A Physically Based Computer Model for Mobile Air Conditioning Condensers Using Ozone-Safe Refrigerants

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ACRC TR-17 April 1992

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Prepared as part of ACRC Project 03
Condenser Performance
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ABSTRACT

A steady-state computer simulation based on fundamental principles was developed to model air-cooled condensers. It consists of dividing the total condenser length into a few segments which are further divided into several modules, as in the tube-by-tube approach. Air and refrigerant heat transfer coefficients, as well as refrigerant pressure drop, were calculated using the existing correlations. The model provides increased flexibility in terms of flow circuitry and refrigerant type.

An experimental test matrix covering a wide range of conditions was used to validate the simulation. Two typical cross-flow condensers were modeled and the error between experimental and calculated condenser capacities obtained with refrigerant R-134a was within 10%.

INTRODUCTION

As evidence of the environmental damage caused by chlorofluorocarbon refrigerants grows, so does the political pressure calling for their phase-out on a national as well as an international level. Although several alternative refrigerants are being seriously considered, the lack of a definite replacement demands greater flexibility in the design of new systems, a process in which experimental testing has often played a dominant role. For this reason, the ability to accurately model a system or component for an arbitrary set of testing conditions is becoming increasingly important. This paper presents a model that was developed to simulate the heat transfer mechanisms present in a typical air-cooled condenser with refrigerant tubes of circular cross-section. Such a model may be used to optimize the condenser design and to perform a sensitivity analysis on various condenser parameters.

Other condenser models found in the literature may be grouped in the following categories: purely empirical, semi-empirical, semi-theoretical and fundamental principles models [1], depending on their reliance on experimental test data and/or physical laws. The primary objective of the present model was to develop a design tool capable of predicting the performance of R-134a in condensers with a minimum amount of experimental data. For this reason a fundamental principles approach was chosen, requiring the following input variables: refrigerant inlet pressure and temperature, air and refrigerant mass flow rates, air inlet temperature, pressure and relative humidity, and the geometric dimensions of the coil. Table 1, on the next page, compares the present model with some of the other models currently available.

Another objective was to expand the application of fundamental principles models to mobile air-conditioning systems. Current models have been validated for heat pump applications. These include models developed by Domanski & Didion[2] and Oak Ridge National Laboratory (ORNL)[3]. Mobile air-conditioning systems are subjected to more severe environmental conditions than heat pumps. This, along with the differences in size, justifies the need for separate experimental validations.

In the model presented here, a typical condenser is divided into segments, each segment is in turn represented in terms of several modules. Heat transfer and pressure drop relationships are solved for each module in a segment. The number of modules in each segment are determined by the user, considering the trade-off between lower discretization errors and longer run times.

Finally, an effort was made to provide flexibility in terms of circuitry arrangements. A tube-by-tube approach is adopted by many system models to reduce execution time as opposed to detailed finite difference methods. This modular approach represents a further consolidation while retaining circuiting flexibility. Most of the condenser geometries used in industry may be represented by combining several segments. Further modifications to the model are needed to fully automate the selection for an arbitrary geometry.

Table 1. Comparison of the Current Model with other Models from the Literature

Model	Date	Solution Scheme	Inlet Air Adjusted for Downstream Rows	Air Side Heat Transfer Coefficient	Applications	Rate Equation
Domanski & Didion [2]	1983	NR**	no	Briggs & Young [5]	Heat Pumps	LMTD
ORNL [3]	1983	SS	no	McQuiston [6], Yoshii [7], Senshu [8]	Heat Pumps	ε-NTU
SERCLE [4]	1991	NR & SS	no	Overall UA	Single Evaporator Refrigerator	LMTD
Current Model	1991	NR	yes	Colburn j-factor	Mobile AC	ε-NTU

** NR: Newton Raphson , SS: Successive Substitution

ANALYSIS OF A TYPICAL SEGMENT

Each segment may be analyzed independently from the rest of the condenser given the following input parameters: refrigerant inlet pressure and temperature, air and refrigerant mass flow rates, air inlet temperature, pressure and relative humidity, and the geometric dimensions of the coil. A distinction must be made between air and refrigerant inlet properties to a segment and to the full condenser, which are not necessarily the same. Figure 1 shows a typical module within a segment. Although modules of uniform length were assigned by default, the size of each module may be specified by the user.

