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MODELING AND EXPERIMENTAL PERFORMANCE EVALUATION OF PARALLEL FLOW MICRO-CHANNEL CONDENSERS

MODELO MATEMATICO Y EVALUACION EXPERIMENTAL DEL DESEMPEÑO DE UN CONDENSADOR DE FLUJO PARALELO CON MICROCANALES

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ABSTRACT

This paper reports results obtained from a theoretical and experimental study of micro channels/louvered fins condensers for automotive applications. A computer model has been developed based upon three zones related to the thermodynamic states of the refrigerant in the condenser. Results from the model for refrigerant HFC-134a are compared to experimental those from an experimental bench developed for that purpose.

RESUMEN

Este trabajo presenta resultados obtenidos de un estudio teórico y experimental de condensadores con micro canales con aletas *louver* para aplicaciones en automóviles. Se desarrolló un modelo matemático basado en tres zonas que caracterizan el estado termodinámico del fluido refrigerante en el condensador y se hizo la simulación computacional. Los resultados del modelo para refrigerante HFC-134a son comparados con los obtenidos de forma experimental en la bancada construida con esta finalidad.

Keywords: Parallel Flow Condenser, Micro Channels, Automotive, Air Conditioning Systems.
Palabras Clave: Condensador de Flujo Paralelo, Micro Canales, Automóviles, Sistemas de Aire Acondicionado.

1. INTRODUCTION

Tight limitations in space and weight of automobile air conditioning condensers, in addition to high thermal performance requirements, have led manufacturers to develop new and ingenious geometries. One of such geometries, which is prevailing in present days, is constituted of blades of extruded aluminum micro channels (typical dimension of the order of 1 mm), with the space between blades being filled with louvered fins. Blades are brazing soldered to vertical headers where the refrigerant is distributed to the micro channels, as clearly shown in Figure 1. One of the main advantages of the micro channels condenser is the distribution of the refrigerant through a higher area, what enhances the heat transfer performance.

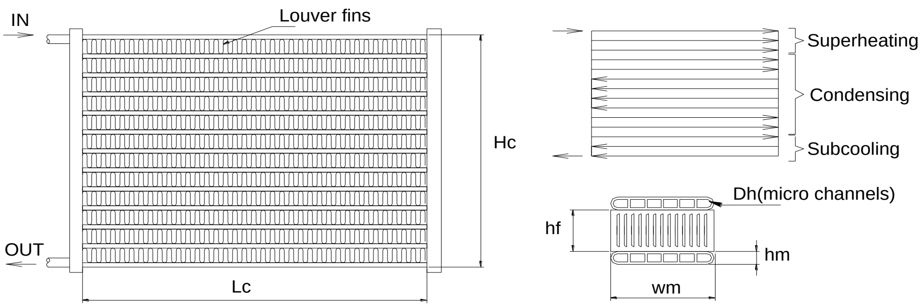

Figure 1 - Schematic representation of parallel flux micro channels condenser.

Despite the intensive research being carried out around the world regarding the heat transfer mechanisms prevailing in these condensers, not many studies have been published in the open literature about their overall performance. One such study has been conducted by Awf [1], who developed a mathematical model for a parallel flow micro channels condenser as part of a comprehensive research involving the simulation of an automobile air conditioning system.

Present paper reports the development of a computer simulation program of a parallel flow micro channels condenser, and its validation through results obtained from an experimental set up. The experimental bench has been developed from actual parts and components of an automobile air conditioning system, as described further on, and it was operated under typical field conditions.

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2. DESCRIPTION OF THE EXPERIMENTAL BENCH

A schematic description of the experimental set up is shown in Figure 2. As noted before, parts and components were obtained from a commercial auto air conditioning system so that field conditions could be adequately reproduced in the laboratory. The compressor was one with a capacity control system, making the refrigeration circuit to operate at a constant suction pressure of approximately 2 bar. It was run by an electric motor with a frequency converter to allow for rotational speed control so that experiments covering the actual range of speeds could be performed. The cooling coil, installed in its actual casing, was a standard aluminum tube and fins whereas a thermostatic expansion valve was used as expansion device. Thermal load was imposed by both, mixing the return with the ambient air, and by electrically heating the incoming air. The condensing air was provided by twin fans installed as in the actual vehicle, and run by their original motors with electrical power provided by a regular 12V automobile battery. The condenser/fan arrangement was installed at the downstream far end of a small wind tunnel constructed for that purpose, as shown in Figure 2. The incoming air temperature was adjusted by a coil of electrical heaters installed at the entrance of the wind tunnel. A voltage converter controlled the electrical power output. A double screen was installed at mid-way in the tunnel in order to further rectify the air stream.

