Influence of Working Gas and Oil Injection Rate on Efficiency of Oil Injected Twin Screw Air Compressor

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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. To generate machine independent experimental data, two similar compressors with different capacities have been built to asses the performance of air compressor when applied to compress nitrogen, argon and helium gases apart from air. Also this paper addresses the effect of oil injection quantity and its temperature on volumetric and power efficiency.

(**Key words**: screw compressor, oil injection rate, oil temperature, gas 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	
M_1	fresh gas mass admitted into the suction cavities during suction process	(kg)
M_g	Mass of gas in the working space	(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_{il}	Interlobe leakage gas mass leaked into the suction chamber during previous compression process at (P_s ,T_1)	(kg)

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$\dot{m}_{gi}$	Leakage gas mass flow rate through flow path into the working chambers during compression-discharge process	(kg/s)
$\dot{m}_{go}$	Leakage gas mass flow rate through flow path out of the working chambers during compression-discharge process	(kg/s)
$\dot{m}_{e}$	Actual discharged gas mass rate at (P_s ,T_1)	(kg/s)
$\dot{m}_{ts}$	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_e	Energy meter constant	
n_e	Number of revolutions of energy meter in time t_e	
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	Time	(s)
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_s	Gas temperature at suction condition	(K)
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

ρ_s	Density of gas at suction condition	(kg/m^3)
ρ_l	Density of oil at suction condition	(kg/m^3)

η_{ra} Adiabatic efficiency at (P_s , T_1)

η_{rea} Experimental adiabatic efficiency at (P_s , T_1)

η_{rv} Volumetric efficiency at (P_s , T_1)

η_{rev} Experimental volumetric efficiency at (P_s , T_1)

η_a Adiabatic efficiency

η_v Volumetric efficiency

η_{tv} Theoretical volumetric efficiency at (P_s , T_s)

η_{ev} Experimental volumetric efficiency

η_{ta} Theoretical adiabatic efficiency at (P_s , T_s)

η_{ea} Experimental adiabatic efficiency

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 gas used and by the type of lubricant used. Thermodynamic efficiency of the compression process also depends greatly on the oil to gas heat transfer.

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. Singh and Phillips [3] developed a mathematical model to calculate the heat transfer rate between oil and gas assuming that the oil is injected in the form of non-intersecting spherical droplets. 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.

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

2. Measurement of Volumetric And Adiabatic Efficiencies

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. The specifications and other parameters of compressors used for test purpose is given in Table 1 and Table 2. 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. The actual gas mass contained in the suction volume at the end of suction process at condition (P_s ,T_1) can be estimated by the relation:

$$M_{t1} = \frac{P_s V_{t1}}{R T_1} \tag{1}$$

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

$$\dot{m}_{ts} = \frac{M_{t1}}{t_s} \tag{2}$$

But the total gas mass at (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. Thus:

$$M_{t1} = M_1 + M_{il} \tag{3}$$

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

$$M_{il} = \dot{m}_{gl} \cdot t_s \tag{4}$$

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

$$\dot{m}_{gl} = \dot{m}_{gi} - \dot{m}_{go} \tag{5}$$

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

$$\dot{m}_{dt} = \dot{m}_{ts} - \dot{m}_{gl} \tag{6}$$

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

$$\eta_{tv} = \frac{\dot{m}_{dt}}{\dot{m}_{ts}} \tag{7}$$

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:

$$\eta_{ev} = \frac{\dot{m}_e}{\dot{m}_{ts}} \tag{8}$$

The actual gas volume flow rates 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] :

$$\dot{m}_e = \rho_s V_m \sqrt{\frac{P_s \cdot T_r}{P_r \cdot T_s}} \tag{9}$$

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 the 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:

$$\eta_{ta} = \frac{\dot{W}_{ta}}{\dot{W}_{sys,e}} \tag{10}$$

The theoretical adiabatic compression work is given by the expression:

$$\dot{W}_{ta} = \dot{m}_{dt} c_p T_s \left[\left(\frac{P_d}{P_s} \right)^{\frac{k-1}{k}} - 1 \right] \tag{11}$$

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:

$$\dot{W}_{sys,e} = \frac{3600 \cdot n_e}{N_e \cdot t_e} \tag{12}$$

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:

$$\eta_{ea} = \frac{\dot{W}_{ea}}{\dot{W}_{sys,e}} \tag{13}$$

The experimental adiabatic compression work is given by the expression:

$$\dot{W}_{ea} = \dot{m}_e c_p T_s \left[\left(\frac{P_d}{P_s} \right)^{\frac{k-1}{k}} - 1 \right] \tag{14}$$

3. Experimental Studies

Before the compressor put into a plant, it need to be tested independently for volumetric and power efficiency, and the temperature rise of gas and lubricant/coolant used during compression process. Study of surface temperature of the compressor is also essential, when the compressor designed for air is applied to compress different working gases.

The main motive of the present test is to study the performance of the oil injected twin-screw air compressor when compressing helium, argon and nitrogen gases. Two test benches have been installed in the laboratory, which are identical in all aspects except for the compressor capacities. The photographs of experimental test rigs have shown in Figures 1 and 2 and the photograph of screw rotors and their casings used for experimentation to validate the theoretical data have shown in Figure 3.

Figure 1: A photograph of 5.5kW experimental test rig

Figure 2: A Photograph of 37kW experimental test rig

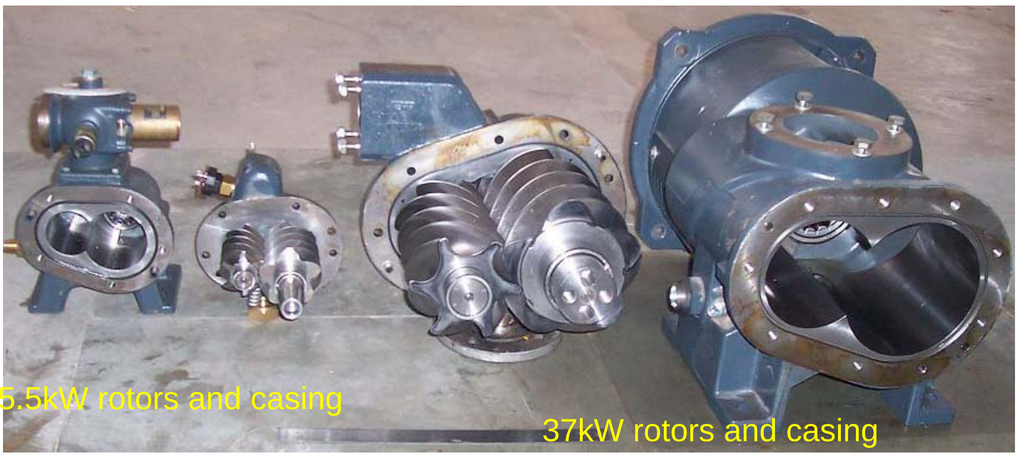

Figure 3: Photograph of 5.5 kW and 37 kW compressor rotors and their casings.

