Study Vapour Compression Refrigeration and Compare the Performance of Working Refrigerants

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Abstract

Part-load and dynamic performance were studied in this research. The aim of the part-load analysis is to estimate the performance of a refrigerator to variations in operating conditions resulting from either changes in condenser sink temperature, evaporator source temperature, coolant or chilled fluid flows or compressor speed, where dynamic study involved developing equations which predict the movement of refrigerant into a system and the change of state of refrigerant in the system components over time. The way chosen has been the development of a mathematical model of the components of a refrigeration system suitable to predict the behaviour of the relevant process. Simulation models were used to compare the performance of working refrigerants R12, R134a, R413a, R-22 and R-407c.

Key words: Refrigeration, refrigerant, dynamic.

Introduction

The classical approach to the mathematical analysis of a refrigeration system describes the physical events taking place with equations derived by applying a steady flow balance to each section in turn. These steady state values are used to determine the size and thermal capacities of components, which make up the system but equipment that operates efficiently under design conditions may perform poorly at off-design conditions.

The goal of current research first is to study the part-load behaviour of vapour compression refrigeration systems using the $\varepsilon - NTU$ method and Wilson plot and following study dynamic behaviour of the system.

Part-Load Study

$\varepsilon - NTU$ method is the most practical method to use to predict the performance of the vapour compression refrigeration systems under part-load conditions. Wilson (1915) devised a method to determine individual resistance from an overall resistance. The heat transfer coefficient for the coolant side is a function of the Reynolds and Prandtl numbers. Wilson proposed a plot of $\frac{1}{UA}$ based on mass flow rate

$$\frac{1}{UA} = C \ \frac{1}{(\dot{m})^n} + RH \tag{1}$$

Where:

$$RH = \frac{1}{\alpha_o A_o} + \frac{x}{k A_m}$$

α_o is refrigerant side heat transfer coefficient and C is a constant available from the manufacturers.

$n = 0.6$ for water or ethylene glycol flowing through shell [3]

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$n = 0.8$ for water or ethylene glycol flowing inside tubes for turbulent flow [3]

Method of Solution

It was assumed that the overall heat transfer coefficient at the design-point UA_d, has already been calculated from first principles (design conditions) and will be given as input data for part-load performance.

Once the coolant heat transfer coefficient, is obtained, the sum of the refrigerant and wall resistance $(RH)_d$, at the design point can be determined. $(RH)_d$ is assumed to remain constant throughout and is used as input of the program.

The heat transfer coefficient for the coolant side is a function of the Reynolds and Prandtl numbers. If the coolant temperature change is relatively small throughout, then the properties of the coolant can be assumed to remain constant, and therefore the heat transfer coefficient can be simplified to

$$\alpha_i = C_1 (C_r^n) \quad (2)$$

Where C_1 is a constant independent.

The overall resistance at part-load can then be calculated from:

$$\frac{1}{UA} = C_2 \frac{1}{(C_r)^n} + RH \quad (3)$$

The idea can be followed using the coolant mass flow rate instead of its mean velocity.

$$\dot{m} = \rho A C_r \quad (4)$$

The overall resistance at part-load can then be calculated from

$$\frac{1}{(UA)_{off}} = C_4 \frac{1}{(\dot{m}_{off})^n} + RH_d \quad (5)$$

$\dot{m}_{off}$ = Coolant mass flow rate at part-load

Beans (1992) used the following relationship for overall resistance of the heat exchanger for the part-load performance of air-to air refrigeration systems:

$$(UA)_{off} = (UA)_d \left(\frac{\dot{m}_{off}}{\dot{m}_{cd}}\right)^{0.4} \quad (6)$$

Computer Model

The simulation is initiated by assuming a starting value for evaporation temperature, and condensing temperature. The program begins by determining $(UA_e)_{off}$ and $(UA_c)_{off}$ for the evaporator and condenser. Given the input values to specify the refrigeration system, the operating conditions representing the part-load situation are numerically searched for. To accomplish this, the computer program logic has to satisfy the energy balance between refrigerant and coolant. The model has been applied to predict the performance of working refrigerants R12, R134a, R22, R407c and R413a in a water-to-water refrigeration system. It is necessary to select design-point conditions, prior to carrying out the simulation model. Design temperatures for the condenser and evaporator were selected as 50 °C and -5°C respectively.

