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SIMULATION OF THE CONDENSATION PROCESS WITH LIQUID RECIRCULATION IN COMPRESSION REFRIGERATION SYSTEMS

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ABSTRACT

The recirculation of liquid from the liquid reservoir to the condenser inlet is being proposed as a new method to reduce and control the condensing pressure in compression refrigeration systems. In this work the condensation process of different refrigerants with liquid recirculation in an air cooled condenser has been analyzed. The analysis is based on a mathematical model implemented in a computer program. The influence of the liquid recirculation ratio on the condensing pressure (condensing temperature) has been studied.

Results of the operating conditions established in the condenser for different liquid recirculation ratios, such as, heat transfer coefficients, heat transfer rate, pressure drop, etc are shown and discussed in the paper.

INTRODUCTION

Finned-tube heat exchangers are widely used as condenser in compression refrigeration systems. These heat exchangers require little maintenance and offer an economic, effective and reliable operation. Despite of these type of condensers have been operated for many years, at present, they are continuously studied and improved by experimentation, simulation, or combining both methods. These research works are related to many aspects such as heat transfer performance and pressure drop characteristics, different tube arrangement to improve the heat transfer, or even economic analysis to find an optimal size-price and weight relation for investment.

Lozza and Merlo [1] carried out an extensive investigation about the performance of various fin configurations aimed to enhance the heat transfer capabilities of air-cooled condensers. They concluded that when comparing fin configurations, the situation cannot be fully depicted by general criteria, not keeping into account the actual interaction between the heat exchanger and the fan driving the air flow. Wang et al. [2] investigated the effects of circuits on the performance of a wavy-finned condenser when R22 was used as the working fluid. Some conclusions were obtained, such as that the counter-cross arrangement would give a better performance when compared to the in-line arrangement though some modifications were needed.

NOMENCLATURE

a	[m]	Element fin width
A	[m^2]	Area
b	[m]	Element fin height
Cp	[J/kg K]	Specific heat
h	[J/kg]	Specific enthalpy
k	[W/m K]	Thermal conductivity
L	[m]	Element length
m	[kg/s]	Mass flow rate
Q	[W]	Heat transfer rate
$R_{\%}$	[%]	Percentage of liquid recirculation
r	[m]	Radius
t	[m]	Thickness
U	[W/m^{2}K]	Overall heat transfer coefficient
ΔT	[ºC,K]	Temperature difference

Special characters

α	[W/m^{2}K]	Heat transfer coefficient
θ	[K, ºC]	Temperature
η		Efficiency

Subscripts

A	Air
F	Fin
in	Inlet,inside
iw	Inner wall
out	Outlet,outside
ow	Outer wall
R	Refrigerant
T	Tube
s	Surface
o	Overall, referred to the outer finned tube surface

Lee et al. [3] performed an experimental study of a fin and tube condenser using two different configurations of condenser paths (U and Z type) and two refrigerants (R22 and R407C). They found that the difference in performance between refrigerant was evident in the Z-type path experiments, where R22 performed better than R407C, whereas the ternary mixture R407C performed similarly to R22 in the U-type path condenser.

To carry out different analysis to improve the behaviour of the air-cooled condensers with various refrigerants, the use of simulation models represents a cheaper and faster alternative when compared to experimental analysis. Many developed models have been presented in the technical literature. These models generally can be classified into two broad categories. The first category is the zone-based models, in which the heat exchanger is analysed divided into several parts, depending basically, on the number of phases that the refrigerant shows during the heat transfer process. Ge and Cropper [4] developed a simulation model for air-cooled finned-tube condensers using a four-section lumped modelling method. Cuevas et al. [5] developed a condenser three zones model. The second category includes detailed models based on a local analysis, in which the heat exchanger is divided into segments or multiple control volumes, with the outlet of one control volume being the inlet to an adjacent control volume, following the refrigerant path. Examples of this type of model were developed by Bensafi et al. [6] and Liang et al. [7].

