Improvement Of Energy Efficiency Ratio Of Refrigerant Compressor

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Abstract: - At first glance, calculating the efficiency of any type of compression system seems to be straight forward Comparing the work required of an ideal compression process with the work required of an actual compression process. But the difficulty lies in defining appropriate system boundaries that include losses associated with the compression process. Generally the losses that occur in the refrigerant compressor are Electric motor losses, Mechanical power distribution losses, Heat losses, Cycle losses, Volumetric efficiency losses.The EER (Energy Efficiency Ratio) of the refrigeration and Air Conditioning system is improved by improving the cooling capacity (or) by reducing the power input to the compressor and by reducing the losses. In this project the parameters affecting the performance of reciprocating compressor are analyzed and appropriate modifications have been made which resulted in improving the performance (or) EER of the system.The modifications made are Surface finish improvements and usage of low viscosity compressor oils to reduce frictional losses between the mating parts. Reducing the displacement volume and suction valve leaf thickness to reduce the volumetric losses. Finally changing the suction head material from metal to plastic so as to reduce the heat losses and thereby improving the thermal efficiency

INTRODUCTION

Refrigeration is the process of removing heat from an enclosed space or from a substance in order to lower the temperature below the surrounding temperature, wherein the cooling medium, or refrigerant, goes through a cycle so that it can be recovered for reuse. The commonly used refrigeration systems are vapour- compression, vapour absorption, steam-jet or steam- ejector, and air refrigeration. The Vapour Compression Refrigeration Cycle is a process that cools an enclosed space to a temperature lower than the surroundings. To accomplish this, heat must be removed from the enclosed space and dissipated into the surroundings. During the cycle, a substance called the refrigerant circulates continuously through four stages. The stage such as Evaporation, Compression , Condensation, and Expansion . It has many applications, including, but not limited to: household refrigerators, industrial freezers, cryogenics, air conditioning, and heat pumps. The refrigerant is a heat carrying medium which during the cycle (that is compression, condensation, expansion and evaporation) in the refrigeration system absorb heat from a low temperature source and discard the heat so absorbed to a higher temperature sink. There are many refrigerants which are most commonly used refrigerants in vapour compression refrigeration system are R-11,R-12,R-13,R-22, R-13, R-717 .

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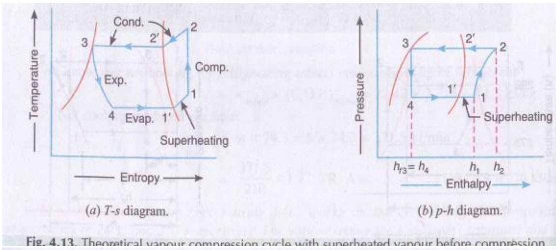
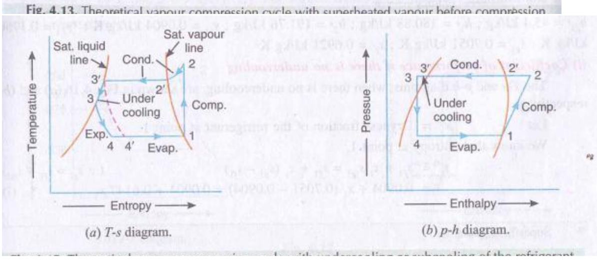
VAPOUR COMPRESSION CYCLE WITH DRY SATURATED VAPOUR AFTER COMPRESSION

A vapour compression cycle with dry saturated vapour compression is shown on T-s and p-h diagrams in Fig.2.1.1 (a) and (b) respectively. At Compression process, let T_1, p_1 and s_1, be the temperature, pressure and entropy of the vapour refrigerant respectively. The vapour refrigerant at low pressure p_1 and temperature T_1 is compressed isentropically to dry saturated vapour as shown by the vertical line 1-2 on T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rises from p_1 to p_2 and T_1 to T_2 respectively. At Condensing process, The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure p2 and temperature T2, as shown by the horizontal line 2-3 on T-s and p-h diagrams.At Expansion process, The liquid refrigerant at pressure $p_3 = p_2$ and temperature $T_3 = T_2$ is expanded by throttling process through the expansion valve to a low pressure $p_4 = p_1$ and temperature $T_4 = T_1$, as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on p-h diagram.At Vaporising process: The liquid vapour mixture of the refrigerant at pressure $p_4 = p_1$ and temperature $T_4 = T_1$ is evaporated and changed into vapour refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams.

WITH SUPERHEATED VAPOUR BEFORE COMPRESSION:





A vapour compression cycle with superheated vapour before compression is shown on T-s and p-h diagrams in Fig. 2.1.2 (a) and (b) respectively. In this cycle, the evaporation starts at point 4 and continues up to point 1^1, when it is dry saturated. The vapour is now superheated before entering the compressor up to the point 1.