Figure 1 provides a comparison of cooling capacity for those working refrigerants over a range of compressor speed. The cooling capacity of all working refrigerants increased as the compressor speed increased. Increasing compressor speed decreases the evaporator pressure, resulting in a decrease in the refrigeration effect and consequently cooling capacity. On the other hand, an increase in refrigerant mass $(\dot{m} = \rho\, v\, n\, \eta_{vol})$ [3] compensates for the decrease in the refrigeration effect, resulting an increase in the cooling capacity. The cooling capacity of R22 and R407c doubled as the compressor increased from 500 rpm to 2000 rpm. The cooling capacity of R413a, R134a and R12 are similar at lower compressor speeds, whilst R12 shows higher cooling capacity compared with that of R134a and R413A at higher compressor speeds.

Figures 2 compare the coefficient of performance against condenser water inlet temperature. The graphs show that reducing condenser water inlet temperature increases the COP of the refrigeration system. Reducing the condenser water inlet temperature reduces the pressure differential across the compressor, resulting in a decrease in the compressor power. On the other hand a decrease in the condensing temperature is associated with an increase in the evaporator refrigeration effect and consequently evaporator cooling capacity and COP.

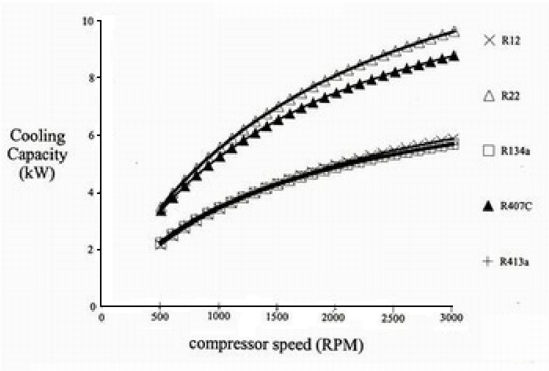

Fig.1 Cooling capacity against compressor speed

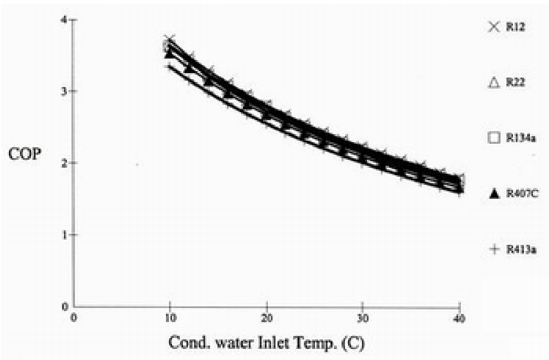

Fig. 2 COP against condenser water inlet temperature

Application 2: Dynamic Study

Evaporator model:
The evaporator was divided into two regions, evaporation region and superheating. The approach to analysis is based on the lump model for two regions. The temperature in the evaporation region is assumed equal for the vapour and liquid and thus the heat transfer between the vapour and liquid is neglected.

Evaporation region
The evaporation region is considered as one lumped model for coolant, tubes and refrigerant, having variable evaporation length and consisting of three zones, the refrigerant, the coolant and the tube.

Refrigerant Model:
The energy balance equation for refrigerant in the evaporation region can be written:

$$\frac{d}{dt}(M_{ve}h_{ve}+M_{le}h_{le})=\dot{m}_v h_{ei}-\dot{m}_{com}h_{ve}+\alpha_{re}A_{ee}(T_{pe}-T_e) \quad (7)$$

Tube model:
By applying the law of conservation of energy to the tube wall the following equation can be written:

$$M_{pe}(c_p)_{pe}\frac{dT_{pe}}{dt}=\alpha_{ce}A_{eo}(T_{cea}-T_{pe})$$
$$-\alpha_{re}A_{ee}(T_{pe}-T_e) \quad (8)$$

Coolant model:
From the law of conservation of energy,

$$M_{cea}(c_p)_{cea}\frac{dT_{cea}}{dt}=\dot{m}_{ce}(c_p)_{cea}(T_{cei}-T_{ceo})\frac{L_e}{L_t}$$
$$-\alpha_{ce}A_{eo}(T_{cea}-T_{pe}) \quad (9)$$

Continuity equation:
The continuity equation for the evaporation region can be presented in a combined form of vapour and liquid as follows:

$$\frac{d}{dt}(M_{ve}+M_{le})=\dot{m}_v-\dot{m}_{com} \quad (10)$$

Mass of vapour refrigerant accumulated in the evaporation region, M_{ve} , is given by:

$$M_{ve}=\alpha_{vfe}\,A_e\,\rho_v\,L_e$$

Mass of liquid refrigerant accumulated in the evaporation region, M_{le}, is given by:

$$M_{le}=(1-\alpha_{vfe})A_e\,\rho_l\,L_e$$

Where $A_e=n_e\pi\frac{d_i^2}{4}$

Hughmark [2] proposed an equation that describes the mean void fraction in the evaporator as a function of refrigeration mass flow rate, evaporation temperature and inlet quality .

