Diss. ETH Nr. 9870

A Contribution to Heat Pump Design by Simulation

A dissertation submitted to the
SWISS FEDERAL INSTITUTE OF TECHNOLOGY
ZURICH

for the degree of
Doctor of Technical Sciences

presented by
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Zürich 1992

Juris Druck + Verlag Dietikon
1992
To my mother and

to the memory of my father, and

to Béatrice and Oliver!

Eles não sabem que o sonho
é uma constante da vida
tão concreta e definida
como outra coisa qualquer,...

António Gedeão - Pedra Filosofal
Poesias Completas
FOREWORD

This work was carried out at the Laboratorium für Energiesysteme - Eidgenössische Technische Hochschule Zürich, under the direction of Prof. Dr. Peter Suter. To him my most sincere gratitude for his guidance, and for the many discussion opportunities. I thank Prof. Dr. Daniel Favrat of the Laboratoire d’Energetique Industrielle - Ecole Polytechnique Fédérale de Lausanne, and Prof. Dr. Georges Yadigaroglu of the Laboratorium für Kerntechnik - Eidgenössische Technische Hochschule Zürich, for being the co-examiners to this thesis, and for their careful reading and insightful comments. The Eidgenössische Energie und Verkehrsdepartment, especially the head of the Sektion Energietechnik - Mr. Hans-Ulrich Schärer - are sincerely acknowledged for their interest in this work and for the financial support. Thanks go as well to the heat pump manufacturer Ernst Schweizer AG in Hedingen - Zürich for their interest in this research and for the support and help during the early stages of the experimental work. I thank the technicians of the Laboratory, Mrs. Martin Meuli and Max Hard, and the late Ernst Reich for their help in the construction of the experimental setup. Special thanks are due to special people who, in one way or another influenced me or my work during this challenging period of my life: to Prof. Dr. Eduardo de Oliveira Fernandes of the Faculdade de Engenharia da Universidade do Porto - Portugal, for his guidance, vision and uninterested help; to Prof. Dr. Jan Berghmans, of the Departement Werktuigkunde - Katholieke Universiteit Leuven - Belgium, for his support during my first contact with heat pump research at his laboratory; to the Calouste Gulbenkian Foundation, in Lisboa - Portugal, for their financial assistance during the early times of my heat pump research; and last but never the least to my wife, Béatrice, for her understanding and bountiful support all along.

Manuel de Resende Conde
Zürich, Spring 1992
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VITA
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Heat pumps, both mechanically and thermally driven, do not always yield thermodynamic efficiencies according to the expectations. The reasons for this are that the efficiencies of heat pump components are lower than practically possible. Various methods to increase the efficiency of heat pumps may be followed: Better integration with the heating system, using for example thermal storage or variable speed; Better integration (matching) of the components in the heat pump itself; And, improvement of the heat pump's individual components. Although none of these methods excludes the others, this work deals with the better integration of the components in vapour compression heat pumps, and yielded as by-product tools that may as well be used to improve some of the components individually.

The approach presented uses mathematical modelling of the physical processes in the components for simulation by computer. The concept of the computer program has been developed as a modular framework permitting the simulation of various heat pump configurations, and of various heat transfer fluids for source and sink. The configuration detailed here as an example, is an air-to-water heat pump with a hermetic reciprocating compressor, a coiled-coaxial condenser, a thermostatic expansion valve, and a plate finned-tube coil evaporator. An experimental test rig has been built based on this configuration, and instrumented in detail to produce data for the verification of the simulation program. The results of this verification are discussed in detail. Studies of the effects of variation over large ranges of individual variables are also reported and discussed.

The simulation program will serve as a basis for future developments, which include the modelling of other types of components and configurations, and extension to variable speed heat pumps. A critical aspect of the whole process is the unavailability of good experimental data for verification and validation of the mathematical models and program modules.
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Zusammenfassung


Das Simulationsprogramm dient als Grundlage für die Modellierung von weiteren Komponententypen und Wärmepumpekonfigurationen sowie für die Ausdehnung auf drehzahlgeregelte Wärmepumpen. Eine der Schwierigkeiten bei der Verifikation und der Validierung von mathematischen Modellen und Programmen ist das Fehlen von zuverlässigen experimentellen Daten.
Nomenclature

- Angle brackets \( (\cdot) \) indicate values averaged over the cross section.
- Angle brackets with a subscript \( (\cdot)_k \) indicate values averaged over the part of the cross section occupied by the phase \( k \).

\[
\begin{align*}
A & \quad \text{Heat transfer surface area, area of free flow section, total channel surface area} \\
A_c & \quad \text{Heat transfer area of cell in the simulation of the evaporator} \\
A_f & \quad \text{Fin surface part of the heat transfer area} \\
A_{\text{fin}} & \quad \text{Fin surface area} \\
A_{\text{frontal}} & \quad \text{Frontal area of the heat exchanger (evaporator)} \\
A_i & \quad \text{Internal (water side) heat transfer surface area} \\
A_m & \quad \text{Logarithmic mean heat transfer surface area} \\
A_{\text{min}} & \quad \text{Area of the minimum free flow section on the air side} \\
A_o & \quad \text{Outer (total) finned tube surface area, External heat transfer surface area} \\
A_p & \quad \text{Area of the fin cross section} \\
A_{r} & \quad \text{Tube part of the heat transfer area (condenser)} \\
A_{s} & \quad \text{Surface area of a bare tube} \\
A_{\text{t}} & \quad \text{Total air side heat transfer area per tube} \\
A_{\text{tube}} & \quad \text{Tube side heat transfer area} \\
C_p & \quad \text{Specific thermal capacity at constant pressure} \\
C_v & \quad \text{Specific thermal capacity at constant volume} \\
D & \quad \text{Compressor volumetric displacement, generic diameter} \\
D_b & \quad \text{Curvature diameter of a return bend} \\
D_h & \quad \text{Hydraulic diameter} \\
D_o & \quad \text{Outer tube diameter (over fin collar)} \\
d & \quad \text{Inner tube diameter, minimum inner tube diameter in grooved tubes, fan rotor diameter} \\
e & \quad \text{Fin height} \\
F_i & \quad \text{Internal fouling resistance} \\
F_o & \quad \text{Outer fouling resistance} \\
F_{pd} & \quad \text{Fin pattern depth (wavy fins)} \\
F_{pdf} & \quad \text{Fin pattern depth under frost} \\
F_s & \quad \text{Fin spacing} \\
F_t & \quad \text{Mean frost thickness}
\end{align*}
\]
Friction coefficient
Fanning friction factor \( f = 4 f_f \)
Acceleration of gravity
Enthalpy, Coil pitch (coiled-coaxial condenser), Level difference

total enthalpy of phase k, given as

\[
h_{\mathrm{o}}^k = h_k + \frac{\nu_k^2}{2} - g z \cos \theta
\]

Electric current
Specific enthalpy of condensation (vaporization)
Specific enthalpy of sublimation
Specific enthalpy of condensation or desublimation of water
Colburn j-factor
Colburn j-factor for dry operation
Colburn j-factor for dry operation in the McQuiston's correlation
Total Colburn j-factor in the McQuiston's correlation
Overall heat transfer coefficient, entrainment factor
Kinetic energy
Loss rate (electrical)
Flow length
Length of fin strip
Mass
Mass flow rate
Mass flux
Air-side mass flux through the minimum free flow section
Rate of condensation at a cell in the evaporator
Flow rate of condensate across a cell in the evaporator
Rotation speed (rpm)
Number of fin patterns per longitudinal tube pitch
Number of tube rows in the heat exchanger
Tube row order number
Exponent in the Blasius equation, rotation speed (rps)
Number of fin strips per longitudinal tube pitch
Pressure
Potential energy (gravity)
\( P_i \)  
Interfacial perimeter

\( P_{h,k} \)  
Heated wall perimeter in contact with phase k

\( P_f \)  
Total fan pressure head

\( P_{w,k} \)  
Wall perimeter wetted by phase k

\( p \)  
Wetted perimeter of the fin cross section

\( P \)  
Fin pitch

\( Q \)  
Thermal power (heating rate)

\( q \)  
Heat flux

\( q'_{i,k} \)  
Interfacial heat flux from the interface into phase k

\( q'_{w,k} \)  
Heat flux from the wall into phase k

\( q''_k \)  
Internal heat generation rate in phase k

\( R \)  
Resistance (electrical), Radius of the equivalent fin

\( R \)  
Refrigeration rate (cooling capacity)

\( R \)  
Universal gas constant

\( R_o \)  
Radius of the equivalent circular fin (same area)

\( S_L \)  
Longitudinal tube spacing (in the air flow direction)

\( S_T \)  
Transversal tube spacing (normal to the air flow direction)

\( s \)  
Slip ratio

\( T \)  
Temperature (absolute)

\( T_{\text{air,sat,surf}} \)  
Saturated air temperature at the fin surface

\( T_{DP} \)  
Dew-point temperature of the humid air

\( t' \)  
Normalized temperature \((t + 273.15) / 273.15\)

\( t \)  
Temperature (°C), Fin thickness

\( t_b \)  
Fin thickness at the root

\( U \)  
Voltage

\( u \)  
Velocity

\( u_c \)  
Velocity of the condensate film

\( u_i \)  
Interface velocity (relative)

\( V \)  
Volume

\( \dot{V} \)  
Volumetric flow rate

\( \nu \)  
Specific volume

\( W \)  
Power (electric)

\( W_s \)  
Width of the fin strips

\( w_c \)  
Width of cell in the evaporator

\( X \)  
Opening displacement (thermostatic expansion valve)

\( X_{tt} \)  
Lockhart-Martinelli parameter for turbulent-turbulent flow

\( x \)  
Vapour mass quality of the refrigerant
\( x_a \)  
Humidity ratio of the humid air

\( x_{do} \)  
Refrigerant mass quality at the onset of the wall dryout

\( z \)  
Length

**Greek symbols**

\( \alpha \)  
Convective heat transfer coefficient, Thermal diffusivity

\( \alpha_f \)  
Heat transfer coefficient on the vertical fin surface

\( \alpha_h \)  
Heat transfer coefficient on the horizontal tube part

\( \alpha_m \)  
Mass transfer coefficient

\( \alpha_{nb} \)  
Heat transfer coefficient in nucleate boiling

\( \alpha_{TP} \)  
Refrigerant two-phase heat transfer coefficient

\( \beta \)  
Coefficient of thermal expansion

\( \Gamma \)  
Property index, volumetric mass generation rate (positive for vaporization)

\( \gamma \)  
Exponent of the isentropic

\( \Delta \)  
Difference, variation

\( \delta_c \)  
Thickness of the condensate (frost) layer

\( \epsilon \)  
Surface rugosity, Flow void fraction, Clearance volume ratio, emissivity

\( \epsilon_k \)  
Fraction of the cross section occupied by phase k

\( \zeta \)  
Fraction of the energy dissipated at the interface that gets transferred to the gas phase, single phase friction coefficient (Friedel), momentum change coefficient (Eck)

\( \eta \)  
Efficiency

\( \eta_{fin} \)  
Fin efficiency

\( \eta_{surf} \)  
Extended surface heat transfer efficiency

\( \theta \)  
Span arc of a bend, Temperature difference between fluid and fin surface, poppet head angle (thermostatic expansion valve)

\( \theta_0 \)  
Temperature difference between fluid and fin surface at the fin root

\( K \)  
Local pressure loss coefficient

\( \kappa \)  
Exponent of a polytropic

\( \Lambda \)  
Heat transfer coefficient multiplier

\( \lambda \)  
Thermal conductivity, fan dimensionless power

\( \mu \)  
Dynamic viscosity

\( \nu \)  
Kinematic viscosity

\( \xi \)  
Mass concentration (refrigerant in oil-refrigerant mixture), friction coefficient (Gnielinski)

\( \rho \)  
Density
Surface tension, Stephan-Boltzmann constant, polytropic index

Interfacial shear stress acting on the gas phase

Fan dimensionless volumetric flow rate

Shear stress between the wall and phase k

Two-phase friction multiplier, boiling heat transfer multiplier

Pressure ratio (compression), fan dimensionless total pressure

Angle between the positive z direction and the acceleration of the gravity

Superheating adjustment angle (thermostatic expansion valve), arc spanning the wetted perimeter in stratified flow

Specific compression work, Concentration of the vapour phase in the noncondensible gas

Boiling number \( (Bo = \dot{q} / (\dot{m} l_g)) \)

Convection number \( (1/x - 1)^{0.8} \left( \rho_G / \rho_L \right)^{0.5} \)

Dean number

Froude number \( (Fr = \dot{m}^2 / \rho_L^2 g d) \)

Graetz number \( (Gz = Re Pr D_h / (N_r S_L)) \)

Nusselt number

Prandtl number

Reynolds number \( (Re = \dot{m} d / \mu) \)

Stanton number

Air, humid air, adjustment

Actual

Thermostatic bulb, phial (thermostatic expansion valve)

Bubble suppression regime

In the stream

Convective boiling regime

Condenser

Catalog

Chisholm correlation

Compressor

Coupling
cr  At the critical state
curv  Curvature
D  Duct
d  Drive
da  Dry air
diaph  Diaphragm
Disch  Discharge
dp  At the discharge port
dp,s  At the discharge port, by an isentropic process
dy  Dynamic (flow)
e  Electrical, evaporator
eff  Effective (real)
elec  Electrical
eq  Equivalent
fg  Boiling, condensation (enthalpy)
G  Gas or vapour phase
GO  Gas or vapour alone
inl  Inlet
is  Insulation
L  Liquid
LO  Liquid alone
mech  Mechanical
mot  Motor
N  Nominal
nb  Nucleate boiling regime
o  Outer
P  Phase
r  Refrigerant, fin root, reduced (pressure, temperature)
s  Smooth, slip ratio, isentropic, saturated, spring, straight
sat  Saturated
sg  Solid to gas or vapour (sublimation, desublimation)
SH  Superheating
sp  At the suction port
st  Stall (fans), static
suc  Suction
T  Throat (thermostatic expansion valve)
TP  Two-phase
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<thead>
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<th>Description</th>
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<tr>
<td>$t$</td>
<td>Total</td>
</tr>
<tr>
<td>VS</td>
<td>Vapour to saturation</td>
</tr>
<tr>
<td>wo</td>
<td>Wall outer surface</td>
</tr>
<tr>
<td>wm</td>
<td>Metallic wall</td>
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1. INTRODUCTION

The wheel’s hub holds thirty spokes, utility depends on the hole through the hub. The potter’s clay forms a vessel, it is the space within that serves. A house is built with solid walls, the nothingness of window and door alone renders it useful. That which exists may be transformed, what is nonexistent has boundless uses!

Lao-Tse
Ever since the proposition of its principle by Sir William Thomson in 1852, the heat pump has been lurking as an interesting solution to the generation of low grade heat. It has gone through various waves of enthusiastic interest and frustrating decline, since the first practical applications in the 1930s (Haldane 1930). Although they are no panacea to the problem of covering the heating needs in cold and temperate climates, heat pumps do hold the potential for a significant contribution, that is, they are part of the solution. The more so, as the environmental consequences of the widespread use of the competing firing systems are seen as important generators of greenhouse active gases.

The reasons that prevent heat pumps from getting a larger share of the generating duties of low grade heat are manifold. Those originating in the heat pumps themselves are of utmost interest here, but it shouldn’t go without pointing out the exogenous causes: The current pricing system for fuels in general is unfair. The fuel prices respond to the market forces, but the supply is only limited by the production capacity, and not by their general availability in a satisfactory human time scale (Hubbert 1979). On the other hand, prices reflect almost exclusively the costs upstream of consumption, that is, an unlimited disposal capacity is assumed. In this setting, the only way for the number of heat pump applications to grow is through improved efficiency and reliability. The reliability question is nowadays considered settled, but the efficiency of heat pump systems still offers significant opportunities for improvement. These improvements must go beyond the simple overcoming of the limitations of the current pricing structure, and help reduce the environmental effects of the power generation process. The opportunities for improvement of heat pump systems lie at three levels: Component design, machine design, and heating systems design. These three levels were considered when defining the scope of this work. Although heating systems design, using heat pumps, offers many possibilities for improvement, it must be recognized that a badly designed heating system will make a good heat pump perform poorly, but no good heating system design will make a bad heat pump perform better. Therefore, a contribution to either
component or machine design, or both, seemed the more desirable. This
contribution might take many forms, though the general availability of cheap
computing power, that started in the mid 1980's, offered an excellent way for
developing affordable design tools, applicable to both individual components and
whole machines. These design tools consist of mathematical models of the
components, and are based on the detailed description of the physical phenome-
na taking place in them. The models are part of a general framework for the
simulation of vapour compression heat pumps and refrigeration machines
(reverse Rankine cycle). Its purpose is the study, by simulation, of machine and
component designs, in particular their ability to match properly under variable
operating conditions. The ENHANCED\textsuperscript{1} heat pump simulation framework has
been implemented into the computer program \textit{HPDesign}, and applied to an air-to-
water configuration. Air-to-water heat pumps result in the lowest capital costs,
and represent about 70\% of all heat pumps installed in Switzerland for space
heating (Schärer 1992). They are as well the most affected by variable operating
conditions. So, special attention is given to the low pressure side, because even
small improvements of the air-heated evaporator have a significant effect on the
overall performance. The expansion device, in particular a thermostatic expansion
valve, plays in this regard a prominent role as it should feed the evaporator with
the right amount of refrigerant. Thus, although all the component models were
developed anew, it is on these two components - thermostatic expansion valve
and air heated evaporator - that the strongest modeling effort has been done.
The \textit{ENHANCED} heat pump simulation framework represents a step beyond
previous simulation models, as discussed in Chapter 2. There I review and
classify the modeling methods known from the literature. The models already
developed in the \textit{ENHANCED} simulation framework are thoroughly described in
Chapter 3, including the general solution algorithm. The data required by the
individual component models are discussed in Chapter 4. The Chapter 4 also

\textsuperscript{1} So called in relation to simpler models developed earlier (Conde 1987, Afjei 1989), but
essentially because it permits the detailed observation of how the behaviour of the
individual components contributes to overall performance.
includes a method developed to generate reciprocating compressor characteristics for new refrigerants from those already known. The experimental test rig built on purpose to obtain data for verification of the models is described in Chapter 5. The results of simulation and experiment are compared and discussed as well in Chapter 5, which also includes the results of a systematic study of the effects of variations of some key variables. The general conclusions of this work are summed up in chapter 6, while chapter 7 proposes a comprehensive program for further developments.
2. HEAT PUMP SIMULATION MODELS IN THE LITERATURE

He had bought a large map representing the sea,
without the least vestige of land:
And the crew were much pleased when they found it to be
A map they could all understand.

Lewis Carrol
2.1 Classification of Models

The simulation of vapour compression reverse Rankine cycle machines, with emphasis on either the refrigeration or the heating effect, has been the subject of extensive research in the past twenty years\(^1\). The availability of general purpose computers induced, as in many other fields of science and engineering, the development of mathematical models suitable for implementation into computer programs. Starting by the demonstration of feasibility and economy, these models and the computer programs derived from them, were rapidly applied in the optimization under various perspectives: They were used to study control strategies, to evaluate economic advantages of competing systems under real application conditions, to analyze the response of the systems when submitted to various kinds of perturbations, etc. A variety of models have been proposed according to the nature and objective of the studies, and there are various classification possibilities for them. A classification based on complexity will be used here to generate a clear picture of the modelling effort done in this field. The following six categories seem to cover all types of models known to date:

- Cat. A: Simple steady state models
- Cat. B: Simple transient behaviour models
- Cat. C: Steady state models with simple description of the individual components
- Cat. D: Transient behaviour models with simple description of the individual components
- Cat. E: Steady state design models
- Cat. F: Detailed transient behaviour models.

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\(^1\) Emphasis is naturally put on heat pumps here, understood as machines from which the useful effect is heating.
2.2 Models in the Literature

Cat. A: Simple steady state models

The steady state input-output\textsuperscript{1} models of the whole machine represent the machine response to the operation variables (source and sink temperatures) by one or more algebraic equations, giving for example, the heating rate and required power as functions of these variables exclusively. Models of this type are particularly useful in the study of more complex systems such as a building or industrial plant. The literature is rich in this kind models, and it is easy to apply corrective factors accounting for various kinds of phenomena, such as frosting in the case of air source heat pumps. Models of this type are reported or used by Sisk, Makiel and Veyo (1976), Jaster and Miller (1980), Goldschmidt and Hart (1982), Rosell, Morgan and McMullan (1982), Rice, Fischer and Emerson (1984), Havelský (1986), Conde (1987), Crawford and Shirey (1987), Halozan (1987), and Afjei (1989).

Cat. B: Simple transient behaviour models

The transient behaviour models with one or more time constants are similar to the previous type, but include as well the machine response to sudden variations of the operation variables, in particular at startup and shutdown. They are useful in the study of system control strategies, and in the analysis of the stability of more complex systems. Examples of this category of models were reported by Groff and Bullock (1976), Dutré, Berghmans and Debosscher (1978), Ofermann (1981), and Tree and Weiss (1986).

\textsuperscript{1} Also named black-box models.
Cat. C: Steady state models with simple description of the components

Steady state models with input-output description of the individual components are the logical next step to the previous two categories. They are inherently simple and provide some information regarding the internal status of the machine. Practical applications are, for example the rapid verification of design decisions, and, depending on the accuracy of the data they are based upon, may constitute an essential tool for the equipment designer. Models of this type have been proposed by Jones et al. (1975), Ahrens (1980), McMullan and Morgan (1981), Tassou, Marquand and Wilson (1982), Gruber-Johnson and Wehrli (1983), Hamam and Rocaries (1983), Krakow and Lin (1983, 1987), Hawken and Lemal (1984), Cecchini and Marchal (1987, 1991), Domanski and McLinden (1990), Ney (1990), Silvestri and Buckman (1990), Armand et al. (1991), Mondot (1991), and Rogers and Tree (1991).

Cat. D: Transient behaviour models with simple description of the components

The transient behaviour models with simplified description of the individual components, represent the components' response to perturbations of the operation variables by first order ordinary differential equations. The internal status of the machine may be described in time, so the mutual influences of the components' responses may be analyzed. Models of this kind have been reported by Bruijn, van der Jagt and Machielsen (1978, 1980), Cleland (1983), MacArthur (1984), Murphy and Goldschmidt (1985), Rajendran and Pate (1986), Gruhle (1987), MacArthur (1987), Sami et al. (1987), Wong and James (1987), Melo et al. (1988), and MacArthur and Grald (1989).

Internal perturbations and their effect upon the systems coupled with the heat pump may be successfully studied with such models.
Cat. E: Steady state design models

Steady state design models describe the operation of every component in detail. The processes taking place in the various components are assumed stationary. The description of the components themselves requires accurate geometric and material data, and the knowledge of their limitations, and operating principle. In the cases of the compressor and expansion device, the operation characteristics are also required. Design models describing accurately the operation of the individual parts, may successfully replace expensive testing, and provide information for operating conditions difficult to realize even in the best test rigs. They permit the study of variations in component design and may eventually allow optimization for the most common operating conditions. In particular, the analysis of component matching is an important characteristic of this kind of models. It is possible to trace the origin of all models of this kind to two seminal works. In the USA, Hiller and Glicksman (1976) reported the very first simulation model - MIT model - of the steady state operation of compression heat pumps. All further developments of steady state design models in the USA represent stepwise improvements to this original model. In France, Haberschill (1983) submitted his dissertation, which includes a steady state model of compression heat pumps. Haberschill's model is later used by Hamdad (1988) to study ground-source heat pumps.

The publication of the MIT model spawned intense modelling activity both at the Oak Ridge National Laboratory with the ORNL series of models, and at the National Institute of Standards and Technology with the NIST series of models. The MIT model was developed to study compressor capacity control schemes as a means of improving heat pump performance. Consequently, the mathematical description of the compressor is very elaborate, as required for such a study. On

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1 The National Bureau of Standards, NBS, in Washington DC changed its name to National Institute of Standards and Technology, NIST, in 1990.
the other hand, the models of the heat exchangers and expansion device are comparatively simple. The simulation algorithm requires the specification of the thermodynamic state of the refrigerant at the evaporator outlet (vapour superheating) and the state of the refrigerant at condenser outlet is assumed subcooled and left fluctuating. The expansion device considered is a thermostatic expansion valve described as a simple orifice. Besides requiring the specification of the refrigerant superheating at the evaporator outlet, this model did not accept the specification of the source temperature, which was an output of the simulation.

At the ORNL (Ellison and Creswick 1978, Ellison et al. 1979, Fischer and Rice 1981, 1983) the compressor description of the MIT model was simplified, bringing it more in line with the complexity of the other components’ descriptions. A model for capillary tubes as expansion devices was added, and that of the thermostatic expansion valve was improved. Still the version I. of the ORNL models kept the source temperature as an output of the simulation. This required a trial and error approach together with interpolation procedures to obtain results for a given set of operating conditions. Further improvements were successively added, namely eliminating the requirement to specify the refrigerant superheating at evaporator outlet (Fischer, Rice and Jackson 1988). Other improvements to the ORNL model are planned (op. cit.) including the calculation of the refrigerant charge inventory (Rice 1987). These improvements have not been reported so far.

The series of simulation models developed at the NIST (Chi 1979, Domanski and Didion 1983, Domanski 1986) were as well a response to the limitations of the MIT model at first, and then to those of the ORNL model too. Chi’s model (Chi 1979) is similar to first ORNL model, requiring still the specification of some refrigerant states, and giving the source air conditions as output. The second NIST model (Domanski and Didion 1983) improves over the MIT and the first ORNL models, by using a physically based description of the capillary tube - the ORNL model used functions adjusted to data in the literature. It also avoids the
specification of refrigerant states, and requires only the source and sink temperatures, humidities and flow rates. The refrigerant superheating at the evaporator outlet is calculated through the refrigerant charge inventory, although considerable discrepancies were found between predicted and effective refrigerant charge.

The heat pump model reported by Haberschill (1983) is specific to a water-to-water heat pump. Although the model is rather detailed, some of the simplifications adopted affect its value as a design model: The heat exchangers are assumed isobaric (no pressure drop of the refrigerant), and divided into three and two lumps, respectively for the condenser and for the evaporator; Despite their very special geometry - coiled-coaxial with surface enhancements - they are described as plain, straight horizontal tubes. The compressor considered is an open type, and the compression is described as a polytropic process. The throttling device is a thermostatic expansion valve described by equations adjusted to manufacturer's data. Also no charge inventory is made, and the pressure and heat losses in the refrigerant piping are neglected. Contrary to the first models developed in the USA, this model does not require the specification of any refrigerant state, all being determined out of the conditions of the source and sink fluids.

The simulation model reported by Hamdad (1988) retakes Haberschill's approach with no significative modification. The model described by Yuan et al. (1989) applies as well to water-to-water heat pumps only. The heat exchangers are, in this case, of the shell-and-tube type with condensation and vaporization on the shell side. The assumption of counterflow seems difficult to justify. On the other hand, dividing the condenser and the evaporator into regions is only required as consequence of that assumption. The expansion device is simulated as a diaphragm with one parameter identified from tests.
The development of a model of this kind is perhaps the most often set objective in the research on heat pumps. However, establishing a true model of the transient behaviour of all components in a heat pump is a formidable task. The tentative made to date are mere approximations. Only with great simplifications, and with the use many equations obtained for steady state operation, is this at all possible. All that is known about thermal processes involving convective transport, relate to steady state conditions and other idealizations. In order to give at least approximate answers in the simulation of transient processes the use of machine specific empirical parameters is required. The advantages of a true model of the transient behaviour have long been recognized, though: The optimization of the machine under any kind of perturbation, especially during the most common instationary conditions, such as startup, shutdown and defrosting, for example. There are anyway a few models reported that approximate this ideal in a reasonable way; Dhar (1978), Yasuda et al. (1981), Chi and Didion (1982), Belth (1984), Beckey (1986), Upmeier (1989), Ney (1990), and Chen and Lin (1991).

2.3 Conclusions on Design Models

An overview of the steady state design models reported in the literature allows the following conclusions:

All design models reported are oriented towards a specific type of heat pump either air-to-air, or water-to-water.

The models of the individual components have attained a high level of sophistication, with most of the phenomena identified to date described at least in a simplified manner. Exceptions to this are:
The model of the thermostatic expansion valve, recognized as unsatisfactory.

The models of the refrigerant lines, including the refrigerant distributor supplying the direct expansion evaporator for air-source heat pumps, which do not consider energy and momentum losses, and storage of refrigerant.

The calculation at the local level of refrigerant holdup in the heat exchangers.

The calculation at the local level of humidity condensation, or desublimation in direct expansion evaporators for air-source heat pumps. The effects of this condensate as it flows down is also not considered.

The possibility of calculation time economy, by taking advantage of eventual symmetry of the circuitry arrangements in air-refrigerant heat exchangers.

Components such as fans and pumps, in many cases part of the heat pump itself, are described by rudimentary models, or not at all.

Flexibility on the type of refrigerant used is not built-in the computer programs reported.

Simulation of multiple components such as multiple compressors, condensers or evaporators has not been considered to date.

Variable compressor speed is allowed by some models, but it is not intended as an operation variable, this is, a fixed speed is assigned to each run.
2.4 Features of Design Models

From this overview, it is possible to establish a list of desirable features and improvements to existing design models:

A. The solution algorithm should be general and independent of the type of heat pump and of its components;
B. The models of components such as the thermostatic valve and of the air-refrigerant evaporator should describe better the hardware;
C. The refrigerant lines' models should be more realistic;
D. The refrigerant charge inventory should be made for all components, including the heat exchangers, compressor and piping.
E. The auxiliary components that are part of the heat pump should be described by more appropriate models;
F. Simulation of multiple components for the same function should be introduced;
G. Capacity variation via compressor speed control, with speed as an operation variable, should be simulated.

Considerable complexity is involved in the development of mathematical models and of their corresponding algorithms to implement all these desirable features. It should be noted as well that a generalized concept is required as a guiding principle for this implementation.

The ENHANCED simulation model concept is thought as a response to this demand, and as a framework within which the simulation of vapour compression reverse Rankine cycle machines shall be carried out easily.

From the above list of desirable features of design models, only the last two, F and G, were not treated in the present form of the ENHANCED simulation model. They should remain a top priority in future developments, however.
3. THE *ENHANCED SIMULATION MODEL*

It is quite a three pipe problem, and I beg you won't speak to me for fifty minutes.

Sherlock Holmes - The Red-Headed League
Sir Arthur Conan Doyle
A machine operating on the reverse Rankine cycle with vapour compression, is built of four basic components; the compressor, the condenser, the expansion device, and the evaporator, Fig. 3.1. Besides the basic components, others are also necessary to build an operational machine, such as ventilators, pumps and controls. The simulation for design, or other purposes, of machines of this kind requires all components to be described by models with a similar degree of detail. When establishing mathematical models of real machine components for simulation by computer, it is necessary to make a certain number of assumptions that, though not affecting the ability of the models to reproduce the observable physical reality, may reduce their complexity to an acceptable level. It is naturally open to discussion what an acceptable complexity level means: On one hand there is the need to reproduce accurately the observable reality, to predict the equipment behaviour, and most important of all, to obtain information allowing the eventual improvement of the parts studied; On the other, it is necessary to compromise on the computation time, on the size of the computer program, and on the amount of data required. The fundamental assumptions made for the development of the models reported in the following are:

- Equilibrium thermodynamic relationships may be applied in general;
- The effects of the presence of lubricating oil dissolved in the refrigerant upon the various heat transfer processes may be neglected;
- The amount of refrigerant (charge) present in the machine is assumed to correspond exactly to that required at the operating conditions simulated. That is equivalent to say that the liquid receiver is so sized as to damp effectively the effects of the charge over the operation of the various components¹.

¹ In machines with control of the evaporator outlet superheating (thermostatic and electronic expansion valves) the condenser subcooling and pressure may be affected by the charge. In machines with constant flow section expansion devices (capillary or short tube, fixed orifice) both the condenser and evaporator operation are affected by the charge.
In this chapter, I describe the mathematical models of the components of a typical air-to-water heat pump and the algorithms to obtain the solutions to those models individually and when used together to simulate an operational machine. The general solution method proposed is not confined to the air-to-water type of machine, but obeys the criteria for steady state design models in general, as discussed in chapter 2.

**Fig. 3.1** - Schematic representing the basic components of a reverse Rankine cycle machine working with vapour compression, and its operating cycle on a T-s Diagram.
3.1 Compressor

Several types of compressors are used currently in vapour compression reverse Rankine cycle machines. The ranges of capacities\(^1\) covered by each type are schematically depicted in Fig. 3.2. Although the capacity is not the unique criterion of selection, it serves as an indicator of the most used types. In the range covered by the simulation model discussed in this work, the reciprocating is the most common type.

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Fig. 3.2 - Compressor types and ranges of refrigeration capacity, adapted (Jakobs 1989).

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\(^1\) Capacities refer to refrigeration capacity under defined operating conditions.
The state-of-the-art reciprocating compressor has been improved over the years, and stands such demanding applications as air-to-water heat pumps: large pressure ratios and wide range of operating conditions.

Because compressors are sophisticated engineered components, they are not purpose designed, at least for the residential and light commercial applications. Rather, a compressor is chosen from manufacturers' catalogs. This fact conditions the approaches that may be considered in the development of models with acceptable complexity. The compressor model described here applies essentially to reciprocating compressors, and is based on the information usually available from the manufacturers, and on readily measurable magnitudes.

