Journal of Applied Thermal Engineering. Volume 30, Issue 4, March 2010, Pages 327-339
DOI- http://dx.doi.org/10.1016/j.applthermaleng.2009.09.010

Theoretical and Experimental Studies on Oil Injected Twin Screw Air Compressor When Compressing Different Light and Heavy Gases

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Abstract

Oil injected twin-screw compressors are widely used for medium pressure applications in many industries. Low cost air compressors can be adopted for compression of helium and other gases, leading to significant cost saving. The efficiency, delivery rate of the compressors (medium and small size) has been analyzed and presented in the study. The heat transfer coefficient required to generate the theoretical efficiencies has been determined experimentally. To generate machine independent experimental data, two similar compressors with different capacities have been built to test the performance of air compressors when applied to compress nitrogen, argon and helium gases apart from air. Also this paper addresses the gas delivery rate and heat of compression (temperature) on volumetric and power efficiency.
(**Key words** :(screw compressor, gas delivery rate, oil injection temperature, working gas, volumetric efficiency, and adiabatic efficiency)

Nomenclature

C_p	Specific heat of gas at constant pressure	(J/kg.K)
k	Ratio of specific heats	(dimensionless)
M_1	Mass of fresh gas admitted into the suction cavities during suction process	(kg)
M_{t1}	Mass of gas in a pair of male and female cavities at the end of suction process at (P_s , T_1)	(kg)
M_{ts}	Theoretical gas mass in a pair of male and female cavities at suction condition (P_s , T_s)	(kg)
M_{il}	Interlobe leakage gas mass leaked into the suction chamber during previous compression process at (P_s ,T_1)	(kg)

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$\dot{m}_e$	Actual discharged gas mass rate at (P_s ,T_1)	(kg/s)
$\dot{m}_t$	Theoretical inducted gas mass rate at (P_s ,T_1)	(kg/s)
$\dot{m}_{dt}$	Theoretical discharged gas mass rate at (P_s ,T_1)	(kg/s)
$\dot{m}_{gl}$	Net gas mass leakage rate from a pair of male and female cavities during compression and discharge process	(kg/s)
N_m	Rotational speed of male rotor	(rps)
N_e	Energy meter constant	(dimensionless)
n_m	Number of lobes on male rotor	(dimensionless)
n_e	Number of revolutions of energy meter in time t_e	(dimensionless)
P_d	Discharge pressure	(N/m^2)
P_r	Reference pressure of gas during rotameter calibration	(N/m^2)
P_s	Suction pressure	(N/m^2)
R	Gas constant	(J/kg.K)
t_e	Time taken for n_e revolutions of energy meter	(s)
t_s	Time required for suction process	(s)
T_1	Temperature of the gas at the end of suction process	(K)
T_{li}	Injected oil temperature	(K)
T_s	Gas temperature at suction condition	(K)
U_{RSS}	Combined uncertainty	(dimensionless)
V_m	Volume flow rate of gas	(m^3/min)
V_{t1}	Geometrical volume of a pair of male and female grooves	(m^3)
W_{ta}	Theoretical adiabatic power of the compressor	(W)
W_{ea}	Experimental adiabatic power of the compressor	(W)
$W_{sys,e}$	Actual input power to the compressor system	(W)

Greek Symbols

η_a	Adiabatic efficiency	(dimensionless)
η_v	Volumetric efficiency	(dimensionless)

η_{tv}	Theoretical volumetric efficiency at (P_s , T_s)	(dimensionless)
η_{ev}	Experimental volumetric efficiency	(dimensionless)
η_{ta}	Theoretical adiabatic efficiency at (P_s , T_s)	(dimensionless)
η_{ea}	Experimental adiabatic efficiency	(dimensionless)
ρ_s	Density of gas at suction condition	(kg/m^3)

1. Introduction

The twin-screw compressor is a positive displacement machine that uses a pair of intermeshing rotors housed in a suitable casing to produce compression. These are capable of high-speed operation over a wide range of operating pressures. In screw machines, oil is deliberately injected into the compression chambers to provide sealing, lubrication, corrosion resistance and cooling effect. Screw compressor performance is influenced both by the refrigerant and by the type of lubricant used. Thermodynamic efficiency of the compression process also depends greatly on the heat exchange between gas and the oil.

A mathematical model has been developed by the authors [1-2] on the basis of the laws of perfect gas and standard thermodynamic relations for calculating the compressor performance. Fujiwara *et al*. [3] determined the heat transfer coefficient experimentally and used in numerical simulation. Stosic *et al.* [4-6] have developed mathematical modelling and experimental investigation on oil injection and its influence on thermodynamic process in twin-screw compressors. Peng *et al.* [7] investigated the oil injection phenomenon to get a better understanding of compressor performance experimentally. Hammerl *et al.* [8] examined several variants of the oil injection technique into the suction port of the compressor. Sangfors [9] studied the effect of oil injection parameters on compressor performance. Recently, Depaepe *et al*. [10] have built a test rig to asses the performance of different types of atomizers for oil atomization in twin-screw compressor.

2. Measurement of Volumetric And Adiabatic Efficiencies

Screw compressor performance is governed by the interactive effects of thermodynamics, heat transfer and machine geometry. This can be calculated reliably only by their simultaneous consideration. The performance index of a compressor is characterized by its volume or mass handling capacity and specific power consumption. The theoretical volumetric and adiabatic efficiencies of twin-screw compressor are estimated after computing the leakage flow rates of gas and oil [1]. The efficiency definitions are taken from standard thermodynamic textbooks and from reference [11].

Volumetric Efficiency

The volumetric efficiency can be defined either in terms of volume flow rate or in mass flow rate terms to yield the same value. Using the ideal gas equation of state, the total amount of gas mass, including leakage gas occupying the suction volume at (P_s ,T_s) calculated by the expression:

$$M_{ts} = \frac{P_s V_{t1}}{R T_s} \tag{1}$$

The actual gas mass contained in the suction volume at the end of suction process at (P_s ,T_1) can be estimated by the relation:

$$M_{t1} = M_{ts} \frac{T_s}{T_1} \tag{2}$$

The theoretical gas flow rate at temperature T_1 over a suction duration (t_s) is:

$$\dot{m}_t = \frac{M_{t1}}{t_s} \tag{3}$$

But the total gas mass at condition (P_s ,T_1) in the suction cavities at the end of suction process is the sum of fresh charge inducted and the leaked gas during previous compression. Thus:

$$M_{t1} = M_1 + M_{il} \tag{4}$$

The gas leakage into the suction cavities over the suction duration is estimated by the relation[1]:

$$M_{il} = \dot{m}_{gl} \times t_s \tag{5}$$

The leakage gas mass rate is given as [1]:

$$\dot{m}_{gl} = \dot{m}_{gi} - \dot{m}_{go} \tag{6}$$

The theoretical discharged gas flow rate is estimated by the relation:

$$\dot{m}_{dt} = \dot{m}_t - \dot{m}_{gl} \tag{7}$$

Hence, the theoretical volumetric efficiency in terms of mass flow rate is calculated by using the relation:

$$\eta_{tv} = \frac{\dot{m}_{dt}}{\dot{m}_{ts}} \tag{8}$$

In general, the experimental gas flow rate is less than the theoretical inducted mass rate due to imperfect nature of ports, wall friction and other frictional losses apart from leakage loses. Experimental volumetric efficiency is calculated after measuring the actual gas flow rate as shown below:

$$\eta_{ev} = \frac{\dot{m}_{e}}{\dot{m}_{ts}} \tag{9}$$

The actual gas volume flow rate is measured with a rotameter, which has been calibrated for volume flow rate at reference condition (P_r , T_r). The gas flow rate of compressor at condition (P_s , T_1) is obtained by using the formula [12] :

$$\dot{m}_{e} = \rho_s V_m \sqrt{\frac{P_s \times T_r}{P_r \times T_s}} \tag{10}$$

where ρ_s is the density of gas at (P_s , T_1) and V_m is the volume flow rate of the gas. The gas density has been taken from standard thermodynamic tables [13].

