Analysis of Energy Performance of a Laboratory Tested Refrigeration System

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Abstract

This paper presents the experimental performance analysis of a laboratory tested chiller system performed at Brunel University London using R404A as a refrigerant. The effect of the different parameters of performance analysis: compressor power consumption, refrigeration cooling capacity, cooling COP, isentropic efficiency, volumetric efficiency, overall compressor efficiency, Carnot COP and actual COP were investigated for various evaporator water leaving temperature and condenser water leaving temperature. The results obtained from the laboratory test of the refrigeration system: Cooling Capacity of 27.05KW; Refrigerant mass flow rate of 0.2668 Kg/s; Carnot COP of 4.67; Actual COP of 2.28; isentropic efficiency of 57.23%; volumetric efficiency of 73.60%; and Overall compressor efficiency of 55.60%. These results could be compared to international standards values for performance improvement of the refrigeration system; maximum possible energy savings; and emission control in commercial buildings.

Keywords: Chiller system; Coefficient of performance; Energy performance standards; Cooling Capacity; Volumetric efficiency; Isentropic efficiency; Overall compressor efficiency.

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1. Introduction

The effective use of energy in measures to counter global warming and address various energy problems is a critical issue of human activity. At the same time, the ever-greater desire for comfort in the living environment highlights the importance of air conditioning. Air conditioning and refrigeration consume significant amount of energy in buildings and in process industries [1]. The energy consumed in air conditioning and refrigeration systems is sensitive to load changes, seasonal variations, operation and maintenance, ambient conditions etc. and therefore, the performance evaluation will have to take into account to the extent possible all these factors. It is of utmost importance to optimize their efficiency or coefficient of performance (COP) in order to reduce the energy consumption and carbon footprint of commercial buildings. Many central cooling systems in air-conditioned buildings have multiple chillers operating in parallel to meet the varying cooling load requirements [3]. The energy performance or COP (cooling output overpower input) of chillers depend on the heat rejection medium, ambient conditions, compressor efficiency and the load carried by the chillers [4]. Energy consumption of an air-conditioning system is huge in summer, especially in the subtropical region where the summer is hot and humid, causing peak load in electricity. In the air-conditioning system, the chillers system is one of the major equipment in energy consumption. Therefore, how to operate the chillers system for minimizing the energy consumption in different cooling loads becomes an important issue and also to accomplish carbon emissions control, many engineering systems are required to comply with prescriptive energy performance. The minimum energy efficiency levels required by standards and guidelines provide benchmarking criteria for assessing the energy performance of engineering systems [5].Furthermore, regular updates on performance requirements would help phase out low efficient products and drive continuous performance improvement of engineering systems. Energy reduction targets are generally set in the building sector to reduce carbon emissions. For medium to large scale buildings with cooling demand, chiller systems are commonly installed to provide cooling energy and their operation accounts for the major proportion in the overall electricity consumption of buildings. Setting the minimum energy performance and energy efficient practices for chiller systems is a direct way to improve building energy efficiency.

This paper investigates the operating characteristics and thermodynamic performance of commercial reciprocating chillers as performed at Brunel University Laboratory with the aim to understand the physical, thermodynamic and technological parameters that quantitatively determines the optimal operating conditions for reciprocating chillers. The significance of this study is to provide insights into how to effectively evaluate the energy performance of chillers system under actual operating conditions with the aim of enhancing their energy performance.

2. Methodology

2.1. Experimental Test Facility

The test facility is based around a chiller of nominal cooling capacity of 25 kW. The chiller is equipped with shell and tube heat exchangers, an externally equalised thermostatic expansion valve, a 4 cylinder hermetic compressor designed for use with R404A and appropriate controls. The compressor displacement is 45.07 m^3/hr at 1450 rpm. A Danfoss DQ5 electronic 4 expansion valve is mounted in parallel to the thermostatic valve to enable performance comparison of the two valves to be carried out. Modifications carried out on the system enable easy introduction and testing of alternative types of compressor and variable speed drives. Test conditions on the chiller are achieved through two water-storage tanks and 3-way mixing valves. The valves mix appropriate quantities of chilled and hot water from the respective tanks to achieve the required temperatures at the condenser and evaporator. A circulatory air tunnel acts as a balancing mechanism between the hot and cold sides of the system. The tunnel can also be used to simulate ambient conditions for air cooled evaporators and condensers. The test facility is comprehensively instrumented to provide the necessary data for control and optimisation studies. Data logging is performed automatically with a microcomputer based data logging system. The computer records refrigerant temperatures and pressures at four points in the cycle, before and after each major component, water temperatures at the inlet and outlet of the condenser and evaporator, water flow rates in the condenser and evaporator and the power consumption of the compressor motor. The system is very flexible, allowing variation in the number of the logged cycles and the logging cycle time. The logging time used in the experiment was 4 s per cycle. The system is capable of producing all the measured data in an Excel format for subsequent analysis. The instrumentation points are shown in Figure 1.

