NUMERICAL MODEL TO OPTIMIZE THE REFRIGERANT CHARGE FOR MAXIMUM REFRIGERATION CAPACITY

by

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Refrigeration systems require optimal amount of refrigerant for maximum system performance. Undercharged or overcharged systems experience reduced efficiency and accessories deterioration. Optimal amount of refrigerant to be charged in a refrigeration system depends on the physical and thermal dynamic properties of the evaporator and the refrigerant. This paper presents formulation of a numerical model that can be used in determination of optimal amount of refrigerant charged in a system for maximum cooling rate as hence maximum system performance. Rayleigh's method of dimensional analysis was used obtain the relationship between the maximum cooling rates of direct expansion evaporators as a function of thermodynamic properties of refrigerant R-134a, Different sizes of evaporator were fitted in the refrigeration system and charged with systematically varying amount of refrigerant until a maximum cooling rate was determined. The variation of pressures and temperatures both at the inlet and exit of the evaporator were observed and analyzed. The cooling rate of the numerical model formulated was compared with the cooling rate of the actual physical refrigeration system. A t-test of 95% confidence interval indicated no significance difference between the numerical model, and the physical refrigeration system.

Key words: *dimensional analysis, R-134a, direct expansion evaporators, refrigeration systems*

Introduction

A sight glass installed in the liquid line of a refrigeration system is used in the determination of optimal amount of refrigerant on a commercial refrigeration system. Optimal amount of refrigerant is assumed when the liquid refrigerant flows smoothly in the sight glass without the presence of bubbles according to Ballaney [1]. Dmitriyev and Pisarenko [2] observed that the smooth flow of the liquid refrigerant in the sight glass was hampered by formation of bubbles from evaporation of the liquid refrigerant. There was also turbulent flow in the thermal expansion valve (TEV), and fluid friction in the pipes according to flow of fluids, in pipes by Douglas and Matthews [3]. Primal *et al*. [4] argued that due to interference of the smooth flow of the liquid in the sight glass, the optimal amount of refrigerant cannot be accurately by a sight glass, Dembi [5] showed that the optimal amount of refrigerant could be determined by the hissing sound produced by the expansion of the liquid in the (TEV). However different evaporators

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produce varying intensity of hissing sound and thus the sound cannot quantify the optimal amount of refrigerant in a refrigeration system.

This paper investigates the relationship between pressures and temperatures or refrigerant R-134a as it enters and leaves at the exit of direct expansion evaporators when charged with systematically varying amount of refrigerant. The physical parameters and the thermal dynamic properties of refrigerant R-134a were investigated to determine their influence to the optimal cooling rate on evaporators. Nae and Jin [6], and Raja *et al*. [7] established various parameters that influence the evaporator cooling rates and showed that there exists a maximum cooling rate that corresponds to optimal amount of refrigerant charge in the evaporators. Raj *et al*. [8] argued that there exists a linear relationship between evaporator cooling rates the pressures and temperatures of the refrigerant in the evaporator, together with the enthalpy of evaporation of the liquid refrigerant. Rayleigh's method of dimensional analysis as stipulated by Douglas and Matthews [3] was used to determine the relationship between the physical, thermodynamic properties of the evaporator and the refrigerant, respectively.

Experimentation

Nae and Jin [6], Raja *et al*. [7], and Raj *et al*. [8], identified the parameters that influence the evaporator cooling rates of the direct expansion evaporator as mass flow rate of the refrigerant in a refrigeration system (M), temperature change between the inlet and exit (ΔT), rate of pressure changes between the inlet and exit of the evaporator (ΔP), effective surface area of the evaporator (A), and the specific heat capacity of the refrigerant (C_P). The five parameters were used to establish the maximum cooling rate achievable in a refrigeration system, with the corresponding amount of refrigerant charged in the system as illustrated by Douglas [3].

Rayleigh's method of dimensional analysis was used to express the maximum evaporator cooling rate (Q) as a function of the five variables as:

$$Q = \phi(C_P, M, \Delta P, \Delta T, A) \tag{1}$$

where ϕ is a function of five variables. The variables were expressed in their respective indices as:

