International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

1990

Computing Design of Heat Pumps with Thermal Storage

V. Havelsky
Slovak Technical University

IMPACT OF THE CAPILLARY TUBE AND CONDENSER MODELLING APPROACH ON THE PERFORMANCE OF A DYNAMIC SIMULATION PROGRAM FOR DOMESTIC REFRIGERATORS

C. Melo (*), R.T.S. Ferreira (*), R.H. Pereira (**), A.L.M. Aranda (**)

(*) Department of Mechanical Engineering
Federal University of Santa Catarina
P.O. Box 476
88049 - Florianópolis - SC - Brazil

(**) Empresa Brasileira de Compressores S/A
Application Engineering
P.O. Box 476
89200 - Joinville - SC - Brazil

ABSTRACT

The objective of this work is to assess the sensibility of a computer simulation program, employed to analyse the transient behaviour of domestic refrigerators, to the adopted capillary tube and condenser mathematical modelling approach. Although all the comparisons, presented in this work, are related to a specific computer program, it is believed that, at least in a qualitative way, they are also valid for most of the computer simulation programs of vapor compression refrigerating systems, available in the open literature.

NOMENCLATURE

A	area, m^2
C	thermal capacity, kJ/K
Cp	specific heat, J/kgK
D	diameter, m
Dh	hidraulic diameter, m
e	fin thickness, m
f	friction factor
G	mass velocity based on tube cross-sectional area, kg/sm^2
H	stagnation enthalpy, kJ/kg
h	specific enthalpy, kJ/kg
hc	conv. heat transfer coef., W/m^2K
j	Colburn factor ($hcPr^{2/3}/QCp$)
K	entrance loss factor
k	thermal conductivity, W/mK
L	length, m
ℓ	distance between adjacent fins, m
M	mass, kg
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number (hcD/k)
P	pressure, N/m^2
Pr	Prandtl number ($\mu Cp/k$)
Q	mass velocity in the coil where the minimum area occurs, kg/sm^2
R	thermal resistance, K/kW
Re	Reynolds number ($4Q/\pi D\mu$)
St	Stanton number (hc/QCp)
T	temperature, K
t	time, s
V	volume, m^3
v	specific volume, m^3/kg
x	vapor quality
Xa	transverse tube spacing, m
Xb	longitudinal tube spacing, m
y	fin height, m
Z	compressibility factor
β	constant
ΔT	temperature drop across condensate film, K
λ	latent heat, J/kg
μ	absolute viscosity, kg/ms
ρ	density, kg/m^3
σ	ratio of minimum flow area to frontal area

Subscripts

a	air
i	inlet, inside
f	fin
ℓ	liquid
o	outlet, outside
t	two-phase flow
v	vapor
w	wall
1	superheated region
2	saturated vapour region
3	saturated liquid region
4	subcooled region
∞	surroundings
*	previous time step

1. INTRODUCTION

For decades the only available technique to ascertain the performance of domestic refrigerators was to perform experimental tests, according to some specific standard, as for example the one presented in reference [1]. Such tests require, normally, a time period of, approximately, 24 hours, considering both the test period and the necessary time for the environmental test chamber to reach the initial steady state conditions. Nowadays the need for energy conservation and mainly, the need to replace CFC 12, in refrigeration systems, increased the required number of tests to such a level that most of the application laboratories are not able to manage. One way to speed up such procedure is to employ computational techniques to numerically simulate the refrigerator performance.

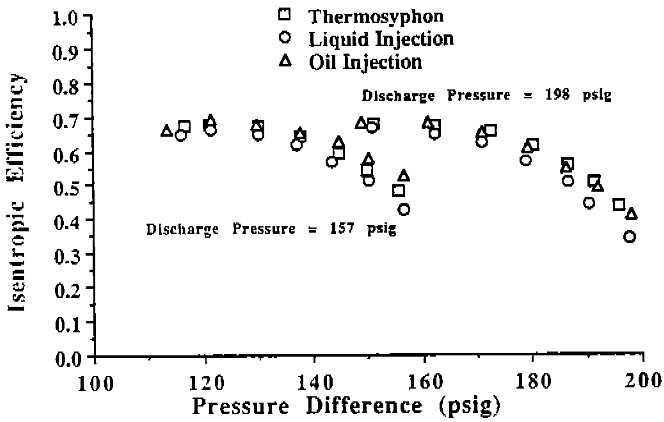

Figure 9 Isentropic Efficiency as a Function of the Discharge to Suction Pressure Ratio for the Oil Cooling Systems.

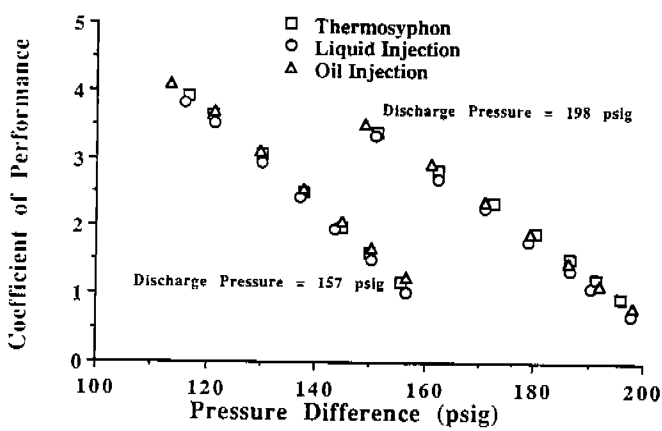

Figure 10 Coefficient of Performance as a Function of the Discharge to Suction Pressure Ratio for the Oil Cooling Systems.