The flow rate of air was measured by both procedures: calorimetric and mapping out the cross section with an electronic Pitot tube. A Coriolis type meter measured the mass flow rate of refrigerant. Pressures and temperatures along the refrigeration circuit were measured and monitored in locations shown in Figure 2. Analog signals from transducers were processed by a data acquisition system from Strawberry Tree, USA. Accuracy of measurements were as follows: (a) temperature(measured with T thermocouples): 0,7°C ; (b) pressure: 20 kPa ; (c) mass flow rate: 0,15% of full scale. Further details of experimental procedures can be found in Ianella (1998).

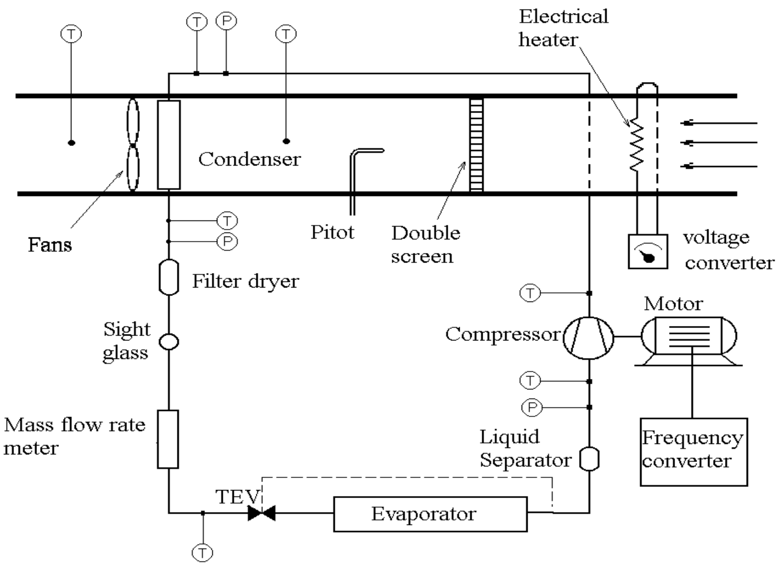

Figure 2 - Schematic diagram of the experimental bench.

3. CONDENSER MODEL

The condenser mathematical model assumes that the heat transfer surface is divided into three regions associated to the state of the refrigerant: the superheated vapor, the saturated and the subcooled liquid regions. Refrigerant and air total flow rates, and the entrance conditions of the refrigerant and air are assumed to be known. Each region is considered as an independent heat exchanger with the total air flow divided for each region, as shown in the procedure below, and the exit state of the refrigerant of one region corresponds to the inlet of the subsequent one.

The overall heat transfer coefficient for each region is determined by assuming that thermal resistances from fouling, metal and contact are negligibly small. The following equation can then be written:

$$1/(UA) = 1/(hA)_{ref} + 1/(\eta hA)_{air} \qquad (1)$$

where η corresponds to the finned surface efficiency, $\eta = 1 - (A_f/A_e)(1-\eta_f)$. The internal value for η is one, since the surface of the micro channels is assumed smooth, with no fins on it. The fin efficiency, η_f, for the louvered air side surface has been determined by assuming fins as plane with rectangular cross section. Correlations for the heat transfer coefficient associated to the refrigerant in each region are given in Table 1. Shah's correlation, Eq. (3), depends upon the quality (local correlation), requiring the evaluation of an average coefficient. This has been accomplished by assuming a constant heat flux along the condensing region, and integrating Eq. (3) in terms of the quality, from 0 to 1.

TABLE 1 - HEAT TRANSFER COEFFICIENTS ASSOCIATED TO THE REFRIGERANT FLOWING INSIDE THE MICRO CHANNELS.

