My sincere thanks to my industry guide Mr. Hitesh H Patel for his constant cooperation, valuable suggestions and guidance throughout this dissertation work. I am very much thankful to Mr. Sachin Pagnis for his constant cooperation, valuable suggestions and guidance in solving my problem with ANSYS Fluent. I am very thankful to PDM analysis and Dr. kovacive who has given me SCORG software trial version on our demand. I am very much thankful and due respect to Dr. Sham Rane who has helped us consistently and gave their valuable time to solve our problem with SCORG and ANSYS CFX. Dr. P N Tekwani (Director, IT, NU) and all the faculty members of Mechanical Engineering Department who have directly or indirectly helped me during this dissertation work. My special thanks to my friends of M.Tech Thermal Engineering. At the end I am thankful to god, my family, and colleagues who have directly or indirectly helped me during this dissertation work and for their encouragement.

MILAN PATEL

Abstract

The twin oil flooded twin screw compressor is a positive displacement machine used for compressing gases to moderate pressures. These compressors are widely used in industries. By studying fluid flow and heat transfer behavior inside of the compressor, it is possible to improve performance of the screw compressor. This study can be possible experimentally, analytically and numerically. Out of these numerical and analytical simulation of the screw compressor is included in the present study. The analysis of air flow through 22kW oil injected screw compressor, made up of N35 rotor profile, was carried out. Effect of various oil injection parameters such as oil to gas ratio, oil injection temperature, injection angle and droplet diameter on the performance of screw compressor is studied using thermodynamic modeling. Detailed study of flow physics through oil injected screw compressor can be possible using CFD modeling which is explored in present work.

Key words: screw compressor, oil injection parameters, thermodynamic modeling

Contents

List of Figures

List of Tables

Nomenclature

$\mathring{m}_{lg}$	Leakage entrance
Q	Heat Interaction
W	Work
p	Pressure
T	Temperature
V	Compressor volume
A	Heat transfer area
$\mathring{m}_{ll}$	Leakage leaving
h	Coefficient of heat transfer
ξ	leakage flow resistance coefficient
Z_1	Number of lobes on male
ρ	Fluid density
T_{oil}	Oil temperature
m_{oil}	Oil mass
c_o	specific heat of oil
P_{hp}	High pressure
m_{gas}	gas mass
u_g	internal energy of gas
u_o	internal energy of oil
R	Gas constant
λ	Thermal conductivity
h_{tot}	Total enthalpy
U_g	mesh velocity
Γ	Diffusion coefficient
S_ϕ	Source term
∂v	Boundary of control volume
n	current level
$n+1$	Next level
$\frac{dV}{dt}$	Time derivative of the control volume
A_{cg}	Area of clearance gap
P_o	Total pressure
T_o	Total temperature

Abbreviations

DISCO	Design Integration for Screw Compressor
SCROG	Screw Compressor Rotor Geometry Grid generator
SCOCFD	Screw Compressor Computaional Fluid Dynamics
SCOCAD	Screw Compressor Compter Added Design
CFD	Computaional Fluid Dynamics
SCOCCM	Screw Compressor Computational Continuum Mechanics
PIV	Particle Image Velocimeter

Chapter 1

Introduction

1.1 Gas Compressors

Gas compressors are the devices used for increasing the pressure of gas or vapor either by decreasing its volume (positive displacement) or by applying work in form of high kinetic energy which is converted into pressure by the help of smaller cross sectional outlet area (dynamically).

Types of compressors used commonly are reciprocating, twin screw, single screw, centrifugal, scroll and rotary vane. The application of compressor is vast such as in refrigeration, food preservation, industrial, hospitals etc.The efficiency of compressor can be improved; but should be economical enough to justify the expense of research and development to achieve the improvement.

1.2 Twin Screw Compressor

Twin screw compressor is a machine consist of casing contain two inter-meshing rotors with minimum clearance which produces compression. In twin screw compressor each rotor consist of helical groove affixed to the shaft. One of them is male rotor which is comparatively larger than other rotor called female rotor. The number of lobes in male rotor and number of flutes in female rotor is different for different compressor. Generally number of flutes is kept more compared to lobes on the male rotor for better efficiency. Male rotor is driven by either electric motor or an engine which intern drives the female rotor. Twin screw compressor is basically divided in two groups.

1.2.1 Oil free compressor

Oil free operation implies that the gas compression space is entirely free from oil contamination. These compressors are suitable for compression of a wide variety of gases requiring relatively low pressure ratios at fairly constant volume flow rates. oil free compressor have low pressure ratio. Excessive heating in casing may damage the rotor

1.2.2 Oil flooded compressor

In an oil injected twin-screw compressor, the lubricating oil is injected into the gas stream to absorb the heat of compression. The purpose of oil injection is cooling, lubrication and sealing. This enables much higher pressure ratio in a single stage without inter-cooling and provides significant protection against corrosive gases and rotor damage. No timing gears are required and the male rotor usually drives the female rotor through an oil film between them.

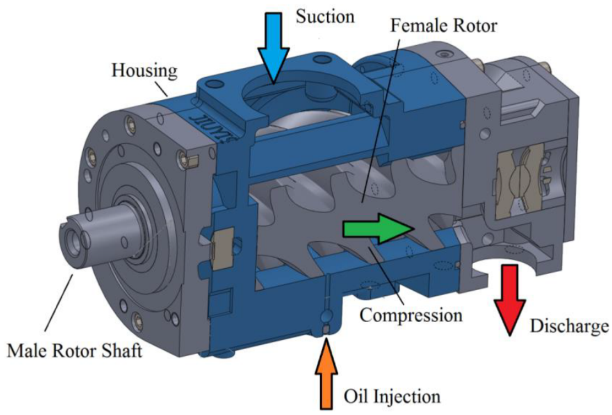

Figure 1.1: Oil injected screw compressor[1]

1.3 Working principle

Figure 1.2 shows the pressure variation in the working chamber in one compression cycle. The numbers on the diagram indicate the state of gas in the process. Number 1 represents start of the process. Point 2 identifies the position at which lobes of the male and female rotors engage at one end to form the control volume which continuously increases after that point. This increase in volume reduces the internal pressure which induces the suction process. Further rotation of the rotors eventually leads to closing of the suction port at which point the working chamber separates from suction – 4. Point 5 indicates the compression process during which the volume reduces and the pressure increases. Further rotation leads to the exposure of the control volume which contains pressurized fluid to the outlet port at which point the discharge process begins. – 6.

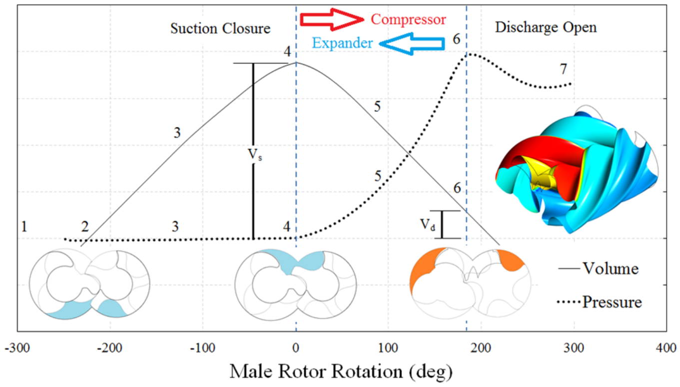

Figure 1.2: Pressure and volume variation in screw compressor[1]

1.4 Motivation

Major advantages of screw compressors in comparison with other compressor types lie in their compactness, durability and wide operating range. The performances of a screw compressor are very sensitive to a number of design parameters and operating parameters which govern the flow process. A particularly strong influence is exerted by the injection of oil which in addition to the lubrication purpose provides better sealing and air cooling.

It is possible to study fluid flow physics of a oil flooded screw compressor with analytically, numerically and experimentally. Experiments requires test setup and instruments also study of parametric variation of design parameters requires numbers of compressors. Thus experimental study is expensive and time consuming where as analytical study and numerical study though theoretical are less expensive and time consuming. Also parametric study can be done very easily compare to experimental one. By using geometrical data of screw compressor and varying various parameters like oil injection flow rate, oil inlet temperature, droplet size, injection location, oil jet speed and angle one can predict the performance of screw compressor can be predicted and based on this performance optimized condition of operation can be determined.

