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Performance Analysis of Oil Injected Twin Screw Compressor

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Abstract

Oil injected twin-screw air compressors are widely used for medium pressure applications in cryogenic industries. Conversion of these compressors for helium applications is in great demand due to their inherent advantages. A mathematical model of an oil injected twin-screw compressor has been constructed basing on the laws of perfect gas and standard thermodynamic relations to evaluate compressor efficiencies. The complete model has been validated using experimental data. Flow and heat transfer coefficients required for computer simulation are experimentally obtained and used in performance prediction, the working medium being air or helium. For example, the heat transfer coefficient is determined from experimentally obtained volumetric efficiencies at varying inlet gas temperatures. Influence of injected oil temperature and quantity, pressure ratio and rotational speeds on volumetric and adiabatic efficiency for sample rotors is analyzed. Conversion of oil injected twin-screw air compressor for helium compression is aimed at, based on numerical and experimental results.

(Key words: screw compressor, internal leakage, clearances, oil injection, simulation and heat transfer)

Nomenclature

A	Heat transfer area (m^2)
A_b	Area of blowhole (m^2)
A_c	Clearance leakage area (m^2)
A_e	Experimental adiabatic power developed by the compressor (W)
A_f	Cross sectional area of female rotor groove (m^2)
A_i	System input power to the compressor (W)
A_m	Cross sectional area of male rotor groove (m^2)
A_t	Theoretical adiabatic power developed by the compressor (W)
a	Clearance between lobe tip and housing(m)
C	Flow coefficient (dimensionless)
c_l	Specific heat of oil (J/kg K)
c_p	Specific heat of gas at constant pressure (J/kg K)
c_v	Specific heat of gas at constant volume (J/kg K)
D	Rotor diameter (m)
h	Heat transfer coefficient between gas and oil (W/m^2–K)
H_g	Enthalpy of gas (J)
k	Ratio of specific heats
M_g	Mass of gas in the working space (kg)
M_l	Mass of oil in the working space (kg)
M_{ts}	Theoretical gas mass inducted into a pair of male and female cavities at suction temperature T_s (kg)
M_{t1}	Theoretical gas mass in a pair of male and female cavities at the end of suction process (kg)
M_{il}	Interlobe leakage mass leaked into the suction cavity during previous compression process (kg)
m	Mass flow rate through flow path (kg/s)
m_b	Leakage mass flow rate through blowhole (kg/s)
m_d	Leakage gas mass flow rate through clearance between rotor end and casing wall (kg/s)
m_e	Actual discharged gas mass rate (kg/s)
m_t	Theoretical inducted gas mass rate in to the displacement volume at suction temperature (kg/s)
m_{ic}	Leakage gas mass flow rate through interlobe clearance (kg/s)
m_{lr}	Leakage mass flow rate of oil through rotor tip housing clearance (kg/s)
m_{gl}	Total gas leakage rate (kg/s)
m_{ds}	Theoretical discharged gas mass rate (kg/s)
N	Rotational speed of rotor (RPM)
P	Pressure in the working space (N/m^2)
P_d	Discharge pressure (N/m^2)
P_s	Suction pressure (N/m^2)
Q	Transferred heat between gas and oil (J)
q	Leakage volume flow rate through flow path (m^3/s)
R	Gas constant (J/kg-K)
R_m	Modified gas constant of oil gas mixture (J/kg-K)
r	Pressure ratio (P_2/P_1)
S	Sealing line length along the rotor (m)
T	Temperature (K)

[1]Corrosponding author

t Time (s)
T_1 Mean temperature of gas and leaked oil at the end of suction process (K)
T_g Temperature of gas in the working space (K)
T_l Temperature of oil in a working space (K)
T_{oil} Mean temperature of leaked oil in suction cavity(K)
T_s Suction gas temperature (K)
t_s Time required for suction process (s)
U Internal energy (J)
V Volume of the working space (m^3)
V_t Rotor tip speed (m/s)
V_1 Inducted gas volume at suction condition (m^3)
V_2 Leakage gas volume at inlet pressure (m^3)
V_{t1} Geometrical volume of a pair of male and female cavities of working chamber (m^3)
V_{th} Geometrical volume of all cavities of the working chambers for one revolution (m^3)
W Gas work (J)
w_t Lobe tip width (m)

