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Experimental Investigation of the Performance of a Design Model for Vapour Compression Refrigeration Systems

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In this study, a design model for vapour compression refrigeration systems developed by Akintunde (2003a) was used to develop a practical refrigeration system. The system developed by using the design model (rig a) was compared experimentally with another system, built out of original components (rig b) and of the same capacities, under the same experimental conditions. The experimental investigations were based on important parameters such as refrigeration inventory, condensing, and evaporating temperatures, coefficient of performance (COP) and refrigeration efficiency. Previous analysis and experimental investigations carried out by Akintunde (2003a) showed that the model results were comparable to the experimental ones and the performance data were comparable to those in the ASHRAE handbook and other literature. The present results show that the maximum absolute deviations are within the ranges of 19 % and 35 % for rig a and rig b, respectively, as compared with the model results.

Keywords: Vapour compression, system, refrigeration capacity, COP, comparison.

1. Introduction

Vapour compression refrigeration systems present some peculiarities with respect to other refrigeration systems because it is commonly used in a wide range of commercial and industrial applications. Vapour compression refrigeration systems represent a substantial fraction of the installed refrigeration systems. The system consists of compressor, evaporator, condenser and expansion device all connected in a closed loop.

Many models have been developed for the evaluation and design of refrigeration components (compressor, condenser, evaporator, and expansion devices), (**Glocker *et al.* 1998; Janssen *et al.* 1988; Aluares** and **Trepp 1987; Spausdus 1987; Melo *et al.* 1988; Lee *et al.* 2000; Lee *et al.* 2002**). In as much as these components are connected in a closed loop in the vapour compression refrigeration cycle, **Akintunde (2003b)** argued that for these components to function effectively, a balanced point must be reached between these components. There is no available information about this balanced point (or "compromised point") except that of **Stoecker** and **Jones (1982)** which was for a single unit refrigeration system. He developed a mathematical model for vapour compression refrigeration systems based on balanced point, which can be applied for the design and optimisation of vapour compression refrigeration systems. This is made possible by using a mathematical model, with the aid of a computer programme to generate balanced point for vapour compression systems, subjected to different operational conditions. A vapour compression refrigeration system was then fabricated based on the computer-generated data, tested and used to validate

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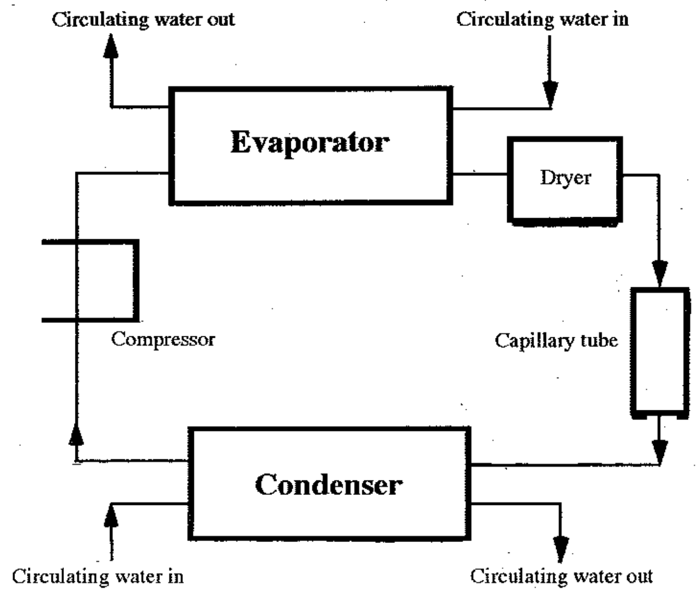

FIGURE 1: *The Schematic Diagram of the Experimental Rig (simplified)*

the developed model. There was a comparison of results between the available data, generated data and the experimental results.

In this work, the model developed by **Akintunde (2003a** and **b)** was used to design a refrigeration system (denoted as rig a in this work). The system constructed out of the design model (rig a) was compared experimentally with another system, built out of original components (denoted as rig b in this work) and of the same capacities, under the same experimental conditions. The experimental methods follow those of **Jabardo *et al.* (2002)** and **Khan** and **Zubair (1999)**. The objectives of the experimental bench were of two fold: to test and check system and components performance and to obtain support data for a computer simulation model of the refrigeration system.

