Numerical simulation of fin and tube condenser in a R22 system charged with R407C

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Present study simulates condensation heat transfer and pressure drop characteristics of R407C, when it is charged in an air-cooled fin and tube condenser designed for R22 utilizing the local method of analysis. In this analysis, 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, and so on. The properties of refrigerants viz., liquid as well as vapour phase density, viscosity, specific heat, thermal conductivity and enthalpy are calculated using subroutines from National Institute of Standards and Technology software package REFPROP. Refrigerant side as well as airside heat transfer and pressure drop are computed using reported correlations.

Keywords: R22, R407C, Fin and tube condenser, Heat transfer, Local method of analysis, Numerical simulation

Introduction

Refrigeration and air-conditioning systems had been using chlorofluorocarbons (CFC’s) as refrigerants, which were phased out due to high ozone depletion potential (ODP). Now, hydrochlorofluorocarbons (HCFCs), such as R-22, are predominantly used. The only short-term drop-in substitute for R22 is R407C because its characteristics are sufficiently similar to those of R22. Air-cooled finned-tube condensers are widely used in refrigeration and air-conditioning applications. Martin Costa & Parise1 developed a mathematical model for performance prediction of air-cooled condensers. Chitti & Anand2 proposed a simple calculation procedure for predicting local heat transfer coefficients during two-phase forced convective condensation for annular flow configuration. Bensafi et al3 developed a computational model for the detailed design of finned coils. Judge & Rademacher4 developed a heat exchanger simulation for transient and steady state cycle simulations of mixtures and pure components. Chung et al5 conducted a test using HFC407C as a drop in substitute for HCFC22 in a split air conditioner. Yan & Lin6 investigated characteristics of condensation heat transfer and pressure drop for refrigerant R134a flowing in a horizontal small circular pipe (diam, 19 mm). Boissieux et al7 presented local heat transfer results obtained during condensation of Isceon 59, R407C and R404A in a smooth horizontal tube. Cavallini et al8 reported experimental heat transfer coefficients and pressure drops measured during condensation inside a smooth tube when operating with pure HFC refrigerants (R134a, R125, R236ea, R32) and nearly azeotropic HFC refrigerant blend R410A. Lee et al9 performed an experimental study of a fin and tube condenser using U and Z type configurations of condenser paths and R-22 and R-407C as working fluids. Aprea et al10 measured quasi local heat transfer coefficients of R22 and R407C in coaxial counter flow condenser of a refrigerating vapour compression plant. Jung et al11 measured flow condensation heat transfer coefficients (HTCs) of R22, R134a, R407C, and R410A inside horizontal plain and micro fin tubes (9.52 mm, outside diam; 1 m, length) at condensation temperature of 40°C with mass fluxes of 100, 200, and 300 kgm⁻²s⁻¹ and a heat flux of 7.7–7.9 kWm⁻². Ge & Cropper12 built a simulation model for air-cooled finned-tube condensers utilizing lumped method.
This study presents development of a numerical simulation model for air-cooled fin and tube condenser in a R22 system charged with R407C, using local analysis based on the work of Lee et al\(^9\). Heat exchanger model has been divided into three distinct zones (de-superheater, condenser and sub-cooler) based on the works of Martin Costa & Parise\(^1\).

**Model Simulation using Local Method**

Condenser heat exchanger configuration is cross flow, fin and tube type. Refrigerant flows through tubes, and a fan forces air between fins and over tubes. Refrigerant enters condenser as a superheated vapor and exits as a sub-cooled liquid. Condenser is separated into three sections: i) De-superheating section; ii) Condensing section; and iii) Sub-cooling section. In superheated and sub-cooled sections, fluid is in a single phase, while in saturated section, two-phase flow correlations are needed. For refrigerant side, mass conservation is automatically satisfied due to steady flow assumption. Moment equation can be represented by calculation of pressure drop in each section. Energy equation can be expressed as

\[
\dot{Q} = m_r (h_{in} - h_{out})
\]

...(1)

These conservation equations are also suitable for airside. Since air-facing velocity is kept unchanged across pipes, mass equation is also automatically satisfied. Pressure drop calculation is used instead of moment equation for airside. There is a heat balance between air and refrigerant side for each section and NTU-e method is utilized to carry out part of the calculation of heat transfer.