Method of Solution:
Substituting for M_{ve} and M_{le} into Equation (10) gives:

$$\frac{d}{dt}(\alpha_{vfe} A_e \rho_v L_e + (1-\alpha_{vfe}) A_e \rho_l L_e)$$

$$= \dot{m}_v - \dot{m}_{com} \qquad (11)$$

Substituting for α_{vfe}, M_{ve} and M_{le} and using the following relations:

$$\frac{d\rho_v}{dt} = \frac{d\rho_v}{dT_e}\frac{dT_e}{dt}$$

and

$$\frac{d\rho_l}{dt} = \frac{d\rho_l}{dT_e}\frac{dT_e}{dt}$$

Partial equations will be converted to ordinary differential equations.

To keep the numerical solution simple for the dynamic part, a single step method, "Euler's method", has been used for solving the differential equations. It uses a truncated Taylor series expansion and may be expressed as follows:

$$f_{(f+dt)} = f_t + \frac{df_t}{dt} dt \qquad (12)$$

The global error of the predicted solution is of the order of the time step, Δt. Hence, to achieve reasonable accuracy using Euler's method, it is necessary that the time step is made sufficiently small.

Evaporator Superheat Region Model

The superheat region connects the evaporation region to a single pipe in which the feeler bulb is attached. The length of the superheat region may be approximated by:

$L_s = L_t - L_e$

Like the evaporation region, the evaporator superheat region is considered as one lumped model consisting of three zones, the refrigerant, the coolant and tube wall Assuming the superheated refrigerant is incompressible makes the continuity equation trivial, therefore only the energy equation is used.

Refrigerant model:

Energy balance equation for superheated vapour refrigerant:

$$M_{se} c_{se} \frac{dT_s}{dt} = \alpha_{es} A_{es} L_s (T_{pes} - T_s)$$

$$- \dot{m}_{com} c_s (T_{sh} - T_e) \qquad (13)$$

Where; $M_{se} = \rho_s\, n_e\, \pi\, d_i\, L_s$ and $A_{es} = n_e\, \pi\, d_i\, L_s$

Tube model:

Energy balance equation for the tube walls:

$$\rho_{pes}(c_p)_{pes} A_{pe} \frac{dT_{pes}}{dt} = \alpha_{ce} \pi n_e d_o (T_{ceas} - T_{pes})$$

$$- \alpha_{es} \pi\, n d_i (T_{pes} - T_s) \qquad (14)$$

Coolant model:

Energy balance equation for the coolant;

$$\rho_{ceas}(c_p)_{ceas} A_{cea} \frac{dT_{ceas}}{dt} = \frac{(c_p)_{ceas} \dot{m}_{ce}}{L_t}(T_{cei} - T_{ceso})$$

$$- \alpha_{ce} \pi\, n d_o (T_{ceas} - T_{pes}) \qquad (15)$$

Similar solution as the evaporation region used here. The Euler's method again was used to solve these equations for $\frac{dT_s}{dt}$, $\frac{dT_{ps}}{dt}$ and $\frac{dT_{ceas}}{dt}$.

Condenser

The same general assumptions made for the evaporator model were also applied to the condenser. The approach taken in modelling the condenser is to consider it as one lumped region (condensation region). The time step of the dynamic model of the condenser is assumed to be the same as the time step of the evaporator, to avoid problems when combining the models.

Since the pressure drop of the refrigerant in the condenser shell was ignored, only the continuity and energy equations should be found as basic equations to model the condenser.

Thermostatic expansion valve

The mass flow rate through the expansion valve is assumed to be depend on the orifice throat area, condensing pressure and evaporator pressure [4].

$$\dot{m}_l = C_d\, A_v \sqrt{\rho_l (P_c - P_e)} \qquad (16)$$

A_v is a linear function of the degree of superheating. The valve area can be approximated by;

$$A_v = \frac{T_{sh} - T_e - SSH}{MSH} A_t \qquad (17)$$

Thus the mass flow rate through the expansion valve is:

$$\dot{m}_v = C_d \left(\frac{DSH - SSH}{MSH} A_t \right) \sqrt{\rho_l (P_c - P_e)} \qquad (18)$$

Bulb model

The model here is a simplified version of Yasuda et al. [6], where feeler bulb was described by three lumped dynamic models making use of separate control volumes, refrigerant line, bulb wall and bulb contents.

Compressor Model

The compressor can be described by a quasi-static equation because its time constants are found to be very short compared with those of the evaporator and condenser.