Adiabatic Efficiency

The actual power input to the compressor for only gas compression will be calculated from the area of the indicator diagram. The area of the indicator diagram is the actual power for compression work, which is obtained from an experimentally measured p-v curve. This curve may be obtained through difficult and expensive process of conducting compressor tests with pressure transducers located with in the pair of cavities. To overcome this difficulty, the efficiency is defined in terms of system power, which can be measured reasonably accurately by an energy meter. The system power is the power required for an entire compression system and it is some of the shaft power and several additional power requirements due to the presence of controllers and other peripheral systems.

The overall theoretical adiabatic efficiency is defined as:

$$\eta_{ta} = \frac{\dot{W}_{ta}}{\dot{W}_{sys,e}} \tag{11}$$

The theoretical adiabatic compression work is given by the expression:

$$\dot{W}_{ta} = \dot{m}_{dt} c_p T_s \left[\left(\frac{P_d}{P_s} \right)^{\frac{k-1}{k}} - 1 \right] \tag{12}$$

The power consumed by the compression system is measured with energy meter. The actual power consumed by the compressor is inclusive of that consumed by cooling fan of the compressor, transmission and electric motor losses, as well as mechanical losses. The power consumed by the compression system is calculated by the relation:

$$\dot{W}_{sys,e} = \frac{3600 \times n_e}{N_e \times t_e} \tag{13}$$

where N_e is the energy meter constant, and t_e is the time taken for n_e revolutions of energy meter disc.

The experimental adiabatic power can be interpreted as the power required to compress the gas adiabatically that produces the actual discharged gas flow at a given pressure ratio. The experimental adiabatic efficiency is defined as:

$$\eta_{ea} = \frac{\dot{W}_{ea}}{\dot{W}_{sys,e}} \tag{14}$$

The experimental adiabatic compression work is given by the expression:

$$\dot{W}_{ea} = \dot{m}_e c_p T_s \left[\left(\frac{P_d}{P_s} \right)^{\frac{k-1}{k}} - 1 \right] \tag{15}$$

3. Experimental Studies

Before the compressor put into a refrigerator or liquefier system, it needs to be tested independently for the energy conversion efficiency during compression process. The experimental study has also been aimed at to study the influence of machine capacity on efficiency. Study of specific power per unit delivery of gas of the compressor at different pressure ratios is also essential, when the compressor designed for air is applied to compress different low molecular weight and higher molecular weight gases. The Photograph of 5.5kW experimental test rig has shown in Figure 1. The experimental setup for medium size compressor has been scattered in three rooms because of its bulkiness. Partial setup has been photographed and shown in Figure 2. The photograph of screw rotors of 5.5 and 37kW coompressors and their casings have shown in Figure 3.

Figure 1: A photograph of 5.5kW experimental test rig

Figure 2: A Photograph of 37kW experimental test rig (partial)

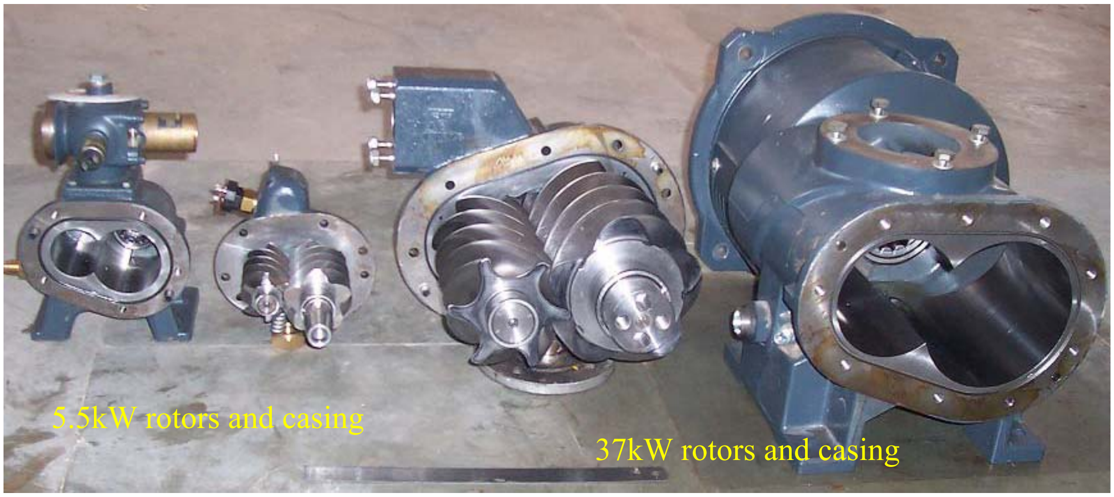

Figure 3: Photograph of 5.5 kW and 37 kW compressor rotors and their casings.

The main motive of the present test is to study the performance of the oil injected twin-screw air compressor when compressing different gases such as helium, argon and nitrogen gases apart from air. Two test benches have been installed in the laboratory, which are identical in all aspects except the compressor capacities. The specifications of small and medium size compressors have been shown in Table 1 and 2.

Table 1: **Rotor specifications of 5.5 kW and 37kW prototype compressors**

Name of constant	**5.5kW compressor**	**37kW compressor**	**Units**
Number of lobes on the male rotor	5	5	none
Number of lobes on the female rotor	6	6	none
Male wrap angle	300	300	degree
Diameter of male rotor	72	152	mm
Diameter of female rotor	54	118	mm
Length of the rotor	90	235	mm
Cross sectional area of male rotor groove	210	1075	mm^2
Cross sectional area of female rotor groove	170	1020	mm^2
Lobe tip width of male rotor	0.5	0.9	mm
Lobe tip width of female rotor	1.8	2.6	mm
Rotational speed of male rotor	4350	2950	rpm

Table 2: Clearances and Sealing line lengths obtained from experiment/actual measurements of 5.5 kW and 37kW compressors.

Constants	***5.5kW***	***37kW***	***Unit***
Inter lobe clearance	0.027	0.035	mm
Rotor tip-housing clearance	0.03	0.03	mm
Clearance between rotor and discharge end face plate	0.03	0.029	mm
Leading blowhole area	2	3.3	mm^2
Trailing blowhole area	2	3.3	mm^2
Interlobe sealing line length	36	74	mm
Sealing line length of rotor tip housing of male rotor	180	460	mm
Sealing line length of rotor tip housing of female rotor	143	345	mm
Sealing line length at leading discharge end face of male rotor	18	34.5	mm
Sealing line length at leading discharge end face of female rotor	16	30	mm
Sealing line length at lagging discharge end face of male rotor	28	62.5	mm
Sealing line length at lagging discharge end face of female rotor	20	44	mm

4. Error analysis

The error in the measurement of volumetric and adiabatic efficiencies has been estimated from the measurement of individual variables for one set of data. All the individual variables which influence the result (volumetric and adiabatic efficiencies) have been measured N times and the associated precision error $S_{\bar{x}}$ along with other variables is shown in Table 3. Bias and random errors in the measurement of volumetric and adiabatic efficiencies have been calculated using the values listed in table 3. The table is specific for calculation of volumetric and adiabatic efficiency. This error estimation is extended to yield an average error in the experimental procedure for the determination of volumetric and adiabatic efficiency.

Table 3: Average values and their error values for N samples.