Greek Symbols

η_{tv} Theoretical volumetric efficiency (dimensionless)
η_{ev} Experimental volumetric efficiency (dimensionless)
η_{ta} Theoretical adiabatic efficiency (dimensionless)
η_{ea} Experimental adiabatic efficiency (dimensionless)
ϕ Mass ratio of oil and gas
θ Rotor rotational angle (degrees)
φ Male rotor wrap angle (degrees)
β Modified adiabatic index (dimensionless)
μ Dynamic viscosity of oil (N-s/m^2)
ρ Density (kg/m^3)
ε Interlobe clearance (mm)

Subscripts

1 Up stream
2 Down stream
b Beyond flow path
f Female rotor
g Gas
i Gas or oil going in to the working space
l Oil
m Male rotor
o Gas or oil going out of the working space

Introduction

The screw compressor is a positive displacement machine that uses a pair of intermeshing rotors housed in a suitable casing to produce compression. It is capable of high-speed operation over a wide range of operating pressures. In screw machines, oil is deliberately injected into the compression chamber to obtain better sealing, lubrication and cooling effect. Rotary dual screw compressors are widely used in the refrigeration and cryogenic industries. They are particularly suitable for small and intermediate size cryogenic refrigerators and liquefiers.

A Computerized method for twin-screw rotor profile generation and analysis has been suggested by Pawan, J. Singh etal [7]. The same author [8] developed a generalized performance computer program for oil flooded twin-screw compressors. Due to the high cost of energy, particularly in India, it is necessary that all machines are efficient in operation. This can be achieved only when machine performance is well understood and predictable. Unlike other compressors, the mechanism of gas compression in oil injected screw compressors is extremely complex. It is difficult to estimate compressor performance analytically. On the other hand, experimental studies are prohibitively expensive because a new rotor needs to be fabricated using expensive machining techniques for every change in rotor geometry.

Modeling of Compressor Cycle

Analysis of power and volumetric efficiency is essential to know how efficient the compressor is. The main objective of the present performance analysis is to develop an analytical model to ascertain the performance of a prototype compressor when it is applied to compress different gases. Efficiency of any compressor depends on the processes involved. Major compressor processes are suction, compression and discharge. The analysis of these processes is required to model the compressor cycle.

Suction Process

Volumetric efficiency of the compressor depends on the amount of gas mass inducted into the suction cavity. This in turn depends on the temperature of the cavity walls. Analysis of suction process gives the temperature of gas in the suction cavity at the end of suction process. Since the pressure and temperature fluctuations in the process are generally negligible [1], calculation of this process is simplified with the following quantities assumed to be constant.

- Inlet velocities of gas and oil
- Temperature of gas and oil
- Pressure drop across the inlet port
- Heat flow from gas to oil (or from oil to gas)

Using the laws of mass energy conservation with the above assumptions, the states of gas and oil at the end of suction process can be analytically obtained as explicit formulae. Equation (1) has been developed from the analysis of suction process.

$$\begin{aligned}&-c_p M_{il} T_1^2 + T_1\left[c_p M_{t1} T_s + c_p M_{il} T_s - hA(T_{oil} - T_s)t_s\right]\\&-c_p M_{t1} T_s^2 = 0\end{aligned} \quad (1)$$

Solving the above quadratic equation, the temperature at the end of the suction process can be calculated. The inducted gas mass can be calculated after knowing the average temperature at the end of suction process.

