Analysis of VCRS by replacement of reciprocating compressor with screw compressor

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Abstract

Refrigeration is one of the buzzing words in society as well as in industry to keep the items at lower temperature without any harm. Further it is designed in various ways for particular purpose. It has been reported that it consumes more energy to provide the comfort ability compressor is one of the key components in refrigeration that consumes more energy while running of the refrigeration plant. Many researchers are reported to be feasible choices for many types of compressor including reciprocating, centrifugal and screw compressor so that the less are consumed. Hence, the more energy saving is possible in forms of electricity. The main advantages of using this alternative compressor are its renewability, compatibility and lesser power consumption. It is environmentally friendly alternative to conventional for use in existing system. It is expected that the replacement of compressor will give a new insight to the refrigeration plant. To keep all above in the mind the research objective will be set for the present research work.

Keywords

Power consumption , Screw compressor , Reciprocating compressor , Energy saving , VCRS.

1.Introduction

A compressor is the heart of vapour compressor system. It pump and circulates refrigerant through the just as the heart pump and circulates blood through the body. In general, a compressor performs following function; It remove low temperature and low pressure vapour from the cooling coil through the line called suction line. It compresses this vapour by increasing the pressure and temperature resulting in an increase of boiling point of refrigerant. It discharge the vapour in high temperature and pressure from the condenser through the line called discharge line, The refrigeration is different from those used in other application, such as or gas compressor. The function of the air gas compressor is totally unrelated to use to which compressed air or gas is put to. So their capacity, susceptibility to break down etc., are governed by what happens to compressed fluid once it is discharge to the receiver, further on the other hand, the function of refrigerator compressor, being an integral part of the system, is related to related to other components. Hence its capacity, life, breakdown, etc. are very much influenced by the performance of other components like condenser, evaporator, throttling device etc.

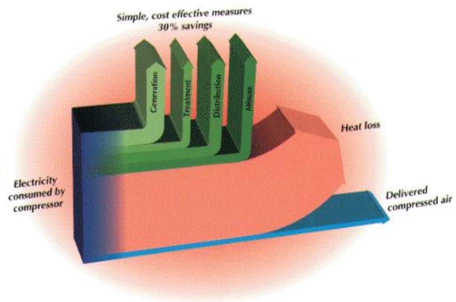

Figure 1 Compressor Diagram

1.1Refrigeration systems

Refrigeration systems are also used for providing cooling and dehumidification in the pharmaceutical industry to maintain the specific temperature of medicine. The first refrigeration systems were used for industrial as well as comfort air conditioning. Eastman Kodak installed the first air conditioning system in 1891 in Rochester, New York for the storage of photographic films. An air conditioning system was installed in a printing press in 1902 and in a telephone exchange in Hamburg in 1904. Many systems were installed in tobacco and textile factories around 1900. The first domestic air conditioning system was installed in a house in Frankfurt in 1894. A private library in St Louis, USA was air conditioned in 1895, and a casino was air conditioned in Monte Carlo in 1901. Efforts have also been made to air condition passenger rail coaches using ice. The widespread development of air conditioning is attributed to the American scientist and industrialist Willis Carrier. Carrier studied the control of humidity in 1902 and designed a central air conditioning plant using air washer in 1904. Due to the pioneering efforts of Carrier and also due to simultaneous development of different components and controls, air conditioning quickly became very popular, especially after 1923. At present comfort air conditioning is widely used in residences, offices, commercial buildings, air ports, hospitals and in mobile applications such as rail coaches, automobiles, aircrafts etc. Industrial air conditioning is largely responsible for the growth of modern electronic, pharmaceutical, chemical industries etc. Most of the present day air conditioning systems use either a vapour compression refrigeration system or a vapour absorption refrigeration system. The capacities vary from few kilowatts to megawatts. Figure 1.1 shows the basic components of a vapour compression refrigeration system. As shown in the figure the basic system consists of an evaporator, compressor, condenser and an expansion valve. The refrigeration effect is obtained in the cold region as heat is extracted by the vaporization of refrigerant in the evaporator. The refrigerant vapour from the evaporator is compressed in the compressor to a high pressure at which its saturation temperature is greater than the ambient or any other heat sink. Hence when the high pressure, high temperature refrigerant flows through the condenser, condensation of the vapour into liquid takes place by heat rejection to the heat sink. To complete the cycle, the high pressure liquid is made to flow through an expansion valve. In the expansion valve the pressure and temperature of the refrigerant decrease. This low pressure and low temperature refrigerant vapour evaporates in the evaporator taking heat from the cold region. It should be observed that the system operates on a closed cycle. The system requires input in the form of mechanical work. It extracts heat from a cold space and rejects heat to a high temperature heat sink. A refrigeration system can also be used as a heat pump, in which the useful output is the high temperature heat rejected at the condenser. Alternatively, a refrigeration system can be used for providing cooling in summer and heating in winter. Such systems have been built and are available now.

