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INVESTIGATION OF THE REFRIGERANTS CHARACTERISTICS IN VAPOR COMPRESSION SYSTEMS

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ABSTRACT: The energy efficiency improvement of the refrigeration system to improve the operation quality makes it unavoidable to strive for the refrigeration system operation. Nevertheless, the processes taking place in it should be as accurate as possible to describe the underlying physical and mathematical model development and refinement. The experimental investigation of any refrigeration system is usually very complicated, mainly due to the financial costs and the large number of variables involved. The use of numerical models can reduce the costs and also facilitate understanding the phenomena related to the problem. The article aims to present and analyze the behavior components of the vapor-compression refrigeration system in case of various refrigerants. Refrigerants included in the present analysis are R22/R134a/R407C/R410A. The simulation program is based upon steady state mathematical models of the refrigeration circuit including the compressor, heat exchangers and thermostatic expansion valve. The simulation results have been presented in a graphic.
KEYWORDS: Refrigeration system, Refrigeration, Mathematical model, Simulation, COP

INTRODUCTION

The European Commission accepts a proposal package [1] of future indicators at the beginning of 2008. The aim of this is to decrease the rapidly increasing greenhouse gas emissions based on the 2007 data. Furthermore to increase the renewable energy sources in the total energy consumption in the proportion of 20% by the year 2020.

The utilization of renewable energy sources is influenced by several factors. Besides the natural environment, the economic conditions are also major factors affecting the case of renewable energies. The fossil fuel prices and conditions of other energy costs are significantly determined by the demand for renewable as well as the amount of state aid and government fiscal policy application.

The energy efficiency improvement of the refrigeration system to improve the operation quality makes it unavoidable to strive for the refrigeration system operation. Nevertheless, the processes taking place in it should be as accurate as possible to describe the underlying physical and mathematical model development and refinement.

In addition to the structural design and dimensions of components of the refigeration system, the refrigerant is also a major influencing factor. The refrigerant disposes of a lot of thermodynamic characteristics which differ among themselves substantially. The thermodynamic characteristics are major contributors to the suitable choice of refrigerant, the choice of components of the refrigeration and cooling system, the refrigerant may be necessary when replacing.

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Many researchers dealt with the description of the steady state behavior of a vapor compression refrigeration system such as Koury et al. [2] proposed a model for a refrigeration system with distributed parameter model for heat exchanger, Jong Won Choi et al.[3], Bellman et al.[4], Yang Zhao et al.[5] and S. A. Klein et al.[6].

DESCRIPTION OF PHYSICAL SYSTEM

The vapor-compression refrigeration cycles consists of the four main components: evaporator, compressor, condenser and expansion valves. The refrigerant is the working fluid of the refrigeration system.

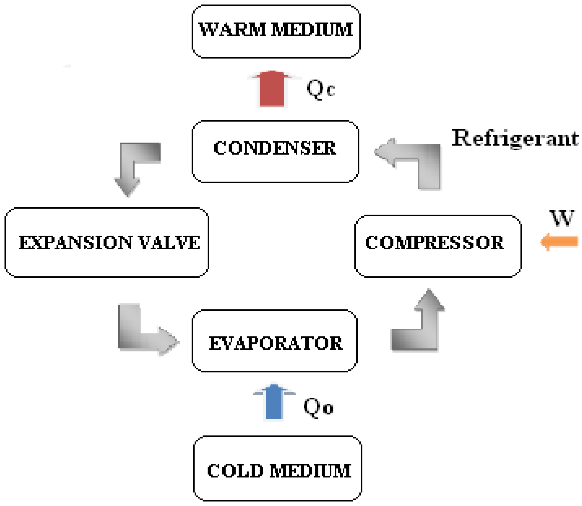

Figure 1. Energy flow diagram of the refrigeration system

In the evaporator, the refrigerant takes over the heat from the low-temperature primary fluid by vaporization. In the evaporator, the refrigerant is superheated and vapor is sucked in by compressor. With the invested mechanical work, it is brought higher level of energy. On the discharge side of the