Region	Correlation	Eq. No.
Single phase (liquid/vapor)	Dittus&Boelter (1936): $h = (k/D)0{,}023 Re^{4/5} Pr^{0,3}$	(2)
Condensing	Shah (1979): $h(x) = h_{lo}\left[(1-x)^{0.8} + \frac{3{,}8x^{0.76}(1-x)^{0.04}}{P_{red}^{\ 0,38}}\right]$	(3)

The air side heat transfer coefficient has been determined from the following correlation for louvered air surfaces suggested by Awf [1]:

$$j = 0{,}91Re_a^{-0,5} \tag{4}$$

It must be noted that the correlation has been obtained for the air side geometry considered in this study, i. e. louvered fins brazed on plane blades.

The airside pressure drop has been neglected, whereas its refrigerant side counterpart has been evaluated, since it can affect the refrigerant temperature distribution. Two effects, which tend to compensate each other in the case of a condensing vapor, cause the pressure drop: friction and acceleration (inertia). Table 2 presents the correlations for friction effects used in the computer model.

TABLE 2 - CORRELATIONS OF FRICTION EFFECTS FOR REFRIGERANT FLOWING IN MICRO CHANNELS

Region	Correlation	Eq. No.
Single phase (liquid/vapor)	Yang e Webb(1996); $\Delta P = f\,\frac{2G^2L}{\rho D}$; $f = 0{,}0676\,Re^{-0,22}$	(5)
Condensing	Yang e Webb(1996); $\Delta P = f_{eq}\,\frac{2G_{eq}^{\ 2}L}{\rho_l D}$; $(f_{eq}/f)=0{,}435Re_{eq}^{\ 0,12}$; $G_{eq}=[(1-x)+x(\rho_l/\rho_g)^{1/2}]$	(6)

Inertia effects are neglected in the single-phase regions. In the condensing region the refrigerant state varies from saturated vapor to saturated liquid, so that these effects cannot be ignored, due to the significant density variation. The following correlation applies:

$$\Delta P_{inertial} = G^2\,(1/\rho_g - 1/\rho_l) \tag{7}$$

Pressure drop effects of headers, and inlets and exits of the micro channels are neglected.
Regarding the thermal performance of each region, the (ε, NTU) method has been implemented. According to this procedure, the heat exchanger effectiveness is defined as

$$\varepsilon = f\,(arrangement,\, NTU,\, C^*) \tag{8}$$

Correlations for the heat exchanger effectiveness, ε, and the Number of Transfer Units, NTU, are summarized in Table 3. Equation (9) stands for the single-phase regions, for which the heat exchanger is considered of the cross flow type with both fluids unmixed. As for the condensing region, the temperature of the refrigerant is assumed constant throughout the region though it must be recognized that a small variation can occur due to the pressure drop. Actually, the saturation temperature correspondent to the inlet pressure is assumed in the model to simplify the procedure. The (ε, NTU) correlation for this region is the one for counterflow heat exchangers, Eq. (10).

TABLE 3 - (ε, NTU) CORRELATION FOR THE REGIONS OF THE CONDENSER

Region	Correlation	Eq. No.
Single phase (liquid/vapor)	$\varepsilon = 1 - exp\left[\frac{1}{C^*}NTU^{0.22}\left(exp\left(-C^*NTU^{0.78}\right)-1\right)\right]$	(9)
Condensing	$\varepsilon = 1 - Exp(-NTU)$	(10)

Energy balances are performed for each region and for the whole condenser as follows:

Air side	$Q_{region} = C_a\,[(T_a)_{out} - (T_a)_{in}]$	(11)
Refrigerant	$Q_{condensing} = m_r\, i_{lg}$, condensing region $Q_{region} = C_r\,[(T_r)_{in} - (T_r)_{out})]$, single-phase regions	(12) (13)
Overall condenser	$Q_c = Q_{superheat} + Q_{condensing} + Q_{subcooled}$	(14)

4. NUMERICAL PROCEDURE

The set of equations constituting the condenser mathematical model has been solved through the software Engineering Equation Solver, EES, from F-Chart, USA. In addition to providing solutions for systems of algebraic (and differential) equations, the software incorporates transport properties of several substances including a number of halocarbon refrigerants.

The numerical procedure for the solution is as follows:

(1) Reading of input parameters: condenser geometry, inlet air and refrigerant temperatures and pressures, mass flow rate of refrigerant and face velocity of the air.