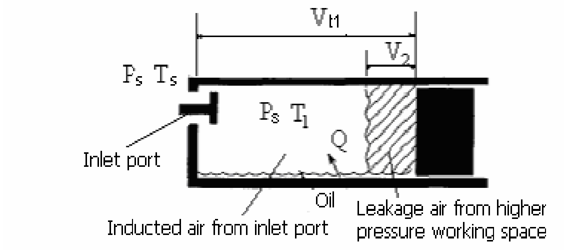


Figure 1 Model of the working space of an oil injected twin-screw compressor cavity at the end of the suction process [1]

Compression and Discharge Process

The process fluid in the compression chamber is compressed to higher pressures by the rotational movement of the rotors. Change in state of gas in the compression and discharge process is numerically simulated in the present work.
The following factors are taken into account in the model

1. volume change due to rotor rotation
2. mass and enthalpy flows of gas, entering or leaving the space through discharge port and leakage paths
3. mass and enthalpy flows of oil, entering or leaving the space through discharge port and leakage paths
4. heat exchange between gas and oil

To simplify the calculations, the following assumptions are made;

§ Gas and oil temperatures are homogeneous at any instant in the working space
§ Gas and oil never change phase
§ Pressure is uniform throughout the working space at any stage
§ Working gas is an ideal gas
§ Oil is an incompressible fluid
§ Heat exchange between gas and oil is in proportion to the temperature difference between them

Fujiwara and Osada [1] first derived the fundamental equations based on standard thermodynamic laws and the laws of perfect gas. The following equations are based on those derived in the reference [1] with modifications and other details introduced as needed.
The rate of change of process gas temperature during compression and discharge process is derived and written as

$$\begin{aligned}\frac{dT_g}{dt} &= -\frac{(k-1)T_g}{V_g}\left(\frac{dV}{dt}-\frac{P_b}{P}q_{li}+q_{l0}\right)\\&+\frac{m_{gi}}{M_g}(kT_{gb}-T_g)-\frac{m_{go}}{M_g}(k-1)T_g-\frac{hA}{c_v M_g}(T_g-T_l)\end{aligned} \qquad (2)$$

The first right-hand term of the above equation relates to the change in volume including the rate of leakage of oil volume. The second and third term represent the effect of gas leakage into and out of the compressor cavity respectively. The last term is derived from the heat transfer from gas to oil.

The rate of change of pressure is expressed as

$$\frac{dP}{dt}=\frac{1}{V_g}\begin{bmatrix}-kP\left(\frac{dV}{dT}+q_{l0}\right)+(kP_b-P_b+P)q_{li}\\+k\frac{T_{gb}PV_g}{T_g M_g}m_{gi}-k\frac{PV_g}{M_g}m_{g0}\\-\frac{PV_g hA}{c_v M_g T_g}(T_g-T_l)\end{bmatrix} \qquad (3)$$

The rate of change of net gas volume due to leakage is expressed as [1]

$$\frac{dV_g}{dt}=\frac{dV}{dt}-q_{li}+q_{lo} \qquad (4)$$

The rate of change of gas mass due to leakage

$$\frac{dM_g}{dt}=m_{gi}-m_{go} \qquad (5)$$

The oil mass also varies during compression process due to leakage

$$\frac{dM_l}{dt}=m_{li}-m_{lo} \qquad (6)$$

The rate of change of oil temperature is obtained from the energy balance in terms of leakage oil temperature and heat transferred from air:

$$\frac{dT_l}{dt}=(T_{lb}-T_l)\frac{m_{li}}{M_l}+\frac{hA}{M_l c_l}(T_g-T_l) \qquad (7)$$

The above sets of differential equations are adequate to simulate the compression and discharge processes.