Parameter	Average value ($\bar{X}$)	Bias error (B_r)	Precision error ($S_{\bar{x}}$)	Sample Size (N)
Suction pressure (P_s) bar	1.113	0.015	0.023	6
Suction Temperature(T_s) ^0C	308	0.005	0.3	6
Discharge pressure(P_d) bar	9.65	0.015	0.25	6
Speed of male rotor(N_m) rpm	4350	15	20	6
Volume flow rate of gas(V_m) L/min	675	25	12	6
Volume of a pair cavities(V_{t1}) cm^3	342	0.5	1	6
No.of revolutions of energy meter(n_e)	5			
Time for n_e rotations of energy meter disc (t_e) sec	38	0.05	0.2	6
Experimental mass flow rate(m_e) kg/s	0.0139	0.1	0.5	6
Theoretical mass flow rate(m_t) kg/s	0.0156	-	-	-
Experimental adiabatic power(W_{ea}) kW	3.642	-	-	-
Experimental system power(W_{sys}) kW	6.3	-	-	-

4.1. Uncertainty in Determination of Volumetric Efficiency

Bias Error:

Gas flow rate has been measured with rotameter, which has been calibrated for air. The gas density and gas characteristic constants have been taken from standard thermodynamic tables [13]. The detailed error analysis has been explained in [14]. The volumetric efficiency equation is rewritten as:

$$\eta_{ev} = \frac{\dot{m}_e}{\dot{m}_{ts}} \tag{16}$$

The bias in the measurement of volumetric efficiency can be estimated by using the relation:

$$\left(\frac{\Delta\eta_{ev}}{\eta_{ev}}\right)^2 = \left(\frac{\Delta\dot{m}_e}{\dot{m}_e}\right)^2 + \left(\frac{\Delta\dot{m}_{ts}}{\dot{m}_{ts}}\right)^2 \tag{17}$$

The mass flow rate of the compressor at suction condition can be obtained by using the formula:

$$\dot{m}_e = \rho_s V_m \sqrt{\frac{P_s \times T_r}{P_r \times T_s}} \tag{18}$$

The bias in the measurement of mass flow rate is estimated by using the above formula:

$$\left(\frac{\Delta \dot{m}_e}{\dot{m}_e}\right)^2 = \left(\frac{\Delta V_m}{V_m}\right)^2 + \left(\frac{1}{2}\frac{\Delta P_s}{P_s}\right)^2 + \left(\frac{1}{2}\frac{\Delta T_s}{T_s}\right)^2 \qquad (19)$$

The bias in the experimental mass flow rate is calculated to be:

$$\left(\frac{\Delta \dot{m}_e}{\dot{m}_e}\right)^2 = 0.00142 \qquad (20)$$

The theoretical mass flow rate is calculated using the equation of state for perfect gas at the same condition and is given by the equation:

$$\dot{m}_{ts} = \frac{P_s V_{t1}}{RT_s} n_m N_m \qquad (21)$$

The bias in the measurement of theoretical mass flow rate is calculated using the relation:

$$\left(\frac{\Delta \dot{m}_{ts}}{\dot{m}_{ts}}\right)^2 = \left(\frac{\Delta V_{t1}}{V_{t1}}\right)^2 + \left(\frac{\Delta P_s}{P_s}\right)^2 + \left(\frac{\Delta T_s}{T_s}\right)^2 + \left(\frac{\Delta N_m}{N_m}\right)^2 = 0.0004073 \qquad (22)$$

Substituting the values from equations (20) and (22) in equation (17), the bias error in the measurement of volumetric efficiency is:

$$\left(\frac{\Delta \eta_{ev}}{\eta_{ev}}\right) \times 100 = \pm 4.27\%$$

Precession Error

The randomness in the determination of volumetric efficiency can be written as:

$$\left(\frac{\Delta \eta_{ev}}{\eta_{ev}}\right)^2 = \left(\frac{\Delta \dot{m}_e}{\dot{m}_e}\right)^2 + \left(\frac{\Delta \dot{m}_{ts}}{\dot{m}_{ts}}\right)^2 \qquad (23)$$

The precision error in the measurement of experimental mass flow rate is estimated from the relation:

$$\left(\frac{\Delta \dot{m}_e}{\dot{m}_e}\right)^2 = \left(\frac{\Delta V_m}{V_m}\right)^2 + \left(\frac{1}{2}\frac{\Delta P_s}{P_s}\right)^2 + \left(\frac{1}{2}\frac{\Delta T_s}{T_s}\right)^2 = 0.000142 \qquad (24)$$

The precision error in the measurement of theoretical mass flow rate is given by:

$$\left(\frac{\Delta \dot{m}_{ts}}{\dot{m}_{ts}}\right)^2 = \left(\frac{\Delta V_{t1}}{V_{t1}}\right)^2 + \left(\frac{\Delta P_s}{P_s}\right)^2 + \left(\frac{\Delta T_s}{T_s}\right)^2 + \left(\frac{\Delta N_m}{N_m}\right)^2 = 0.00386 \qquad (25)$$

Substituting the values of $\left(\frac{\Delta \dot{m}_e}{\dot{m}_e}\right)$ and $\left(\frac{\Delta \dot{m}_{ts}}{\dot{m}_{ts}}\right)$ from equations (24) and (25) in equation (23), the precision error in volumetric efficiency measurement is given as shown below:

$$\left(\frac{\Delta\eta_{ev}}{\eta_{ev}}\right)^2 = \pm 6.33\%$$

The number of degrees of freedom υ_r associated with combined precision P_r is found by using the Welch-Satterthwaite formula [15].

$$\upsilon_r = \frac{\left[\sum_{i=1}^{J}(\theta_i P_i)^2\right]^2}{\sum_{i=1}^{J}\left[(\theta_i P_i)^4 / \upsilon_i\right]} = \frac{P_r^4}{\sum_{i=1}^{J}\left[(\theta_i P_i)^4 / \upsilon_i\right]} = 3.98 \approx 4$$

From t-distribution table [15] for the degree of freedom 4, the t-value at 95% confidence level is 2.776. The combined uncertainty in the measurement of volumetric efficiency is estimated by using the equation [15] given below:

$$U_{RSS} = \left[B_r^2 + (t \times P_r)^2\right]^{\frac{1}{2}} \tag{26}$$

$$U_{RSS} = [(4.27)^2 + (2.776 \times 6.33)^2]^{\frac{1}{2}} = 18.08\%$$

Therefore, the overall uncertainty in the measurement of volumetric efficiency is:

$$\left(\frac{\Delta\eta_{ev}}{\eta_{ev}}\right) = \pm 18.08\%$$

4.2 Uncertainty in Determination of Adiabatic Efficiency

Bias Error:

The bias in the measurement of adiabatic efficiency has also been computed in the similar fashion as that of volumetric efficiency. The experimental adiabatic efficiency is rewritten as:

$$\eta_{ea} = \frac{\dot{W}_{ea}}{\dot{W}_{sys,e}} \tag{27}$$

The bias in adiabatic efficiency determination can be written as:

$$\left(\frac{\Delta\eta_{ea}}{\eta_{ea}}\right)^2 = \left(\frac{\Delta\dot{W}_{ea}}{\dot{W}_{ea}}\right)^2 + \left(\frac{\Delta\dot{W}_{sys,e}}{\dot{W}_{sys,e}}\right)^2 \qquad (28)$$

The experimental adiabatic efficiency is:

$$\dot{W}_{ae} = \dot{m}_e c_p T_s \left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}} - 1\right] \qquad (29)$$

The bias in the measurement of experimental adiabatic power is obtained by the relation:

$$\left(\frac{\Delta\dot{W}_{ea}}{\dot{W}_{ea}}\right)^2 = \left(\frac{\Delta\dot{m}_e}{\dot{m}_e}\right)^2 + \left(\frac{\Delta T_s}{T_s}\right)^2 + \left(\frac{(k-1)}{k}\times\frac{\Delta P_d}{P_d}\right)^2 + \left(\frac{(k-1)}{k}\times\frac{\Delta P_s}{P_s}\right)^2 = 0.001694 \qquad (30)$$

The power consumed by the compressor system is rewritten as:

$$\dot{W}_{sys,e} = \frac{3600\times n_e}{N_e\times t_e} \qquad (31)$$

The bias associated with measurement of the system power is:

$$\left(\frac{\Delta\dot{W}_{sys,e}}{\dot{W}_{sys,e}}\right)^2 = \left(\frac{\Delta n_e}{n_e}\right)^2 + \left(\frac{\Delta t_e}{t_e}\right)^2 = 0.000107 \qquad (32)$$