The model developed avoids the complexities of those reported by Röttger (1975), Hiller and Glicksman (1976), and by Haberschill (1983), but does consider the internal transport phenomena in more detail than the compressor models used in the ORNL (Fischer and Rice 1981, 1983) and in the NIST (Domanski and Didion 1983) heat pump models.

3.1.1 Scope and Limitations

In its current state no capacity control method is considered; neither cylinder offset nor variable speed. It is certainly possible to apply it in the simulation of other types of positive displacement compressors, provided that the characteristic data are available in a suitable form. The application of this model to the simulation of variable capacity machines may require major changes.

The three constructive types of reciprocating compressors currently in the market - hermetic, semi-hermetic, and open - are simulated differently in respect to losses, but the basic model is the same. Limitations in the size of the machines this model is suitable for, result only from the absence of capabilities to simulate capacity control, which is mostly an attribute of large machines.
3.1.2 Theory and Assumptions

3.1.2.1 Ideal (Reversible) Processes

In simulating positive displacement compressors, the compression process may be assimilated to a continuous flow process, provided that either steady state conditions or large\(^1\) time steps are considered.

The first law of thermodynamics applied to such a process gives the work input necessary

\[
\omega_{in} = \Delta Ke + \Delta Pe + \int v dP
\]  

[3.1]

Assuming that the variations in the kinetic and potential energy terms, respectively \(\Delta Ke\) and \(\Delta Pe\), are negligible in relation to \(\int v dP\), the work input may be readily calculated once the process law is known. Vapour compression and expansion processes may be mathematically described by an equation of the form

\[
P v^\gamma = Cte.
\]  

[3.2]

where \(\gamma\) is the isentropic exponent of expansion for a reversible process, or the corresponding polytropic exponent in irreversible cases.

For an ideal gas, \(\gamma\) equals the ratio of the specific thermal capacities at constant pressure and volume, \(\gamma = Cp / Cv\). For real gases \(\gamma\) is given by

\[
\gamma = -\frac{v}{P} \left( \frac{\partial v}{\partial P} \right)_s = -\frac{v}{P} \frac{Cp}{Cv} \left( \frac{\partial P}{\partial v} \right)_T
\]  

[3.3]

The specific work input between states 1 and 2 is

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\(^1\) Large enough in relation to the duration of the complete compression cycle.
3.1.2.2 Real Processes

3.1.2.2.1 Qualitative Analysis

Real compression processes are affected by irreversibilities due to the use of a real gas or vapour, and by the limits imposed by existing compressors. Real compressors impose pressure drops across the cylinder intake and exhaust ports, and heat interactions of the incoming gas or vapour with the piston, cylinder walls and valve assemblies. Actual pistons and crank mechanisms are not frictionless, and inertia and valve behaviour do not allow for complete filling of the cylinder with fresh suction vapour. Furthermore, allowances necessary in the manufacturing process and for safe operation let some compressed vapour re-expand in the cylinder before every new intake. Piston blow-by reduces further the net throughput per cycle, as high pressure vapour leaks past the piston rings to the low pressure side of the piston. Heat interactions with the intake vapour reduce both the isentropic and the volumetric efficiencies. These two figures of merit describe the deviations of the real compression process from the ideal one. Other parameters are required to account for the compressor's mechanical losses (friction) and for the drive's losses.

3.1.2.2.2 The Compression Process

Deviations from the ideal compression process are accounted for using the isentropic and volumetric efficiencies. The first accounts for deviations from the ideal work done on the refrigerant, while the second accounts for deviations in the mass of refrigerant delivered by the process in relation to the ideal case.

\[
\omega_{in} = \int_{1}^{2} v dP = P_1 v_1 \frac{\gamma}{\gamma - 1} \left[ \left( \frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]
\]
The isentropic efficiency $\eta_s$ is the ratio of compression work theoretically necessary to that actually done on the refrigerant during the compression. Mathematically,

$$\eta_s = \frac{h_{dp,s} - h_{sp}}{h_{dp} - h_{sp}} \quad [3.5]$$

The volumetric efficiency $\eta_v$ is the ratio of the volume of refrigerant intaken, at the suction port conditions, to the volume swept by the piston(s) in one stroke. Mathematically,

$$\eta_v = 1 - \varepsilon \left( \Psi^{1/\gamma} - 1 \right) \quad [3.6]$$

where $\Psi$ is the compression ratio, $\Psi \equiv P_d / P_s$.

The geometric clearance volume ratio $\varepsilon$ is not accurately known because the clearance volume in the cylinder is not available in the manufacturers' data, and its experimental measurement is difficult. On the other hand it is possible to determine from catalog data an effective clearance volume ratio, $\varepsilon_{eff}$. This effective clearance volume ratio is determined using the theoretical expression of the volumetric efficiency with the effective volumetric efficiency:

$$\varepsilon_{eff} = \frac{1 - \eta_{v,eff}}{\Psi^{1/\gamma} - 1} \quad [3.7]$$

The effective volumetric efficiency $\eta_{v,eff}$ is less than the theoretical value, due to valve leakage and piston blow-by. The relationship between effective and theoretical volumetric efficiencies is not readily obtainable, even from catalog data. The flow rate of refrigerant through a real compressor is then calculated as

$$\dot{m}_r = \eta_{v,eff} \frac{N}{60} \frac{D}{v_{sp}} \quad [3.8]$$
The adiabatic compression work done on the refrigerant is

\[ W = \dot{m}_r (h_{dp} - h_{sp}) \]  \[ 3.9 \]

The suction port conditions \( h_{sp} \) are easily estimated, and the only discharge port parameter known is the discharge pressure \( P_{dp} \).

The isentropic efficiency \( \eta_s \) is derived from catalog data for the actual operating pressures (suction and discharge), as the compression work may also be calculated from

\[ W = \dot{m}_r \frac{h_{dp,s} - h_{sp}}{\eta_s} \]  \[ 3.10 \]

Finally, the power required to drive the compressor is

\[ \dot{W}_d = \frac{\dot{W}}{\eta_{mech,comp} \times \eta_d} \]  \[ 3.11 \]

In the case of an electric motor drive, the drive efficiency is the product of three factors:

\[ \eta_d = \eta_{e,mot} \times \eta_{mech,mot} \times \eta_{coup} \]  \[ 3.12 \]

The transmission efficiency \( \eta_{coup} \) is unity for direct coupling (hermetic and semi-hermetic construction types).

### 3.1.2.3 Assumptions

The main assumptions underlying the application of this model to reciprocating compressors, in steady state, are:

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1 The heat transfer between the cylinder wall and the refrigerant both inside and outside of the cylinder are negligible (Röttger 1975, Ney 1990).
Geometric and mechanical relationships are kept independent of the particular working point.

The mechanical efficiencies of the compressor and drive are independent of the operating conditions, for individual rotating speeds.

The isentropic efficiency is mostly dependent upon the compression ratio, other influences being negligible, e.g. the level of suction superheating.

The compressor rotation speed may either be considered constant at the nominal speed value, or dependent upon the compression ratio $\psi$, when the variation law is known.

Discharge and suction conditions are considered at the compressor boundary, i.e. immediately after the discharge port and immediately before the suction port, in the direction of the refrigerant flow.

In hermetic and in the semi-hermetic construction types the driving motor is cooled by the suction vapour.

The drive power and refrigeration capacity (rate) at catalog conditions are described by equations of the form

$$W(\dot{R}) = A + B T_e^* + C T_c^* + D T_e^* T_c^* + E T_e^{*2} + F T_c^{*2}$$  \[3.13\]

The refrigerant at compressor inlet is supposed to have some degrees of superheating.
3.1.3 The Compresor Model

3.1.3.1 Derivation of the Characteristic Parameters from Manufacturers' Data

Based on the foregoing theory and assumptions, the data available from the compresor manufacturers to the heat pump designer are used to obtain the characteristic parameters of the compressor simulation model. The calorimetric data supplied by the manufacturers usually includes refrigeration capacity and drive power at given conditions of subcooling and superheating of the refrigerant.

For hermetic and semi-hermetic construction types, the characteristics of the drive are supplied as well, i.e. what appears in the data sheets is the overall drive power. For the open construction type instead, the shaft power is known, but the drive and transmission characteristics are to be obtained separately.

The characteristic parameters of the compression process, isentropic and volumetric efficiencies, are derived by the same method for the three construction types. The thermal losses and refrigerant state at the suction port are calculated by different methods as explained in the following.

3.1.3.1.1 Hermetic Construction Type

In this type of construction the driving motor, compressor, and discharge manifold are inside the same welded shell. The suction vapour comes in contact with all three and with the lubricating oil in the oil sump, before admission into the cylinder. Heat interactions with the discharge manifold, oil surface and external surfaces of the driving motor and compressor body add some degrees of superheating to the suction vapour. In most cases the vapour is sucked through the motor windings, and along the cylinder wall in order to cool them, picking up some more degrees of superheating in the process. Finally, passing through the
valve assembly again some important heat interactions occur. This model considers only the refrigerant state outside of the valve assembly. As the temperature of the suction vapour increases due to the heat interactions mentioned, the intensity of these interactions has to be determined in order to establish the suction vapour state at the suction port. A feedback loop consisting of four parts may be recognized;

- Gain from the discharge manifold
- Gain from the motor body
- Gain from the compressor body
- Gain from the oil sump.

**Fig. 3.3** - Schematic representation of a hermetic compressor.
Fig. 3.4 - Flow diagram describing the algorithm to determine the characteristic parameters of the compression process in hermetic compressors.
The existence of this feedback loop imposes an iterative solution to determine the refrigerant state at the suction port. Fig. 3.4 shows a flow diagram describing the corresponding algorithm. The model for the electrical and mechanical losses from induction motors is discussed in Appendix A, and that for the mechanical losses of the compressor is presented in Appendix B.

In the hermetic case all energy losses from the motor, the compressor, and the discharge manifold result in heat interactions with the suction vapour in the shell. The suction vapour interacts with the shell wall, and through this with the environment. Direct losses to the environment through metallic parts (supports, etc.) are assumed negligible. The suction vapour in the shell comes into contact with the motor and the compressor, and with the discharge manifold and the lubricating oil. All these parts are at temperatures higher than the incoming vapour, so the contact results in an increase of the suction vapour temperature. The assumption of steady state implies no variation of the oil-refrigerant mixture composition, thus only heat interactions are considered. Given the complex geometry of the various surfaces, excepting that of the oil, an exact treatment of these heat interactions is very difficult. The importance of their effects upon the compressor operation requires a simplified, though physically meaningful treatment. A two prong solution is adopted: On one hand the losses from the motor and compressor can only be taken away by the refrigerant, and on the other convective transport equations have to be applied to calculate the exchanges with the lubricating oil and the discharge manifold. The electrical analogy diagram depicted in Fig. 3.5 shows the paths for energy exchange in a hermetic compressor (Conde and Berghmans 1983). Although it is already a simplification of the energy exchange paths, the diagram is still complex so further simplifications are in order. As said above, losses through metallic contact are neglected. This is equivalent to set $R_{MSW} = \infty$ in the diagram. On the other hand, all the losses of the motor and compressor are taken away by the suction vapour, which is equivalent to set $R_{MO} = \infty$ and $R_{MS} = 0$. To further simplify the numerical treatment, the losses from the shell to environment are calculated in
two parts. One corresponds to the shell surface in contact with the lubricating oil, and the other to that part in contact with the suction vapour. Thus $R_{SWE}$ is made of two parts, and no heat conduction is considered between the two nodes so obtained.

$R_{DO}$ - Discharge to Oil  
$R_{DS}$ - Discharge to Shell Refrigerant  
$R_{MO}$ - Motor/Compressor to Oil  
$R_{MS}$ - Motor/Compressor to Shell Refrigerant  
$R_{MSW}$ - Motor/Compressor to Shell Wall

$R_{OS}$ - Oil to Shell Refrigerant  
$R_{OSW}$ - Oil to Shell Wall  
$R_{SSW}$ - Shell Refrigerant to Shell Wall  
$R_{SWE}$ - Shell Wall to Environment

Fig. 3.5 - Electrical analogy schematic of the energy exchange paths in a hermetic compressor (D - Discharge; S - Suction; M - Motor/Compressor; O - Lubricating oil; SW - Shell wall; E - Environment).

The areas of these surfaces are calculated from the lubricating oil volume and the external geometry of the shell. The shell is assumed a cylinder of revolution for
this purpose. In cases where this is not so, an equivalent shell diameter is calculated when generating the compressor data file.

The calculation method for the energy exchanges between the discharge manifold and the lubricating oil with the suction vapour is explained in the Appendix C.

The heat interaction of the shell with the environment takes place by radiation and free convection. A heat transfer coefficient, derived from measurements, takes the value of 2.0 W/m² K, and is assumed constant for the whole working range, in order to simplify the calculations. The losses corresponding to the oil part are proportional to the temperature difference between the discharge vapour and the environment. Those corresponding to the rest of the shell are proportional to the temperature difference between the suction vapour and the environment, and may be either positive or negative.

3.1.3.1.2 Semi-Hermetic Construction Type

![Fig. 3.6 - Schematic representation of a semi-hermetic compressor.](image)

1 - Driving Motor
2 - Compressor
3 - Oil Sump
4 - Shell Wall
Contrary to the hermetic, this type of construction permits the access to the various parts of the compressor from outside. The discharge vapour gets directly into the discharge line without any further heat interaction with the suction vapour besides that taking place at the cylinder port assembly. The driving motor is cooled down by the suction vapour. Thus it is necessary to account for the heat interaction of the suction vapour with the driving motor and of the motor's shell with the environment. Again the calculation of this heat interaction requires an iterative solution. The iteration is done until two successive values of the losses to the environment agree within an acceptable limit. When the convergence is attained, the parameters volumetric and isentropic efficiencies, and the specific volume of the refrigerant vapour at the suction port are calculated. The flow diagram depicted in Fig. 3.7 illustrates the corresponding algorithm. The model for the electrical and mechanical losses of the driving motor is described in Appendix A, and that for the compressor in Appendix B. Refer to the previous section for the losses to the environment.
Fig. 3.7 - Flow diagram describing the algorithm to determine the compression parameters of a semi-hermetic compressor.
3.1.3.1.3 Open Construction Type

In this type of construction the drive is completely independent of the compressor itself. No heat interactions take place between the suction vapour and any part of the compressor or drive beyond those at the valve assembly. As consequence the determination of the parameters for the simulation model is much simpler in this case. Only the compressor losses must be added to the net work done on the refrigerant. The derivation of the volumetric and isentropic efficiencies, and of the suction vapour state at the suction port are straightforward.

Fig. 3.8 - Schematic representation of an open compressor.
3.1.3.2 Modeling Actual Performance

3.1.3.2.1 Hermetic Construction Type

The characteristic parameters of a compressor derived from catalog data are now applied to predict the values of the operating variables. Those parameters are derived for the same operating pressures at the catalog conditions (suction line superheating). At the actual working conditions the vapour state at the suction port is certainly different from that specified by the catalog. The isentropic efficiency is assumed independent of variations in the vapour superheating at the suction port. Its value is then taken equal to that derived from the catalog data. The volumetric efficiency, on the other hand, is strongly affected by the specific volume of the refrigerant vapour at intake. Hence, this parameter is corrected to account for the actual refrigerant vapour state at the suction port.

The volumetric efficiency may be expressed as

\[ \eta_v = 60 \frac{m_r}{D N} v_{sp} \]  

[3.14]

All other conditions being constant, the theory shows that the rate of mass flow is inversely proportional to the specific volume at the suction port

\[ \frac{m_r}{m_{r,cat}} \propto \frac{v_{sp,cat}}{v_{sp}} \]

Dabiri and Rice (1981) observed that the constant of proportionality is not unity. This may be explained by the fact that the valves are pressure actuated: Different specific volumes generate different flow velocities through the port, and consequently different drag forces, thus changing the valve opening and eventually increasing the pressure drop across. Dabiri and Rice, op. cit., proposed the following relationship between the flow rates and the specific
volumes at the suction port,

\[ \frac{\dot{m}_r}{\dot{m}_{r,cat}} = 1 + F \left( \frac{v_{sp,cat}}{v_{sp}} - 1 \right) \]  \[3.15\]

Applying this to the definition of volumetric efficiency, it results

\[ \frac{\eta_v}{\eta_{v,cat}} = F + v_r(1 - F) \]  \[3.16\]

with \( v_r = \frac{v_{sp}}{v_{sp,cat}} \)

The value of 0.75 suggested by those authors for \( F \) is maintained, although it might have to be modified to account for different compressor designs (e.g. different valve stiffness). A plot of this equation is depicted in Fig. 3.9.

![Graph](image)

**Fig. 3.9** - Relationship between specific volume ratio and volumetric efficiency ratio.
Fig. 3.10 shows a flow diagram of the algorithm devised for simulation of this type of compressor. In the simulation, first all heat interactions are initialized to null (this corresponds to assuming that the suction vapour goes directly from the suction line into the cylinder). The first iteration gives the first discharge conditions, from which the heat interactions are evaluated, the flow rate calculated, etc. The iteration proceeds until two successive values of the flow rate differ by less than 1%. Out of these calculations result the values for flow rate, drive power and temperature of the discharge vapour.

3.1.3.2.2 Semi-Hermetic Construction Type

As described in the previous section for the hermetic construction type, the parameters

- Volumetric efficiency
- Isentropic efficiency
- Specific volume of the vapour at the suction port

are now applied to predict the actual performance of the compressor. Corrections to the volumetric efficiency previously derived, apply in this case as well. A flow diagram for the simulation algorithm of this case is depicted in Fig. 3.11.

3.1.3.2.3 Open Construction Type

The procedure to calculate the actual performance of an open type compressor is similar to that used for the previous two types. However, as no heat interaction with the suction vapour takes place, the simulation is simplified accordingly. A flow diagram for the simulation algorithm to this case is shown in Fig. 3.12.
Fig. 3.10 - Flow diagram describing the algorithm of the simulation model of the hermetic construction type.
Fig. 3.11 - Flow diagram describing the algorithm to the simulation model of the semi-hermetic case.
Fig. 3.12 - Flow diagram describing the algorithm to the simulation model of the open case.
3.1.4 Refrigerant Mass Balance Inside a Compressor

The mass of refrigerant held inside a compressor may be a significant part of the total refrigerant charge, especially in the case of hermetic compressors in small machines. For a correct account of the refrigerant distribution among the various components, an estimate of the refrigerant mass held in the compressor is required. Refrigerant is held in the compressor in three main places:

- As suction vapour in the free volume of the shell
- As discharge vapour in the discharge manifold
- As liquid in solution in the lubricating oil.

The refrigerant mass held in the vapour phase is easy to estimate once the holding volumes are known. This is theoretically simple, although in practice data on the holding volumes are many times difficult to gather.

The refrigerant mass held in solution with the lubricating oil may be approximately calculated when the equilibrium properties of the solution are known. The concentration of the refrigerant in solution depends upon the vapour pressure of the refrigerant onto the free surface of the oil, and upon the solution temperature. The refrigerant-oil solutions may have one or more equilibrium compositions at given conditions. In the cases where more than one equilibrium concentration is possible (e.g. HCFC22 with most oils) it is necessary to decide which will be the dominant concentration. Considering that the amount of refrigerant available is much larger than that that may enter into solution, it seems reasonable to assume that the higher concentration will be dominant. Fig. 3.13 shows a plot of equilibrium concentrations of CFC12-mineral oil solutions, as function of temperature and pressure (Bambach 1955), whereas Fig. 3.14 depicts an example of HCFC22-oil solutions. In this case the miscibility limit curve separates the region where two and only one equilibrium concentrations are possible (Conde 1990).
The mass of refrigerant in the vapour phase is calculated as

\[ M_{vapor} = V_{free} \times \rho_v \]  

[3.17]

and that held in the lubricating oil is

\[ M_{in \ oil} = \frac{\xi}{1-\xi} V_{oil} \times \rho_{oil}(T_{oil}) \]  

[3.18]

with

Fig. 3.13 - Equilibrium concentration of CFC12 solutions in mineral oil (Bambach 1955).
\[ \rho_{oil}(T_{oil}) = \rho_{oil}(T_0) - 0.58 \times (T_{oil} - T_0) \]  

(3.19)

\( T_0 \) is a temperature for which the oil density is known (ASHRAE 1986).

Fig. 3.14 - Equilibrium concentration of HCFC22-lubricating oil solutions (Conde 1990).
Fig. 3.15 - Flow diagram describing the algorithm for calculation of the mass of refrigerant held in the compressor.
3.2 Heat Exchangers - Condenser and Evaporator

3.2.1 Generalities

Both the condenser and the evaporator in vapour compression heat pumps or refrigeration machines, exchange heat between refrigerant, mostly in two-phase flow, and a secondary fluid, usually air, water or a brine. For the types and capacities of the condensers and evaporators covered by this model, refrigerant condensation and vaporization take place in channels of either circular or annular geometry. Hence, the description of two-phase flow mechanisms and transport processes is confined to these geometries.

The general forms of the conservation equations for such processes are presented in this introductory section, and will be applied later to the particular cases of evaporator and condenser considered. Conservation equations for the air and water (or brine) sides will be discussed in the evaporator and condenser sections, respectively.

The Conservation Equations on the Refrigerant Side

The conservation equations for one dimensional steady state two-phase flows, related to phase $k$, are

*Continuity Equation*

\[
\frac{1}{A} \frac{d}{dz} \left[ \rho_k \langle \varepsilon_k \rangle \langle u_k \rangle_k A \right] = \Gamma \tag{3.20}
\]
Conservation of Linear Momentum

\[
\frac{1}{A} \frac{d}{dz} \left[ \rho_k \langle \varepsilon_k \rangle \langle u_k \rangle_k \right] = 
\]
\[
- \langle \varepsilon_k \rangle \frac{dP}{dz} + g \rho_k \langle \varepsilon_k \rangle \cos \Theta 
\]
\[
- \frac{P_{w,k} \tau_{w,k}}{A} + \frac{P_i \tau_i}{A} + u_{i,k} \Gamma 
\]  

Energy Equation

\[
\frac{1}{A} \frac{d}{dz} \left[ \rho_k \langle \varepsilon_k \rangle \langle h_0 \rangle_k \langle u_k \rangle_k \right] = 
\]
\[
\dot{q''}_{w,k} \langle \varepsilon_k \rangle + \frac{\dot{q'}_{i,k}}{A} \frac{P_i}{A} + \frac{\dot{q'}_{w,k}}{A} \frac{P_{h,k}}{A} + \Gamma h_0 \dot{\varepsilon}_{i,k} + \zeta \frac{P_i}{A} \tau_i u_{i,k} 
\]

This results in six conservation equations to be satisfied simultaneously. Besides requiring complex boundary conditions, such a model is computationally too heavy for use in the simulation of components such as heat exchangers that are part of more complex devices. Further simplifications are required.

The simplification of this model carries with it two main consequences; Information is lost in what regards phase interactions, and must again be supplied in the form of empirical correlations. If, further, the phase equations are added, the interface terms cancel out. There are then three conservation equations left where the closure laws needed are those for wall shear, wall heat flux, and a relationship between quality and void fraction. It must as well be assumed that both phases are saturated at the local pressure, which is itself assumed uniform over the whole cross section.

Setting \( \varepsilon = \langle \varepsilon_{vapour} \rangle \), the resulting equations (mixture model) become
Continuity equation

\[ \frac{d}{dz}[\varepsilon \rho_G u_G + (1 - \varepsilon) \rho_L u_L] = 0 \quad [3.20'] \]

Conservation of linear momentum

\[ \frac{d}{dz}[(1 - \varepsilon) \rho_L u_L^2 + \varepsilon \rho_G u_G^2] = \]
\[ - \frac{dP}{dz} - g \cos \theta [\rho_L (1 - \varepsilon) + \rho_G \varepsilon] \]
\[ - \frac{1}{A} (P_{w,L} \tau_{w,L} + P_{w,G} \tau_{w,G}) \quad [3.21'] \]

Energy equation

Assuming no internal heat generation, and that the kinetic and potential parts in \( h_k^o \) are negligible in face of \( h_k \), and neglecting the interface dissipation as well, the energy equation reduces to

\[ \frac{d}{dz} [\rho_L (1 - \varepsilon) h_L u_L + \rho_G \varepsilon h_G u_G] = \frac{\dot{q}_w}{A} P_w \]
\[ \dot{q}_w = \alpha_r (T_w - T_r) \quad [3.22'] \]

The method of solution adopted is by finite differences, with the boundary conditions defined by the inlet or outlet property values, depending on the direction of the integration scheme in relation to the flow of refrigerant.
The finite difference form of the conservation equations is

**Continuity**

\[
\Delta \left[ \dot{M}_G + \dot{M}_L \right] = 0 \tag{3.20*}
\]

**Conservation of linear momentum**

\[
\Delta P = -\frac{1}{A} \Delta \left[ \dot{M}_L \ u_L + \dot{M}_G \ u_G \right] \\
- g \cos \theta \left( \rho_L (1 - \varepsilon) + \rho_G \varepsilon \right) \Delta z \\
- (F_{w,L} + F_{w,G}) \Delta z \tag{3.21*}
\]

The first term on the RHS is the acceleration term, that may be worked out to give

\[
\Delta P_{acc} = \dot{m}^2 \Delta \left[ \frac{x^2}{\varepsilon \rho_G} + \frac{(1 - x)^2}{(1 - \varepsilon)\rho_L} \right] \tag{3.23}
\]

The second term on the RHS is the gravity effect, which is assumed negligible for the typical layout of these components, and the last term represents the friction losses. As has become common in two-phase flow calculations, the friction pressure losses are calculated using a two-phase friction multiplier to the friction pressure loss of one of the phases, assuming the whole flow in that phase.

\[
\Delta P_{fricc} = \Psi (\Delta P)_{LO} \tag{3.24}
\]
Energy

\[ \dot{M} \Delta[(x - x)h_L + xh_G] = \alpha_R(\bar{T}_w - \bar{T}_R) P_w \Delta z \]  

[3.22]

The conservation of linear momentum and the energy equations are coupled, not only through the refrigerant properties, but through the mutual dependence among the vapour quality \( x \), the void fraction \( \varepsilon \), and the two-phase friction multiplier \( \Psi \).

The iterative solution to the energy equation is found using an electrical analogy for heat transfer, with the temperature of the separating wall as iteration variable (voltage divider), Fig. 3.16. The thermal resistance of the tube wall is usually more than one order of magnitude smaller than the convective resistances \( \Sigma R_S \) and \( \Sigma R_R \).

Fig. 3.16 - Thermal resistances and heat flux across a wall.

Closure laws are required relating vapour quality and void fraction, two and single phase friction coefficients and for the heat transfer coefficients. They involve assumptions regarding the two-phase flow patterns, nature of the channel surfaces, etc.

The Inventory of Refrigerant Mass in the Heat Exchangers

The amount of refrigerant residing in the heat exchangers is the integral of the volumetric distribution of the phases weighted with the respective densities.
3.2.2 Condenser Model

The condenser type considered in this investigation, is a coiled-coaxial type with condensation in the annulus and the cooling, or secondary, fluid (water or brine) flowing in the inner tube in countercurrent. This geometry is widely used for condensing powers up to 50 kW, and offers the advantage of a low ratio of volume to heat transfer area. The outer tube - shell - is usually a seamless copper (or copper alloy) tube internally smooth. The inner tube - core - is usually

\[
M = A \int_{0}^{Z} \left[ \rho_L (1 - \varepsilon) + \rho_G \varepsilon \right] \, dz
\]  

**Typical Geometric Data**

- \( \phi_0 \): Shell Tube Inner Diameter (31 mm)
- \( \phi_t \): Fin Top Diameter (22 mm)
- \( \phi_b \): Fin Base Diameter (19 mm)
- \( \phi_i \): Core Tube Inner Diameter (17 mm)
- \( p_t \): Fin Pitch (1024 Fins/metre)

**Fig. 3.17 - Arrangement of the coaxial tubes.**
finned on the outside with low integral fins, fluted inside due to the manufacturing process - rolling. This ensemble is coiled helically, with the core tube(s) maintained in place by means of spacers. The schematic in Fig. 3.17 shows this geometry, and the geometric parameters required by the simulation model.

3.2.2.1 Heat Transfer Coefficient - Refrigerant Side

The condensation of pure vapours on the outside of tubes has been studied by several authors since the basic work of Nusselt. Most authors considered simpler geometries. Beatty and Katz (1948), studied the condensation of pure vapour on low integral finned tube surfaces in the horizontal position. In their model, these authors assume that the condensate is evacuated from the finned surfaces by gravity, and that there are no shear forces exerted by the vapour upon the condensate. They considered no condensate subcooling as well. Their solution consists of the superposition of two parts, one considering the condensation on the horizontal tube surface, using the Nusselt solution for laminar film condensation onto horizontal tubes, and the other considering the condensation on a vertical plate, using the Nusselt solution for this geometry.

\[ \alpha_o \eta = \alpha_h \frac{A_r}{A} + \alpha_f \eta_f \frac{A_f}{A} \]  \[3.26\]

The Nusselt solutions for both these cases are, respectively,

\[ \alpha_h = 0.725 \left[ \frac{\lambda_L^3 \rho_L (\rho_L - \rho_G) g h_{fg}}{\mu_L \Delta T_{vs} D} \right]^{1/4} \]  \[3.27\]

\[ \alpha_f = 0.943 \left[ \frac{\lambda_L^3 \rho_L (\rho_L - \rho_G) g h_{fg}}{\mu_L \Delta T_{vs} L_f} \right]^{1/4} \]  \[3.28\]
The characteristic length for the horizontal tube is the tube diameter $D$. That for the fin surface is defined by Beatty and Katz as the ratio of the area of one side of the fin to the fin tip diameter, eq. [3.30].

$$L_f = \frac{\pi (D_o^2 - D_r^2)}{4 D_o}$$  \[3.29\]

On this basis, the resulting equation for $\alpha_o$ is

$$\alpha_o \eta = 0.725 \left[ \frac{\lambda L^3 L (\rho L - \rho G) g h_{lg}}{\mu L \Delta T_{vs}} \right]^{1/4} \times \left[ \frac{A_r}{A} D^{-1/4} + 1.3 \frac{\eta_f A_f}{A} L_f^{-1/4} \right]$$ \[3.30\]

The surface and fin efficiencies, $\eta$ and $\eta_f$, respectively, may be considered unity for low integral fins.

Several improvements were made to the Beatty and Katz model, considering namely that the resulting condensate is not saturated and that some of the condensate may be retained between the fins, thus reducing the surface where condensation may take place. In his first solution of the condensation problem Nusselt (1916) considered already that the condensate might be subcooled, but suggested no general way to take this into account. Rohsenow (1956) analyzed this problem and suggested a modification to the Nusselt solution that accounts for condensate subcooling. This is done by correcting the enthalpy of condensation to include the subcooling part. Rohsenow's correction is done using the Jacob number, as

$$h_{lg} = 1 + C Ja$$ \[3.31\]
where $C$ was given the value $3/8$. The Jacob number is defined as the ratio of the sensible to the condensation enthalpy

$$J_a = \frac{C_p f \Delta T}{h_{fg}} \quad [3.32]$$

Sadasivan and Lienhard (1987) improved Rohsenow's modification establishing a relationship between the constant $C$ and the Prandtl number of the condensate. They found that $C$ is expressed as function of the Prandtl number of the condensate as

$$C = 0.68 - 0.23 \Pr_L^{-1} \quad [3.33]$$

With these modifications, the term $h_{fg}$ in the Beatty and Katz equation, Eq. [3.31], should be replaced by $h'_{fg}$ calculated as

$$h_{fg} = h_{fg} \left[ 1 + J_a \left( 0.68 - 0.23 \Pr_L^{-1} \right) \right] \quad [3.34]$$

Webb and co-workers (Rudy and Webb 1985, and Webb, Rudy and Kedzierski 1985) studied the problem of condensate retention between the fins. They found that the Beatty and Katz's assumptions of no retention could not be realized even for fluids of low surface tension. In their study it is shown that a fraction of the tube surface is always flooded with condensate, thus contributing little to the condensation process. They modified the Beatty and Katz model to take into account this observation (B-K-W model)

$$\alpha_0 \eta = (1 - C_b) \left( \alpha_h \frac{A_r}{A} + \alpha_f \frac{A_f}{A} \right) + C_b \alpha_B \quad [3.35]$$

where $C_b$ is the fraction of the circumference that is flooded with the condensate.
According to those authors, the last term \( C_b \alpha_b \) represents less than 2% of \( \alpha_o \eta \), and may be neglected for engineering calculations.

The flooded fraction of the surface is a function of the geometry and of the surface tension of the condensate (Webb and Rudy 1985).

\[
C_b = \frac{1}{\pi} \cos^{-1} \left[ 1 - \frac{2 \sigma (p - t_b)}{\rho_L g D_o (p_f e - A_p)} \right] \tag{3.36}
\]

---

**Fig. 3.18** - Comparison of measured and calculated average condensation heat transfer coefficients.
A comparison of $\alpha_0$ determined by this model - B-K-W - to $\alpha_0$ obtained from measurements shows important deviations, Fig. 3.18. This may be attributed to

- the differences in geometry (the tube is coiled).
- the flow is normal to the fins. This gives rise to shear stress effects of the vapour upon the condensate.
- the presence of spacers in the annulus, to keep the core tube in position, which disrupt the flow of the condensate and increase its turbulence.