(2) Thermal evaluation of the superheated region

- (a) Calculation of the overall heat transfer coefficient and pressure drop
- (b) An iterative procedure is then followed for the determination of the other parameters. Initially, the mass flow rate of air corresponding to this region is assumed to be equal to 10% of the overall air mass flow rate.
- (c) Evaluation of the refrigerant pressure drop.
- (d) Evaluation of the air capacity, C_a, along with C_{min} , ε, and the NTU for the region. It must be noted that the actual heat transfer rate for the region is known, since the inlet and outlet temperatures of the refrigerant are available.
- (e) Evaluation of the product (U A) and the area of the region from NTU.
- (f) With the external heat transfer area, the mass flow rate of air for that region can be determined as

$$(m_a)_{superheated} = m_a\,(A_{superheated}/A_{total}) \tag{15}$$

- (g) The obtained mass flow rate is compared to the initially assumed value, step (b). If the variation is within 1% or less, the program is instructed to deviate to step (2), otherwise to step (b) to start the calculation over again.

(3) Once the superheated region has been solved, the condensing one is started following a similar procedure. The initial assumption for the air mass flow rate for this region is 80% of the overall mass flow rate.

(4) The procedure for the subcooled region is simpler since the mass flow rate of air can easily be determined as the difference between the overall mass flow rate and the combined flow rate of previous regions.

(5) Finally, the exit average air temperature is evaluated by adiabatically mixing the exit air from the three regions along with the total condenser heat transfer rate and the overall value of the product (U A), given by

$$(U\,A)_{cond} = (U\,A)_{superheat} + (U\,A)_{condensing} + (U\,A)_{subcooled} \tag{16}$$

Further details regarding the procedure can be found in Mamani (1997).

5. DISCUSSION OF RESULTS

Results from the proposed model have been compared with those obtained from an actual condenser tested in the experimental set up described herein. The condenser was of the type described in Figure 1, with the following dimensions (please refer to Figure 1):

Micro channels					**Overall condenser dimensions**			
Number of channels/blade	Cross section	h_f [mm]	h_m [mm]	w_m [mm]	L_c [mm]	H_c [mm]	Overall fin area [m^2]	A_e/A_i
16	Square (D_h=1 mm)	8,6	1,5	20.25	580	360	4.344	4.035

A constant face velocity of the cooling air of 3 m/s was maintained throughout the tests, corresponding to a flow rate provided by the twin fan system.

Results have been compared in terms of three important overall physical parameters of the condenser: heat rejection rate, refrigerant pressure drop, and heat transfer coefficient. Figures 3 (a) and (b) present the comparison between experimental and simulated results of the first two parameters.

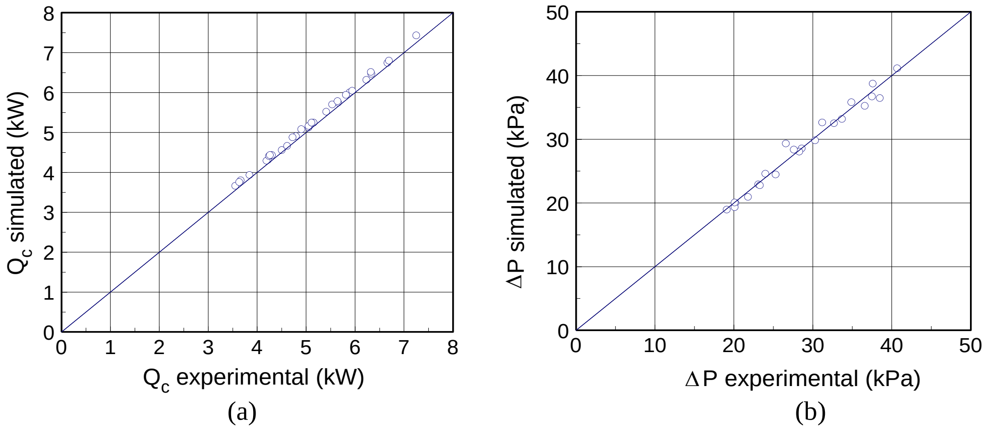

Figure 3 - Comparison of simulated with experimental results in terms of: (a) heat rejection rate; (b) refrigerant pressure drop. Refrigerant: HFC-134a; inlet air temperature: 40°C; refrigerant mass flow rate: 0.0198 to 0.0335 kg/s; inlet temperature of the refrigerant: 70.2 to 94.4°C; saturation temperature of the refrigerant at inlet pressure: 49.2 to 56.7°C.