2. Materials and Methods

Experimental rig was designed based on the model-generated-data and some of the operational parameters predicted by the model were compared with the experimental results.

The diagram of the experimental rig is as shown in **Figure 1**. The components of the rig (rig a) and the main parameters are listed as follows:

1. **Compressor:** Reciprocating type. Power 0.746 kW, Refrigerant R-12, with cylinder stroke volume of 32.7 cm^2.

2. **Evaporator:** Bare coil, immersed in water being cooled, pipe inner diameter 5.588 mm, outer diameter 6.35 m and tube length 30.71 m

3. **Condenser:** Bare coil, pipe inner diameter 5.588 mm, outer diameter 6.35 mm and tube length 14.362 m.

4. **Circulating Water:** In order to change the load of the evaporator, water is circulated at various flow rates.

The objectives of this experimental rig is to:

i) Test and check system and components performance.

ii) Obtain data for justification of a computer model.

The second experimental rig (rig b) was made up of bought out components of a vapour compression refrigeration system of equal capacity as rig a. This system is an assembly of 0.746 kW compressor and equivalent condenser, evaporator and capillary tube.

The refrigeration cycle performance was evaluated through the following parameters: Cooling load or rate of heat absorption (Q_e) condensing load or rate of heat rejection (Q_c), Compressor power (P), Coefficient of performance (COP) and the refrigeration efficiency (η). These parameters were evaluated with the aid of measured temperatures, pressures and the Steam Table in conjunction with equations (1) to (5) suggested by **Jabardo *et al.* (2002)**.

$$Q_e = m_r\left(h_{ee} - h_{ei}\right) \tag{1}$$

$$Q_c = m_r\left(h_{ce} - h_{ci}\right) \tag{2}$$

$$P = m_r\left(h_{epe} - h_{epi}\right) \tag{3}$$

$$COP = \frac{Q_e}{P} \tag{4}$$

$$\eta = \frac{COP_{system}}{COP_{Carnot}} \tag{5}$$

the subscripts are:

ee	evaporator exit
ei	evaporator inlet
ce	condenser exit
ci	condenser inlet
cpe	compressor exit
cpi	compressor inlet
r	refrigerant

Further details of the procedure can be found in the work of **Akintunde, (2003b)**. The model developed was tested and compared with theoretical values.

3. Results and Discussion

The overall refrigeration system performance was strongly influenced by operational parameters such as evaporating and condensing temperatures, flow rate of circulating water, ambient temperatures and compressor speed. The tests with the experimental rigs in the present study was planned in such a way to allow for the evaluation of the effect of the aforementioned parameters on system performance. In this study two experimental rigs were used. They were obtained as stated above.

The test data was used to assess the quality of the computer-simulated results. The summary of the experimental and simulated system performance results and their comparison is presented alongside an additional feature related to the effect of refrigerant charge, which is of major concern to refrigeration designers and refrigeration manufacturers (**Jabardo *et al.* 2002**).

3.1 Effect of Charging

As reported by **Lee *et al.* (2000)** the expected charge for the capacity considered in this study should range between 750 g and 900 g of refrigerant. In order to evaluate the effect of the refrigerant charged, tests were performed with an initial refrigerant charge of 1.1 kg. In subsequent experiments, the refrigerant charged was reduced by stepwise decrement of 50 g at a time, down to a minimum of 600 g, when a significant amount of bubbles started to appear in the liquid line sight glass. This was accomplished by placing the refrigerant cylinder on a scale and withdrawing refrigerant into it via a valve.