The plot of the ratio of $\alpha_0$ measured to $\alpha_0$ calculated from this model against the Reynolds number of the vapour at inlet, Fig. 3.19, shows that a correlation exists between both. This correlation may be established in two parts; a lower part, for $Re < 3 \times 10^5$ and an upper part for $Re > 3 \times 10^5$. This Reynolds number takes into account the geometry of the coil through the hydraulic diameter, Eq. [3.38].

$$D_H = (D_{i,o} - D_o) \left[ 1 + \left( \frac{h}{\pi D_{coil}} \right)^2 \right]$$

[3.37]

Using a Reynolds number based factor, $f(Re_{coil})$ permits the reproduction of the measurements within ±10% of the calculated value for $\alpha_0$, Fig. 3.19. This correction factor introduces the effects of the geometry ($D_H$ in the Reynolds number) and of the shear stress of the vapour upon the condensate. A similar method to account for the effects of the shear stress over the condensate has been suggested by Vierow and Schrock (1991). It remains unclear, however, which is the relative importance of the effect of spacers in the annulus. With measurements on a single geometry this last aspect could not be clarified.
Fig. 3.19 - Correlation of the ratio of condensation heat transfer coefficient with the Reynolds number of the vapour flow at condenser inlet.

The general form of the equation for \( \alpha_0 \) becomes then

\[
\alpha_0 \eta = f(Re_{coil}) \left( 1 - C_b \right) \left[ \frac{\lambda_L^3 \rho_L (\rho_L - \rho_g) g \dot{h}_g}{\mu_L (T_{sat} - T_{wall})} \right]^{1/4} \times D_{eq} \tag{3.38}
\]

with

\[
D_{eq} = \left[ \frac{A_f}{A} D_r^{-1/4} + 1.3 \eta_f \frac{A_f}{A} L_f^{-1/4} \right] \tag{3.39}
\]
and

\[ f(Re_{\text{coil}}) = A_1 + A_2 Re_{\text{coil}} \]  \[3.40\]

with

\[ A_1 = -0.71 \]
\[ A_2 = 1.30 \times 10^{-5} \quad \text{for } Re_{\text{coil}} \leq 3.0 \times 10^5 \]

and

\[ A_1 = -9.20 \]
\[ A_2 = 4.13 \times 10^{-5} \quad \text{for } Re_{\text{coil}} > 3.0 \times 10^5 \]

In the regions with single phase flow, before condensation starts and after complete condensation, the heat transfer coefficient is calculated as for the secondary fluid, with the difference that here the channel has an annular geometry.

3.2.2.2 Heat Transfer Coefficient - Secondary Fluid

The heat transfer coefficient for the secondary fluid side is calculated on the basis of the Chilton - Colburn analogy between heat and momentum interactions. This analogy is expressed mathematically as

\[ j = St \ Pr^{2/3} = \frac{f_c}{8} \]  \[3.41\]

A number of studies dealing with the friction factor in helically coiled tubes have been published, although only smooth surfaces have been considered (White 1929, Ito 1959, Mishra and Gupta 1978, Gnielinski 1983). The solutions proposed may be described as

\[ f_{\text{coiled tube}} = (\text{straight tube term} + \text{curvature effects}) \]
where the straight tube term for smooth surfaces is calculated with the Blasius' equation.

\[ f_s = 0.3164 \, Re^{-1/4} \] \[3.42\]

By analogy, for nonsmooth surfaces, the straight tube term is replaced by a friction factor determined when considering the surface roughness \( \varepsilon \). In this case the Colebrook (1939) equation was selected and inserted in the solution proposed by Mishra and Gupta (1978). Their solution is presented in the following with the modification for rough surfaces. Secondary flows apparently retard the transition to turbulent flow. These secondary flows are affected by the radius of curvature of the coiled tube. The curvature diameter of a helically coiled tube may be approximated as

\[ D_{curv} = D_{coil} \left( 1 + \left( \frac{h}{\pi D_{coil}} \right)^2 \right) \] \[3.43\]

An equation due to Schmidt (1966) gives the transition Reynolds number for coiled tubes as

\[ Re_{cr} = 2300 \left[ 1 + 8.6 \left( \frac{d}{D_{curv}} \right)^{0.45} \right] \] \[3.44\]

For laminar flow \( Re_{cr} > Re \), the friction factor is calculated as function of the Dean Number (Mishra and Gupta 1978)

\[ De = Re \sqrt{\frac{d}{D_{curv}}} \] \[3.45\]

as
\[ f_c = K \frac{64}{Re} \left[ 1 + 1.178E-3 \ (Ln(De))^4 \right] \]  

[3.46]

for \( De_{cr} > De > 1 \). The term \( K \) is a correction for nonisothermal flows:

\[ K = \left( \frac{\mu_{wall}}{\mu_{bulk}} \right)^{0.27} \]  

[3.47]

Fig. 3.20 - Comparison of two correlations for pressure drop using the best value of the internal rugosity of the core tube (\( e/d = 0.0065 \)).
For turbulent flow regimes, \( R_{e_c r} \leq R_e \), the friction factor is

\[
f_c = \left( f_s + 0.03 \sqrt{\frac{d}{D_{curv}}} \right) K
\]  

[3.48]

where \( f_s \) is obtained from the Colebrook equation

\[
f_s^{-1/2} = -2.0 \log \left( \frac{\varepsilon}{3.7 \, d} + \frac{2.51}{R_{e_d} \, f_s^{1/2}} \right)
\]  

[3.49]

Although this equation is implicit in \( f_s \), it is easily solved by the Newton-Raphson method, with the seed solution obtained from the Blasius equation for the same Reynolds number. The wall rugosity \( \varepsilon \), is obtained from catalog information on pressure drop using the same model. Fig. 3.20 shows approximations to the catalog data using two different correlations and the best value for the rugosity \( \varepsilon \), obtained by trial and error, \( \varepsilon/d = 0.0065 \).

A comparison of the values obtained for \( \alpha_i \) using this model, with those calculated from the measurements shows a tendency of the model to underpredict the measurements, Fig. 3.21. Furthermore the ratio of both values at the same point is well correlated to the Reynolds number of the secondary fluid flow in a coiled geometry.
Fig. 3.21 - Comparison of calculated and measured heat transfer coefficients inside the core tube.

A linear function of the Reynolds number may be used as correcting factor with excellent results, Fig. 3.22. As may be observed in this figure, the corrected model represents most the measured points within ±10%. The adjusted heat transfer coefficient is

$$\alpha_i = \frac{1}{8} \left( 0.711 + 1.288 \times 10^4 \ Re^{-1} \right) \rho \ u \ Cp \ f_c \ Pr^{-2/3} \quad [3.50]$$
3.2.2.3 Friction Pressure Losses - Refrigerant

The pressure loss due to friction during condensation is calculated from the single-phase pressure loss, using a two-phase friction multiplier $\Psi$. All correlations for $\Psi$, in the literature, were obtained for straight smooth channels. In this case, the channel is neither straight nor smooth. So, a comparison with measured values was made in order to find out which correlation would be the most appropriate, Fig. 3.23.
Fig. 3.23 - Comparison of predictions of two pressure drop correlations with experimental values.

The Baroczy-Chisholm (Baroczy 1966, Chisholm 1983) correlation gives the best approximation. Still the calculated values are about one order of magnitude lower than those obtained from measurements, Fig. 3.23. Thus an additional correcting factor is required, in the form

$$\psi_{\text{coil}} = f(Re_{G,in}) \cdot \psi_{Ch}$$  \hspace{1cm} [3.51]

The Baroczy-Chisholm two-phase friction multiplier is calculated with the Chisholm's B model (Chisholm 1983), as follows
The property index $\Gamma$ is defined as

$$\Gamma = \left[ \frac{\rho_L}{\rho_G} \left( \frac{\mu_L}{\mu_G} \right)^n \right]^{\frac{1}{2}} \quad [3.53]$$

$n$ is the exponent of the Reynolds number in the Blasius equation (0.25 for smooth surfaces). For nonsmooth surfaces $n$ is obtained as suggested by Chisholm (1983) from

$$\frac{f_{LO}}{f_{GO}} = \left( \frac{\mu_L}{\mu_G} \right)^n \quad [3.54]$$

The $B$ parameter is

$$B = \frac{C\Gamma - 2^{2-n} + 2}{\Gamma^2 - 1} \quad C = \frac{2000}{m} \left( \frac{\rho_L}{\rho_G} + \sqrt{\frac{\rho_G}{\rho_L}} \right) \quad [3.55]$$

The $D$ parameter is

$$D = (1-x)^{2-n} + \left( 2^{2-n} - 2 \right) x \left( 1-x \right)^{2-n} \left( 1-x \right)^{2-n} - 1 \quad [3.56]$$

The effect of the channel roughness on the parameter $B$ is calculated from

$$\frac{B_{rough}}{B_{smooth}} = \left\{ \begin{array}{ll} 0.5 & \left( 1 + \left( \frac{\mu_G}{\mu_L} \right)^2 - \frac{600 \varepsilon}{D_H} \right) \left( \frac{0.25 - n}{0.25} \right) \end{array} \right\} \quad [3.57]$$

The factor $f(Re_{G,in})$ accounts for the effects of the coiled and annular geometry and was obtained from measurements as
\[ f(Re_{G,in}) = 0.0398 \ Re_{G,in}^{0.4166} \iff \frac{Re_{G,in}}{100000} \leq 2.5 \]
\[ f(Re_{G,in}) = 19.6 \left( \frac{Re_{G,in}}{100000} \right)^{0.8} - 27.7 \iff \frac{Re_{G,in}}{100000} > 2.5 \]

### 3.2.2.4 Friction Pressure Losses - Secondary Fluid

The secondary fluid friction pressure loss coefficient is calculated with the Mishra and Gupta (1978) model for coiled tubes as explained in section 3.2.2.2. The pressure loss is given as

\[ \Delta P_S = f \frac{z \dot{m} S^2}{\rho S D} \]

### 3.2.2.5 Void Fraction during Condensation

The volumetric distribution of the phases during condensation in a coiled annulus has not been the subject of reports in the open literature to this date. Its knowledge, however approximate, is required to estimate the pressure loss due to acceleration of the flow, and to carry out the inventory of the refrigerant residing in the condenser under given operating conditions. A qualitative consideration of the flow in such a geometry leads to the assumption that the condensate entrainment, by the higher velocity vapour, is certainly important. An annular flow pattern, may eventually form onto the outer tube, due to centrifugal forces, and will coexist with the mist pattern of entrained droplets. These considerations lead to the choice of the Smith (1969) model for void fraction, which is based on the assumption that the two-phase mixture flows in thermodynamic equilibrium, i.e. the local vapour quality may be calculated from an energy
balance with an annular liquid part, and a central core of homogeneous mixture with the same velocity head as the liquid. The following expression for the slip ratio between the phases is derived from these assumptions.

\[
s = K + (1 - K) \left( \frac{\rho_L + K \frac{1 - x}{x}}{\rho_G + K \frac{1 - x}{x}} \right) \]  

[3.60]

where \(K\) is the entrainment factor, suggested by Smith to take the value 0.4. An interesting feature of this model is that for \(K=1\) (full entrainment), it defaults to homogeneous flow values, \(s=1\).

The void fraction is then

\[
\varepsilon = \frac{1}{1 + \frac{1 - x}{x} \frac{\rho_G}{\rho_L} s} \]  

[3.61]

3.2.2.6 Solution Method of the Condenser Model - Algorithm

The solution to the condenser model is obtained by a finite difference scheme for fractions of the length, following the refrigerant flow from inlet to the outlet. The length fractions have varying magnitudes. They are smaller at the inlet of the refrigerant (high temperature differences) and larger after the refrigerant vapour temperature attains saturation, or the vapour quality becomes smaller than 0.8. Fig. 3.24 depicts a general section \(j\) of the condenser, with the corresponding boundary conditions.
Fig. 3.24 - Schematic representation of the heat transfer process and boundary conditions at a condenser section.

The thermal energy transferred across the wall is

\[ \dot{Q}_j = \frac{\Delta T_{\text{log},j}}{RT_R + RT_S} \]  

[3.62]

The logarithmic mean temperature difference for the section \( j \) is

\[ \Delta T_{\text{log},j} = \frac{(T_{Gi,j} - T_{So,j}) - (T_{Go,j} - T_{Si,j})}{LN \left( \frac{T_{Gi,j} - T_{So,j}}{T_{Go,j} - T_{Si,j}} \right)} \]  

[3.63]
The thermal resistances are

$$R_{TR} = \frac{1}{\alpha_{R,j} A_{o,j}} + F_R \quad \quad \quad RTS = \frac{1}{\alpha_{S,j} A_{S,j}} + F_S \quad [3.64]$$

where $F_R$ and $F_S$ are the fouling resistances on the refrigerant and the secondary fluid side, respectively.

The energy balance for section $j$ is

$$\dot{M}_R \left\{ [x_{i,j} h_{Gi,j} + (1 - x_{i,j}) h_{Li,j}] - [x_{o,j} h_{Go,j} + (1 - x_{o,j}) h_{Lo,j}] \right\} = \dot{Q}_j \quad [3.65]$$

The following constraints apply to the calculation of the refrigerant enthalpy before the condensation is complete

$$h_L = H(P) - l_{fg}(P)$$
$$h_G = H(P, T_G) \quad \iff T_G > T_{sat}(P)$$
$$h_G = H(P) \quad \iff T_G = T_{sat}(P)$$

and after complete condensation

$$h_L = H(P_{sat}(T)) - l_{fg}(P_{sat}(T))$$
$$h_G = 0$$

The temperature $T_{Go,j}$ must satisfy the energy balance, and the conservation of linear momentum, which yields

$$P_{o,j} = P_{i,j} - \Delta P_{R,j} \quad \quad \Delta P_{R,j} = \Delta P_{fric,j} + \Delta P_{acc,j} \quad [3.66]$$

The vapour mass quality at outlet, $x_{o,j}$, results

$$x_{o,j} = \frac{x_{i,j} h_{Gi,j} + (1 - x_{i,j}) h_{Li,j} - h_{Lo,j} - \dot{Q}_j / \dot{M}_R}{h_{Go,j} - h_{Lo,j}} \quad [3.67]$$
The global solution for the section \( j \) is attained when the average wall temperature for the section \( \overline{T_{W,j}} \) converges to within 0.05 K for two consecutive iterations, Fig. 3.24.

\[
\overline{T_{W,j}} = \frac{RTS_j + RTR_j}{RTR_j RTS_j} \left[ \frac{T_{R,j}}{RTR_j} + \frac{T_{S,j}}{RTS_j} \right] \tag{3.68}
\]

Fig. 3.25 depicts a flow diagram illustrating the algorithm to solve the condenser model, while Fig. 3.26 shows the algorithm used to solve the heat exchanger problem for each length section.
Fig. 3.25 - Flow diagram illustrating the algorithm of the condenser model.
Fig. 3.26 - Flow diagram of the algorithm solving the heat exchanger problem in the condenser model.
3.2.3 Evaporator Model

The evaporator type considered in this work is the so-called dry expansion plate finned-tube coil. A plate-finned-tube heat exchanger is made up of one or more parallel rows of horizontal tubes arranged either inline\(^1\) or in a staggered pattern, with continuous thin plate (mostly aluminum) fins outside. The air (or other gas being handled) flows transversally to the tubes, i.e. parallel to the fins. The surface of the fins may be smooth or patterned. Several patterns are in use today. Sometimes the plates are just embossed with some type of relief. In this case the pattern is said 'wavy'. In other cases some parts of the fin plate are cut and deflected to another plane, 'louver fin', or slits are cut and offset from the fin plane, 'offset strip fin'. For special applications, the surfaces of the fin plate may be given special treatments to augment the heat transfer. The tubes are mostly internally smooth. However, the improvements obtained on the air side, now justify the use of internally enhanced tubes, e.g. spiral-grooved. Various types of internal inserts are used often too. The tube material is copper for halogenated refrigerants, steel for ammonia. The tubes are associated in several parallel circuits, which may be regarded as heat exchangers in themselves. The refrigerant is fed to them in approximately equal amounts by means of a distributor nozzle, a manifold, or capillary tubes. The objective of this partition into multiple parallel circuits is to reduce the refrigerant pressure drop to a reasonable value. The refrigerant flow rate through each circuit is naturally a function of the pressure gradient along the tubes that are part of it. As boiling is taking place inside the tubes, the pressure gradient along them is not constant. On the other hand, a different arrangement of the tubes (e.g. circuits with different number of tubes in a given row) or an uneven distribution of the air flow, lead to different pressure drops, thus to different flowrates. This justifies the separate simulation of different circuits in an enhanced simulation model. It is supposed that circuits with equal layouts may be simulated just once, assuming that the air flow is

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\(^1\) Not very common in evaporators, given the low heat transfer performance.
uniformly distributed across the frontal surface of the heat exchanger, which in practice is only an approximation. So, the basic element to be simulated by this model, is a section of finned tube with refrigerant flowing inside and air outside, in crossflow, Fig. 3.27.

**Fig. 3.27 - Element of the evaporator simulation model.**

The conservation equations for the refrigerant side are as described in section 3.2.1. Those for the air side are:
Continuity

\[ \frac{1}{P_w} \dot{M}_{da} \frac{dX_a}{dz} = \Gamma_{H_2O} \]  \text{[3.69]}

Momentum

\[ \Delta P_{elem} = \frac{\dot{m}_{a,\text{max}}}{2\rho_a} \left( \frac{S_L}{D_h} + K_i + K_o \right) \]  \text{[3.70]}

Energy

\[ \frac{1}{P_w} \dot{M}_{da} \frac{dh_a}{dz} + \Gamma_{H_2O} \left[ l_{H_2O} + C_{PH_2O} (T_{DP,a} - T_{surf}) \right] = \dot{q}''_{w,o} \]

\[ \dot{q}''_{w,o} = \frac{\alpha_a}{C_{P_a}} \eta_{surf} (h_a - h_{a,\text{sat,surf}}) \quad T_{surf} \leq T_{DP,a} \]  \text{[3.71]}

\[ \dot{q}''_{w,o} = \alpha_a \eta_{surf} (T_a - T_{surf}) \quad T_{surf} > T_{DP,a} \]

The surface efficiency \( \eta_{surf} \) is

\[ \eta_{surf} = 1 - \frac{A_{\text{fin}}}{A_o} (1 - \eta_{\text{fin}}) \]  \text{[3.72]}

\( h_a \) is the enthalpy of the humid air, and \( T_{DP,a} \) its dew-point temperature.

The fin efficiency \( \eta_{\text{fin}} \) is calculated as explained in Appendix D, considering the transport phenomena actually taking place (Rich 1966, McQuiston and Tree 1972, O'Brien and Turner 1965, Webb 1980). Special surface treatments such as surface roughning (Itoh et al. 1982) or polishing (Yoshii et al. 1971) are not considered. The method of solution adopted is a finite difference scheme, following the refrigerant path from the outlet of a circuit to its inlet. The reasons for this procedure are explained later in this chapter.
3.2.3.1 Heat Transfer Coefficient - Refrigerant

In order to establish the method to calculate the heat transfer coefficient to the vaporizing refrigerant, the basic processes that occur during saturated flow boiling inside tubes must be understood. Although all the transitions depicted in Fig. 3.28 may take place in a dry expansion evaporator, the transition $1 \rightarrow 2$ is not likely to happen in reality, it may show up during the simulation though.

**Fig. 3.28** - Transition processes during vaporization inside tubes.
Boiling inside tubes is a thermodynamic process with many applications, from power generation to cryogenics. Despite this fact, no single theory, such as Nusselt's for condensation, has yet been formulated. The reasons for this cannot be underestimated, and are at the origin of an enormous body of literature. The methods available to calculate the heat transfer coefficient for saturated flow boiling are based on empirical equations, most of them derived from experiments carried out in limited ranges of the influencing parameters. However, when developing an evaporator simulation model, it is necessary to provide for some generality, while keeping the complexity of the model within acceptable bounds, and limiting the possibilities considered to those representative in the field, e.g. horizontal tubes.

**Saturated Flow Boiling Heat Transfer Coefficient**

- Global
  - Vertical
  - Horizontal

- Local
  - Flow Pattern
    - Stratified
    - Annular
    - Spray
    - Others
  - Orientation
    - Vertical
    - Horizontal
    - Others
  - Mechanisms
    - Nucleate Boiling
    - Convective Boiling
  - Fluid Dependent
  - Fluid Independent
  - Surface Geometry
  - Tube Wall Properties

**Fig. 3.29** - Options considered in the development of the method to calculate the boiling heat transfer coefficient.
The schematic in Fig. 3.29 depicts the options considered when developing the method to calculate the heat transfer coefficient during flow boiling in this evaporator model. Global methods, such as proposed by Pierre (1964 a,b) and by Slipcevic (1972), are applicable in limited ranges of the refrigerant vapour quality, and do not provide insight leading to eventual optimization. They are very useful, however, to determine the basic dimensions of evaporators. Thus, methods allowing the calculation of the local heat transfer coefficient are preferred for a detailed simulation. These methods abound in the literature, but as said before, the way in which the various influencing parameters are considered varies widely. Most of the methods or correlations follow the method proposed by Chen (1966) to calculate the flow boiling heat transfer coefficient as a linear combination of heat transfer coefficients calculated for the two basic boiling mechanisms: nucleate and convective boiling, Eq. [3.73].

\[ \alpha_{TP} = E \alpha_{cb} + S \alpha_{nb} \]  

[3.73]

\( E \), is called the \textit{enhancement factor}, and is a measure of the augmentation in the liquid convection due to boiling. \( S \), is called the \textit{supression factor}, and is a measure of the effect of the flow upon bubble nucleation. Although Chen’s method was developed for water boiling in vertical tubes, it was later extended to other tube orientations and fluids. The fundamental limitation of the methods based on Chen’s, is their difficulty in handling nonsymmetrical flow patterns as they appear in tube orientations other than vertical. Equations for the heat transfer coefficient to forced flow boiling fluids that use Chen’s model are proposed, among others, by Bjorge et al. (1982), Shah (1982), Gungor and Winterton (1986,1987), Kandlikar (1987,1990,1991), and Murata and Hashizume (1991). When applied to horizontal and near horizontal tube orientations, some ways of accounting for a partially wetted perimeter are proposed. In the equations mentioned, one method is based on the Froude number of the liquid phase (Shah, Gungor and Winterton, and Kandlikar), and the other on the Lockhart-Martinelli parameter (Bjorge et al., Murata and Hashizume).
A desirable method, *in absencia* of a single theory, is a local one that takes account of the tube wall properties and surface geometry (enhancements), is independent of the fluid (no fluid specific parameter), considers the specificity of the boiling mechanisms, reflects the effects of the tube orientation\(^1\), and of the flow patterns.

The tube wall properties, especially its thermal conductivity, play an important role in the heat transfer, and the more so when the tube perimeter is not fully wetted by the same phase. In such cases, particularly when the conductance of the wall is low \(\lambda_w t_w < 1 \text{ W/K}\), a kind of fin effect allows a nonuniform perimetral temperature variation which reduces the \(\Delta T\) between the air and the tube wall (Müller-Steinhagen 1984, Schmidt 1986, Klimenko 1988).

The geometry of the internal surface of the tubes has been given increasing attention in recent years. Enhanced surfaces (fins, inserts, etc.) augment the heat transfer by increasing the surface area, the fraction of wetted area or both. Traditionally, smooth plain tubes have been extensively used. However, with the improvements on the air side, the controlling heat transfer coefficient shifted to the refrigerant side, leading to the use of surface enhancements. The first enhancements studied were the star-profile inserts and low internal fins, e.g. Lavin and Young (1965). More recently, most of the research efforts in this field have been directed to the study of surfaces such as the spiral-grooved, that improve the heat transfer without causing any significative increase in the pressure drop. Numerous studies of the vaporization of refrigerants in internal spiral-grooved tubes have been published: Ito et al. (1977), Ito and Kimura (1979), Kubanek and Miletti (1979), Lazarek (1980), Mori and Nakayama (1980), Kimura and Ito (1981), Khanpara et al. (1986), Panchal et al. (1986), Khanpara et al. (1987), Reid et al. (1987), and Yoshida et al. (1987) all dealt with some aspects of the application of this type of surface to the vaporization of refrigerants.

\(^1\) In this type of evaporator the tubes are always horizontal.
rants. Twisted-tape inserts were studied by Jensen and Bensler (1986), although they are not as favourable. The main conclusions of these studies may be summarized as follows:

Internal spiral-grooved tubes with groove depths less than 0.20 mm promote a significative augmentation of the vaporization heat transfer coefficient without any meaningful change in the pressure drop.

The groove inclinations that maximize the improvement of the heat transfer coefficient are $-10^\circ$ and $-90^\circ$, though the $90^\circ$ groove inclination promotes a larger pressure drop.

The most important variable affecting the improvement of the heat transfer coefficient, besides the geometry, is the mass flux. The effects of the heat flux are of secondary importance.

In order to be able to simulate the vaporization of refrigerants on this type of surfaces, it is proposed to use a multiplier, $\Lambda_f$, giving the enhancement of the heat transfer coefficient in relation to that of the internally smooth tube of the same diameter. In the derivation of this multiplier account is taken of the fact that at low (< 50 kg/m$^2$s) and at high (> 300 kg/m$^2$s) mass fluxes the heat transfer coefficient tends to the smooth tube value (Yoshida et al. 1987).

Flow boiling may take place at the liquid vapour interface (convective boiling), or at the tube liquid interface (nucleate boiling) or both. The dominance of one mechanism over the other is controlled by three parameters: Local wall superheating, heat flux, and mass flux. Fig. 3.30 shows a typical dependency of the boiling heat transfer coefficient upon these three parameters, from measurements on CFC12 by Iwicki and Steiner (1979). In Fig. 3.30, the horizontal part of the curves represents the region of convective boiling, which show no dependence upon heat flux, but are strongly dependent on the mass flux. The portion
of the curves to the right depend markedly on the heat flux, and less upon the mass flux. They represent the region where nucleate boiling dominates.

![Graph showing dependence of boiling heat transfer coefficient upon heat flux, mass flux, and wall superheating.](image)

**Fig. 3.30** - Dependence of the boiling heat transfer coefficient upon heat flux, mass flux, and wall superheating (adapted from Iwicki and Steiner 1979).

The two flow boiling mechanisms apply as long as the tube wall is at least partially wetted. After the boiling crisis\(^1\), the remaining liquid is suspended in the vapour flow and vaporization takes place at the surface of the droplets, when the vapour has acquired enough superheating.

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\(^{1}\) Boiling crisis is understood here as dryout. In a fluid heated system the wall superheating and the heat flux are controlled by the heating fluid temperature.
The problem of wall wetting brings the two-phase flow patterns into the picture. Extensive research results exist for flow boiling inside vertical tubes, as consequence of safety concerns in nuclear power plants, but less data has been collected for horizontal tubes. Zahn (1964, 1966) was perhaps the first to observe that two-phase flow pattern maps, such as Baker's (Baker 1954), that were established from observation of adiabatic flows, do not represent the situation found when boiling heat transfer takes place, Fig. 3.31.

Fig. 3.31 - Two-phase flow pattern map using the Baker parameters, with superposition of the boundaries suggested by Zahn (1964, 1966), and by Hashizume (1983).
The parameters in the coordinates of the Baker's map are\(^1\)

\[
\lambda = \left( \frac{\rho_G}{\rho_a} \frac{\rho_L}{\rho_{H_2}O} \right)^{1/2} \quad \Psi = \frac{\sigma_{H_2}O}{\sigma_L} \left( \frac{\mu_{H_2}O}{\mu_L} \left( \frac{\rho_{H_2}O}{\rho_L} \right)^2 \right)^{1/3}
\]

The most important differences between the maps for adiabatic and nonadiabatic two-phase flow patterns, are the reduction of the region where annular flow is observed, and the enlargement of those regions where stratified and spray flow patterns occur, Fig. 3.31. This implies a significative decrease of the fully wetted tube wall area. Zahn's observations are confirmed by Hashizume (1983).

Another important observation by Zahn, regards the effects of return bends upon the flow patterns. He observed that after each return bend, and independently of the flow direction, the whole perimeter of the tube was wet for a length of about 10 to 15 diameters, returning to the flow pattern observed in the previous tube, afterwards. Zahn's observations agree with others, such as by Worsøe-Schmidt (1959) and by Grønnerud (1975). This behaviour affects the heat transfer coefficient, and must be considered when developing a model based on local values. Still in relation to flow patterns, let's look now in more detail to the situation after the boiling crisis, where the spray flow pattern is observed.

It seems reasonable to assume, in the particular case of air-heated refrigerant evaporators, that due to the relatively small temperature difference between the wall and the refrigerant\(^2\), both the vapour and the liquid phases are close to saturation while the tube wall is at least partially wetted. Immediately after dryout, both vapour and liquid are at about the same temperature, and heat transfer

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1 The values of the air and water properties are taken at 20 °C.

2 A different situation occurs in experimental test rigs where the heat flux is constant (electric heating).
takes place to gaseous phase, except for the few droplets that impinge on the wall. It requires some length for the vapour to acquire enough superheating to completely vaporize the liquid droplets. Thus, in this region thermal equilibrium between vapour and liquid does not exist.

Although various authors (Miropol'skiy 1963, Groenveld and Delorme 1976, Mayinger and Langner 1978, Saha 1980, Schnittger 1982, Hein and Köhler 1986, Rohsenow 1988) propose models to deal with this nonequilibrium process, their solutions apply only to straight tubes. The effects of the return bends in an evaporator of the kind considered here, may promote the formation of rewetted patches in the post-dryout region depending on the flow velocity and on the size of the droplets. On the other hand, as the temperature difference between the tube wall and the vapour becomes small, the heat transfer rate decreases and so decreases the energy available to vaporize the remaining liquid. Therefore, the fraction of the total heat transfer surface area necessary for this process may be significant. In spite of these considerations, I will assume, for simulation purposes, that the vapour and liquid phases are in equilibrium after dryout, for this approach simplifies the model significantly.

From the foregoing analysis it seems reasonable to consider three regions in the evaporator, according to the fraction of the tube perimeter that is wetted: Partially wetted (stratified and stratified-wavy flow regimes), fully wetted (slug and annular flow regimes) and dry (mist flow regime after dryout). The sequence in which the analysis of the flow regimes is made is similar to that suggested by Steiner (1983): First check for stratified flow, then if the flow is not stratified check whether the flow pattern is spray, and if it is none of these, the wall is fully wetted. The check for stratified flow is made with a method proposed by Klimenko and Fyodorov (1990), that considers the hydrodynamics of both the liquid and vapour phase, and was derived from flow boiling data. The Klimenko's criterion says that the flow is stratified if the inequality
is verified. This criterion is graphically depicted in Fig. 3.32.

\[ 0.074 \left( \frac{d}{b} \right)^{2/3} \left( Fr_G + 8 \left[ 1 - \left( \frac{\rho_G}{\rho_L} \right)^{0.1} \right]^2 \right) Fr_L \leq 1 \]  

Fig. 3.32 - Graphical representation of the criterion to identify stratified flow according to Klimenko and Fyodorov (1990).

The criterion to identify the spray flow regime is based on the observations of Zahn (1964, 1966) and Hashizume (1983). The coordinates of the Baker flow regime map, Fig. 3.31, are used for this purpose, and the lower bound of the area where Zahn observed spray flow is delimited with the empirical equations

I adjusted to Zahn's data:
\[ Y = 37.5 \, X^{0.097} \quad 0.1 \leq X < 6.2 \]
\[ Y = 56.3 \, X^{-0.124} \quad 6.2 \leq X \leq 123 \]

Heat Transfer Coefficient for Boiling inside Smooth Tubes

Fig. 3.33 - Schematic of the distribution of the wetted and unwetted areas in a horizontal tube section with boiling two-phase flow.

Heat transfer to the vaporizing refrigerant inside the tubes is calculated as a weighted average of the heat transfer coefficients to the boiling liquid and the vapour. Heat transfer with fully wetted and dry wall involves only one phase, while with a partially wetted wall both phases must be considered. The general situation is depicted in Fig. 3.33, considering the dry perimeter to be defined by
the angle $\phi$. If the section considered is downstream of a return bend, and the flow pattern is not spray, then a portion of section, $\Delta z_{\text{ann}}$, is in annular flow regime, while the rest operates with a degree of wetting defined by the arc $\phi$. The total wetted fraction, $F_L$, of the tube section is

$$F_L = \frac{\Delta z_{\text{ann}} + \left(1 - \frac{\phi}{2\pi}\right) \Delta z_{\text{strat}}}{\Delta z}$$ \hspace{1cm} [3.76]$$

and the dry fraction is $F_G = 1 - F_L$.