\[
\dot{Q} = m_a C_{p,a} [T_{in} - T_{out}] = e(G_c)_{min} (T_{hin} - T_{cin})
\]

...(2)

When cross-flow is assumed in heat exchanger, effectiveness \(e\) can be calculated for single-phase region [Eq. (3)] and for two-phase region [Eq. (4)] as...
\[ \varepsilon = \left(1 - \exp\left(\frac{NTU^{0.22}}{(G_e)_{\min}/(G_e)_{\max}}\right)\right) \left(\exp\left(-\frac{(G_e)_{\min}}{(G_e)_{\min}} \times NTU^{0.78}\right) - 1\right) \]  

... (3)
Start

Read input parameters mr, ma, Ts, Tc

Read first section

Calculate refrigerant vapour and air properties viz., specific heat, density, viscosity, thermal conductivity

Calculate ha, h, fin surface efficiency, pressure drop, NTU, Q for de-superheat

Calculate section outlet pressure and temperature

If section outlet temp. = sat. temp

Yes

De-superheat region solved with 'M' sections

No

Read next section

Saturation region solved with 'N' sections

Calculate liquid and vapour phase refrigerant properties

Calculate pressure drop, h, NTU, Q for saturation

Calculate section outlet pressure and vapour quality

If vapour quality = 0

No

Read next section

Yes

Calculate liquid refrigerant properties, pressure drop, h, NTU, Q for sub-cooling

Calculate section outlet pressure and temperature

Sub-cooling region solved with 'P' sections

Stop

Fig. 2 — Flow chart of simulation model for local method of analysis
the last section. From this local analysis, condenser outlet pressure and temperature and air outlet temperature shall be determined. Also, heat transfer rate, area and pressure drop for all the three regions shall be determined.

Simulation for fin and tube R22 condenser charged with R407C was written as a program in MATLAB. Performance of heat exchanger was analyzed along condenser length and by varying input parameters (refrigerant mass flow rate, condenser inlet temperature, condensation temperature and ambient temperature). Air mass flow and airside heat transfer coefficient have little impact on performance of condenser. Following input data was used for analysis of condenser performance: refrigerant mass flow rate, 0.035 kg/s; condenser inlet temperature, 70°C; condensation temperature, 55°C; and ambient temperature, 35°C.

Results and Discussion

Performance Analysis along Condenser Length

In de-superheat region, temperature drop is steep due to sensible heat rejection to air (Fig. 3). Temperature saturation is reached at the condenser relative length of 0.09 for R22 and 0.1 for R407C (15°C drop over 1.3 m length). In saturation region, phase change occurs and latent heat is rejected to the air. However, temperature drops a little for R22 by 0.34°C over 9 m length due to frictional pressure drop in two phase region. For R407C, temperature drop occurs in saturation region due to temperature glide associated with a zeotropic mixture as well as due to frictional pressure drop. A temperature drop of 5°C occurs over a length of 10.6 m. In condensation of a zeotropic mixture, pressure drop augments temperature glide. In sub-cooling region, temperature drop is again steeper due to sensible heat removal. Sub-cooling is 10.8°C over a length of 3.1 m for R22 and 4.5°C over a length of 1.5 m for R407C.

Local heat transfer coefficient increases slightly in de-superheat region because vapour specific heat increases and vapour viscosity decreases due to temperature and pressure drops (Fig. 4a). Also, specific heat of R407C vapour is higher than that of R22 vapour. So, heat transfer coefficient of R407C is higher than that of R22. When phase change starts, local heat transfer coefficient steeps into a very high value, because combined effect of latent heat and buoyancy induces large heat transfer rates, which lead to a higher value of heat transfer coefficient in the two phase region. Buoyancy is induced by surface tension between liquid-vapour interface and density difference between the two phases. However, as the quality of refrigerant vapour decreases along condenser length due to condensation, local heat transfer coefficient decreases, due to combined effect of latent heat and buoyancy that are reduced due to heat rejection as well as decrease in liquid-vapour interface. Heat transfer coefficient in the two phase region is also higher for R407C compared to R22, due to higher Reynolds and Prandtl numbers for R407C in liquid as well as vapour phase. When refrigerant reaches saturated liquid state, local heat transfer coefficient reaches minimum and remains almost constant in sub-cooling region, because variation in thermo-physical properties of liquid refrigerant is negligible along condenser length. In this region also, heat transfer coefficient is higher for R407C. Fig. 4b shows variation of local values of overall heat transfer coefficient along condenser length for R22 and R407C.

