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Research Article

Experimental evaluation of the effect of leakage in scroll compressor

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ABSTRACT

This research gives an experimental investigation of the scroll compressor, with an emphasis on the effect of leakage on performance enhancement. The effect of gas leakage losses on compressor performance is studied both experimentally and conceptually. In the present study, we have modified the scroll compressor to bypass the refrigerant through an orifice called leakage and experimentally investigate the effect of valve opening area and angle to observe the effect of leak gas on compressor performance, compressor capacity loss, discharge line temperature rise, and discharge gas temperature. Experimental results indicated that the maximum percentage rise in suction superheat is observed to be 7.13% at a maximum effective valve opening area of 0.33 m^2, whereas the rise in discharge line temperature lies in the range of 0.8% to 2.75% over the entire range of effective leak area. In addition, based on experimentation the 8.9 % maximum compressor capacity loss is observed.

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INTRODUCTION

At this time, mankind's common aims include environmental protection, sustainable development, and energy conservation. To achieve the common aims, many researchers are focusing on ways to improve performance, reduce machinery energy consumption, and increase efficiency. A scroll compressor is a type of positive displacement machine that is extremely efficient. Scroll compressors are commonly employed in refrigeration, air conditioning, gas compression, and pressurized pump products, among other applications. With the scroll compressor's rapid development and widespread use in recent years, its amazing properties have piqued the interest of many researchers [1,2]. It has a basic structure, low noise, good dependability, minimal vibration, and a tiny footprint. Scroll compressors are intrinsically more efficient than other types of compressors for a variety of reasons, including the lack of pistons for gas compression, which allows scroll compressors to achieve 100% volumetric efficiency, resulting in lower energy costs and re-expansion losses [3]. The volumetric efficiency of such a compressor depends upon many variables like pressure ratio, clearance volume, etc. The uncommon aspect of each piston stroke encountered in reciprocating models is eliminated in the scroll compressor. In addition, the scroll compressors have fewer moving parts and no valve (port) losses. The scroll compressor is much quieter in operation

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than other types of compressors. Gas pulsation is also reduced, if not eliminated, allowing them to run with fewer resources. As the need for increasing energy demands, the compressor efficiency requirements have increased and have become more stringent. Various leakages largely affect the scroll compressor efficiency causing increased power consumption and higher discharge temperature. In this section, research on leakage modeling has been performed which provides a guideway for project work. A literature survey has been conducted to study the compressor leakage causes, modeling, and validation. The literature survey is divided accordingly into the following four sections:

A small amount of improvement in the efficiency of the compressor will lead to a decrease the energy consumption. While designing scroll compressors, one should try to reduce the frictional power loss between the compressor elements and from the refrigerant leakage power loss at each clearance. One possible approach is to reduce the internal leakage of a scroll compressor. There are thermal and pressure losses in the compressor. These thermal losses include suction gas superheating, refrigerant gas heating due to motor winding, etc. The pressure losses in the compressor are wire drawing losses at the suction and discharge side, friction losses, and gas blow-by loss due to clearance. The researcher has carried out the study based on the effect of compressor leakage causing mathematical modeling, CFD simulation, and experimentation, which provides a guideway for the development of scroll compressors. Yu Chen et al. [4] have developed a comprehensive R-410A scroll compressor model. This comprehensive compressor model consists of a combination of the details of the compression process model, and an overall compressor model and is used in the investigation of the compressor's performance under different operating conditions and subjected to changes in design. Lemort Vincent [5] has thoroughly analyzed the impact on the R410A refrigeration scroll compressor due to convective heat transfer, flank, and radial leakage to propose the working process of a mathematical model for a refrigeration scroll compressor. Panpan Song [6] presented a three-dimensional numerical technique accessible for modeling the radial leakage that flows through the axial clearances at the tip and root of the scroll wrap. The radial leakage flow patterns of both axial clearances were investigated. Marco Diniz et al.[7] have predicted the temperature distribution of scroll compressors with gas superheating in the suction process by developing lumped parameter thermal model. H Imdad et al. [8] have simulated the mixing pattern of two parallel gas streams which were separated by a splitter plate. In this paper, numerical models to predict gas mixture flows were classified into two categories. Computations were performed for two fluid systems; one with particles of about equal masses and another with particles of quite distinct masses, and the corresponding results were compared. N Nojima et al. [9] have developed new scroll compressors for R410A by upgrading existing air conditioner scroll compressors. The existing scroll compressor was studied, and various technologies were incorporated in developing these R410A new scroll compressors. Shikalgar N D et al. [10] developed instrumentation, data acquisition, and analysis techniques for the refrigerator with an inverter compressor. These techniques featured the measurement of shaft speed and instantaneous pressures within the scroll elements in conjunction with the use of digital oscilloscopes. M J Maertens et al. [11] have investigated the performance of a scroll compressor designed for the traditional design point. The scroll compressor simulation model was used by the researcher to evaluate the effects of compression ratio, compressor operating envelope, and system operating envelope on compressor efficiency. M M Tukiman et al. [12] have performed the numerical simulation using the OPEN FOAM 1.6 (commercial Computational Fluid Dynamics simulation software). CFD simulations were executed to predict the flow patterns, and profiles of velocity, pressure, and vena contracta in the orifice meter. N Ishii [13] has investigated evaluation for the refrigerant gas leakage flow through axial and radial clearances between the fixed and orbiting scrolls in scroll compressors. R22 was used as a working fluid for the compressor. Evandro et al. [14] have investigated heat transfer in scroll compressor suction and compression chambers and created a model to predict fluid flow and heat transfer inside the suction and compression chambers. In the simulations, a low Reynolds number turbulence model was used. The numerical model was created using a commercial CFD code that used the finite volume method to solve the governing equations of mass, momentum, and energy while accounting for the unstable effects caused by the circling scroll's motion. Yong et al. [15] have created theoretical models to simulate radial and tangential leakages in the Meso(micro) scroll compressor. The compressor's working fluid was R134a. The compressor had a suction temperature of 282°C, a discharge temperature of 340°C, and a rotational speed of 4000 revolutions per minute. The impact of leaks on compressor performance (COP, suction temperature, discharge temperature) was investigated and a graphical representation was created. Tingchao et al [16] conducted an innovative experimental analysis of the effect of the orifice-to-pipe diameter ratio on leakage flow rate for a variety of internal/external flow circumstances and orifice holes of various shapes. An experimental orifice discharge facility was created to measure leakage volumes by imitating the leakage of a pressurized pipe beneath the water with air as an external source of leakage. Herbert et al. [17] have constructed the formulae for calculating pure gas flow rates through very small orifices and capillaries. For both compressible and incompressible flow, leak flow through orifice and capillary was studied under laminar and turbulent flow conditions. To generate flow equations, a smooth flow surface, constant flow area at restriction, and an adiabatic flow process were assumed. It also studied flow equations, coefficient of discharge, and expansion factor for compressible flow. Shikalgar Niyaj et al. [21] have presented the energetic and exergetic experimental performance analysis of a domestic refrigerator with HWAC and BSTWC condenser is studied to check the possibility of energy conservation and exergy efficiency. A significant improvement in system performance and reduction in energy consumption and exergy destruction and irreversibility of a refrigerator is observed with a BSTWC condenser. Prashant et al. [22] have presented the investigations which put forth the development of a novel double wall vented rotary fluid heating device. In this device, water is used as a process fluid and is heated by the combustion of sugarcane bagasse. The proposed combustion method is found to provide the use of a more systematic fuel transport system and ensure an efficient heat transfer process to the fluid. Shikalgar Niyaj et al. [23] have presented the problem of energy conservation by developing a domestic refrigerator system capable of maintaining the cooling effect for more than 15 hours without energy input and being compact as well as cost-effective. He also tries the Secondary objective to reduce the global warming potential caused by HCFC refrigerants. Taking into account the very needs of an average rural household in India

Scroll compressor leakages have a significant impact on compressor performance (capacity, SEER, COP, and power consumption), and while complete eradication is challenging, scroll leakages can be minimized to improve compressor performance. To model leakage flow, researchers proposed and studied various flow models such as Fanno flow, Dall tube flow, and nozzle flow models, all of which were in good agreement with experimental results. CFD analysis with proper boundary conditions and flow model could be used to accurately predict leakage flow rate, heat interactions, and discharge line temperature in agreement with experimental results. Copeland's general product Guide [18] covers the specification and technical information of hermetic and semi-hermetic scroll compressors for low, medium, and high-temperature applications. The investigated compressor was chosen from the Copeland product handbook. The operating envelope of the scroll under investigation was investigated. Different forms of compressor controls were also investigated. There are different sources of (Radial and axial leakages, Suction Superheat loss, and internal seal leaks). The impact of losses and leakages on compressor performance needs to be understood. As a part of this research paper, a detailed study and analysis of the internal seal leakages, and suction superheat losses was carried out. Correlations of the above losses with capacity/performance drop are studied experimentally.