The arc $\phi$ spanning the dry perimeter, is a function of the void fraction in the stratified flow regime, is null in the annular flow regime, and unity after dryout. The void fraction of the stratified flow is calculated according to Rouhani (1969), as suggested by Steiner (1983).

$$\varepsilon = \frac{x}{\rho_G} \left(1 + 0.12 \left(1 - x\right)\right) \left(\frac{x}{\rho_G} + \frac{1 - x}{\rho_L}\right) + \frac{1.18 (1 - x) \left(g \sigma (\rho_L - \rho_G)\right)^{0.25}}{\dot{m} \rho_G^{0.5}}$$ \hspace{1cm} [3.77]$$

The arc $\phi$ is calculated iteratively from

$$\phi = 2\pi \varepsilon + \sin \phi$$ \hspace{1cm} [3.78]$$

The heat transfer coefficient to the dry fraction is calculated using the Gnielinski (1983) equation for turbulent flow, Eq. [3.79].

$$\alpha_G = \frac{\lambda_G}{d} \frac{(\xi/8) \Pr_G (Re_G - 1000)}{1 + 12.7\sqrt{\xi/8} \left(Pr_G^{2/3} - 1\right)}$$ \hspace{1cm} [3.79]$$

$$\xi = \frac{1}{(1.82 \log_{10}Re_G - 1.64)^2}$$
For the wetted fraction, the heat transfer coefficient depends on whether the dominant boiling mechanism is convective or nucleate boiling. The heat transfer coefficient for the boiling liquid is calculated according to Shah (1982). Shah defines two dimensionless numbers, the boiling number $Bo$, and the convection number $Co$ that he uses to correlate the data for fully wetted boiling heat transfer:

$$ Bo = \frac{\dot{q}}{m \Delta h_g} \tag{3.80} $$

$$ Co = \left( \frac{1}{x} - 1 \right)^{0.8} \sqrt{\frac{\rho G}{\rho L}} $$

and relates the two-phase heat transfer coefficient $\alpha_{TP}$ to that of the liquid phase flowing alone in the tube $\alpha_{LO}$

$$ \alpha_{LO} = 0.023 \left( \frac{\dot{m}(1 - x) d}{\mu_L} \right)^{0.8} \left[ \frac{Cp_L \mu_L}{\lambda_L} \right]^{0.4} \frac{\lambda_L}{d} \tag{3.81} $$

by a multiplier $\psi$

$$ \psi = \frac{\alpha_{TP}}{\alpha_{LO}} \tag{3.82} $$

For the nucleate boiling dominated regime $Co > 1.0$, $\psi$ is the largest of $\psi_{nb}$ and $\psi_{cb}$ calculated as

$$ \psi_{nb} = 230 \sqrt{Bo} \quad Bo > 0.3 \times 10^{-4} $$

$$ \psi_{nb} = 1 + 46 \sqrt{Bo} \quad Bo \leq 0.3 \times 10^{-4} \tag{3.83} $$

$$ \psi_{cb} = 1.8 \times C_o^{-0.8} $$
Shah defines two regions in the nucleate suppressed boiling regime, where the multiplier $\psi$ is the largest of $\psi_{bs}$ and $\psi_{cb}$:

$0.1 < Co \leq 1.0$

\[
\psi_{bs} = F \sqrt{Bo} \times e^{(2.74 \cdot Co^{-0.1})}
\]  \hspace{1cm} [3.84]

$Co \leq 0.1$

\[
\psi_{bs} = F \sqrt{Bo} \times e^{(2.47 \cdot Co^{-0.15})}
\]  \hspace{1cm} [3.85]

The factor $F$ is given as

\[
F = 14.7 \quad Bo \geq 11 \times 10^{-4}
\]

\[
F = 15.43 \quad Bo < 11 \times 10^{-4}
\]

The choice of the method described is the result of the consideration of increasingly complex methods. Starting from global correlations (Pierre 1964a, Slipcevic 1972) which would not yield acceptable results when applied locally, I moved on to use the Gungor and Winterton (1987) correlation, which having been developed from a large data set including a lot of refrigerant data, should provide a better agreement with the measurements (Conde and Suter 1991). The Gungor and Winterton's correlation does not distinguish clearly among the various flow regimes.

The method I propose here applies the Shah (1982) correlation only to the wetted wall regions, while the heat transfer coefficient for dry wall regions is calculated as for single phase vapour. This represents the limit of application of the equilibrium approach. Further improvements will only be obtained with the more complex nonequilibrium approach.
Boiling Heat Transfer Coefficient inside Enhanced Tubes

The only kind of enhancement considered here is the spiral-grooved tube, as it is becoming more and more important in the design of compact evaporators. The data currently available are those of Ito and Kimura (1979). A reasonable approximation to those data is depicted in Fig. 3.34.

**Fig. 3.34** - Comparison of the heat transfer coefficient for the internal spiral-grooved tube to that of the smooth tube.
3.2.3.2 Heat Transfer Coefficient - Air

The heat transfer on the air side depends upon a large number of variables, which may be grouped as follows:

- Geometry
- Constructive aspects (surface enhancements, etc.)
- Coexisting transport phenomena

although there are mutual interferences such as geometry modifications due to frost deposition, for example. The calculation of the heat transfer coefficient is based upon works reported in the literature. Starting from a basic geometric configuration, with smooth plate fins, and considering only sensible heat transfer (no condensation or desublimation), the heat transfer coefficient is successively corrected, according to the geometry at hand and to the occurring transport phenomena.

3.2.3.2.1 Basic Geometry

The basic geometric dimensions of a plate finned-tube are depicted in Fig. 3.35. The minimum free flow area is between two consecutive tubes in the same layer for inline tube arrangements, but may occur as well between tubes in two consecutive layers for staggered tube patterns. In the first case the minimum free flow area is

\[
A_{\text{min}} = (S_T - D_o) (F_s - t)
\]  

[3.86]

while in the second case it becomes

\[
A_{\text{min}} = (F_s - t) \left( \sqrt{S_T^2 + 4S_L^2} - 2D_o \right)
\]  

[3.87]

\[S_T < \sqrt{S_T^2 + 4S_L^2}\]
The eventual presence of a layer of condensate, or of frost, also affects the geometry of the air flow channels. Assuming an uniform thickness of this layer, $F_p$, both the fin thickness $t$ and the tube diameter over the fin collar $D_o$ are modified to take this into account: $t$ becomes $t + 2F_p$ and $D_o$ becomes $D_o + 2F_t$.

### 3.2.3.2.2 Basic Heat Transfer Coefficient

The heat transfer coefficient for the basic geometry, with no condensation or desublimation effects is calculated according to Rich (1975). Rich presented the data as Colburn $j$-factors, Eq. 3.88, using as characteristic length the longitudinal tube spacing $S_L$ for the individual tube rows (layers) of the heat exchanger.
His experiments covered fin densities from 157 to 550 fins/meter. In fact, Rich did not find an influence of the fin spacing for dry operation, see also Webb (1980), but found it to be the best correlating characteristic length for wet operation (Rich 1973). Fig. 3.36 depicts the data published by Rich (1975) and an approximation to them obtained by the least squares method.

\[ j = \frac{\alpha_a}{m_a C_p a} Pr_a^{2/3} \left( \frac{\mu_{a,w}}{\mu_{a,b}} \right)^{0.14} \]  

[3.88]

Fig. 3.36 - Approximation to the data of Rich (1975), and extrapolation to a sixth row.
3.2.3.2.3 Surface Enhancement Effects and other Constructive Aspects

Enhanced surfaces are created by giving the fin surface either special treatments (Yoshii et al. 1971, Itoh et al. 1982) to obtain drop or film condensation of the air humidity for example, or some kind of pattern to augment the air side heat transfer. Earlier studies by Kays and London (1964), considered flat plate finned-tubes (smooth fins). In current use today there are wavy (also known as corrugated or herringbone) fins, by far the most common, louver (slits are cut and deflected to another plane), and offset strip fins (Mori and Nakayama (1980), Nakayama and Xu (1983)). These surfaces are effective at increasing the heat transfer coefficient but also the friction losses, which impose a penalty in terms of fan power. Ito et al. (1977) refer an increase of the heat transfer coefficient up to 42% of the louver and offset strip fins, in relation to smooth fins, while Nakayama and Xu (1983) report improvements of the order of 30%, and Mori and Nakayama (1980) claim heat transfer coefficient augmentations of the order of 60%. The pressure drop is only slightly increased in relation to the wavy surface, as shown by all authors, when no burrs are left on the borders of the offset parts. Webb (1987) reports values of the heat transfer coefficient enhancement of the order of 50 to 70% for wavy fins over smooth plate fins, and of 70 to 90% for offset strip fins.

Plate finned-tube heat exchangers are manufactured with either inline or staggered tube arrangements Fig. 3.35. Inline tube arrangements have bad heat transfer qualities (Shah and Webb (1983), Webb (1983a, 1983b, 1987, and 1990b)) and are seldom used as evaporators. The manufacturing process of these heat exchangers has evolved remarkably since their first applications in the 1920s. Taborek (1987) reports on this evolution, and on the problems found, with extended use, on the bond resistance between fins and tubes. Gardner and Carnavos (1960), and Eckels and Rabas (1977) studied the problem posed by the loosening of the fins from the tubes. Sheffield et al. (1985) developed an equation to calculate this resistance. The thermal resistance between fins and
tubes is extremely dependent upon the manufacturing process (Nho and Yovanovich 1989) and upon the amplitude of the temperature cycles imposed on the exchanger. On the other hand, as observed by Manzoor et al. (1984), whenever good contact between fin collar and tube is achieved for at least 10 percent of the contact surface, the contact conductance is almost equal to the perfect contact conductance. Because in the application as heat pump evaporator the amplitude of the temperature variations is small, and because today's manufacturing processes result mostly in high quality products, the effect of the contact resistance upon the overal thermal resistance is assumed negligible.

Although inline tube arrangements may also be considered, the model will largely overpredict their performance (Webb 1990b). The effect of the fin surface patterns on heat transfer is considered according to Nakyama and Xu (1983) for louver and offset strip fins, and according to Beecher and Fagan (1987) for the wavy ones.

Wavy, Corrugated or Herringbone Fin Patterns

Beecher and Fagan (1987) studied several wavy fin patterns including a smooth surface (null pattern) fin. The data obtained is presented as Nusselt number as function of the Graetz number (see Webb 1990a in regard to this form of data presentation). The Beecher and Fagan's data for patterned fins may be related to those for the null pattern, all other dimensions being equal, as

\[
\frac{Nu_{patt}}{Nu_{flat}} = f \left( Gz, \frac{Fin \ Pattern \ Depth}{Fin \ Spacing} \right)
\]

For this expression to be useful in the context of the present model, a relationship between the Nusselt number, as defined by these authors, and the Colburn j-factor must be established. As defined by Beecher and Fagan the Nusselt number is
\[ Nu = \frac{\alpha_m D_H}{\lambda_a} \quad [3.89] \]

This definition is based on the arithmetic temperature difference, as pointed out by Webb (1990a), but included in the ratio above will not affect this derivation.

The \( j \)-factor may be expressed in terms of the Nusselt number as

\[ j = \frac{Nu \lambda_a}{D_H \rho C_p m_{a,max}} Pr \frac{2}{3} \left( \frac{\mu_{a,w}}{\mu_{a,b}} \right)^{0.14} \quad [3.90] \]

so the heat transfer coefficient multiplier, \( \Lambda_2 \), for wavy patterned fins is

\[ \Lambda_2 = \frac{j_{\text{patt}}}{j_{\text{flat}}} = \frac{D_{H,\text{flat}}}{D_{H,\text{patt}}} f \left( \frac{Gz}{F_i}, \frac{\text{Fin Pattern Depth}}{\text{Fin Spacing}} \right) \quad [3.91] \]

The function \( f \) is depicted in Fig. 3.37.

**Louver and Offset Strip Fin Patterns**

The heat transfer coefficient multiplier for these types of fin patterns is that proposed by Nakayama and Xu (1983).

\[ \Lambda_2 = 1 + 1093 \left( \frac{F_S}{F_S - t} \right)^{1.24} \phi_s^{0.944} Re_{D_H}^{-0.58} \]
\[ + 1.097 \left( \frac{F_S}{F_S - t} \right)^{2.09} \phi_s^{2.26} Re_{D_H}^{0.88} \quad [3.92] \]

\[ \phi_s = \frac{2 n_s - 1}{S_T S_L - \frac{\pi D_o^2}{4}} W_s L_s \]
The limits of application of this multiplier are

\[ 0.2 \leq \phi_s \leq 0.35 \]
\[ 250 \leq Re_{DH} \leq 3000 \]
\[ 1.8 \leq F_S \leq 2.5 \text{ mm} \]
\[ 0.15 \leq t \leq 0.2 \text{ mm} \]

**Fig. 3.37** - Wavy fin pattern depth effects (Beecher and Fagan 1987).
3.2.3.2.4 Simultaneous Heat and Mass Transfer Effects

The transport phenomena, on the air side, range from the simple single phase convective heat transfer to simultaneous heat and mass transfer, either with condensation or desublimation. Situations where ice accumulates onto the surfaces of the fins and tubes, due for example, to previous defrost operations, are not handled by this model. An approximation to simulate this situation is possible however, by specifying a frost thickness different from zero. The eventual ice layer will be considered as a frost layer (lower density and lower thermal conductivity, in relation to ice). The Chilton-Colburn analogy between convective heat and mass transfer is the basis for the calculation of the heat transfer coefficient multiplier when simultaneous heat and mass transfer occur (Eckels and Rabas 1987, McQuiston 1975, 1976, 1978a, 1978b, 1981, Isaji and Tajima 1973). The heat transfer coefficient multiplier \( \Lambda_3 \) accounts for the effects of the simultaneous heat and mass transfer, and is defined by McQuiston (1978b) as

\[
\Lambda_3 = \frac{j_{IM}}{j_{dM}}
\]

\[
j_{IM} = 0.0014 + 0.2618 \cdot JP \cdot JFS
\]

\[
j_{dM} = 0.0014 + 0.2618 \cdot JP
\]

with

\[
JP = ReD_o^{-0.4} \left( \frac{A}{A_t} \right)^{-0.15} \]

\[
A = \frac{4}{\pi} \frac{S_L}{D_H} \frac{S_T}{D_o} \frac{A_{min}}{A_{frontal}}
\]

\[
D_H = \frac{4 A_{min} S_L}{A} \]

\[
ReD_o = \frac{\dot{m}_{a, max} D_o}{\mu_a}
\]
and

\[ JFS = \left(0.95 + 4 \times 10^{-5} \, Re_{Fs}^{1.25}\right) \left(\frac{F_s}{F_s - t}\right)^2 \]  
\[ Re_{Fs} = \frac{\dot{m}_{a,max} F_s}{\mu_a} \]  

Finally, the air side heat transfer coefficient \( \alpha_a \) is calculated with all generality as

\[ \alpha_a = \Lambda_2 \Lambda_3 j \, C p_a \, \dot{m}_{a,max} \, Pr_a^{-2/3} \left(\frac{\mu_{a,w}}{\mu_{a,b}}\right)^{0.14} \]  

which includes the effects of row order, fin spacing, fin surface enhancements and the occurrence of simultaneous heat and mass transfer.

3.2.3.3 Friction Pressure Losses - Refrigerant

Pressure losses by friction on the refrigerant side take place both in the straight portions and in the return bends. Besides these, there are important pressure losses in the refrigerant distribution system (nozzle, manifold) due to the refrigerant flowing in two-phase.

**Pressure Losses across Sudden Contractions in Two-Phase Flow**

In a manifold type of supply, the pressure loss in the connection to the tube in the core of the heat exchanger may be calculated as that across a sudden contraction, Fig. 3.38. The pressure loss across the nozzle supply type, Fig. 3.40, may be calculated as for an ideal nozzle. Weisman et al. (1978) and Husain et al. (1978) studied the flow of two-phase vapour-liquid mixtures across abrupt area changes and suggested that a homogeneous model of the flow is suitable for the
calculation of the pressure losses across sudden contractions, while a slip model gives better results for losses across sudden expansions. Mayinger (1982) introduced several simplifications into the equations proposed by Weisman et al. (1978). The simplified equation for the pressure drop across sudden contractions is

\[ \Delta P = \frac{m_2^2}{2} \left[ 1 - \left( \frac{A_2}{A_1} \right)^2 + \left( \frac{1}{a_c} - 1 \right)^2 \right] \left[ \frac{x}{\rho_G} \frac{1 - x}{\rho_L} \right] \]  

[3.97]

\[
\text{Flow}
\]

\[
\begin{array}{c}
A_1 \quad P_1 \\
\hline
1 \quad C \quad 2
\end{array}
\]

\[
A_2 \quad P_2
\]

Fig. 3.38 - General schematic of a sudden contraction. C is the vena contracta plane.

\(a_c\) is the ratio of the flow section at the vena contracta to that downstream. An approximation to the data published by Mayinger (1982) for \(a_c\) is depicted in Fig. 3.39.
Fig. 3.39 - The vena contracta area ratio for a sudden contraction in adiabatic two-phase flow. Approximation to the data of Mayinger (1982).

\[
a_c = 0.578 \exp \left[0.525 \left(\frac{A_2}{A_1}\right)^2\right]
\]

Fig. 3.40 - Schematic representation of a refrigerant nozzle distributor.
Pressure Losses in Return Bends

The pressure loss in return bends represents the larger part of all the refrigerant pressure loss in the evaporator. The bends are assumed adiabatic as they are not in the air stream, so the vapour mass quality at outlet of the preceding tube is used in the calculations. Geary (1975) reports on pressure drop in return bends with two-phase flow and discusses a method for its calculation. This method is applicable for vapour mass qualities up to 0.80. From there to complete vapour flow, he suggests in the discussion of the paper, that a logarithmic interpolation between the value predicted by his method at \( x = 0.8 \), and that calculated for the whole flow as vapour flowing alone, should be used.

A more comprehensive method is proposed by Grønnerud (1975), which allows a detailed calculation of the pressure losses in return bends. Grønnerud calculates the pressure drop in bends as a linear combination of the losses generated by each of the individual phases alone. In contrast to Geary’s method, this approach presents the advantage of coherency from single phase liquid to single phase vapour. The total pressure drop for phase \( i \), is

\[
\Delta P_i = \Delta P_{st,i} + \Delta P_{fricc,i} + \Delta P_{mom,i} \tag{3.98}
\]

and the linear combination for both phases, as proposed by Grønnerud is

\[
\Delta P_{b, x} = (1 - \varepsilon)^{0.385} \Delta P_L + x^{0.775} \Delta P_G \tag{3.99}
\]

where the void fraction \( \varepsilon \) is calculated as for homogeneous flow.

The static pressure change of the gaseous phase \( \Delta P_{st,G} \) may be neglected in comparison with the other terms, that of the liquid phase \( \Delta P_{st,L} \) must account for the direction of the flow, which depends on the circuit layout, and the inclination of the evaporator, Fig. 3.41. The description of circuit layouts is discussed in section 3.2.3.6. In terms of the nomenclature in that section, the sign of the static pressure change is
Fig. 3.41 shows the characteristic dimensions of a return bend, and the angle of inclination of the evaporator as used in this model. The individual components of

\[ -1 \iff \text{[col supplier tube]} < \text{[col current tube]} \]
\[ +1 \iff \text{[col supplier tube]} > \text{[col current tube]} \]
\[ 0 \iff \text{[col supplier tube]} = \text{[col current tube]} \]

In reality, this is not absolutely correct, especially for staggered tube layouts, as it may take the three values -1, +1, 0.
the pressure loss in a return bend are

\[
\Delta P_{st,i} = 2R_b \sin \beta
\]

\[3.100\]

\[
\Delta P_{fric,i} = \frac{0.3164}{\left(\frac{m_i d}{\mu_i}\right)^{0.25}} \frac{\pi R_b \dot{m}^2}{2 \rho \dot{m}}
\]

\[3.101\]

\[
\Delta P_{mom,i} = \frac{1}{2} \zeta \frac{\dot{m}^2}{\rho_i}
\]

\[3.102\]

where \(i\) stands for either the liquid or the gaseous phase.

The momentum change coefficient \(\zeta\) is obtained from Eck (1966) as recommended by Grønnerud,

\[
\zeta = 1.497 \theta^{0.75} \sum_{j=-1}^{3} \chi_j \left(\frac{d}{R_b}\right)^j
\]

\[3.103\]

with the bend span (\(\pi\) for return bends) \([\theta] = \text{rad}\), and the \(\chi_j\)s

\[
\chi_{-1} = 7.76052 \times 10^{-3}
\]

\[
\chi_0 = -9.21210 \times 10^{-3}
\]

\[
\chi_1 = 0.35970
\]

\[
\chi_2 = -0.18024
\]

\[
\chi_3 = 9.55261 \times 10^{-2}
\].
Friction Losses with Vaporization in Tubes

The friction pressure losses during vaporization inside tubes, are calculated using a two-phase friction multiplier, $\Psi$. The correlation developed by Friedel (1975, 1979, 1984) is used for this purpose. It applies to internally smooth pipes, and is regarded as one of the best available. The two-phase friction multiplier as established by Friedel (1984) is

$$\Psi = (1 - x)^2 + x^2 \frac{p_L}{\rho_G} \frac{\zeta_G}{\zeta_L} +$$

$$3.43 \times 0.685 (1 - x)^{0.24} \left(\frac{p_L}{\rho_G}\right)^{0.8} \left(\frac{\eta_G}{\eta_L}\right)^{0.22} \times$$

$$\left(1 - \frac{\eta_G}{\eta_L}\right)^{0.89} Fr_L^{-0.047} We_L^{-0.0334}$$

[3.104]

The single phase friction coefficients $\zeta_i (i = G, L)$ are calculated as suggested by Friedel (1979).

$$\zeta_i = \left[0.86859 \frac{ LN \left( \frac{Re_i}{1.964 \frac{LN(Re_i)}{3.8125}} \right)^2 }{-1.964 LN(Re_i) - 3.8125} \right]$$

$Re_i > 1055$

[3.105]

$$\zeta_i = \frac{64}{Re_i}$$

$Re_i \leq 1055$

The definitions of the Froude and Weber numbers are

$$Fr_L = \frac{m^2}{g D_H p_L^2} \quad We_L = \frac{m^2 D_H}{\rho_L \sigma}$$
The single phase pressure drop, for phase $j$, is

$$\Delta P_i = \zeta_i \frac{z}{D_H} \frac{m^2}{2 \rho_j}$$  \hspace{1cm} [3.106]

### 3.2.3.4 Friction Pressure Losses - Air

Contrary to the heat transfer on the air side, the calculation of the friction coefficient is carried out on a global basis, assuming all the exchanger operating at the same conditions of the current tube. The pressure loss is calculated tube-by-tube, however. The friction coefficient is calculated according to McQuiston (1978b, 1981). McQuiston analysed existing data and presented a correlation based on a parameter proposed earlier by Jameson (1945). Gray and Webb (1986) also developed a correlation for the dry operation of smooth plate finned-tube heat exchangers, and compared it to McQuiston's. Their analysis suggests that the McQuiston correlation would give very large deviations, but a comparison of the equations they give as McQuiston’s with the original references, show they are in error in more than one exponent and one constant. For enhanced surfaces, wavy, louver and offset strip fins, a multiplier due to Nakayama and Xu (1983) is applied to the friction factor calculated using the correlation of McQuiston. The Reynolds number based on the tube outer diameter correlates well data for dry operation, but that based on the fin spacing correlates better the data with simultaneous mass transfer (Rich 1973). The Reynolds number based on the tube outer diameter is

$$Re_D = \frac{\dot{m}_{a,\text{max}} D_o}{\mu_a}$$  \hspace{1cm} [3.107]
The Reynolds number based on the fin spacing is

\[ Re_{F_s} = \frac{m_{a,max} F_s}{\mu_a} \]  \[ (3.108) \]

McQuiston (1978b) developed his correlation so as to use the same basic factors for both dry and wet operation, one defaulting to unity for dry operation. The general form of the McQuiston's fanning friction factor equation is

\[ f_f = A + B \left( F_p F(F_s) \right)^2 \]  \[ (3.109) \]

\[ F_p = Re_D^{-0.25} \left( \frac{D_o}{D_H} \right)^{0.25} \left[ \frac{S_T - D_o}{4 (F_s - t)} \right]^{-0.4} \left[ \frac{S_T}{D_H} - 1 \right]^{-0.5} \]  \[ (3.110) \]

\[ D_H = \frac{D_o}{A_{lubes}} \frac{F_s A_{total}}{S_T - D_o + F_s} \]  \[ (3.111) \]

\( F(F_s) \) introduces the effects of the filmwise humidity condensation, and is given by McQuiston (1978b) as

\[ F(F_s) = \frac{0.6 + Re_{F_s}^{-0.15}}{\left( \frac{F_s}{F_s - t} \right)^3} \]  \[ (3.112) \]

with simultaneous heat and mass transfer. \( F(F_s) \) defaults to unity for dry operation. Finally, the friction factor \( f \) is given as

\[ f = 4 \ f_f \ K_f \]  \[ (3.113) \]

where \( K_f \) reduces to unity for smooth fins, and is given by Nakayama and Xu (1983)

\[ K_f = 1 + 0.0105 \ Re_{D_H}^{0.575} \]  \[ (3.114) \]
for any patterned type of fin, as may be concluded from Hosoda et al. (1979). The pressure drop becomes then

$$\Delta P_{tube} = \left( f \frac{S_L}{D_H} + K_i + K_o \right) \frac{m_{a,max}^2}{2 \rho_a}$$  \[3.115\]

3.2.3.5 Void Fraction during Vaporization

The local void fraction during flow boiling depends upon the local flow pattern. As discussed before in section 3.2.3.1, three kinds of flow patterns are considered in this model: stratified, annular, and spray. For the stratified pattern, the Rouhani equation [3.78], is recognized to give good results (Steiner 1983). The Tandon equation (Tandon et al. 1985) was developed for annular flow. Tandon's equation uses the Lockhart-Martinelli property index for turbulent-turbulent flow as correlating parameter.

$$\varepsilon = 1 + A \left( \frac{Re_L^\beta}{F(X_{tt})} \right) + B \left( \frac{Re_L^\beta}{F(X_{tt})} \right)^2$$  \[3.116\]

with the following values for the constants A, B, and β

<table>
<thead>
<tr>
<th>$Re_L$</th>
<th>A</th>
<th>B</th>
<th>β</th>
</tr>
</thead>
<tbody>
<tr>
<td>&gt; 1125</td>
<td>-0.38</td>
<td>0.0361</td>
<td>-0.088</td>
</tr>
<tr>
<td>&gt; 50 ≤ 1125</td>
<td>-1.928</td>
<td>0.9293</td>
<td>-0.315</td>
</tr>
<tr>
<td>≤ 50</td>
<td>-2.828</td>
<td>2.0</td>
<td>-0.45</td>
</tr>
</tbody>
</table>

where

$$Re_L = \frac{\dot{m} (1 - x) \, d}{\mu_L}$$  \[3.117\]

is the Reynolds number calculated for the liquid phase.
\[ F(X_{ft}) = 0.15 \left[ X_{ft}^{-1} + 2.85 \ X_{ft}^{-0.476} \right] \] 

\[ X_{ft} \] is the Lockhart-Martinelli parameter for both phases in turbulent flow, defined by the equation

\[ X_{ft} = \left( \frac{1 - x}{x} \right)^{0.9} \left( \frac{\rho_G}{\rho_L} \right)^{0.5} \left( \frac{\mu_L}{\mu_G} \right)^{0.1} \] 

Finally for spray flow, the droplets are assumed to have the same velocity as the vapour phase, so the homogeneous form of the void fraction may be used.

### 3.2.3.6 Solution Method of the Evaporator Model - Algorithm

As explained before, this type of evaporator is usually divided in a number of parallel circuits which are themselves complete heat exchangers. In order to carry out the simulation at the local level, a convenient method of description of the refrigerant paths must be used. Furthermore, such method shall also be suitable for analysis of circuit similarity. Similar circuits are those having the same number of tubes, and the same sequence of tubes in the various tube layers. Similar circuits need only be calculated once, potentially reducing the computation time.

**Description of the Heat Exchanger Geometry and Analysis of Similarity**

A finned-tube coil heat exchanger has several parallel layers of tubes in the air flow direction – called *rows* in the following – and several tubes in the direction normal to the air flow – called *cols* in the following. The identification of a tube in such a matrix requires the tube position \([\text{row}, \text{col}]\) and those of the tubes supplying or being supplied by it. Common practice shows that no more than two tubes supply to, or receive from, any tube. Thus, five pairs \([\text{row}, \text{col}]\) are required to fully describe a single tube. The following scheme is adopted, and the numbering conventions are illustrated in Figure 3.42.
Fig. 3.42 - Definitions for refrigerant circuitry specification.

The circuitry layout data are generated by means of interactive graphics. The generation of these data must obey a certain number of rules, and is an excellent candidate for a rule-based expert system. Seven rules, which are an extension of those proposed by Hogan (1980) and Ellison et al. (1981), govern the specification of the refrigerant circuitry, and are as follows:
RULE 1  A heat exchanger is built up of a finite number of sub heat exchangers (Hogan 1980).

RULE 2  In a sub heat exchanger the refrigerant never reverses flow direction relative to the air stream. This is equivalent to: A sub heat exchanger must start at one face of the coil and terminate at the opposite face.

RULE 3  Connections from one row to the other are made to the nearest tube.

RULE 4  All sub heat exchangers manifest the same net flow direction (Hogan 1980).

Fig. 3.43 - A nonsplitted circuit layout.
RULE 5 If no splitting or joining occurs, Fig. 3.43, then no tube can supply or receive refrigerant to or from more than one tube, respectively.

RULE 6 After a splitting or a joining, Fig. 3.44, respectively joining or splitting in the same circuit is not allowed.

RULE 7 The number of tubes assigned to a circuit may only differ by multiples of 2 from the number assigned to any other circuit (need to have all the circuit outlets at the same side of the exchanger).
The similarity among the different refrigerant circuits that constitute an evaporator is verified by counting the number of tubes in each row and comparing them. The methodology for this analysis is explained by the flow diagram shown in Fig. 3.45.

**The Distribution of Refrigerant among the Parallel Circuits of a Dry Expansion Evaporator**

When there are different circuit layouts, the distribution of the refrigerant flow among them is not necessarily even, due to different pressure gradients. The parallel circuits in a dry expansion evaporator may be considered as a network of flow impedances associated in parallel. An electrical network analogy is evident here. The voltage difference is analog to the pressure difference, while the current is analog to the mass flow rate squared.

\[
\Delta P = \frac{2 f L M^2}{\pi D^3 \rho} \leq [U = R \times I] 
\]  

[3.120]

This is so for single phase flow along a rectilinear pipe. By analogy, one may write the same as an approximation for two-phase flows. This approximation is also used for a dry expansion evaporator, although none of \(f, D\) and \(\rho\) are necessarily constant along the whole path of the refrigerant, from the TXV to the outlet of the circuits in the evaporator. By sticking to the analogy, even in the case of mixed two and single phase flows, and variable diameter along the path, one may calculate the hydraulic impedance by dividing the path total \(\Delta P\) by the mass flow rate squared. For an evaporator with \(n\) parallel circuits, grouped together according to their geometric layout similarity, there are \(m\) different patterns, and \(Z_1 \ldots Z_m\) circuits per pattern \(\Sigma_m Z_i = n\). The following system of linear equations permits the calculation of the flow rate distribution.
Fig. 3.45 - Analysis of the refrigerant circuitry layout.
where the $p_i$s are defined as

$$\beta_i = -\left(\frac{R_i}{R_1}\right)^{1/2} \quad [3.122]$$

An initial estimate is required to initialize the flow rate distribution, as the pressure drop is not known \textit{a priori}. The flow impedances are made equal to the hydraulic length of the circuits in the first iteration. For the second and further iterations, the flow impedances are determined from the previous iteration, and the flow distribution recalculated. The solution is considered satisfactory when at least 50% of the individual circuit layout patterns' pressure drop differ less than 10% from the average pressure drop of all different layout patterns.

**General Solution Method**

In the most general case, an evaporator of the type considered, may be seen as a collection of parallel refrigerant circuits, which are themselves a collection of tubes connected by return bends. Each circuit is supplied with refrigerant by a circuit feeder, and delivers the vaporized refrigerant to the outlet manifold. The air flows in cross-countercurrent to the overall flow direction of the refrigerant. Carrying out the simulation at the local level means following, in an organized manner, from the least to the most complex of these structures, that is from a
fraction of the tube length to the whole tube and the associated return bend, to the circuit, to the whole exchanger.

The flow diagram in Fig. 3.46 explains the solution algorithm to the heat exchanger problem for a fraction of a tube. As the simulation proceeds against the refrigerant flowstream direction, the outlet boundary conditions on the refrigerant side, and those at the inlet on the air side are known, and so are the flow rates of the refrigerant and the air. The solution process is iterative, with the main iteration carried out on the tube wall temperature, assumed constant for each fraction of tube length. The refrigerant inlet conditions are determined within this process, and are the boundary conditions for the next step. The air outlet conditions are as well determined in this process, and are used to determine the boundary conditions for the corresponding length fraction of the next tube layer. The air is considered fully mixed between tube layers for each length step.

The algorithm to simulate a refrigerant circuit is illustrated in the flow diagram of Fig. 3.47. This part ensures that the proper sequence of tubes is followed, manages the boundary conditions and the balances of momentum, mass and energy on both sides of the heat exchanger.