1.2Compressor

A compressor is the heart of vapour compressor system. It pump and circulates refrigerant through the just as the heart pump and circulates blood through the body. In general, a compressor performs following function; it removes low temperature and low pressure vapours from the cooling coil through the line called suction line. It compresses these vapours by increasing the pressure and temperature resulting in an increase of boiling point of refrigerant. It discharge the vapours in high temperature and pressure from the condenser through the line called discharge line, The refrigeration is different from those used in other application, such as or gas compressor. The function of the air gas compressor is totally unrelated to used to which compressed air or gas is put to. So their capacity, susceptibility to break down etc., are governed by what happens to compressed fluid once it is discharge to the receiver on the other hand, the function of refrigerator compressor , being an integral part of the system, is related to related to other components. Hence its capacity, life, breakdown, etc. are very much influenced by the performance of other components like condenser, evaporator, throttling device etc.

A semi-hermetic screw compressor is provided to ensure high operational efficiency and reliable performance. Capacity control is achieved through slide valve. The compressor is a positive displacement type characterized by two helically grooved rotors, which are manufactured from forged steel. The 50 Hz motor operates at 2975 rpm to directly drive the male rotor, which in turn drives the female rotor on a light film of oil. Each compressor is direct drive, semi-hermetic, rotary twin screw type and includes the following items: Two screw rotors, manufactured from forged steel. Cast iron compressor housing precisely machined, a built-in high efficient oil separator an internal discharge check valve can prevent rotor backspin during shutdown. A suction vapour cooled, high efficient and reliable semi hermetic motor has overload protection: thermostat and current overload protection. A suction vapour screen and a serviceable oil filter are installed in the compressor housing. Refrigerant vapour is sucked into the void created by the unmeshing of the five lobed male and seven lobed female rotors. Further meshing of the rotors closes the rotor threads to the suction port and progressively compresses the vapour in an axial direction to the discharge port. The vapour is compressed in volume and increased in pressure before exiting at a designed volume at the discharge end of the rotor casing. Since the intake and discharge cycles overlap, a resulting smooth flow of vapour is maintained.

2.Literature review

work on reciprocating compressor worldwide, there are no standard guidelines for the correct measurement and calculation of turbocharger maps at low speeds. In collaboration with a French automotive manufacturer, a special method was therefore designed and applied within the laboratory LGP2ES at Cnam Paris in order to obtain the compressor low speed map. A special torque meter was fitted in a cold turbocharger test bench, affording measurements from 30,000 rpm to 120,000 rpm[1-7]. The experimental results presented in this paper show the combined effect of the lubricating oil temperature and pressure on the compressor performance, expressed in terms of compression ratio, compressor power, isentropic efficiency and mechanical efficiency. These results afford a better estimation of the compressor map at low speeds. Author suggested optimize the efficiency of reciprocating compressor.

This paper deals with the design and analysis of screw compressor. The twin-screw compressor is a positive displacement machine used for compressing air to moderate pressures. It comprises of a pair of intermeshing rotors with helical grooves machined on them, contained in a casing which fits closely around them. The rotors and casing are separated by very small clearances. The rapid acceptance of screw compressors in various industries over the past thirty years is due to their relatively high rotational speeds compared to other types of positive displacement machines which make them compact, their ability to maintain high efficiencies overa wide range of operating pressures and flow rates and their long service life and high reliability. Every time generation of different profiles and evaluate performance of those profiles by experiments is very difficult and these are expensive and time taking process. By using CFD can find out performance of different profiles easier. A suitable procedure for optimization of the screw compressor shape, size, and dimension is described here, which results in the most appropriate design. Compressors thus designed achieve higher delivery rates and better efficiencies than those using traditional approaches[6-9].