LEAKAGES IN THE SCROLL COMPRESSOR

To model the scroll compressor leakages, it is necessary to evaluate the existing leakage sources in the compressor and the losses caused by the leakages. Sources of losses in the scroll compressor consist of a scroll and non-scroll losses. Internal losses are occurred due to leakage of refrigerant and heat transfer, while external losses occur due to frictional and motor losses. There are two scroll leakages in the compressor. The first loss is formed due to the gap between the bottom or top plate and at the tip of the scroll and another at the path that is formed is path which is formed by a gap between the flanks of the two scrolls.

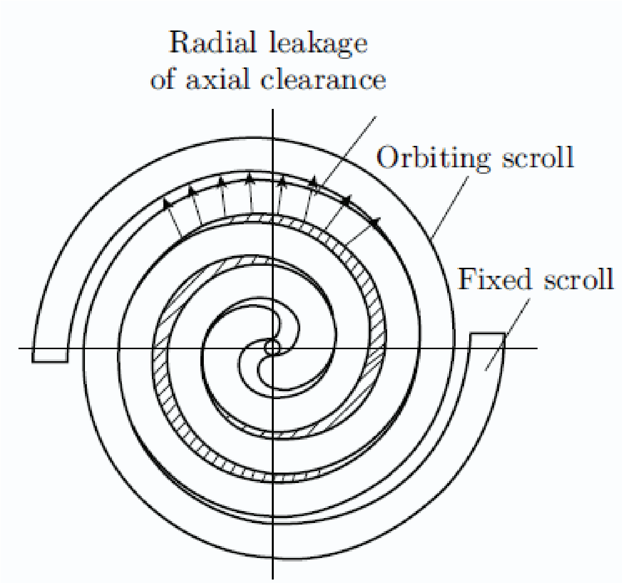

Figure 1. Leakages in a scroll compressor [13].

Figure 1 shows the path of the gas formed due to the leak from the high-pressure chamber to the low-pressure chamber. The gas leaks from the passage between the flanks or walls of orbiting and fixed scroll. Also, gas leaks between the scroll top and the base plate of the scroll known as radial leakage. The scroll leakages can be reduced by providing a very thin oil layer between the scroll wraps. It acts as a sealing layer and avoids the leakages past the scroll wraps also it will reduce the friction between the scrolls which reduces the frictional losses and improves the COP of the system. The scroll compressor consists of various parts such as a floating seal, thermally operated disc, and pressure relief valve. As there is imperfect contact, there is always a leak from high pressure to low pressure. Leakages are present inherently in all compressors. Driving energy for leakages is the pressure differential developed by the compressor. There is a flow from the high-pressure side to the low-pressure side region due to the leakage caused by the unwanted flow of refrigerants inside the scroll compressor. To analyze the effect of leak flow on compressor performance, flow needs to be controlled manually. Flow control devices are used to control the flow rate accurately. The magnitude of leak flow through leak paths is very small so, the flow control device must be able to handle minute variations in inflow. Also, it should have good repeatability and reproducibility. Leakage gas has higher pressure and temperature as compared to ambient. So, the device selected should withstand much higher pressure and temperature without affecting its performance. For controlling such miniature flow, the flow control device should have a gradually opening area. Needle valves are used generally for this purpose. For this experimentation, a gradually opening hot gas bypass needle valve is used which is made up of brass and can withstand the pressure and temperature of leak gas. Complete opening of the valve requires approximately eight and a half turns which ensures that valve area opening is gradual and valve can control minute leakage flow.

Experimental Investigation

An experimental test facility is shown in Figure 2. Experimentation is performed using Nitrogen as a working medium. The high-pressure cylinder is used as a pressure source. The pressure of 200 PSI is supplied towards the upstream of the orifice. Two Pressure transducers are used to measure upstream and downstream of the orifice. The upstream pressure transducer was located at a distance exceeding 2.5 D (D is pipe diameter) and the pressure transducer downstream is located at a distance of 8D. A mass flow meter is connected downstream of an orifice. The mass flow meter, in this case, was not able to measure smaller variations in the flow so the mass flow meter is used to check the repeatability and reproducibility of the experiment.