The evaporator simulation algorithm is described by the flow diagram in Fig. 3.48. The number of circuits to simulate - geometrically similar circuits are simulated just once - is determined here, and so are the distribution of the refrigerant flow among the various circuits, and the entrance and exit losses of both air and refrigerant.

This approach to the simulation of the evaporator assumes an uniform air flow distribution over the frontal surface of the evaporator and equal temperatures at the outlet of all circuits. These assumptions permit simplifications such as the calculation of only one circuit for each different circuit layout. It allows as well the simulation to follow the air flow direction, which reduces the number of iteration
loops required\(^1\). The assumption of an uniform air flow distribution over the frontal surface of the evaporator is that that is farther from the reality. However, excepting the case of a single tube layer evaporator, any specified nonuniform distribution, in particular along the fins as used by Domanski (1989), applies only to the first layer of tubes. For further layers, and unless a three dimensional model is used on the air side, the air flow distribution will be unknown. A nonuniform flow distribution along the tubes may eventually maintain itself throughout if the fins are continuous: no offset strip or louvered fin patterns.

The specification and use of a nonuniform air flow distribution on the frontal surface poses no great problem. What such distribution looks like is difficult to determine in most cases, let alone the increased complexity of the computer code. Previous models of plate finned-tube coil evaporators (Fischer and Rice 1983, Domanski 1989) are characterized by rather simple approaches to the calculation of the heat transfer process. The present model innovates in particular in the methods to specify the layout of the refrigerant circuits, in the calculation of the boiling heat transfer and refrigerant pressure losses, and in the consideration of the eventual presence of frost, though the frost deposition process itself cannot be simulated (instationary process).

\(^1\) The thermodynamic state of the humid air is defined by a triplet while that of the refrigerant is defined by a doublet all over the evaporator.
Fig. 3.46 - Flow diagram describing the algorithm to simulate an element of tube.
Fig. 3.46 - Flow diagram describing the algorithm to simulate an element of tube (Cont'ed).
Fig. 3.47 - Flow diagram describing the algorithm to simulate a refrigerant circuit.
Fig. 3.48 - Flow diagram describing the algorithm of the evaporator model.
3.3 Throttling Device

The liquid refrigerant, at the outlet of the condenser, must be throttled to a lower pressure for its vaporization to be possible at the low temperature of the source, in the evaporator. The process of refrigerant throttling may take place in various kinds of components which have a common operating principle: The liquid is accelerated to a relatively high velocity (25 + 80 m/s), so that its pressure will eventually be lower than that corresponding to saturation, and flashing occurs contributing to further pressure reduction. In the course of flashing the acoustic velocity is almost always attained, having as consequence that the throttling process is practically independent of the downstream pressure.

Fig. 3.49 - Schematic representation of a typical thermostatic expansion valve.
Traditionally, constant flow section throttling devices have been used for relatively small refrigeration capacities, and in other cases as well when the operating conditions are not expected to deviate significantly from the design conditions. For higher capacity applications, or in cases where rather variable operating conditions are expected, variable section throttling devices are more common. Examples of constant flow section devices are capillary tubes, short nozzles, fixed orifices. Expansion valves are variable flow section devices that may be either thermostatically or electronically activated.

The success of a simulation model as that proposed here for reverse Rankine cycle machines, depends on the adequacy of the models and data of the various components to represent the phenomena to simulate. Previous attempts at modelling thermostatic expansion valves (Habberschill 1983, Fischer and Rice 1983) failed because the data available from the manufacturers are plainly insufficient to develop a reasonable physically based model. Other attempts to model this device as a multiple or single fixed diameter orifices (Huelle 1971b, Hamman and Rocaries 1983, MacArthur 1984, Beckey 1986) also failed because such models cannot represent the valve behaviour under very different operating conditions. Thus, other methods had to be used.

The model of the thermostatic expansion valve described in the following is conceptually simple, though the parameters required for its use are not readily available. The method proposed here to obtain them, Appendix E, demands some experimental effort, but the necessary means, though not available to all potential users of the model, are common in most laboratories. The essential difficulty lies in the need for detailed geometric data of the valve's seat.

An alternative approach, leading eventually to a still simpler model, would be based on the block diagram depicted in Fig. 3.50. The functions describing each block, or even associations of blocks, could be experimentally obtained. The limitation here is that a much more sophisticated test plant is required in order to
cover the whole operating range of the valve for all the important parameters. In particular, the parameters for the alternative approach cannot be obtained from an experimental test rig as that described in chapter 5.

3.3.1 Physical Description and Operation of Thermostatic Expansion Valves

Fig. 3.49 depicts schematically a typical thermostatic expansion valve. The thermostatic bulb, or phial, 4, is attached to the evaporator outlet tube, thus sensing the temperature of the refrigerant at the evaporator outlet. The pressure generated inside the phial acts upon the upper face of the diaphragm 3, and is a function of the evaporator outlet temperature, and of the type of charge in the bulb. Under the diaphragm acts the actual evaporation pressure\(^1\). Thus, the diaphragm "computes" the evaporator outlet superheating by the play of these two pressures. The spring 7 under the diaphragm, ensures that the phial pressure is always higher than the evaporation pressure. The tension on this spring may be adjusted by turning the screw 8, Fig. 3.49. The TXV may be adapted to the actual evaporator using this screw, which sets the opening superheating, in fact promoting a translation of the valve characteristic. It also helps in setting the TXV to satisfy the stability criteria, (Huelle 1971 a). The diaphragm acts upon the valve poppet 5 through the actuator 2, displacing it and thus generating a wider or narrower flow section. A TXV performs several functions at once in a vapour compression refrigeration or heat pump machine: It throttles the refrigerant from the condensation to the evaporation pressure while maximizing the use of the evaporator heat transfer area, and protecting the compressor from liquid surges. These functions may be divided by their nature into control and throttle functions. However, they are not independent, and a simulation model should reflect their mutual influence.

---

\(^1\) For evaporator systems with a small pressure drop, this pressure may be the TXV outlet pressure, taken inside the valve (internally equalized valve), while for evaporators with a large pressure drop this pressure should be the refrigerant pressure at evaporator outlet (externally equalized valve).
Fig. 3.50 - Block diagram of the control function of a thermostatic expansion valve with external equalization.
As control devices TXV's may be described as proportional controllers with a time delay imposed by the evaporator they work with. Fig. 3.50 depicts a block diagram of the control function of a TXV. Although the static (no flow) opening characteristics of the valve are normally linear, as shown in Figs 3.51 through 3.53, the effect of the forces generated by the flow acceleration may modify these characteristics significatively. In order to model the steady-state operation of a TXV, it must be assumed that the stability criteria (Huelle 1971a) are fulfilled. However, the model shall be able to predict extreme operating conditions, e.g. maximum opening, which are an indication that those criteria are not met. The model shall as well predict the minimum required adjustment of the valve static superheating settings necessary to avoid extreme operating conditions.

Fig. 3.51 - Opening characteristic at -20 °C evaporation temperature, and its displacement by adjusting the setting screw by one turn clockwise (Conde and Suter 1992).
3.3.2 The Mathematical Model

The TXV operates in steady-state when the equilibrium of the forces acting upon the diaphragm, Fig. 3.49, is attained. The equilibrium is mathematically expressed as

\[ F_b = F_r + F_s + F_{dy} \]  \hspace{1cm} [3.123]

where

- \( F_b \) is the force generated by the bulb (phial) pressure on the upper face of the diaphragm
- \( F_r \) is the force generated by the operating fluid (refrigerant) on the lower face of the diaphragm
- \( F_s \) is the sum of the forces exerted by both springs 7 and 9 (Fig. 3.49)
- \( F_{dy} \) is a force resulting from the imbalance of pressures around the poppet, due to the acceleration of the flow.

The force \( F_b \) depends exclusively upon the current temperature at the thermostatic bulb 4 and on the kind of charge in it. The spring force \( F_s \) may be decomposed into three components

\[ F_s = F_{s,0} + F_a + K_s X \]  \hspace{1cm} [3.124]

where

- \( F_{s,0} \) is the sum of the forces exerted by the both springs 7 and 9 with the valve completely closed, at the factory settings
- \( F_a \) results from an eventual adjustment of the static opening superheating of the valve, by means of the screw 8
- \( K_s X \) is the component due to the poppet displacement. \( K_s \) is assumed constant (linear springs).
Fig. 3.52 - Opening characteristic at 0 °C evaporation temperature, and its displacement by adjusting the setting screw by one turn clockwise (Conde and Suter 1992).

There are several unknowns in this model that are difficult to determine explicitly. The calculation of $F_b$ requires the knowledge of the pressure-temperature relationship of the phial charge, Fig. 3.54, which is unknown but may be determined by experiment as described in Appendix E. The calculation of $F_s$ is virtually impossible: Although $K_s$ is easy to determine, the factory set tension $F_{s,0}$ cannot be easily measured. The relationship between the poppet displacement and the flow area at the valve throat may be calculated when the geometry of the poppet and of the orifice are known. From the geometry as shown in Fig. 3.55, it results:
\[ A_T = \pi X \left( D \sin \theta - \frac{X \sin^2 2\theta}{\cos \theta} \right) \]  

and the hydraulic diameter at the throat is

\[ D_{H,T} = \frac{4 A_T}{\pi (2 D - X \sin 2\theta)} \]  

Fig. 3.53 - Opening characteristic at 10 °C evaporation temperature, and its displacement by adjusting the setting screw by one turn clockwise (Conde and Suter 1992).
Fig. 3.54 - The pressure-temperature relationship of the bulb charges (measured in two valves, one designed to operate with HCFC22 and the other with HFC134a).

Although this model is simple, it seems at first impractical. It may, however, be put to work if the difficulties mentioned are circumvented. One way to overcome them, is to establish a direct relationship between the poppet displacement $X$, and the measurable variables that determine it, on one hand, and between the forces due to flow acceleration and the excess superheating required to equilibrate them, on the other.
Under static (no flow) conditions, it is possible to establish a relationship between the valve opening displacement $X$, and the superheating applied to the phial to produce it. Within the physical limits of the valve — totally closed and fully open — this relationship may be assumed linear, Figs 3.51 - 3.53. The parameters describing the static opening characteristic — slope and intercept — depend upon the evaporation pressure, as expressed, in general, by Eq. [3.127].

\[ X = f_1(P_{ev}) + f_2(P_{ev}) \Delta T_{SH, st} \quad 0 \leq X \leq X_{max} \quad [3.127] \]

The form and the parameters of the functions $f_1$ and $f_2$ are determined from experimental data as those depicted in Figs. 3.51 through 3.53. Further, the effect of an eventual adjustment of the static opening superheating must be considered in relation to the factory settings. Since the springs are assumed linear, the adjustment of the static opening superheating may only promote a translation of the valve static opening characteristic, measured as from the factory (experimentally verified in Fig. 3.51 - 3.53). Hence, a correction term may be determined that permits the use of the static opening characteristic, as from the factory, in the calculation of the actual throat area $A_T$. The actual static superheating is then corrected to account for the static opening superheating adjustment, Eq.[3.128]. The term $\Delta(\Delta T_{SH})_{turn}$ depends upon the evaporation pressure, and is determined through experiment (Appendix E).

\[ \Delta T_{SH, actual, static} = \Delta T_{SH, factory, static} + \Delta(\Delta T_{SH})_{turn} \frac{\phi}{360} \quad [3.128] \]

It results from above, that it is possible to determine the free flow area at the throat $A_T$ required for the intended flow rate, and that it is as well possible to find out the non-flow superheating necessary to generate such an area. The actual operating superheating results from the consideration of the term $F_{dy}$ in Eq.[3.123], generated by the imbalance of pressures around the poppet. Hence, a simultaneous solution of the upper static model and of the conservation equations is required. What is obtained from that solution is the value of the force $F_{dy}$. The excess superheating necessary to equilibrate $F_{dy}$ is calculated from the
pressure-temperature relationship of the charge in the phial, Fig. 3.54, taking into account the surface area of the upper face of the diaphragm.

\[ \Delta T_{SH,dy} = \frac{\Delta P_{dy|T_{phial}}}{\left( \frac{dP_{phial}}{dT} \right)_{T_{phial}}} \]  \[3.129\]

This equation yields the excess superheating \( \Delta T_{SH,dy} \) required to balance the forces produced at the diaphragm by the acceleration of flow across the throttling section. Other types of phial charges (gas cross, MOP, adsorption, etc.), may require a different formulation of this dependence.

**Fig. 3.55** - Schematic of the throttling section of a thermostatic expansion valve.
Dynamic Forces Actuating upon the Valve Poppet

Fig. 3.56 - Schematic showing the thermostatic expansion valve poppet and actuator, displaying the forces applied to them.

The balance of forces due to the refrigerant flow, around the poppet of the valve, yields the excess force $F_{dy}$ that the pressure in the phial-diaphragm system must equilibrate. Fig. 3.56 shows the details of the poppet and actuator, and all the forces concerned. The force $F_3$, see also Fig. 3.55, is calculated from the conservation equations applied to the control volume CV1, of the operating fluid, Figs. 3.55 and 3.57. Then, the additional forces applied to the poppet, $F_5$ and $F_6$, and to the actuator $F_4$, are calculated from the balance of the control volume CV2, Figs. 3.55 and 3.58. Several assumptions are required to write the conservation equations for these control volumes, namely:

- the flow may be considered unidimensional
- the flow is incompressible while the fluid is in the liquid phase
- flashing of the superheated liquid occurs only after the throat
- the two-phase mixture is in thermodynamic equilibrium at the section $A_o$, and the flow is considered homogeneous.
The conservation of linear momentum applied to CV1, in the direction parallel to the $\xi$, yields,

\[ F_1 - F_2 + F_3 \sin \theta - F_T \cos \theta = \dot{M}(u_T \cos \theta - u_1) \]  \[3.130\]

the continuity equation for the same control volume is,

\[ u_T = \frac{A_1}{A_T} u_1 \]  \[3.131\]

and the energy equation is the Bernoulli equation under the assumptions made,

\[ P_1 + \frac{1}{2} \rho_1 u_1^2 = P_T + \frac{1}{2} \rho_1 u_T^2 \]  \[3.132\]

The solution to this system of equations, returns the flow area at the throat $A_T$ the pressure at the throat $P_T$, and the force $F_3$. The static operating superheating is that required to produce $A_T$, and is determined from the static opening characteristic. The forces $F_2$ and $F_3$ are calculated as integrals of the pressure distribution on the respective surfaces, Fig. 3.56. As the flow accelerates converging to the throat, there is a significant variation of the local pressure.
(typical velocities at the throat are 60 m/s). Average velocities at each section in the convergent are considered.

The conservation of linear momentum applied to CV2, yields,

\[ F_T \cos \theta + F_5 \sin \theta + F_6 - F_o = \dot{M}(u_o - u_T \cos \theta) \]  \[3.133\]

from where \( F_5 \sin \theta + F_6 \) are necessary to calculate \( F_{dy} \), Fig. 3.56.

\[ F_{dy} = F_p - F_3 \sin \theta - F_5 \sin \theta - F_6 + F_4 \]  \[3.134\]

Although it is not necessary to care for what happens within CV2, it is necessary to be aware of the flow conditions at the section \( A_o \), that is it is necessary to know whether the flow chokes there, or whether it remains subsonic. The condition to remain subsonic is that the flow velocity \( u_o \) be lower than the acoustic velocity corresponding to the thermodynamic state of the refrigerant at \( A_o \). It should be remembered that the acoustic velocity for a two-phase liquid-vapour mixture is substantially lower than that of the saturated vapour, Fig. 3.59. This done, the excess pressure required on the diaphragm is

\[ \Delta P_{phial,dy} = \frac{F_{dy}}{A_{diaph}} \]  \[3.135\]

where \( A_{diaph} \) is the active area of the diaphragm. The excess superheating necessary to generate this excess pressure, is now determined by calculating the variation of the temperature of the phial charge that satisfies Eq.[3.129].

The actual operating superheating is the sum of that required to generate the throat area \( A_T \) (static model), with the excess superheating necessary to balance the dynamic forces, \( \Delta T_{SH,dy} \)

\[ \Delta T_{SH} = \Delta T_{SH,sl} + \Delta T_{SH,dy} \]  \[3.136\]
Fig. 3.59 - Acoustic velocity of homogeneous two-phase liquid-vapour mixtures of HCFC22, calculated according to Michaelides and Zissis (1983) (see also Hilfiker 1975).

The first output variable of the model, the operating superheating $\Delta T_{SH}$ is then calculated. The second variable is the refrigerant vapour quality at the TXV outlet, which is easily calculated applying the continuity and energy equations, and assuming the flow at the TXV outlet to be homogeneous, and in thermodynamic equilibrium.
3.3.3 Solution Algorithm of the Thermostatic Expansion Valve Model

Fig. 3.60 - Flow diagram describing the algorithm to solve the expansion valve model.
3.4 Refrigerant Piping

Refrigerant piping are like any other piping system, with the exception that two-phase flow may occur in some of the piping lines. Pipes have heat interactions with the surroundings, promote pressure losses and store refrigerant. The simulation model is described in its three important aspects; Heat transfer, pressure drop, and resident mass of refrigerant in the tubes.

3.4.1 Heat Transfer

Heat transfer to and from refrigerant piping takes place by convection and radiation. Some tubes may eventually be submitted to forced convection on the outside surface. The general case, however, is that with free convection and radiation. Heat transfer under a given threshold of the temperature difference between the refrigerant and the surroundings is assumed negligible. As general approach, the radiative heat interaction is considered to take place with a view factor of unity, between the tube surface and a fictitious surface at the surrounding air temperature. The convective heat interaction depends on the pipe orientation. For a vertical pipe the characteristic length is its length, while for a horizontal one, it is its diameter. Most pipes have horizontal and vertical portions, or even other tube orientations. For simulation purposes, the pipe is considered vertical if its major part has this orientation. Otherwise it is considered horizontal. No intermediate cases are considered.

The correlation proposed by McAdams (1954) for heat transfer by free convection is used for both horizontal and vertical tubes. The equations have the same general form in both cases, the characteristic length \( CL \) is the diameter for horizontal tubes, and the length for vertical ones. McAdams equation is
$$Nu = \frac{\alpha(\text{CL})}{\lambda} = c \left(Gr_{CL} Pr\right)^n$$ (3.137)

The Grashoff number $Gr_{CL}$ is

$$Gr_{CL} = \frac{g \beta(T_{wo} - T_0)}{\nu^2} (CL)^3$$ (3.138)

with the coefficient of thermal expansion $\beta$

$$\beta \equiv \frac{1}{\rho} \left(\frac{\partial p}{\partial T}\right)_P$$ (3.139)

The ranges of application defined by the product $Gr_{CL} Pr$, and the values of the parameters $c$ and $n$ are given in Table 3.1.

<table>
<thead>
<tr>
<th>Range</th>
<th>Orientation</th>
<th>$c$</th>
<th>$n$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$10^4 + 10^9$</td>
<td>Horizontal</td>
<td>0.53</td>
<td>1/4</td>
</tr>
<tr>
<td>$10^9 + 10^{12}$</td>
<td>Horizontal</td>
<td>0.13</td>
<td>1/3</td>
</tr>
<tr>
<td>$10^4 + 10^9$</td>
<td>Vertical</td>
<td>0.59</td>
<td>1/4</td>
</tr>
<tr>
<td>$10^9 + 10^{13}$</td>
<td>Vertical</td>
<td>0.10</td>
<td>1/3</td>
</tr>
</tbody>
</table>

The heat interaction inside a tube depends upon the thermodynamic state of the refrigerant, on the geometry of the pipe, and on the flow conditions. Single and two-phase flows are considered. The single-phase heat transfer coefficient is
calculated from correlations as detailed in Table 3.II.

**Table 3.II - Heat transfer correlations in single-phase flow.**

<table>
<thead>
<tr>
<th>$Re_D$</th>
<th>$Pr$</th>
<th>$L/D$</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$&gt; 10^4$</td>
<td>$0.7 \div 100$</td>
<td>$&gt; 60$</td>
<td>Dittus-Boelter</td>
</tr>
<tr>
<td>$&lt; 10^4$</td>
<td>$0.7 \div 160$</td>
<td>$&gt; 60$</td>
<td>Colburn</td>
</tr>
<tr>
<td></td>
<td>$0.7 \div 16700$</td>
<td>$10 \div 400$</td>
<td>Notter and Sleicher</td>
</tr>
</tbody>
</table>

The equations for these three correlations are

**Dittus-Boelter**

$$\alpha = 0.023 \frac{\lambda}{D} Re_D^{0.8} Pr^n$$  \hspace{1cm} (3.140)

with $n = 0.3$ for cooling and $n = 0.4$ for heating.

**Colburn**

$$St \Pr^{2/3} = 0.023 \ Re_D^{-0.2}$$  \hspace{1cm} (3.141)

or

$$\alpha = 0.023 \ \rho \ u \ Cp \ Pr^{-2/3} \ Re_D^{-0.2}$$

**Notter and Sleicher**

$$\alpha = 0.036 \frac{\lambda}{D} Re_D^{0.8} Pr^{1/3} \left(\frac{L}{D}\right)^{-0.055}$$  \hspace{1cm} (3.142)

The two-phase heat transfer coefficient is calculated using the general correlation proposed by Klimenko (1988). Klimenko’s correlation is divided in two parts. The
Convective boiling number \( N_{CB} \) (Eq. [3.146]) is used to define the range of application of each part. The two-phase Nusselt number is defined in a special form by Klimenko, using the Laplace number (Eq. [3.147]) as characteristic length.

\[
Nu_{TP} = \frac{\alpha b}{\lambda_L} \quad (3.143)
\]

The correlation equations have the following ranges and form

\[
\begin{align*}
Nu_{TP} &= Nu_b \quad N_{CB} < 1.6 \times 10^4 \\
Nu_{TP} &= Nu_c \quad N_{CB} > 1.6 \times 10^4
\end{align*}
\]

with

\[
Nu_b = 7.4 \times 10^{-3} Pe^{0.6} K_p^{0.5} Pr_L^{-1/3} \left( \frac{\lambda_w}{\lambda_L} \right)^{0.15} \quad (3.144)
\]

and

\[
Nu_c = 0.087 Re_m^{0.6} Pr_L^{1/6} \left( \frac{\rho_G}{\rho_L} \right)^{0.2} \left( \frac{\lambda_w}{\lambda_L} \right)^{0.09} \quad (3.145)
\]

The convective boiling number \( N_{CB} \) is

\[
N_{CB} = \left( \frac{Re_m}{Re_*} \frac{\rho_L}{\rho_G} \right)^{2/3} \quad (3.146)
\]

and the Laplace number \( b \) is

\[
b = \sqrt{\frac{\sigma}{g(\rho_L - \rho_G)}} \quad (3.147)
\]
The average Reynolds number $Re_m$ is

$$Re_m = \frac{u_m b}{v_L} \quad (3.148)$$

The mean velocity $u_m$ is

$$u_m = \frac{G}{\rho_L} \left[ 1 + x \left( \frac{\rho_L}{\rho_G} - 1 \right) \right] \quad (3.149)$$

The modified Peclet number $Pe_*$ is

$$Pe_* = \frac{\dot{q} b}{\dot{q}_f \rho_L \alpha_L} \quad (3.150)$$

$\alpha_L$ is the thermal diffusivity of the liquid phase $\alpha_L = \frac{\lambda_L}{\rho_L C_p_L}$. 

The dimensionless parameter $K_p$ is defined as

$$K_p = \frac{P}{\sqrt{\sigma g (\rho_L - \rho_G)}} \quad (3.151)$$

and the modified Reynolds number $Re_*$ is

$$Re_* = \frac{\dot{q} b}{\dot{q}_f \rho_G v_L} \quad (3.152)$$

The general heat transfer problem between the refrigerant tubes and the surroundings is solved using the electric resistance analogy, as shown in Fig. 3.61. The tubes may be insulated or not. For bare tubes $R_{is}$ is set to null, and $T_{mw} = T_{wo}$. 
Fig. 3.61 - Schematic representation of the resistance network equivalent to the thermal system of a tube.

The thermal resistance due to the radiative heat interaction is

$$R_{\text{rad}} = \frac{1}{\sigma \varepsilon A (T_{w_0}^3 + 2T_{w_0}T_0 + T_0^3)}$$  \hspace{1cm} (3.153)

As both the radiative and the convective heat interactions on the outer surface depend upon its temperature, this problem requires an iterative solution. The iteration is carried out on $T_{w_0}$ and the algorithm used is the so-called Newton-Raphson method. The initial value given to $T_{w_0}$ is the mean value of the temperatures of the refrigerant and of the environment. The heat transfer equations are

$$\dot{Q} = \frac{T_i - T_0}{R_{wm} + R_{is} + R_o(T_0)}$$  \hspace{1cm} (3.154)

$$\dot{Q} = \frac{T_i - T_{w_0}}{R_{wm} + R_{is}}$$  \hspace{1cm} (3.155)
\[ Q = \frac{T_{wo} - T_0}{R_0(T_{wo})} \]  

\[ \frac{1}{R_0(T_{wo})} = \frac{1}{R_{\text{conv}}(T_{wo})} + \frac{1}{R_{\text{rad}}(T_{wo})} \]  

Solving equations [3.156] and [3.157] for \( T_{wo} \), results

\[ T_{wo} = \frac{T_i R_0(T_{wo}) + T_0 (R_{wm} + R_{is})}{R_0(T_{wo}) + R_{wm} + R_{is}} \]  

Defining \( F \) or the Newton-Raphson algorithm as

\[ F = T_{wo} - \frac{T_i R_0(T_{wo}) + T_0 (R_{wm} + R_{is})}{R_0(T_{wo}) + R_{wm} + R_{is}} = 0 \]

one obtains for the iteration

\[ T_{wo,new} = T_{wo,old} - \frac{F(T_{wo,old})}{\left( \frac{dF}{dT_{wo}} \right)_{T_{wo,old}}} \]

with the derivative

\[ \frac{dF}{dT_{wo}} = 1 - \frac{T_i - T_0}{[R_0(T_{wo}) + R_{wm} + R_{is}]^2} \times (R_{wm} + R_{is}) \frac{dR_0(T_{wo})}{dT_{wo}} \]
and

\[
\frac{dR_o(T_{wo})}{dT_{wo}} = -\frac{1}{[R_o(T_{wo})]^2} \sigma \varepsilon A \left( 3 T_{wo}^2 + 2 T_o \right)
\]  \quad (3.162)

The change of the refrigerant state is calculated once the energy transport by radiation is known. Here the direction followed by the simulation process, in relation to the flow direction, is taken into account. The discharge and liquid lines are simulated in the same direction as the refrigerant flow, while the suction and evaporator feed lines are simulated against the flow direction. The losses are subtracted in the first case and added in the second, in order to estimate the refrigerant enthalpy at the other end of the tube. For two-phase flows the vapor quality is also corrected.

### 3.4.2 Pressure Drop

The pressure drop in a refrigerant pipeline may be due to three main causes; friction, gravity effects, and momentum change. This last component is mostly due to vaporization or condensation in the pipes. It is assumed negligible for all tubes in this model.

The pressure drop due to friction in refrigerant pipes is calculated for both single and two-phase flow cases. The general equation for the calculation of the friction pressure drop of a refrigerant flow is

\[
\Delta P = \Psi \rho \frac{u^2}{2} \left( \frac{fL}{D} + \Sigma K \right)
\]

For single-phase flow \(\Psi\) is unity. For two-phase flows, the Baroczy-Chisholm (Chisholm 1983) correlation for friction pressure losses is used. Originally, \(\Psi\) should affect only the friction factor \(f\). It is assumed however, that local losses
represented here by $\Sigma K$ are equally affected by the two-phase flow. So, in fact the two-phase friction multiplier is applied here to the single phase pressure loss that would occur had the whole flow been single-phase liquid, Eq.[3.164].

$$\Delta P = \Psi \Delta P_{LO}$$  \hspace{1cm} (3.164)

The pressure change due to gravity effects may be important in some cases. Thus it is considered in this simulation model. Again, it is calculated for both single and two-phase flows. Furthermore, the relative directions of the flow and of the simulation must be taken care of. The signal of the pressure change by gravity is explained in Table 3.III.

<table>
<thead>
<tr>
<th>FLOW DIRECTION</th>
<th>FORWARD SIMULATION</th>
<th>BACKWARD SIMULATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Up</td>
<td>-</td>
<td>+</td>
</tr>
<tr>
<td>Down</td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td>Horizontal</td>
<td>Null</td>
<td>Null</td>
</tr>
</tbody>
</table>

The pressure change by gravity is given as

$$\Delta P_g = \rho \ g \ \Delta h$$  \hspace{1cm} (3.165)

$\Delta h$ represents the altimetric distance between both ends of the tube. For two-phase flows, an average value of the density is calculated using the average void fraction of the flow $\varepsilon$. 
\[ \bar{\rho} = \rho_L (1 - \bar{\varepsilon}) + \rho_G \bar{\varepsilon} \quad (3.166) \]

The void fraction is calculated with Eq. [3.61] (Smith 1969). The refrigerant state at the other end of the tube, is corrected considering the total pressure loss. The vapor quality of two-phase flows is also corrected, assuming the pressure loss to be isenthalpic. The heat transfer losses are taken into account separately. The local pressure losses in pipe bends and fittings are calculated using the local loss coefficient \( K \). The values of \( K \) are given in Appendix F for various types of tube fittings. There are components which could be thought as pipe fittings, but that cannot be described by a single value for \( K \). A typical case is the filter-drier usually fitted to the liquid line. The calculation of the pressure drop in this component is discussed in Appendix F.

### 3.4.3 Mass of Refrigerant Resident in a Pipe

The mass of refrigerant held by a tube is given, in general, by

\[ M = V \left[ \rho_G \bar{\varepsilon} + \rho_L (1 - \bar{\varepsilon}) \right] \quad (3.167) \]

which is valid for all cases. For liquid-phase alone the void fraction \( \varepsilon = 0 \) and \( M = V \rho_L \), and inversely, for vapor phase alone \( \varepsilon = 1 \) and \( M = V \rho_G \). For two-phase flows \( \varepsilon \) has a nontrivial value that is calculated using a suitable correlation. The correlation of Smith (1969) is adopted in this model. It is an entrainment based correlation, which requires the least number of substance properties, of all the correlations available. The entrainment factor \( K \) set equal to 0.4 as suggested by Smith, Eq.s [3.60 and 3.61].
3.5 Fans and Ductwork

Air-source heat pumps require in most cases a fan to promote the forced circulation of the air through the evaporator. In many cases some ductwork is also used and, though not an integral part of the heat pump itself, it does affect the heat pump performance. Hence, models for these components have been developed as well.

Usually small heat pump units, of the split type, are fit with axial fans in the outdoor, evaporator part. Packaged units, especially for indoor installation, are fit with centrifugal fans in order to reduce noise emission. Although in most cases the centrifugal fans are single speed, the advent of speed controlled heat pump units in the market, and in particular their design, requires the possibility of accurate simulation of these fans. Axial fans for the smaller units are usually single speed fixed blade fans. Also in this case, further progress in the design techniques may require in the near future the use of either variable pitch blades, or variable speed, in particular when optimization is either desired or necessary. Variable-capacity variable-speed heat pumps are microprocessor controlled, so the application of a simple, though accurate fan model is useful in the development of general hardware capable of acceptance, by soft programming, of a few model specific parameters. Such control hardware would be the same at least for a scaled-up series of units.

The main difficulty in simulating ductwork comes from the fittings, elbows, filters, dampers, etc., usually present. These local losses are considered in a standardized manner in the ductwork model.
3.5.1 Fan Model

3.5.1.1 Dimensionless Parameters used in the Simulation Models of Fans

Eck (1973), showed that the characteristics of a family of similar fans may be represented using three dimensionless variables:

- $\Phi$ - the dimensionless volumetric flow rate,
- $\psi$ - the dimensionless total pressure head, and
- $\lambda$ - the dimensionless fan power.