1.5 Problem Definition

One of the important problems which require careful investigation is the influence of oil injection into the working volume of most type of screw compressor for the purpose of sealing, cooling and lubrication. Better sealing means less gas leakage and the improvement of compression and delivery rates. Better cooling increase the compression ratio and lower the outlet temperature of air which reduces the oil cooler size and subsequent components. Lubrication reduces the frictional losses and hence reduce power requirement to drive the compressor. Thus in present work performance analysis of screw compressor with oil injection is planned.

1.6 Objectives of Study

Objectives of present work are as under

- To carry out performance study of screw compressor with oil injection.
- To do parametric study of screw compressor performance by varying oil injection parameters.
- To find optimum oil injection parameters.

It is planned to carry out above mentioned study using thermodynamic modeling and CFD modeling.

1.7 Organization of Thesis

The thesis has been arranged into five chapters. Chapter 1 deals with a general classification of compressors and introduction to the twin screw compressor in more detail and enumerates the objective of the present investigation. In chapter 2, a brief review of relevant literature covering theoretical and experimental studies has been presented. Chapter 3 covers the mathematical modeling and CFD modeling of screw compressor with oil injection. Chapter 4 includes results and discussions. The final chapter is confined to some concluding remarks and for outlining the scope of future work.

Chapter 2

Literature Review

SCORG [1] is a tool for the design and CFD pre-processing of rotary twin screw machines. Apart of the main functionality to construct deforming grids in the rotor domain of a twin screw machine, the software includes additional modules for handling rotor profiles, executing a basic thermodynamic calculation based on chamber model and generating the pre-processing setup routines for selected commercial CFD solvers.

The first practical compressor was invented by Lysholm in 1934 [2] and was mainly developed by SRM (Svenska Rotor Maskiner) of Sweden. In the 1960's, the twin-screw compressor come to existence, providing high capacity with reduced size and cost, together with an option to operate with high compression ratios allowing single stage systems for gas compression and low temperature refrigeration requirements. More recently, there has been a lot of research activity is going on at the City University of London on methods to improve rotor design as well as volumetric and adiabatic efficiencies. Despite the rapid growth in screw compressor usage, public knowledge of the scientific basis of their design is still limited.

2.1 Oil injection in screw compressor

2.1.1 Mathematical and Experimental study literature

Lj milutinovic et.al.[3] Presented some results of experimental investigation of the influence of oil injection upon screw compressor working process. Several parameters that characterize the oil injection were varied over ranges that the compressor performances were evaluated on the basis of measurements of all important bulk parameters. The collected data served to verify the earlier developed model of the oil injection. The obtained experimental results helped in modifying the oil injection system which resulted in saving of compressor energy consumption up to 7%.

Experimental investigations of two prototypes of screw compressor (one of them equipped with the standard and another with the new oil injection system), enabled the effects of all relevant parameters upon the screw compressor performances.

Xueyuan Peng et. al.[4] collected individual process related to oil injection experimentally. They prepared performance characteristic of oil injected screw compressor from number of performance data. P-V diagrams at various conditions for a twin screw compressor have been recorded and analyzed. The beginning of discharge phase is apparently different for different discharge Pressure. Oil atomization and distribution has been observed with the aid of PIV system. Many oil droplets are observed flying in the working chamber. A thin film of oil always exists on the rotor surfaces, and a large amount of oil is foamed and many bubbles are presented in the working chamber.

Stosic and Kovacevic [5] presented some result of mathematical modeling of the oil injection into the screw compressor and its influence upon thermodynamics process. They concluded that the flow, pressure and temperature of working media are the function of rotational angle of male rotor. An important influence of the oil temperature and oil injection location has been noticed, the oil to gas mass ratio, and the oil type have a surprisingly small influence on the compressor process.

Hammerl and Knittl-Frank [6] has modified the oil injection system. They provide different injection variant in which oil is injected by nozzles into suction port of compressor. But they found that there was a deterioration of specific power and volumetric efficacy compare to conventional injection method with its compact oil jet of relatively large diameter.

Paepe et.al. [7] give some experimental result of cooling of compressor with oil atomization. Experiments show that lowering the oil droplet diameter results in a considerably higher heat transfer. Growing oil flow rate, also gives a better cooling effectiveness. In parallel, a thermodynamic model is developed by which the compression process can be calculated for every degree of revolution of the male-rotor. Paper shows that trying to reach isothermal compression through oil atomization is not possible. The importance of the cooling effectiveness in the thermodynamic process is too small to have a significant influence.

N.Seshaiah et.al. [8] presented experiment result of 5.5 kW and 37 kW air compressors. Apart from air as a working gas, the effect of using other working gases such as nitrogen, argon and helium are studied experimentally and mathematically. The result shows that lowering of oil inlet temperature is more effective than injecting a higher quantity of oil.

2.1.2 Numerical study literature

Jianhua Wu and Gang Wang [9] provided a general method for analyzing the oil supply system of a rotary compressor by using computational fluid dynamics. It is found that the main bearing oil flow rate varies circularly along with the rotation of the shaft. The shape and inclination angle of the spiral groove also influence the main bearing oil flow rate.

Rui huang et. al. [10] investigated pressure distribution along water film thickness in single screw compressor. In this paper, they investigated the pressure distribution of the water film with cavitation model. Results were compared to the pressure distribution obtained by one-phase model using N-S equations. Results show that difference of pressure distribution obtained by the two models is very similar for the positive pressure distribution. For the negative pressure distribution, the saturated vapor pressure could be applied to replace the results obtained by one-phase model with very small errors. Therefore, one-phase model will be effective to simulate pressure distribution along the tooth flank by using the vapor pressure to replace the negative pressure.

N. Stosic [11] gives an outline of two methods of computing heat transfer in a screw compressor by means of a quasi-one dimensional differential model and by three dimensional computational fluid dynamics . The 3D CFD procedure is more accurate but requires a far longer running time. Heat in oil flooded screw compressor is low.

Kovacevic et. al.[12] suggested that for the analysis of screw compressor moving, stretching and sliding mesh has to be produced. They have developed design integration package that increases interoperability called DISCO. The interface basically consists of five modules named SCORPATH, SCROG , SCOCAD, SCOCFD and SCONOISE.

Sun-Seok Byeon et. al. [13] performed numerical simulation to investigate the performance of oil injected twin screw air compressor. Finite volume scheme and density based solver with coupled scheme were applied in the computational process. Standard k- turbulent model, implicit formulations were used. The dynamic mesh method utilized to generate the set of meshes required for every time step in FLUENT.

N. Stosic et. al.[14] assessed three strategies of grid deformation; diffusion equation mesh smoothing, user defined nodal displacement and key-frame remeshing. All attempts to solve flow within a twin screw compressor by use of Key-Frame Re-meshing failed, due to the complexity of the numerical mesh.

IVA Papes [15] performed 3D analysis of twin screw expander. The mesh motion is handled by an in-house code which generates a block-structured grid with the help of solutions of Laplace problems. The grid in the inlet, outlet and oil injection ports is stationary (tetrahedral cells) and therefore is not handled by the grid generation algorithm mentioned in the introduction. The grid in the casing is built by stacking two-dimensional structured (rectangular) grids in slices of the casing.

Sham rane et. al. [16] perform the study of multiphase flow at the suction of screw compressor. The purpose of this research is to show whether injection of water or other atomize oil is useful to control the discharge temperature and reduction in energy consumption. They used SCORG for the grid generation and ANSYS CFX solver for calculations. In the multiphase model case water was injected via a 3 mm diameter nozzle in one meter long pipe at the suction to the pipe. It was evident from the CFD simulation that there was no slip between the two phases and both air and water flow mixture can be assumed as homogenous flow. It was found that liquid and gas phase flowing at the same velocity in the upstream pipe and most of the injected water will flow through the middle part of the screw compressor suction port.