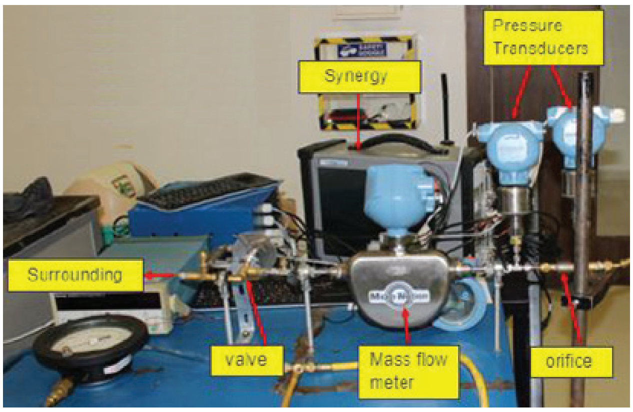

Figure 2. Experimental Test Facility [14].

A valve is connected the downstream of mass flow meter and it was used as a device for applying backpressure another end of the valve was open to the atmosphere. To measure the valve rotation, the angle protractor and pointer assembly are attached to the valve, and the valve shaft was coupled with a potentiometer via a cylindrical coupling. The potentiometer output signal is calibrated for the angle from 00 to 1500 of valve rotation. Signal generated by a Pressure transducer, mass flow meter, and potentiometer is acquired by a data acquisition system (DAQ). The purpose of the experimentation is to develop a mechanism to simulate the variable controlled leak flow which can be mounted on the compressor for performance testing. To quantify variable leaks, a relationship between the valve opening area and leak mass flow rate needs to be established. To study the effect of leakages on compressor performance, an experimental investigation has been performed. In the gas cycle stand, the discharged refrigerant flow from the compressor is circulated via an oil separator and heat exchanger only. The advantage of this type of load stands the phase of refrigerant has remained gases phase so an evaporator is needed which reduces the costs of the gas cycle stand significantly. A schematic of the gas stand is shown in Fig. 3. The different states that the refrigerant assumes during one cycle are shown by the schematic.

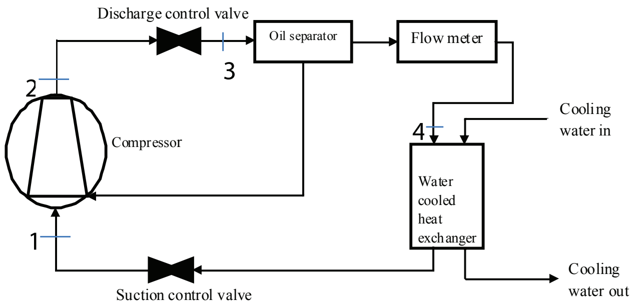

Figure 3. Schematic of gas cycle stand.

A compressor performance test has been performed on a gas cycle stand which allows the measurement of compressor performance compressor power consumption, capacity, discharge line temperature, and suction mass flow. These measurements were used to evaluate the effect of leakages on compressor performance on a macroscopic basis. In addition, pressure transducers and thermocouples are installed to measure additional pressure and temperature values at different locations of the compressor, discharge temperature, cooling water temperature, and pressure. Top, mid and bottom shell temperatures are also measured. Suction mass flow and oil circulation rate are monitored. The operating condition of the compressor gas cycle stand is shown in Table 1. The ambient is maintained at a constant temperature as provided by AHRI 540 standard test conditions.

Table 1. Operating conditions of the compressor

Parameters	Operating Conditions
Evaporating Temperature	45^0F
Condensing Temperature	130^0F
Degree of superheat	20^0F
Degree of subcooling	15^0F
Refrigerant	R410A