$$Q = C(C_P^b, M^d, \Delta T^x, \Delta P^y, A^z) \tag{2}$$

where C is a constant. The values of indices b, d, x, y, and z were obtained by expressing the variable in their respective fundamental dimensions of M, L, T, and K. The variables were expressed in their fundamental units as, $\dot{Q}$. Rate of evaporator cooling rate per unit length (M, L, T^{-3}, K^0), C_p – specific heat capacity of the refrigerant (M^0, L^2, T^{-2}, K^{-1}), M – mass flow rate of refrigerant in the system (M, L^0, T^{-1}, K^0), ΔT – temperature change per unit length of evaporator (M^0, L^{-1}, T^0, K), ΔP – rate of pressure drop in the evaporator, (M, L^{-1}, T^{-3}, K^0), and A – area of evaporator (M^0, L^2, T^0, K^0). The corresponding right side indices were equated to the left side indices of eq. 2 leading to eq. 3:

$$\begin{aligned} MLT^3K^0 &= C[(M^0L^2T^2K^{-1})^b(ML^0T^{-1}K^0)^d \\ &(M^0L^{-1}T^0K)^x(ML^{-1}T^{-3}K^0)^y(M^0L^2T^0K^0)^z] \end{aligned} \tag{3}$$

The left side dimensions were equated to the corresponding dimensions on the right hand side. The following equations were formed by equating left side and right side equations:

$$M: 1 = d + y \tag{4}$$

$$L: 1 = 2b - x + y - 2z \tag{5}$$

$$T: -3 = -2b - d - 3y \tag{6}$$

From eq. 4, $d = y - 1$.

Substituting d with $y - 1$ in eq. 6, $b = 1 - y$. Substituting b in eq. 5, $x = y - 1$, while substituting x in eq. 5, $z = y$.

Indices b, d, x, y, and z were expressed in terms of y where: $b = 1 - y$, $x = y - 1$, $d = 1 - y$, and $z = y$.

Equation 1 was thus expressed as:

$$\dot{Q} = C[C_p^{(1-y)} M^{(1-y)} \Delta T^{(1-y)} \Delta P^y A^y] \tag{7}$$

and simplified as:

$$\dot{Q} = C\left[C_P M \Delta T \left(\frac{\Delta PA}{C_P M \Delta T}\right)^y\right]$$

and was found to be similar to $\ln Q = \ln R + y \ln N$.

The value of y was obtained by plotting a graph of $\ln R$ against $\ln N$. Direct expansion evaporators of different lengths (L) and diameter (d) and of similar thickness were measured and their effective surface areas calculated as πdL [m^2]. Evaporators of effective surface areas of 0.01571 m^2, 0.03142 m^2, 0.04712 m^2, and 0.062832 m^2, were randomly selected and flanged on both ends and each evaporator was connected to the refrigeration system. The system was evacuated using a refrigerant recovery unit to a pressure of 100 kPa, by connecting a refrigerant recovery unit to the vacuum discharge valve. The system was systematically charged with a predetermined amount of refrigerant R-134a. This was achieved by placing the refrigerant cylinder on a digital display weighing scale and connecting the cylinder to the system, via a low pressure charging valve using a hose as shown in fig. 1. The amount of refrigerant charged in the system was obtained by the difference between the final and initial weight reading of the digital weighing scale.

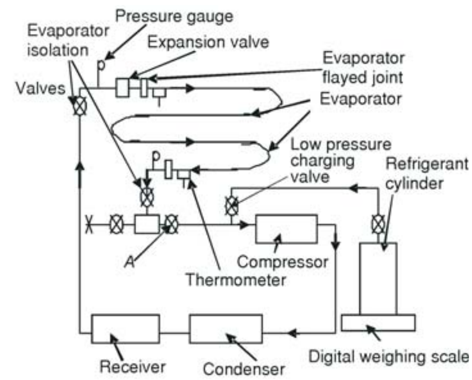

Figure 1. Schematic equipment set-up for charging the system

Cooling rate (Q) for each evaporator was determined as a function of the mass of refrigerant charged in the system (M), specific heat capacity (C_P) of the refrigerant R-134a, and the temperature difference of the refrigerant at the exit and inlet (ΔT). Digital probe thermometers whose accuracy was ±0.1 was attached at the two points were used to obtain the temperatures differences, when stable conditions were attained in terms of inlet and exit temperatures and pressures their values were read and recorded. The cooling rate was obtained by $Q = MC_P\Delta T$ the amount of refrigerant charged in the system was systematically increased for each size of evaporator and the cooling rate determined, alongside the corresponding amount of refrigerant charged in the system. The specific heat capacity at each temperature was obtained from Bitzer (Bitzer technical information, Catalogue 73/4 Refrigerants table (2000), Bitzer Kuhlmaschinenbau GmbH, Sindelfingen, Germany, www.bitzer de/eng/Home) [9]. The specific heat capacity was compared with their corresponding enthalpies obtained from Rohsenow and Griffin [10], and Wyatt [11]. The process was repeated three times, for each amount of refrigerant added in the system and an average value was calculated. After the maximum cooling rate for each evaporator was obtained, further increase of the refrigerant resulted in a decrease of the rate of cooling rate. The graphs showing cooling rate against amount of refrigerant of different evaporator sizes of were as obtained.