Momentum and energy balances are applied to the modules used to represent the segment, resulting in a set of residual equations which are solved simultaneously with the Newton-Raphson method. Each module is treated as a separate heat exchanger and the amount of heat transfer and refrigerant pressure change in that module were calculated with the appropriate correlation. The module outlet enthalpies and pressures become the variables in the iterative solution scheme, which proceeds until the set of residual equations balance. The two residuals associated with the i^{th} module have the form,

$$R = h_r(i) - h_r(i-1) - \frac{Q(i)}{m_r} \tag{1}$$

$$R = p_r(i) - p_r(i-1) - \Delta P(i) \tag{2}$$

where,

$h_r(i)$	: Refrigerant enthalpy at the outlet of module 'i'
$p_r(i)$	: Refrigerant pressure at the outlet of module 'i'
$\Delta P(i)$	: Refrigerant pressure drop in module 'i'
m_r	: Refrigerant mass flow rate in module 'i'
$Q(i)$	: Heat transfer rate in module 'i'

A discretization error is introduced with a fixed module length approach due to transition modules, those where the refrigerant changes phase, because heat transfer and pressure drop correlations depend on refrigerant phase. For simplicity, the refrigerant in a given module is assumed to be fully in its exiting phase. The errors associated with this assumption may be reduced by increasing the number of modules used to represent a given segment. Several alternative solution schemes were considered to eliminate the aforementioned problem, but all presented drawbacks that made them unsuitable for the practical design of condensers.

HEAT TRANSFER AND PRESSURE DROP CALCULATIONS

The amount of heat rejected by a module is calculated with the effectiveness-NTU method for a cross-flow heat exchanger. The ε-NTU approach is preferred because other methods can lead to numerical instability if too large a module is selected. An overall heat transfer coefficient is based on the inside and outside heat transfer coefficients, as well as the tube wall thermal resistance. Refrigerant thermodynamic properties are evaluated with subroutines developed by NIST [9].

Several refrigerant heat transfer coefficient correlations from the literature are available to the program. These include the Dittus-Boelter and the Petukhov-Popov [10] correlations for single phase refrigerant, and the Cavallini-Zecchin [11], Traviss et al. [12] and Shah [13] correlations for two-phase refrigerant. The results presented in this paper were obtained with the Dittus-Boelter and the Cavallini-Zecchin correlations for single and two-phase refrigerant respectively.

On the air side, a Colburn j-factor vs. Reynolds number correlation was obtained experimentally for the condenser used to validate the simulation, the j-factor being a non-dimensional form of the heat transfer coefficient ($j=St*Pr^{2/3}$). This format was chosen because it is used in most published air-side data [14]. More generalized air-side heat transfer coefficient correlations may be substituted in the code at a later time. The experimental data were analyzed with the modified Wilson plot technique to separate the air from the refrigerant side resistance to heat transfer.

Changes in pressure due to friction, acceleration and gravity effects are calculated for both single and two-phase refrigerant. The Fanning equation and the Lockhart-Martinelli method [15] are used to evaluate friction pressure drop for single and two-phase refrigerant respectively . Momentum effects are calculated with an expression obtained by integrating the momentum equation from module inlet to outlet. A homogeneous model is used for gravity related pressure changes. Finally, the effect of tube bends on the pressure drop is neglected.

MODELING OF A FULL CONDENSER

The air inlet temperature is an important parameter in the segment-by-segment analysis of a full condenser. It determines, together with the air inlet pressure and its relative humidity, the air inlet enthalpy. An energy balance between the air and the refrigerant sides is used to obtain the enthalpy of the air at the outlet of each module, from which the outlet air temperatures are calculated. Finally, the outlet temperatures are weighted by the area of the module to the total area and summed to determine the average outlet air temperature of the segment.