INVESTIGATED SCROLL COMPRESSOR

The compressor investigated during research work is a vertical type of scroll compressor, which utilizes a low-pressure shell, i.e. the entire compressor shell is under suction pressure. A compressor is tested for performance at the standard ARI test point. The compressor uses R410A as a working refrigerant fluid. A schematic block diagram of the compressor is shown in Figure 4. The investigated scroll set has an upper scroll as a stationary scroll and a lower scroll as orbiting scroll with orbiting motion provided by 1- a phase AC motor. The upper scroll has a seal for high and low-pressure gas separation and a top plate for sound attenuation from the compressor. The refrigerant R-410A is drawn into the suction chamber in the pump assembly through a suction pipe that enters the scroll from the periphery of the scroll set. Part of the refrigerant is circulated over the motor windings for effective motor cooling causing the refrigerant to be superheated at suction. The refrigerant is compressed in the scroll set assembly and discharged through the discharged opening into the compressor head (top cap) and the compressor head is exposed to the surrounding. From the compressor head, the compressed gas flows through a one-way flow valve to the discharge outlet. The compressor shell body and top cap are welded together. The main bearing housing which supports the scroll set is a 3-point press fitted in the shell. The motor is mounted onto the compressor shaft, driving the orbiting scroll, which is prevented from spinning by an Oldham coupling in the compressor assembly. Lubricating oil is supplied to the bearing assembly through an oil port connected to a drilling port in the shaft. One end of the shaft is immersed in an oil sump at bottom of the compressor. Another end of the shaft is coupled to the bottom orbiting scroll. Since the compressor is of vertical type, the compression region is located at the top and the oil sump is located at the bottom.

MATHEMATICAL MODELLING OF COMPRESSOR CAPACITY LOSS

The refrigerant leaving the evaporator enters the suction line of the compressor. The refrigerant enters the scroll suction pockets along the periphery as shown in Figure 4, which is compressed as it travels from a low-pressure pocket to a high-pressure pocket. The compression process, presence of friction & compression heat loss makes the process polytrophic. Sensible and latent heat removal from refrigerants takes place inside the condenser. The refrigerant is expanded using an expansion device. Low-pressure and low-temperature refrigerants are used to extract heat from the evaporative chamber. Refrigerant leaving the evaporator is dry saturated

Theory

The principle of photo-thermal energy conversion is converting the energy of the incident radiation to thermal energy, solar thermal systems collectors are one of the examples. The energy equation for the solar system collector, accounting for the volumetric heat release, can be written as:

Compressor capacity

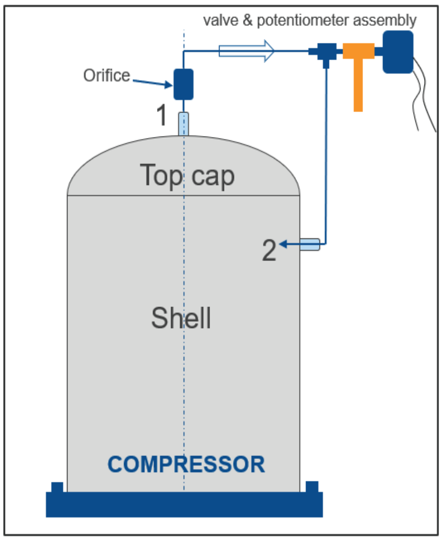

Figure 4. Schematic of leak paths.

$$m^{\cdot}{}_{1} * (h_1 - h_4\ ) \tag{1}$$

$m^{\cdot}{}_1$, is the mass flow rate at compressor suction. Consider the valve potentiometer assembly region as the control volume, Due to the presence of a leak path; the mass flow rate of refrigerant that is circulated through the system ($m^{\cdot}{}_1{}'$) is reduced. Applying the mass conservation for a control volume $(\dot{m}_{in})_{CV} = (\dot{m}_{in})_{CV}$

$$m^{\cdot}{}_1 = m_1{}' + \dot{m}_{leak} \tag{2}$$

Compressor capacity without leak = $\dot{m}_1(h_1 - h_4)$, and compressor capacity with leak=$m_1{}'(h_1 - h_4)$, and compressor capacity loss due to leak = $\dot{m}_1(h_1 - h_4)$ - $m_1{}'(h_1 - h_4)$. Leak gas is added to the incoming compressor suction gas. The temperature of the leaking gas is higher than compared of the suction gas. Due to such temperature difference between suction gas and leak gas, heat interaction occurs in the control volume where both gases mix.

$$\dot{m}_a\left(h + \frac{V^2}{2g} + z\right)_a + \dot{m}_b\left(h + \frac{V^2}{2g} + z\right)_b + Q = \dot{m}_c\left(h + \frac{V^2}{2g} + z\right)_c + W \tag{3}$$