For this experiment, the following parameters were used: Evaporating temperature (T_1) = 5°C, Condensing temperature (T_2) 40°C, Ambient temperature ((Ta)) 35°C and a constant compressor speed of 1750 rpm. The experimental result is shown in **Figure 2**. As indicated in **Figure 2**, the refrigerating capacity (Q_e) and the coefficient of performance (COP) are not strongly affected over a wide range of refrigerant charges. It can be seen that between 950 g and 700 g both Q_e and COP are relatively constant for both. The results from the experiments fall within

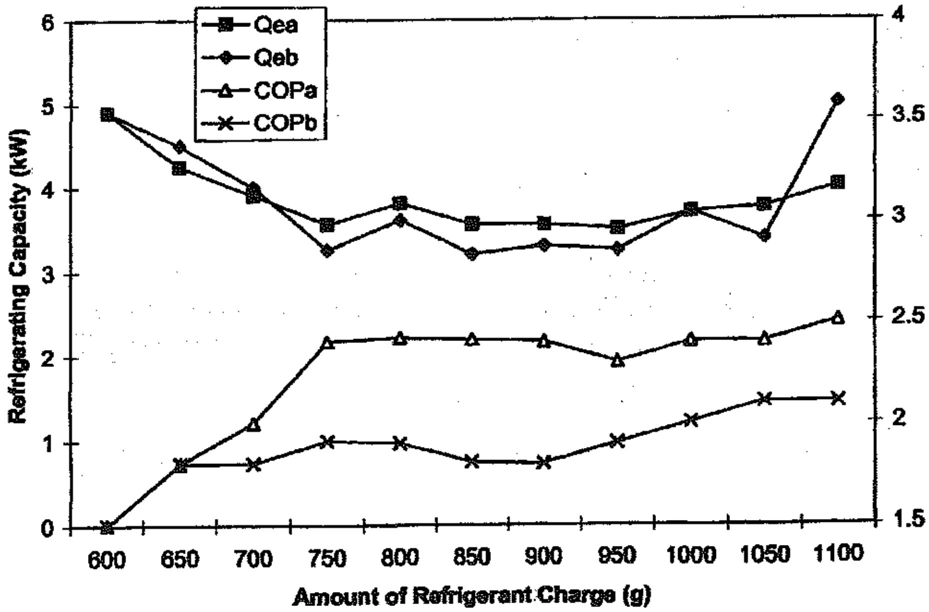

FIGURE 2: *Effects of Refrigerant Charge on COP and Refrigerating Capacity*

the expected range for 750 g and 900 g [9] of refrigerant charge where the influence of the refrigerant charge on the performance is considered negligible. It has been known that flooding of the system is promoted at a charge above 950 g and a large volume of vapour is produced at a charge below 700g. These result in poor system performance. At constant ambient temperature, the condensing pressure increases and so does the liquid subcooling at the condenser exit. This results in a slight gain in the refrigeration effect.

Figure 3 shows that within the expected refrigerant charge, the mass flow rate is relatively constant. This effect also will not influence the expected result negatively.

3.2 Effect of Compressor Speed

In this investigation, four compressors are used; these compressors are of equal capacity but of varying speeds as indicated by the manufacturer. These compressors, therefore, are of different bore diameters. The aim is to investigate the effect of compressor speed on refrigeration effect (Q_e), coefficient of performance (COP) and the mass flow rate (m_r). In this experiment, condensing and evaporating temperatures were kept