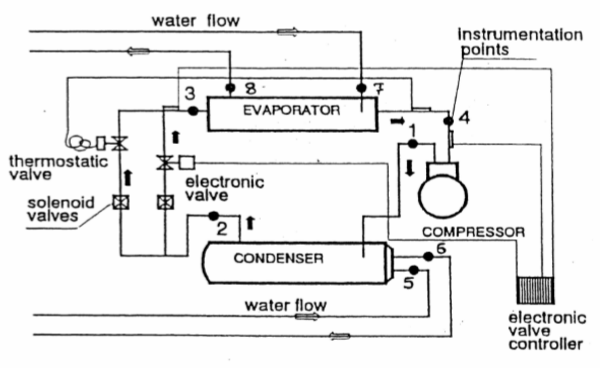

Figure 1. Experimental Test Facility

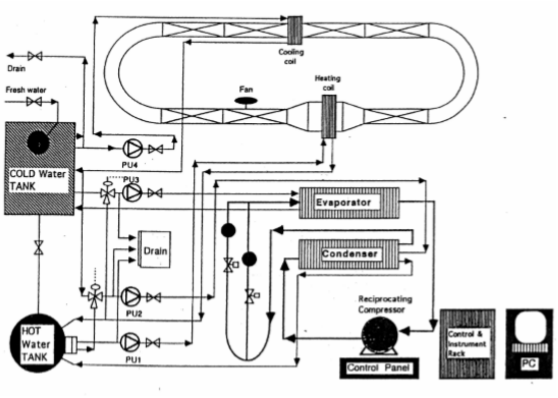

Figure 2. Schematic diagram of Test Rig.

2.2. Experiment

Three sets of experimental tests were carried out for condenser water leaving temperatures of 30°C, 35°C and 40°C. The condenser water leaving temperature approximately corresponds to a constant refrigerant condensing temperature (saturated vapour). For each condenser water leaving temperature the evaporator water leaving temperature which approximately corresponds to a constant refrigerant evaporating temperature (saturated vapour) was varied from 5°C to 10°C in steps of approximately 1°C. In experimental investigations it is very difficult to ensure that conditions are kept constant at exactly the required value and you will see from the results that the evaporator water leaving temperature fluctuates between +/- 0.25°C about the set point.

A number of readings were taken for each set-point and accurate results were obtained from averaging these values using Excel. Sets of readings which deviated considerably from the set point were disregarded. After averaging the readings the results ended up with 6 sets of reading for each of the condenser water leaving temperature, and all the graphs plotted were based on these readings. The reading for the refrigerant mass was disregarded. Calculation of the refrigerant mass flow rate was based on an energy balance between the refrigerant and the water across the condenser.

3. Results Discussion

The results of the averaging data obtained from the experiment performed in the laboratory: compressor power consumption, refrigeration cooling capacity, cooling COP, isentropic efficiency, volumetric efficiency, overall compressor efficiency, carnot COP and actual COP were investigated for various evaporator water leaving temperature and condenser water leaving temperature. These were calculated and plotted on graphs for better understanding of the chiller performance and hence optimization.

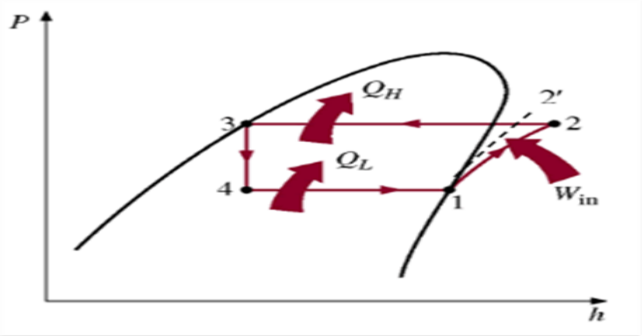

Figure 3. The P-H Diagram of the Refrigerating System [2]

3.1 Three performance characteristics are plotted against evaporator water leaving and condenser water leaving temperatures

In this task, three performance characteristics are plotted against evaporator water leaving and condenser water leaving temperatures and they include: Power consumption, Cooling Capacity and Coefficient of performance (COP).