4. Discussion

Theoretical and experimental studies have been conducted on commercially available twin-screw air compressors using air, nitrogen, argon and helium as working fluids. The influence of injected oil temperature on different working gases at a fixed pressure ratio is shown in Figure 4. From the figure, it is observed that lower injected oil temperature results in better volumetric efficiency. The influence of oil injection rate at a fixed pressure ratio and injected oil temperature is presented in Figures 5 and 6 for different working gases. It may be observed that the influence of oil injection rate on volumetric efficiency is not same for different working gases. The influence of injection quantity on volumetric efficiency is relatively stronger with helium than with the other three gases on both the compressors.

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 enhances the leakage and thus degrades the efficiency. It has also been seen from the figure that volumetric efficiency can be improved by lowering the injected oil temperature. Volumetric efficiency decreases with increase in pressure ratio and decreases with increases of oil inlet temperature. 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 is a light gas and leaks more easily. On the other hand, monatomic gas generates higher temperature leading to decrease in oil viscosity and consequently higher leakage rate. This is the case with argon.

The variation of volumetric efficiency with oil to gas mass ratio has been presented in Figures 8 and 9. It is observed that increase in volumetric efficiency with mass ratio is marginal for air, nitrogen and argon in the range of mass ratio between 15 and 18. Helium being a lighter gas, operating oil to gas mass ratio is nearly 7 times of that with air.

The gas delivery rates of both the compressors have been presented in Figure 10. Experiments have been conducted at two different injected oil temperatures on 5.5 kW compressor keeping the inlet gas pressure constant. 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 figure 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 injected oil temperature on adiabatic efficiency at a fixed pressure ratio with the 5.5 kW compressor is presented in Figure 11. It may be observed that the increase in efficiency with decrease in injected oil temperature is an obvious consequence of the increase in volumetric efficiency and it is similar for all the gases. The variation of adiabatic efficiency with injected oil temperature at constant pressure ratio has been studied on both the compressors and results are compared in Figure 12.

Variation of adiabatic efficiency with oil injection rate at fixed pressure ratio and fixed injection temperature is shown in Figures 13 and 14. The influence of oil injection rate on adiabatic efficiency, while compressing air, nitrogen or argon is rather weak beyond a certain injection rate. But it plays a significant role in determining adiabatic efficiency while compressing a light gas like helium which can easily leak through narrow gaps if sufficient oil is not available to seal the gaps. Additionally, helium being a monatomic gas develops higher temperature on compression, which requires higher oil injection rate to maintain acceptable temperature.

The relation between adiabatic efficiency and pressure ratio for all the working gases at two different injected oil temperatures have been shown in Figure 15. 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 figure 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 variation of adiabatic efficiency with oil to gas mass ratio is shown in Figures 16 and 17. It may be observed that the operating oil to gas mass ratio is in the range of 15 to 18 for air, nitrogen and argon, whereas that for helium is nearly 7 times higher due to its low molecular weight.

The specific power consumption of the compressors against pressure ratio is shown in Figure 18. It may be observed that it 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.

Apart form volumetric and adiabatic efficiencies, the oil and gas mixture temperature at compressor discharge for all the working gases at a fixed pressure ratio and oil injection temperature is presented in Figure 19 as a function of oil injection rate. It is seen from the figure that the oil and gas mixture temperature decreases for all the gases with oil injection rate. As expected, argon, a monatomic gas with high molecular weight shows the highest discharge temperature. At higher injected oil temperature, mixture temperatures are indistinguishable for the different working gases because the high oil to gas mass ratio dominates the prevailing temperature.

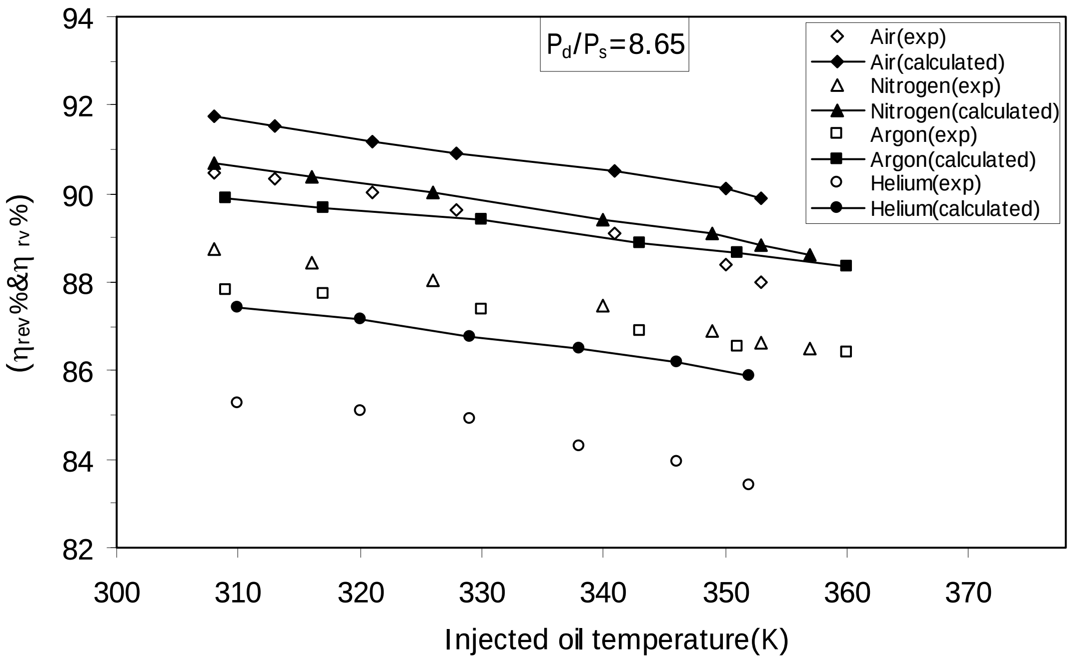

Figure 4: Influence of injected oil temperature on volumetric efficiency with different working gases of 5.5 kW compressor at constant suction pressure.