WITH UNDER COOLING OR SUBCOOLING OF REFRIGERANT:

Some times the refrigerant, after condensation process $2^1 - 3^1$, is cooled below the saturation temperature (T_3^1) before expansion by throttling. Such a process is called under cooling .The ultimate effect of the undercooling is to increase the value of coefficient of performance under the same set of conditions. And also increased by adopting both the superheating and undercooling process .

EXPERIMENTAL SETUP

The experimental test unit is an integral of pressure gauges, thermocouples, flow meters, energy meter, voltmeter, to measure the temperatures and pressures at various places to measure compressor and condenser inlet and outlet pressures and temperatures, mass flow rate of refrigerant in the circuit connected before evaporator and Power consumption. All the parameters are displayed on the Dash board as shown in the Fig.5. The dash board displays readings like temperatures, pressures, mass flow rate, power consumption, voltage. There are provisions to measure the compressor top shell temperature, bottom shell temperature, and the air flow around the compressor and compressor cabin temperature measurement. There are special drivers to convert the measured analog readings to digital form. The calorimeter test rig uses Cal 4 software. There is a provision to save and take a print out of the results from the calorimeter test rig.

METHODS OF EXPERIMENT

Connecting the Compressor, Evacuation and Charging Refrigerant

The compressor after fixing the thermocouples and pressure gauges is kept in the compressor chamber of the calorimeter. The suction and discharge tube connections are made depending upon the capacity of the compressor. Wells are provided for inserting return gas and discharge temperatures RTD sensors. The flexible hoses with threaded adaptors are provided for connecting the compressor quickly in the chamber to the refrigeration system. After connecting the compressor to the primary refrigeration system, the following procedure is followed:1. All the valves provided in the primary refrigeration system are fully opened. The valves are provided at the following locations: Compressor Discharge and Compressor Suction, Valve in the liquid line after the Mass Flow Sensor, Valve at the receiver outlet ,Valve at the drier outlet.2. The system is pressurized with dry nitrogen up to a pressure of 25 bar. The joints made are checked for leakage. While applying the nitrogen pressure the suction pressure gauge is closed by operating the Hand Shutoff valve provided below the gauge to prevent the damage to the gauge on account of the very high nitrogen pressure. 3.The nitrogen pressure is released from the primary system and a good quality two stage, rotary, high vacuum pump is connected at both, the suction and discharge process connections provided in the compressor chamber. The entire system is evacuated to at least 75 microns of pressure. 4. Vacuum is broken with the refrigerant system. There are three sight glasses in the liquid line. The first one is at the receiver, the second is just before the mass flow sensor and the third is before the expansion valve, that is, inside the secondary pot chamber.

METHODS FOLLOWED TO IMPROVE COMPRESSOR EER

In this experiment various changes were made to improve the EER of the compressor by reducing the various losses like Mechanical losses (Surface Finish improvements, Fine boring, Pin grinding, Journal grinding, Crank bore grinding,Cylinder and Piston grinding) ,Thermal losses (changing suction head material from metal to plastic) and volumetric losses(suction, design optimization and displacement).Views of the modified parts are shown in Fig. 5.2.1

Fig.5.2.1 Existing model Suction head, Modified model Suction head and Suction Valve leaf.

TESTING THE COMPRESSOR

After charging the refrigerant into the compressor the connection of the compressor in the calorimeter test rig as shown in the Fig.5.4. The calorimeter is connected to the inverter to get the different load conditions. The thermo couples are connected to digital unit and pressure probes are directly connected to the pressure gauges. Compressor is started and the experiment is conducted in the different load conditions. The readings are taken after attaining the steady state in every load step. All the readings at various loads are tabulated. Another experiment is conducted to measure the shell temperatures at various locations on the compressor shell. A production AW compressor is taken and the horizontal lines and vertical lines are marked using marker on the outer side of the shell. At the intersections of horizontal lines and vertical lines the temperatures are measured. As described earlier compressor is fitted to the calorimeter and charged. This experiment is also conducted at various loads. All the temperature readings over the shell are taken by optical pyrometer at the intersections of the horizontal and vertical lines. All these readings are tabulated in the table to determine the capacity, volumetric efficiency, energy efficiency ratio and mechanical, thermal information such as power loss requirements for the compressor.