Screw compressors are at present a widely used means of compressing air, process gas and refrigerants. Computer simulation programs, verified by experimental data and improved by recent advances in mathematical analysis, form a powerful tool for process analysis within these machines and hence for a comprehensive check of innovative suggestions and design optimization. One consequence of this is that screw rotors lobe profiles have been developed over the past few years which lead to enhanced machine performance. Although efficient operation of screw compressors is mainly dependent on the rotor profile and clearance distribution, other components, such as housing ports, bearings, seals and the lubrication system must be designed to take full advantage of their potential if maximum performance gains are to be achieved. A review of the most recent achievements in application of innovative details into the screw compressor practice is given at the end of this paper[10-18].

During the last decade, substantial research activities have been undertaken regarding refrigeration cycles and systems with particular emphasis on the replacement of refrigerants like R134a by refrigerants like hydrocarbons which have negligible GWP (Global Warming Potential).Besides using eco-friendly refrigerants, thrust has been given upon devising methods to increase the efficiency of the refrigeration cycle/system, which will also contribute to reducing emission of GHG (Green House Gases). This paper provides a review of the efforts to replace the HFCs (hydro fluro carbons) which are harmful to the environment.

Recently in most of the studies meso scale vapour compression refrigerator is integrated in high power microelectronic packaging and it offers a new cooling solution. Thermodynamic efficiency of such system depends on its operating temperatures. However, important practical issues like system design, sizing, cost, safety, reliability etc. depends on the type of refrigerant selected for given application. Now a days, global warming and ozone layer depletion are the prior issues to be considered while selecting the refrigerant. This work presents a comparison between four refrigerants and finding out the most suitable one for miniature refrigerator. In this work it is found that NH_3 gives the highest COP but because of its drawbacks like corrosive nature, fouling smell and highly toxic nature R134a is selected for the system[19-20].

In his paper the cycles' performances are analyzed for the viewpoints of both the first and second laws of thermodynamics. The waste heat from intercooler and pre-cooler of the gas turbine- modular Helium reactor (GT-MHR) is utilized to drive organic Rankine and ejector refrigeration cycles for performance enhancement, in three different configurations. Energy analysis revealed that the compressor or recuperator has the second highest energy destruction after the reactor.

proposed in this study to improve the entrainment performance of thermal vapour compressors (TVCs) that are widely used in multi-effect distillation (MED) desalination systems by preheating entrained vapour. In order to confirm its effectiveness and study the influences of the entrained vapour superheat on the TVC entrainment ratio, a four-effect-parallel-flow MED-TVC experimental system was set up. Due to the poor performance of the TVC used in this study, further verification experiments of the proposed method need to be conducted in future work.

a conventional heat pump exhibits performance degradation even though larger heating capacity is needed as the outdoor temperature declines. As a way to prevent the performance degradation, a heat pump with an inverter-driven two-stage rotary compressor and vapor injection (VI) cycle was investigated for an air-to-water heat pump (AWHP) system employing a flash tank. The volume ratio of two cylinder of a two-stage rotary compressor has significant effect on the performance of the AWHP so that it was experimentally investigated. Based on this result, a two-stage rotary compressor designed with an optimized volume ratio was manufactured and incorporated into the AWHP system. It was found that the VI AWHP system improved the heating capacity by 48% and the COP by 36% compared to those values for the conventional AWHP at water temperature of 60 °C and ambient temperature of −15 °C. This VI AWHP system can be used for cold climate applications.

In this paper the experimental data were extrapolated to predict the annual performance of ASHP. The coefficient of performance (COP) varied between 4.7 to 5.7 as the outdoor air temperature changed from 33^{0}C to 16^{0}C, respectively. Author suggest that these enhancements are due to the reduction in the operating cyclic and relatively lower operating speeds of the device

3.Results and discussion

3.1Measurement data with reciprocating compressor

Table 1 shows the performance of Reciprocating chiller in 24 hours; in this table it is shown some important parameters such as Flow rate of chilled water, temperature of CHW (Chilled water) inlet, & temperature of CHW outlet. These parameters are measured every hour by Ultrasonic flow meter, mercury glass thermometer and digital temperature indicator respectively.