compressor, the now hot and highly pressurized vapor is cooled in the condenser.
In the condenser, vapor provides the heat to the secondary fluid and then the refrigeration condenses. Condensation in horizontal tubes may involve partial or total condensation of the vapor. Depending on the application, the inlet vapor may be superheated, equal to 1.0 or below 1.0.
Superheated vapor enters the horizontal tube, which has a temperature below the saturation temperature of the vapor. The flow at this point in the tube is single-phase vapor flow. After the vapor cools and becomes saturated, condensation starts to occur on the inner wall of the tube.
Near the outlet of the horizontal tube, the vapor quality reduces to zero and the flow in the tube becomes single-phase liquid flow.
The condensed refrigerant then passes through a pressure-lowering expansion device.
The expansion device is to reduce the pressure and to regulate the refrigerant mass flow rate. The widely utilized expansion device is the thermostatic expansion valves. The thermostatic expansion is a valve for controlling the refrigerant flow by a sensor bulb placed in the evaporator discharge line and hence controls the mass rate by the degree of superheat.
The low pressure, refrigerant leaving the expansion device enters the evaporator, in which the refrigerant absorbs heat and boils. The refrigerant then returns to the compressor and the cycle is repeated.
The observed heat exchangers are counter-cross flow, shell and tube type. Tubes are made of copper and have a staggered layout. In the current case, the refrigerant flows through in finned tube bundle of heat exchangers, while primary and secondary fluid flow in the shell across the tube bundle.
In this case, the compression occurs on the principle of displacement reciprocating compressor. While the throttle is isenthalpic and occurs with variable cross-section of expansion valve.

MATHEMATICAL MODEL OF REFRIGERATION SYSTEM. Heat exchangers

The evaporation and the condenser are approached in a similar manner from the modeling point of view. Both are divided into regions associated to the phase of the refrigerant.
In the case of the condenser the superheated vapor, and the condensation are considered, whereas the evaporator is divided into the evaporating are superheated vapor regions.
For the each region, the overall heat transfer coefficients is evaluated by assuming that thermal resistance due to wall conduction, contact and fouling are negligibly small.
Heat exchanger energy balances:
Water side

$$Q = \dot{m} \cdot c_{pw} \cdot \Delta T_w \tag{1}$$

Refrigerant side
Condensing region

$$Q = \dot{m}_{ref} \cdot \Delta i_{lv} \tag{2}$$

Evaporating region

$$Q = \dot{m}_{ref} \cdot (i_v - i_i) \tag{3}$$

Single phase regions

$$Q = \dot{m}_{ref} \cdot c_{pref} \cdot \Delta T_{ref} \tag{4}$$

Overall Heat Exchangers

$$Q = A \cdot U \cdot LMTD \tag{5}$$

Condenser

$$Q_C = Q_{desuperheated} + Q_{condensing} \tag{6}$$

Evaporator

$$Q_o = Q_{evaporating} + Q_{desuperheating} \tag{7}$$

LMTD is the logarithmic mean temperature difference defined by:

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} \tag{8}$$

The heat transfer correlation can be written as:

$$\frac{1}{U} = \frac{A_o}{A_i \cdot \alpha_w} + \frac{1}{\eta_o \cdot \alpha_r} \tag{9}$$

Where is the finned heat transfer surface effiency given by the well known correlation:

$$\eta_o = 1 - \left(\frac{A_f}{A_o}\right) \cdot \left(1 - \eta_f\right) \tag{10}$$

The heat transfer coefficient of single-phase refrigerant vapor was calculated by the Dittus-Boelter correlation [7]

$$\frac{\alpha_{vap} \cdot d_i}{\lambda} = 0.023 \cdot \left(\frac{G \cdot d_{in}}{\mu}\right)^{0.8} \cdot \left(\frac{\mu \cdot c_p}{\lambda}\right)^{0.33} \tag{11}$$

Water-side heat transfer coefficient for staggered horizontal tubes is given by [7]

$$\frac{\alpha_w \cdot d_{out}}{\lambda} = C \cdot \left(\frac{G \cdot d_{out}}{\mu}\right)^{0.6} \cdot \left(\frac{\mu \cdot c_p}{\lambda}\right)^{0.33} \tag{12}$$

Kandlikar correlation [8] was used for the prediction of the heat transfer coefficient in flow boiling. The final correlation consists of two sets of constants. One for the convective evaporation dominated regime and the other for the nucleate boiling dominated regime. The correlation is:

$$\alpha_{kf} = \alpha_f \cdot \left(C_1 \cdot (Co)^{C_2} \cdot (25 \cdot Fr_f)^{C_5} + C_3 \cdot (Bo)^{C_4} \cdot Fn\right) \tag{13}$$

and the constants are given in the table below.