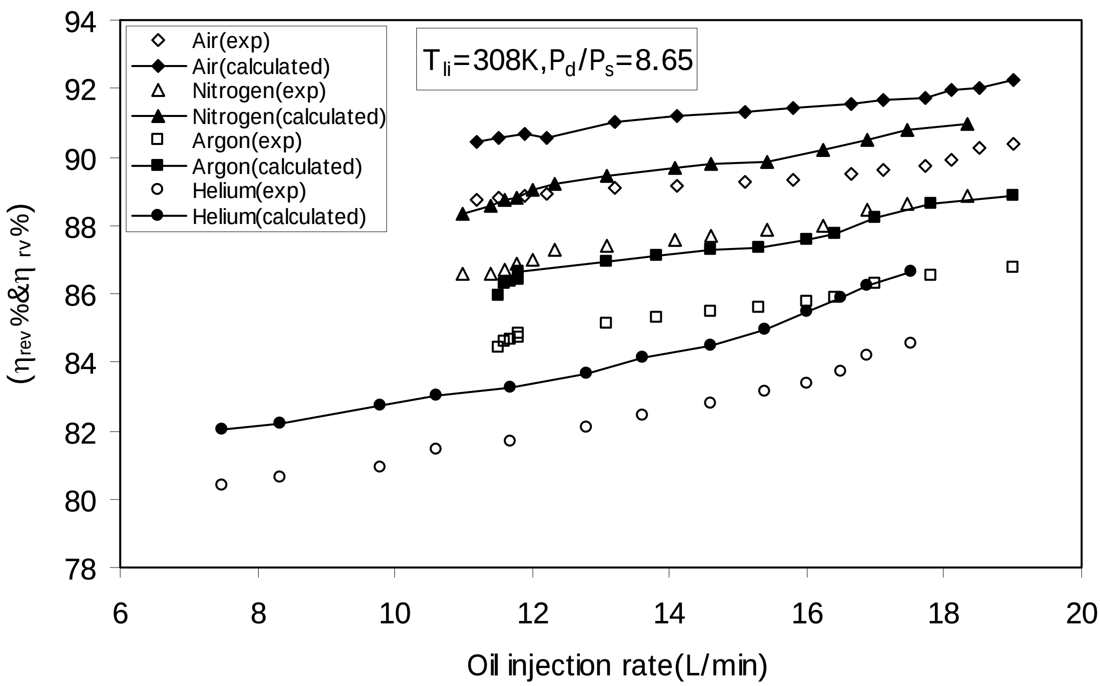

Figure 5: Influence of injected oil quantity on volumetric efficiency of 5.5 kW compressor at constant suction pressure.

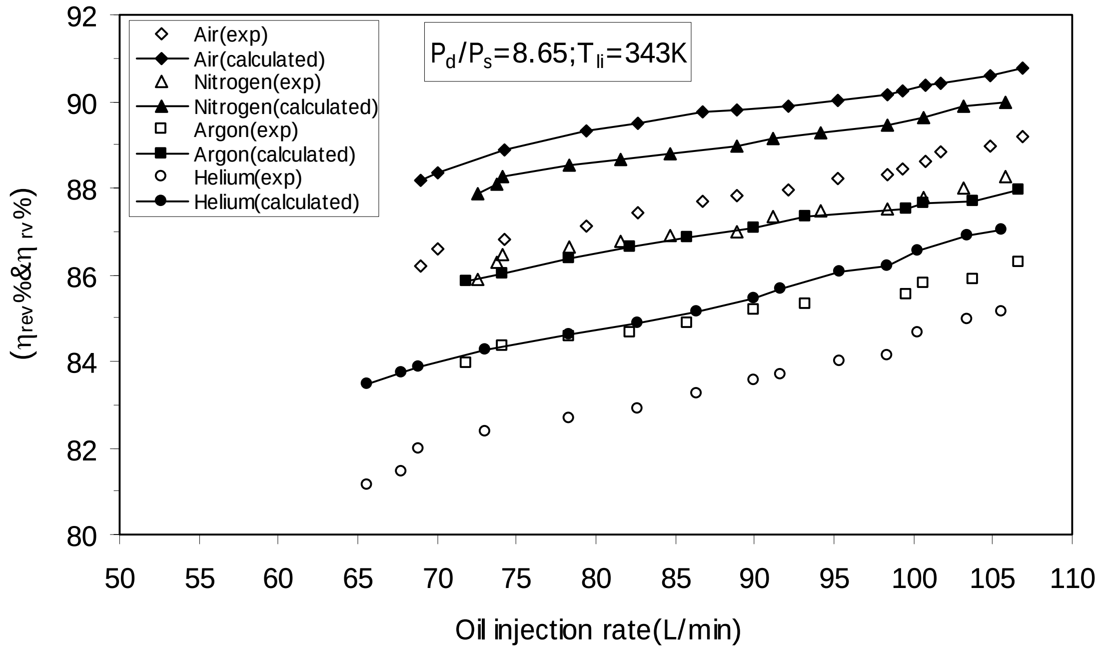

Figure 6: Influence of injected oil quantity on volumetric efficiency of 37 kW compressor at constant suction pressure.

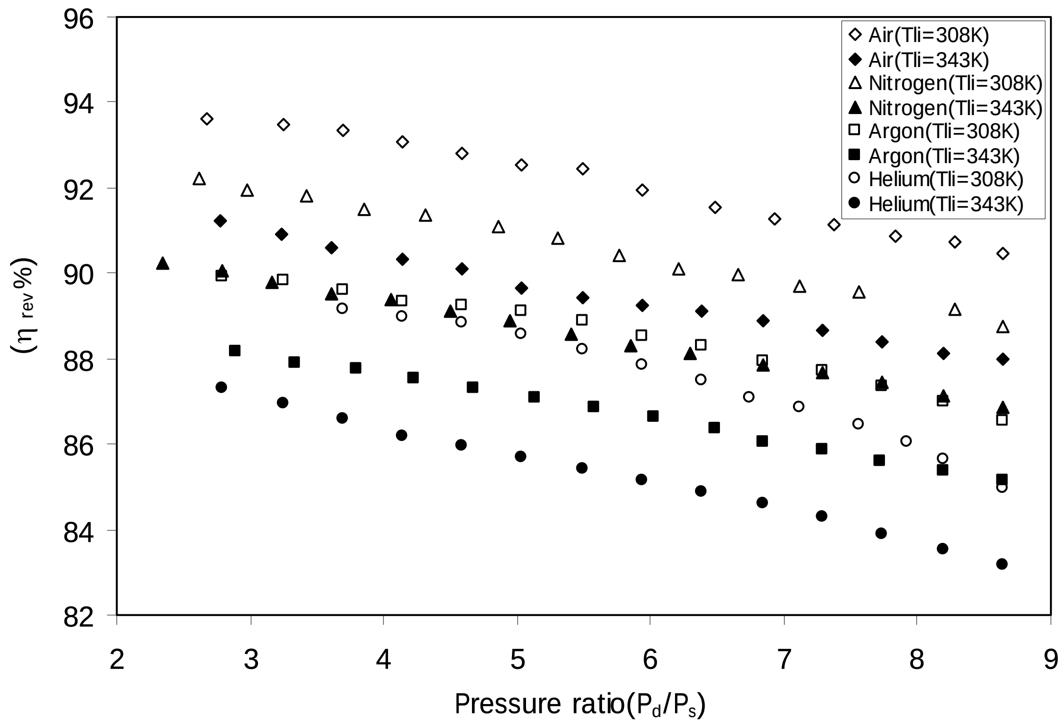

Figure 7: Variation of experimentally measured volumetric efficiency of 5.5 kW compressor at two different injected oil temperatures.