W.S. Lee et.al. [17] performed theoretical and numerical models for oil free and oil flooded screw compressor. The effect of oil-injected angle, oil temperature, oil flow rate, built-in volume ratio and other operation conditions on the performance of twin screw air compressors are investigated. It shows that the volumetric efficiency becomes higher as the injected temperature becomes lower. Volumetric and compression efficiency increase with amount of oil increase but specific power decrease. Specific power increase as oil injection angle moved to discharged side.

Chapter 3

Modeling of oil injected screw compressor

3.1 Thermodynamic Modeling

The thermodynamic module is used to perform thermodynamic calculation based on the chamber thermodynamic model which integrates of equations of conservation of mass and internal energy applied to a control volume. The compressor working chamber changes the shape and size with time, depending of the rotor position. The mass and energy flows in and out of the control volume affect the quantity of mass and internal energy of the fluid inside the working chamber. The rate of change of the mass and energy within the working chamber are defined by the conservation laws of mass and energy respectively expressed in terms of differential equations.

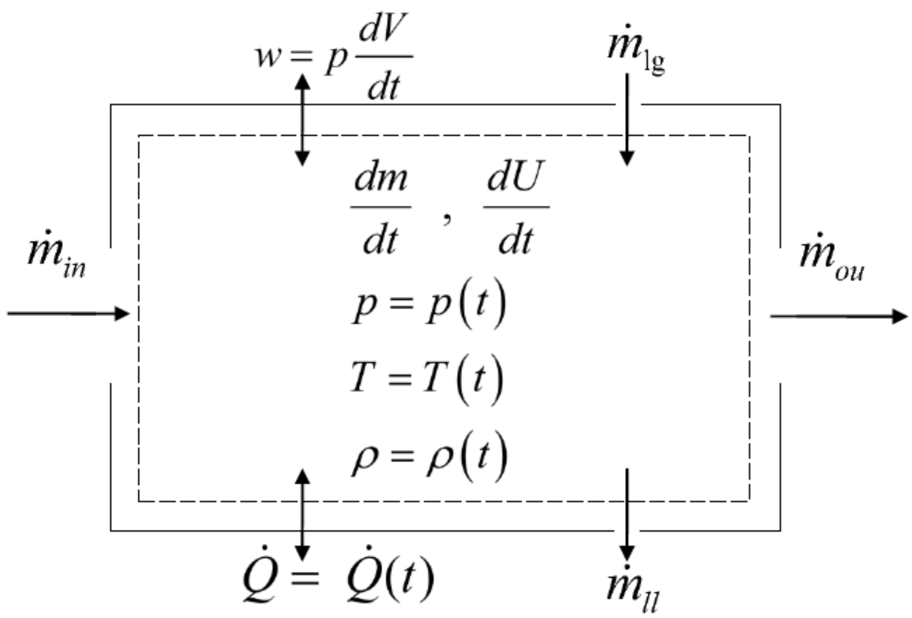

Figure 3.1: Screw compressor control volume for one-dimensional analysis[1]

3.1.1 The mass conservation law

The equation of mass conservation which describes mass variation in the control volume. The mass inflow into the control volume consists of the suction flow when the chamber is connected to this port, the flow of injected fluid and the leakage flow which enters the control volume.

$$\mathring{m}_{in} = \mathring{m}_{suc} + \mathring{m}_{inj} + \mathring{m}_{lg} \tag{3.1}$$

The mass outflow from the control volume consists of the discharge flow and the leakage flows which leave the chamber by

$$\mathring{m}_{out} = \mathring{m}_{dis} + \mathring{m}_{ll} \tag{3.2}$$

The leakage flows $\mathring{m}_{lg}$ and $\mathring{m}_{ll}$ depend on the size of leakage paths.

3.1.2 The energy conservation law

The conservation of energy is in this model represented by equation of internal energy. Since the velocity of the fluid within a screw compressor is relatively low, the kinetic energy of the fluid is negligible in comparison to its internal energy and is therefore neglected. The conservation of internal energy within the control volume may then be written as follows

$$\frac{dU}{dt} = \mathring{m}_{in}h_{in} - \mathring{m}_{out}h_{out} + \mathring{Q} - p\frac{dV}{dt} \tag{3.3}$$

The rate of change of the internal energy within the control volume consists of the difference of energy fluxes, the heat transfer $\mathring{Q}$ and the thermodynamic work $p\frac{dV}{dt}$. The heat exchange with environment is relatively small.

The overall enthalpy of fluids which flow into the control volume is given as

$$\mathring{m}_{in}h_{in} = \mathring{m}_{suc}h_{suc} + \mathring{m}_{inj}h_{inj} + \mathring{m}_{lg}h_{lg} \tag{3.4}$$

The total enthalpy outflow from the control volume is

$$\mathring{m}_{out}h_{out} = \mathring{m}_{dis}h_{dis} + \mathring{m}_{ll}h_{ll} \tag{3.5}$$

3.1.3 Oil injection

In addition to lubrication, the major purpose for injecting oil into a compressor is to cool the gas. The mass flow of injected oil through the oil port is calculated through oil-to-gas mass ratio.

$$\overset{\circ}{m}_{oil} = \frac{\overset{\circ}{m}_{oil}}{\overset{\circ}{m}_{gas}} \frac{Z_1}{2\Pi} \tag{3.6}$$

In SCORG model [1], a simpler procedure is adopted in which the heat exchange with the gas is determined from the differential equation for the instantaneous heat transfer between the surrounding gas and an oil droplet. The heat exchange between the droplet and the gas can be expressed in terms of a simple cooling law

$$Q_o = h_o A_o \left(T_{gas} - T_{oil}\right) \tag{3.7}$$

Where A_o is the Sauter mean diameter of the droplet and h_o is the heat transfer coefficient on the droplet surface.

The above approach is based on the assumption that the oil-droplet time constant is smaller than the droplet traveling time through the gas before it hits the rotor or casing wall, or reaches the compressor discharge port.

3.1.4 Leakage flows

The mass flow of the leaking fluid through the clearance gap is calculated by simplified continuity equation assuming constant temperature, as described by Stosic et al [5]

$$\overset{\circ}{m}_{lg} = \rho w_{cg} A_{cg} = A_{cg} \left(\frac{\Upsilon \left(p_{hp}^2 - p_{lp}^2\right)}{a^2 \left(\xi + \ln \frac{p_{hp}}{p_{lp}}\right)} \right)^{1/2} \tag{3.8}$$

Fluid flow through the leakage gaps is caused by the pressure difference, with the flow direction from the high pressure towards the low pressure.

3.1.5 Thermodynamic properties

Once the mass and energy within the control volume are solved, the thermodynamic properties of the fluid can be obtained at each angle of rotation. The fluid density can be calculated from a known volume of the chamber as

$$\rho = \frac{m}{v} \tag{3.9}$$

The working fluid in current release of SCORG is considered to be an ideal gas and its properties are obtained from equation of state. The fluid temperature is then calculated from the internal energy of the fluid within the chamber as:

$$T = \frac{\Upsilon - 1}{R_g}\frac{U}{m} \tag{3.10}$$

The pressure is then estimated from the equation of state.

$$p = \rho R_g T \tag{3.11}$$

3.1.6 Selection of operating and design parameters

Working conditions define conditions on the boundaries of the thermodynamic model. Table 3.1 describes data that is required as input to the model.

Table 3.1: Inputs for Thermodynamic model

Variable	Description
Wtip	Main rotor tip speed
Rotor speed	Main rotor speed for a specified Wtip and outer diameter
P0	Suction pressure
Pr	Discharge pressure
T0	Suction temperature
Tr	Discharge temperature
Tevp	Evaporator suction temperature (for real fluids)
Tcond	Condenser saturation temperature (for real fluids)
Ts	Average temperature of casing
X	dryness fraction at suction

3.1.6.1 Oil injection settings

The thermodynamic model allows the user to define oil injection in the compression chamber during calculations. Data for oil injection is in terms of the geometric description of the oil injection and also the properties of the injected oil Table 3.2

Table 3.2: Oil injection settings

Variable	Description
Ratio	Oil to Gas mass ratio
P	Pressure in Oil reservoir
T	Oil injection temperature
Injection Angle	Oil injection angle
Axial Position	Axial position of oil injection port
Port Diameter	Oil injection port diameter
Doil	Oil droplet mean diameter
CpOil	Oil droplet mean diameter
ρ	Density of injected oil
Viscosity of Oil	Kinematic viscosity of injected oil at T

3.2 CFD Modeling

This chapter includes CFD modeling details of oil flooded screw compressor.