In this research, the condensation process of different refrigerants with liquid recirculation in an air-cooled condenser has been analysed. The analysis was based on a detailed mathematical model implemented in a computer program. The geometry of the condenser is known and the heat transfer coefficients are calculated by using correlations from the open literature. The influence of the liquid recirculation ratio on the condenser operating parameters was analysed. In the paper, the main results are presented and discussed.

SYSTEM DESCRIPTION AND MATHEMATICAL MODEL

The air-cooled condenser considered in this research operates in a compression refrigeration systems. A schematic diagram of the liquid recirculation system integrated in a compression refrigeration plant is shown in Figure 1.

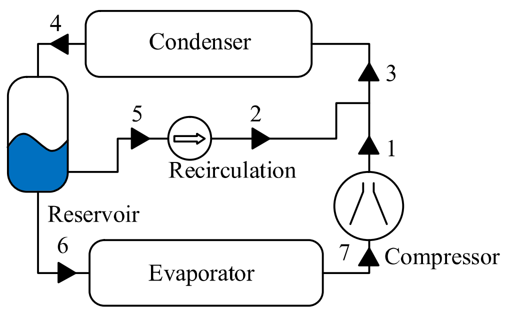

Figure 1 Schematic diagram of the compression refrigeration system with liquid recirculation to the condenser.

The liquid recirculation system to the condenser consists of pumping liquid refrigerant from the liquid recipient to the condenser inlet. The liquid mixes with the vapour from the compressor in the discharge line before going into the condenser. The mass flow rate of liquid recirculated is controlled by a pump.

The air-cooled condenser analyzed consists of multi-row tubes with fins pressed over the tubes. Due to symmetry, the condenser was modelled following a distributed method. In the model, the condenser is considered as one tube divided into various small control volumes, from the refrigerant inlet to the refrigerant outlet. Each control volume is composed by a tube with an equal length to the distance between two fins and the corresponding portion of the fin, as show in Figure 2. The control volumes are connected in the refrigerant flow direction.

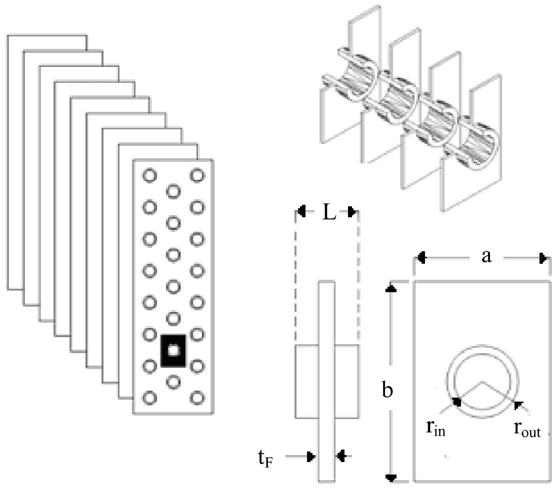

Figure 2 Tube-fin-element for a tube-fin condenser (left), four element arrangement (right top) and tube fin control volume geometry (right bottom)

The model equations were formulated from the mass and energy balances applied to each one of the systems in each element, in addition to the heat transfer equations. The control volume considered in the analysis is shown in Figure 3.

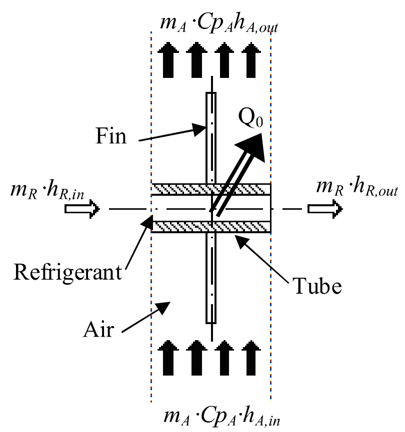

Figure 3 Representative domains used for modelling

In formulating the model, the following assumptions were considered:

(1) The refrigerant liquid and vapour phase are in thermodynamic equilibrium.
(2) The physical properties of the refrigerant and tube wall are considered uniform in the heat exchanger transversal section.
(3) The axial heat conduction in the tubes and the variations of the refrigerant potential and kinetic energy are neglected.