The equations used for the calculations include:

Cooling Capacity = EWFR × Cp × (T_7 –T_8) [2]

Cooling COP = $\frac{Cooling\ Capacity}{Power\ Consumption}$

Where: EWFR = Evaporator water flowrate (Kg/s), Cp = 4.187KJ/Kg.°C, T7 = Evaporator inlet water temperature, T8 = Evaporator outlet water temperature.

3.1.1 Power consumption:

The results show that the compressor power consumptions tend to increase slightly with increasing evaporator and condenser water leaving temperatures as deduced from fig. 4 and 5. The increase seems to be more with the 40 degree temperature.

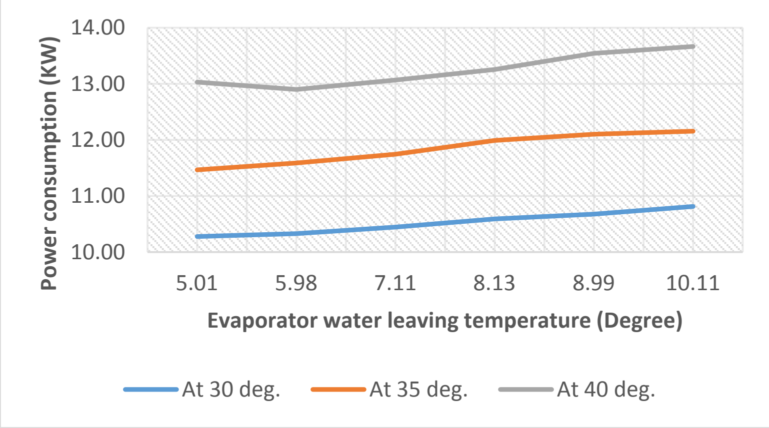

Figure 4. Effect of Evaporator Water Leaving Temperature on Power Consumption

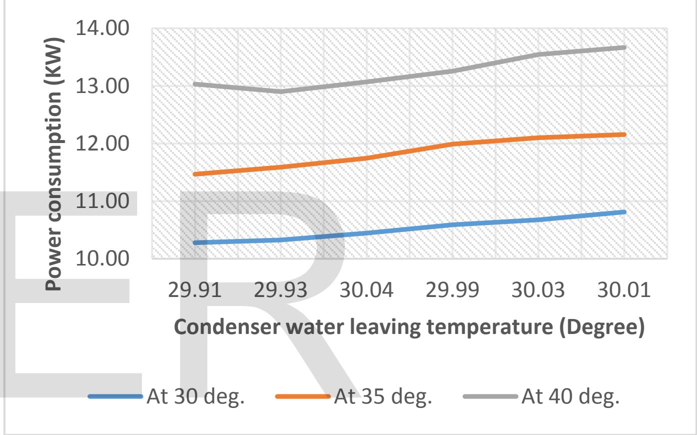

Figure 5. Effect of Condenser Water Leaving Temperature on Power Consumption

3.1.2 Cooling Capacity:

Figure 6 and figure 7 show that the cooling capacity of the refrigeration system increases with increasing evaporator and condenser water leaving temperatures. These increments seem uniform with the 30, 35 and 40 degree temperatures respectively