Table 1 Measurement data with reciprocating compressor

Measurement data with reciprocating compressor							
S. No.	**Time**	**Design Pump Capacity (M^3/Hr)**	**Actual Pump Capacity (M^3/Hr)**	**Chw Inlet Temp (^{0}c)**	**Chw Outlet Temp (^{0}c)**	**Cp (Kj/Kg^{0}c)**	**Tr**
1	6:00	165	150	7	6	1	50
2	7:00	165	150	8	6	1	99
3	8:00	165	150	10	7	1	149
4	9:00	165	150	9	6	1	149
5	10:00	165	150	11	9	1	99
6	11:00	165	150	11	10	1	50
7	12:00	165	150	10.5	10	1	25
8	13:00	165	150	11	10	1	50
9	14:00	165	150	11	9	1	99
10	15:00	165	150	11	10	1	50
11	16:00	165	150	10	9	1	50
12	17:00	165	150	11	9	1	99
13	18:00	165	150	10	8.8	1	60
14	19:00	165	150	10.2	8.5	1	84
15	20:00	165	150	9.8	8.2	1	79
16	21:00	165	150	10.2	8.9	1	64
17	22:00	165	150	10.8	8.4	1	119
18	23:00	165	150	10	8.6	1	69
19	0:00	165	150	11	8.5	1	124
20	1:00	165	150	11	9.2	1	89
21	2:00	165	150	11	9.6	1	69
22	3:00	165	150	10.5	9.2	1	64
23	4:00	165	150	10	9	1	50
24	5:00	165	150	10	9.3	1	35
Avg. Tr							**78**

3.2Power calculation with reciprocating compressor

The compressor power can be measured by a portable power analyses which would give reading directly in kW. If not, the ampere has to be measured by the available on-line ammeter or by using a tong tester. The power can then be calculated by assuming a power factor of 0.9.

Table 2 Power calculation with reciprocating compressor

Power calculation with reciprocating compressor						
Sl. No.	Time	Design Motor (Kw)	Voltage (Volt)	Current (Amp)	Power Factor	Power (Kw)
1	6:00	130	410	140	0.9	89.5
2	7:00	130	410	155	0.9	99.1
3	8:00	130	410	165	0.9	105.5
4	9:00	130	410	180	0.9	115.0
5	10:00	130	410	175	0.9	111.8
6	11:00	130	410	180	0.9	115.0
7	12:00	130	410	165	0.9	105.5
8	13:00	130	410	160	0.9	102.3
9	14:00	130	410	170	0.9	108.6
10	15:00	130	410	160	0.9	102.3
11	16:00	130	410	155	0.9	99.1
12	17:00	130	410	155	0.9	99.1
13	18:00	130	410	160	0.9	102.3
14	19:00	130	410	165	0.9	105.5
15	20:00	130	410	162	0.9	103.5
16	21:00	130	410	165	0.9	105.5
17	22:00	130	410	160	0.9	102.3
18	23:00	130	410	170	0.9	108.6
19	0:00	130	410	165	0.9	105.5
20	1:00	130	410	165	0.9	105.5
21	2:00	130	410	160	0.9	102.3
22	3:00	130	410	160	0.9	102.3
23	4:00	130	410	160	0.9	102.3
24	5:00	130	410	155	0.9	99.1
AVG. KW.						**104.0**

3.3Specific power data with reciprocating compressor

Specific power data is commonly referred to as efficiency of compressor, but actually power input to compressor motor divided by tons of cooling produced, or kilowatts per ton (kW/ton). Lower kW/ton indicates higher efficiency.

Table 3 Actual specific power data with reciprocating compressor

Actualspecific power data with reciprocating compressor						
SL.NO.	TIME	DESIGN TR	ACTUAL (TR)	DESIGN (KW)	ACTUAL (KW)	KW/TR
1	6:00	165	49.6	130	89.48	1.80
2	7:00	165	99.2	130	99.06	1.00
3	8:00	165	148.8	130	105.45	0.71
4	9:00	165	148.8	130	115.04	0.77
5	10:00	165	99.2	130	111.84	1.13
6	11:00	165	49.6	130	115.04	2.32
7	12:00	165	24.8	130	105.45	4.25
8	13:00	165	49.6	130	102.26	2.06
9	14:00	165	99.2	130	108.65	1.10
10	15:00	165	49.6	130	102.26	2.06

11	16:00	165	49.6	130	99.06	2.00
12	17:00	165	99.2	130	99.06	1.00
13	18:00	165	59.5	130	102.26	1.72
14	19:00	165	84.3	130	105.45	1.25
15	20:00	165	79.4	130	103.54	1.30
16	21:00	165	64.5	130	105.45	1.64
17	22:00	165	119.0	130	102.26	0.86
18	23:00	165	69.4	130	108.65	1.56
19	0:00	165	124.0	130	105.45	0.85
20	1:00	165	89.3	130	105.45	1.18
21	2:00	165	69.4	130	102.26	1.47
22	3:00	165	64.5	130	102.26	1.59
23	4:00	165	49.6	130	102.26	2.06
24	5:00	165	34.7	130	99.06	2.85
AVG KW/TR						**1.61**

3.4Measurement data with screw compressor

Table 4 shows the performance of Screw chiller for 24 hrs; in this table we shown some following important parameters such as Flow rate of chilled water, temperature of CHW inlet, & temperature of CHW outlet. These parameters are measured every hour by Ultrasonic flow meter, mercury glass thermometer and digital temperature indicator respectively.