The percentage of recirculated liquid is defined in equation 1, according to the nomenclature used in Figure 1.

$$R_{\%} = \frac{m_2}{m_1} \cdot 100 \tag{1}$$

The recirculated liquid is assumed as saturated at compressor discharge pressure. The enthalpy of the mixture at state 3, according Figure 2, is obtained from equation 2.

$$h_3 = \frac{(m_1 \cdot h_1) + (m_2 \cdot h_2)}{m_1 + m_2} \tag{2}$$

In the condenser, the total heat flux transferred from the refrigerant to the air is given by equation 3, according to the control volume shown in Figure 3.

$$Q_0 = U_0 \cdot A_0 \cdot \Delta T_{LM} \cdot F(R,P) \tag{3}$$

where, U_0 is the overall heat transfer coefficient referred to the outer finned tube surface, A_0 is the element total outer surface area, ΔT_{LM} is the logarithmic mean temperature difference defined in equation 4, and $F(R,P)$ is the correction factor for the logarithmic mean temperature difference which in this case is considered equal to 1, as corresponds to a cross flow with the hot fluid at constant temperature.

$$\Delta T_{LM} = \frac{(\theta_R - \theta_{A,in}) - (\theta_R - \theta_{A,out})}{Ln\left[\frac{\theta_R - \theta_{A,in}}{\theta_R - \theta_{A,out}}\right]} \tag{4}$$

The overall heat transfer coefficient is obtained according to equation 5.

$$\frac{1}{U_0} = \frac{A_0}{2 \cdot \pi \cdot r_{in} \cdot L \cdot \alpha_{R,T}} + \frac{A_0 \cdot Ln\left[\frac{r_{out}}{r_{in}}\right]}{2 \cdot \pi \cdot L \cdot k_T} + \frac{1}{A_0 \cdot \alpha_{T,A} \cdot \eta} \tag{5}$$

The overall outer surface efficiency (η) *is* given by equation 6.

$$\eta = 1 - \frac{A_F}{A_0} \cdot (1 - \eta_F) \tag{6}$$

In eq. 6, η_F is the fin efficiency obtained from the empirical solution given by Schmidt [8]. The overall outer surface area in of each element and the element fin area are calculated from equations 7 and 8, respectively, according to the Figure 2.

$$A_0 = 2 \cdot \pi \cdot r_{out} \cdot (L - t_F) + 2 \cdot A_F \tag{7}$$

$$A_F = (a \cdot b) - (\pi \cdot r_{out}^2) \tag{8}$$

The mass and energy balances, and the heat transfer equation at the refrigerant side, provide equations 9 and 10, respectively.

$$Q_0 = m_R \cdot (h_{R,in} - h_{R,out}) \tag{9}$$

$$Q_0 = [U \cdot A]_{R,T} \cdot (\theta_R - \theta_{T,iw}) \tag{10}$$

The energy conservation applied to the air and the heat transfer to the air from the outer heat exchanger surface yield equations 11 and 12, respectively.

$$Q_0 = m_A \cdot Cp_A \cdot (\theta_{A,out} - \theta_{A,in}) \tag{11}$$

$$Q_0 = [U \cdot A]_{T,A} \cdot \frac{(\theta_{T,ow} - \theta_{A,in}) - (\theta_{T,ow} - \theta_{A,out})}{Ln\left[\frac{\theta_{T,ow} - \theta_{A,in}}{\theta_{T,ow} - \theta_{A,out}}\right]} \tag{12}$$

The correlation of Gnielinski [9] is used to calculate the convective single-phase (vapour or liquid) heat-transfer coefficients in the tube. The condensation heat-transfer coefficient is obtained using the correlations proposed by Traviss et al. [10]. The heat transfer coefficient between the outer finned surface and the cooling air is obtained from the Colburn j-factor evaluated from McQuiston and Parker [11]. The void fraction in the two phase flow is computed by the Martinelli correlation [12]. The pressure drop is calculated using the correlations proposed by Friedl [13]. The thermodynamic properties of the fluids (refrigerant and air) are obtained from the Refprop Database [14].