Cumulative heat rejection rate for R22 is high compared to R407C (Fig. 5a). In de-superheat region, heat rejection rate is higher for R407C than for R22 because specific heat of R407C vapour is higher than that of R22 vapour. However, in saturation region, heat rejection rate is higher for R22 than for R407C, due to

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Table 1 — List of correlations used in this study

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Applicable zone</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat transfer coefficient</td>
<td>Air side(^{13,14})</td>
</tr>
<tr>
<td></td>
<td>Single phase – refrigerant side(^{15})</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>Single phase – refrigerant side(^{12,16})</td>
</tr>
<tr>
<td>Friction factor</td>
<td>Single phase – refrigerant side(^{17})</td>
</tr>
</tbody>
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Fig. 3 — Variation of refrigerant temperature along condensers
temperature glide; the drop in saturation temperature is more for R407C, which in turn reduces heat rejection rate. Due to higher heat rejection rate, R22 occupies less heat transfer area in two phase region compared to R407C, which results in larger sub-cooling heat quantity and area for R22.

Cumulative pressure drop is high for R22 compared to R407C (Fig. 5b). In de-superheat region, pressure drop is more for R407C than for R22, since heat transfer area is more for R407C in de-superheat region due to high heat quantity. However, in two phase region, friction factor for R22 is very high compared to R407C due to high velocity in both liquid and vapour phases. Also, pressure drop is higher in sub-cooling region for R22, since it occupies more area.

**Performance Analysis by Variation of Refrigerant Mass Flow Rate**

Condenser performance is analyzed by varying refrigerant mass flow rate (0.025-0.045 kg/s) and by keeping other parameters constant (condenser inlet temperature, 70°C; condensation temperature, 55°C; and ambient temperature, 35°C). As mass flow increases, heat rejection rate increases (Fig. 6a). Cumulative heat rejection rate for R22 is high compared to R407C. In de-superheat region, heat rejection rate is higher for R407C than for R22 because specific heat of R407C vapour is higher than that of R22 vapour. However, in saturation region, heat rejection rate is higher for R22 than for R407C, due to temperature glide; drop in saturation temperature is more for R407C, which in turn...
reduces heat rejection rate. Due to higher heat rejection rate, R22 occupies less heat transfer area in two phase region compared to R407C, which results in large subcooling heat quantity and area for R22. At higher flow rates, difference in heat rejection between R22 and R407C is more, because at higher flow rates, effect of temperature glide is more for R407C, which results in delayed condensation and reduced heat rejection rate.

The pressure drop increases as the mass flow increases, because refrigerant velocity increases due to increase in mass flow, which in turn leads to increase in pressure drop (Fig. 6b). Cumulative pressure drop is high for R22 compared to R407C. In de-superheat region, pressure drop is more for R407C than R22, due to more heat transfer area for R407C than R22. However, in two phase region, friction factor for R22 is very high compared to R407C due to high velocity in both liquid and vapour phases. Also, pressure drop is higher in subcooling region for R22, since it occupies more area.

Performance Analysis by Variation of Condenser Inlet Temperature

Condenser performance is analyzed by varying condenser inlet temperature (70-90°C) by keeping other parameters constant (refrigerant mass flow rate, 0.035 kg/s; condensation temperature, 55°C; and ambient temperature, 35°C). As condenser inlet temperature increases, heat rejection rate increases due to higher degree of superheat (Fig. 7a). Degree of superheat increases, due to increase in condenser inlet temperature, which in turn increase de-superheat. Since refrigerant mass flow rate and condensation temperature remain constant, two-phase region heat removal from refrigerant is constant. For a given condenser configuration, since de-superheat occupies more area, sub-cooling region becomes small. Cumulative heat rejection rate for R22 is high compared to R407C. Pressure drop almost remains constant as condenser inlet temperature increases (Fig. 7b), because condenser inlet temperature
has little impact on refrigerant velocity, which influences pressure drop. Pressure drop is more for R22 when compared with R407C.