Table 4 Measurement data with screw compressor

Measurement data with screw compressor							
SL.NO.	TIME	DESIGN PUMP CAPACITY (m^3/hr)	ACTUAL PUMP CAPACITY (m^3/hr)	CHW INLET TEMP(^{0}C)	CHW OUTLET TEMP (^{0}C)	Cp (KJ/KG^{0}C)	TR
1	6:00	165	150	8	6	1	99
2	7:00	165	150	9	6	1	149
3	8:00	165	150	10	7	1	149
4	9:00	165	150	11	9	1	99
5	10:00	165	150	11	8	1	149
6	11:00	165	150	11	9	1	99
7	12:00	165	150	10.5	9.2	1	64
8	13:00	165	150	11	9	1	99
9	14:00	165	150	11	9	1	99
10	15:00	165	150	11	9	1	99
11	16:00	165	150	10	7.3	1	134
12	17:00	165	150	11	8.2	1	139
13	18:00	165	150	10	7.8	1	109
14	19:00	165	150	10.2	8.3	1	94
15	20:00	165	150	9.8	7.2	1	129
16	21:00	165	150	10.2	7.8	1	119
17	22:00	165	150	10.8	8.2	1	129
18	23:00	165	150	10	8	1	99
19	0:00	165	150	11	8.3	1	134
20	1:00	165	150	11	9	1	99
21	2:00	165	150	11	9.2	1	89
22	3:00	165	150	10.5	8.6	1	94
23	4:00	165	150	10	8	1	99
24	5:00	165	150	10	8.3	1	84
AVG.TR							**111**

3.5Power calculation with screw compressor

The compressor power can be measured by a portable power analyses which would give reading directly in kW. If not, the ampere has to be measured by the available on-line ammeter or by using a tong tester. The power can then be calculated by assuming a power factor of 0.9.

Table 5 Power calculation with screw compressor

Power calculation with screw compressor						
Sl.No.	**Time**	**Design Motor (Kw)**	**Voltage (Volt)**	**Current(Amp)**	**Power Factor**	**Power(Kw)**
1	6:00	130	410	110	0.9	70
2	7:00	130	410	130	0.9	83
3	8:00	130	410	140	0.9	89
4	9:00	130	410	140	0.9	89
5	10:00	130	410	120	0.9	77
6	11:00	130	410	110	0.9	70
7	12:00	130	410	170	0.9	109
8	13:00	130	410	110	0.9	70
9	14:00	130	410	100	0.9	64
10	15:00	130	410	110	0.9	70
11	16:00	130	410	120	0.9	77
12	17:00	130	410	155	0.9	99
13	18:00	130	410	120	0.9	77
14	19:00	130	410	130	0.9	83
15	20:00	130	410	100	0.9	64
16	21:00	130	410	102	0.9	65
17	22:00	130	410	125	0.9	80
18	23:00	130	410	112	0.9	72
19	0:00	130	410	110	0.9	70
20	1:00	130	410	98	0.9	63
21	2:00	130	410	120	0.9	77
22	3:00	130	410	110	0.9	70
23	4:00	130	410	110	0.9	70
24	5:00	130	410	110	0.9	70
AVG.KW						**76**

3.6Specific power data with screw compressor

Specific power is commonly referred to as efficiency, but actually power input to compressor motor divideby tons of cooling produced, or kilowatts per ton (kW/ton). Lower kW/ton indicates higher efficiency.