Specification of the N35, 22 kw oil flooded screw compressor model is mentioned in below Table 3.3

Table 3.3: Screw compressor specification

Compressor detail	Specification
Profile Relative	N35
length (L/D)	1.6
Wrap angle	285
Axis distance	93 mm
Interlobe gap	0.06 mm
Axial gap	0.06 mm
Radial gap	0.06 mm

3.2.1 Geometry creation

The geometry of rotor and casing Figure 3.2 is prepared by SCORG itself by providing different inputs as mentioned in Table 3.3. The geometry of suction and discharge port can prepare separately in any CAD software but in this work it was provided. The geometry calculation need to be calculated in order to provide inputs for thermodynamic and grid module.

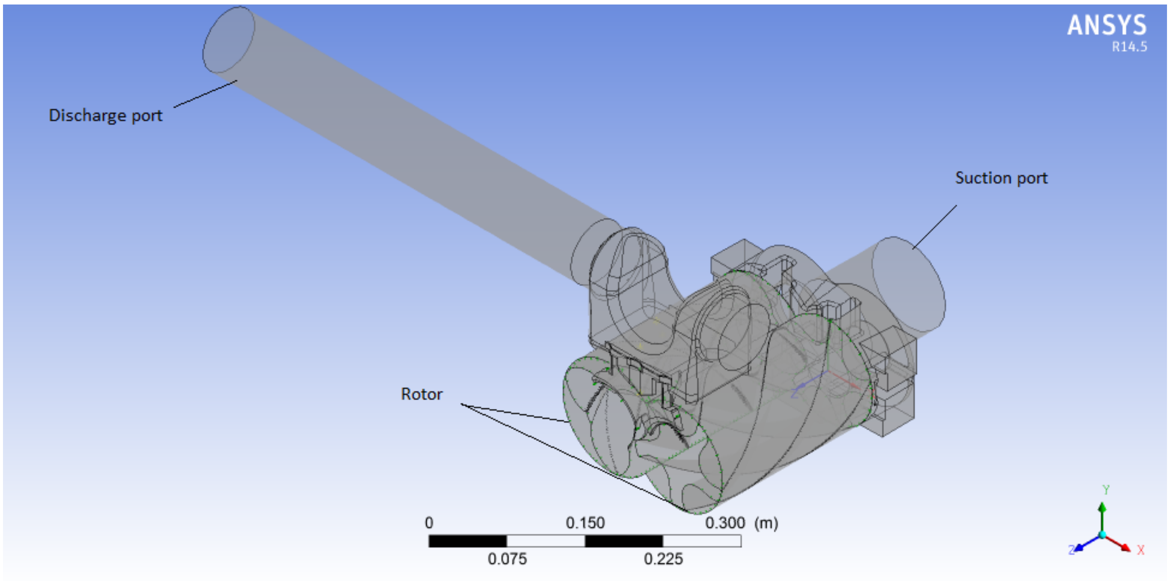

Figure 3.2: Isometric view of oil flooded screw compressor

3.2.2 Grid Generation

One of the most important process in any CFD analysis is meshing or grid generation of the geometry. The success of any CFD analysis depends on the type of meshing and its quality. In case of screw compressor it is most challenging part to generate dynamic mesh which cannot fail while simulation. In screw compressor the gap between rotors and rotor and casing is in microns so to create dynamic mesh in this gap is more complex. It is difficult to handle this type of problem using popular available software.

There are other commercial software like SCORG and Twin-mesh specially developed to create mesh for positive displacement rotary machines. Out of these two SCORG is utilized in present work to prepare dynamic mesh of screw compressor geometry.

Stages for grid generation of screw machine in SCORG is shown in Figure 3.3

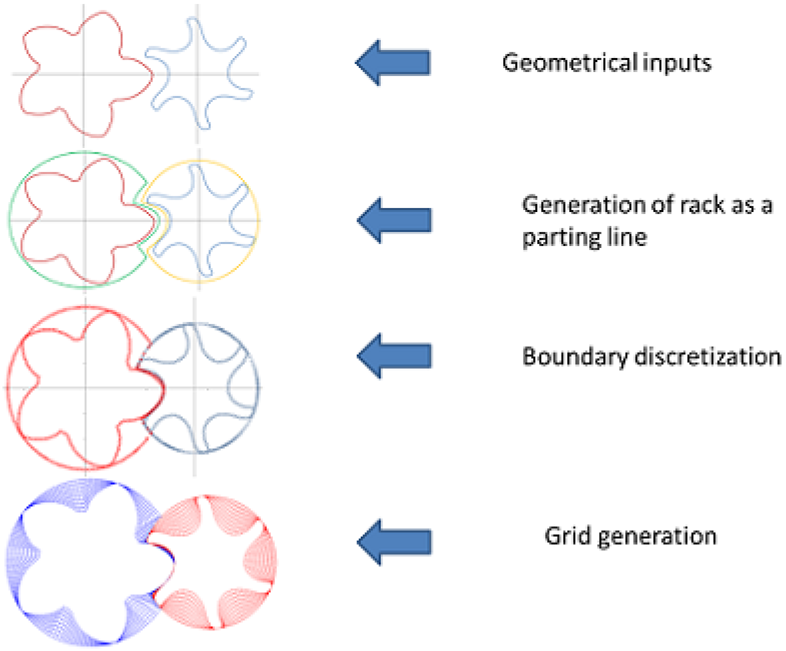

Figure 3.3: Step of grid generation using SCORG[12]

3.2.2.1 Grid type and parameters

SCORG generates structured 'O' type grid for rotor domain. Grid for high pressure, low pressure are hexahedral. Rack to rotor conformal meshing is recommended for oil injected compressor. Numerical grids for rotor domains and compressor port domains are connected through non-conformal conservative interfaces in the solver selected for CFD calculation.

The parameters set for rotor domain in SCORG is shown in Table 3.4

Table 3.4: Parameters for rotor domain in SCORG

parameters	value
Number of profile points	400
Interlobes divisions	60
Vertex file starts	1
Vertex files ends	50
Regularization factor for male	3.5
Regularization factor for female	0.25
Rack smoothing factor	0.6
Mesh Orthogonalisation	1
Smoothing iteration	1
Smoothing coefficient	0.2

3.2.2.2 Domain grid and grid statistics

Whole fluid domain grid Figure 3.4is shown in and its statistics shown in Table3.5

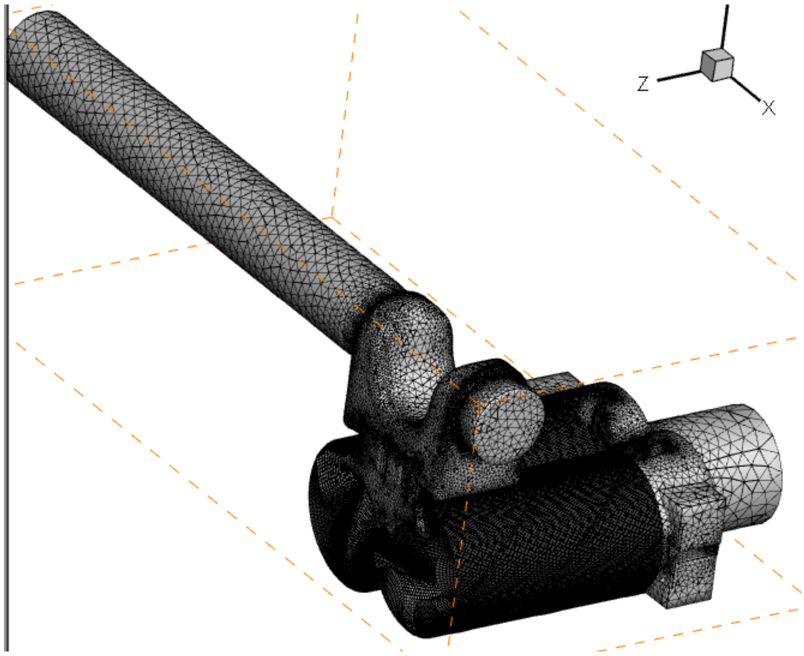

Figure 3.4: Fluid domain grid

Table 3.5: Grid statistics

Name of component	Type of grid	No. of nodes	No. of elements
Suction	Tetrahedral	94201	484090
Discharge	Tetrahedral	76484	398885
Rotor	Hexahedral	411026	355416
Assembly		581711	1238391

3.3 Boundary conditions

It is very essential to specify the boundary conditions like inlet, outlet, oil injection inlet, interface at all zones. A zone-type specification defines the physical and operational characteristics of the model at its boundaries and within specific regions of its domain. The boundary conditions are pressure inlet and pressure outlet.