Leakage rates of gas and oil

The gas and oil mass vary continuously due to leakage during compression process. The following set of equations give the rate of change of gas and oil mass.
The leakage of oil and gas mixture through leakage paths (except at lobe tip-housing clearance) is assumed to follow the process of a convergent nozzle. Fujiwara et.al. [3], [6] used the equation (8) to estimate the leakage of gas mass only for their oil free compressors. If the leakage of oil gas mixture is homogeneous, the oil gas mixture leakage may be written as

$$m=(m_g+m_l)=\frac{CA_cP_l}{\sqrt{T_l}}\sqrt{\frac{2\beta(r^{\frac{2}{\beta}}\quad r^{\frac{\beta+1}{\beta}})}{(\beta\quad 1)R_m}} \qquad (8)$$

for $r>\left(\frac{2}{\beta+1}\right)^{\frac{\beta}{\beta-1}}$

or

$$\mathrm{m} = \frac{CA_c P_1}{\sqrt{T_1}} \sqrt{\frac{\beta\left(\frac{2}{\beta - 1}\right)^{\frac{\beta+1}{\beta-1}}}{R_m}} \quad \text{for } r \le \left(\frac{2}{\beta + 1}\right)^{\frac{\beta}{\beta-1}}$$

Due to presence of oil in gas, exact determination of adiabatic index is rather difficult. The apparent ratio of specific heats of oil and gas mixture is calculated by the formula [1]

$$\beta = \frac{c_p + \varphi c_l}{c_v + \varphi c_l} \tag{9}$$

Similarly, the modified gas constant of the mixture is also calculated using the formula [1]

$$R_m = \frac{R}{1 + \varphi} \tag{10}$$

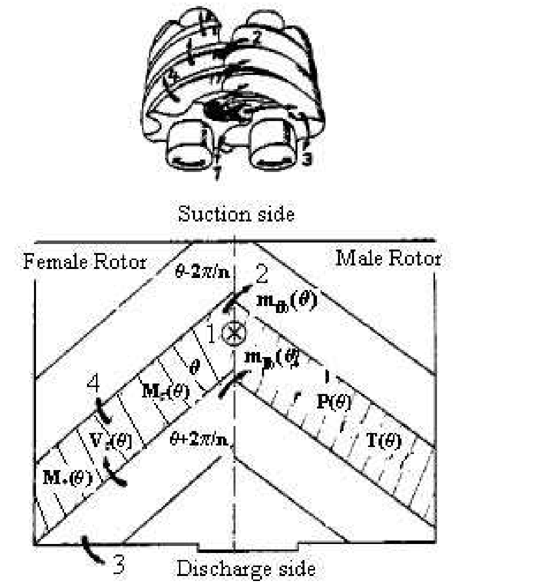
Sangafors [2] used the assumption that the gas and oil mixture leakage in oil injection compressors are homogeneous and equal to the mixture ratio at discharge port. Basing on this assumption, the mass ratio of oil to gas in the working volume as well as through all leakage paths is defined as

$$\varphi = \frac{M_l}{M_g} = \frac{m_l}{m_g} \tag{11}$$

The average leakage area is determined by multiplying sealing line length with an average gap (clearance) for each type of leakage. Normally, the average gap/clearance is determined from the actual clearance measurements in the compressor. The discharge or flow coefficients are empirically selected to account for the presence of oil in each instance of leakage flow.

At the lobe tip, the clearance fills with the oil due to the action of centrifugal force, and the oil leakage flow is in single phase. Therefore, the leakage flow rate of oil through this clearance can be calculated using the equation of incompressible viscous flow through a narrow gap [5].

$$m_{lr} = S\rho_l \left[\frac{V_t a}{2} - \frac{(P_1 - P_2)a^3}{12\mu_l w_t} \right] \tag{12}$$



1-Interlobe leakage flow, 2-Leakgage through trailing blowhole, 3-Leakage through discharge end clearance, 4-Leakage through rotor tip clearance, θ-Male rotor rotation angle

Figure 2 Overview of the different types of leakages through different gaps during compression process [4]