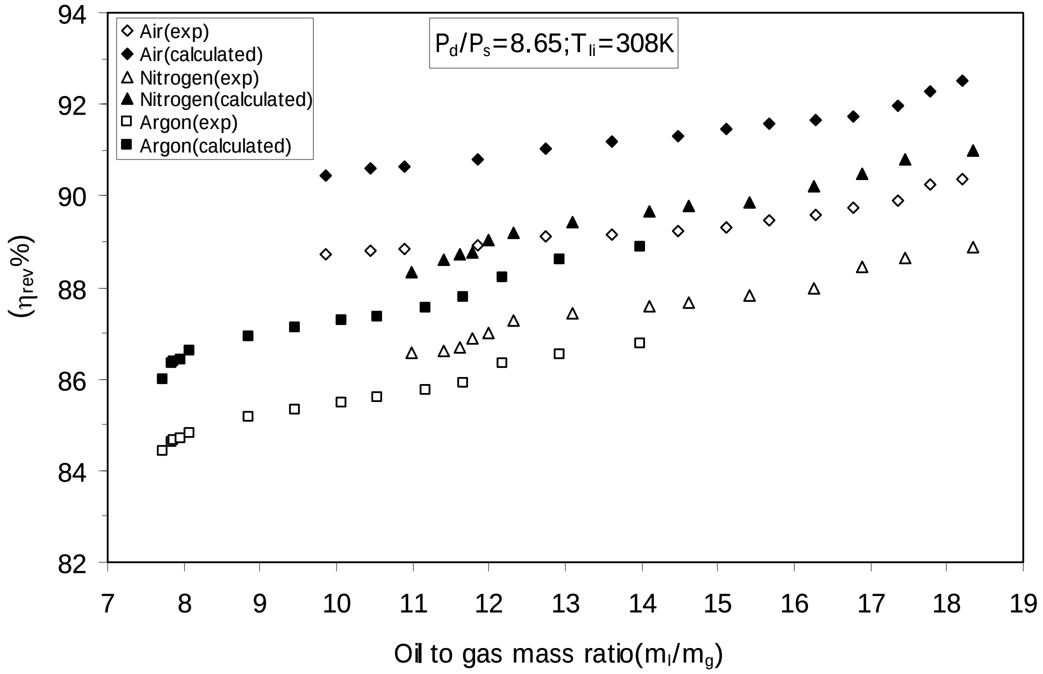

Figure 8: Influence of oil to gas mass ratio on volumetric efficiency of 5.5 kW compressor with different working gases at constant pressure ratio, fixed oil injection temperature and at constant suction pressure.

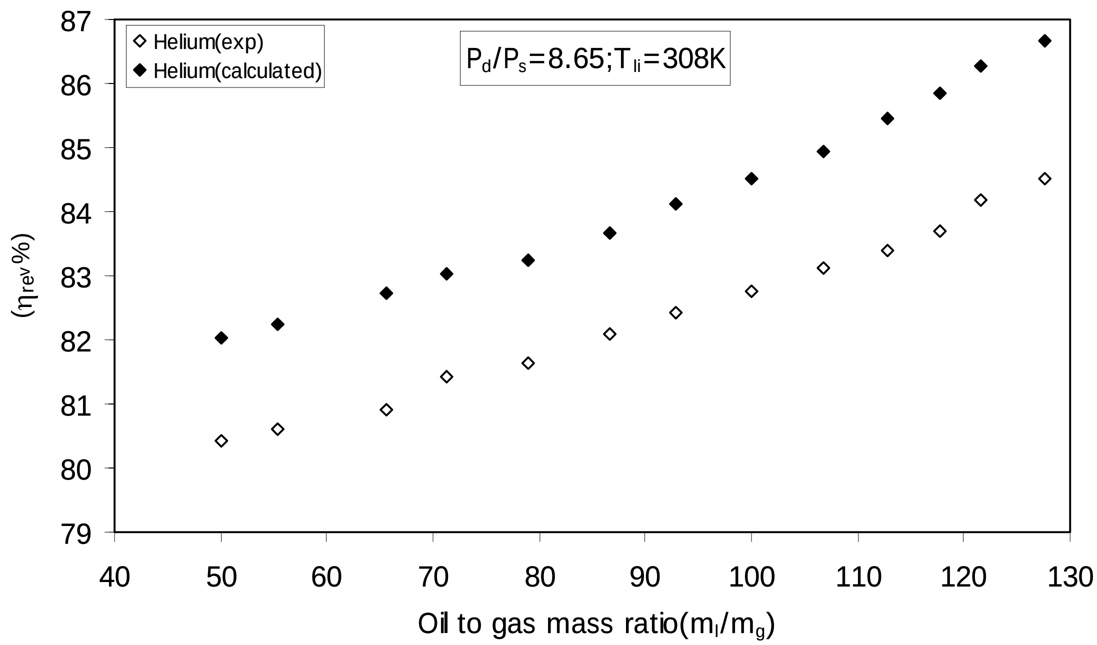

Figure 9: Influence of oil to gas mass ratio on volumetric efficiency of 5.5 kW compressor with helium as working gas at constant pressure ratio and at fixed injected oil temperature with constant suction pressure.