- At inlet of suction port : Boundary type : Opening, pressure 1 bar, temperature 300 K, air volume fraction 0.9999, oil volume fraction 0.0001.

- At outlet of discharge port: Boundary type : Opening, pressure 2 bar, temperature 350 K, air volume fraction 1, oil volume fraction 0.

Three interfaces are needed to be defined between following specified domains.

- Domain interface 1 : Suction port and axial interface of rotor

- Domain interface 2 : Discharge port and radial interface 1 and 2

- Domain interface 3 : Discharge port and axial interface of rotor

3.4 Governing equations

In an Eulerian reference, the conservation of mass, momentum and energy applied to fluid flow in a control volume can be defined by coupled, time dependent, partial differential equations.

continuity equation

$$\frac{\partial(\rho U)}{\partial t} + \nabla * (\rho U) = 0 \tag{3.12}$$

Momentum eqation

$$\frac{\partial(\rho U)}{\partial t} + \nabla * (\rho U * U) = -\nabla p + \nabla * \tau \tag{3.13}$$

Total energy equation

$$\frac{\partial(\rho h_{tot})}{\partial t} - \frac{\partial p}{\partial t} + \nabla * (\rho U h_{tot}) = \nabla * (\lambda \nabla T) + \nabla * (U\tau) \tag{3.14}$$

With respect to dynamic meshes, the integral form of the conservation equation for a general scalar on an arbitrary control volume whose boundary is moving can be written as follows

$$\frac{d}{dt}\int_{V} \rho\phi dV + \int_{\partial V} \rho\phi(u - u_g) * dA = \int_{\partial V} \Gamma\nabla_{\phi} * dA + \int_{V} S_{\phi} dV \tag{3.15}$$

By using a first-order backward difference scheme, the time derivative term in can be written as

$$\frac{d}{dt}\int_{V} \rho\phi dV = \frac{d}{dt}\int \frac{(\rho\phi V)^{n+1} - (\rho\phi V)^{n}}{\triangle t} \tag{3.16}$$

3.5 Simulation in CFX

CFX is very useful tool to predict the flow undergoing in various parts of compressor. Using SCORG screw rotor domain is divided in subdomains and further into a series of 2D domains from which numerical mesh is generated for every position of the rotor which stores in the form of fortron code. CFX uses this 2D domain grids using fortron complier (Intel parallel studio). Ansys is commercially available CFX software which consists of different turbulence models, was used to simulate the flow. In calculation, finite volume method was used for the discretization of governing equations.

Turbulence Model

Turbulence model is used to predict the effects of turbulence. SST (Shear Stress Transport) turbulence model is a widely used in Computational Fluid Dynamics. This model combines the k-omega turbulence model and K-epsilon turbulence model such that the k-omega is used in the inner region of the boundary layer and switches to the k-epsilon in the free shear flow.The formulation of the SST model is based on physical experiments and attempts to predict solutions to typical engineering problems.

Heat transfer model

The total energy option activates the full enthalpy equation, including compressibility effects. It is used for high speed flows or anywhere else where compressibility is important.

The thermal energy option activates the enthalpy/temperature equation, but does include compressibility effects. It is used for low speed thermal models like AC flows, combustion, heat exchangers, etc.

3.5.1 Other relevant parameters related to modeling

- Discretization : Finite-Volume Method
- Analysis type : Transient
- Iteration/time step : Minimum = 1, Maximum = 10, Time step : 1500
- Air and oil were used as a working fluid.
- Initial condition :
  - Pressure : 101325 Pa
  - Temperature : 298 K
- Models used for turbulence, heat transfer and multi-phase are
  - Turbulence : Shear stress transport (SST)
  - Heat transfer : Total energy and Thermal energy
  - Multi phase : eulerian-eulerian model where air and oil are continuous fluid
- Convergence criteria used for different equations are

  - Continuity equation 0.001
  - Momentum equation 0.001
  - Energy equation 0.001

- Pressure velocity coupling
  - A collocated layout was used for the pressure velocity coupling.

- Linearization technique :
  - For momentum and turbulence, second order upwind scheme was used.

- Ansys CFX solver settings
  - Double precision, the double precision was selected because it is more accurate and free from round of errors.
  - The grid was checked by the solver itself.
  - Parallel processing is done.

Chapter 4

Results and Discussion

4.1 Result of thermodynamic modeling

In this section the effect of oil injection parameters on performance and discharge temperature of screw compressor is presented.

The operating condition table 4.1 remains same but the oil injection parameters very.

Table 4.1: operating condition

Working fluid	Air
Tip speed	53.39 m/s
Inlet pressure	1 bar
Outlet pressure	3 bar
Inlet temperature	20 degree
Outlet temperature	90 degree
Port diameter	20 mm
Axial position	100 mm

4.1.1 Effect of oil-gas ratio

The oil to gas ratio very from 0.1 to 6 and other parameter keep constant Table 4.2 Its effect on performance, discharge temperature and bearing is shown Figure 4.1

Table 4.2: oil-gas ratio parameter

oil to gas ratio variation	0.1 to 6
oil injection temperature	45 C
oil injection angle	40 deg
droplet diameter	0.01mm