The mass flow through the compressor can be calculated by:

$$\dot{m}_r = \rho \ V \ \eta_{vol} N \qquad (19)$$

where the volumetric efficiency of the compressor, η_{vol} is calculated by:

$$\eta_{vol} = 0.9 + C - C(\frac{P_c}{P_e})^{\frac{1}{n}} \qquad (20)$$

The component models are combined to configure a dynamic simulation model for the entire refrigeration system. The specification and operating conditions of all components and initial values for all the independent variables are given as input values in the simulation program. Once the integration time step and maximum operation time are determined, a series of subroutines are used to calculate the mass and energy transfer in the condenser, evaporator, compressor and expansion valve. In each iteration the local properties of the refrigerant in the system are determined. This is repeated for each time step until the end of the transient simulation is reached.

Results and Discussions

The model was employed to predict and compare the dynamic behaviour of the working refrigerants R12, R134a, R22, R413a, and R407c.

Figure 1 through 3 is samples of the transient behaviour (start-up). These figures show the variation of system pressure and temperature. Numerical results depicted in these figures indicate that the system stabilised after periods of 300 seconds from start-up. At time equal to zero, evaporating and condensing temperatures are equal. Immediately after the compressor is started, the condensing pressures and temperatures rise rapidly and then gradually approach their steady state value. The evaporating pressures and temperatures drop rapidly at first and then gradually approach a steady state value. The sudden drop in evaporating pressure and following that in the evaporator temperature causes the refrigerant in the evaporator to vaporise rapidly. The evaporator pressure and temperature increase after about 20 seconds. That represents the instant when the feeler bulb pressure at the outlet of the evaporator reaches a point where it forces the expansion valve to open, thus increasing the evaporating pressure and temperature to steady values. After the first time step, the compressor delivers refrigerant to the condenser against a rapidly increasing pressure differential. As the evaporator pressure decreases the density of the refrigerant entering the compressor decreases. On the other hand as the compressor pressure ratio increases the volumetric efficiency of the compressor decreases.

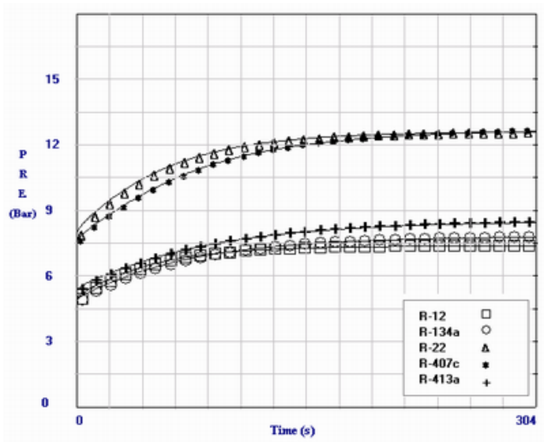

Figure 3 Variation of condenser pressure

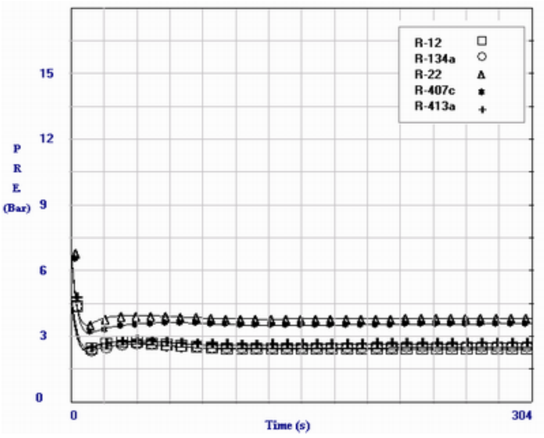

Figure 4 Variation of evaporator pressure

Conclusions

The goal of the present work was to develop a mathematical model for simulating the part-load and starting-up transient behaviour of a water-to-water vapour compression refrigeration system.

The model was derived with an emphasis on the evaporator and expansion valve where a $\varepsilon - NTU$ model used for part-load study. repeated for each time step until the end of the transient simulation was reached.

The model was used to study and compare the behaviour of working refrigerants of R12 and R22 with their substitutes R134a, R-413a (a refrigerant blend of R134a/R218/R600a, 88% /9% /3%, composition by mass) and R407c (a zeotropic mixture of R134a, R125 and R32 in a 52% 25% 23% composition by mass) in a water-to-water refrigeration system.. A comparative study has been made to compare the performance of R134a was shown to have an excellent performance compared with that of R12. It results also showed that, R-413a have satisfactory performance compared with that ofR12 and R134a and the behaviour of R407c was shown to be comparable with that of R22.

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