These variables are defined from the fan physical magnitudes as:

$$\Phi = \frac{4 \dot{V}}{\pi^2 d^3 n}$$

[3.168]

$$\psi = \frac{2 P_{tf}}{\rho_a (\pi d n)^2}$$

[3.169]

$$\lambda = \frac{8 \dot{W}}{\pi^4 d^5 n^3 \rho_a}$$

[3.170]

3.5.1.2 Centrifugal Fans

Manufacturers' data for centrifugal fans are typically represented in graphical form. The calculation of the parameters for the mathematical model requires that a reasonable number of data points be abstracted from those graphs, or from tables when available. Although no equation requires more than four points,
Theoretically, their representation of the data becomes more precise as the number of data points increases. The geometry of the fan must be known, and a suitable software for the determination of the parameters should be available as well (e.g., an application of the least squares method). Centrifugal fans are fixed blade fans, so the data include only curves for the various operating speeds. This model uses the dimensionless parameters to build general equations of the form:

\[ \psi = e^{\left( \sum_{i=0}^{3} A(i) \phi^i \right)} \]  \[3.171\]

\[ \lambda = e^{\left( \sum_{i=0}^{3} B(i) \phi^i \right)} \]  \[3.172\]

\[ \eta = \frac{\Phi \psi}{\lambda} \]  \[3.173\]

respectively for the total pressure and power characteristics. The efficiency is obtained from the other two dimensionless parameters. Fig. 3.62 shows plots of these three dimensionless characteristics for a typical centrifugal fan. In this case a computer program that fits polynomial functions of any degree, within a predefined coefficient of variation, has been used. The coefficients of variation of all the fits depicted in Fig. 3.62 is less than 1%, and the maximum deviation less than 3%.
3.5.1.3 Axial Fans

Axial fans may have both fixed blades or variable pitch blades. Typically, fixed blades are used in the smaller units, and variable pitch blades in the larger ones. Small and medium size heat pumps when fitted with axial (propeller) fans, use the fixed speed fixed blade fans. The introduction in the market of variable speed driven heat pumps may require this type of fans to be operated at variable speed, another alternative is to use fixed speed variable blade pitch fans, although these may be more expensive and more difficult to control.
Twizell and Bright (1981), suggested that the performance of an axial fan may be described by an equation of the form:

\[ P = A \left( \dot{V}_0 - \dot{V} \right)^\alpha \]  

[3.174]

These authors also suggested that \( \dot{V}_0 \) is a function of the blade pitch alone, of the form

\[ \dot{V}_0 = \sum_{i=0}^{2} A_0(i) \theta^i \]  

[3.175]

The volumetric flow rate at null total pressure, \( \dot{V}_0 \) is not always available in the manufacturers' data. An alternative representation of the fan characteristic is proposed here. It is proposed to consider the fan stalling characteristics instead of the flow rate at null pressure head. The Eq. [3.174] would then take the form

\[ P = P_{st} - A \left( \dot{V} - \dot{V}_{st} \right)^\alpha \]  

[3.176]

where the subscript \( st \) refers to stalling conditions. The stalling pressure and volumetric flow are usually available in the manufacturers' catalogs. In dimensionless form the Eq. [3.176] is

\[ \Psi = \Psi_{st} - A \left( \Phi - \Phi_{st} \right)^\alpha \]  

[3.177]

with

\[ \Psi_{st} = \psi(\theta) \quad \Phi_{st} = \Phi(\theta) \quad A = f_1(\theta) \quad \alpha = f_2(\theta) \]

Fig. 3.63 depicts the dependence of \( \Phi_{st} \) and \( \psi_{st} \) from the blade angle \( \theta \), for a typical axial fan. The dimensionless power of an axial fan may be described as function of the dimensionless flow rate \( \Phi \) and dimensionless total pressure \( \psi \), as
The efficiency is calculated from the Eq. [3.173].

\[ \lambda = \sum_{i=0}^{2} D(i)(\Phi\Psi)^i \]  

Fig. 3.63 - Dimensionless stalling characteristics of an axial fan.

The manufacturers' data for axial fans are usually given in graphical form for a number of rotation speeds. The abstraction of the data from the graphs is time consuming, although it is required to determine the parameters of the model. In order to represent the fan characteristics with reasonable accuracy, a relatively large number of data points should be taken. The determination of the
coefficients is better done using a least squares method. Fig. 3.64 depicts the dimensionless $\tilde{V} - P$ characteristics of a typical axial fan. The data represented in the graph show an excellent reproduction of the manufacturer's data by the model.

Fig. 3.64 - Dimensionless pressure - volume characteristic of an axial fan. Comparison of catalog data with model.
3.5.1.4 Model of Induction Motor Fan Drives

The shaft power required by a fan may be written as

\[ \dot{W} = \eta_{transm} \eta_{elec} \dot{E}_{power} \]

\[ \dot{E}_{power,3-Phase} = \sqrt{3} Ul\cos\phi \quad [3.179] \]

\[ \dot{E}_{power,mono} = Ul\cos\phi \]

The transmission efficiency \( \eta_{transm} \) is the product of the mechanical efficiencies of the motor and of the transmission itself (\( \eta_{transm} = 0.93 \times 0.95 \)). The electric efficiency varies with the load at which the drive is submitted

\[ \eta_{elec} = 1 - \frac{\dot{E}_{losses}}{\dot{E}_{power}} \quad [3.180] \]

The electrical losses \( \dot{E}_{losses} \) are proportional to the square of the electric current (Andreas 1982).

\[ \dot{E}_{losses} = 2.7 \times \text{Stator Losses} - i^2 \quad [3.181] \]

Thus one may write

\[ \left( \frac{\dot{E}_{losses}}{\dot{E}_{nominal}} \right)_{actual} = \left( \frac{\dot{E}_{losses}}{\dot{E}_{nominal}} \right)_{nominal} \frac{i^2}{i^2_{nominal}} \quad [3.182] \]

The power factor is load dependent (Fink and Beat 1987), and may be approximated as (Appendix A)

\[ \cos\phi_{actual} = \cos\phi_{nominal} \left( \frac{\dot{E}}{\dot{E}_{nominal}} \right)^{0.4} \quad [3.183] \]

so the electric power required by an induction drive at conditions other than nominal may be obtained, after some algebra, from
\[ \dot{E}_{\text{power}} = \frac{\dot{W}}{\eta_{\text{transm}}} + \left( \frac{\dot{E}_{\text{losses}}}{\dot{E}_{\text{power,nominal}}} \right)^{1.2} \left( \dot{E}_{\text{power}} \right)^{1.2} \]  

[3.184]

This equation has an implicit form, thus has to be solved by iteration. The Newton-Raphson method is appropriate for this.

### 3.5.2 Ductwork Model

The pressure losses in a complex ductwork are usually calculated as the sum of the losses due to the singularities (bends, filters, dampers, etc.) and those due to friction. Mathematically, for a ductwork with \( m \) legs and \( n \) fittings

\[ \Delta P = \frac{V^2 \rho_a}{2} \left( \sum_{j=1}^{m} f_{D,j} \frac{z_{D,j}}{D_{H,j}} \frac{1}{A_j^2} + \sum_{i=1}^{n} \frac{K_i}{A_i^2} \right) \]  

[3.185]

When a heat exchanger is included in the ductwork, such as in the case of the evaporator of an air-source heat pump, a third term appears in the RHS of the Eq. [3.185]. This third term accounts for the pressure drop in the heat exchanger.

\[ \frac{1}{A_{\text{min}}^2} \left( f_I \frac{A}{A_{\text{min}}} + K_{e,i} + K_{e,o} \right) \]

The friction factors are calculated as functions of the Reynolds number of the flow in the particular section they concern. For straight sections of duct, the friction factor is calculated according to Churchill (1977), Eq. [3.186]. In order to estimate the air flow rate when starting a simulation, the friction factor of the heat exchanger \( f_I \) is required. An equation due to Kays and London (1964) is used for this purpose, Eq. [3.187].
\[ f = \left( \frac{8}{Re_{DH}} \right)^{12} + \frac{1}{(A + B)^{2/3}} \]  

\[ A = \left[ 2.457 \ln \left( \frac{7}{Re_{DH}} \right) + \frac{0.9}{D_H} + \frac{0.27 e}{D_H} \right]^{16} \]

\[ B = \left( \frac{37530}{Re_{DH}} \right)^{16} \]

\[ f_f = 0.339 \ Re^{-0.321} + 0.72 \ Re^{-0.447} \]  

With the Reynolds number \( Re \) based on the hydraulic diameter of the minimum free flow area in the heat exchanger. For later iterations the evaporator module exports the pressure loss on the air side.

**Local Loss Coefficients (Singularities)**

The local loss coefficients are obtained from tables for the various types of singularities. When specifying the ductwork layout the loss coefficients are added in the form specified by Eq. [3.185]. A given section of ductwork is characterized by its geometry, length and sum of singularity loss coefficients. Two ductwork sections are usually considered in the case of an air-source heat pump: Inlet and outlet duct.

**3.5.3 Algorithm of the Fan - Ductwork System Model**

Finding the solution of a fan - ductwork system consists in solving simultaneously the ductwork equation with fan’s characteristic equation, and out of this solution determine the power required by the fan’s driving motor. This system of equations is not always well behaved. As shown in Fig. 3.65 when the iteration starts by the
largest absolute slope the solution will not converge (cases 2 and 3). Alternatively, a stepwise approximation may be used at first, and then the bisection method will converge rapidly. The flow diagram depicted in Fig. 3.66 presents the method of solution adopted.

Fig. 3.65 - Divergent and convergent approaches to the solution of the fan ductwork system.
Fig. 3.66 - Flow diagram describing the algorithm to solve the fan-ductwork system model.
3.6 Fluid Properties

The models of the components described in the foregoing sections require the knowledge of a large number of thermodynamic and transport properties of the fluids they handle. In a model conceived for design purposes those properties must be calculated for real fluids, as stressed by Black (1986). The fluids used with heat pumps include that undergoing the cyclic process - the refrigerant - and the source and sink fluids, mostly humid air, water or a brine. The thermodynamic and transport properties of the refrigerant are required for the liquid and the vapour phases, and for the two-phase liquid-vapour region as well. Water and brine properties are mostly necessary for the liquid phase, although solid water (frost and ice) properties may as well be required when the source fluid is atmospheric air, or an ice producing heat pump.

3.6.1 Properties of Refrigerants

Thermodynamic Properties

The refrigerant, operating in the reverse Rankine cycle, goes through many processes requiring accurate and consistent thermodynamic property values for their calculation. All thermodynamic properties\(^1\) are calculated on the basis of an equation of state (P-V-T) for the gaseous phase, and on four auxiliary equations for the saturated liquid density, the specific thermal capacity at the ideal gas state (null pressure), the liquid saturation pressure as function of temperature, and the Clausius-Clapeyron equation. All other properties are derived from this basic set using their thermodynamic definitions. Of the many equations of state available (see Martin 1967, and an excellent review by Bejan 1988), the Martin-Hou (1955) equation was chosen as it is the most widely used in the refrigeration industry. The form of the Martin-Hou equation adopted,

\(^1\) Enthalpy, entropy, enthalpy of vaporization, specific thermal capacities at constant pressure and volume, isentropic exponent and acoustic velocity for the vapour phase.
Eq. [3.188], is a modified version of the original equation, as published by Chan and Haselden (1981). The parameters of this equation are available for a large number of substances (Downing 1974, Ekroth 1979, Chan and Haselden 1981, Wilson and Basu 1988, Basu and Wilson 1989).

\[
P = \frac{\mathcal{R} T}{v - b} + \sum_{j=2,4} A_j + B_j T + C_j \left( \mu e^{-\frac{kT}{T_c}} + \frac{v}{T^3} \right) + \sum_{j=3,5} A_j + B_j T + C_j e^{-\frac{kT}{T_c}} \left( \mu e^{-\frac{kT}{T_c}} (v - b) \right)^{j} + \frac{A_6 + B_6 T + C_6 e^{-\frac{kT}{T_c}}}{e^{\alpha v} (1 + C' e^{\alpha v})} \]

[3.188]

The consistency in the calculation of the thermodynamic properties is ensured through the adoption of convenient reference conditions. Properties of the liquid phase are required only very close to saturation, so they are approximated by the saturated liquid properties at the liquid temperature.

**Transport properties**

The transport properties of refrigerants both in liquid and gaseous phases are based on a large set of published data as shown in Table 3.IV. In some cases, such as vapour dynamic viscosity and thermal conductivity, existing models were adopted. In all other cases empirical equations were adjusted to the data.
Table 3.IV - Reference sources for the transport properties of refrigerants.

<table>
<thead>
<tr>
<th>Property</th>
<th>References</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity of liquid</td>
<td>Touloukian et al. (1970), Reid et al. (1977), Yata et al. (1984), Altunin et al. (1987), Shankland et al. (1988)</td>
</tr>
</tbody>
</table>

3.6.2 Properties of Humid Air

*Thermodynamic Properties*

The thermodynamic properties of atmospheric air - humid air - are calculated from a virial equation of state (Himmelblau 1960, Mason and Monchick 1963,

**Transport Properties**

The transport properties of humid air are calculated from various works as shown in Table 3.V.

**Table 3.V - Reference sources for the transport properties of humid air.**

<table>
<thead>
<tr>
<th>Property</th>
<th>References</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity</td>
<td>Mason and Saxena (1958), Cheung et al. (1962),</td>
</tr>
<tr>
<td></td>
<td>Theiss and Thodos (1963)</td>
</tr>
<tr>
<td>Dynamic viscosity</td>
<td>Wilke (1950), Kestin and Whitelaw (1963),</td>
</tr>
<tr>
<td></td>
<td>Lo et al. (1966)</td>
</tr>
<tr>
<td>Prandtl number</td>
<td>-</td>
</tr>
<tr>
<td>Schmidt number</td>
<td>-</td>
</tr>
<tr>
<td>Diffusivity of water vapour in air</td>
<td>Rossie (1953)</td>
</tr>
</tbody>
</table>

**3.6.3 Properties of Water**

The properties of water are necessary for both the liquid and solid phases. They are calculated from a variety of sources as shown in Table 3.VI. In some cases equations were adjusted to the data, but in most cases the equations published in those sources are used.
Table 3.VI - Reference sources for the properties of water.

<table>
<thead>
<tr>
<th>Property</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamic viscosity</td>
<td>Theiss and Thodos (1963), Grigull et al. (1968), Touloukian et al. (1975), Kestin et al. (1978)</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>Theiss and Thodos (1963), Yonko and Sepsy (1967), Brian et al. (1969), Touloukian et al. (1970), Biguria and Wenzell (1970), Hayashi et al. (1977)</td>
</tr>
<tr>
<td>Surface tension</td>
<td>Grigull et al. (1984)</td>
</tr>
<tr>
<td>Specific thermal capacity</td>
<td>Touloukian and Makita (1970), Kell (1975)</td>
</tr>
<tr>
<td>Density</td>
<td>Kell (1975), Brian et al. (1969), Biguria and Wenzell (1970), Hayashi et al. (1977)</td>
</tr>
<tr>
<td>Saturation pressure of liquid</td>
<td>Hyland and Wexler (1983 b)</td>
</tr>
<tr>
<td>Saturation pressure of frost</td>
<td>Hyland and Wexler (1983 b)</td>
</tr>
<tr>
<td>Enthalpy of vaporization</td>
<td>Hyland and Wexler (1983 b)</td>
</tr>
<tr>
<td>Enthalpy of sublimation</td>
<td>Keenan et al. (1969), Zemanski and Dittman (1981)</td>
</tr>
<tr>
<td>Prandtl Number</td>
<td>Basu et al. (1979)</td>
</tr>
</tbody>
</table>
3.7 General Solution Method

The models of the individual components described represent local solutions. They must now be integrated in a global solution through the boundary conditions to the model of each component. The boundary conditions on the refrigerant side are essential to this integration.

![Diagram of the enhanced simulation model](image-url)

Fig. 3.67 - Information flow diagram of the enhanced simulation model.

The main hypothesis of the general solution method is that whatever the source and sink fluids, and component types, the refrigerant boundaries of each
component require always the same information from the other components. This has the important consequence that the general solution method here proposed shall be independent of the source and sink fluids, and from the kind of components building up the machine\(^1\). A complete information flow diagram is depicted in Fig. 3.67 with, in this case, air and water as source and sink fluids, respectively.

Although the simulation strategy is not readily apparent in the information flow diagram, a closer analysis shows that the direction of the arrows on the refrigerant side depart from the compressor block, to converge onto the expansion device block. This means that the calculation starts by the compressor and progresses in two directions: through the condenser on one side and through the evaporator on the other, taking into account the piping, to the expansion device, at the exit of which the global check for convergence is made. The solution method is a little more involved, as depicted in the flow diagram of Fig. 3.68. First the high pressure side is calculated, iterating on the temperature difference in the condenser\(^2\), to the inlet of the expansion device, piping included. The refrigerant subcooling both at the condenser outlet (complete condensation) and at the expansion device inlet, must be greater than zero, and is assigned an interval (0-5 K) where the solution is acceptable. Then, the low pressure side is calculated, iterating on the temperature difference on the evaporator\(^3\), to the outlet of the expansion device including the piping. The expansion device is now calculated and the convergence is verified at its exit.

---

\(^1\) So stated, this is the objective. There are a number of limitations that hinder the generality of the solution procedures proposed here. The main one is that the compressor speed cannot be adjusted during the simulation.

\(^2\) In fact the temperature difference considered here is that between the refrigerant saturation temperature at compressor outlet, and the outlet temperature of the secondary fluid in the condenser which is one of the data characterizing the working point.

\(^3\) Here too, this temperature difference is between the refrigerant saturation temperature at the compressor inlet, and the source fluid inlet temperature, which is as well one of the variables characterizing the working point.
Fig. 3.68 - Flow diagram describing the algorithm of the global solution method.
Finally, the most external loop requires that the calculated and assumed compressor inlet superheating agree within the limits of convergence.

The initial conditions, temperature differences at the condenser and the evaporator, are determined as functions of the temperature lift (temperature difference between sink and source). These functions are depicted in Fig. 3.69.

Fig. 3.69 - Initial values of the temperature differences at the condenser and the evaporator.

The solution to the high pressure side - first loop in the flow diagram of Fig. 3.68 - is obtained with the Brent algorithm (Press et al. 1989), which requires the specification of an interval containing the solution. The upper bound of this interval is determined by the function in Fig. 3.69, and the lower bound is fixed. The initial value of the temperature difference on the low pressure side $\Delta T_{ev}$ is obtained from function as shown in Fig. 3.69. Values for later iterations are
obtained from the previous iteration by calculating an average global conductance of the evaporator 

\[(KA)_i = \frac{\hat{Q}_{ev,i}}{\Delta T_{ev,i}} \]  

[3.189]

and then using the last calculated enthalpy difference between the evaporator and the thermostatic expansion valve outlets to determine the new temperature difference 

\[\Delta T_{ev,i+1} = \frac{M_{r,i} (h_{ev,\text{out},i} - h_{TXV,\text{out},i})}{(KA)_i} \]  

[3.190]

The typical number of iterations required on the high pressure side is seven, whereas that for the whole is three. The iteration on the condenser subcooling, Fig. 3.68, has only been necessary in a very few cases of those tested. The running times of the simulation program are difficult to give with accuracy. They depend on too many variables, though some have stronger influence than others. The 'heaviest' model of all is the plate finned-tube evaporator. The complexity and number of refrigerant circuit layouts affect strongly the running time required. Another strong influence also in the evaporator, is the occurrence of air dehumidification, which increases the number of iterations necessary for convergence at each step. The typical running times observed with the machine configuration described in Chapter 5, range from two to fifteen hours on a high end AT\textsuperscript{®} class Personal Computer\textsuperscript{®1}.

\[1\] i80386 and i80387\textsuperscript{®} processors running at 33 MHz, using memory and disc caches, and most disc operations with a virtual RAM disc.
Leer - Vide - Empty
4. DATA REQUIRED BY THE ENHANCED SIMULATION MODEL
The mathematical models described in the foregoing chapter require various types of characteristic data for their useful application. While for some models the geometry and material properties suffice, for others operational characteristics are also necessary. Models that require operational characteristics are by their nature less flexible than those that do not. In fact, the decision to use operational characteristics is a response to the need for simplicity in the cases where the mathematical description of the component's function from the basic principles alone would result in very complex models. On the other hand, the most complex components (compressor, thermostatic expansion valve, etc.) are not tailored for each potential customer, but are designed as series which range of operation covers the broad range of applications possible. In these cases, the optimization of the components by the manufacturers, does not respond to specific application demands, but rather to market segments representing the largest part of the customers of the brand. Therefore, little would be gained by establishing very detailed models for these components. A case where it would still be rewarding is when the operational characteristics may be adapted in order to match the other components in the machine. These considerations are basic in determining what kind of model to use or develop, and consequently the data required. Table 4.1 lists some of the components (in a broad sense) used in heat pumps, and shows as well the sort of data required, the kind of model, the sources of the data for each component, and whether a model is currently available in the ENHANCED model framework.

4.1 Data Resources

Some terms in Table 4.1 require, perhaps, clarification; Basic Principles, suppose the application of the physical laws governing the phenomena at hand, with a few parameters obtained from closure relationships (similarity), normally derived from experimental data. Empirical Model, means some kind of interpolation formula adjusted to the manufacturers data (catalog) or operational data. Design, refers to data specified on a trial basis, for example when checking design changes.
Table 4.1 - Summary of component types and their models, data required and where to get the data from.

<table>
<thead>
<tr>
<th>COMPONENT</th>
<th>Type(s)</th>
<th>Kind of Model</th>
<th>Sort of Data</th>
<th>Source of Data</th>
<th>Avail.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reciprocating Compressor</td>
<td>Hermetic Semi-Hermetic Open</td>
<td>Empirical Basic Principles</td>
<td>Operational Geometry</td>
<td>Manufacturer Measurements Literature</td>
<td>Yes</td>
</tr>
<tr>
<td>Condenser</td>
<td>Coiled-Coaxial</td>
<td>Basic Principles</td>
<td>Geometry</td>
<td>Manufacturer Measurements Design</td>
<td>Yes</td>
</tr>
<tr>
<td>Expansion Device</td>
<td>Thermo. Expansion Valve</td>
<td>Basic Principles Empirical</td>
<td>Geometry</td>
<td>Manufacturer Measurements Design</td>
<td>Yes</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Plate-Finned Tube Coil Air Heated</td>
<td>Basic Principles</td>
<td>Geometry</td>
<td>Manufacturer Measurements Design</td>
<td>Yes</td>
</tr>
<tr>
<td>Refrigerant Piping</td>
<td>-</td>
<td>Basic Principles</td>
<td>Geometry Fittings</td>
<td>Manufacturer Design</td>
<td>Yes</td>
</tr>
<tr>
<td>Liquid Pipes</td>
<td>-</td>
<td>Basic Principles</td>
<td>Geometry Fittings</td>
<td>Manufacturer Design</td>
<td>Yes</td>
</tr>
<tr>
<td>Air Ducts</td>
<td>-</td>
<td>Basic Principles</td>
<td>Geometry Fittings</td>
<td>Manufacturer Design</td>
<td>Yes</td>
</tr>
<tr>
<td>Fans</td>
<td>Centrifugal Axial</td>
<td>Basic Principles Empirical</td>
<td>Operational Geometry</td>
<td>Manufacturer Measurements Design</td>
<td>Yes</td>
</tr>
<tr>
<td>Liquid Pumps</td>
<td>Circulator</td>
<td>Basic Principles Empirical</td>
<td>Operational</td>
<td>Manufacturer Measurements</td>
<td>Yes</td>
</tr>
<tr>
<td>Fluids</td>
<td>Refrigerants Lub. Oil Water Air Brines</td>
<td>Basic Principles Empirical</td>
<td>Misc. Solub. Composition</td>
<td>Literature</td>
<td>Yes Yes Yes No</td>
</tr>
</tbody>
</table>

The components listed in Table 4.1 represent those used in air-to-water heat pumps, though they would be the same in an air-cooling device for comfort purposes, for example. The nature and availability of the data would not be much different either if, instead of an air-to-water heat pump, another type of heat pump was considered: The kinds of heat exchangers, which would be the only
components that may vary, are in the list. Naturally the models would be others, but the data would be the same. In a number of cases, data may as well be obtained from measurements, though most of the potential users of the models described are not equipped to carry out those measurements. Despite this consideration, in the current status of the models reported in Chapter 3, there is at least one case where measurements are still required. This is the thermostatic expansion valve, for which the manufacturers data fall quite short of the data necessary to simulate this component properly (Conde and Suter 1992). The experimental procedure to determine the parameters of the thermostatic expansion valve model is described in detail in Appendix D.

The kind and volume of data required by the models varies from component to component, and their handling for simulation purposes cannot be separated from simulation process itself. Therefore, some kind of data organization and management is necessary. We come to the use of data structures, and cannot avoid relating them to computer programming techniques and languages.

4.2 Organization of the Components' Data - Data Structures

In organizing the components' data, the concept of a computer program\(^1\) for heat pump simulation has been always present, and determined the details of this organization. A schematic representation of this concept is depicted graphically in Fig. 4.1. The data for components of the same type are associated in so-called data banks. Each data bank has a dedicated routine, data bank manager, that is capable of deleting, editing and generating component items in the data bank. Most of the data originate in manufacturers' catalogs, usually given as graphs or tables, and are not in the form required by the models. The data bank managers include easy to use input sections, and options to transform the data, to select a default data set, or get one of the data sets existing in the corresponding data

\[\text{1 The program's name as referred in the following, is HPDesign.}\]
bank. This means that, in general, all the data required by the model can put in the right form for the data bank, with the data available to an engineer working by a refrigeration or heat pump OEM\(^1\).

\[\text{Fig. 4.1 - Concept for heat pump simulation with the ENHANCED simulation model, and for component data handling, as applied to an air-to-water heat pump.}\]

The data structures used to organize, store, and handle the data in the data banks, assemble data of different nature (floating point, integer, alphanumeric, boolean, arrays, etc), taking advantage of the possibilities offered by the modern,\(^1\) Original Equipment Manufacturer.
structured computer programming languages. It should be remarked that the
programming language used is a general purpose computer programming
language (Pascal). The data structures used in HPDesign are presented in
Appendix G for all components in Table 4.1.

All the discussion above regards the cases where the data is readily obtainable,
either from the manufacturers or from experiments easy to perform. There are,
however, cases where it is desirable to study the application of a component for
conditions (new refrigerants for example) for which no catalog data exist. The
typical case, at least theoretically interesting, is the reciprocating compressor. The
model for this component is largely based on manufacturers data, so the question
here is how to derive the compressor characteristics for a new refrigerant from
data for an existing one.

4.3 Conversion of Compressor Model Parameters

The recent international agreements (Vienna 1985, Montréal 1987, London 1990),
on the phasing out of some of the most widely used refrigerants, require a rapid
response of the industry to manufacture the new components necessary, and to
generate new data for existing ones, which may eventually be used with the new
fluids. This applies in particular to the compressors used in refrigeration machines
and heat pumps.

The performance data of a reciprocating compressor are fluid dependent, and are
generally obtained from calorimetric tests. This means that even if the design is
maintained, these components should be tested again. In such cases, methodolo-
gies to adapt the data from one fluid to another, once proved their validity, are
economically important. There are problems associated with fluid replacement

\footnote{In responding to this question no account is taken of material compatibility. It is assumed
that the parts would perform in an equally safe manner with the replacing refrigerant, as
they do with the one they were designed for. In the reality it is not necessarily so.}
that cannot be addressed without some testing effort, e.g. compatibility of insulation (electric) and sealing materials, lubricating oil, etc. These however, do not play a prime role in the thermodynamic performance of a compressor. Hence, departing from the basic assumption that the compatibility problems are solved, a method to derive numerically the compressor thermodynamic performance data for new operating fluids, from those data known for existing ones, is discussed in the following. As a means to check the method, the results of a conversion from data for HCFC22 to CFC12 are compared with manufacturers data.

The losses during the compression process may be considered as a fraction of the work done on the vapour

\[ T \, ds = \sigma \, P \, dv \]  

[4.1]

So the real exponent of compression (polytropic) may be expressed as a function of the exponent for the reversible (isentropic) process (Bošnjaković 1988, Suter 1988).

\[ \frac{\kappa - 1}{\kappa} = (1 + \sigma) \frac{\gamma - 1}{\gamma} \]  

[4.2]

It seems reasonable to assume that \( \sigma \) is independent of the working fluid, provided that similar conditions \( \psi \) and \( P_s^* \) are considered:

\[ \sigma_1 \bigg|_{\psi, P_s^*} \equiv \sigma_2 \bigg|_{\psi, P_s^*} \]  

[4.3]

Hence, the polytropic exponent \( \kappa_2 \) for the replacing fluid is calculated from that of the original, and the thermodynamic properties of both.

---

1 Performance data for compressors operating with ozone safe refrigerants were not available for the compressor brand considered.
Furthermore, the effective volumetric efficiency defined as

$$\eta_{V, eff} = 1 - e_{eff} \psi^{1/\kappa} - 1$$

accounts for the volumetric losses due to both reexpansion and leakage. The heat transfer between the cylinder wall and the refrigerant vapour plays a second order role in the volumetric losses (Rottger 1975). So, under the same constraints as above, an effective leakage area may be defined that is independent of the fluid. The isentropic and volumetric efficiencies for a replacing fluid, may be calculated from the assumptions and constraints above, by the application of the conservation equations to the compression process in reciprocating compressors.

4.3.1 Isentropic Efficiency

The isentropic efficiency is the ratio of the work done on the refrigerant vapour in a reversible (isentropic) process to that done in a real (polytropic) process.

$$\eta_s \equiv \frac{\Delta h_s}{\Delta h_{act}} = \frac{P_s v_s \gamma}{\gamma - 1} (\psi^{\frac{\gamma - 1}{\gamma - 1}} \left(1 - \frac{\gamma - 1}{\gamma - 1})\right)$$

$$= \frac{P_s v_s \kappa}{\kappa - 1} (\psi^{\frac{\kappa - 1}{\kappa - 1}} \left(1 - \frac{\kappa - 1}{\kappa - 1})\right)$$

The mean value of $\gamma$ for the compression process, is calculated from the solution for $\gamma$, of
\[
\Delta h_s = P_s v_s \frac{\gamma}{\gamma - 1} (\psi - 1) \tag{4.7}
\]

The polytropic exponent of the actual compression process \( \kappa \) involves the solution for \( \kappa \) of

\[
\frac{\kappa - 1}{\kappa - 1} (\psi - 1) = \frac{1}{\eta_{s,1}} \frac{\gamma - 1}{\gamma - 1} (\psi - 1) \tag{4.8}
\]

Fig. 4.2 - The polytropic exponent as function of the isentropic efficiency at various compression ratios. The curves suggest numerical difficulties when calculating \( \kappa \).

The value of \( \eta_{s,1} \) is obtained from manufacturers data for the original fluid. Fig. 4.2 shows the dependency of \( \kappa \) upon the RHS of Eq. [4.8]. When \( \kappa \) is
known for the original fluid, it is easy to derive the isentropic efficiency for the replacing fluid from the assumptions above.

\[
\eta_{s,2} = \frac{\frac{\gamma_2 - 1}{\gamma_2 - 1} (\psi)_{\gamma_2} - 1}{\frac{\kappa_2 - 1}{\kappa_2 - 1} (\psi)_{\kappa_2} - 1}
\]  

[4.9]

4.3.2 Volumetric Efficiency

The fraction of the volumetric displacement \( D \) corresponding to the volumetric losses, expressed at the suction port conditions, is

\[
\varepsilon D \left( \psi^{1/\gamma} - 1 \right) + V_{\text{leak}} = (1 - \eta_v) D
\]

[4.10]

where \( \varepsilon \) is the geometric clearance volume \( V_c / D \), Fig. 4.3. Thus, the effective volumetric efficiency may be expressed as

\[
\eta_{v,\text{eff}} = 1 - \left[ \varepsilon \left( \psi^{1/\gamma} - 1 \right) + \frac{V_{\text{leak}}}{D} \right]
\]

[4.11]

The leakage volume \( V_{\text{leak}} \) is a function of the density of the vapour, of the area of the leakage gap, of the mean velocity of leakage, and of the time during which leakage occurs.

---

1 Leakage occurs in a reciprocating compressor due to delays in closing of the suction and discharge valves, and between the piston or piston rings and the cylinder wall (piston blow-by).
The area of the leakage gap may be calculated, if the following assumptions are made:

- The leakage gap may be described as an ideal nozzle
- Leakage occurs only when the pressure ratio across the gap is higher or equal to the critical pressure ratio (the leakage flow is sonic)
- The pressure in the cylinder may be described by an ideal compression cycle between the maximum and the minimum pressures of the cycle, Fig. 4.4.

$$V_{\text{leak}} = \frac{\rho(P)}{\rho_s} \overline{A_I} \overline{\bar{u}_I} \Delta \tau_I$$  \[4.12\]
The time during which leakage occurs, is a function of the rotating frequency of the compressor, and of the critical pressure ratio $\psi_{cr}$. A finite difference integration method is used to solve the integral

$$\int \rho \ u \ d\theta$$

where $d\theta$ is calculated from the piston displacement necessary to generate a step increase (decrease) in pressure during the compression (reexpansion).

During the compression, there is leakage from the point where the pressure in the cylinder attains $P_s \ \psi_{cr}$ up to the Top Dead Point (TDP), while during reexpansion, leakage occurs from the TDP down until the pressure attains again $P_s \ \psi_{cr}$ The free volume in the cylinder, corresponding to the pressure at the
current step, is calculated from the law for a polytropic transformation

\[ P_s \left[ (\varepsilon + 1) D \right]^\kappa = P_{\text{step}} V_{\text{cyl}}^\kappa \]  \hspace{1cm} [4.10]

for compression, and

\[ P_{d} (\varepsilon D)^\kappa = P_{\text{step}} V_{\text{cyl}}^\kappa \]  \hspace{1cm} [4.11]

for reexpansion.

\( \theta \) for the step, results from the kinematic equation for the cylinder volume (see Fig. 4.3 for notation).

\[ V_{\text{cyl}} = \varepsilon D + \frac{\pi \phi_{\text{cyl}}^2}{4} \times \\
\left[ R \left( 1 - \cos \theta \right) + L \left( 1 - \sqrt{1 - \frac{L^2}{R^2} \sin^2 \theta} \right) \right] \]  \hspace{1cm} [4.12]

Finally, the area of leakage is

\[ \bar{A}_l = \frac{V_{\text{leak}}}{60 \frac{1}{N} \frac{1}{\rho_s} \frac{1}{2\pi} \int_{\rho} u d\theta} \]  \hspace{1cm} [4.13]

This area of leakage should depend only upon the compression ratio, so that it may be considered independent of the fluid and other working conditions.