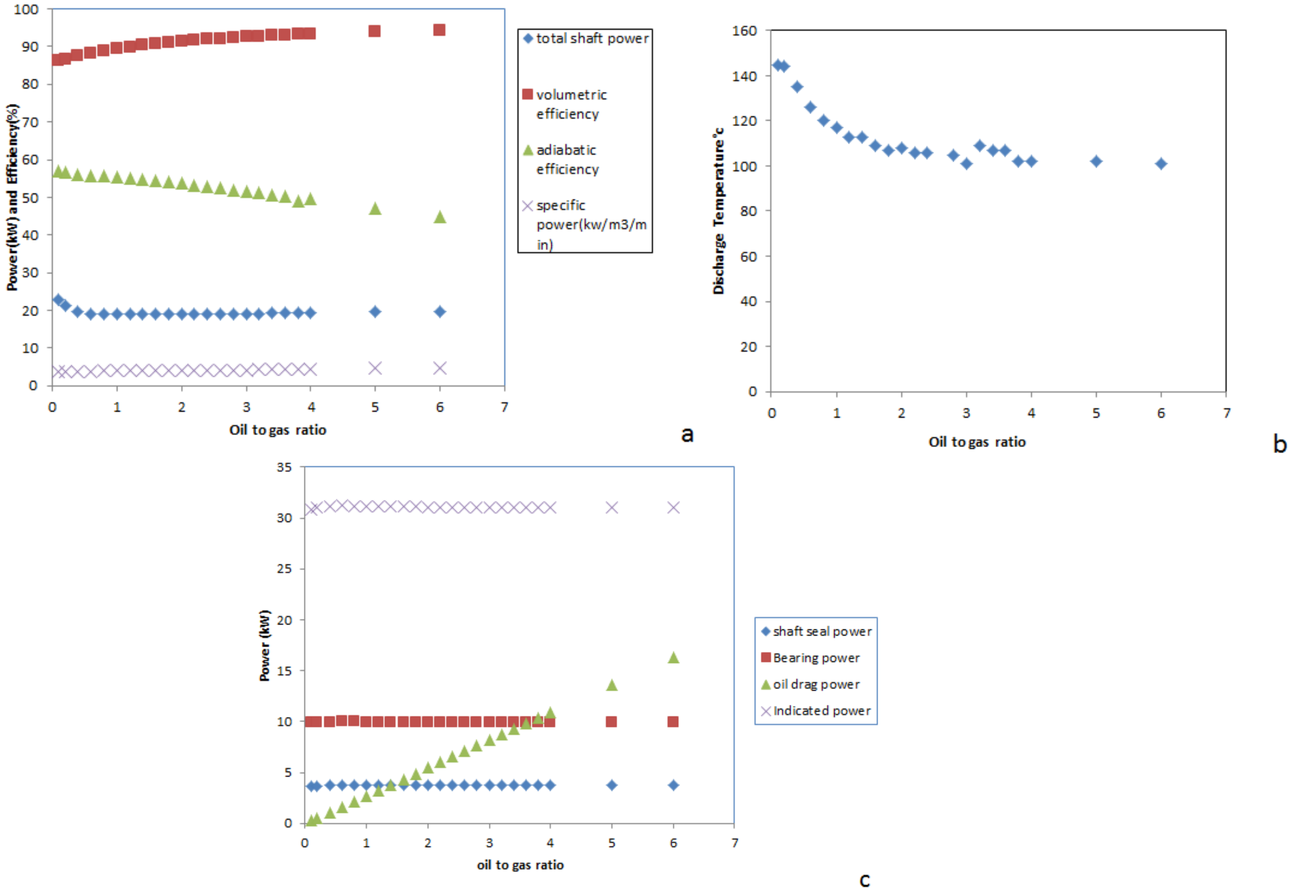

Figure 4.1: Effect of oil to gas ratio

The volumetric efficiency increase because there is reduction of discharge temperature and on other hand there is decrease in adiabatic efficiency beyond the oil to gas ratio of 1.6. As the mass of oil increase there is better cooling of rotors hence discharge temperature decrease and size of oil cooler, fan can be optimize but oil drag power linearly increase with ratio which increase unnecessarily power consumption of compressor. There are almost negligible effect on bearing power and shaft seal power. Indicated power slightly increase up to 1.6 then remain constant.

4.1.2 Effect of oil injection temperature

The oil injection temperature vary from 25 to 100 °C. The effect of oil injection temperature on performance, discharge temperature and bearing power is shown in Figure4.2

Table 4.3: Injection temperature parameters

injection temperature variation	25 to 100 °C
oil to gas ratio	1.6
oil injection angle	45 deg
droplet diameter	0.01mm

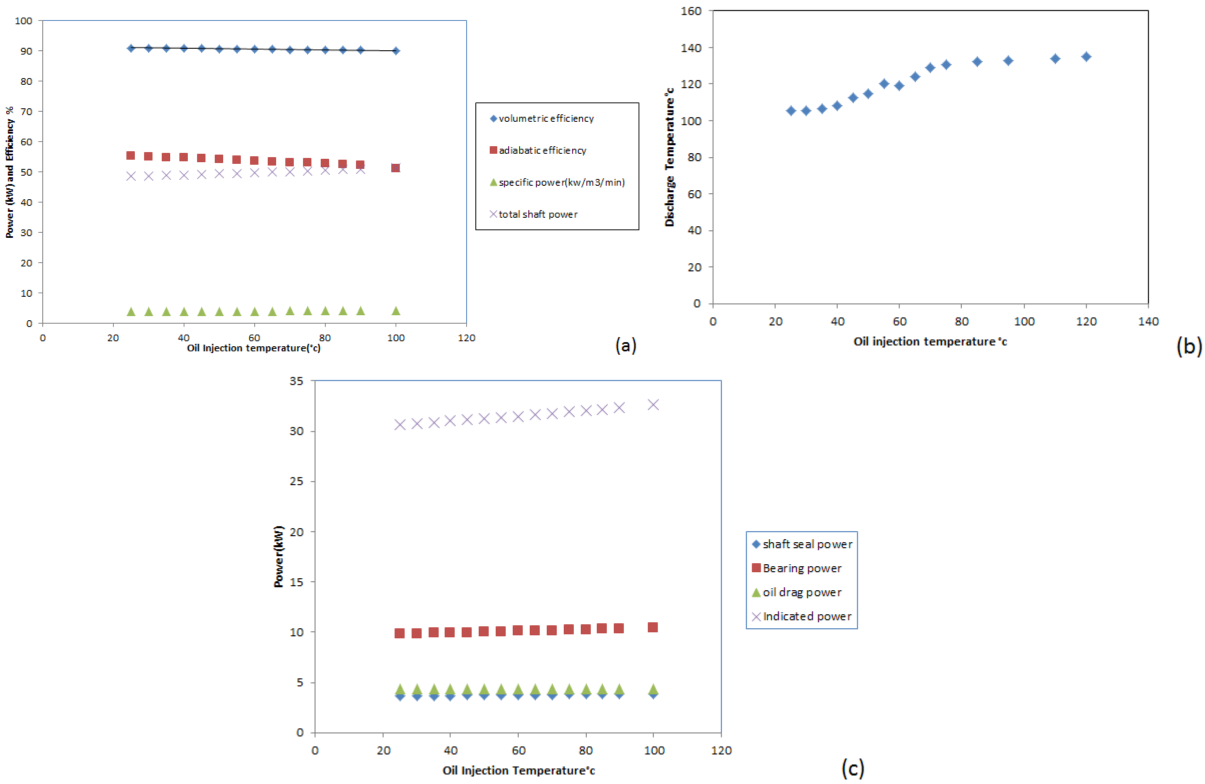

Figure 4.2: Effect of injection temperature

It is obvious that as oil injection temperature increase the discharge temperature also increase. Both volumetric and adiabatic efficiency decrease and specific power and total shaft power increase with oil temperature which is not desirable. Higher injection temperature is not recommended. The oil injection temperature should be higher then its condensation temperature if temperature inside casing goes below to its condensation temperature there is solid particle formation which chock up rotor and may damage the rotors. Here, the optimum point of injection temperature is 45 °C.

4.1.3 Effect of oil injection angle

The effect of oil injection angle on performance, discharge temperature and bearings is shown in 4.3

Table 4.4: Injection angle parameter

injection angle variation	30 to 200 degree
oil to gas ratio	1.6
oil injection temperature	45 °C
droplet diameter	0.01mm