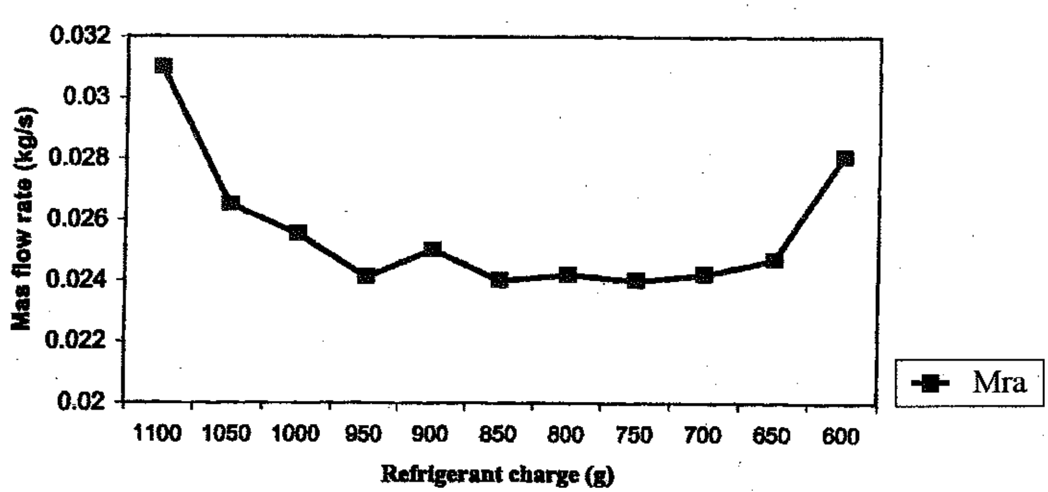

FIGURE 3: *Effect of Refrigerant Charge on Mass Flow Rate*

constant at 40°C and 5°C, respectively. As reported by **Jabardo *et al.* (2000)**, in systems with a variable capacity compressor, the running speed should be of no major concern regarding the refrigerating capacity. In other words, refrigerating capacity should not vary with running speed. The results obtained in this experiment justify the statement.

The speeds of these compressors are: 1500; 1750; 2000 and 2250 rpm. The result of this investigation is as presented in **Figure 4**. From **Figure 4**, the refrigerating capacities Q_{em}, Q_{ea} and Q_{eb} remain constant over the whole range of speeds, so also the corresponding mass flow rate. This is not so for the COPs which diminishes with the compressor speed as a result of the increment in compression work. This is also related to the rise in the discharge temperature associated with the increased compression irreversibility with speed. The mass flow rate remains constant (**Figure 4**) and that it is independent of rotational speed as exemplified in **Figure 3**.

Figures 4 (a – c), plotted for clarification of **Figure 4** show that the results from the simulated model compared very well with the experimental results. As indicated by the above-mentioned Figures, it can be seen that rig a performed better than rig b. This justifies the initial argument that a "balance point" must be sought between the components of the system. The deviation between the simulated data and the experimental data is about 3 %. This deviation is likely to be as a result of heat loss, which is very dificult to be controlled.

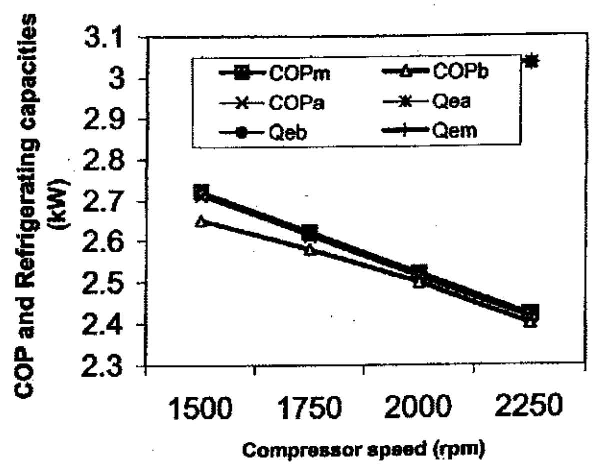

FIGURE 4: *Effect of Compressor Speed on COP, Refrigerating Capacities and Flow Rates*

3.3 Influence of Charge Loss on Performance

Figure 5 shows the sensitivity of coefficient of performance (COP) to insufficient and excesses refrigerant charge. Performance sensitivity to charge is primarily as a result of heat transfer effects and more specifically to evaporator starvation.