Table 6 Actual specific power data with screw compressor

Actualspecific Power Data With Screw Compressor						
Sl.No.	**Time**	**Design Tr**	**Actual (Tr)**	**Design (Kw)**	**Actual (Kw)**	**Kw/Tr**
1	6:00	165	99	130	70	0.7
2	7:00	165	149	130	83	0.6
3	8:00	165	149	130	89	0.6
4	9:00	165	99	130	89	0.9
5	10:00	165	149	130	77	0.5
6	11:00	165	99	130	70	0.7
7	12:00	165	120	130	109	0.9
8	13:00	165	99	130	70	0.7
9	14:00	165	99	130	64	0.6
10	15:00	165	99	130	70	0.7
11	16:00	165	134	130	77	0.6
12	17:00	165	139	130	99	0.7
13	18:00	165	109	130	77	0.7

Actualspecific Power Data With Screw Compressor						
Sl.No.	Time	Design Tr	Actual (Tr)	Design (Kw)	Actual (Kw)	Kw/Tr
14	19:00	165	94	130	83	0.9
15	20:00	165	129	130	64	0.5
16	21:00	165	119	130	65	0.5
17	22:00	165	129	130	80	0.6
18	23:00	165	99	130	72	0.7
19	0:00	165	134	130	70	0.5
20	1:00	165	99	130	63	0.6
21	2:00	165	89	130	77	0.9
22	3:00	165	94	130	70	0.7
23	4:00	165	99	130	70	0.7
24	5:00	165	84	130	70	0.8
AVG.Kw/TR						0.68

The results obtained from the above tables are shown by means of graphs as shown below:

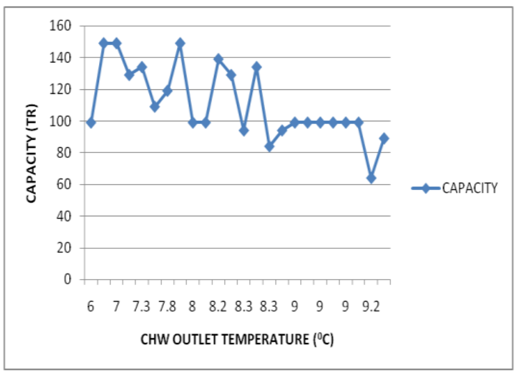

Figure 2 Graph between capacity and cooling water outlet Temperature

The graph drawn between CHW outlet temperature in ^{0}C and TR (Capacity) clearly depicts that once the chilled water outlet temperature drops down the capacity of the compressor 9 as down erratically. This erratic behavior is only because of the inlet temperature.

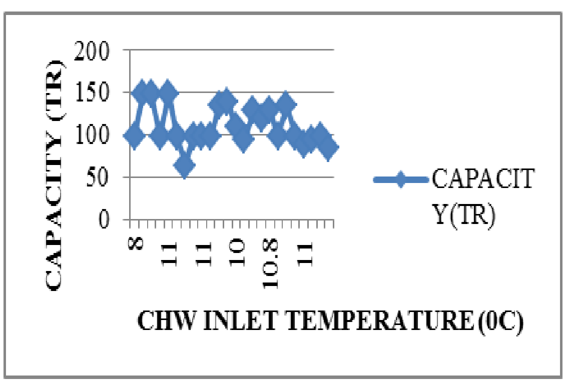

Figure 3 Graph between capacity and CHW inlet temperature

On the basis of physical data made available a graph 3.2 has been drawn between CHW inlet temperature and the capacity. In this case to the nature of graph is erratic. This simply depicts that the erratic behavior is on account of the external factors.

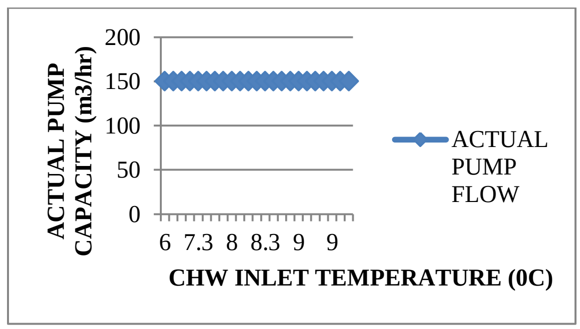

Figure 4 Graph between actual pump capacity and CHW inlet temperature

The Figure 5 in this case drawn between the readings obtained for CHW inlet temperature and the actual pump capacity. The nature of the graph shown is alternated against the different readings obtained and is a straight line curve showing clearly that it is ineffective on account of the inlet temperature.