This information is especially relevant when dealing with a segment located in a downstream row. Ellison et al. [16] assumed air reaching the back rows to be at the same temperature as air leaving the front rows if it all exchanged heat with the front row. Experimental data from our test facility suggests this is not the case. A considerable amount of fresh air bypasses the front row of the condenser which reduces the temperature of the air approaching the second row. Given the lack of a satisfactory analytical description of the problem, an empirical approach was chosen. The following experimentally determined mixing factor was introduced:

$$\phi = \frac{T_{ib} - T_{if}}{T_{of} - T_{if}} \tag{3}$$

T : Average air temperature

<u>Subscripts:</u>

if : Inlet of front row
of : Outlet of front row
ib : Inlet of back row

This non-dimensional factor was assigned a constant value, since no dependence on the condenser geometry or the air mass flow rate could be found from the experimental data. A value of 0.5 gave the best agreement between the calculated and experimental heat capacities for the range of test conditions covered in the experimental validation of the model.

Two different coils, described in figures 2a and 2b, were analyzed with this model. The program was originally developed for the coil in figure 2a. This coil was divided into two segments, one for the front rows and one for the back rows. The outlet temperatures from the front segment were used along with the ϕ factor to determine the inlet temperature for the second segment.

This sequential analysis would not work, however, for the third segment of the coil represented in figure 2b. This third segment could be divided in half but could not be analyzed in the same way because the inlet air temperature in the first half (as the refrigerant flows) was dependent on what happened in the second half. Conversely, the front half of the segment (as the air flows) could not be analyzed before the first half because the inlet refrigerant conditions were dependent on what happened in the first half.

The model was modified to let the inlet temperature to the first half become a variable in the iterative solution. The difference between this variable and the estimated value of this intermediate air temperature provides an additional residual equation. The same ϕ factor approach described above was used to estimate the intermediate air inlet temperature to the first half.

This gives the model added flexibility in modeling the circuitry of heat exchangers. Figures 3a and 3b illustrate the type of segments which can be analyzed. The parallel-cross flow segment, figure 3a, can be divided into two segments and analyzed sequentially since all the inlet conditions to the back half can be determined after analyzing the front half. The counter-cross flow segment needs to be analyzed simultaneously as described in the previous paragraph.

On the refrigerant side, energy balances were applied at segment junctions to evaluate the refrigerant inlet conditions to adjoining segments. Only the cases in which two tubes were merged into one and vice versa were considered. Equal mass flow rates were assumed in parallel condenser tubes.

RESULTS

An experimental test matrix, covering the range of conditions encountered in automotive applications, was used to test the accuracy of the model results. The data were collected with a condenser coil having a fin density of 18 fins per inch, a fin thickness of 0.005 inches and a width of 1.625 inches. The sensitivity of the results to a varying number of modules was examined. For the test conditions considered, increasing the number of modules beyond twelve changed the capacity by less than 1%.

The effect of varying the air mixing factor, ϕ, was also investigated. Figure 4 shows the dependence of the capacity on this factor for a typical set of test conditions. Discretization errors are responsible for the discontinuities in this plot, and correspond to changes in the exiting phase of certain modules. As ϕ increases the amount of heat transferred in second row decreases. This shifts the phase change further down the tube and when the phase change moves to a new module a discrete change in the heat transfer rate occurs.

The experimental capacities measured with the first coil were compared in figure 5 with the simulation results. The same ϕ factor value, 0.5, was used for all data points and both the front and back tube rows were modeled using twelve modules. The experimental results were calculated based on the measured refrigerant properties.

Data obtained with a second coil were also analyzed. The second coil was different with a fin density of only 10 fins per inch, a fin thickness of 0.004 inches and a width of 1.75 inches. A number of approximations were made when analyzing this data with the simulation. First of all, the original j-factor correlation, developed with data from the first coil, was used. In addition, the air mixing factor was also left unchanged. Given these two major assumptions along with the differences in circuitry, the prediction of the condenser capacity shown in figure 6 is remarkable.

CONCLUSIONS

These results demonstrate that a reasonable level of accuracy may be obtained with a fundamental principles model. Generality of the model was supported by successfully modeling a second coil with a significantly different geometry. Many flow circuitry arrangements currently in use can be simulated with the model. Other conclusions which can be made are:

1. The effectiveness-NTU method provides stability in the numerical solution. Very stable behavior was obtained for all the test conditions.

