Experimental Investigation of The Performance of a Water-Cooled Serial 700 Packaged Reciprocating Chiller

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Abstract- The widespread use of packaged units of all sizes requires interpretation of catalogue data by application engineers, sales engineers and others including end users. The first step is to be certain of the basis of the published data and consider in what ways this will be affected by different conditions. Hence, compressor capacity and power requirements are determined accurately only by actual testing of the compressor in a laboratory. This project, therefore aims at testing the performance of a packaged water-cooled reciprocating chiller manufactured by DUNHAM-BUSH. A performance testing rig was constructed to achieve this purpose and in the instrumentation of the test facility, effort was made to ensure that all sensors are fixed or placed to reflect actual situations. Instruments are interfaced to a microcomputer, both to record data and to process them automatically. From the results of the operation tests, it was found that the capacity of the chiller varied from a maximum value of 27.83KW at start-up to a value of about 20KW at steady state. To achieve a better coefficient of performance, it is recommended to increase the tube side heat transfer in the condenser by keeping the water velocity as high as possible and a water temperature rise of 5K are good targets to aim for.

Indexed Terms- Cooling Capacity, Coefficient of Performance, Heating Capacity, Reciprocating Chiller, Thermal System

I. INTRODUCTION

The A large proportion of refrigeration and air-conditioning equipment is bought from catalogue data published by manufacturers indicating rating and application data for their products based on standard test conditions and for the most common range of uses. They cannot be expected to have accurate figures for every possible combination of conditions for an individual purpose, although most will produce estimates if asked.

The widespread use of packaged units of all sizes requires interpretation of catalogue data by application engineers, sales engineers and others including end users. The first step is to be certain of the basis of the published data and consider in what ways this will be affected by different conditions. Revised figures can then usually be determined. For extensive interpretation work, simple mathematical models of performance can be constructed. However, taking a compressor for example, it is an accepted fact that mathematical evaluation of all the factors which influence the compressor performance is very difficult to predict. Hence, compressor capacity and power requirements are determined accurately only by actual testing of the compressor in a laboratory.

II. LITERATURE REVIEW

Refrigeration deals with the process of reducing and maintaining the temperature of a space or material below the temperature of the surroundings. The main measuring parameter to check the performance of a system is the coefficient of performance which is defined as the ratio of the cooling effect produced and work input (Akanmu, 1988). This cooling can be improved in two ways: firstly, by increasing heat abstraction rate in the evaporator, and secondly, by reducing the work done in a compressor. Using Nanoparticles will increase heat transfer in cooling systems thus improving system performance. Nanoparticles are either suspended in the refrigerant or in the lubricating oil (Subramani et al., 2011).

Yilmaz (2020) evaluated the performance of a refrigeration system using nano lubricant and found out that the most suitable amount of 0.5 vol % of Cu/Ag alloy and CuO nano lubricants provided the COP increments of 20.88% and 14.55% respectively, with comparison to that of the compressor oil without nano additives due to tribological enhancement of the nanoparticles in the lubricant.

Hydrocarbon refrigerants have become a suitable substitute for conventional refrigerants in most of the refrigeration systems used in developed countries (Banjo, 2021, Raghunatha, et al., 2019, Ajayi et al., 2018). Besides the excellent effect of the refrigerant on the refrigeration system, the extended heat transfer surface (EHTS) contributes to the performance of the system as this enhances the heat transfer rate. The EHTS has layers of tubes with fins attached to ease the heat transfer, and there are three types of EHTS: water-cooled, air-cooled and evaporative. Bolaji (2020) evaluated the performance of a domestic refrigeration system using refrigerant mixtures with similar specific volume characteristics, low global warming, and zero ozone depletion as a substitute to the hydrofluorocarbon (R134a) refrigerant by a theoretical approach. The selected refrigerant mixtures have higher latent heat than R134a, which enhances their heat transfer rate and cooling capacity. Karim et al. (2020) investigated three types of condensers: water-cooled, evaporative, and air-cooled condensers. A spray technique was used to improve the heat rejection on the evaporative condenser, which resulted in a drop in the condenser's pressure. The refrigeration system performed better while working with an evaporative condenser in terms of energy reduction and coefficient of performance enhancement. Banjo et al. (2019) conducted experiments on a vapour compression system that uses refrigerant R134a and hydrocarbon refrigerants (R-600a) simultaneously by retrofitting the system. The results show that the non-conventional refrigerant performed excellently when compared to the conventional refrigerants in terms of coefficient of performance (COP), pull-down time, energy consumption, and heat transfer rate. The COP of the HC- refrigerated system increased by 32.2%, and the energy reduction rate was 4.5%.

III. EXPERIMENT

Plate 1: The Chiller that was investigated.

Plate 2: The experimental rig in the laboratory

The test rig was set up as shown in plate 2. Measurements that were taken and recorded automatically are the temperatures of water into and out of the evaporator and the condenser and that of the water flow rates in these two circuits.

For the temperatures, the K-type thermocouples are used, and for water flow rates, "rotameters" are used. These readings were automatically taken and recorded for every minute from start-up to the next 26 minutes. 26 minutes was adopted because by that time, the capacities would have reached steady state conditions.

The data logging system used in this project consists of the following four items:
1) The computer and its accessories.
2) The IEEE-488 bus.
3) The Microlink.
4) The isothermal box.