Substituting the values from equations (30) and (32) in equation (28), the bias in adiabatic efficiency measurement is shown below:

$$\left(\frac{\Delta\eta_{ea}}{\eta_{ea}}\right)\times 100 = \pm\,4.24\%$$

Precession Error

The randomness in the measurement of adiabatic efficiency can also be computed following the above procedure. The precision error in the measurement of adiabatic efficiency is:

$$\left(\frac{\Delta\eta_{ea}}{\eta_{ea}}\right)^2 = \left(\frac{\Delta\dot{W}_{ea}}{\dot{W}_{ea}}\right)^2 + \left(\frac{\Delta\dot{W}_{sys,e}}{\dot{W}_{sys,e}}\right)^2 \qquad (33)$$

where

$$\left(\frac{\Delta\dot{W}_{ea}}{\dot{W}_{ea}}\right)^2 = \left(\frac{\Delta\dot{m}_e}{\dot{m}_e}\right)^2 + \left(\frac{\Delta T_s}{T_s}\right)^2 + \left(\frac{(k-1)}{k}\times\frac{\Delta P_d}{P_d}\right)^2 + \left(\frac{(k-1)}{k}\times\frac{\Delta P_s}{P_s}\right)^2 = 0.000232 \qquad (34)$$

The precision error associated with measurement of the system power is:

$$\left(\frac{\Delta \dot{W}_{sys,e}}{\dot{W}_{sys,e}}\right)^2 = \left(\frac{\Delta n_e}{n_e}\right)^2 + \left(\frac{\Delta t_e}{t_e}\right)^2 = 0.00177 \tag{35}$$

Substituting the values of $\left(\frac{\Delta \dot{W}_{ea}}{\dot{W}_{ea}}\right)$ and $\left(\frac{\Delta \dot{W}_{sys,e}}{\dot{W}_{sys,e}}\right)$ from equations (34) and (35) in equation (33), the precision error in measuring adiabatic efficiency is:

$$\left(\frac{\Delta \eta_{ea}}{\eta_{ea}}\right) \times 100 = 4.5\%$$

The number of degrees of freedom υ_r associated with combined precision P_r is also found by using the below equation:

$$\upsilon_r = \frac{\left[\sum_{i=1}^{J} (\theta_i P_i)^2\right]^2}{\sum_{i=1}^{J}\left[(\theta_i P_i)^4 / \upsilon_i\right]} = \frac{P_r^4}{\sum_{i=1}^{J}\left[(\theta_i P_i)^4 / \upsilon_i\right]} = 13.98 \approx 14$$

From t-distribution table using degrees of freedom value14, the t-value at 95% confidence level is 2.145 [15]. The combined uncertainty in the measurement of adiabatic efficiency using equation is:

$$U_{RSS} = \left[B_r^2 + (t \times P_r)^2\right]^{\frac{1}{2}}$$

$$U_{RSS} = [(4.24)^2 + (2.145 \times 4.5)^2]^{\frac{1}{2}} = 10.54\%$$

Hence, the overall uncertainty in the measurement of adiabatic efficiency is:

$$\left(\frac{\Delta \eta_{ev}}{\eta_{ev}}\right) = \pm 10.54\%$$

5. Discussion

Theoretical and experimental analysis has been conducted on commercially available twin-screw air compressors of two different capacities using air, nitrogen, argon and helium as working gases. Figure 4 gives the influence of inducted gas temperature on volumetric efficiency with different working gases at a fixed pressure ratio and injected oil temperature in the 5.5 kW compressor. It is observed that the volumetric efficiency increases with inlet temperature. This effect is attributed to the fact that the temperature rise of fresh gas mass at higher induction gas temperature during suction process is less. This led to the minimum disturbance in suction cavities during suction process and hence increases in volumetric efficiency.

Figure 5 gives the variation of volumetric efficiency with suction gas pressure at a fixed pressure ratio in the 5.5kW compressor. The increase in volumetric efficiency with increase in suction pressure may be assigned to the increase in volume flow rate of gas without proportionate increase in oil flow rate. Most of the leakage during compression process is through interloped clearance. The amount of leakage gas mass depends on pressure difference between suction cavity and discharge end. If the suction cavity pressure increases, the pressure difference decreases and hence the volumetric efficiency increases. The comparison of volumetric efficiency under similar conditions has shown in Figure 6. It has been observed that the volumetric efficiency of small compressor is slightly more than the big compressor for almost all the gases. This is attributed to be the manufacturing deviations of the machine rotors.

The variation of volumetric efficiency with pressure ratio is shown in Figure 7. This variation is due to the fact that increase in pressure ratio (i.e discharge pressure) enhances the leakage and thus degrades the efficiency. For the same pressure ratio and the same oil injection temperature, volumetric efficiency is the lowest for helium followed by argon, nitrogen and air. This may be due to the fact that helium being a light gas and leaks more easily. On the other hand, monatomic gas generates higher heat of compression leading to decrease in oil viscosity and consequently higher leakage rate. This is the case with argon.

The theoretical and experimental gas delivery rate of the 5.5kW compressor for different working gases is shown in Figure 8. Experiments have been conducted at two different injected oil temperatures on 5.5 kW compressor, keeping the inlet gas pressure constant. The gas delivery rates have been compared at two different oil injection temperatures and presented the results in Figure 9. It is observed that the gas volume delivery rate decreases with increase in pressure ratio and with increase of oil injection temperature. The causes are the increase of leakage rate at higher pressures and reduction of inducted volume at higher temperature. These figures also show the relative variation of volume flow rates of different working gases. The lowest gas delivery rate is with helium followed by argon, nitrogen and air.

Influence of inducted gas temperature on adiabatic efficiency with different working gases is shown in Figures 10. The nature of variation with inducted gas temperature is similar for all the working gases. The increase of adiabatic efficiency for a particular gas species is due to the increase in volumetric efficiency which decreases the actual amount of work consumed per unit mass of gas. Adiabatic efficiency is highest for argon followed by air, helium and nitrogen. The variation of adiabatic efficiency with suction pressure is shown in Figures 11. The adiabatic efficiency is increasing with increase in suction pressure. This is due to the reduction in interloped leakage rate because of reduction in pressure difference.

The relation between adiabatic efficiency and pressure ratio for all the working gases at a fixed injected oil temperature on both the compressors has been shown in Figures 12. It has been observed from the results that the nature of variation is similar for all the working gases. It may be observed from the figures that adiabatic efficiency curves show maximum values at different pressure ratios for different working gases. This is due to the fact that adiabatic efficiency increases with pressure ratio and volumetric efficiency decreases with pressure ratio. At higher pressure ratios, there is decrease in mass flow due to fall in volumetric efficiency, which in turn lowers the adiabatic efficiency. The specific power consumption of the compressors against pressure ratio is shown in Figures 13 to 15. It may be observed that the power consumption is highest with helium followed by argon, nitrogen and air. Helium has the highest specific power consumption in mass terms due to higher work of compression for a lower volume flow rate, as well as its low density.

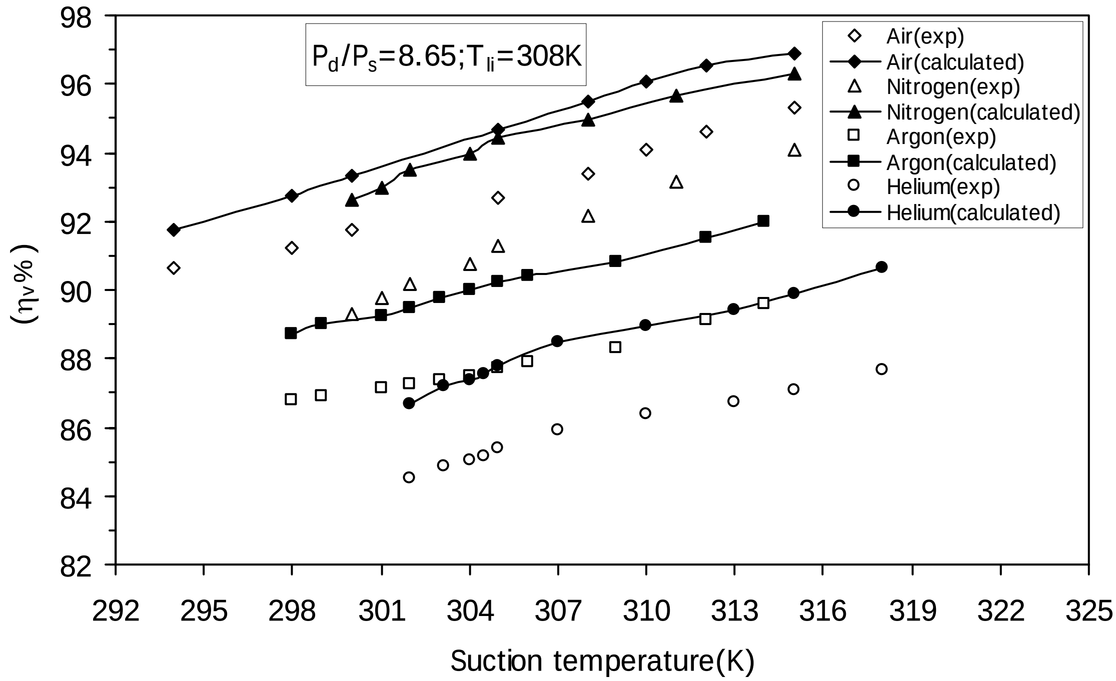

Figure 4: Influence of inlet gas temperature on volumetric efficiency with different working gases of 5.5 kW compressor at fixed injected oil temperature