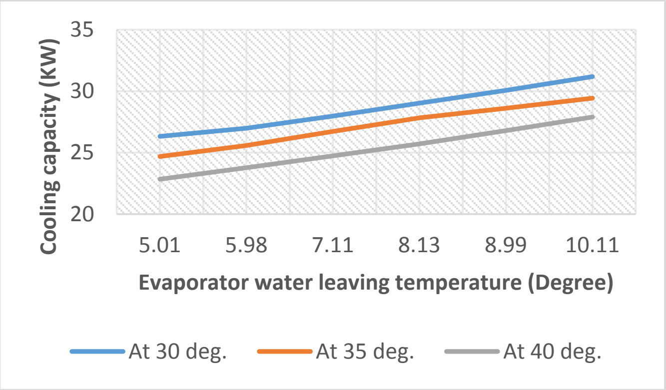

Figure 6. Effect of Evaporator Water Leaving Temperature on Cooling Capacity

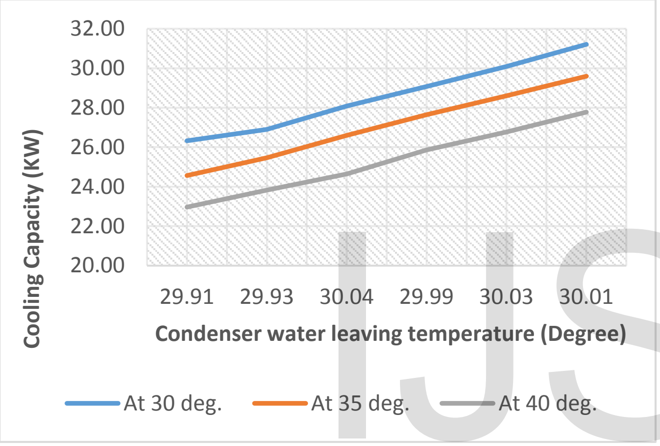

Figure 7. Effect of Condenser Water Leaving Temperature on Cooling Capacity

3.1.3 Cooling COP:

The cooling COP increases gradually with increasing evaporator and condenser water leaving temperatures as seen in fig. 8 and 9.

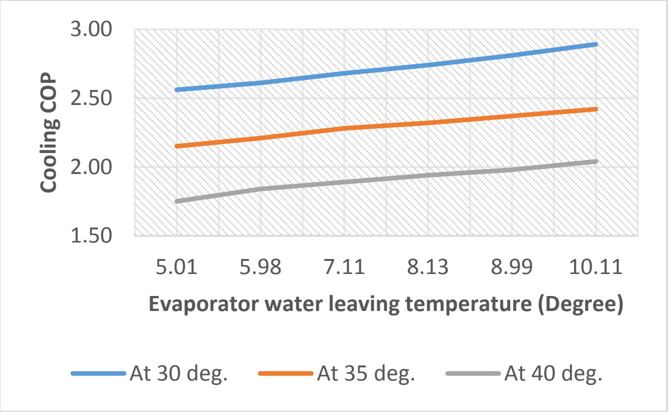

Figure 8. Effect of Evaporator Water Leaving Temperature on Cooling COP

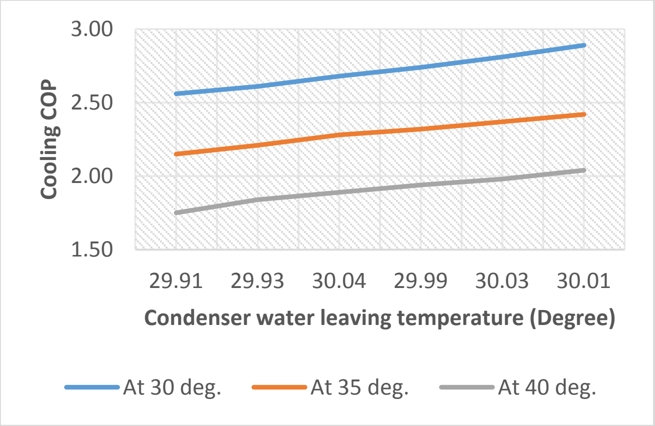

Figure 9. Effect of Condenser Water Leaving Temperature on Cooling COP

3.2 Three characteristic efficiencies are calculated and plotted against Evaporator water leaving and condenser water leaving temperatures

Here, three characteristic efficiencies are calculated and plotted against Evaporator water leaving and condenser water leaving temperatures. These are: Isentropic efficiency, volumetric efficiency and combined efficiency.

From figures 10 and 15, the isentropic efficiency and overall compressor efficiency increase sharply as both the evaporator and condenser water leaving temperatures increase; with a slight change noticed in the 40 degree water leaving temperature, probably due to equipment error. While the volumetric efficiency increases gradually with increasing evaporator and condenser water leaving temperatures.