IV. DISCUSSION AND ANALYSIS OF RESULTS

The packaged chiller is a standard factory fabricated and assembled unit. The unit is charged with refrigerant - R22. The machine operates on the reversed heat engine cycle and is supplied with work W, absorbs a quantity of heat Q_1 at a temperature $T_1$and rejects a quantity of heat Q_2 at a temperature T_2. Applying the first law of Thermodynamics to the cycle.

$\sum dQ = \sum dW$ or $Q_1 - Q_2 = -W$ and $COP = \frac{Q_1}{W}$

The cooling/heating capacity is directly proportional to the mass flow rate, specific heat and the temperature difference by the cooled/heated water in passing through the evaporator/condenser. The cooling/heating capacities, and the COP were then calculated and plotted for each minute.

$Q_e = M_{we} \times C_{pw} \times (T_5 - T_6)$ and $Q_c = M_{wc} \times C_{pw} \times (T_8 - T_9)$

Where

Q_e = the heat loss at the evaporator in kilowatts.
m_{we} = the mass flow rate of water in evaporator in kilograms per second or liters per second.

C_{pw} = the specific heat of the cooled water (kJ/kg K)
(T_5-T_6) the temperature fall of cooled water in the evaporator in K.

Q_c = the heat gain at the condenser in kilowatts.
m_{wc} = the mass flow rate of water in condenser in kilograms per second or liters per second.

(T_8-T_9) the temperature rise of heated water in the condenser in K.

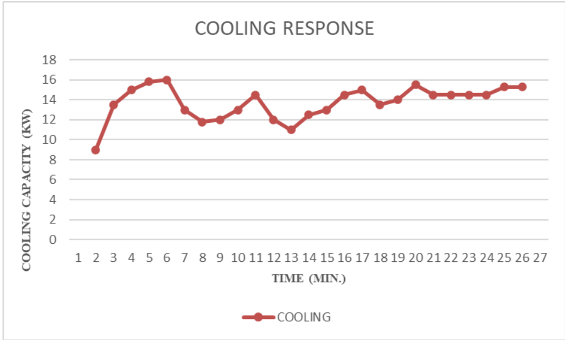

Fig. 1: The cooling capacity of the evaporator against time.

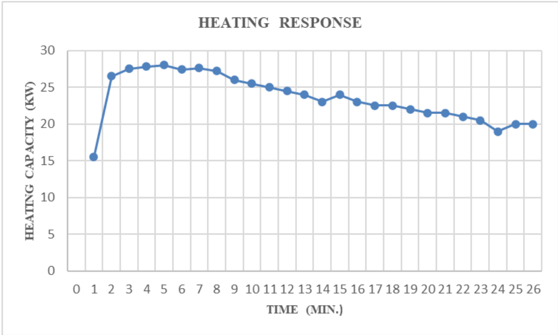

Fig. 2: The heating capacity of the condenser against time.

Graphs in Fig.1 and Fig. 2 show the results of cooling and heating capacities against time. It can be seen that the capacities rise rapidly and then gradually tend towards steady state values. This is because of the rapid fall in the suction pressure, the refrigerant flow rate to the evaporator increases and the refrigerant boils rapidly at a low temperature and thus creates a rapid increase in the capacity. As the rate of evaporation of the refrigerant in the evaporator balances with the rate at which liquid refrigerant is formed in the condenser, the mass flow rate of the refrigerant flowing through the expansion valve and the compressor balances and the capacities tend towards a steady state value.

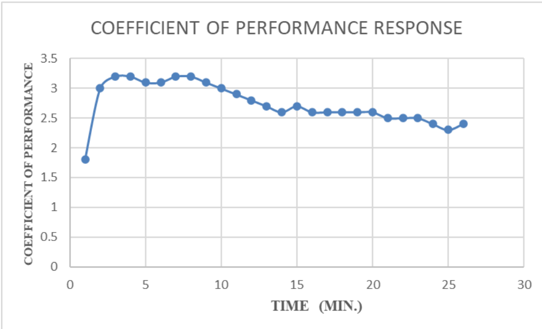

Fig. 3: The COP of the chiller against time.

Fig.3 shows the coefficient of performance (COP) of the chiller plotted against time. During the start-up of the compressor, the COP rises very rapidly and then falls and levels up to a steady value of about 2.3. The reason for this is that, for the first few cycles, the enthalpy of the refrigerant flowing into the evaporator is much higher than the enthalpy of vapour flowing out of the evaporator. As soon as the liquid refrigerant starts flowing from the condenser, the COP reduces and then levels up to the steady state value.

CONCLUSION

The performance of the reciprocating chiller were investigated experimentally in the laboratory and the following are the major findings from the experiment.

The published catalogue accompanying the packaged chiller (ref. 19) indicated the capacity rating of the chiller to be between 20 - 25KW. From the results and graphs of the operation tests, it was found that the capacity of the chiller varied from a maximum value of 27.83KW at start-up to a value of about 20KW at steady state. To achieve the maximum rated capacity and even better coefficient of performance, increasing the tube side heat transfer in the condenser by keeping the water velocity as high as possible (- consistent with reasonable pumping power and freedom from piping erosion) could be an idea. Moreover, a water temperature rise of 5K and a temperature approach of 5K between water exit temperature and condensing temperature are good targets to aim for.

The suction pressure and temperature have to be kept as high as possible while condensing temperature and pressure should remain as low as possible.

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