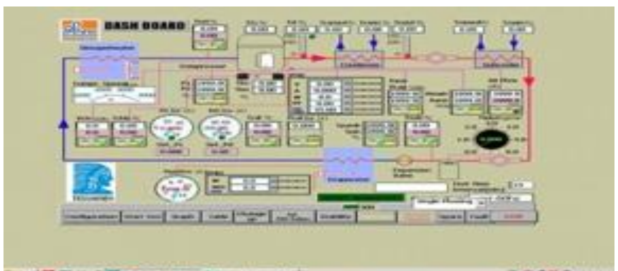


Fig. 5 Dash Board diagram of Calorimeter Test Rig
Fig. 5.4 Compressor performance testing unit

After testing the compressor in Calorimeter lab the following readings are noted in table 1

Pressures	Actual value Bar	Set value bar	Actual value Bar	Set value bar
	Existing of Model AWZ5528EXN		Modified Model AWT5528EXN	
Suction	6.2366	6.254	6.2314	6.254
Discharge	21.458	21.46	21.458	21.46
Calorimeter out	6.248	6.254	6.248	6.254
Temperatures ^{0}C	Actual value	Set value	Actual value	Set value
Evaporating	7.11	7.20	7.08	7.20
Condensing	54.39	54.40	54.39	54.40
Degree of superheat	27.91	27.80	27.90	27.80
Degree of sub-cooling	8.27	8.30	8.27	8.30
Return gas temperature	35.02	35.0	35.02	35.0
Liquid to expansion valve	46.12	46.1	46.12	46.1
Calorimeter outlet temperature	35.02	35.0	35.02	35.0
Compressor chamber ambient	35.0	35.0	35.0	35.0
Top shell	48.46	48.50	46.46	48.50
Bottom shell	60.48	62.1	60.48	62.1
Middle at shell	50.94	51.0	50.94	51.0
Discharge Line	106.3	108.0	106.22	108.0
Electrical Parameter	Actual value	Set value	Actual value	Set value
Frequency HZ	60.01	60.0	60.01	60.0
Compressor voltage V	230.44	230	229.7	230
Compressor current 1	13.53	13	11.43	13
Compressor power	2950W	2950W	2604W	2604W
Calorimeter heater energy	8698W	8698W	8233W	8233W
ECR	346.52	346.52V	329.96V	329.96

results Existing of model AWZ5528EXN

Location	Temperatures(°C)	Temperatures(°C)
Suction Inlet	62.77	48.21
Suction Cavity	67.13	46.14
Discharge Muffler inside	129.326	102.02
Discharge muffler surface	108.99	80.59
Cylinder head inside	130.15	105.29
Cylinder head surface	110.85	96.19
Winding temperature	84.76	67.12
Oil temperature	93.50	82.43
Lamination (Stack)	94.10	76.33
Crank case surface	83.58	87.21
Suction cavity	6.179 bar	6.25 bar
Cylinder head discharge pressure	21.737 bar	21.76 bar

Table 6.2 AWZ5528EXN Compressor test report

Performance Calculations of Model AWT5528EXN:

Stages	Suction	Discharge	Evaporation	Condensing	Calorimeter out
Pressure (bars)	6.23	21.48			
Temperature °C	35	46.12	7.2	54.3	35

Capacity	Enthalpy kJ / kg	suction	discharge	subcooling
		428.8	464.3	257.6
	Corrected		Uncorrected	8698 Watts

Mass of Refrigerant Pumped	Enthalpy of gas out of calorimeter tank h1 = 428.83 kJ/kg	Enthalpy of liquid (saturated) at 46.12°C h4 = 257.6 kJ/kg	Net Refrigeration effect (h1 h4) = 171.23 kJ/kg	Uncorrected Capacity / Net Refrigeration effect = Mass of Refrigerant Pumped 8.698 kJ/sec / 171.23 kJ/kg = 0.05079 kg/sec.
Liquid entering expansion valve	Enthalpy of suction gas h1 = 428.843 kJ/kg	Enthalpy of liquid (saturated) at 46.1°C h4 = 257.42kJ/kg	Net refrigeration effect (h1 h4) = 171.41 kJ/kg	Mass of Refrigerant Pumped × Net Refrigeration effect = Corrected capacity 0.05079 kg/sec × 171.23 kJ/kg = 8.696 KW

Energy Efficiency Ratio Corrected capacity / Compressor power Watts = EER
8241.4 Watts / 2604 Watts = 3.165
Power factor Compressor power Watts /Compressor amps / Compressor volts = Power factor
2604 Watts / 11.43 amps / 229.7 volts = 0.99