Table 1: Constants in Kandlikar (1990) correlation

constant	Convective evaporation	Nucleate boiling
C_1	1.1360	0.6683
C_2	-0.9	-0.2
C_3	667.2	1058
C_4	0.7	0.7
C_5	0.3	0.3

The correlation is calculated twice using each set of constants and the greater of the two values is used as the heat transfer coefficient.

$$\alpha_{tp} = \max\left|\alpha_n, \alpha_c\right| \tag{14}$$

For the two phase regions, in the condenser, Shah correlation [9] was selected to proposed heat transfer coefficient. The Shah correlation is a modified version of Dittus-Boelter single-phase heat transfer correlation. The two-phase model the reduced pressure refrigerant takes into account.

$$\alpha_{kf} = \alpha_f \cdot \left[(1-x)^{0.8} + \frac{3.8 \cdot x^{0.76} \cdot (1-x)^{0.04}}{p^{*0.38}} \right] \qquad (15)$$

Where the reduced pressure is: $p^* = \dfrac{p}{p_{crit}}$

In Eqs. (11) the single phase heat transfer coefficients are determined by the Dittus-Boelter correlation.

Compressor model

Refrigerant mass flow rate through the compressor is a function of compression ratio, refrigerant density and compressor speed, that is,

$$\dot{m} = f\left(\frac{p_c}{p_e}, \rho_{sz}, N\right) \qquad (16)$$

The relationship between compressor exit and inlet temperatures is given by:

$$T_{ny} = T_{sz} \cdot \left[\left(\frac{p_{ny}}{p_{sz}}\right)^{\frac{k-1}{k}} - 1 \right] \qquad (17)$$

Neglecting the thermal inertia effects, the indicated work is given by:

$$W = \frac{k}{k-1} \cdot p_{sz} \cdot v_{sz} \cdot \left[\left(\frac{p_{ny}}{p_{sz}}\right)^{\frac{k-1}{k}} - 1 \right] \qquad (18)$$

Isentropic effiencies [10] is correlated as a function of the pressure ratio between condensation pressure and evaporation pressure.

$$\eta = A + B \cdot \tau + C \cdot \tau^2 \qquad (19)$$

where A, B and C are the regression coefficients.

THERMOSTATIC EXPANSION VALVE MODEL AND REFRIGERANT

Thermostatic expansion valve is the valve that controls the refrigerants mass flow rate by sensing the degree of suction vapor superheat temperature. The enthalpy is assumed to be constant. The refrigerant mass flow is calculated by the following equation

$$\dot{m} = C \cdot \sqrt{2\rho \cdot (p_c - p_e)} \qquad (20)$$

where C is the characteristic constant of the valve.
The refrigerants used in this study are the R22, R134a, R407C, R410A. Calculation of the refrigerants and transport properties is performed by correlations written as computer code function, using thermodynamic properties coming from SOLKANE database [11]. The effect of the circulation of oil is not taken into account in the model.