3.2.1 Isentropic Efficiency

Isentropic Efficiency, $\eta_{is} = \frac{work\ required\ for\ isentropic\ compression}{Work\ done\ on\ the\ gas\ in\ the\ compressor}$

$$\eta_{is} = \frac{W_{is}}{W} = \frac{\dot{m}_r\,(h_1' - h_4)}{\dot{m}_r\,(h_1 - h_4)}$$

$$\eta_{is} = \frac{(h_1' - h_4)}{(h_1 - h_4)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

From properties table, averaged:

$h_1' = 404.6370$ KJ/Kg $\quad h_1 = 421.9971$ KJ/Kg $\quad h_4 = 381.4089$KJ/Kg

$$\eta_{is} = \frac{(h_1' - h_4)}{(h_1 - h_4)} = \frac{404.6370 - 381.4089}{421.9971 - 381.4089} = 0.5723$$

Therefore, isentropic efficiency

η_{is} = 57.23%

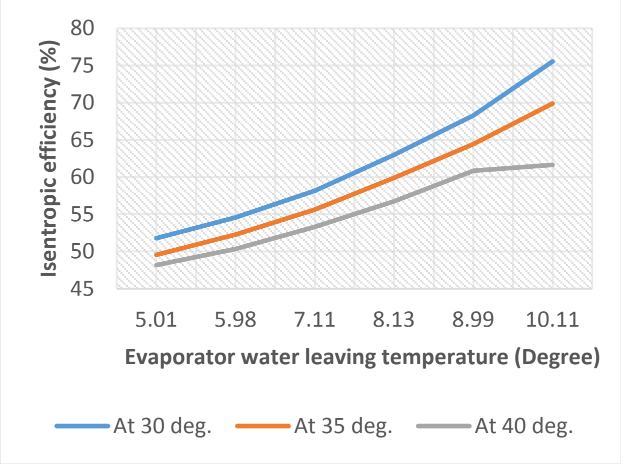

Figure 10. Effect of Evaporator Water Leaving Temperature on Isentropic Efficiency

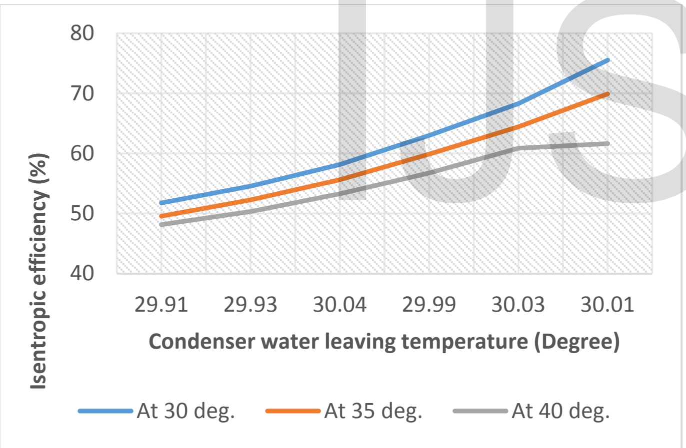

Figure 11. Effect of Condenser Water Leaving Temperature on Isentropic Efficiency

3.2.2 Volumetric Efficiency:

Volumetric Efficiency, $\eta_v = \frac{actual\ volume\ of\ fresh\ gas\ induced}{compressor\ swept\ volume}$

$\eta_v = \frac{V_{ind}}{V_s} \gg \eta_v = \frac{\dot V}{\dot V_s}$

$\eta_v = \frac{\dot V}{\dot V_s}$ …………………………………...……..... (2)

Assuming an ideal gas and constant density:

Volume = $\frac{Mass}{Density} = \frac{m}{\rho}$

$\frac{dv}{dt} = \frac{1}{\rho} \times \frac{dm}{dt}$, hence, $\dot V = \frac{\dot m_r}{\rho}$ …………………… (3)

Compressor displacement, $\dot V_s$ = 45.07 m^3/hr = $\frac{45.07}{3600} m^3$/sec.

$\dot V_s$ = 0.012519 m^3/sec.

Compressor speed, N = 1450rpm = $\frac{1450}{60}$ rev. /sec.

N = 24.16667 rev. /sec.

From property table, averaged density,

ρ = 28.9772 Kg/m^3

Averaged refrigerant mass flowrate,

$\dot m_r$ = 0.2668 Kg/sec.