Performance Analysis by Variation of Condensation Temperature

Condenser performance is analyzed by varying condensation temperature (48-55°C) and by keeping other parameters constant (refrigerant mass flow rate, 0.025 kg/s; condenser inlet temperature, 65°C; and ambient temperature, 35°C). Cumulated heat rejection increases with increase in condensation temperature (Fig. 8a). For a given condenser configuration, when de-superheat and two-phase region heat decrease, subcooling heat quantity becomes more, because as condensation temperature increases, degree of superheat decreases, thereby reducing de-superheat in condenser. Also, when condensation temperature increases, latent heat to be removed from refrigerant decreases, thereby reducing two-phase region heat. Since, increase in subcooling heat quantity is more than decrease in heat quantity of de-superheat in two-phase regions, total heat rejection increases with increase in condensation temperature. Heat rejection slope is more at lower temperatures and becomes flat at higher temperatures, because increase in subcooling heat rejection is more than decrease in de-superheat and two phase heat rejection at lower condensation temperatures and is almost equal at higher temperatures. Cumulative heat rejection rate for R22 is high compared to R407C.

Overall pressure drop decreases as condensation temperature increases (Fig. 8b), because as condensation temperature increases, de-superheat and saturation heat transfer area decrease. So pressure drop is also reduced in these two regions. Since, subcooling area becomes...
larger, pressure drop is more in this region. Pressure drop is more for R407C at lower condensation temperatures and less at higher temperatures when compared to R22, because at lower condensation temperatures, de-superheat area is more and at higher condensation temperatures, sub-cooling area is more. In de-superheat region, pressure drop is more for R407C than for R22, because in spite of R22 having a high vapour velocity and friction, heat transfer area is more for R407C in de-superheat region due to high heat quantity. Pressure drop is higher in sub-cooling region for R22 since it occupies more area than R407C.

Performance Analysis by Variation of Ambient Temperature

Condenser performance is analyzed by varying ambient temperature (28-38°C) and by keeping other parameters constant (refrigerant mass flow rate, 0.035 kg/s; condenser inlet temperature, 70°C; and condensation temperature, 55°C). Cumulated heat rejection rate decreases as ambient temperature increases (Fig. 9a). Cumulative de-superheat and saturation heat almost remain constant. However, heat rejection rate becomes less in both regions due to increase in ambient temperature. So, these two heat quantities occupy more area to reach saturated vapour state and saturated liquid rate respectively. This results in reduced heat transfer area for sub-cooling, which becomes less leading to reduction in cumulated heat rejection rate. Heat rejection rate is more for R22 compared to R407C, because cumulative de-superheat is more for R407C; even though two phase region heat is less for R407C. Total pressure drop increases as ambient temperature increases (Fig. 9b). De-superheat and two phase region heat quantity occupy more area due to increase in ambient temperature, which results in increase in pressure drop in these two regions. Total pressure drop is more for R22 when compared with R407C.

Performance Degradation

Performance degradation for R22 condenser charged with R407C increases, as mass flow rate increases (Fig. 10). When R22 condenser is operated, at a condensation temperature of 55°C, complete condensation occurs up to mass flow rate of 0.045 kg/s, whereas for R407C, it occurs up to 0.035 kg/s, because of delayed condensation that occurs due to temperature glide associated with R407C. When condensation temperature is increased by 2°C, complete condensation occurs for R407C up to 0.045 kg/s similar to R22. Performance degradation is also minimized due to increase in condensation temperature.

Conclusions

There is a potential degradation in performance of R22 condenser when it is charged with R407C, due to temperature glide associated with zeotropic mixture R407C. When R22 is used, drop in saturation temperature across two phase region is a maximum of 0.5°C, due to frictional pressure drop, which has little impact on condenser heat rejection capacity. When R407C is used, drop in saturation temperature across two phase region is around 5°C, due to temperature glide associated with mixture as well as frictional pressure drop. This leads to delayed condensation. At higher mass flow rates of R407C, heat rejection capacity is also reduced in addition to incomplete condensation. So, caution must be exercised in selecting the operating parameters, when R22 condenser is charged with R407C. Performance degradation shall be minimized by selecting a slightly higher condensation temperature for R407C (1 or 2 °C higher than that of R22), without changing mass flow rate. Increase in condensation temperature compensates for temperature drop due to glide associated with R407C. Though, increasing condensation temperature may not improve overall performance of refrigeration system, complete condensation shall be ensured.
References