MODEL IMPLEMENTATION

A finite difference approach is used to solve the model equations. The system of discretized equations is solved in space, element by element, beginning with the control volume where the vapour or the vapour-liquid mixture enters the evaporator, and continues by following the flow direction. An explicit integration scheme is used to establish estimates for nodal temperatures and heat flows.

The inlet temperature, pressure and mass flow of the refrigerant, the initial tube and fin properties, the evaporator geometry, the air inlet temperature and velocity are used as inputs. The calculation procedure is summarised below.

(1) Determine refrigerant properties at the condenser inlet from equation 2.
(2) Guess: $\theta_{A,out}$, $\theta_{T,iw}$ and $\theta_{T,ow}$.
(3) Calculate Q_0 from equation 3.
(4) Calculate $\theta_{T,ow}$ from equation 12.

(5) Check the $\theta_{T,ow}$. If verified, continue, otherwise guess a new value and go to step 2.
(6) Calculate $\theta_{T,iw}$ from equation 10.
(7) Check the $\theta_{T,iw}$. If verified, continue, otherwise guess a new value and go to step 2.
(8) Calculate $\theta_{A,out}$ from equation 11.
(9) Check the $\theta_{A,out}$. If verified, the calculation procedure ends, otherwise guess a new value and go to step 2.

The mathematical model was implemented in a computer program using Visual Basic.

RESULTS

The simulation model was used to perform a parametric analysis to evaluate the influence of the liquid recirculation ratio on the thermal performance of the air-cooled condenser. The liquid recirculation ratio was varied from 0% to 80%. The data considered in the analysis are specified in Table 1. Four refrigerants were considered in the analysis: R134a, R404A, R507A and NH_3.

Figure 4 shows the inner heat transfer coefficient evolution for refrigerant R134a as a function of the condenser length for liquid recircultion ratios from 0% to 80%. When the condenser is operating without recirculation (R% = 0), it can be identified three zones as a function of the refrigerant phase in the condenser: the vapour zone, liquid-vapor zone (two-phases zone) and the liquid zone.

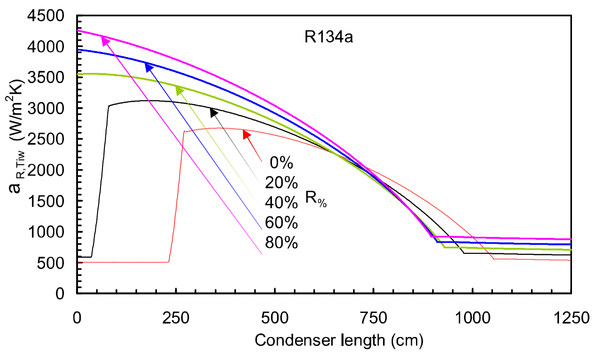

Figure 4 Inner convection heat transfer coefficient with R134a, $\alpha_{R,Tiw}$, as a function of the condenser length for different percentage of recirculated liquid

The first zone corresponds to the vapour zone, where the refrigerant is de-superheated. In this zone, the inner convection coefficient ($\alpha_{R,Tiw}$) is low and remains nearly constant. In the second zone, the two-phases zone, the inner convection heat transfer coefficient ($\alpha_{R,Tiw}$) increases up to a maximum value, obtained with a vapour quality of around 0.9, and then decreases until reaches a minimum when all the refrigerant is condensed. In the third zone, the liquid zone, the refrigerant is sub-cooled and the inner convection coefficient ($\alpha_{R,Tiw}$) shows a slight decreasing trend.

If liquid from the reservoir is recirculated to the condenser inlet, the portion of the surface in the condenser used to de-superheating (vapour zone) is reduced, while the two-phase zone increases.

Figure 5 shows the evolution of the vapour quality for the refrigerant R-134a as function of the condenser length for different liquid recirculation ratios.

Table 1. Technical data and operating conditions of the condenser considered in the analysis.