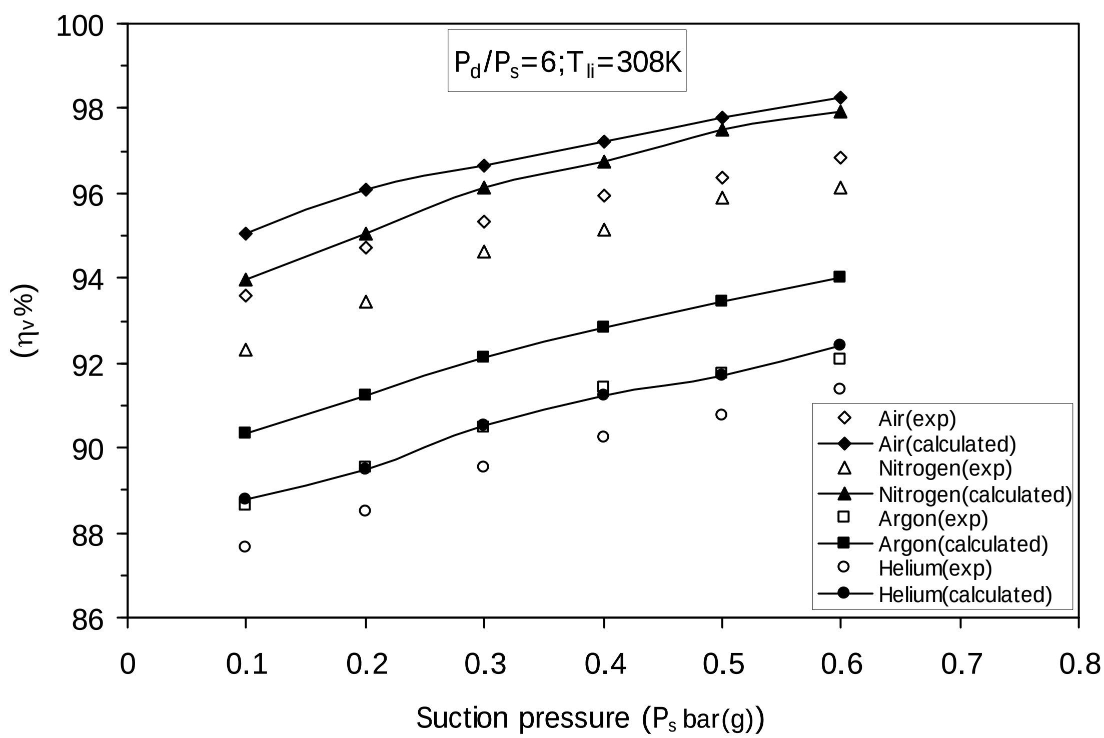

Figure 5: Effect of suction pressure on volumetric efficiency with different working gases of 5.5 kW compressor at fixed injected oil temperature.

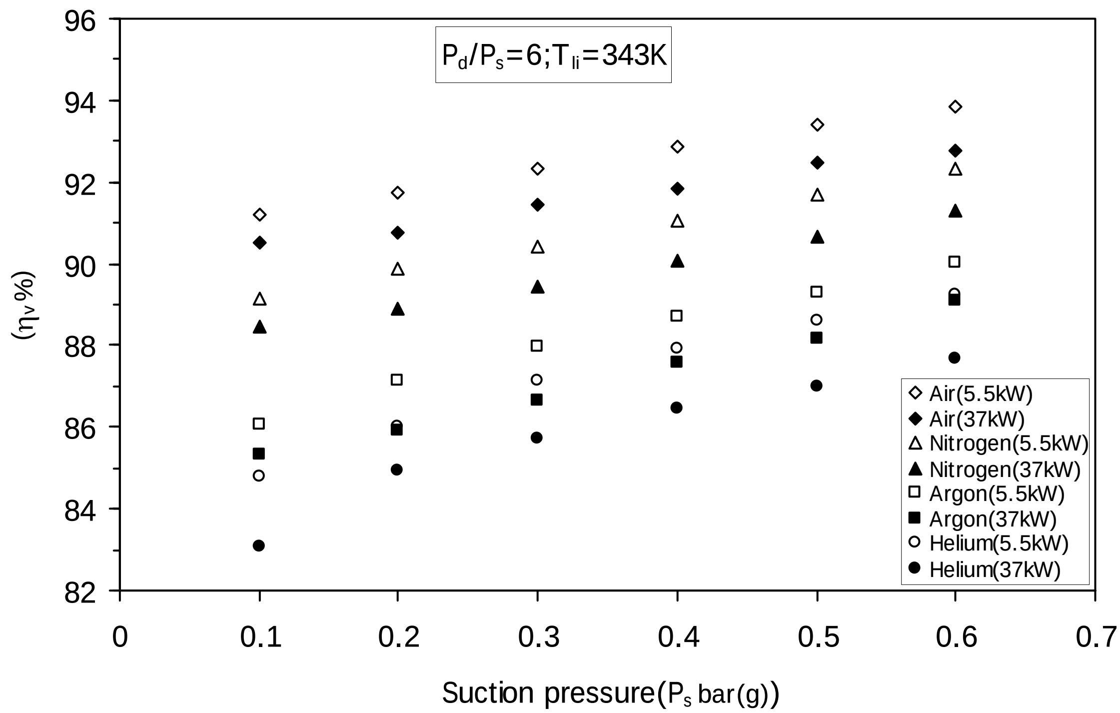

Figure 6: Comparison of experimentally measured volumetric efficiencies of 5.5 & 37 kW compressors at fixed injected oil temperature.