2. It is important to adjust the inlet temperature for downstream rows since this affects the performance of the coil.

3. A tentative conclusion that the Cavallini-Zecchin correlation for the convective heat transfer coefficients on the refrigerant side accurately predicts the performance of R-134a seems reasonable. Additional experimental measurements are needed to verify this conclusion.

4. An advantage of the modular approach is that it provides very detailed information about the distribution of various refrigerant properties along the condenser tubes. Such information may be used to determine the relative importance of the resistance to heat transfer on the air and the refrigerant sides, or the compare the performance of various refrigerant types.

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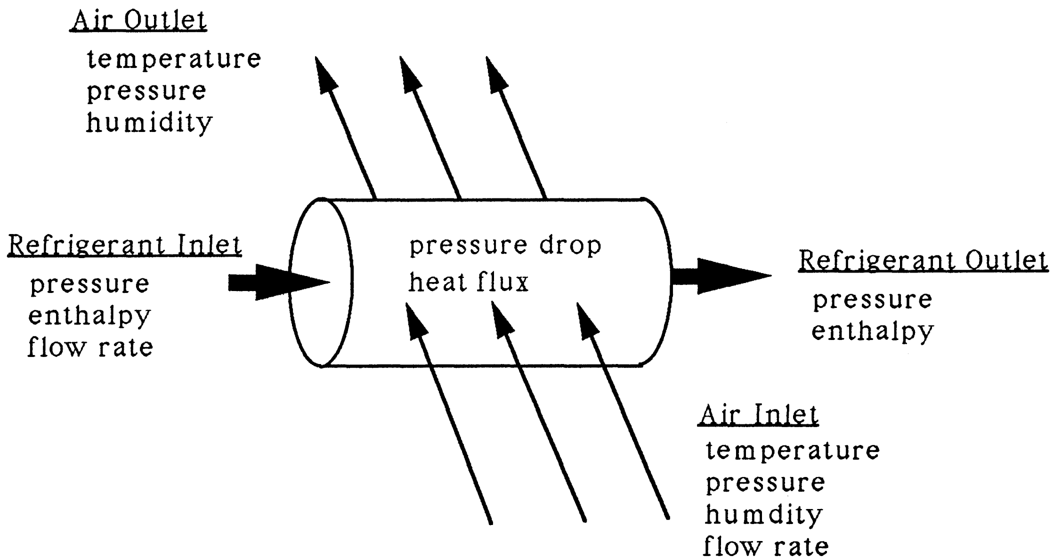