Where, for adiabatic gas mixing, changes in velocity head and datum head are negligible

$$\dot{m}_a h_a + \dot{m}_b h_b = \dot{m}_c h_c \tag{4}$$

$$\dot{m}_1{}'h_1 + \dot{m}_{leak}h_{leak} = \dot{m}_{mix}h_{mix} \tag{5}$$

$$h_{mix} = \left(\frac{\dot{m}_1{}'h_1 + \dot{m}_{leak}h_{leak}}{\dot{m}_{mix}}\right) \tag{6}$$

RESULTS AND DISCUSSION

To study the effect of leakages on compressor performance, an experimental investigation has been performed. In the experimentation, a mechanism was developed for external leak path control of the actual scroll compressor by accompanying a few modifications in the compressor. A compressor performance test has been performed on a Gas cycle stand which allows the measurement of compressor performance compressor power consumption, capacity, discharge line temperature, and suction mass flow. These measurements were used to evaluate the effect of leakages on compressor performance on a macroscopic basis.

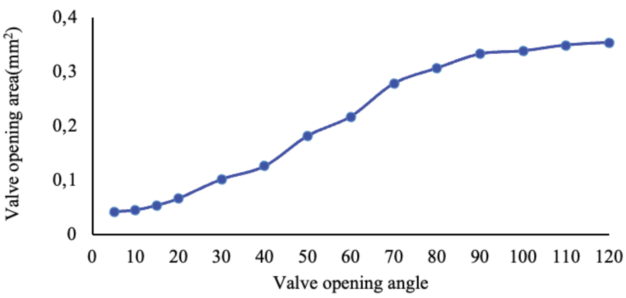

Figure 5. Variation of valve opening area with valve opening angle.

It can be observed from Fig. 5 that the valve area gradually opens with a gradual increase in opening angle. The relationship is observed to be nonlinear. When the 100° mark is reached for valve opening, the mass flow remains the same even after the increase in valve opening angle. This condition represents that the valve opening area is equal to the orifice area at a 100^0 valve opening angle.

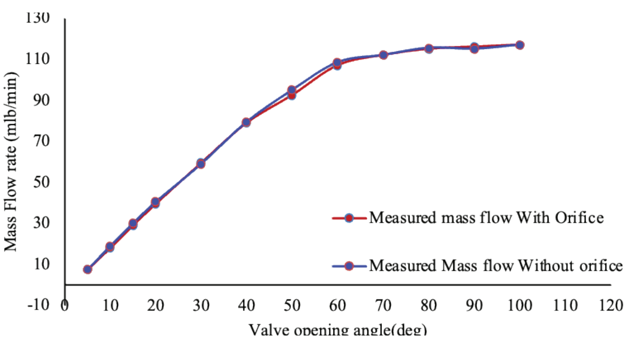

Figure 6. Variation of mass flow concerning valve opening area.

It is observed from Figure 6, which the valve area gradually opens with a gradual increase in opening angle. The relationship is observed to be nonlinear. When the valve opening angle is reached at the 80^0 mark, the mass flow starts to be constant even after an increase in valve opening angle. This condition represents that the valve opening area is equal to the orifice area at a 100^0 valve opening angle. The mass flow meter is attached between the orifice & needle valve. The mass flow meter in this setup is used only to check the repeatability & reproducibility of flow. Pressure drop across mass flow is observed to be very low. Mass flow meter readings are used to check the mass flow rate with & without the orifice (blocked orifice). It is observed that the mass flow rate with and without an orifice is similar. It is concluded that the orifice does not affect leakage mass flow.

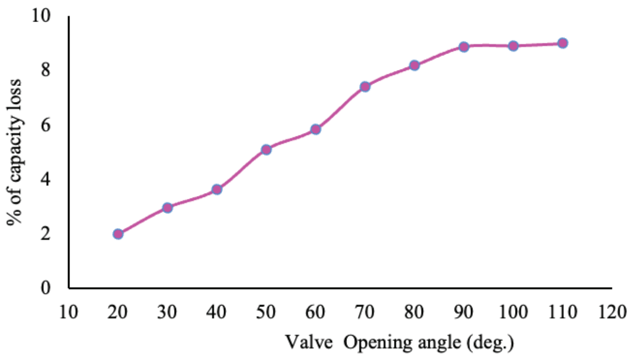

Figure 7. Variation of mass flow concerning valve opening area.