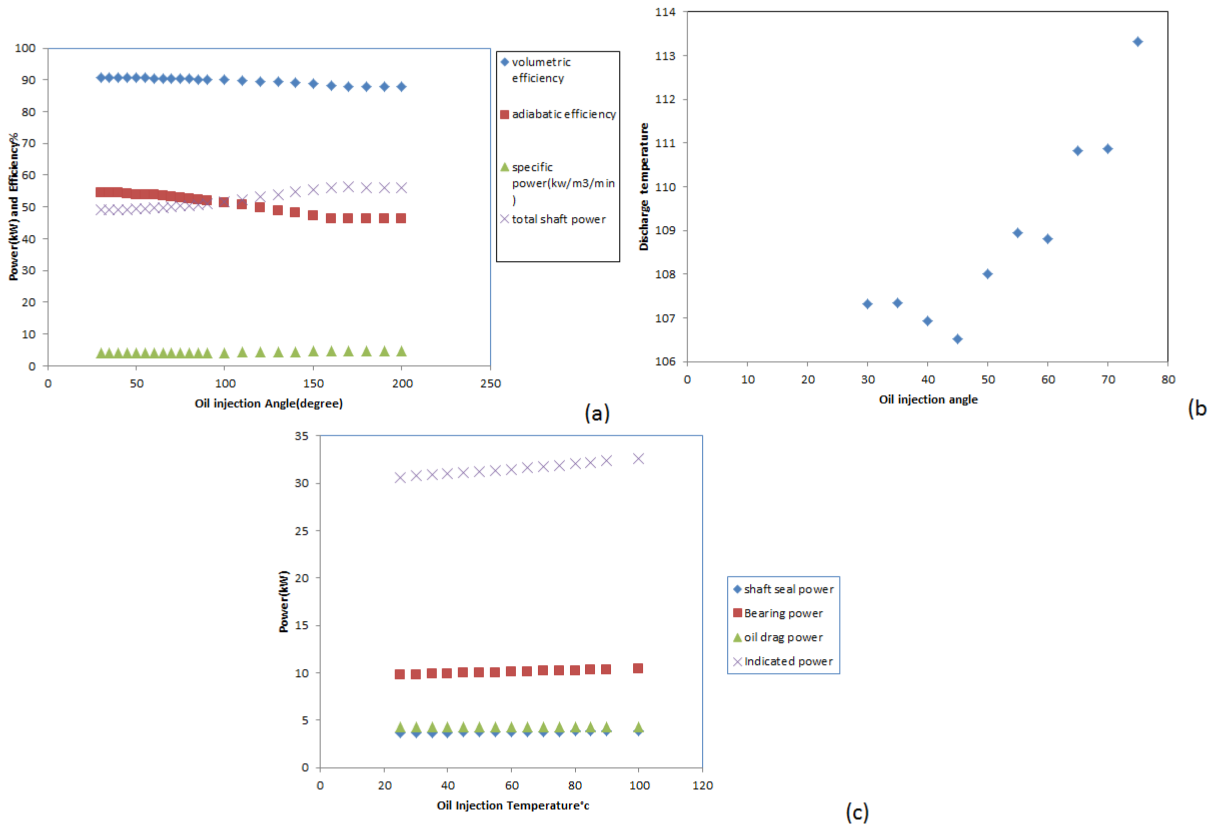

Figure 4.3: Effect of oil injection angle

The influence of oil injection angle on discharge temperature is initially decrease but as angle increase the temperature also increase drastically. The specific power increase and adiabatic and volumetric efficiency decrease. The oil drag and shaft seal power is nearly constant but indicated power increase. With regards to discharge temperature and compressor efficiency the optimal injection angle is 40 to 45 degree. Earlier angle near to discharge port gives better result on performance of screw compressor.

4.1.4 Effect of oil droplet diameter

The oil droplet diameter vary from 0.001 to 2 mm. The effect of oil injection angle on performance, discharge temperature and bearings is shown in Figure 4.4

Table 4.5: oil droplet diameter effect

droplet diameter variation	0.001 to 2 mm
oil to gas ratio	1.6
oil injection angle	45 deg
oil injection temperature	45 °C

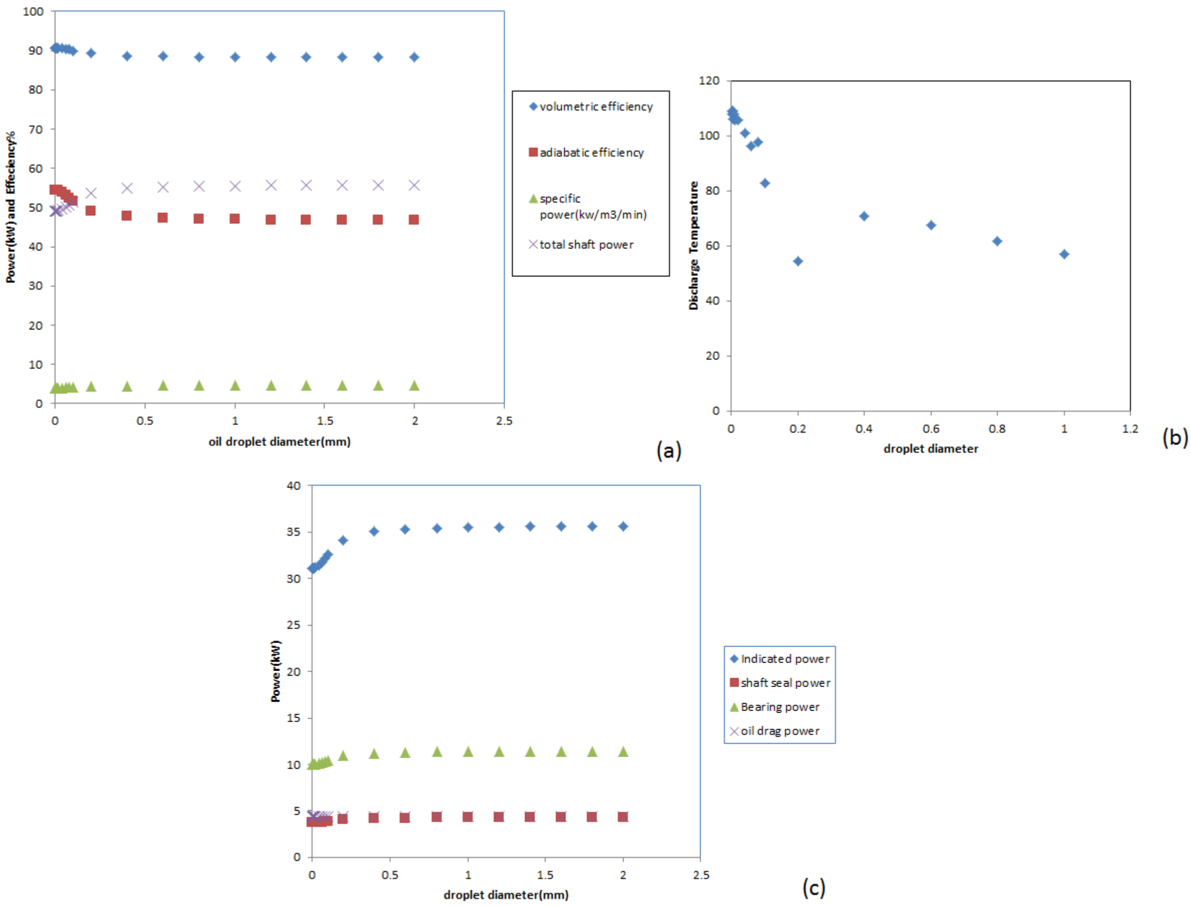

Figure 4.4: oil droplet diameter effect

The specific power and total shaft power increase with droplet diameter which increase the oil drag and power consumption of screw. There is higher volumetric and adiabatic efficiency at small droplet which increase better cooling and heat interaction with air particles. Smaller droplet diameter can be achieved by nozzle or by increasing oil injection pressure with simple orifice. The discharge temperature decrease with bigger droplet hence compromise must be made between efficiency and discharge temperature. With regard to efficiency and discharge temperature the optimum droplet diameter is 0.01 mm.

4.1.5 Optimized parameters

After study we are getting following optimized parameters Table 4.6.

Table 4.6: Optimized parameters

Parameter	Value
Oil to gas ratio	1.6
Oil injection temperature	45 ^{o}c
Oil injection angle	45 degree
Droplet diameter	0.01 mm

4.2 Result of CFD modeling

The result of CFD modeling of side oil injection screw compressor up to 240 time step is shown in figure 4.5. Total simulation time step is 1500 but solver get stable after 900 time step. The oil injection holes are normal to the rotor lobes when they pass through every cycle. On the gate rotor side oil free rotor profile have larger width and this can block the oil flow or produce solver instability in the clearance gaps so we are able to do simulation up to 240 time step after that we found error of solver handler and fatal overflow.

Here, pressure goes 2 bar, maximum temperature 442 K.

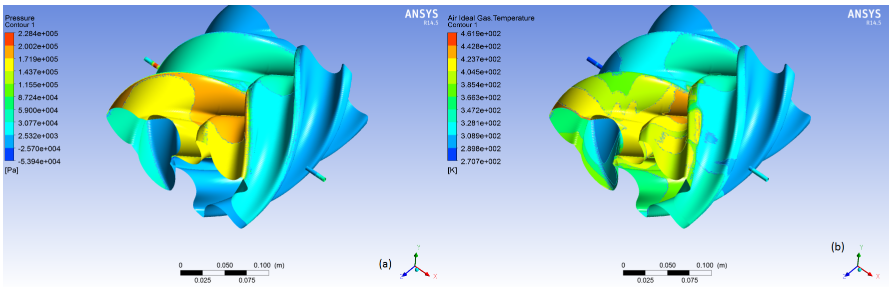

Figure 4.5: contour (a) pressure, (b) temperature up to 240 time step

4.2.1 Oil Injected from suction port

After getting error and difficulty in solving oil injection from side of rotors we assume that oil enters the domain from the main suction pipe in the form of a uniform spread of droplet. Prediction may be possible on the effect of oil flow from suction. Comparison between oil free and oil injected compressor at 1500 time step and 545 time step respectivelly and with boundary condition as shown in section figure 3.3.