Parameter	Value
Condenser operating conditions	
Condensing temperature	50 ºC
Compressor discharge temperature	80 ºC
Condenser sub-cooling degree ($R_{\%} = 0$)	8 ºC
Cooling air inlet temperature	25 ºC
Cooling air inlet velocity	10 m/s
Condenser technical data	
Coil metal	Copper
Fin metal	Aluminum
Fin pitch	1 fin/cm
Tube outside diameter	16 mm
Tube wall thickness	0.5 mm
Tube total length	12.5 m

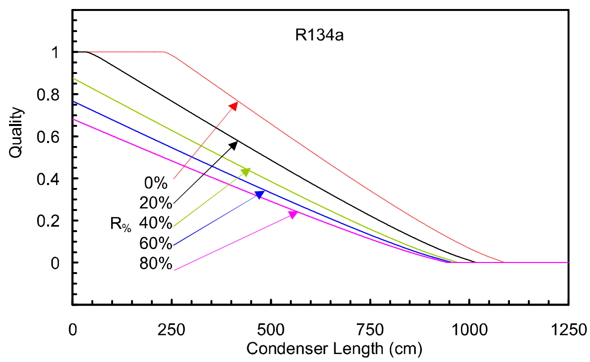

Figure 5 Evolution of the R-134a vapour quality as a function of the condenser length, for liquid recirculation ratios of 0, 20, 40, 60 and 80%.

Results in Figure 5 show that, for liquid recirculation ratios higher than 20%, the refrigerant is a vapor-liquid mixture at the condenser inlet, and the de-superheating zone does not exist. In this case, it can be seen in Figure 5 that, the higher the ratio of the recirculated liquid, the lower the refrigerant quality at the condenser inlet.

The recirculation of liquid to the condenser produces two opposite effects which influence the condenser thermal performance. The reduction of the vapor zone significantly improves the inner convection heat transfer coefficient. As a result, the total heat flux transferred from the refrigerant to the air tends to rise. On the other hand, the refrigerant temperature at the condenser inlet is reduced and the liquid recirculation increases the refrigerant mass flow rate to be condensed. Consequently, at a first glance, the liquid recirculation could either improve or deteriorate the condenser heat transfer performance. Therefore, the aim of this preliminary work is to investigate the effects of the liquid recirculation ratio on the condenser thermal performance. The condenser thermal performance is evaluated based on the refrigerant sub-cooling degree at the condenser outlet.

Taking into account the reasons cited above, Figure 6 shows the sub-cooling degree at the condenser outlet as a function of the liquid recirculation ratio for refrigerants R134a, R404A, R507A and NH_3. Results in Figure 6 allow the comparison of the sub-cooling degree obtained for different liquid recirculation ratios with the sub-cooling degree obtained without recirculation (0% recirculation ratio).

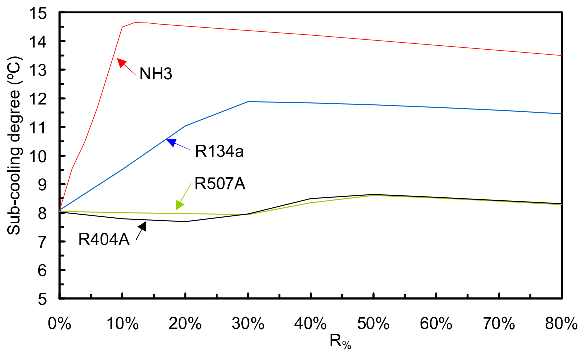

Figure 6 Sub-cooling degree at the condenser outlet as a function of the liquid recirculation ratio, for refrigerants R134a, R404A, R507A and NH_3.

Results in Figure 6 point out that, for each refrigerant, there is an optimum liquid recirculation ratio which provides the highest sub-cooling degree at the condenser outlet. The optimum values are around: 12% for NH_3, 30% for R134a, 50% for R404A and 50% for R507A.

On the other hand, the results in Figure 6 clearly show a quite different thermal performance of the condenser depending on the refrigerant used. NH_3 and R-134a show the same trend. For these refrigerants, the sub-cooling degree increases with increasing the liquid recirculation ratio up to a maximum value. Once the maximum value of the sub-cooling degree is attained, it slightly decreases with increasing the liquid recirculation ratio. NH_3 shows the highest increase of the sub-cooling degree at the condenser outlet, from 8 to 14.6 ºC, followed by R134a from 8 to 11.9 ºC. On the other hand, the refrigerants R404A and R507A show a similar trend between them but different from NH_3 and R134a. For these refrigerants, and despite of the liquid recirculation ratio, the sub-cooling degree at the condenser outlet (and the condenser thermal performance) remains nearly constant. Consequently, the results shown in Figure 6 for R404A and R507A realize that the use of a liquid recirculation ratio does not seem to be a good solution to improve the condenser thermal performance.