It is observed from Figure 7, which the valve area gradually opens with a gradual increase in opening angle. The relationship is observed to be nonlinear. When the valve opening angle is reached at the 80^0 mark, the mass flow starts to be constant even after an increase in valve opening angle. This condition represents that the valve opening area is equal to the orifice area at a 100^0 valve opening angle. The mass flow meter is attached between the orifice & needle valve. The mass flow meter in this setup is used only to

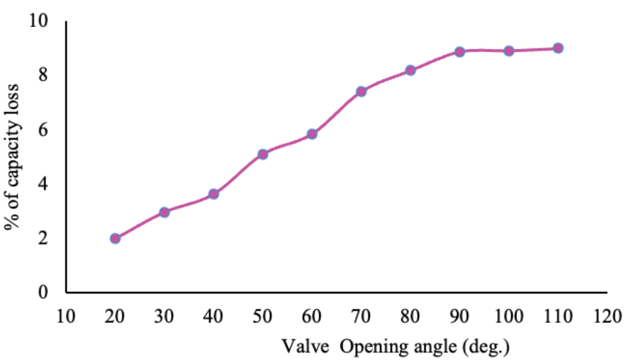

Figure 8. Capacity loss concerning valve opening angle.

check the repeatability & reproducibility of flow. Pressure drop across mass flow is observed to be very low. Mass flow meter readings are used to check the mass flow rate with & without the orifice (blocked orifice). It is observed that the mass flow rate with and without an orifice is similar. It is concluded that the orifice does not affect leakage mass flow.

Experimental results of the percentage of capacity loss with valve opening angle and leak area are represented in Figure 8, it is observed that the percentage of compressor capacity loss with gradual valve opening area nearly follows a linear relationship. It can be observed that the rise in capacity loss is steeper from the valve opening angle of 20^0 to 90^0. The capacity loss line flattens as the valve angle is between 90^0-100^0. The maximum valve area contributing to loss is 0.3 mm^2 in the presence of a leak path; the mass flow rate of refrigerant that is circulated through the system ($\dot{m}_1'$) is reducing from valve opening angle 20^0 to 90^0. This indicates an effective valve opening angle and area affecting compressor capacity loss. The Maximum experimental capacity loss settles at a constant value of 9%.

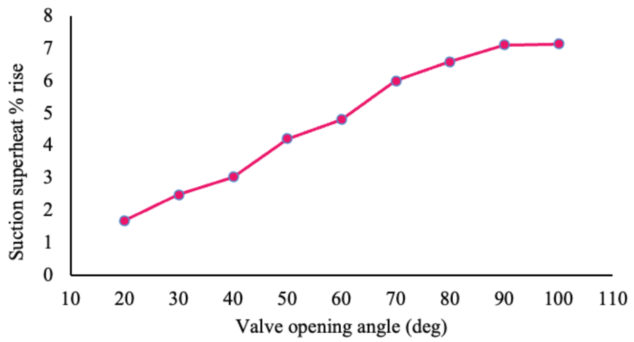

Figure 9. A rise in suction gas superheat concerning valve opening angle.

Before entering the suction chamber, the refrigerant temperature rises by roughly 2 K due to suction gas heating. The greater the temperature difference between the refrigerant and the suction tube, the higher the refrigerant temperature. As a result, when performing a performance analysis, suction gas heating must be considered. Temperature and pressure are essentially unchanged at the end of the suction since there is no leakage or heat transfer. The high-temperature leak gas gets added to the low-pressure region, causing suction to superheat represented by Figure 9. The maximum rise in suction superheat is found to be 7.13% as compared to the condition with no leak at 20° valve opening. A rise in suction superheat is nearly linear and flattens at a 100° valve opening angle because the refrigerant absorbs heat from the scroll and attains the equilibrium condition. This type of superheat by leak gas does not contribute to the refrigeration effect and causes a rise in discharge gas temperature and a reduction in compressor capacity.

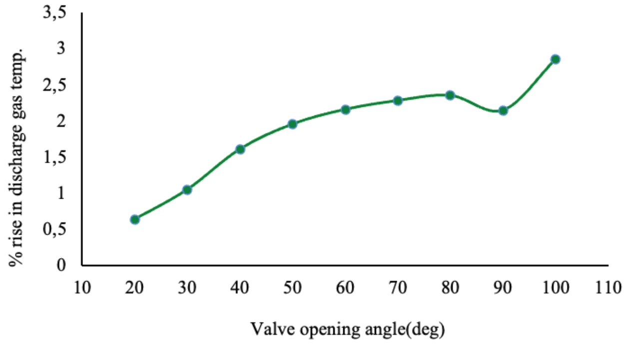

Figure 10. Rise in Discharge gas temp concerning valve opening angle.