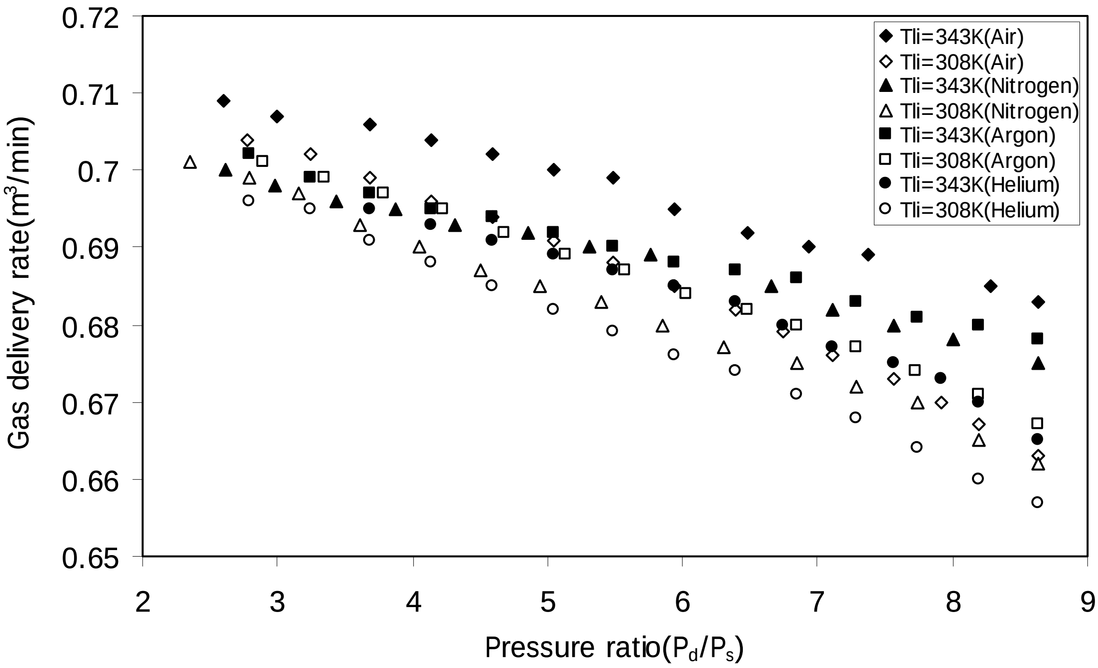

Figure 10: Variation of gas delivery rate in 5.5 kW compressor with pressure ratio at two different injected oil temperatures and fixed suction pressure.

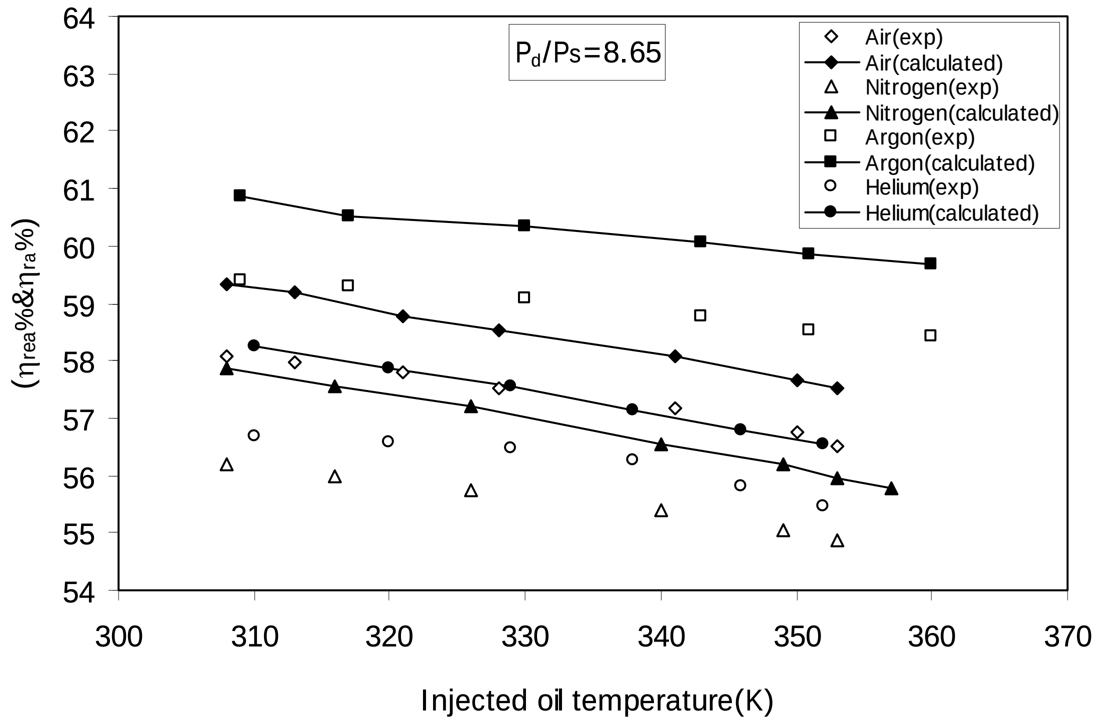

Figure 11: Influence of injected oil temperature on efficiency of 5.5 kW compressor with different working gases at fixed pressure ratio.

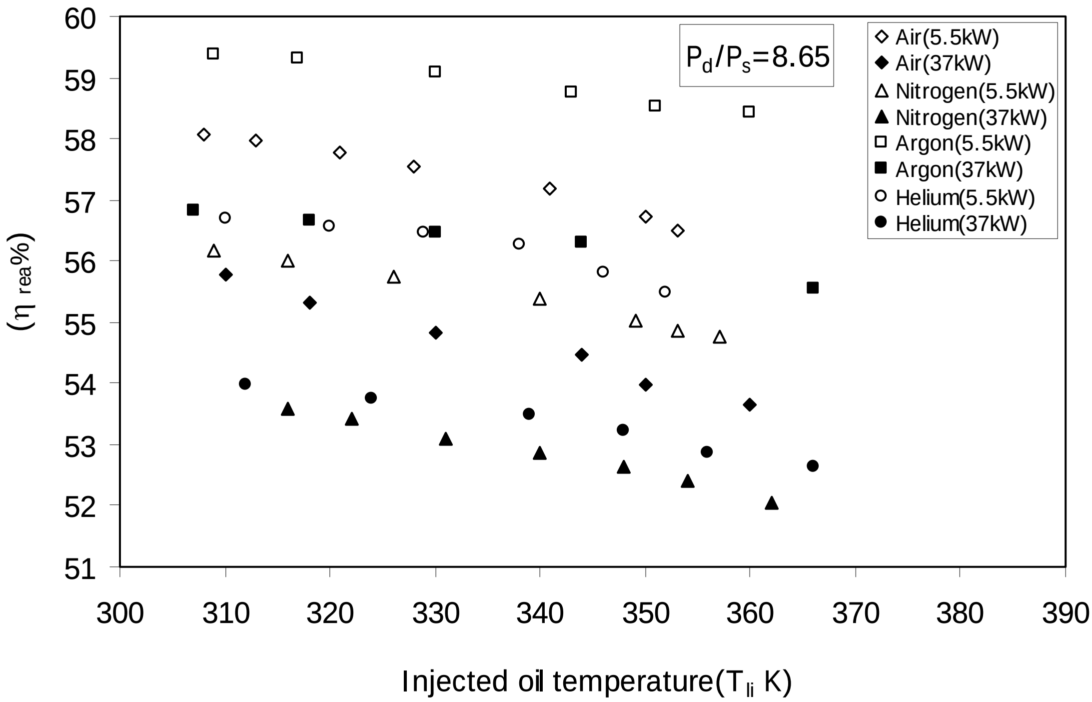

Figure 12: Variation of experimentally measured adiabatic efficiency with injected oil temperature at constant pressure ratio.