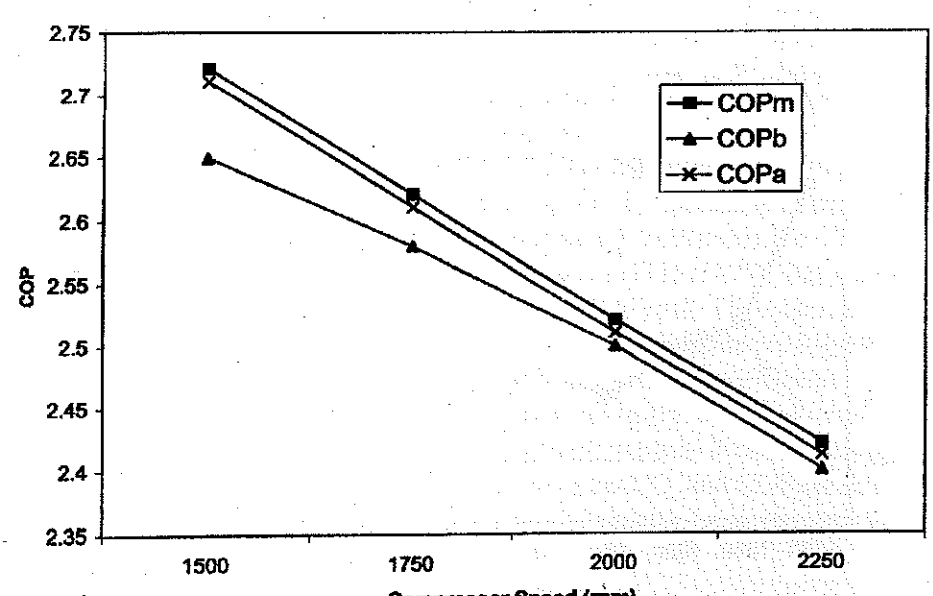

FIGURE 4a: *Effect of Compressor Speed on COP*

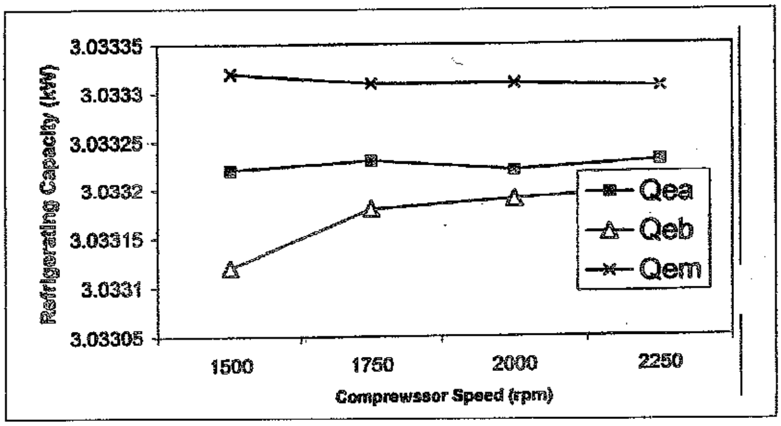

FIGURE 4b: *Effect of Compressor Speed on Refrigerating Capacity*

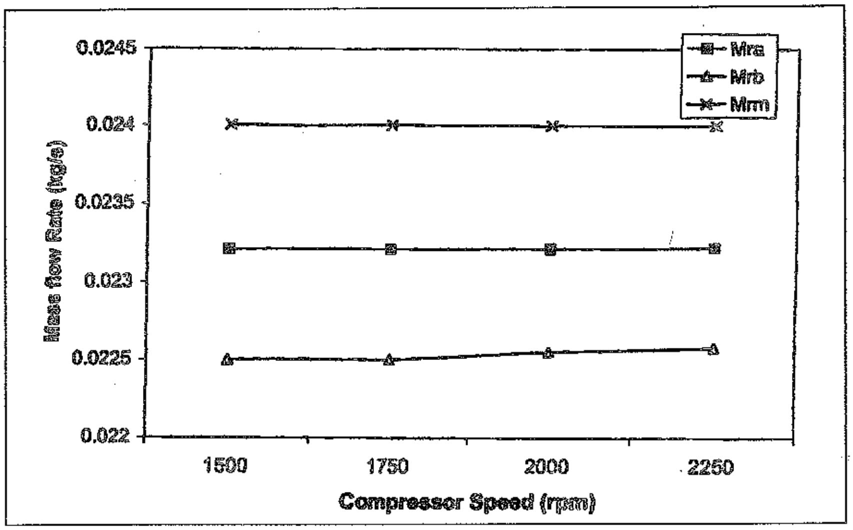

FIGURE 4c: *Effect of Compressor Speed on Mass Flow Rate*

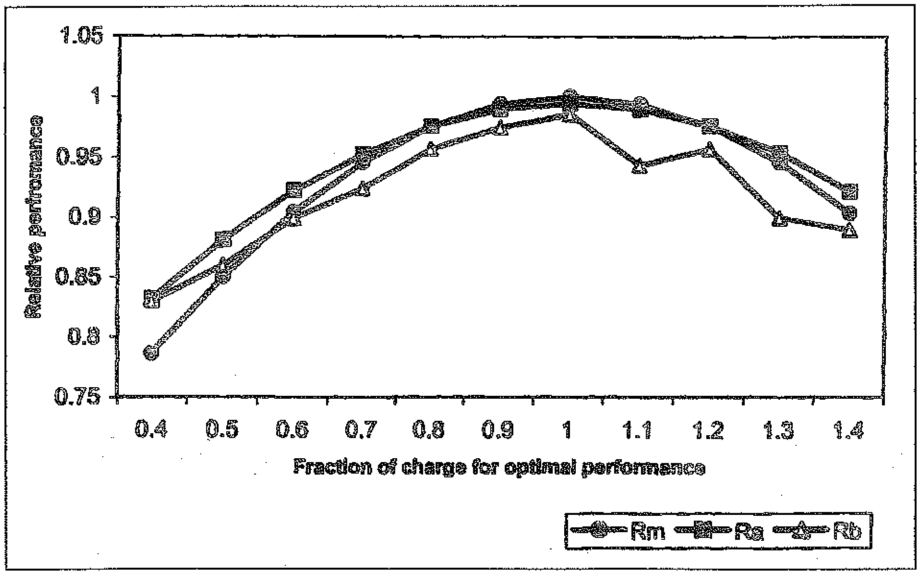

FIGURE 5: *Influence of Charge Loss on Performance*

From **Figure 5**, it could be seen that the three systems agreed well on the under charging side. Nevertheless, rig b shows the greatest discrepancy. It is expected that at the optimal COP, the three systems will indicate a relative performance of unity (1.0). At this point, however, only the model gives the expected value, while rig a gives a value of 0.994 (about 6 %) difference. This may be attributed to heat losses, since rig a could not be isolated from the immediate environment and also to the level of accuracy of the measuring instruments used. In the case of rig b, a value of 0.986 (about 1.4 % difference relative to the simulation model and 0.81 % relative to rig a). These figures show that the relative performance of rig a is about 2.3 times that of rig b. Since the experiments were performed under the same conditions and the same instruments were used in the measurements, this discrepancy may be attributed to the effect of reaching a balanced point for the system components. This effect is also strongly pronounced on the over-charging side of the curve. It could be seen that performance characteristics of rig b deviated widely from the normal as predicted by the simulation model.

There is a strong correlation between the simulated model and rig a, even at undercharging and overcharging conditions. This shows that the simulated model can be used for the optimal design of refrigeration systems.

4. Conclusion

The developed model is accurate for the design and optimisation of vapour compression refrigeration systems.

The present analysis has shown that simulated results are comparable to the actual system from both the quantitative and qualitative point of view. Under the same operational conditions, maximum absolute deviations of the variable parameters – mass flow rate, condensing temperature and compressor speed – are within the range of 18.5%, though most of the simulated results are within the range of 6.54 % from an ideal cycle.

As presented in this work, the COP and all other system parameters are calculated as accurately as possible (within ±1.5% of the actual values). The model is demonstrated to be a useful tool for the design and performance evaluation of vapour compression refrigeration systems.

Nomenclatures

C	=	Heat capacity (kJ/kg.K)
COP	=	Coefficient of Performance
Q	=	Rate of Heat Rejection or Absorption (kW)
R	=	Relative Performance
T	=	Temperature (K)

Subscripts

a	=	rig a
b	=	rigb
c, cond	=	Condenser
e, evap	=	Evaporator
1	=	Evaporator
2	=	Condenser
c	=	Calculated
m	=	Model

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