Therefore, $\eta_v = \frac{\dot m_r}{\rho\, \dot V_s} = \frac{0.2668}{28.9772 \times 0.012519} \times 100\%$

η_v= 73.6%

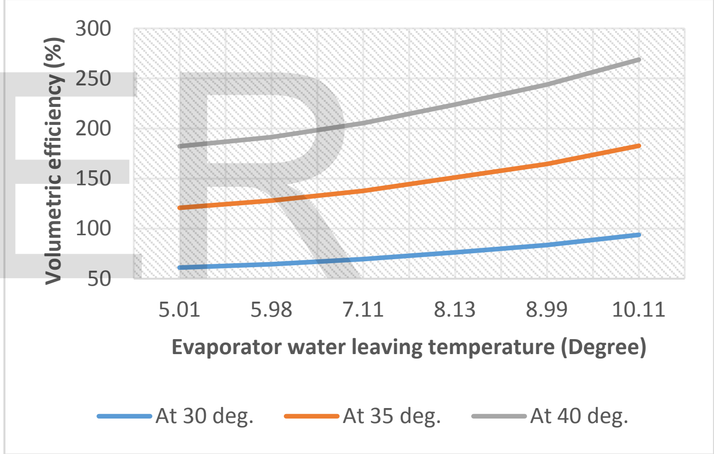

Figure 12. Effect of Evaporator Water Leaving Temperature on Volumetric Efficiency

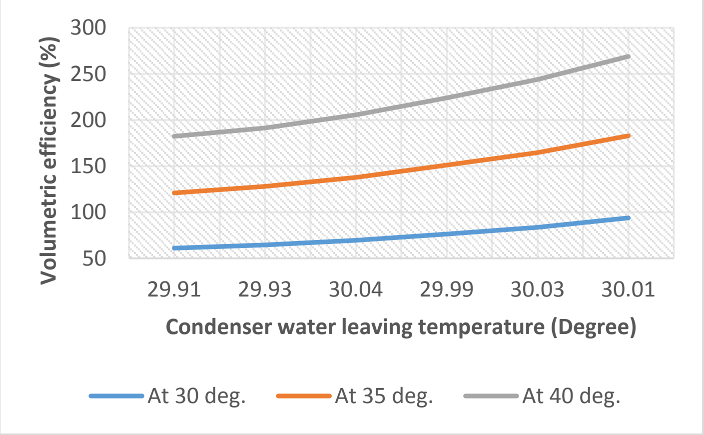

Figure 13. Effect of Condenser Water Leaving Temperature on Volumetric Efficiency

3.2.3 **Overall Compressor Efficiency:**

$$\eta_{cs} = \frac{Isentropic\ Workdone}{Power} = \frac{W_{is}}{W} = \frac{\dot{m}_r(h_1'-h_4)}{W} \ldots\ldots\ldots\ldots\ldots\ldots\ (4)$$

From the experimental results, averaged values for:

W = 11.14 KW, $\dot{m}_r$ = 0.2668 Kg/sec. h_4 = 381.4089 KJ/Kg and h_1' = 404.6370 KJ/Kg

Therefore, $\eta_{cs} = \frac{0.2668(404.6370-381.4089)}{11.14} = 0.5563 \times 100$

η_{cs}= 55.6 %

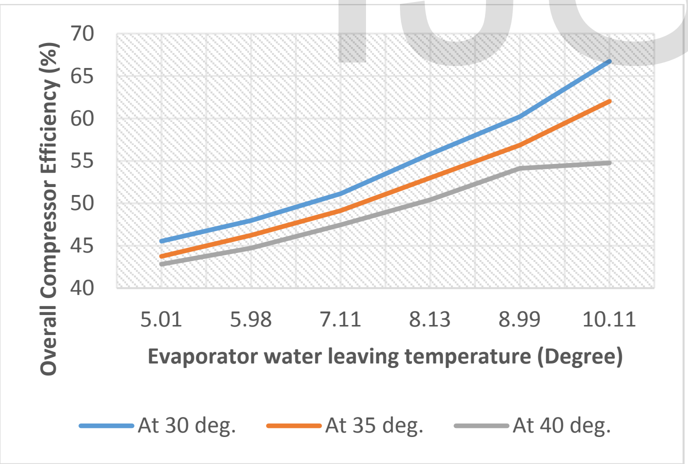

Figure 14.Effect of Evaporator Water Leaving Temperature on Overall Compressor Efficiency