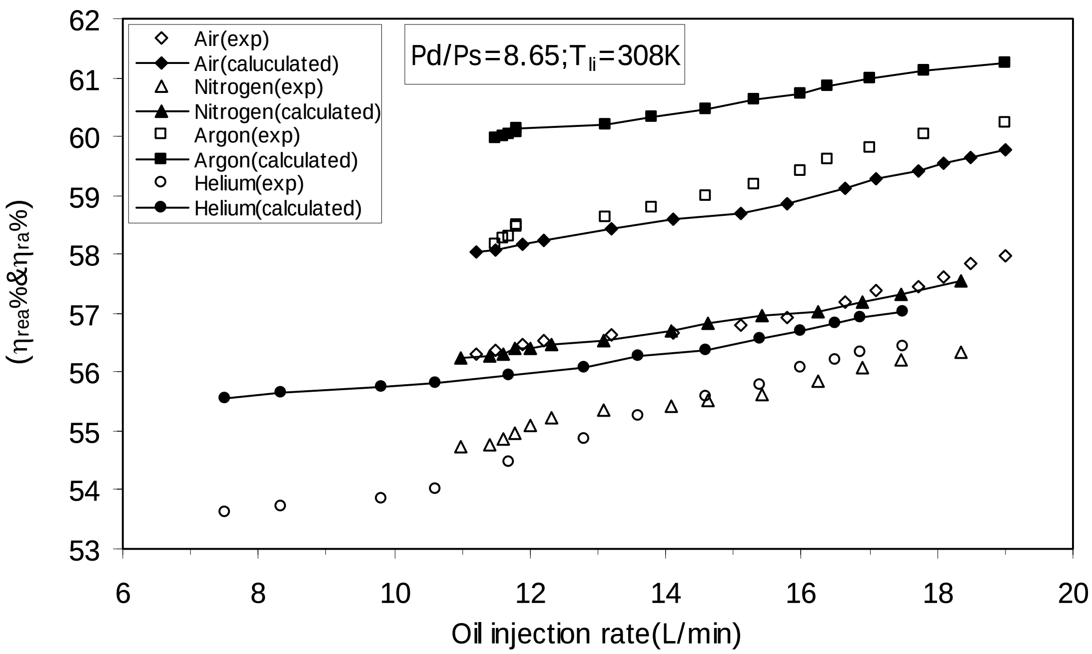

Figure 13: Influence of oil injection quantity on adiabatic efficiency at constant pressure ratio and fixed injected oil temperature in 5.5 kW compressor.

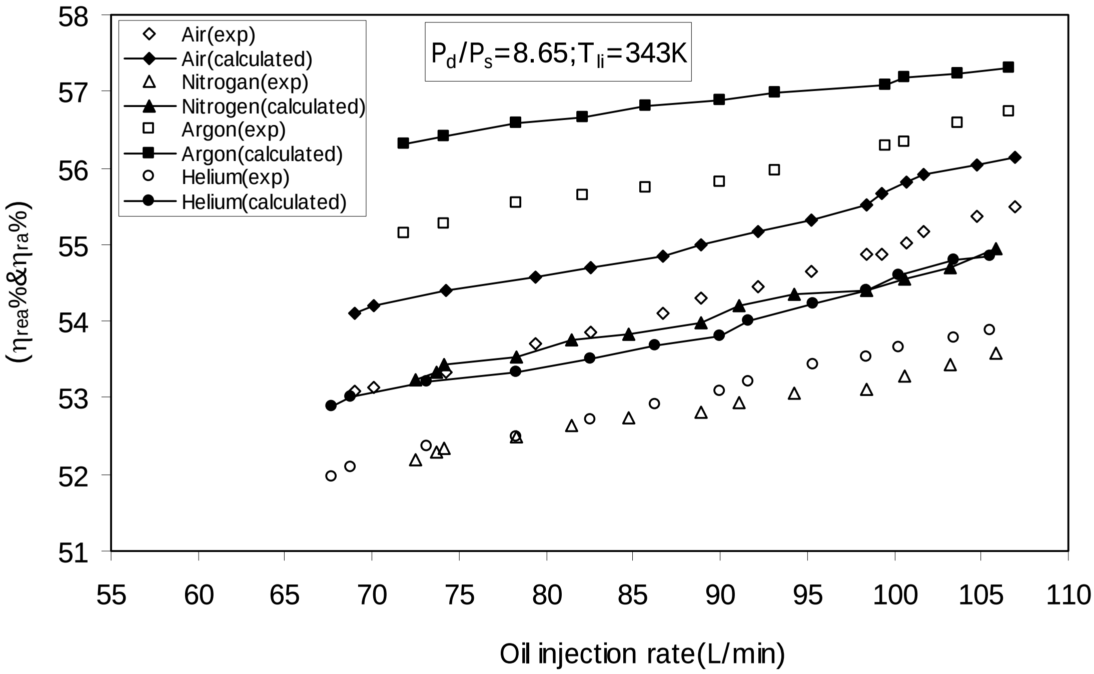

Figure 14: Influence of oil injection quantity on adiabatic efficiency of 37 kW compressor with different working gases at constant pressure ratio and fixed injected oil temperature.