The leakage of gas mass in to the working chamber is through leading blowhole, and the clearance between leading cavity end and casing wall from both male and female rotors. Therefore, the total leakage gas flow rate in to the working volume is

$$m_{gi} = (m_{bi} + m_{dmi} + m_{dfi})/(1 + \varphi) \tag{13}$$

The leakage gas mass going out of the working volume during will take place through trailing blowhole, through interlobe clearance, and through clearance between lagging cavity end and casing wall of both male and female rotors. Hence, the total leakage of gas going out of the working volume is

$$m_{go} = (m_{bo} + m_{ico} + m_{dmo} + m_{dfo})/(1 + \varphi) \tag{14}$$

Similarly, the rate of oil mass leaking into the working volume is

$$m_{li} = \varphi m_{gi} + m_{lrmi} + m_{lrfi} \tag{15}$$

and the leakage rate of oil from the working volume to the succeeding groove is

$$m_{lo} = \varphi m_{go} + m_{lrmo} + m_{lrfo} \tag{16}$$

Equations (8) to (16) are adequate to calculate the rate of change of gas and oil mass during compression and discharge process.

Table 1 Rotor specifications of prototype compressor

Profile	Sigma
Combination of number of teeth	5+6
Outer diameter of male rotor (mm)	69.5
Outer diameter of female rotor(mm)	56
Wrap angle(degree)	300
Rotor length (mm)	75
Theoretical volume of one male and one female rotor groove (cm^3)	72.7
Built in volume ratio	5.2

Table 2 Operating conditions

Suction pressure (MPa)	0.1
Discharge pressure (MPa)	0.95
Rotational speed of male rotor (rpm)	4350
Suction gas temperature (K)	294.6
Supplied oil temperature (K)	343
Supplied oil rate (l/min)	23.4

Table 3 Flow coefficients at different leakage paths

Interlobe clearance	0.7
Lobe tip and casing bore	0.7
Leading blowhole	0.7
Lagging blowhole	0.6
Between rotor end and casing wall	0.4

Table 4 Clearances at different leakage paths and blowhole areas

Interlobe clearance (mm)	0.03
Clearance between lobe tip and casing bore(mm)	0.03
Clearance between rotor end and casing and wall(mm)	0.03
Leading blowhole area (mm^2)	2.8
Lagging blowhole area (mm^2)	2.8

Figure 3 A view of experimental setup

Heat Transfer Coefficient

To find the rate of heat transfer between gas and oil, the heat transfer coefficient between them is required. Fujiwara and Osada [1] determined the flow and heat transfer coefficients using experimental performance data. In this study, the same method is followed to find the flow and heat transfer coefficients.
The heat transfer coefficient between gas and oil

$$h = \frac{kP_s V_{t1}}{(k-1)At_s}\frac{d\eta_v}{dT_s} \tag{17}$$

Equation (17) relates "h" to the tangent of the η_v-T_s curve. Applying experimental test data to this equation, heat transfer coefficient is determined. Figure 4 shows typical volumetric efficiency curve plotted against inlet air temperature, in which the supplied oil temperature fixed at $70C^0$.The rise in volumetric efficiency at higher inlet temperature may be attributed mainly due to less temperature rise in the inducted air, as the oil temperature is fixed in this case.

However, there is no exact information available concerning the heat transfer area. The representative heat transfer area defined by Fujiwara and Osada [1]

$$A = V_{t1}^{\frac{2}{3}} \tag{18}$$

where

$$V_{t1} = (A_m + A_f)L$$

Efficiencies

Many standard efficiency definitions exist that qualify the mass flow and power performance characteristics of a compressor. Volumetric and power definitions are taken from standard thermodynamic textbooks and from reference [9].