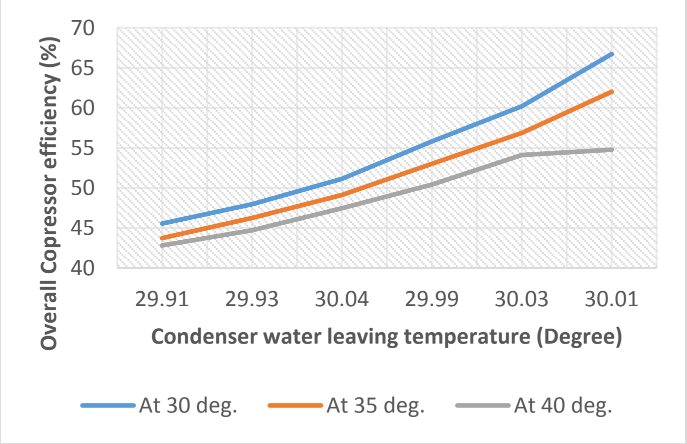

Figure 15. Effect of Condenser Water Leaving Temperature on Overall Compressor Efficiency

3.3 *Calculation and plotting of Carnot COP and comparing it with the actual COP of the system*

Task 3 is for calculation and plotting of Carnot COP and comparing it with the actual COP of the system.

3.3.1 Carnot COP:

Carnot COP = $\frac{Heat\ removed}{Work\ input}$.

Carnot COP = $\frac{T_4}{T_1-T_4}$ ……………………………….. (5)

Where: T_4 = Evaporator outlet/Compressor suction temperature (K).

T_1 = Compressor discharge/Condenser inlet temperature (K).

The averaged values for: T_1 = 76.66 +273 = 349.66K, T_4 = 15.03 +273 = 288.03K

Carnot COP = $\frac{288.03}{(349.66-288.03)} = \frac{288.03}{61.63}$ = 4.67

Carnot COP = 4.67

3.3.2 Actual COP:

Actual COP = $\frac{Cooling\ Capacity}{Power\ Consumption}$

Cooling Capacity = EWFR × Cp × (T_7 –T_8)

Where: EWFR = Evaporator water flow rate (Kg/s), T7 = Evaporator inlet water temperature, T8 = Evaporator outlet water temperature.

Taking their averaged values: EWFR = 0.77Kg/s, Cp = 4.187KJ/Kg.°C, Power Consumption = 11.87 KW, T_7 = 15.94°C, and T_8 = 7.55°C

$$\text{Actual COP} = \frac{0.77 \times 4.187 \times (15.94 - 7.55)}{11.87} = \frac{27.05}{11.87} = 2.28$$

Actual COP = 2.28

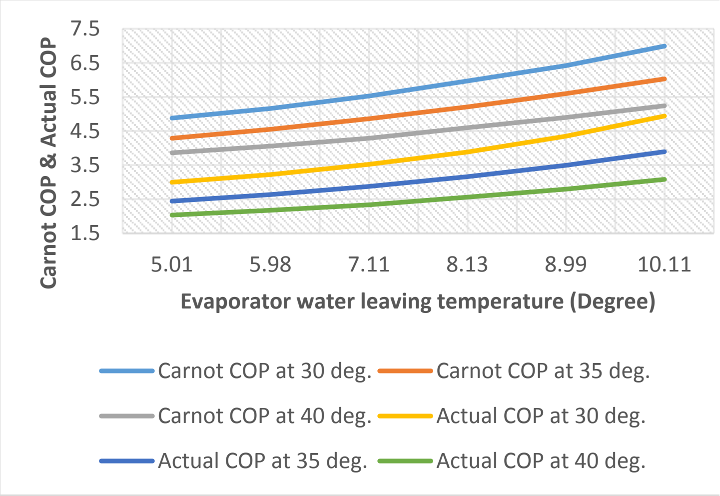

Figure 16. Effect of Evaporator Water Leaving Temperature on Carnot COP & Actual COP