Data points in those figures correspond to ranges of input parameters indicated in the caption. It can be noted that both, the condenser heat rejection rate and refrigerant pressure drop, compare very well. It must be stressed that results involve a relatively wide range of input parameters, what makes the obtained results more significant. Regarding the pressure drop, neglecting effects of headers, and inlet and exit of the micro channels has not significantly affected model results with respect to the experimental ones.

Heat rejection rate at the condenser was correlated in terms of the difference between the average condensing temperature and the inlet air temperature for both results, experimental and simulated. The resulting correlations are plotted in Figure 4. The average overall heat transfer coefficient can be obtained from this kind of plot by neglecting the effects of the superheated vapor and subcooled liquid regions, so that everything happens as if the refrigerant undergoes a condensing process. It can easily be demonstrated that the slope, R, of the curves of Figure 4 is given by the following correlation:

$$R = C_a [1 - \exp(-UA/C_a)] \tag{17}$$

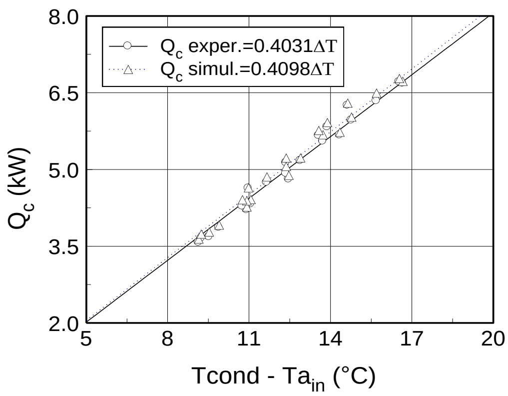

Figure 4 - Q_{cond} vs $[T_{cond} - (T_a)_{in}]$. Refrigerant: HFC-134a; inlet air temperature: 40°C. Same ranges of other input parameters as in Figure 3.

The slope for each data set is indicated in this figure. Thus, given that the mass flow rate of air is known and constant, product (U A) can be determined from Eq. (17). This U value corresponds to the average overall heat transfer coefficient extensive to the range of input parameters. Values of (U A) from experimental and simulated results were, respectively, equal to 0.7890 kW/ºC and 0.8186 kW/ºC. The simulated (U A) deviates 3.75% from the experimental.

This figure is rather low, considering that the simulation model incorporates correlations for heat transfer and pressure drop from the open literature most of them developed for standard size and shape channels. For each set of input parameters, (U A) was also evaluated from Eq. (16), with results being slightly lower than the average from the above procedure. This result was predictable since, in this case, effects of single phase regions are taken into account.

6. CONCLUSIONS

The proposed model for parallel flow micro channels condensers for automobile air conditioning applications has produced physically sound results. Quantitatively, results from the model in terms of overall parameters, namely heat rejection rate, refrigerant pressure drop, and overall heat transfer coefficient, compare very well with the experimental ones. Deviations were rather low, mostly remaining in the range from 1 to 2%. Noteworthy in the model is the fact that heat transfer and pressure drop correlations were obtained from the open literature for standard channel size and geometry.

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8. NOMENCLATURE

A	Area	(m^2)
C	mc_p, heat capacity	($kW/^oC$)
C^*	C_{min}/C_{max}	
D	Diameter	(m)
f	Friction factor	
G	Mass velocity	(kg/m^2s)
h	Heat transfer coefficient	($kW/m^2\ ^oC$)
j	Colburn factor, $Nu/(RePr^{1/3})$	
k	Conductivity	($kW/m\ ^oC$)
L	Length	(m)
m	Mass flow rate	(kg/s)
NTU	Number of transfer units	
P	Pressure	(kPa)
Pr	Prandt Number	
Q	Heat rejection	(kW)
Re	Reynolds number	
T	Temperature	(^oC)
U	Overall heat transfer coeff.	($kW/m^2\ ^oC$)
x	Quality	
ρ	Density	(kg/m^3)

Subscripts

a	Air
c	Either cool fluid or condenser
e	External
eq	Equivalent
f	Fin
g	Either gas or vapor
h	Hydraulic diameter, hot fluid
i	Internal
l	Liquid
lo	Total mass flow as liquid
r	Refrigerant
red	Reduced

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