Figure 1. Inlet and Outlet Conditions of a Typical Module

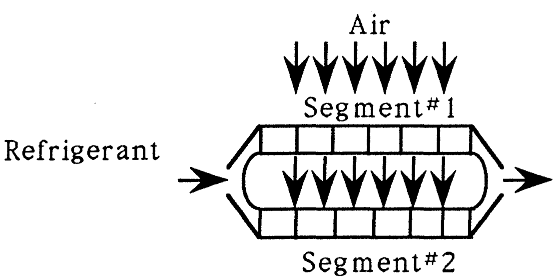

Fig 2.a. Schematic of Condenser #1

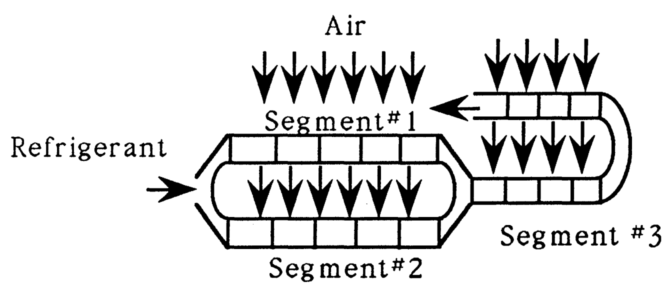

Fig 2.b. Schematic of Condenser #2

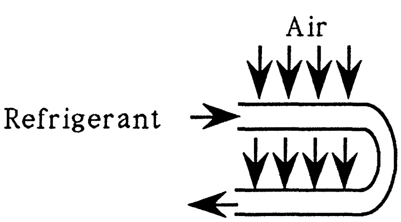

Fig 3.a. Parallel-Cross Flow Section

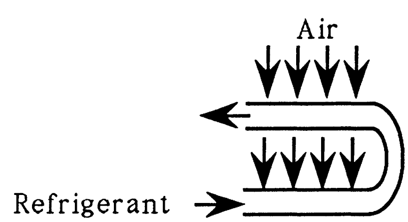

Fig 3.b. Counter-Cross Flow Section

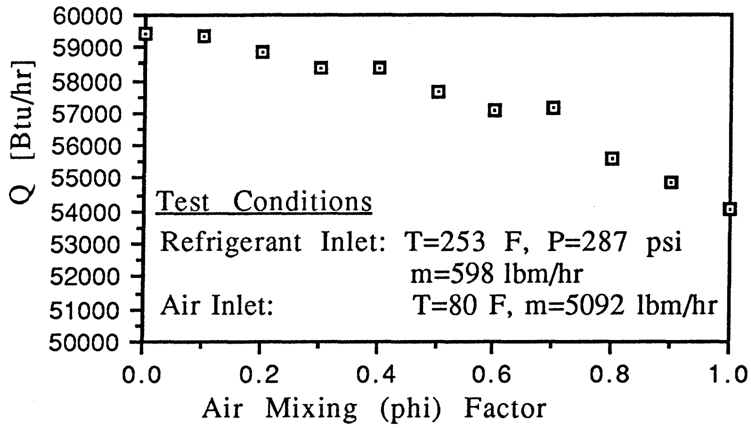

Figure 4.a. Effect of the Air Mixing Factor on the Condenser Capacity