Data obtained from the four evaporators cooling rate, of different sizes was used to plot the ln$Q$ against ln$N$ summary of each evaporator cooling rate was obtained.

The gradient of the graph with a coefficient of relation 0.988 in fig. 2 was equal to 0.1295 as indicated in the equation of line $y = 1.00797 + 0.1295$ of the graph.

The value of the constant C in eq. 7 was obtained by plotting the maximum cooling rate of the physical model and the numerical/mathematical expression of the evaporator whose effective surface areas were; 2.422 m^2, 3.412 m^2, 4.281 m^2, and 5.362 m^2 charged with different amount of refrigerant until the maximum evaporator cooling rate was established for each size of evaporator. The maximum cooling rate were noted from the physical and the numerical were as indicated in fig. 3.

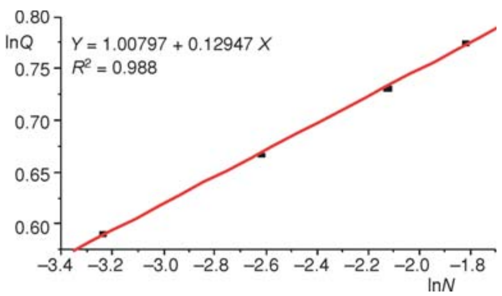

Figure 2. Graph of ln$Q$ against ln$N$

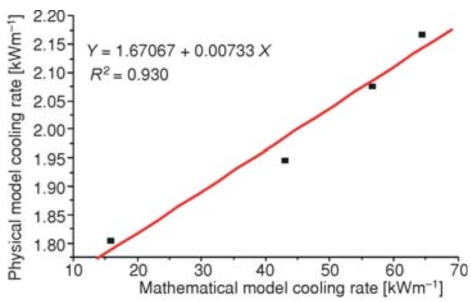

Figure 3. Physical and mathematical maximum model cooling rates

The gradient of the curve plotted of the physical model cooling rate against the numerical model gave the value of the constant C as 0.00733, as indicated in equation of the line in the graph. The maximum evaporator cooling rate was expressed as:

$$Q\,[\mathrm{kWm}^{-1}] = 0.007\left[C_P M\Delta T\left(\frac{\Delta PA}{C_P M\Delta T}\right)^{0.1295}\right] \tag{8}$$

The physical model cooling rates for different mass of refrigerant charged in evaporators against their corresponding maximum cooling rates were plotted and the same compared with values obtained with numerical model. The results were as shown in fig. 4.

A one sample t-test for the physical and numerical/mathematical model maximum evaporator cooling rates performed, indicated that there was no significance difference in the maximum cooling rates between the physical and numerical model cooling rates.

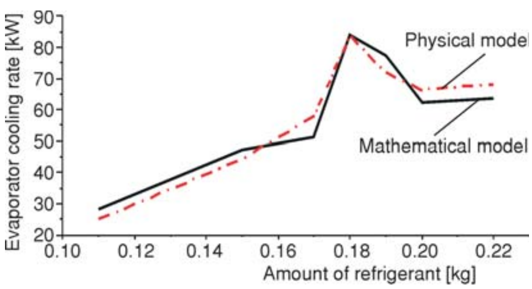

Figure 4. Physical and mathematical evaporator cooling rate curve of area 0.06285 m^2 per meter

Results and discussions

The evaporator cooling rates obtained for different sizes of evaporators were as shown in figs. 5-8. In three graphs 2, 3, and 4, there was a maximum cooling rate that corresponded with the optimal amount of refrigerant charged in the system, followed by a sharp decrease of the cooling rate with further additional of the refrigerant in the system. This was in accordance with Raj [8] who showed that there exists maximum refrigeration effect that corresponds with optimal amount of refrigerant in a refrigeration system. Figure 8 showed a poor drop of cooling rate as compared to the others graphs. This could be due to high ambient temperature around the evaporator. According to Young and Bansal [12], Rationum and Avalian [13] and Cemil *et al*. [14] high evaporator ambient temperatures increases the specific volume of the refrigerant which chocks the evaporator, and thus reducing the rate of drop in the cooling rate. It was however noted that the numerical model indicated a sharp decrease of cooling rate, unlike the physical model. This is a clear indication that the numerical model is less influenced by external factors and is therefore accurate and reliable.