As shown in Fig. 4.5, this is only approximately true, but the spread of the \( \bar{A}_l \) values, 14% at the maximum, seems not to have as much influence over the final results. These are depicted in Figs. 4.6 and 4.7, for drive power and refrigeration capacity. These figures compare the performance characteristics derived by the method described here, with those given by the manufacturer, under the same reference conditions (superheating, subcooling).
Fig. 4.5 - Average area of leakage vs compression ratio for a compressor with 3.6 kW nominal drive power.

Fig. 4.6 - Comparison of method's results with the manufacturer's data. Drive Power.
Fig. 4.7 - Comparison of method's results with manufacturer's data. Refrigeration capacity.
Leer - Vide - Empty
5. VERIFICATION OF THE *ENHANCED SIMULATION MODEL*

Experience does not ever err, it is only your judgement that errs in promising itself results which are not caused by your experiments.

Leonardo da Vinci (c. 1510)
The mathematical models described in chapter 3 are based on many sources in the literature. Given the complexity of the phenomena they describe, simplifying assumptions were made to render the models useful. On the other hand, the concept for the simulation of the whole heat pump has been developed anew. Therefore, some sort of verification\(^1\) of the concept, models and assumptions is necessary. For this purpose, a heat pump was installed in the laboratory and instrumented in enough detail to provide the experimental data required. The heat pump is an out-of-the-shelf machine with a heating power of approximately 10 kW at 7 °C/35 °C. The main components of the heat pump are: a hermetic monocylinder reciprocating compressor, a coiled-coaxial condenser with condensation in the annulus, a plate-finned tube coil evaporator, and a thermostatic expansion valve with external equalization. Besides these, there is a liquid receiver, and an oil separator was also installed in the discharge line.

There are limitations in the nature of the model verification that is possible from the experimental data obtained. Although it is claimed that the modelling approach used in the ENHANCED model is flexible enough to handle all types of heat sources and sinks, it is materially impossible to collect experimental data permitting the full verification of that claim. However, the verification that the simulation algorithm works with the selected machine configuration, does suggest that it will work as well with other configurations, provided that the models of the components satisfy the requirements of the ENHANCED model framework (standard interfaces on the refrigerant side). Measured data from other origins, that might enable the verification of the simulation model for other cases, could not be found.

\(^1\) The terms verification, validation and certification, are used as defined in Terminology for Model Credibility by the SCS Technical Committee on Model Credibility, *Simulation*, 32(3), March 1979, 103-104.
5.1 Experimental Setup

5.1.1 General Description

Fig. 5.1 - Schematic representation of the experimental setup.

The schematic in Fig. 5.1 shows the three main fluid loops in the experimental setup, with the most important components and sensors. On the air loop, it is possible to mix the cold return air with the fresh inlet air, to provide for source temperature variation. The air humidity is not controlled. The range of air
temperatures is limited by both the cooling capacity of the cooling coil \textit{HEX-1}, and by the air temperature in the laboratory. The temperature and relative humidity of the air are measured before and after the evaporator. The air flow velocity is measured in the return leg of the channel with a pitot tube, at a suitable position\(^1\), after the flow straightner \textit{FS-1}. The air flow rate varies with the amount of recirculated air. The electric power of the fan driving motor is measured independently of the total electric power demanded by the heat pump. The fan rotation speed is also measured, using a miniature DC generator as tachometer.

The condenser is water cooled. In order to stabilize the condensing temperature, a closed loop was built using the plate heat exchanger, \textit{HEX-2}. \textit{HEX-2} is supplied through the 3-way valve \textit{TV-1}, which permits the control of the cooling water temperature by mixing cold and hot water from the laboratory mains. The water flow rate, in the closed loop, may be varied with the by-pass valve \textit{BV-1}, and is measured with the flowmeter \textit{FM-1}. The inlet and outlet temperatures of the water at the condenser are also measured.

The refrigerant loop, as depicted in Fig. 5.1, shows only the pressure and temperature sensors at the inlet and outlet of the main components (compressor, condenser, expansion device, and evaporator). It also shows the flowmeter in the refrigerant loop, \textit{FM-2}, mounted on the discharge line, and the power transducer measuring the electric power of the compressor. Many more sensors are installed in the refrigerant loop, but are associated with the individual components.

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\(^1\) The pitot tube may be displaced all over the channel cross section, by positioning screws actuated by small electric motors. A position giving a velocity value close to the average in the channel was found by trial and error, and the pitot tube remained in that position for all measurements.
5.1.2 Instrumentation of the Heat Pump Components

5.1.2.1 Sensors

The temperatures in the experimental setup are mostly measured with type K thermocouples. The thermocouples are connected in opposition to a high temperature (50 °C) reference thermocouple of the same kind. The reference temperature is monitored using a Pt100 thermometer, in order to correct for eventual deviations. This method ensures a precision better than ±0.1 K. Pt100 thermometers are also used for surface temperature measurements (compressor shell). The precision of these sensors is better than ±0.02 K. The four wire technique is used to connect all Pt100 to the data acquisition system.

The pressures are measured with absolute pressure transducers (reference vacuum) with a precision better than ±0.1% Full Scale (FS). On the low pressure side of the heat pump, the range of the transducers is 0..20 bar, whereas in the high pressure side it is 0..50 bar. The effect of temperature is hardware compensated in the range of temperatures the transducers are submitted to in this application. Pressure transducers are also used in the flowmeters, in this case differential pressure transducers. The flowmeters for the refrigerant and water are orifice flowmeters. The precision of the differential pressure transducers is better than ±0.25% FS. The precision of the flow measurement itself is better than ±5% for refrigerant and water. The precision of the differential pressure transducer used with the pitot tube in the air channel is better than ±1% FS, for a range of 0..25.4 mmCW.

A miniature DC generator is used as tachometer measuring the fan rotation speed. It was calibrated with a precision mechanical tachometer. Its precision should be better than ±1%.

The electrical power demanded by the whole heat pump, including auxiliaries,
and that of the fan alone, are measured with three phase power transducers of the precision class 0.25, and a range of measurement 0..800 W. The power transducer for the whole machine required current transformers in order to be in that range.

The relative humidity transducers have a precision better than ±2%. These transducers need to be periodically calibrated (once a week).

In order to monitor the distribution of refrigerant mass in the machine, a level meter was built in the liquid receiver. It operates as an electric capacitor having the refrigerant as dielectric. The uncertainty of the method could not be verified. It gives information only on the refrigerant mass in the receiver. Influences such as the presence of lubricating oil, and of an eventual film of liquid on the walls of the capacitor cannot be excluded.

5.1.2.2 Sensor Distribution

The locations of the temperature and pressure sensors in the refrigerant lines are shown schematically in Fig. 5.1. The positions of the temperature sensors in the compressor are illustrated schematically in Fig. 5.2. There are eight Pt100 thermometers applied onto the surface the compressor shell at three different levels, Fig. 5.2. These measurements permit an estimation of the external losses (gains) through the shell. The refrigerant temperatures are measured within the shell, in the oil heater, and in the discharge line. The temperature of the lubricating oil in the sump is measured as well. The locations of the thermocouples in the condenser are depicted in Fig. 5.3. The coiled-coaxial condenser is shown as straight coaxial to permit a better visualization of the sensor positions. Fig. 5.3 also shows the position of the thermocouples in the circumference. This position was chosen in order to measure preferably the vapour temperature.
Fig. 5.2 - Position of the temperature sensors on the hermetic reciprocating compressor.

Fig. 5.3 - Longitudinal and circumferential positions of the temperature sensors on the coiled-coaxial condenser.
The positions of the thermocouples on the plate-finned tube evaporator are shown in Fig. 5.4 as black dots. Details of the thermocouple mounting are also depicted. The temperature is measured at the outlets of all the seven parallel circuits. Besides this, the refrigerant temperatures in the last tube of each circuit in each tube layer are measured as well for circuits one and four, counted top down.

Fig. 5.4 - Positions of the thermocouples on the evaporator, and detail of the thermocouple mounting in the tubes.
5.1.3 Data Acquisition System

The data acquisition system, schematically shown in Fig. 5.5, consists of a multiplexer (HP3497A®1) with 100 channels, a digital voltmeter, DVM, (HP3497A-OPT001®1), and a controller computer (HP9000/300®1). These three instruments communicate through an IEEE 488 (HP-IB®1) interface. The 75 sensors installed were scanned in series, once a minute, for sixty consecutive times per run, and their readings stored in binary form. These binary voltage readings were later converted to the respective physical magnitudes (temperatures, pressures, 

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1 Trade-Mark of Hewlett-Packard Co.
flow rates, etc.), stored as character (ASCII) strings, and exported to a MS-DOS®1 operated computer for treatment and analysis. The DVM readings were taken at 5 1/2 digits, using the highest possible resolution of the instrument, providing accuracies of 0.007% for voltage readings, and of 0.032% for resistance measurements. This means that the measurements with Pt100 thermometers are affected by the accuracy of the DVM, whereas those with thermocouples, and higher voltage output sensors are not.

5.1.4 Uncertainties in the Measurements

![Comparison of the condenser thermal power](image)

Fig. 5.6 - Comparison of the condenser thermal power calculated from the refrigerant and from the water side. The error bars show both the uncertainty of the measuring system, and of the steady-state testing conditions.

1 Trade-Mark of Microsoft Corporation, Inc.
The measurements made with the experimental setup were carried out under steady state conditions. Therefore, those where a significant amount of frost formed onto the evaporator surface must be discarded, as the continuous build-up of the frost layer promotes a steady variation, in time, of all the magnitudes measured. The rate of sensor scanning is of the order of magnitude (1 min) of the response time of the observed system, and only one measurement method is used. Hence, the measurements are single-sample measurements, for which the representative value is the mean (Kline and McClintock 1975, Taylor 1982, Moffat 1988).

![Fig. 5.7 - Comparison of the evaporator thermal power calculated from the refrigerant and from the air side. The error bars show both the uncertainty of the measuring system, and of the steady-state testing conditions.](image-url)
The interest of this analysis lies in finding the uncertainties resulting for magnitudes calculated from various different measured quantities, particularly flow rates and energy balances. The energy balances of the condenser and of the evaporator may be compared for both sides of the exchangers, and as they require various measured variables, provide a good indication of the overall uncertainty of the measured data. These comparisons are depicted in the Fig.s 5.6 and 5.7 for the condenser and evaporator, respectively. Most of the data fall within a band of ±10% of the reference, taken as the refrigerant side. Particularly for the air side in the evaporator, a considerable scatter of the data is observed. On the other hand, the data for the condenser show a lower scatter, but show some bias of the data for the water side, in relation to the refrigerant side.

In general, it is concluded that the data are consistent, though the spread is due not only to random variations in sensor output, but to instabilities in the steady-state of the machine operation as well.

5.2 Comparison of Simulation Results with Measurements

For this comparison, the values of the independent variables - air inlet temperature, relative humidity and pressure, the water outlet temperature and flow rate, and the ambient temperature - were taken from the measurements and used as input variables to the simulation program HPDesign, which implements into computer code the ENHANCED simulation model. I first compare and discuss global results, and then those for selected local variables.

5.2.1 Global Results

The simulation results and the measurements agree fairly well for most of the global characteristics: Heating COP, total electric power, condenser heating rate, evaporator cooling rate, and refrigerant flow rate. Some points show a relatively
large spread that, as said before, reflects not only random fluctuations of the sensor readings, but instabilities of the steady state regime of the machine as well. The presence of the many sensors in the refrigerant stream, and of an oil separator in the discharge line, lowered the performance of the experimental heat pump. Earlier simulation results, that did not account for the presence of those devices, overpredicted all the magnitudes depicted in Fig.s 5.8 through 5.12. When the effects of those devices are considered the simulation predicts very well the measurements, as shown in the following.

Fig. 5.8 - Comparison of measured and calculated heating COP.

Fig. 5.8 compares the heating COP values from the simulation and from the measurements. Both include the power demand by the auxiliaries (fan and circulator pump).
Fig. 5.9 depicts the measured and calculated total electric power. As before the power of the auxiliaries fan and circulator pump is also included.

![Graph showing comparison of calculated and measured total electric power]

**Fig. 5.9 - Comparison of calculated and measured total electric power.**

The simulation results show an excellent agreement with the measurements in the whole range of the experiments.
Fig. 5.10 shows a comparison of the calculated and measured condenser heating rates, for the refrigerant side. A comparison of the measurements from the water side with those of the refrigerant side is depicted in Fig. 5.6.

![Fig. 5.10](image_url)

**Fig. 5.10 - Comparison of calculated and measured condenser heating rates.**

The data from measurements show a relatively large spread. The simulation results agree well with the measurements.
Fig. 5.11 compares the evaporator cooling rate obtained from the simulation and from the measurements, for the refrigerant side. See Fig. 5.7 for a comparison of the measurements with the air side.

![Graph comparing Simulation and Measures for Calculated Evaporator Cooling Rate](image)

**Fig. 5.11 - Comparison of the evaporator cooling rates obtained from the simulation and from the measurements.**

The same good agreement between the simulation results and measurements of the previous figures is found in this case as well.
The calculated refrigerant flow rates agree well with those measured, as shown in Fig. 5.12. Contrary to the previous figures, the refrigerant flow rate is slightly underpredicted at the low flow rates, and overpredicted at the higher ones.

**Fig. 5.12 - Comparison of calculated and measured refrigerant flow rates.**

The comparison between the measured and the calculated global parameters shows a good agreement in general.
5.2.2 Local Results

The comparison of the measurements with the simulation results for local variables shows that the agreement is not equally good for all points in the cycle. I first discuss a comparison of the refrigerant pressure values from the simulation and from the measurements at four points in the cycle, see Fig. 5.1. Then, I compare simulation results within the condenser and the evaporator.

Fig. 5.13 shows the calculated and measured values of the refrigerant pressure at the compressor inlet. The simulation and the measured data agree fairly well.

![Graph of measured vs. calculated compressor suction pressure](image)

**Fig. 5.13** - Comparison of calculated and measured refrigerant pressures at compressor inlet.
Fig. 5.14 depicts a similar comparison for the pressure values at the compressor outlet. The simulation results agree well with the measurements. There is a tendency to flatten out at the higher values of the pressure. This may be due to the action of the internal safety system of the compressor, a check valve between discharge and suction, which reacts by releasing high pressure vapour to the suction side.

Fig. 5.14 - Comparison of measured and calculated values of the compressor discharge pressure.
This same flattening shows up again in Fig. 5.15, at the inlet of the thermostatic expansion valve. From the comparison of both figures 5.14 and 5.15, it is concluded that the pressure drop calculated for the condenser is slightly lower than the measured values, as the pressure at the TXV inlet tends to be higher than the measured values.

![Graph showing comparison of calculated and measured refrigerant pressures at the inlet to the thermostatic expansion valve.](image-url)

**Fig. 5.15** - Comparison of calculated and measured refrigerant pressures at the inlet to the thermostatic expansion valve.
Fig. 5.16 represents the observed and calculated values of the refrigerant pressure at the outlet of the thermostatic expansion valve. A good agreement between calculated and measured values is found for most observations.

![Graph showing comparison of measured and calculated TXV outlet pressures](image)

Fig. 5.16 - Comparison of the measured and calculated refrigerant pressures at the expansion valve outlet.
Fig. 5.17 shows a comparison of all temperatures measured in the condenser with calculated values. The water outlet temperature (value at 0% length) was input to the simulation program. The condensation temperature in this particular run is slightly overpredicted by the simulation (~3 K).

![Graph showing comparison of measured and calculated temperatures in the condenser.]

Fig. 5.17 - Comparison of measured and calculated temperatures in the condenser.
Fig. 5.18 compares the temperatures measured in two of the seven parallel circuits in the evaporator, with the corresponding calculated values. The measured values of the air temperature are also depicted, but they are global values (inlet and outlet).

Although there is a good agreement at the inlet and outlet for both air and refrigerant, measured and calculated values seem not to be obtained by
corresponding processes. Whereas the measured values of the refrigerant temperature are lower than the calculated ones for the first half of the heat exchanger, they become much higher in the second half. This is coherent with the same temperature values at the extremities of the circuits.

The tendency of the calculated refrigerant temperature to attain rapidly the saturation value is caused by the assumption of thermodynamic equilibrium, in particular in the post-dryout region. The dryout of the upper part of the tube inner surface occurs as soon as a stratified flow pattern is established. On the other hand, complete dryout occurs when the liquid fraction cannot wet the bottom of the tube anymore. Whether a spray flow pattern establishes itself after dryout depends upon the mass and heat fluxes.

The nonverification of the assumption of thermodynamic equilibrium in the post-dryout region does not explain the other differences shown in Fig. 5.18. These suggest that the measured temperatures were taken in the vapour phase region, and further that the vapour temperature was not the saturation temperature. As Fig. 5.4 shows, the tip of the thermocouples mounted in the evaporator tubes are approximately on the centerline. This means that for void fractions higher than 50% in stratified flow, or for annular flow, the thermocouples will sense predominantly the vapour phase temperature. The flow pattern for most of the length of the circuits is stratified flow, with in some cases short lengths in annular flow, Fig.s 5.19, 5.20, for the typical mass flux of 100 kg/m² s. The void fractions are higher than 80% for the whole circuit. The other condition for nonequilibrium in the stratified flow regime (Bonn et al. 1980, Müller-Steinhagen and Schlünder 1984) is a wall conductance less than 1 W/K. In the evaporator considered the wall conductance is approximately 0.15 W/K. These considerations show that the assumption of thermodynamic equilibrium is in general not verified. The important question at this point is whether the consequences of that assumption have a significative impact on the value of the model. The answer must consider three additional questions:
Do the results obtained with the equilibrium assumption affect the general predictions of the model for the whole machine?

Are there methodologies available to establish a fully nonequilibrium model of the vaporization process in these complex geometries, or may they be developed with reasonable effort?

Is the simulation of the whole machine still interesting (computation time, costs, etc.) with an evaporator model much more complex than the model described, which cannot be labeled a simple one?
Fig. 5.20 - Heat transfer coefficient for boiling inside tubes, and vapour mass quality as function of the length.

I believe that the value of the model is affected by the assumption of equilibrium, though there are currently no facts to justify this. A model that describes the physical phenomena with the less assumptions is more likely to produce better results in general. The answer to the first additional question is a qualified yes: The heat transfer in a nonequilibrium model would be worse in some regions and better in others, than in the equilibrium one.

1982, Hein and Kastner 1982, Köhler 1984, Müller-Steinhagen and Schlünder 1984, Hein and Köhler 1986, Katsaounis 1986, 1988, Varone Jr. and Rohsenow 1986, Remizov et al. 1987, Becker et al. 1988, Hwang and Moallemi 1988, Rohsenow 1988, Jones, Jr. 1989) apply in general to water, with a few verifications with CFC12. They were obtained for mass and heat fluxes much higher than usually observed in air heated refrigerant evaporators, and for simpler geometries (straight tubes). I believe, however, that it is possible to extend those methodologies to evaporators of the type considered here, though the effort required may be of the order of magnitude of that invested in the current model.

Whether such a complex model will still be interesting in the simulation of whole machines is not certain, though it is certainly interesting in the simulation for design of the evaporator alone.

It is important to stress in conclusion, that there is plenty of interesting research work on this subject to be done. The simulation model of the evaporator remains valid in its strategy and solution algorithms, and represents a good starting point for the development of a nonequilibrium model of the vaporization process.

A practical conclusion from the simulation results in Fig.s 5.19 and 5.20 is that any method that might help keeping the tube wall wetted for longer stretches would reduce the evaporator size significatively. In particular such methods would be effective in the regions where stratified flow is known to occur, lower quality regions, but would be undesirable in those regions where annular flow might form from the hydrodynamics of the flow. It comes to mind here that, for example twisted tape inserts applied selectively, and not to the whole length of the tubes (remember Zahn's observations), would result in a reduction of the evaporator size, with the concurrent reduction in fan power and increase in the COP.
5.3 Case Studies - Influences of Individual Independent Variables

The influence of individual input variables upon the heat pump performance has been studied by simulation using HPDesign. In each case only one variable varies over the range it is expected to cover in typical operating conditions of an air-to-water heat pump for domestic heating purposes, while all the others are kept constant at typical values. The condenser cooling water outlet temperature was varied from 30 °C to 55 °C, the air inlet temperature from -10 °C to 10 °C, the air inlet relative humidity from 10% to 90%, the condenser cooling water flow rate from 500 kg/h to 1600 kg/h. The default values of the independent variables, when not subject of study are:

- Air inlet temperature 5 °C
- Air inlet relative humidity 60 %
- Condenser water outlet temperature 45 °C
- Condenser water flow rate 1170 kg/h
- Ambient temperature 22 °C
- Atmospheric pressure (Zürich Normal) 970 mbar.

5.3.1 Influence of the Air Dry-Bulb Temperature at the Evaporator Outlet

Fig.s 5.21 to 5.26 show the effect of the variation of the air temperature upon the characteristics of the heat pump, with the other operation variables constant.
Air Inlet Dry_Bulb Temperature [°C]

Source Conditions
60 % Relative Humidity
970 mba Air Pressure

Sink Conditions
45 °C Water Outlet Temperature
1170 kg/h Water Flow Rate
22 °C Ambient Temperature

Fig. 5.21 - Heating and cooling COP,s as function of the air temperature at the evaporator inlet.
Fig. 5.22 - The electric power of condenser and fan, and the heating and cooling rates of the condenser and evaporator, respectively, as function of the air temperature at evaporator inlet.
Fig. 5.23 - Variation of the refrigerant flow rate with the temperature of the air at the evaporator inlet.
Source Conditions
60 % Relative Humidity
970 mba Air Pressure

Sink Conditions
45 °C Water Outlet Temperature
1170 kg/h Water Flow Rate
22 °C Ambient Temperature

Fig. 5.24 - Effects of the temperature of the air at the evaporator inlet over the refrigerant temperature at several points in the cycle.
Fig. 5.25 - Effects of the temperature of the air at the evaporator inlet over the refrigerant pressure at several points in the cycle.
Fig. 5.26 - Effect of the temperature of the air at the evaporator inlet on the air flow rate across the evaporator.
5.3.2 Influence of the Water Temperature at Condenser Outlet

Fig.s 5.27 to 5.31 show the effect of the water temperature at the condenser outlet upon the characteristics of the heat pump, with all the other operation variables constant.

![Graph showing the variation of heating and cooling COP with water outlet temperature.]

Source Conditions
- 5 °C Air Inlet Dry-Bulb Temperature
- 60% Relative Humidity
- 970 mbar Air Pressure

Sink Conditions
- 1170 kg/h Water Flow Rate
- 22 °C Ambient Temperature

**Fig. 5.27** - Variation of the heating and cooling COPs with the water temperature at the condenser outlet.
Fig. 5.28 - Effect of the temperature of the water at the condenser outlet on the heating and cooling rates of the condenser and evaporator, respectively, and on the electric power of the compressor and fan.
Source Conditions
5 °C Air Inlet Dry-Bulb Temperature
60% Relative Humidity
970 mba Air Pressure

Sink Conditions
1170 kg/h Water Flow Rate

22 °C Ambient Temperature

Fig. 5.29 - Variation of the refrigerant flow rate with the temperature of the water at the outlet of the condenser.
Fig. 5.30 - Variation of the refrigerant temperature at several points in the cycle, with the temperature of the water at the outlet of the condenser.
Fig. 5.31 - Effect of the temperature of the water at condenser outlet on the refrigerant pressure at several points in the cycle.
5.3.3 Influence of the Air Relative Humidity at the Evaporator Inlet

Fig.s 5.32 to 5.36 show the effect of the relative humidity of the air at the evaporator inlet upon the characteristics of the heat pump, with all other operation variables kept constant. It is concluded that only a slight effect is noticed for relative humidities higher than 60%, at the air temperature considered.

![Graph showing the variation of heating and cooling COP with relative humidity](image)

**Source Conditions**
- 5 °C Air Inlet Dry-Bulb Temperature
- 970 mba Air Pressure

**Sink Conditions**
- 45 °C Water Outlet Temperature
- 1170 kg/h Water Flow Rate
- 22 °C Ambient Temperature

*Fig. 5.32 - Variation of the heating and cooling COPs with the relative humidity of the air at the evaporator inlet.*
Fig. 5.33 - Effect of the variation of the relative humidity of the air at the evaporator inlet upon the heating and cooling rates of the condenser and evaporator, respectively, and upon the compressor and fan electric power.

Source Conditions
- 5 °C Air Inlet Dry-Bulb Temperature
- 970 mba Air Pressure

Sink Conditions
- 45 °C Water Outlet Temperature
- 1170 kg/h Water Flow Rate
- 22 °C Ambient Temperature
Fig. 5.34 - Variation of the refrigerant flow rate with the relative humidity of the air at the evaporator inlet.
Fig. 5.35 - Variation of the refrigerant temperatures at several points in the cycle, with the relative humidity of the air at the evaporator inlet.
Fig. 5.36 - Effect of the relative humidity of the air at the evaporator inlet on the refrigerant pressure at several points in the cycle.
5.3.4 Influence of the Water Flow Rate in the Condenser

Within the limits of technically possible variation of the water flow rate in the condenser 500 to 1600 kg/h, no sensible variation of the heat pump characteristics was noticed. This is explained by the fact that the controlling heat transfer coefficient in the condenser is the refrigerant's. Whereas the typical heat transfer coefficient of the condensing refrigerant is about 800 W/m$^2$K, that of the water lies at about 15 000 W/m$^2$K.

5.4 Limits of Application

It is natural that mathematical models, and especially the computer programs that implement them, are limited in their ranges of application. These limitations have various origins:

The models describe mathematically components of a certain type, configuration and size. Changes in any of these attributes may not be well represented by the models, e.g. compressors requiring external cooling will not be described by a model that does not include such a possibility, and the whole simulated process would be affected by this.

Inadequate size of the heat exchangers (condenser and evaporator) may lead to large temperature differences, in some cases getting out of the range of validity of the equations used in the calculation of the thermophysical properties of the working fluids.

The initial conditions, temperature differences in the heat exchangers, superheating and subcooling, in the computer program HPDesign may eventually have to be adapted, following different machine configurations.

In its current implementation, HPDesign cannot handle situations with more than one component for each function. This excludes the simulation
of multistage, multi-condenser and multi-evaporator systems, as well as systems including a separate desuperheater.

Most refrigerants in current use, and those considered as substitutes with less harmful characteristics to the environment (HFC134a, HCFC22 and HCFC123) may be used in simulations. Other fluids deserving interest in the future, e.g. ammonia and water, cannot be used in the current form of the program modules calculating the thermophysical properties.

Capacity control methods such as compressor cylinder unloading or variable rotation speed are not considered in the current versions of the models, however various constant speeds of the compressor may be handled easily.

Only operating points where stability is warranted may be successfully simulated. On the other hand, a non-convergence to a solution may be taken as an indication for a nonstable operating point.
Caminante, son tus huellas
el camino, y nada más;
caminante, no hay camino,
se hace camino al andar.
Al andar se hace camino,
y al volver la vista atrás
se ve la senda que nunca
se ha de volver a pisar.

6. CONCLUSIONS

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Antologia Poetica
Mathematical models and computer program modules, for a set of typical components of air-to-water heat pumps, have been developed and tested by comparison with experimental data. The simulation framework created to integrate these component models into a model of a complete heat pump, has proved its suitability to the task.

The general solution algorithm has demonstrated its stability under typical, and extreme conditions of application of air-to-water heat pumps. It has not, however, proved successful at simulating random sets of components, that is, components that have disparate characteristics (would not match at all in their expected range of operation). An optimization scheme that might change a selected component, or its characteristics, until it matches the others, would eventually solve this problem. Such an optimizer requires computer power not available with the computer platform currently used.

The effects of the variation of the individual operation variables - source (air) temperature and relative humidity, and sink (water) temperature and flow rate - over their widest expected range of variation (much larger than allowed by the experimental test rig) further confirmed the stability of the general solution algorithm.

The comparison of the experimental measurements with the simulation results shows a good agreement in most cases, both at the global and the local levels. These simulation runs were done using the experimental operation conditions as input. In the particular case of the evaporator, it is concluded that a fully nonequilibrium model for the refrigerant side is desirable if this component is to be improved through simulation.

Some results of the simulations demonstrate that although the models are only intended for steady state behaviour, there are combinations of the values of the
operation variables for which no steady operation could be attained. Again, component modifications, or adjustment of their characteristics, may be tested in order to find out how to bring stable operation to the desired range of the variables. These modifications or adjustments cannot be done automatically at present.

The eventual shortcomings of the tools developed did not hinder them to produce valuable information in the ranges of the operation variables that could not be reached in the experiments. On the contrary, they have predicted effects which, if experimentally verified, will definitely qualify them for the study of conditions difficult to realize even in the most sophisticated test rig. This certainly confirms the value of the simulation approach in the effort to improve the efficiency of vapour compression reverse Rankine cycle machines, as required for the reasons expounded at the beginning of this work.

The experimental observations, Fig. 5.18, suggest that the performance of air-source heat pumps may be easily improved by the use of smaller evaporators. The vaporization of the refrigerant would not be completed in them, but for example in a small and effective heat exchanger in the liquid line. This permits improved performance and simultaneously a cost reduction: A smaller fan and a smaller evaporator, and an additional liquid line heat exchanger. Other methods to provide for a liquid free vapour at the compressor inlet, may as well be considered.

The comparison of the absolute values of the gradients of the various magnitudes with the source and sink temperatures, Fig.s 5.21 + 5.31, shows that the same investment will yield a better improvement when done on the source (evaporator) side than when done on the sink (condenser) side.
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7. FUTURE DEVELOPMENTS

When you make a thing, a thing that is new,
it is so complicated making it that it is bound to be ugly.
But those that make it after you, they do not have to worry about making it. And they can make it pretty, and so everybody can like it when the others make it after you.

Pablo Picasso
as cited by Gertrud Stein
in *Design for the Real World*
Victor Papanek
A mathematical simulation model for any complex system, and the computer program(s) that implement it, are only useful when the hypotheses, methods and assumptions from which they were developed have been thoroughly proved and validated. The *ENHANCED* heat pump simulation model, and the computer program *HPDesign*, were verified in this work for a fraction of the scope ascribed to them. In order to fully achieve the potential usefulness of the concepts and tools described in the previous chapters, further work is necessary in the following areas:

- Modelling and programming
- Experimentation
- Verification and validation.

7.1 Further Developments in Modelling and Programming

The framework developed in the *ENHANCED* simulation model is not bounded by the type of component, kind of operating fluid, and source or sink fluids. It is bounded, however, by the operating principle of the machine to simulate: It applies to vapour compression reverse Rankine cycle machines, independently of whether the useful effect is heating, cooling or both.

The variety of types of components available, mostly handled as commodities, underlines the importance of filling the 'data banks' as conceived in *HPDesign*, and of having the mathematical models and program modules for them. Therefore, to use the full capabilities of the *HPDesign* concept, it is necessary to develop the models, and implement the program modules for

- compressors of various types, including the scroll and screw
- various configurations of shell-and-tube condensers
- plate-finned tube coil condensers
- various configurations of water heated evaporators, including the coiled-coaxial geometry, welded plate, and shell-and-tube with boiling both on the tube and on the shell side
- the electronic expansion valve with both pulsating and modulating operation principles, and other expansion devices such as the short tube, the capillary tube, etc.

Further, the desirable features of design models not implemented yet - multiple components for the same function, adjustable compressor speed as operation variable - shall be considered prioritary as they are instrumental for optimization studies in specific applications.

The number of characteristic data required by the models of each component is substantial, and while most are readily available, either from manufacturers or from simple measurements, others are difficult to obtain. An effort is necessary to reduce, or eliminate, the need for data requiring measurements. This may be done by either changing the models, or convince the manufacturers that such data should be available.

The calculation of thermophysical properties during the simulation takes a significant portion of the running time. An effort is necessary as well in this area in order to minimize the time required for property calculations. There are different possible strategies to follow in solving this problem. They include the use of look-up tables, at least for some intensively used regions such as the saturated two-phase regions, the use of simpler equations or equations valid across phase boundaries which shouldn't require as many checks as required at present. Still related to this area, and as a necessary step in the replacement of ozone depleting or strong greenhouse active fluids, it is necessary to include the environment safe refrigerants in future versions of the property calculation program modules.

The program modules for the individual components may as well be applied as design tools for the components, independently of the machine they might work in. Such tools may exist at some manufacturers, but are usually limited in their
range of application, to the manufacturer's specific products. The application of the program modules for this purpose requires the development of controlling programs, and of the corresponding user interfaces.

The computer platform\(^1\) chosen to carry out this work determined the kind of implementation used, and presents significant limitations to further developments of the concept. This is to say that giving the implemented code a higher degree of portability, at least to UNIX\(^\circledast\) systems, is desirable, and would open its utilization to both academy and industry.

The systematic update of component's models and program modules using the latest technological developments and the data in the literature, e.g. new enhanced heat transfer surfaces in the heat exchangers, new electric motors, etc., is necessary to both check the conformity of models and modules with new developments, and maintain them up to date.

An interesting development would be to build an optimizer around the existing simulation scheme, with the ability to change model parameters, and maximize the global performance for applications with a given pattern of heat demand and source and sink availabilities. This requires more computing power than available by today's technology on the computer platform where the simulations run currently.