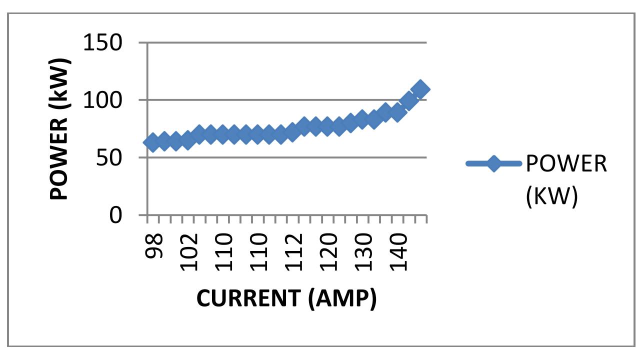

Figure 5 Graph between power and current

In the graph 3.4 drawn is between power and the current drawn which is too simple once the power gone on higher side the current drawn will be also be higher which is evident as seen from the graph. For 60 kw where on the current drawn is 98 Amps in case of 110 kw it is taking 155 amps.

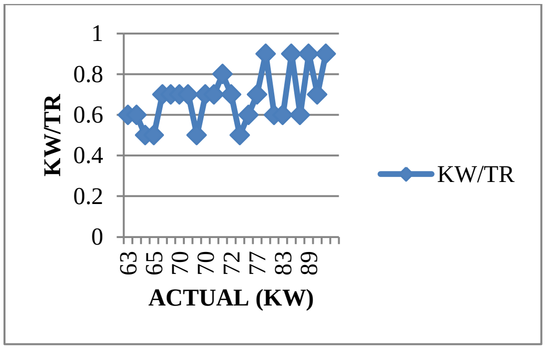

Figure 6 Graph between KW/TR and actual (KW)

This graph 3.5 has been drawn between KW/TR and actual KW. In this case the nature of graph being sinusoidal it erratic in nature.

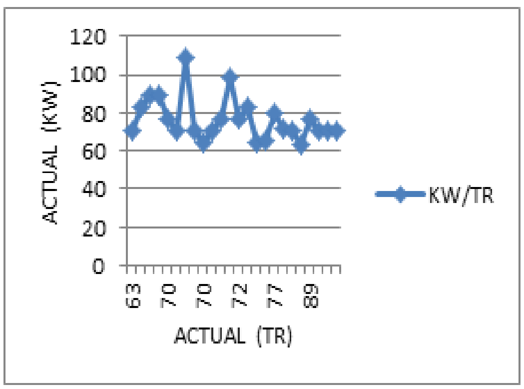

Figure 7 Graph between KW and actual TR

In this Figure 8 has been drawn between the actual KW with respect to actual (TR). In this case to the nature of graph obtained from the reading taken is of erratic in natural.

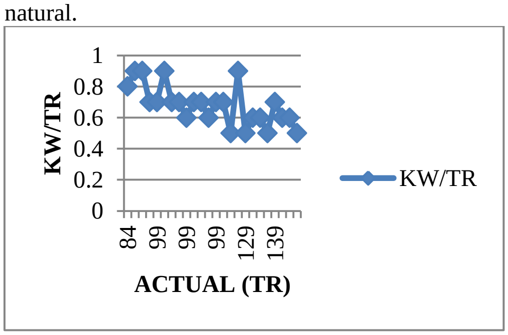

Figure 8 Graph between KW/TR and actual (KW)

This is the Figure 9 drawn between KW/TR and actual KW. This graph depicts the picture of power as unit capacity with the actual power.

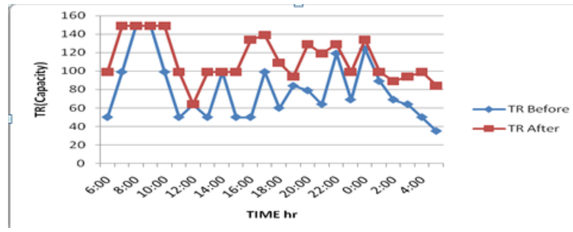

Figure 9 Graph between capacity TR and TIME hr

In this Figure 10 form the figure so obtained as different internal of time are between the capacity and time. This capacity has been shown both before the working and after the working.

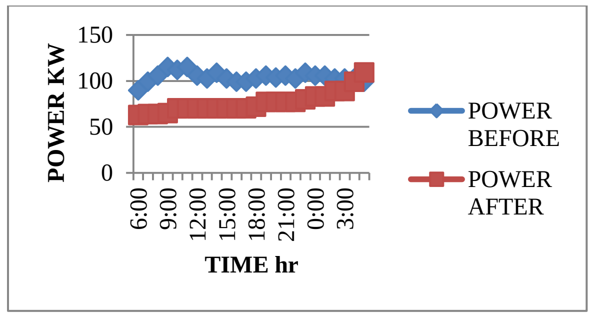

Figure 10 Graph between Power KWand TIME hr

On the similar lines as of Figure 11, in this case the graph has been plotted between the power consumed as different internal of time starting from 6 AM to few days 4 morning calculating the figure on every two hrs lines.