The pressure variation between oil free and oil injected compressor is shown in figure 4.6 In oil free pressure is 2.35 bar while in oil flooded it is 2.50 bar.

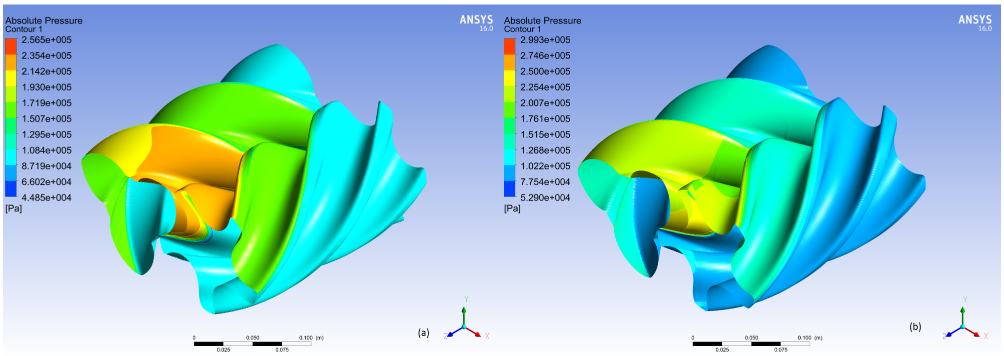

Figure 4.6: Pressure contour (a) oil free (b) oil injected

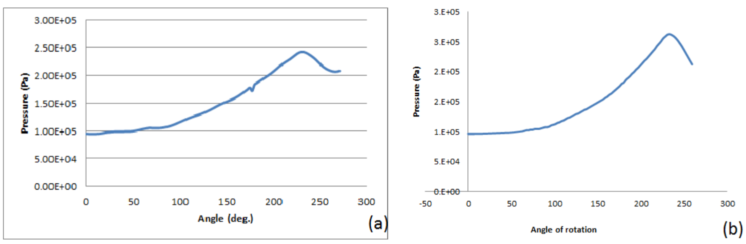

Figure 4.7: Pressure variation with rotation angle (a) oil free (b) oil injected

The temperature variation between oil free and oil injected compressor is shown figure 4.8 . Maximum temperature in oil free is 472 K and temperature in oil injected is 440 K.

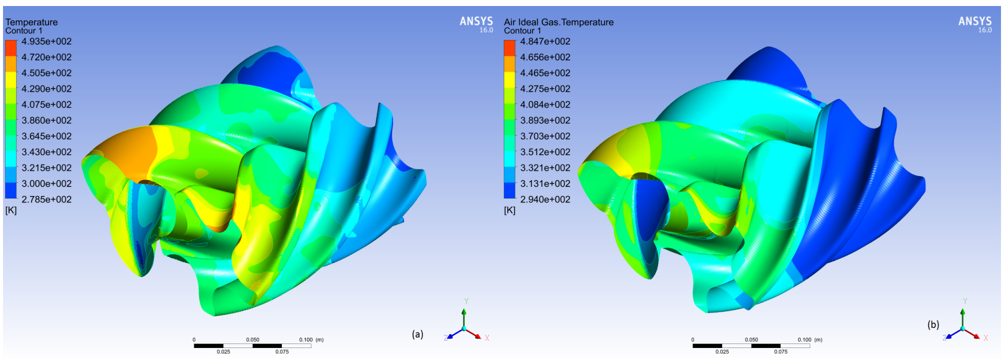

Figure 4.8: Temperature contour (a) oil free (b) oil injected

Chapter 5

Conclusion and Future work

5.1 Conclusion

The following conclusions can be drawn from the present study:

1. By lowering oil to gas ratio volumetric efficiency decrease, adiabatic efficiency increase and discharge temperature. Total shaft power and specific power are not changing much with change in oil to gas ratio. To reduce the discharge temperature oil to gas ratio should be higher but after certain value of oil to gas ratio discharge temperature remain constant. Thus certain value of oil to gas ratio between 1 and 2 gives minimum discharge temperature, better adiabatic efficiency, a little value of volumetric efficiency should be compromised.

2. Higher oil injection angle affects volumetric efficiency, adiabatic efficiency and power and discharge temperature. But out of these it affects discharge temperature a lot. For oil injection below and above 45 degree discharge temperature increase however effect in increase in angle found be much higher than decreasing it. Thus one can say 45 degree of oil injection angle gives optimized result.

3. Lower oil injection temperature gives higher efficiency and shaft power decreases .Also lowering oil injection temperature discharge temperature reduce but care should be taken so that condensation of moisture from air should not take place.

4. Small droplet diameter is useful for better volumetric, adiabatic efficiency, total shaft power and specific power.

5. CFD simulation shows that there is increase in pressure from 2.35 bar to 2.50 bar and reduction in temperature from 472 K to 440 K so in oil injected

compressor one can get higher pressure and lower temperature.

5.2 Future scope

- CFD of side oil injection and its parametric study with size of port diameter, angle, different types of oil, axial position of injection port.
- Parametric study with different rotor profile and different working fluid and oil injection.

Bibliography

[1] SCORG help manual 2015.

[2] nagam seshaisha, 2006, experimental and Computational Studies on oil injected twin-screw compressor. P.hd thesis,NIT Rourkela.

[3] N.Stosi, L j. Milutinovi, K. Hanjalic and A. Kovacevic 1990 experimental investigation of the influence of oil injection upon the screw compressor working process international compressor engineering conference, Paper 687.

[4] Xueyuan Peng, Ziwen Xing, Xiaojun Zhang, Tiansheng Cui, Pengcheng Shu, 2000 Experimental Study of oil Injection and its Effect on Performance of Twin Screw Compressor Division of Compressor, international compressor engineering conference, Paper 1491.

[5] N. Stosic, A. Kovacevic, K. Hanjalic, Lj. Milutinovic 1988 Mathematical modeling of the oil influence upon the working cycle of screw compressors International Compressor Engineering Conference. Paper 645 .

[6] Klaus Hammerli, Laurenz Rinder ,2000, Modification in the Design of the oil-injection System for Screw Compressor, International Compressor Engineering Conference. Paper 1489.

[7] M. De Paepe , W. Bogaert, D. Martens ,2005,Cooling of oil injected screw compressors by oil atomization 1359-4311.

[8] N.Seshaiah, R.K. Sahoo, S.K. Sarangi, 2006, Influence of Working Gas and Oil Injection Rate on Efficiency of Oil Injected Twin Screw Air Compressor. .

[9] Jianhua Wu, Gang Wang, 2013,Numerical study on oil supply system of a rotary compressor, 425-432

[10] Rui Huang Ting Li Quanke Feng Weifeng Wu, Numerical Calculation and Analysis of Lubricating Water Film Cavitation of A Water Flooded Air Single Screw Compressor, International Compressor Engineering Conference. Paper 2241

[11] N. Stosic, 2015 On heat transfer in screw compressors, International Journal of Heat and Fluid Flow 51 285–297.

[12] A.Kovacevic, N. Stosic, E. Mujic, I.K. Smith, 2007 CFD integrated design of screw compressors, Engineering Applications of Computational Fluid Mechanics .

[13] Sun-Seok Byeon, Jae-Young Lee, Youn-Jea Kim, Numerical analysis on the flow characteristics of oil-injected screw air compressor, 440-746.

[14] Sham Rane, Ahmed Kovacevic, Nikola Stosic, Madhulika Kethidi, 2013, Grid deformation strategies for CFD analysis of screw compressors, International journal of refrigeration 36 1883 – 1893.

[15] IVA Papes, JORIS Dedroot, and JAN VIERENDEELS 3D CFD Analysis of a Twin Screw Expander for Small Scale ORC System .

[16] Arjeneh Mohammad, Kovacevic Ahmed, Gavaises Manolis and Rane Sham,2014, Study of Multiphase Flow at the Suction of Screw Compressor.

[17] W.S. LEE R.H. MA S.L. CHEN W.F. And H.W. HSIA Numerical Simulation and Performance Analysis of Twin Screw Air Compressors.