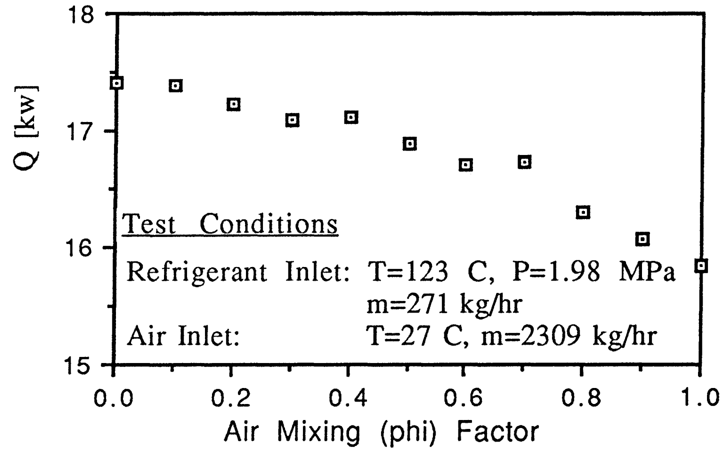

Figure 4.b. Effect of the Air Mixing Factor on the Condenser Capacity

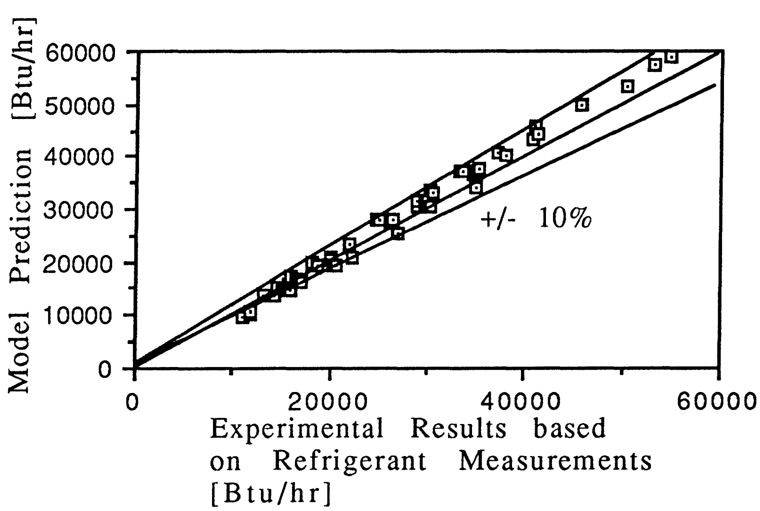

Figure 5.a. Capacity of First Coil

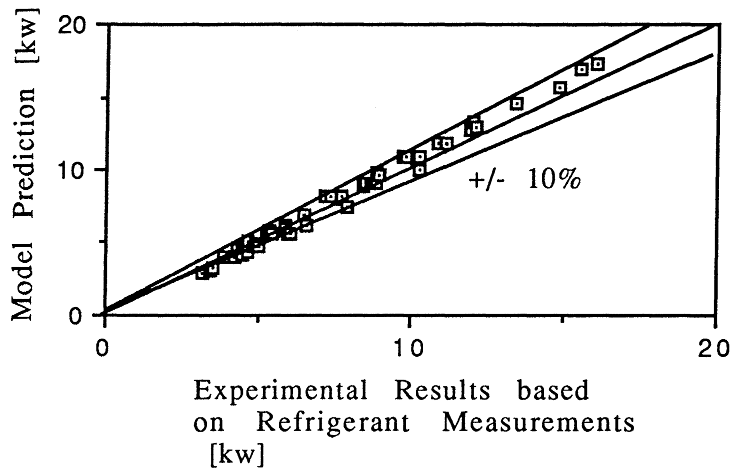

Figure 5.b. Capacity of First Coil

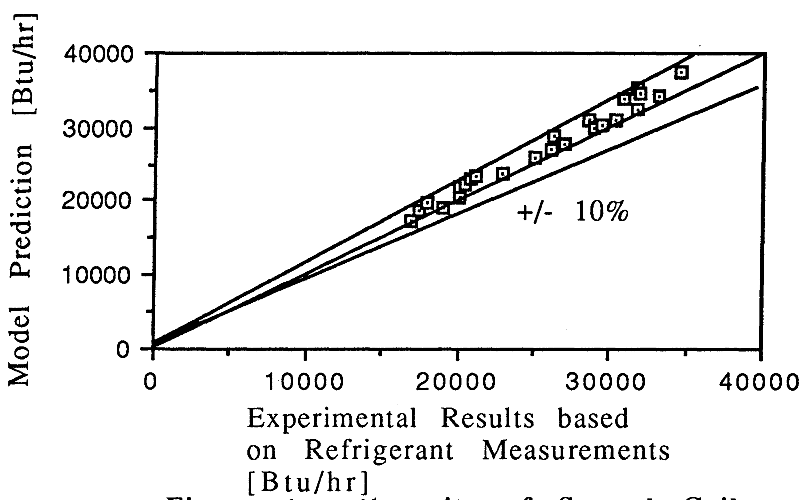

Figure 6.a. Capacity of Second Coil

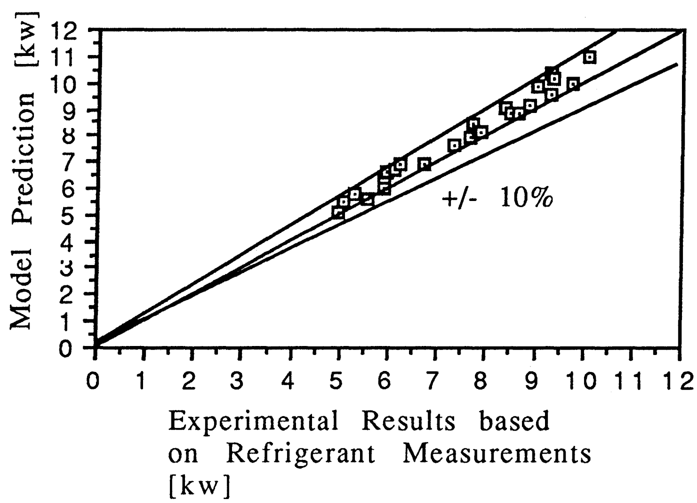

Figure 6.b. Capacity of Second Coil