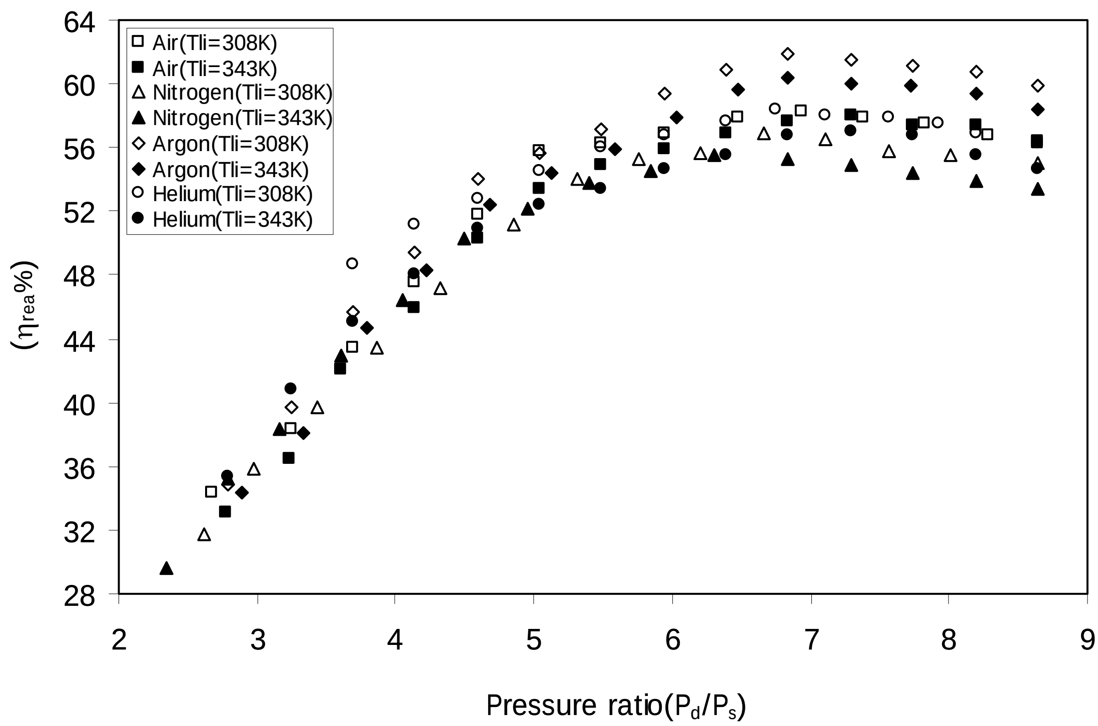

Figure 15: Comparison of experimentally measured adiabatic efficiencies of 5.5kW compressor at different injected oil temperatures.

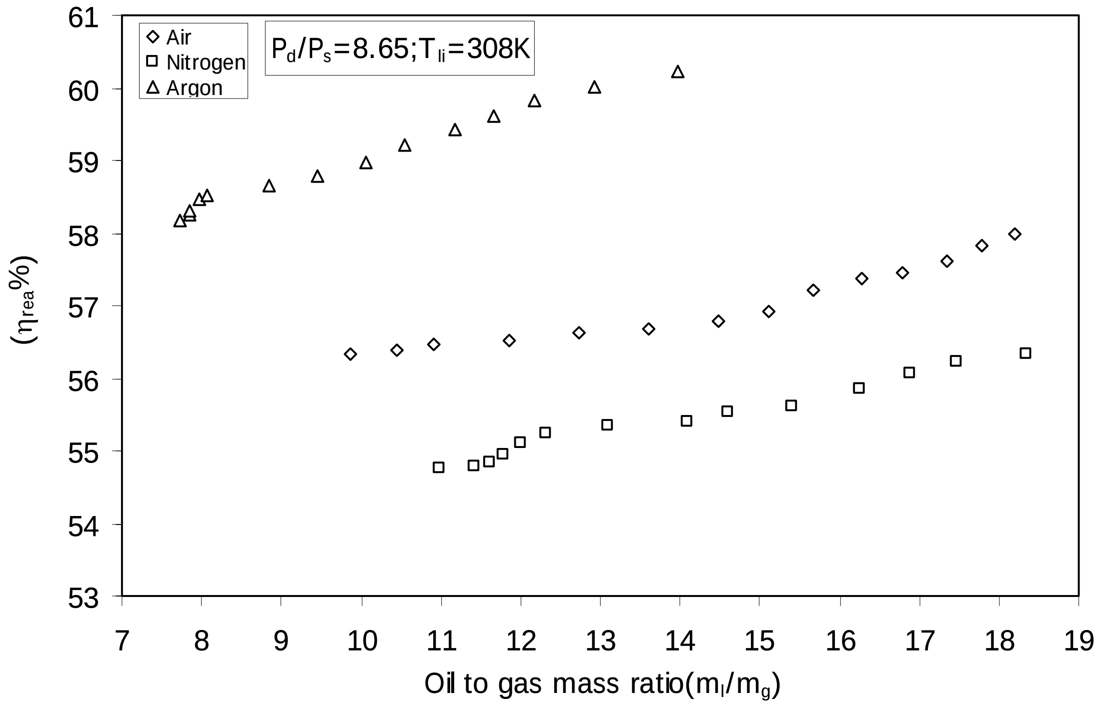

Figure 16: Variation of experimentally measured adiabatic efficiency with oil to gas mass ratio at fixed injected oil temperature and pressure ratio.

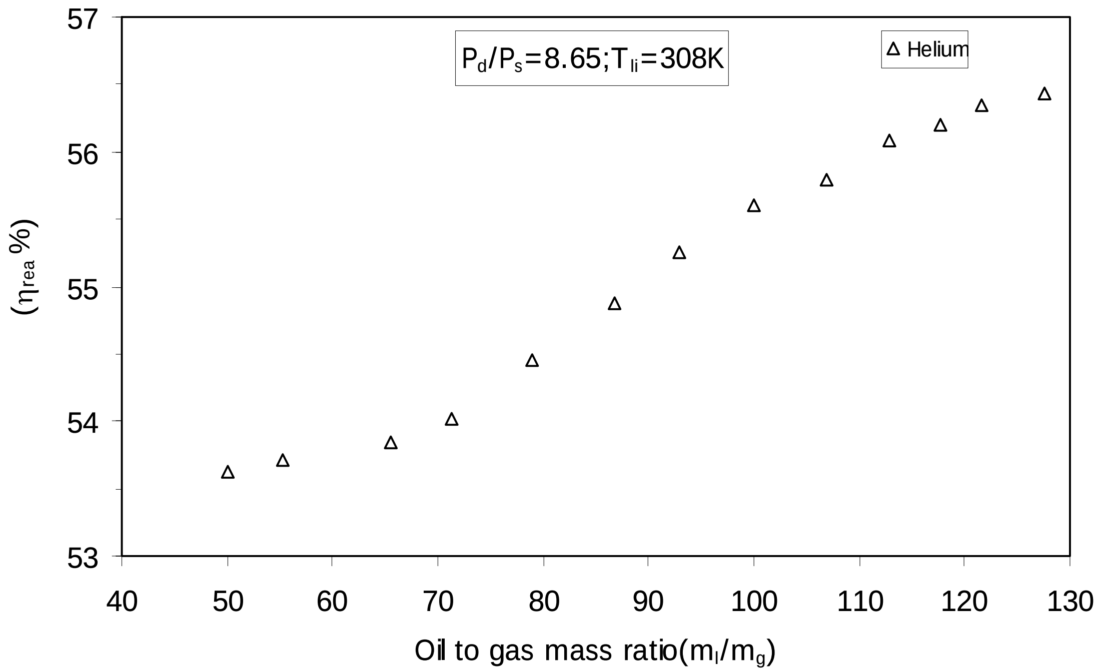

Figure 17: Variation of experimentally measured adiabatic efficiency with oil to gas mass ratio at fixed injected oil temperature and pressure ratio.