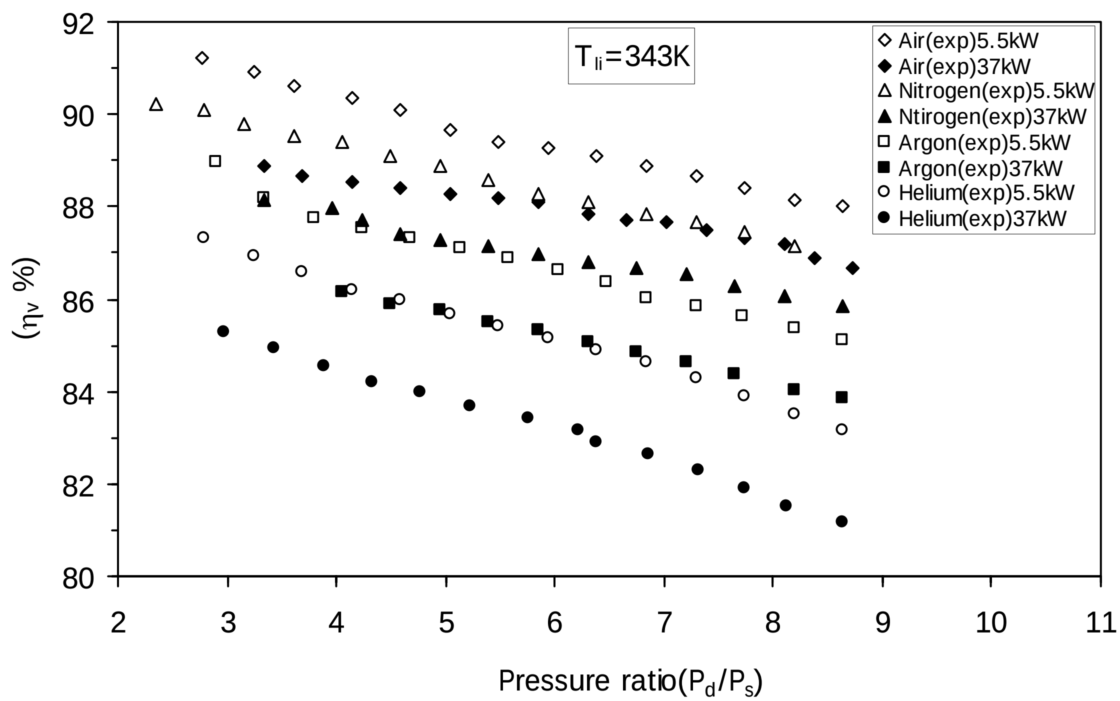

Figure 7: Variation of experimentally measured volumetric efficiency of 5.5 kW & 37 kW compressors with pressure ratio at fixed injected oil temperature and at constant suction pressure.

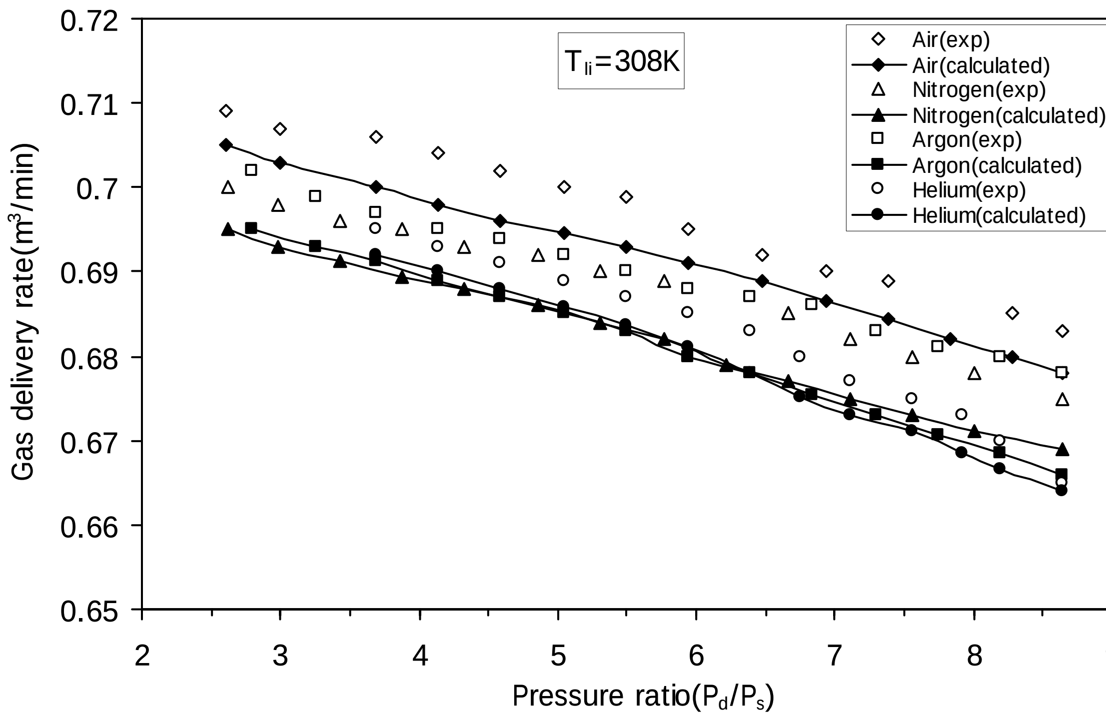

Figure 8: Variation of gas flow rate in 5.5 kW compressor with pressure ratio at fixed injected oil temperature and suction pressure.

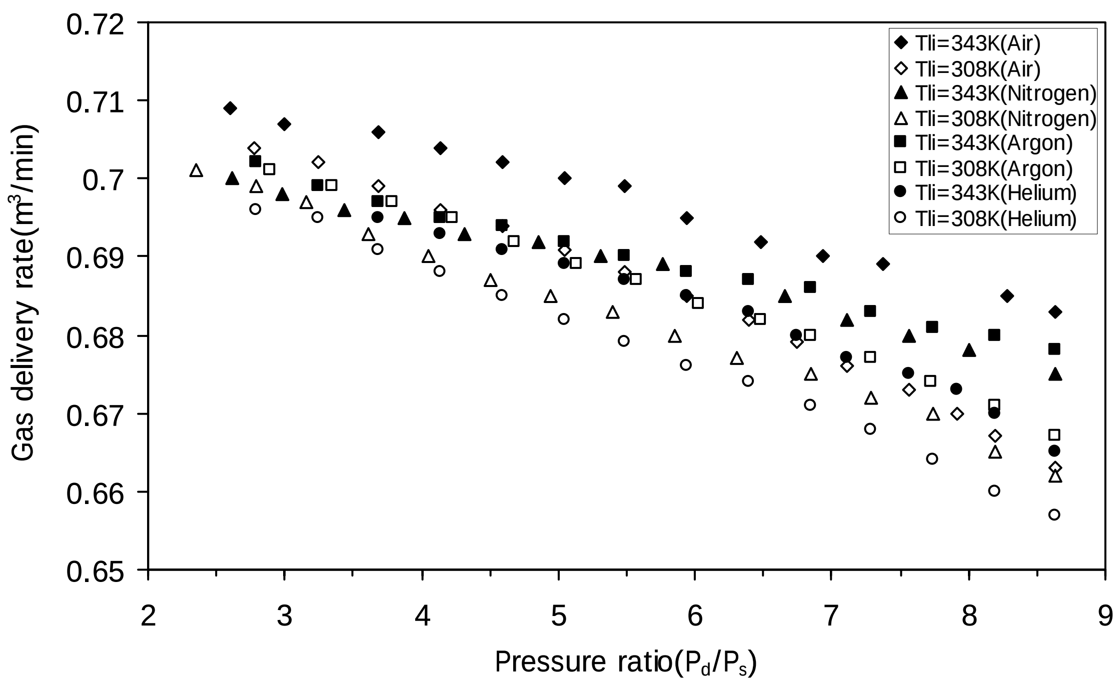

Figure9: Variation of gas flow rate in 5.5 kW compressor with pressure ratio at two different injected oil temperatures and fixed suction pressure.

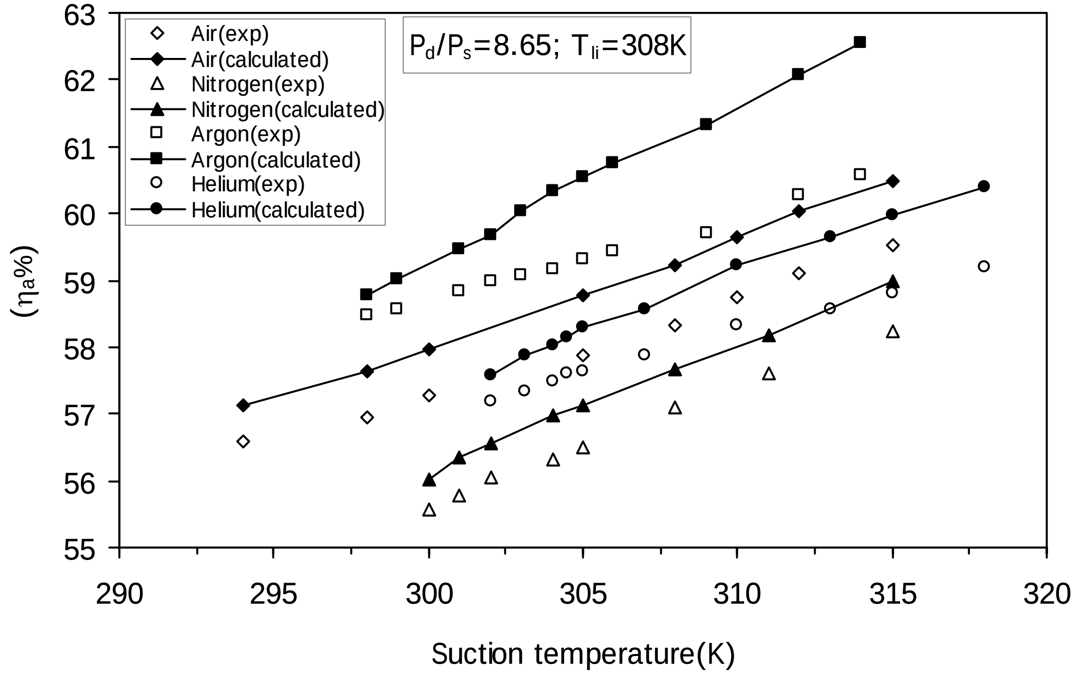

Figure 10: Influence of inducted gas temperature on adiabatic efficiency at constant suction pressure and injected oil rate in 5.5 kW compressor.