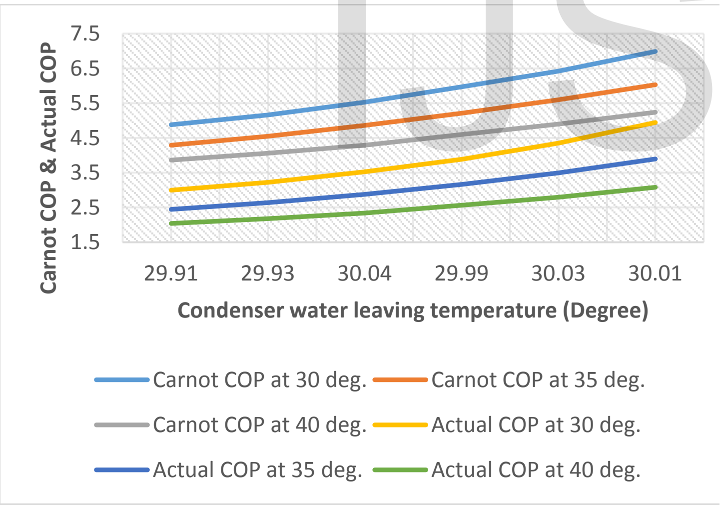

Figure 17. Effect of Condenser Water Leaving Temperature on Carnot COP & Actual COP

3.3.2 Reasons for higher Carnot COP to Actual COP:

Thermal efficiency increases with increase in average temperature at which heat is supplied to the system, or with a decrease in the average temperature at which heat is rejected from the system. The source and sink temperatures that can be used in practice are limited by the maximum temperature that the component can withstand and the lowest temperature that the cooling medium utilized in the cycle. Carnot cycle is independent of this limit, hence higher COP. Also, the relatively higher value of Carnot COP attained is derive from assumptions that it ignores small finite-rate heat transfer losses in the heat exchangers and frictional pressure drop.

3.4 Performance characteristics of the system and their implications with respect to system design and optimal operations are discussed

At this point, the performance characteristics of the system and their implications with respect to system design and optimal operations are discussed.

3.4.1 Volumetric Efficiency:

A higher volumetric efficiency signifies good mechanical design and reduced physical size for a given refrigerating capacity. Its implication to system design and optimal operation is that the higher the volumetric efficiency, the smaller the size of the system and thus lesser cost of design and production of a system for a given capacity.

3.4.2 Isentropic Efficiency:

A higher isentropic efficiency signifies higher orderliness and lesser fluid flow losses in the system. Its implication to system design and optimal operation is that its high values implies higher refrigerant flowrate and hence, more efficient system.

3.4.3 Coefficient of Performance(COP):

The energy performance of chillers is often described as the coefficient of performance (COP) which is the cooling capacity output in kW over the electric power input in kW [6]. It is the ratio of useful cooling provided to work required. Therefore, higher COPs equate to lower operating costs.

4. Conclusion

In this study, the thermodynamic performance of a commercial chiller was investigated in a laboratory and the following results were obtained from the laboratory test of the refrigeration system: Cooling Capacity of 27.05KW; Refrigerant mass flow rate of 0.2668 Kg/s; Carnot COP of 4.67; Actual COP of 2.28; isentropic efficiency of 57.23%; volumetric efficiency of 73.60%; and Overall compressor efficiency of 55.60%. These results were all obtained from averaging the values obtained from the test and these speak volume of any improvement changes needed as compared to international operating standards.

5. References

[1] Shuichi M., Ryo F., et al (2002). High-Performance Absorption Chiller for District Heating and Cooling. Mitsubishi Heavy Industries, Ltd. Technical Review Vol.39 No.2

[2] M.J Gordon and K.C. Ng, Cool Thermodynamics: The engineering and Physics of predictive, Diagnostic and Optimization Methods for cooling systems – Cambridge International Science Publishing (2000). ISBN: 1898326908

[3] Fernandez-Seara J., Uhia F.J., Diz R., Dopazo J.A., Vapour condensation of R22 retrofit substitutes R417A, R422A and R422D on CuNi turbo C tubes , International Journal of Refrigeration, Vol. 33, 148-157, 2010.

[4] Bolaji B.O., Performance investigation of ozone-friendly R404A and R507 refrigerants as alternatives to R22 in a window air-conditioner , Energy and Buildings, Vol. 43, 3139-3143, 2011.

[5] F.W. Yua, K.T. Chan, et al (200). Review of standards for energy performance of chiller systems serving commercial buildings. The 6th International Conference on Applied Energy – ICAE2014

[6] Air-conditioning, heating and Refrigeration Institute (AHRI). AHRI Standard 550/590: Performance rating of water chilling and heat pump water-heating packages using the vapor compression cycle, Author Press; 2011.