The experimental rise in discharge line temperature (DLT) with valve opening angle is represented in Figure 10. Maximum DLT rise is observed to be 2.75 % at 1000 valve opening angle and 0.3 mm^2 leak area. DLT rise at valve opening angle in the range of 400 - 800 is observed to be uniform. Steep rise is observed in the range of 300 – 400 valve opening angle.

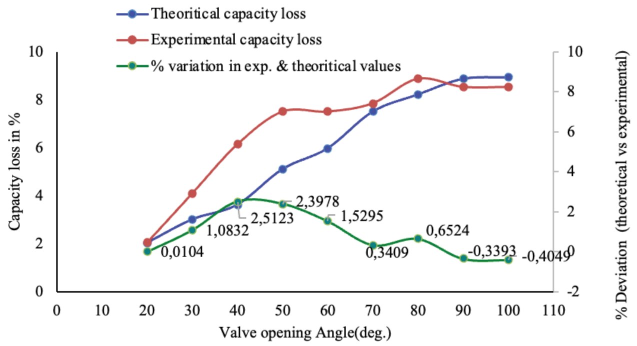

Figure 11. Comparison of compressor capacity loss concerning valve opening angle.

Figure 11 shows the comparison of theoretical and experimental compressor capacity loss. A valve opening angle beyond 60°, the experimental capacity loss agrees with the calculated theoretical loss. Also, it coincides at an angle of 20°. The maximum deviation in theoretical and experimental capacity loss is observed at smaller values of valve opening angle (2.39 % at 50° valve opening). The reason for deviation at smaller valve opening angles is the time taken by the compressor for stabilization in this range. It is higher at the lower valve opening angle as compared to a higher valve opening angle. The deviation in readings can be reduced further by increasing the cycle time for readings at a lower angle and taking readings at the interval of 5° of valve opening in this range. This ensures the operational stability of the compressor and a more detailed trend of capacity loss variation at lower valve opening angles.

CONCLUSION

As a part of research work, leak flows are modeled mathematically and experimentally. The observed effect of leak gas on compressor performance and compressor operating parameters such as capacity loss, DLT rise, and discharge gas temperature are presented. The effects of leakage on compressor performance parameters are studied analytically & experimentally. The same approach can be applied for different leak paths by providing both leak paths (back pressure to low & high to low pressure) on the same compressor. The effect of multiple combinations of leak flow rates from both leak paths on compressor performance can be analyzed. Suction gas temperature rise due to the addition of leak gas at scroll suction, discharge gas temperature rise & rise in DLT can be determined using CFD analysis which will provide great insight into the process of gas mixing & temperature variation along scroll suction & discharge line. The major conclusions from the analysis are presented as follows.

1. The leak control valve has an effective opening area of up to 0.33 mm2 and an opening angle of 1000, beyond which leak flow remains constant and has no effect on compressor performance.
2. The use of a flow restriction device (here orifice) does not affect leak mass flow and is verified experimentally.
3. Maximum capacity loss is 8.9 % at a leak area of 0.33 mm2 or 100° valve opening angle.
4. The maximum percentage rise in suction superheat is observed to be 7.13% at the maximum effective valve opening area, whereas the rise in DLT lies in the range of 0.8% – 2.75% over the entire range of effective leak area.

NOMENCLATURE

$\dot{m}_1$	Mass flow displaced by a compressor, kg
h_1	Enthalpy at compressor suction, kJ/kg
h_4	Enthalpy at the evaporator inlet, kJ/kg
ρ_1	Refrigerant density at the suction, kg/m^3
V_d	Scroll displacement per unit revolution, m^3 / N
N	Rotational shaft speed, rpm.
Q	Rate of heat transfer across control volume, W
W	Rate of work done, W
t	Time, sec
$\frac{V^2}{2g}$	Kinetic energy per unit mass, kJ/kg
Z	Elevation head, m
T	Temperature, °C
u	Velocity of the medium, m/sec.
V	Volume occupied the nanosuspension, m^3

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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