INITIAL CONDITION AND VALUES

The mathematical models are simulated by the use of the software tool Solkane.
The initial conditions and values for the simulation:

- Refrigerant: R22, R134a, R407C, R410A
- Refrigerating capacity: $Q = 2kW$
- Temperature of evaporator: $T_o = 0^\circ C$
- Pressure drop in evaporator: $\Delta p = 0.4bar$
- Superheating temperature: $\Delta T = 5K$
- Temperature of condenser: $T_c = 45^\circ C$
- Pressure drop in condenser: $\Delta p = 0.1bar$
- Isentropic efficiency of compressor: $\eta = 0.8$

SIMULATION RESULT AND DISCUSSION

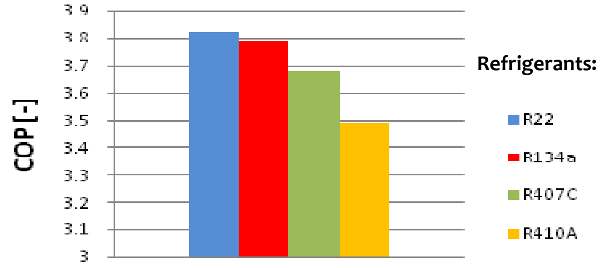

Figure 2. The coefficient of performance of the refrigerator

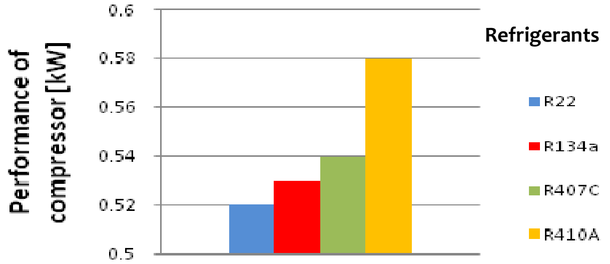

Figure 3. Performance of the compressor

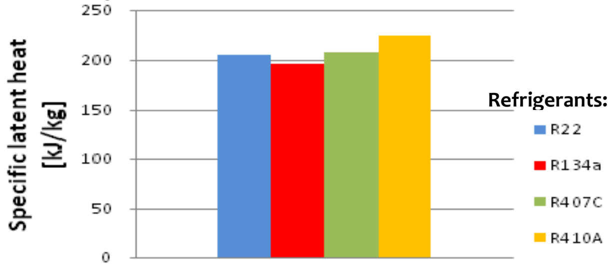

Figure 4. Specific latent heat of the refrigerants

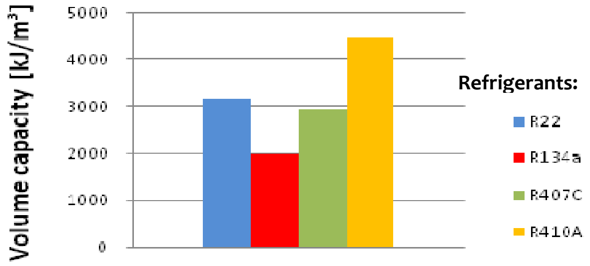

Figure 5. Volume capacity of the refrigerants

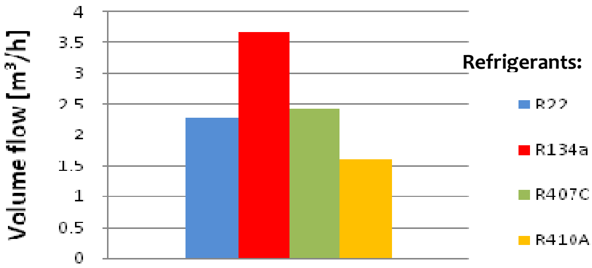

Figure 6.Volume flow of the refrigerants

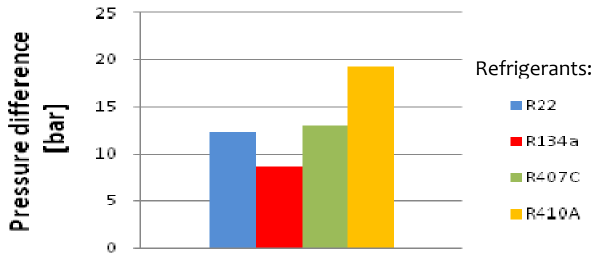

Figure 8. Pressure difference

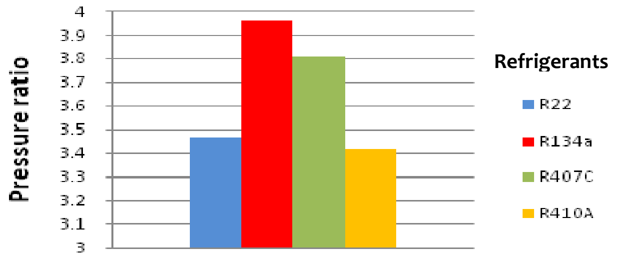

Figure 7. Pressure ratio of evaporator & condenser pressure

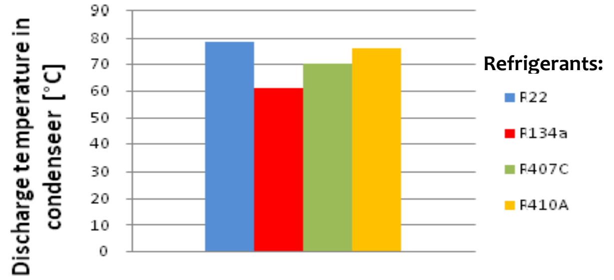

Figure 9. Discharge temperature

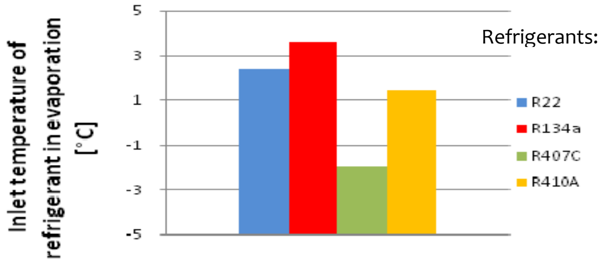

Figure 11. Inlet temperature of refrigerant in evaporator

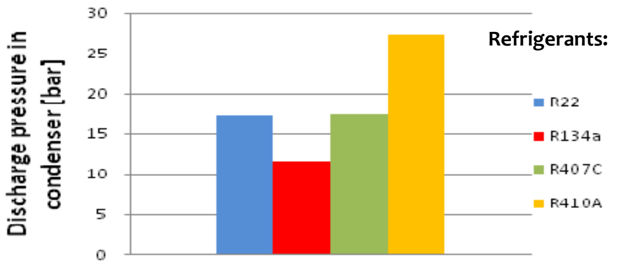

Figure 10. Discharge pressure

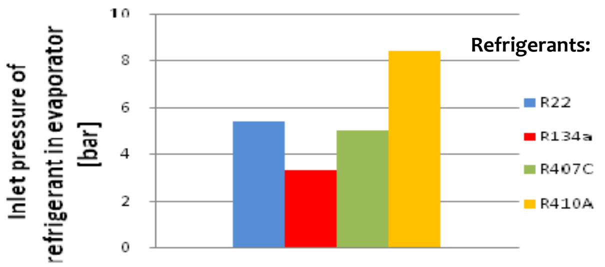

Figure 12. Inlet pressure of refrigerant in evaporator

CONCLUSIONS