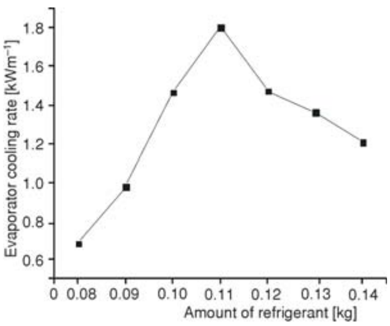

Figure 5. Evaporator cooling rate curve of an evaporator of area 0.015716 m^2 per meter

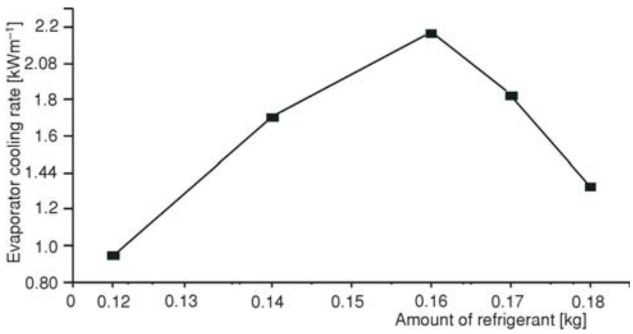

Figure 6. Evaporator cooling rate curve of area 0.06283 m^2 per unit length

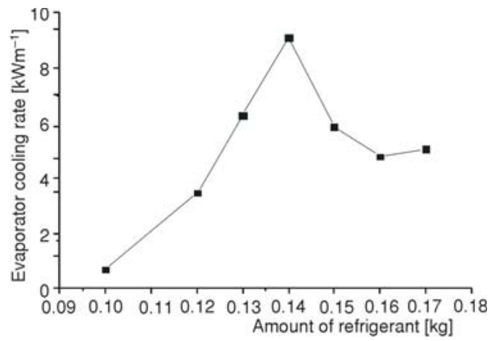

Figure 7. Evaporator cooling rate curve of an evaporator of area 0.03142 m^2

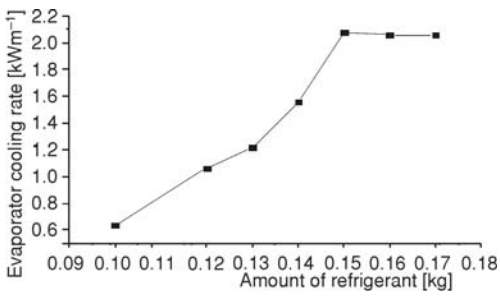

Figure 8. Evaporator cooling rate curve of area 0.04712 m^2 per unit length

The curves obtained by comparing numerical and physical models as shown in fig. 9 showed the expected variation of cooling rates and amount of refrigerant in the system had con-

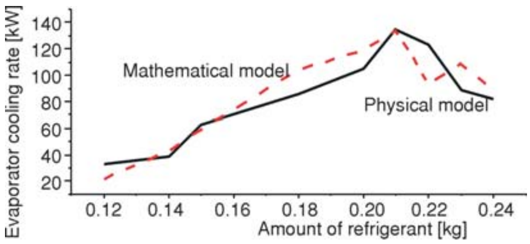

Figure 9. Physical and mathematical evaporator cooling curve of area 0.08180 m^2 per meter

sistency with Ratianum and Avalian [13] Samuel [15] Piret and Isbin [16], and Rogers and Mayhew [17].

The maximum direct expansion evaporators cooling rates was expressed as:

$$Q = 0.007\left[C_P M\Delta T\left(\frac{\Delta PA}{C_P M\Delta T}\right)^{0.1295}\right]$$

Conclusions

Rayleigh's method of Dimensional analysis was successfully used in obtaining a numerical expression relating the optimal amount of refrigerant R-134a with the maximum cooling rate of direct expansion evaporators of different sizes. Increase of the amount of refrigerant R-134a in a refrigeration system increased the rate of evaporator cooling rate to a maximum level which corresponded with the optimal amount of refrigerant in the system. The evaporator cooling rate decreased with further addition of the refrigerant in the system, this compared well with the curves obtained by Samuel [15] when similar experiments were simulated and analysis done by X-ray. The results of maximum evaporator cooling rates obtained in both numerical expression and the physical model were found to have similarities with the *t*-test .The numerical model is thus accurate and dependable in determination of optimal amount of refrigerant in a refrigeration system.

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