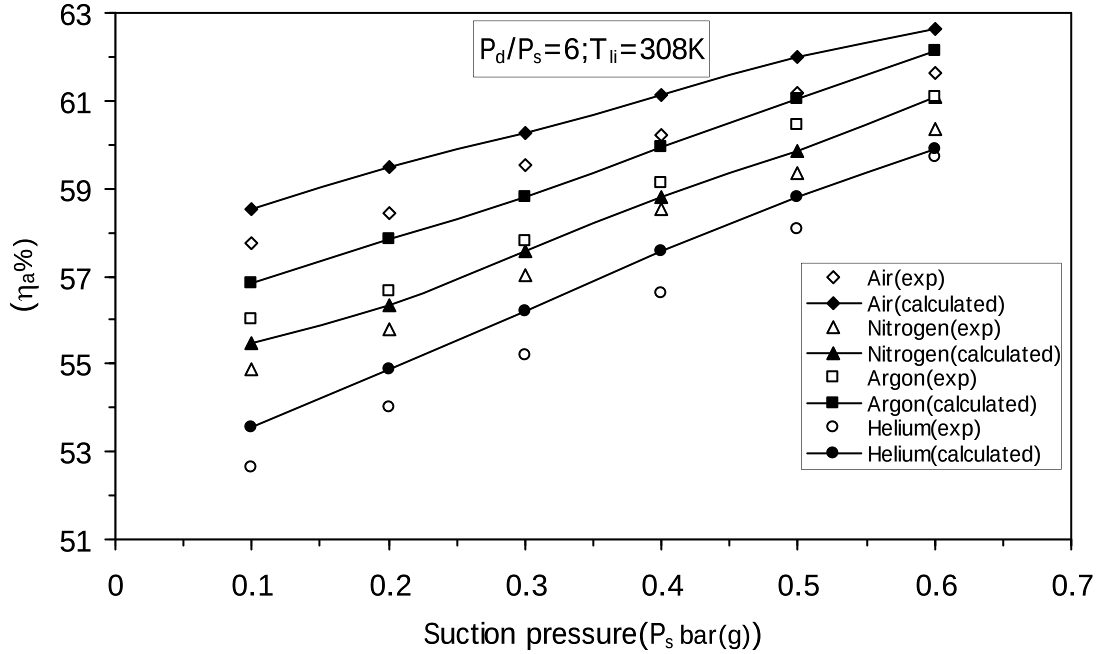

Figure 11: Influence of suction pressure of 5.5 kW compressor on different working gases at fixed injected oil temperature.

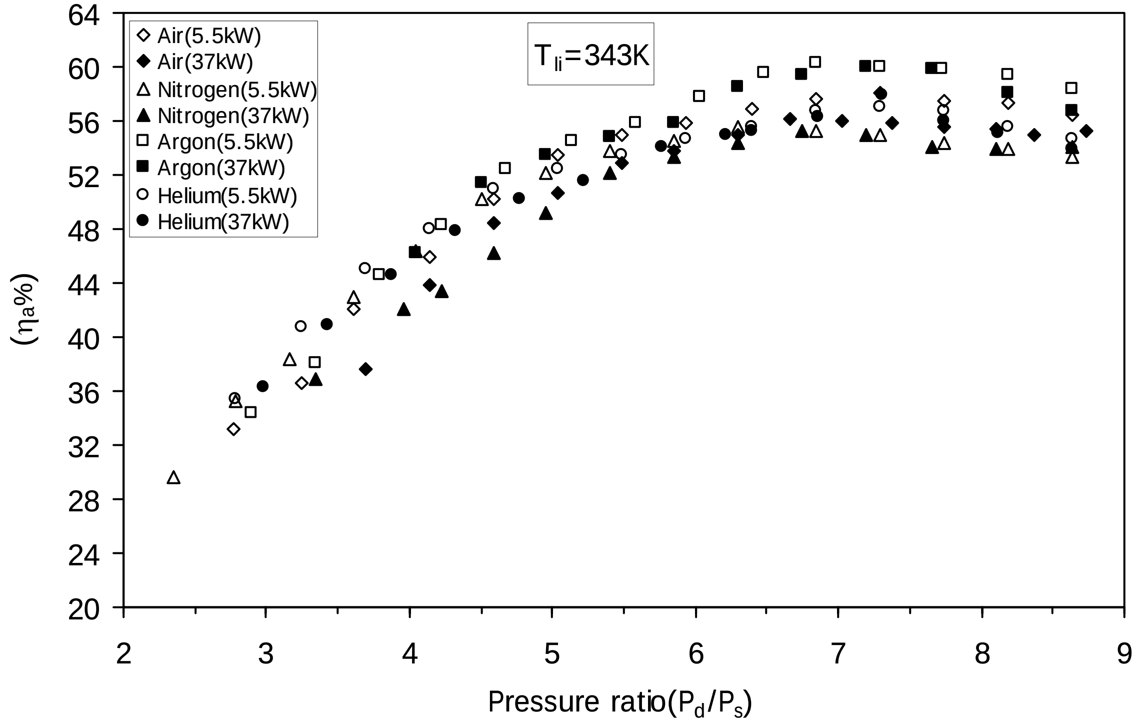

Figure 12: Variation of experimentally measured adiabatic efficiency with compressor capacity at fixed injected oil temperature.

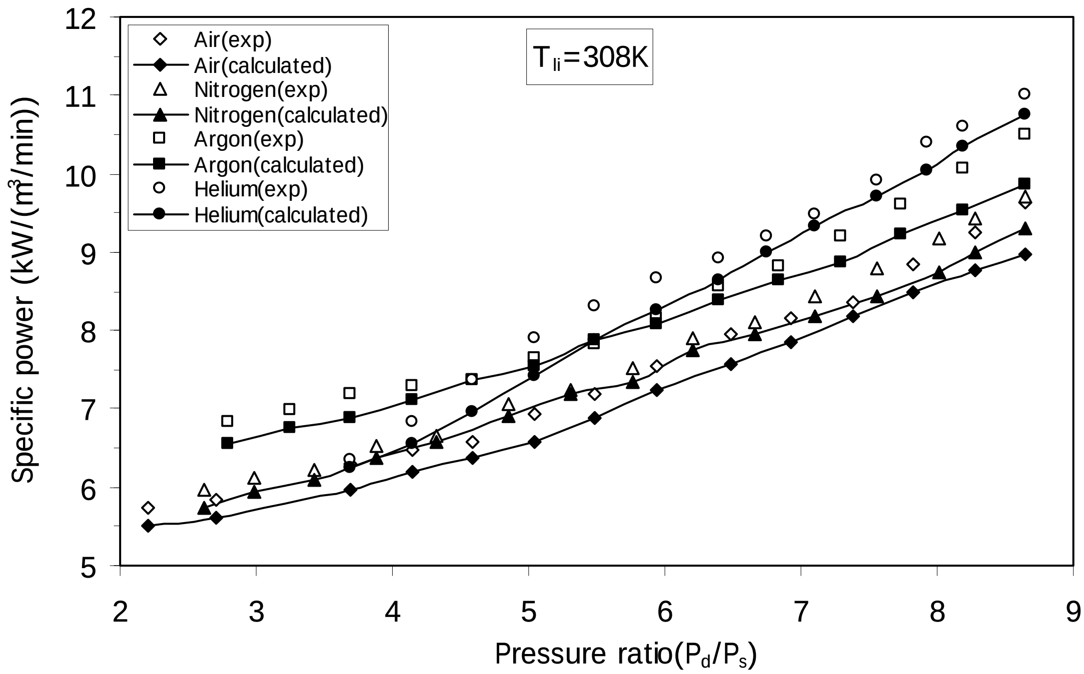

Figure 13: Variation of specific power in 5.5 kW compressor with pressure ratio with different working gases at fixed injected oil temperature and constant suction pressure.