	New model AWT5528EXN			old model AWT5528EXN		
	Heat generated	Frictional Torque N m	Powerless Watts	Heat generated	Frictional Torque N m	Powerless Watts
between piston and cylinder	7.268	1.14016	65.559	9.436	2.6013	149.578
connecting rod and piston pin	5.8862	0.10236	36.981	10.943	0.19016	68.702
crank shaft and connecting rod	10.90	0.18	68	20.2	0.35	127.316
journal and bearing	0.15	2.722 × 10^3	0.98	0.2908	5.0574 × 10^3	1.82715
bearing face and thrust face of crank shaft	2.369	0.01590	5.74	8.467	0.02948	10.65

al input = $\eta_{electrical}$ × power input = 0.84 × 2604 = 2187.36 Watts

$$\eta_{mech} = \times 100 = \frac{\text{mechanical input} - \text{power loss}}{\text{mechanical input}}$$

$$\frac{2187.36 - 177.766}{2360} \times 100 = 91.87\ \%$$

FORCES ON VALVES

Valves	Original pressure P bar	Actual pressure P_s = P − P $_{diff}$	Actual pressure ratio
Suction	6.2314	5.7184	$\frac{21.6605}{5.7184}$
Discharge	21.458	21.6605	3.787

Total Clearance volume V_C = 1737.028 mm^3 and
Diameter of the cylinder D_c = 39.3713 mm
Stroke Length L = 18.59 mm
Displacement volume $V_D = \pi/4 \times d^2 \times L = \times \frac{\pi}{4}$
$39.3713^2 \times 18.59 = 22635.59\ mm^3$

Clearance Factor $C = \frac{V_C}{V_D} = \frac{1737.028}{22635.59} = 0.07674$

Polytropic index n = 1.12

VOLUMETRIC EFFICIENCY:

$$\eta_{vol} = (1+C) \times \left(\frac{P_S}{P_1}\right)^{1/n} - C \times \left(\frac{P_d}{P_1}\right)^{1/n} - 0.01 \times \left(\frac{P_d}{P_s}\right)$$

$$= (1+0.07674) \times \left(\frac{5.7184}{6.2314}\right)^{\frac{1}{1.12}} 0.07674 \times \left(\frac{21.6605}{6.2314}\right)^{\frac{1}{1.12}} - 0.01 \times \left(\frac{21.6605}{5.7184}\right)$$

$$= 0.9972 - 0.2334 - 0.03787$$

$$= 0.72.59$$

$$\eta_{vol} = 72.59\ \%$$

	Volume mm^3	Workdone N − m	Power Watts
Suction	18667.96	10.6751	613.812
Compression	16951.336	19.954	1147.355
Discharge	5684.256	12.312	707.963
Re expansion	3967.632	4.6668	268.341

POWER OUTPUT:

P_{out} = Compressor power + Discharge power – Re expansion power – Suction power
= 1147.355 + 707.963 – 268.341 – 613.812
= 973.165 Watts

THERMAL EFFICIENCY CALCULATION:

Clearance volume $V_C = V_3 = 1737.028\ mm^3$
Displacement volume $V_D = 22635.59\ mm^3$

$$V_1 = V_C + V_D$$

$$= 1737.028 + 22635.59 = 24372.62\ mm^3$$

$$V_2 = \left(\frac{P_s}{P_d}\right)^{\frac{1}{n}} \times V_1$$

$$= \left(\frac{5.7184}{21.6605}\right)^{\frac{1}{1.12}} \times 24372.62 = 7421.284\ mm^3$$

$$V_4 = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} \times V_3 = \left(\frac{21.6605}{5.7184}\right)^{\frac{1}{1.12}} \times 1737.028$$

$= 5704.66\ mm^3$

Isothermal Work = Ps × V_1 × ln $\left(\frac{V_1}{V_{21}}\right)$

$$= 5.7184 \times 24372.62 \times 10^{-4} \times \ln\left(\frac{24372.62}{7077.805}\right)$$

$$= 17.2333\ N - m$$

Isentropic Work = $\frac{n}{n-1} \times P_S \times V_1 \times \left[\left(\frac{P_d}{P_s}\right)^{\frac{n}{n-1}} - 1\right]$

$$= \frac{1.12}{1.12-1} \times 5.7184 \times 24372.62 \times 10^{-4} \times \left[\left(\frac{21.6605}{5.7184}\right)^{\frac{1.12}{1.12-1}} - 1\right] = 19.9512\ N - m$$

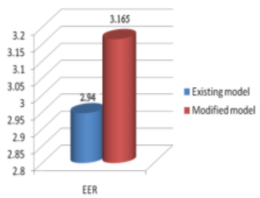
$$\eta\ \text{thermal} = \frac{\text{Isothermal Work}}{\text{Isentropic Work}} \times 100 = \frac{17.2333}{19.9512} \times 100 =$$