- *The performance coefficient of the vapor refrigerant system of the refrigerant R22 is the highest of the investigated refrigerants since this one possesses the smallest refrigerant compressor power requirement and its latent heat is high. However, the applicability of the R22 refrigerant is of finite time, since it contains Cl its release is ceased until 2010, so it can be stated that R134a refrigerant dispose the best properties in terms of efficiency.*
- *Too high pressure is not favorable in the operation process of the equipment.The high partial pressure from the aspects of strength is unfavorable, it requires wall thickness contributing in costs increase.*
- *The pressure ratio in the condenser and the evaporator resulting positively of the piston compressors, while of the centrifugal compressors the small pressure difference between partial pressure is positive. The increase of pressure ratio in the piston compressor reduces volumetric efficiency of the compressor.*
- *When using reciprocating compressors it is advantageous if the volumetric capacity is high because that way the transportable volumetric flow and thus the machine sizes decrease. In the processes of turbo-compressors the high volumetric flow, low volumetric capacity is especially favorable.*
- *In the heat exchanger, the heat transfer is favorable, if the thermal conductivity coefficient is high,while the vapor,the liquid viscosity and surface tension of refrigerant is low.*
- *The refrigerant that meets all the requirements fully is non-existent. In each case the conditions and requirements must be examined in order to choose the most suitable and favorable refrigerant.*

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