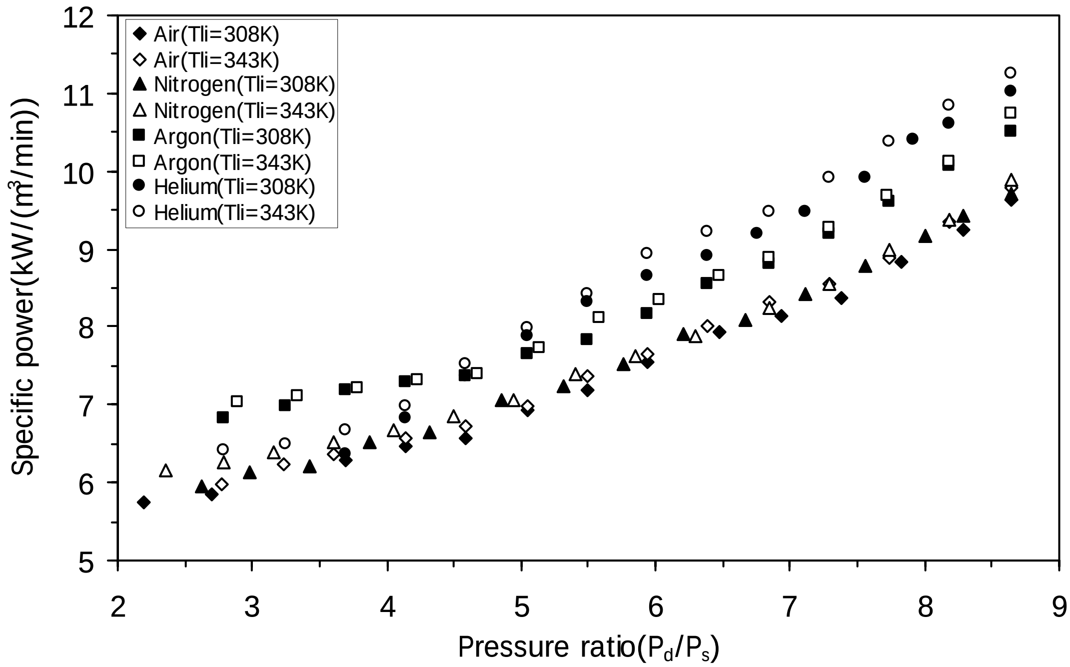

Figure 14: Comparison of specific power at two different injected oil Temperatures with different working gases at constant suction pressure in 5.5 kW compressor.

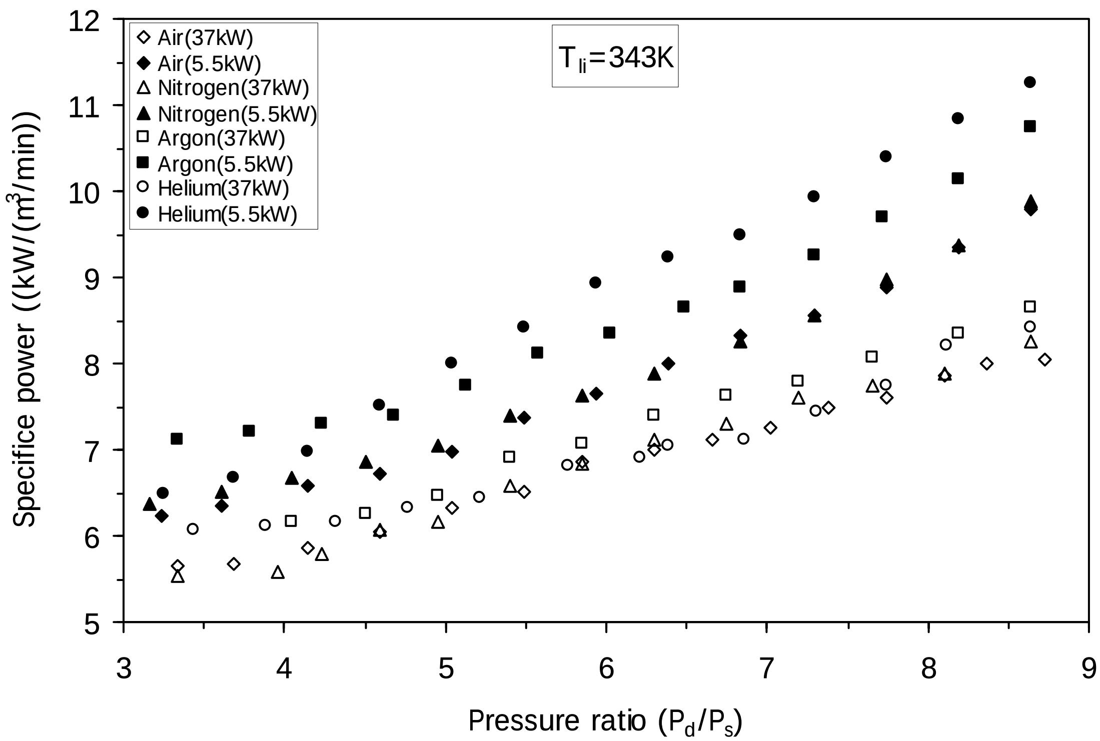

Figure 15: Variation of specific power with pressure ratio in two different compressors at fixed injected oil temperature and constant suction pressure.

6. Conclusions

Inlet temperature of working gas, suction pressure and pressure ratio have been taken as operating parameters to present the variation of volumetric and adiabatic efficiencies in 5.5 kW and 37 kW air compressors. Apart from air as a working gas, the effect of using other heavy and low dense working gases such as nitrogen, argon and helium are studied experimentally and numerically. The volumetric and adiabatic efficiencies compared and presented the results. It has been observed that the volumetric and adiabatic efficiencies are better at lower injected oil temperature. This result shows that lowering of oil inlet temperature is more effective. This may be due to the fact that at lesser temperature of the oil, the amount of fresh inducted gas mass is better and leakage losses are less due to good viscous properties of the oil.

Helium shows the lowest volumetric efficiency followed by argon, nitrogen and air. This is because helium being both light and monatomic in nature has highest leakage rate and attains highest temperature on discharge. This is followed by argon, which is also a monatomic gas although it has a high molecular weight. The higher values of volumetric efficiency with nitrogen and air are due to their higher molecular weight and relatively low heat of compression.

Variation of adiabatic efficiency with suction temperature, suction pressure, injected oil temperature, and pressure ratio has been carried out. In comparison, argon shows the highest efficiency followed that of air, helium and nitrogen in the descending order. It is because argon is a monatomic gas and has high molecular weight. Analysis of uncertainty in experimental measurement shows that the associated uncertainty in the measurement of volumetric efficiency is around 18%, where as the uncertainty in the adiabatic efficiency is around 11%.

Acknowledgements

The authors are grateful to the Board of Research in Nuclear Sciences (Department Atomic Energy) for sponsoring this work. The technical inputs from Mr.Trilok Singh, Head Cryogenics Division, Baba Atomic Research Centre are also gratefully acknowledged. The authors are thankful to the Mechanical Engineering Department, NIT, Rourkela, for providing all testing facilities and guidance.

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