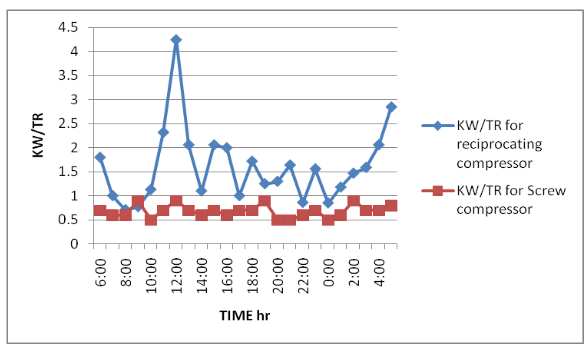

Figure 11 Graph between specific power KW/TR and TIME hr

3.7Maintenance cost of reciprocating compressor

Maintenance of reciprocating compressor is very high due to piston to cylinder line contact, high thermal heat, high vibration in between piston and cylinder due to reciprocating motion. As per company previous maintenance data the cost of compressor maintenance is approx 2 lakh/year. Details are as shown in Table 7.

Table 7 Cost of components

Sr.No	Name of the equipment with specification	Qty.Required	Price/Each (Rs)	Total price (Rs)
1	Air Filter Element W 0849801	1	5022	5022
2	Oil Filter 1903061211	2	951	1902
3	Recipe Lub Oil 20 Ltr Tin #1903016195	1	11321	11321
4	Suction Valve Cover 1903717003	4	1926	7704
5	Delivery Valve Cover 1903717103	1	1646	1646
6	.Oil Wiper Ring Hp 1443151	2	3856	7712
7	O Ring Unloaded Hx2P40Np Cp	15	161	2415
8	Valve Cover O Ring HX2T-50 NP#1903021208	10	198	1980
9	Suction Valve Assembly #1903254090	4	10815	43260
10	Suction Valve Assembly #0450085050	2	11637	23264
11	Discharge Valve AssyLp # 1903254190	4	9189	36756
12	Discharge Valve Assy HP # 1903254390	2	9886	19762
13	Tension Nut #1903671803	2	5943	11986
14	Lp Rider Ring 1903396808	1	9362	9362
15	Hp Rider Ring 1903394408	1	5591	5591
Total Cost (RS.)				**199041/-Rs**

3.8Maintenance cost of screw compressor:-

Maintenance of screw compressor is very less due to piston to cylinder point contact. Low friction loss, low thermal heat, low vibration in between Screw and cylinder due to rotary motion is ensured. As per company previous maintenance data the cost of compressor maintenance is approx 0.5 lakh/year. Detail of spare data is shown in Table 9.

Table 9 Cost of components

Sr. No	Name of the equipment with specification	Qty.Required	Price/Each (Rs.)	Total price (Rs.)
1	.Air Filter Element W 20	1	2000	2000
2	.Oil Filter 1903061211	2	4000	4000
3	RecipLub Oil 20 Ltr Tin #190301265	1	40000	40000
TOTAL COST (RS.)				50000/-**Rs**

4.Conclusion and future scope

Outcome of the investigation and studies made from the experimental data collected from the compressor are as follows:

Replacement of the reciprocating compressor in VCRS system by the introduction of screw compressor results to

Increase in refrigeration effect represented by the temperature drop range.

Reduction in the specific power consumption of compressor

Reduction in the maintenance cost.

Considerable increase in the running period between one consecutive maintenance- shut down (from 45 days to 90 days).

The need and importance of refrigeration cannot be underestimated now days. Especially in reference of heavy refrigeration plant such as ice plant, medical plant etc requires more energy. Hence the prime aim has been focused to minimize the cooling load in overall. In this present research work a major source of enrgy consumption part of plant is compressor. It has been observed that the replacement of reciprocation compressor with the screw compressor saves energy that is it makes refrigerator more energy efficient.

In the same sequence the following two points may be the scope for further research. First, it may be possible to plot the comparative chart for the reciprocating, centrifugal and screw compressor in same working condition. In all, the more efficient one can be selected. Secondly the nano-particle fluid can be inserted into refrigerant for the more efficiency, higher COP and lesser energy consumption.

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