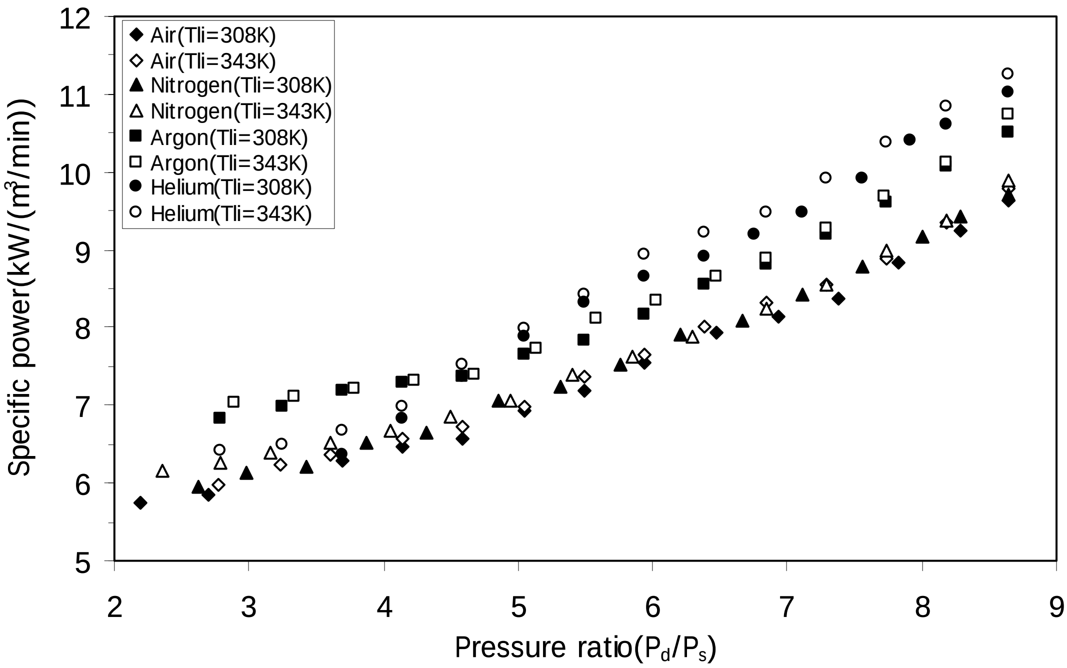

Figure 18: Comparison of specific power at two different injected oil Temperatures with different working gases at constant suction pressure in 5.5 kW compressor.

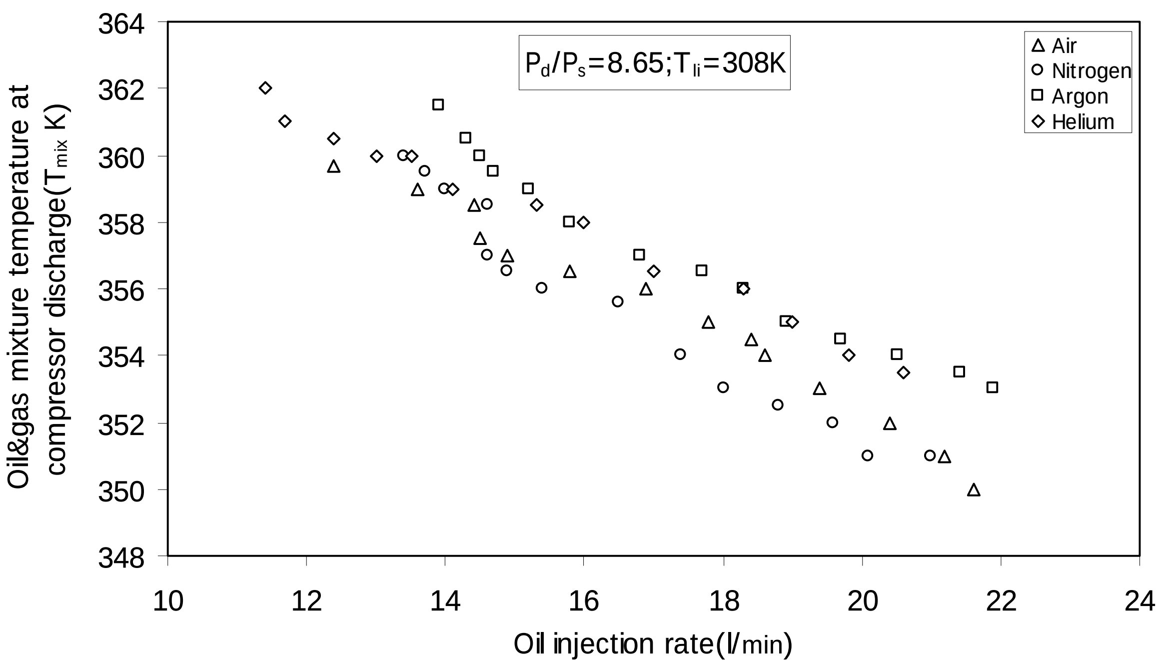

Figure 19: Variation of oil and gas mixture temperature at compressor discharge with oil injection rate at fixed injected oil temperature and constant suction pressure in 5.5 kW compressor.

5. Conclusions

Inlet temperature of oil, oil injection rate, 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 working gases such as nitrogen, argon and helium are studied experimentally and numerically. The volumetric and adiabatic efficiencies are increasing with lowering of injected oil temperature. The volumetric and adiabatic efficiencies are increasing over a certain range of oil injection rate. This result shows that lowering of oil inlet temperature is more effective than injecting a higher quantity of oil. This may be due to the fact that more resident oil obstructs the volume flow of working gas.

The variation of oil and gas mixture temperature with oil injection rate is within the range of 3^0C for different gases. 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 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.

Variation of adiabatic efficiency with injected oil temperature, oil injection rate 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.

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