**7.2 Further Experimental Work**

The measurements available at present for verification and validation come from a single machine. Experimental data for other sizes and configurations are necessary for a more confident validation. Also in relation to model improvement,

\(^1\) The current implementation is bound to MS-DOS operated computers, and requires major changes in order to be ported to other platforms, namely in the input output interfaces.
and to reduction of the characteristic data, further experimental work is required.

The models requiring experimental data for the individual brands, e.g. thermostatic expansion valve, may eventually be improved to avoid time consuming experiments each time, if a large data set for a variety of brands is available.

Compact heat exchangers such as coiled-coaxial geometries or welded plate, for which the basic processes are known, but that by their construction bring new effects into play, should be subjected to more extensive study.

Experimental data for machines using other source and sink fluids (water-water, water-air, and air-air) are necessary to verify the models as they are extended to these combinations of sources and sinks.

7.3 Further Work on Verification and Validation

Data from the specialized literature, when available, or from cooperative international research, should be used for further verification and validation of component models. On the other hand, as new component models are developed, permitting the simulation of new machine configurations, they should as well be verified with the best data available.
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8. LITERATURE


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9. APPENDICES
APPENDIX A - Induction Motor Losses

Induction motors are the most common prime movers used to drive heat pump compressors in residential and light commercial applications. They are even the exclusive solution to hermetic and semi-hermetic compressors. In these two cases, the type of motor used is the squirrel-cage type. Usually NEMA\textsuperscript{1} Design B standard motors.

In order to predict the temperature increase of the refrigerant before it reaches the cylinder port, in the hermetic and semi-hermetic construction types, it is essential to know the losses of the driving motor. According to Andreas (1982), the electric motor losses break down as follows:

- Stator power losses, $I_1^2 R$
- Rotor Power Losses, $I_2^2 R$
- Magnetic core losses
- Friction and windage losses
- Stray load losses.

These losses may be evaluated using the motor's figures of merit, such as the mechanical efficiency, electrical efficiency, etc. The values of these figures of merit are load dependent, and vary with motor size as well. This Appendix presents a method to calculate the losses of induction motors based on their nominal performance and on directly measurable characteristics.

\textsuperscript{1} National Electrical Manufacturers Association (USA).
The shaft power of a 3-phase electric motor may be calculated as

\[ \dot{W} = \eta_{\text{mech}} \eta_{\text{elec}} \sqrt{3} U I \cos \phi \]  

[9.1]

The mechanical efficiency \( \eta_{\text{mech}} \) accounts for the mechanical losses of the motor, and ranges from 0.95 to 0.98. The electrical efficiency \( \eta_{\text{elec}} \) accounts for the electrical losses which vary significantly with the load. The power factor \( \cos \phi \), cosinus of the phase angle between current and voltage, is also load dependent.

For state-of-the-art induction motors Fink and Beat (1987) propose a relationship between the power factor and the load as depicted in Fig. 9.1.

Fig. 9.1 - Dependence of the power factor upon load according to Fink and Beat (1987).
An equation fitting these data points with excellent approximation is

\[ \frac{\cos \varphi}{\cos \varphi_N} = \left( \frac{\dot{W}}{\dot{W}_N} \right)^{0.4} \]  \[\text{[9.2]}\]

The mechanical losses of the motor are determined from its mechanical efficiency. Andreas (1982) gives indications, for integral horse power motors of small and medium capacity, on the mechanical efficiency, which should lie around 0.98, with negligible variations with load and size.

Based on data of Andreas (1982), the electrical losses, which are all load dependent, may be related to the stator power losses and calculated on that basis. Thus the total electric losses, including core losses, may be calculated from the stator losses as

\[ \dot{L}_{e,t} = 2.703 \times \text{Stator Losses} \]  \[\text{[9.3]}\]

The stator losses depend only upon the ohmic resistance of the stator windings. The ohmic resistance may be directly measured from connection terminals. The ohmic resistance per phase is determined according to the windings arrangement \(\Delta\) or \(Y\). For the \(\Delta\) case the resistance per phase is \(R_p = \frac{3}{2} R_{\text{measured}}\) and for the \(Y\) case it is \(R_p = \frac{1}{2} R_{\text{measured}}\). The total stator losses are calculated as

\[ \dot{L}_{\text{stator}} = 3 R_p I^2 \]  \[\text{[9.4]}\]

And the global electric losses become

\[ \dot{L}_{e,t} = 8.11 \times R_p I^2 \]  \[\text{[9.5]}\]
APPENDIX B - Compressor Mechanical Losses

The mechanical losses of the compressor are accounted for using the compressor mechanical efficiency. The value of this parameter depends upon the size and type of compressor.

Hiller and Glicksman (1976) reported mechanical efficiencies ranging from 0.90 to 0.98, with different ranges according to compressor size. The mechanical efficiency of large compressors ranges from 0.94 to 0.98, while that of small ones ranges from 0.92 to 0.96.
APPENDIX C - Energy Exchange inside a Hermetic Compressor

The energy exchange among the discharge manifold, the lubricating oil, and the suction vapour, see Fig. 3.5, affects both the isentropic and volumetric efficiencies of the compressor.

As shown schematically in Fig. 9.2, the transport processes involved are mainly natural convection (from the oil, discharge manifold and motor/compressor surfaces to the suction vapour, and from the oil heater to the oil), and forced convection (inside the discharge manifold and the oil heater, and in the refrigerant flow through the driving motor). Transport by radiation also occurs, but mostly between the various surfaces.

The calculation of the energy exchanges describing all these processes, though feasible, results in an extremely complex model. Hence, I propose the following method in order to overcome this complexity in a reasonable manner:

Assume that all heat interactions promoting a temperature increase of the suction vapour take place with the discharge manifold. This avoids the calculation of the heat interaction of the suction vapour with the lubricating oil, which temperature is anyway almost exclusively dependent on that of the discharge vapour, Fig. 9.3. As the suction vapour in the shell is forced upwards by both the hot oil...
surface, and the surfaces of the motor/compressor and of the discharge manifold, consider this process to be forced convection, so that the following equations may be used:

Discharge vapour (cooling)

\[ Nu_D = K \, Re_D^{0.8} \, Pr_D^{0.3} \]  \[ \text{[9.6]} \]

Suction vapour (heating)

\[ Nu_S = K \, Re_S^{0.8} \, Pr_S^{0.4} \]  \[ \text{[9.7]} \]

and determine \( K \) from experimental data, assuming:

The discharge manifold is a cylinder of revolution with the free height of the shell (total height minus the height of the oil), and the volume of the high pressure side specified to the compressor (compressor databank);

The suction vapour flows in an annulus defined by the free height and the free volume of the shell, and the discharge manifold as defined;

That both streams flow in countercurrent;

The motor losses to be taken 30% on the outer surface of the motor/compressor, and 70% by the refrigerant flow through the motor windings;

The thermal resistance of the wall of the discharge manifold to be negligible.
Fig. 9.3 - Variation of the lubricating oil temperature with the temperature of the discharge vapour.

With these assumptions, the value of $K$ is 0.26, that is an order of magnitude larger than in the Dittus-Boelter equation. This comes from pooling all interactions on the discharge manifold, what greatly simplifies the model, however letting it still respond to the important variables.
APPENDIX D - The Efficiency of Constant Thickness Circular Fins

The continuous fins on a plate finned-tube heat exchanger may be simulated as circular fins with the same area as that associated with a tube (Schmidt 1949). For both tube patterns - inline and staggered - the radius of the equivalent circular fin is:

\[ R_o = \sqrt{\frac{S_L S_T}{\pi}} \]  

[9.8]

The efficiency of a fin is the ratio of the thermal energy actually transferred to the fin to that that would be transferred were the whole surface at the temperature of the fin root. Mathematically,

\[ \eta_{\text{fin}} = \frac{\int_0^{A_{\text{fin}}} \theta \, dA_{\text{fin}}}{\theta_0 \, A_{\text{fin}}} \]  

[9.9]

The efficiency of the whole extended surface is

\[ \eta_{\text{surf}} = 1 - \frac{A_{\text{fin}}}{A_{\text{tot}}} \left( 1 - \eta_{\text{fin}} \right) \]  

[9.10]

Besides the assumptions of isotropy and steady-state, the main assumptions are:

- the convection heat transfer coefficient is constant over the whole fin surface
- the temperatures of the surrounding fluid and of the fin root are uniform
- the effects of radiation are negligible
- temperature gradients inside the fin may be neglected.

The energy balance of an elementary control volume of a circular fin with constant thickness, including latent heat transfer, e.g. condensation of a vapour fraction from a mixture with noncondensable gas, is
\[ d(qA) = 4\pi R \left[ \alpha \theta + \alpha_m \lg \Delta \omega \right] dR \]  \hspace{1cm} [9.11]

\( \Delta \omega \) is taken as the driving potential for the mass transfer process, and \( \theta \) is the driving potential for convective heat transfer.

Fig. 9.4 - Elementary control volume in a circular fin.

In order to integrate this equation it is desirable to relate both driving potentials. For small values of both potentials a relationship may be obtained by local linearization:

\[
\frac{\Delta \omega}{\theta} = \frac{\omega_{\infty} - \omega_{\text{sat,surf}}}{T_{\infty} - T_{\text{surf}}} = C_{\omega,T} \]  \hspace{1cm} [9.12]
The heat and mass transfer coefficients are related using the Chilton-Colburn analogy between heat and mass transfer (Colburn 1933, Chilton and Colburn 1934). In terms of Colburn j-factors\(^1\)

\[ j_H = j_D \Leftrightarrow StPr^{2/3} = St_mSc^{2/3} \tag{9.13} \]

From the definitions of the dimensionless groups above, and considering that \(Le \equiv Sc/Pr\), it results

\[ \alpha_m = \frac{\alpha}{C_p} Le^{-2/3} \tag{9.14} \]

The energy balance equation for the elementary control volume becomes

\[ d(\dot{q}A) = 4\pi\alpha\theta \left( 1 + C_{\omega,T} Le^{-2/3} \frac{I_g}{C_p} \right) R \, dR \tag{9.15} \]

which differentiated and rearranged gives

\[ \frac{d^2\theta}{dR^2} + \frac{1}{R} \frac{d\theta}{dR} - \frac{2\alpha}{\lambda b} \left( 1 + C_{\omega,T} Le^{2/3} \frac{I_g}{C_p} \right) \theta = 0 \tag{9.16} \]

The boundary conditions to integrate this equation are: No heat flux at the fin tip

\[ \left( \frac{d\theta}{dR} \right)_{R = R_o} = 0 \]

and the temperature difference at the fin root is \(\theta = \theta_o\).

The solution to Eq.(9.11) is written in terms of Modified Bessel Functions (MBF) of the first and second kind.

\(^1\) Some authors, e.g. Bedingfield and Drew (1950) dispute the exponent 2/3 on the Sc and Pr numbers. This value is accepted by most authors today (Bejan 1984, Incropera and De Witt 1985).
\[ \frac{\theta}{\theta_0} = \frac{K_1(B) I_0(A) + I_1(B) K_0(A)}{K_1(B) I_0(C) + I_1(B) K_0(C)} \]  

[9.17]

The fin efficiency in terms of MBF's is

\[ \eta_{\text{fin}} = \frac{2}{\mu_b} \left( \frac{I_1(\mu_b) - \beta K_1(\mu_b)}{I_0(\mu_b) + \beta K_0(\mu_b)} \right) \]  

[9.18]

The arguments of the MBF's are

\[ A = RM \quad B = R_0M \quad C = R_iM \]

\[ M = \sqrt{\frac{2\alpha}{\lambda b} \left( 1 + C_{\omega,T} Le^{-2/3} \frac{l_l}{C_p} \right)} \]

\[ \mu_b = C \quad \mu_e = C \frac{R_0}{R_i} \quad \beta = \frac{I_1(\mu_e)}{K_1(\mu_e)} \]

This is a rather complex method to calculate the fin efficiency, unless the MBF's values are tabulated. In order to simplify it for use in computer simulation applications, the approximate method suggested by Schmidt (1949) is used. Schmidt considered the equation for the fin efficiency of linear fins and modified it so that it would also apply to circular fins. The efficiency of linear fins of constant thickness is calculated as

\[ \eta = \frac{\tanh(ML)}{ML} \]

The Schmidt's approximation consists of a change of the length L so that it accounts for the geometry of the circular fin. The length L is defined
The equation for efficiency becomes

\[
\eta_{\text{fin}} = \frac{\tanh[MR_i(p - 1)(1 + 0.35 \ln(p))]}{MR_i(p - 1)(1 + 0.35 \ln(p))}
\]  

[9.19]

with \( M \) as previously defined.

For processes where simultaneous heat and mass transfer occur, both the fin’s thickness and thermal conductivity differ from the values for the metal alone. The fin becomes a composite fin, especially in case when desublimation of the vapour fraction takes place. The thickness and the thermal conductivity must be corrected to account for these conditions. The equivalent thickness is the sum of the individual thicknesses (condensate or frost layer + metal + condensate or frost layer). The equivalent thermal conductivity is the weighted mean of the thermal conductivity of all layers:

\[
\lambda_{\text{fin,eq}} = \frac{2 \delta_c \lambda_c + \delta_{\text{fin}} \lambda_{\text{fin}}}{2 \delta_c + \delta_{\text{fin}}}
\]  

[9.20]

Given that only steady or quasi-steady operation conditions are considered here, the process of frosting cannot be entirely described (variation of thickness, density and thermal conductivity of the accumulated frost). The process may, however, be analyzed statically by imposing a certain frost thickness (the density is held constant, and the thermal conductivity depends upon the surface temperature). When the mass transfer takes place by condensation, the thickness of the condensate depends on the position of the tube considered in the heat exchanger. For the calculation of the thickness of the condensate layer at the current location the following assumptions are made:
- the condensate flows down exclusively under the effect of gravity (no shear effects of the air stream)
- the rate of condensation above the current cell is assumed equal to that of the current cell in order to reduce the complexity of the computational solution method
- for a condensate thickness less than a certain limiting value, its thermal conductance is assumed infinite (all the effects of the mass transfer process are considered, but no additional thermal resistance).

\[
\theta \leq \gamma = 180^\circ
\]

\[
Z = (\text{Current\_Col} - 1) \cdot S_T + \text{OD} \rightarrow \text{Even Rows}
\]

\[
Z = \text{Current\_Col} \cdot S_T - \text{OD} \rightarrow \text{Odd Rows}
\]

**Fig. 9.5 - Area of condensate for the current cell - upright coil.**

The height of the surface above the current cell depends on the inclination of the coil. The calculation of the surface height is explained in Fig.s 9.5 through 9.7, for the various possible cases.
The total mass flow of condensate at the current cell is

\[ \dot{M}_z = \frac{Z \dot{M}_c}{A_c} \quad [9.21] \]

The thickness of the water film at the current cell is calculated from the momentum balance of the film (Leidenfrost and Korenic 1974).

\[ \delta_c = \left( \frac{3 \nu_c \dot{M}_z}{\dot{M}_c g \nu_c} \right)^{1/3} \quad [9.22] \]

The flow velocity is

\[ u_c = \frac{g \delta_c^2}{3 \nu_c} \quad [9.23] \]

The convective heat transfer coefficient between the film and the metallic surface is, according to MacAdams (1954),

\[ \alpha_c = 0.01 \left( \frac{\nu_c g}{3 \nu_c^2} \right)^{1/3} Pr_c^{1/3} Re_{c,cr}^{1/3} \quad [9.24] \]

where the subscript \( c,cr \) refers to the section with the thickest film, \( \delta_{c,cr} \).

\[ \delta_{c,cr} = \left( \frac{3 \nu_c \dot{M}_z}{(\dot{M}_c g) (\dot{M}_c g)} \right)^{1/3} \quad [9.25] \]

The critical Reynolds number \( Re_{c,cr} \) is

\[ Re_{c,cr} = \frac{4 \nu_{c,cr} \delta_{c,cr}}{\nu_c} \quad [9.26] \]
Fig. 9.6 - Area of condensate for the current cell - forward slanted coil.

Z = MIN OF
\[\text{[(Current\_Row - 0.5) \times S_t / \cos(\theta - 180)]}\]
AND
\[\text{[[[(Current\_Col - 1) \times S_t + OD) / \sin(\theta - 180)] \rightarrow Even\_Rows}}\]
OR
\[\text{[[Current\_Col \times S_t - OD) / \sin(\theta - 180] \rightarrow Odd\_Rows}}\]

Fig. 9.7 - Area of condensate for the current cell - backward slanted coil.

Z = MIN OF
\[\text{[(#\_Flows - Current\_Row + 0.5) \times S_t / \cos(\theta - 90)]}\]
AND
\[\text{[[[(Current\_Col - 1) \times S_t + OD) / \sin(\theta - 90)] \rightarrow Even\_Rows}}\]
OR
\[\text{[[Current\_Col \times S_t - OD) / \sin(\theta - 90)] \rightarrow Odd\_Rows}}\]
APPENDIX E - Measurement of the Thermostatic Expansion Valve Characteristics

Fig. 9.8 - Schematic of the experimental setup to measure the static opening characteristic of a thermostatic expansion valve.

The static (no flow) opening characteristics of TXVs were measured using the installation schematically depicted in Fig. 9.8. The evaporation pressure, $P_{ev}$, is generated by pressing low viscosity oil in the cylinder 2, and the phial temperature is adjusted by means of the thermostatic bath 9. The pressure is kept
constant at the desired value, which depends from the refrigerant the valve was designed for. By changing the phial temperature, different values of the superheating are generated, and the valve opening displacement is measured by means of the indicator 5. It is interesting to note here that hysteresis of the valve opening was only found for new valves. After some run in, no significant hysteresis was found any more. Fig.s 3.50 through 3.52 show some results of measurements by this method for one brand of valves. They also depict the translation of the static opening characteristic when the adjusting screw (8 in Fig. 3.48) was turned one full turn in the clockwise direction (increase of the static opening superheat).

![Graph showing the relationship between Reduced Evaporation Temperature and Intercept/Slope.](image)

Fig. 9.9 - Slope and intercept of the static opening characteristic at various evaporation pressures.
Fig. 9.9 shows the slopes and intercepts of the static opening characteristic of the same valve brand, as function of the reduced evaporation temperature $T_{s,p}$ of the operating refrigerant. The functions $f_1$ and $f_2$ describing the slope and intercept are, respectively

$$f_1 = \lambda_0 T^{\lambda_1} \quad f_2 = \lambda_2 T^{\lambda_3} \quad [9.27]$$

The translation of the static opening characteristic by turning the adjusting screw of one full turn in the clockwise direction, is described by Eq. [9.28], and is shown in Fig 9.10 for the example TXV.

Fig. 9.10 - Variation of the opening superheating with the evaporation temperature.

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1 This is the saturation temperature at the evaporator outlet pressure.
The excess superheating required to balance the dynamic forces is calculated from the pressure temperature relationship of the phial charge. This relationship is not known a priori, but may may be experimentally determined. Fig. 9.11 depicts the schematic of the experimental setup used for this purpose.

\[ \Delta(\Delta T_{SH})_{turn} = \lambda_4 \, Tr^{\lambda_5} \]  

**Fig. 9.11** - Schematic of the experimental setup used to determine the pressure-temperature relationship of the phial charge.
The experimental procedure is relatively simple. It requires the destruction of one expansion valve, though. The phial is cut from the capillary connecting it to the diaphragm case in the valve body, and an adaptor is soldered to the capillary to permit it to be connected to the cylinder 2. Actuating on the cylinder 2, pressure is generated so that the valve opening attains values previously measured when determining the static opening characteristic, while the evaporation pressure is maintained constant by means of the cylinder 3. The pressure so generated in cylinder 2 corresponds to the phial temperature used to obtain the static opening characteristic, that is, it is the saturation pressure of the phial charge.

Regarding the destruction of the valve, one must consider the fact that such valves are designed for a relatively large range of nominal capacities, that may be changed just by replacing the nozzle of the valve. So the destruction of only one valve body allows the obtention of the characteristics for many nominal capacities.
APPENDIX F - Pressure Loss in Bends and Fittings of Refrigerant Pipes

The local pressure loss in pipe bends and fittings is given as

\[ \Delta P = \frac{1}{2} \rho u^2 K \]  

[9.29]

The values of K must be found empirically. The values reported in the literature (ASHRAE 1989, Hydraulic Institute 1979) are accurate to ±25%. They depend from brand and type of construction.

The Fig.s 9.12 to 9.20 show fits to the data in the literature for K, of the most common pipe fittings. The local loss coefficient for welded check valves is independent of their size, and takes the value 2.0. Extrapolation beyond the range actually reported in the literature is allowed (the equations adjusted are smooth and monotone), the accuracy range however may become larger than the above figure of 25%. Actually, in the program HPDesign a warning is issued anytime the equations are used out of the range of the data. The equations and the respective parameters are also given in the plots. The basic variable is the internal diameter of the tube corresponding to the fitting, and must be expressed in millimeter.
Fig. 9.12 - Local pressure loss coefficient - Angle Valve (Screwed).

Fig. 9.13 - Local pressure loss coefficient - Angle Valve (Welded).
Fig. 9.14 - Local pressure loss coefficient - Check Valve (Screwed).

Fig. 9.15 - Local pressure loss coefficient - Gate Valve (Screwed).
Fig. 9.16 - Local pressure loss coefficient - Gate Valve (Welded).

Fig. 9.17 - Local pressure loss coefficient - Globe Valve (Screwed).
Fig. 9.18 - Local pressure loss coefficient - Globe Valve (Welded).

Fig. 9.19 - Local pressure loss coefficient - Return and Elbow Bends (Screwed).
The filter drier is a component mounted on the liquid line pipe, and promotes a pressure drop that is large in comparison to other fittings. The particularity of this component is that it cannot be assigned a constant local loss coefficient as for other fittings (Jones 1964, Rosen et al. 1965). The data reported by Jones were adjusted an empirical equation in order to be able to estimate the pressure loss across this component. Typical values of the refrigerant flow in common applications are out of the range of the data reported by Jones (1964). The equation adjusted to the data allows extrapolation by flow similarity. In any case the accuracy range of the data is not known, and own measurements are not available. The equation adjusted is (internal pipe diameter $\phi$ in millimeter)

$$K = \frac{2 \Delta P}{\rho u^2} = 0.496 \phi^{3.478} Re^{-0.04881 - 0.2397 \times \ln(\phi)}$$

[9.30]
Fig. 9.21 depicts a plot of this equation for typical liquid line pipe diameters (indicated as nominal diameter in inches).

\[ \chi = 0.496 \phi^{0.478} \text{Re}^{-0.44E-3} + 0.324 \]

**Fig. 9.21** - Pressure loss coefficient in a filter-drier.
APPENDIX G - Data Structures used in the Modules of HPDesign

G.1 Compressor Data

Cat_Coef = ARRAY[1..6] OF REAL;
Comp_Typ = (Hermetic,Semi_Hermetic,Open);
Comp_Ptr = ^Comp_Rec;
Comp_Rec = RECORD
  Name : DirStr; { identification }
  Typ : Comp_Typ;
  MotorType : (MonoPhase,ThreePhase);
  CompEff : REAL; { mechanical efficiency }
  ShellOD : REAL; { shell outer diameter }
  ShellHeight : REAL;
  ShellFreeVol : REAL;
  ShellHiPressVol : REAL;
  OilVol : REAL;
  CompDisp : REAL; { volumetric displacement }
  CompNormRpm : REAL; { nominal rotation speed }
  MotNormRpm : REAL; { motor nominal rotation speed }
  MotCompTransm : REAL; { motor compressor transm. ratio }
  MotEff : REAL; { motor mechanical efficiency }
  MotNomPower : REAL; { motor nominal power }
  MotNomVolt : REAL; { motor nominal voltage }
  MotNomAmp : REAL; { motor nominal current }
  MotRotorBlkAmp : REAL; { motor current with blocked rotor }
  PhaseOhm : REAL; { ohmic resistance per phase }
  MotHertz : REAL; { motor nominal frequency }
  CatalogSh : REAL; { superheating for map data }
  CatalogSc : REAL; { subcooling for map data }
  CatalogAirTemp : REAL; { ambient temperature for map data }
  Ref : String[4]; { refrigerant identification }
  Speed : ARRAY[0..2] OF REAL; { speed correction }
  CompRefRate : Cat_Coef; { coefficients for refrigeration capacity }
  CompPower : Cat_Coef; { coefficients for drive power }
END;
Comp_Fil = FILE OF Comp_REC;

G.2 Lubricating Oil Data

Oil_Ptr = ^Oil_Rec;
Oil_REC = RECORD
  Oil_Name : DirStr; { identification }
  Refri : String[4]; { refrigerant identification }
  Density : REAL; { oil reference density }
  Ref_Temp : REAL; { temperature for reference density }
  Tcrm : REAL; { reference temperature misc. curve }
  Ecrm : REAL; { reference miscib. concentration }
  Misc_Line : ARRAY[1..2,1..8] OF REAL; { coefficients }
  Solu_Line : ARRAY[1..9] OF REAL; { coefficients }
END;
Oil_Fil = FILE OF Oil_REC;
G.3 Coaxial Condenser Data

Coax_C_Ptr = ^Coax_C_Rec;
Coax_C_Rec = RECORD
    Condenser_Name : DirStr; { identification }
    CounterCurrent : BOOLEAN; { TRUE or FALSE }
    Out_Tube_Diameter : REAL; { OD of shell tube }
    Out_Tube_Wall_Thick : REAL; { shell tube wall thickness }
    Mean_Coil_Diameter : REAL; { diameter of helix }
    Maximum_Coil_Height : REAL; { height -> number of turns }
    Tube_Conductivity : REAL; { thermal conductivity inner tube }
    Tube_ID : REAL; { ID inner tube }
    Tube_Inner_Rough : REAL; { Relative roughness inner tube }
    CASE ln_Tube_Finned : BOOLEAN OF { TRUE or FALSE }
        TRUE : (Fin_Top_Diameter : REAL;
                 Fin_Root_Diameter : REAL;
                 Fin_Mean_Thick : REAL;
                 Fin_Density : REAL);)
        FALSE : (Tube_OD : REAL);
    END;

Coax_C_Fil = FILE OF Coax_C_Rec;

G.4 Thermostatic Expansion Valve Data

TypModel = (M1,M2);
Txv_Ptr = ^Txv_REC;
TXV_REC = RECORD
    V_Name : DirStr; { identification }
    Nominal_Capacity : REAL;
    Refrigerant : Name; { refrigerant identification }
    Ext : BOOLEAN; { external or internally equilibrated }
    ModelTyp : TypModel; { simulation model options }
    Teta,
    Diam, { orifice diameter }
    InDiam, { diameter of inlet section }
    OutDiam, { diameter of outlet section }
    MaxOpen, { maximum opening displacement }
    PoppetDiam, { poppet diameter }
    PoppetTopDiam, { poppet tip diameter }
    PushRodDiam, { pushrod body diameter }
    PushRodTipDiam, { pushrod tip diameter }
    DiaphragmDiam : REAL; { diaphragm active diameter }
    Open : ARRAY[0..3] OF REAL; { model coefficients }
    Sh_Shift : ARRAY[0..1] OF REAL; { model coefficients }
    PhialPressParm : ARRAY[0..4] OF REAL; { model coefficients }
    LimitOfApphc : ARRAY[1..2] OF REAL; { temperature limits }

Txv_Fil = FILE OF Txv_REC;
G.5 Finned-Coil Evaporator Data

```plaintext
Coil_E_Ptr = "Coil E_REC
Coil_E_REC = RECORD
  Evap_Name : DirStr;
  Circ_File_Name : PathStr;
  Tube_Lay_Typ : Tube_Lay_Typ;
  Fin_Surf_Typ : Fin_Surf_Typ;
  In_Tube_Pattem : Tube_Surf_Typ;
  Supply : Ref_Supply_Typ;
  Num_Tube_Layers, { <= 7 }
  Num_Tube_Per_Layer, { >= 1 }
  Num_Ref_Circ : BYTE; { >= 1 }
  Frontal_Area,
  Long_Pitch, { tube spacing air flow direction }
  Transv_Pitch, { tube spacing normal to air flow }
  Tube_Od, { tube outer diameter over fin collar }
  Tube_id, { internal tube diameter }
  Fin_Spacing,
  Fin_Thickness,
  Fin_Pattern_Depth,
  NumFPattSI, { # fin patterns per long_pitch }
  NumStripSI, { # fin strips per long pitch }
  StripHeight, { height of fin strip }
  StripWidth, { width of fin strip }
  StripLength, { length of fin strip }
  Out_Manifold_ID, { ID of outlet manifold }
  Lambda_Fin, { fin thermal conductivity }
  Lambda_Tube : REAL; { tube thermal conductivity }
  CASE Ref_Supply OF
    Manifold : (In_Manifold_ID : REAL);
    Nozzle : (Circ_Feeders_Length,
               Circ_Feeders_ID,
               Tube_to_Nozzle_ID, { tube diameter TXV->NOZZLE}
               Nozzle_Throat_ID : REAL);
  END;
  Coil_E_Fil = FILE OF Coil_E_REC;
```
G.6 Fan Data

Fan_Typ = (Axial,Centrif);
Drive_REC = RECORD
  Drive_Type : (Monophasic,Triphasic);
  MotNomPower : REAL;
  MotVoltage  : REAL;
  MotNomCurrent : REAL;
  MotNomPowerFactor : REAL;
  MotEfficiency : REAL;
  MotNomSpeed : REAL;
  TransmEff   : REAL;
  TransmRatio : REAL;
END;

Fan_Ptr = ^Fan_REC;
Fan_REC = RECORD
  Fan_Name : DirStr;
  Fan_Type : Fan_Typ;
  Fan_Size : REAL;
  Drive_Data : Drive_REC;
  DimLessPower : ARRAY[0..3] OF REAL;
  CASE Type_Fan Fan_Typ OF
    Axial : Fan_Type OF
      Fixed_Blade : BOOLEAN;
      A_Parm       : ARRAY[0..1] OF REAL;
      Alpha_Parm   : ARRAY[0..1] OF REAL;
      Phi_Stall    : ARRAY[0..1] OF REAL;
      Psi_Stall    : ARRAY[0..1] OF REAL;
      LimitOfApphc : ARRAY[0..1] OF REAL;
    Centrif : (DimLessPress : ARRAY[0..3] OF REAL);
  END;
  Duct_Fil = FILE OF Fan_REC;

G.7 Air Ducts Data

Duct_Ptr = ^Duct_REC;
Duct_Geo = (Circ,Rect);

Channel = RECORD
  DPosition : (InBranch,OutBranch);
  DLeng     : REAL;
  DSingular_Coeff : REAL; { singularity loss coefficient ΠK }
  CASE DGeometry : Duct_Geo OF
    Circ : (Diameter : REAL);
    Rect  : (Sizes : ARRAY[1..2] OF REAL);
  END;
  CASE Channel OF
    Circ : REAL;
    Rect : REAL;
  END;
Duct_REC = RECORD
  DName : DirStr;
  Inlet : Channel;
  Outlet : Channel;
END;
Duct_Fil = FILE OF Duct_REC;
G.8 Refrigerant Piping Data

Orientation = (Downward, Horizontal, Upward);
Section_Lst = (Discharge, Liquid, EvapFeed, Suction);
Tube_Material = (Copper, Steel);
TubingPtr = ^Tubing;
Tubing = RECORD
  Material : Tube_Material;
  Tube_Orientation : Orientation;
  Total_Length : REAL;
  Altimetric_Length : REAL; { height between tube ends }
  In_Diameter : REAL;
  Out_Diameter : REAL;
  Local_Loss : REAL;
  Tube_Rough : REAL; { Relative Roughness }
  Insulated : BOOLEAN; { TRUE or FALSE }
  Insulation_Thick : REAL; { insulation thickness }
  Insulation_Lambda : REAL; { insulation thermal conductivity }
END;

Piping_Ptr = ^Piping_REC;
Piping_REC = RECORD
  Machine_Name : DirStr;
  Discharge_Line : Tubing;
  Liquid_Line : Tubing;
  Evap_Feed_Line : Tubing;
  Suction_Line : Tubing;
END;
Piping_Fil = FILE OF Piping_REC;
Leer - Vide - Empty
VITA

Born November 18, 1953, the fifth child of Ana Maria and Ventura Lopes Conde

1960-64 Primary school in Torrao do Lameiro - OVAR
1964-66 Preparatory secondary school at the Escola Industrial de Ovar
1966-69 Study to Technician on Electromechanics at the Escola Industrial de Ovar
1969-70 Preparation for the entrance exam to the Instituto Industrial do Porto
1970-74 Study to Industrial Engineer on Electromechanics at the Instituto Industrial do Porto (renamed Instituto Superior de Engenharia do Porto)
1974-75 Instructor on Electromechanics at the Escola Industrial de Ovar
1975-80 Industrial Engineer at the Administração dos Portos do Douro e Leixões (Douro and Leixoes Harbour Administration)
1977-79 Complementary engineering studies to the degree of Mechanical Engineer at the Faculdade de Engenharia da Universidade do Porto, specializing on heat and fluids engineering
1982-83 Research on the dynamic behaviour of vapour compression heat pumps at the Departement Werktuigkunde - Katholieke Universiteit Leuven - Belgium, under Prof. Dr. Jan Berghmans