86.37 %=

GRAPHS

1. Table and Graph representing the Power loss between various mating parts of the Existing model with Modified model

	Existing model	Modified model
E.E.R	2.947	3.165



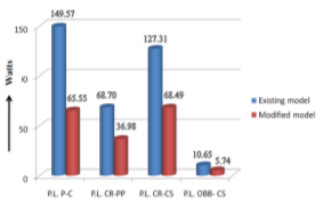
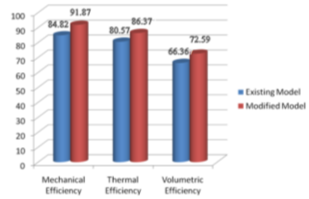


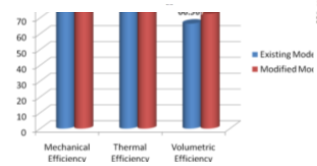
Table and Graph representing the Power loss between various mating parts of the Existing model with Modified model

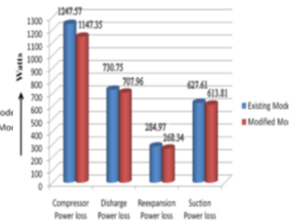
Power loss in various	Existing model	Modified model
Piston – Cylinder	149.57	65.55
Connecting rod –	68.70	36.98
Connecting rod –	127.31	68.49
O.B.B – Crank shaft	10.65	5.74

3. **Table** and Graph representing the Frictional Torque between various mating parts of the Existing model with Modified model

Frictional torque between various matting	Existing model	Modified model
Piston – Cylinder	2.6	1.14
Connecting rod – Piston pin	0.19	0.102
Connecting rod – Crank shaft	0.35	0.189
O.B.B – Crank shaft Thrust	0.029	0.015







4. Table and Graph representing the variation of various efficiencies of the Existing model with Modified model.

Efficiency	Existing model	Modified model
Mechanical	84.82%	91.87%
Thermal	80.57%	86.37%
Volumetric	66.36%	72.59%

5. Table and Graph representing the Power loss at various locations of the VCR cycle of the Existing model with Modified model

Power loss in VCR cycle (Watts)	Existing model	Modified model
Compressor power loss	1247.57	1147.35
Discharge power loss	730.35	707.96
Re- expansion power loss	284.97	268.34
Suction power loss	627.61	613.81

RESULTS AND DISCUSSIONS

With an aim to develop and test a better compressor a number of modifications are thought of. A few modifications are done to the existing compressor. The following are the modifications.1). Suction path is made with plastic (G.E.Valox – 420) to reduce the suction super heat temperature which reduced by 14.56^{0}C by changing the suction muffler from metal to plastic the heat transfer rate is reduced from the cylinder head to the suction muffler, also the flow patterns are changed.2). Suction head is changed to reduce the super heat temperature there by EER is increased. And thermal efficiency is improved by 5.80%.3). Compressor displacement is modified and Pressure drops at suction were reduced by reducing the suction valve leaf thickness to increase the volumetric efficiency and capacity of the refrigerant compressor. It is found that the volumetric efficiency got improved by 6.23%.4). Because of the various losses reduction methods like surface finish improvements, usage of low viscosity compressor oil and selective assembly clearances Mechanical efficiency is improved by 7.05%.From the results the EER of the compressor model AWZ5528EXN (existing model) was found to be 2.947 and for the compressor model AWT5528EXN (Modified model) was found to be 3.165. This shows that after the reduction of losses the EER of the reciprocating compressor is increased by 6.88 %.

Table 7.1Comparive results of Existing model and **Modified model**

	Existing Model AWZ5528EXN	Modified model AWT5528EXN
E.E.R	2.947	3.165
Mechanical efficiency	84.82%	91.87%
Thermal efficiency	80.57%	86.37%
Volumetric efficiency	66.36%	72.59%
Power loss between moving parts	358.073	177.766

CONCLUSION
From the experimental results, it can be concluded that a high efficiency compressor can be developed by applying design modifications which reduces the various losses such as mechanical losses, volumetric losses, and thermal losses. For the existing compressor AWZ5528EXN under consideration the EER was found to be 2.947, after applying design modifications reductions in losses found to be approximately 6.88% and the EER of the compressor was found to improve to 3.165. From the results obtained it is observed that the suction gas is absorbing more heat and getting superheated. It is due to very high discharge plenum and oil temperatures surrounding the cylinder head. By reducing the suction gas superheat temperature EER is improved. Hence, it is concluded that the energy efficiency ratio (EER) can be increased by minimizing the friction losses, thermal losses and volumetric losses which is attained by the proposed design modifications.

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