As can be seen in Figure 6, NH_3 offers the best conditions to improve the condenser performance by means of a liquid recirculation system. If NH_3 is used, the optimal system operation will be achieved with a low recirculation ratio (around 12%) and the performance improvement will be higher than for the other refrigerants. It can be seen in Figure 6 that, for the optimal recirculation ratio of NH_3, the sub-cooling degree at the condenser outlet increases up to 14.6 ºC.

Taking into account the results discussed above, a new analysis was carried out with the aim of evaluating the influence of the liquid recirculation system on the condensing pressure. The analysis consisted of setting the liquid recirculation ratio equals to the optimal value found previously (Figure 6) and reducing the pressure at the condenser inlet till obtain a sub-cooling degree of 8 ºC at the condenser outlet (i.e. the sub-cooling degree obtained at the design conditions without the liquid recirculation system). The other operating conditions were kept constant an equal to the data specified in Table 1.

The results of this last analysis for NH_3 ($R_{\%}$=12%) show that the condensing pressure decreases from 2034 kPa to 1930.5 kPa, i.e. the pressure could be reduced by 103.5 kPa (around 5.1%). Consequently, the condensing temperature (saturation temperature at the inlet pressure) decreases from 50 ºC to around 48 ºC. In the case of R-134a and with a liquid recirculation ratio equals to 30% (optimal ratio), the condensing pressure decreases from 1317.9 kPa to 1268.9 kPa, i.e. 49 kPa. Then, the condensing temperature could also be diminished from 50 ºC to 48.5 ºC.

These results are valuable since the condensing pressure reduction could have a significant effect on the compressor efficiency and the overall refrigeration system performance. However, further investigations should be developed to validate these simulation results and to evaluate the effects of including the liquid recirculation system on the overall system performance. Nowadays, an experimental program using a single stage compression refrigeration system with NH_3 as refrigerant is being conducted at our laboratory. We hope we will be able to validate the theoretical results in a near future.

CONCLUSION

The recirculation of liquid from the liquid reservoir to the condenser inlet is being proposed as an effective method to reduce the condensing pressure in compression refrigeration systems. However, the recirculation of liquid to the condenser produces two opposite effects which influence the condenser thermal performance. Consequently, the liquid recirculation could either improve or deteriorate the condenser heat transfer performance. In this work the condensation process of R134a, R404A, R507A and NH_3 with liquid recirculation in an air cooled condenser has been analyzed. The analysis is based on a mathematical model implemented in a computer program.

The results show that for NH_3 and R-134a there is an optimum liquid recirculation ratio which provides the highest sub-cooling degree at the condenser outlet. NH_3 shows the highest increase of the sub-cooling degree at the condenser outlet, from 8 to 14.6 ºC, followed by R134a, from 8 to 11.9 ºC. However, for refrigerants R404A and R507A, and despite of the liquid recirculation ratio, the sub-cooling degree at the condenser outlet (and the condenser thermal performance) remains nearly constant. Consequently, the results indicate that a liquid recirculation system does not seem to be a good solution to improve the condenser thermal performance with these refrigerants.

The analysis conducted setting the liquid recirculation ratio equals to the optimal value and reducing the pressure at the condenser inlet till obtain a sub-cooling degree of 8 ºC revealed that the condensing pressure decreases 103.5 kPa (around 5.1%) and 49 kPa (around 3.1%) for NH3 and R-134a, respectively.

It is worth pointing out that the simulation results are valuable due to the condensing pressure reduction could have a significant effect on the compressor efficiency and the overall refrigeration system performance. However, further experimental investigations should be developed to validate these simulation results and to evaluate the effects of including the liquid recirculation system on the overall system performance.

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