Volumetric Efficiency

The theoretical volumetric efficiency based on the mass discharge

$$\eta_{tv} = \frac{M_{t1} - m_{gl}}{M_{ts}} \tag{19}$$

Experimental volumetric efficiency basing on experimental discharged mass flow rate is written as

$$\eta_{ev} = \frac{m_e}{m_t} \tag{20}$$

where,

$$m_t = (M_{ts} \times N_m \times n_m) / 60$$

Adiabatic Efficiency

The theoretical adiabatic efficiency is written as

$$\eta_{ta} = \frac{A_t}{A_i} \tag{21}$$

where

$$A_t = m_{ds} c_p T_s \left[(\frac{P_d}{P_s})^{\frac{k-1}{k}} - 1 \right]$$

The experimental adiabatic power can be interpreted as the power required to compress the gas adiabatically that produces the actual discharged mass flow rate at the given pressure ratio.

$$\eta_{ea} = \frac{A_e}{A_i} \tag{22}$$

where

$$A_e = m_e c_p T_s \left[(\frac{P_d}{P_s})^{\frac{k-1}{k}} - 1 \right]$$

The actual input power to the compressor system is measured with energy meter. This power consumption is inclusive of compressor cooling mechanism, transmission and electric motor losses. Mechanical losses are also included in calculating the adiabatic efficiency.

Results and Discussion

Variation of volumetric efficiency with inlet temperature is shown in Figure. 5 for a fixed supplied oil temperature. Volumetric efficiency with increasing inlet temperature is relatively more for helium than air. The rise in volumetric efficiency at higher temperature may be attributed mainly to a lower temperature rise in the inducted gas due to low amount of heat exchange with the oil. The variation of volumetric efficiency with discharge pressure is shown in Figure 6. It is observed that the variation of volumetric efficiency at higher discharge pressures is negligible. This is true because the screw compressor doesn't have any clearance volume at the end of compression process. The result of this analysis is similar to the results obtained by Osada and Fujiwara [1]. Figure. 7 show the effect of interlobe clearance on the P-V relationship for the 5-6 rotor profile. It shows that the pressure in the compression process decreases, as the interlobe clearance is increases. This is because of decreasing mass in the compression chamber due to more interlobe leakage. The interlobe clearance should be kept optimum to optimize the compressor performance.

The rise in temperature during compression of helium is more compared to that for air as shown in Figure 8. The reason for such a rise is attributed to high specific heat ratio. Figure 9 depicts the influence of oil injection quantity on volumetric efficiency. Oil will have less influence on compressing air and significant effect on helium application. The effect of rotational speed on volumetric efficiency is shown in Figure 10.It indicates that at higher lobe tip velocity, rotational speed has little effect on the volumetric efficiency for air and significant effect on helium compression. This is because at lower RPM, available leakage time is more. It is concluded that there is an optimum rpm for the best volumetric efficiency in case of both helium and air applications.

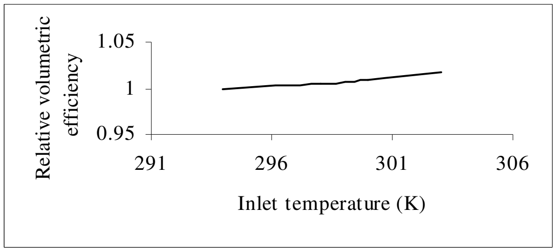


Figure 4 experimental volumetric efficiency curves for air against inlet temperature (efficiencies are relative to inlet temperature of 294K)

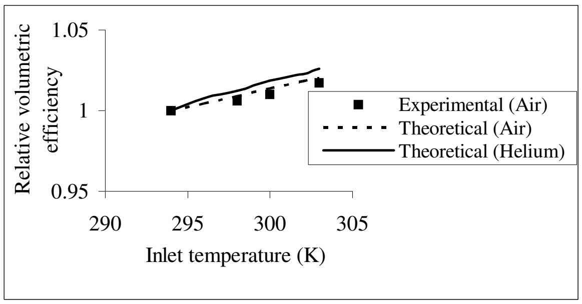


Figure 5 Variation of volumetric efficiency with inlet temperature of gas (efficiencies are relative to inlet temperature of 294K)

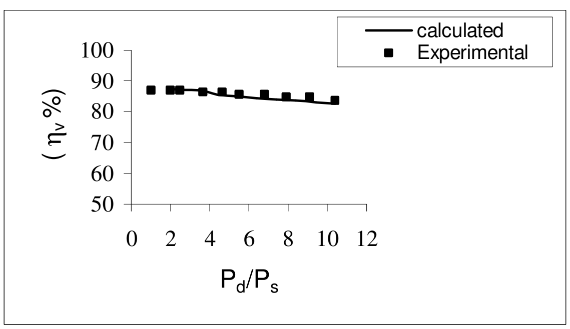


Figure 6 Variation of volumetric efficiency with discharge pressure (Air)

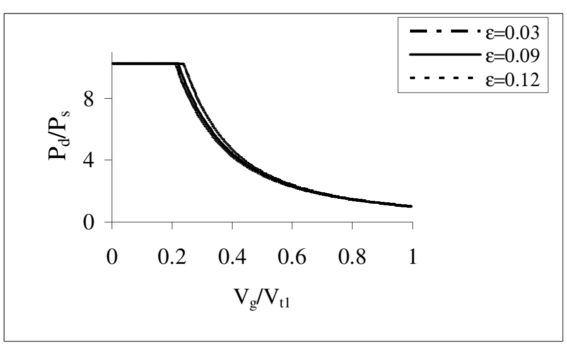


Figure 7 Effect of interlobe clearance on P-V diagram profile

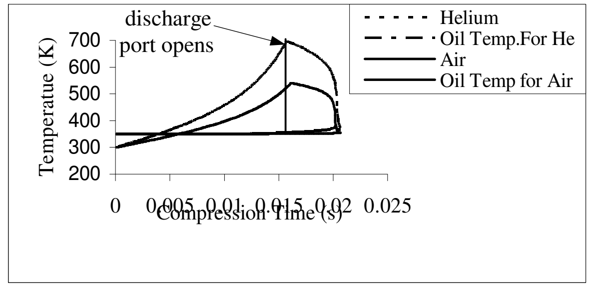


Figure 8 Variation of gas and oil temperatures during compression and discharge process

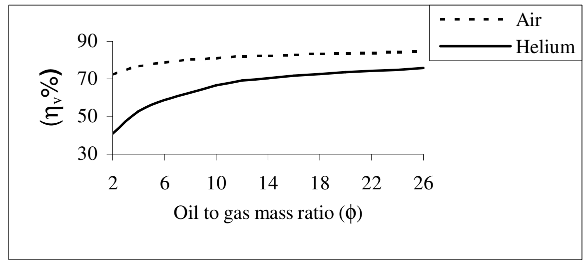


Figure 9 Variation of volumetric efficiency with oil to gas mass ratio

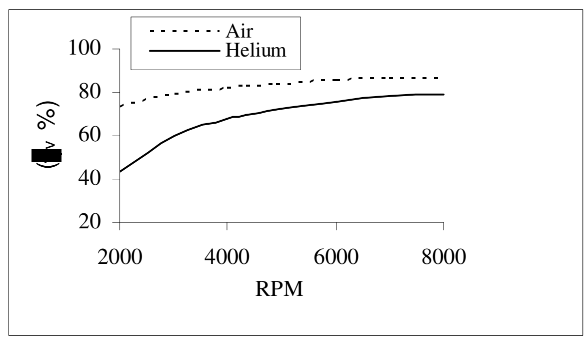


Figure 10 Variation of volumetric efficiency with rotational speed

Conclusions

A numerical model of the oil injected duel screw compressor has been developed covering both suction and compression-discharge steps for evaluating the performance. Heat transfer coefficient between gas and oil has been determined from experimental observations on a commercial compressor in which volumetric efficiency decreases with decreasing inlet temperature. Flow coefficients also have been determined from experimentally obtained compressor efficiencies. Some quantitative results have been obtained to convert an oil injected air compressor for helium compression.

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