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'ART' A COMPUTER CODE TO ASSIST THE DESIGN OF REFRIGERATION AND A/C EQUIPMENT

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ABSTRACT

In this paper a new computer code: ART (Advanced Refrigeration Technologies), recently developed at the Universidad Politécnica de Valencia is presented. The main features of the program are fast calculation and accurate evaluation of the refrigeration unit performance. The paper describes the main characteristics of the global model and of the submodels of the main components considered in ART, as well as of the concerned numerical strategies, showing the capabilities of the code to assist the design of refrigeration equipment.

The model takes into account all the components of the system: compressor, HEs, piping and accessories. For the main components different submodels with different levels of accuracy are included in order to allow the user to concentrate in the optimization of the desired component.

ART is able to calculate a complete refrigeration system, including the detailed modeling of the HEs, in less than 20s. Once minimally adjusted, typical predicted results show a good agreement with experimental results with maximum deviations around 5%.

NOMENCLATURE

A	Cross section area (m^2)
C_p	Specific heat (J/kg K)
D_h	Hydraulic diameter (m)
E	Power (W)
f	Friction factor
g	Gravity (m/s^2)
G	Mass velocity (kg/s m^2)
H	Heat transfer coefficient (W/m^2K)
HE	Heat exchanger
HTC	Heat transfer coefficient
i	Enthalpy (J/kg)
k	Conductivity (W/m K)
NTU	Number of transfer units
m	Mass flow rate (kg/s)
P	Perimeter (m)
PHE	Plate heat exchanger
p	Pressure (Pa)
Q	Heat (W)
T	Temperature (K)
u	Velocity (m/s)
V_s	Swept volume flow (m^3/s)
W	Absolute humidity (-)
x	quality (-)
y_w	Film water thickness (m)
z	Spatial co-ordinate (m)

Greek

α	Void fraction
Δ	Increment
ε	Efficiency
η_c	Compressor efficiency
η_v	Volumetric efficiency
ξ	Heat losses ratio to the environment
Φ_f^2	Two phase friction multiplier
θ	Angle with horizontal
ρ	Density (kg/m^3)

Subscripts

a	Air
C	Sensible heat
D	Latent heat
f	Saturated liquid
g	Saturated vapour
i	Inlet, cell index
is	Isentropic
o	Outlet
r	Refrigerant
w	Wall
wat	Water

INTRODUCTION

ART (Advanced Refrigeration Technologies) is a software package recently developed at the Universidad Politécnica de Valencia for the analysis and optimization of refrigeration equipment based on the vapor compression principle. ART is based on a long experience on the detailed modeling of refrigeration components and is fully targeted to assist the design of components and systems, especially envisaged from the beginning of its development as a design tool for industrial use. The main feature of the program is the accurate evaluation of the refrigeration unit performance including a quite accurate modeling of every single component at the same time. In that way, any modification in one or several components can be always assessed from the perspective of the global performance of the unit.

The aim of the software is to combine a high accuracy with a low CPU time in order to be a feasible tool to support the design of refrigeration equipment.

MAIN FEATURES OF ART

The main features of ART are:

- Evaluation of fluid properties of pure fluids and any mixture. A software tool called GENMAP is included to be able to generate a file with the thermodynamics and transport properties of any refrigerant under the specified range of operation.
- Calculation of the Theoretical cycle. The software allows for the evaluation of the theoretical cycle for the prescribed evaporation and condensation temperatures and cooling or heating duties.
- Calculation of the Real cycle. The software allows for the evaluation of the working cycle for the prescribed components: evaporator, condenser, piping and compressor, at the given temperature and flow rate, or alternatively at specified inlet and outlet temperatures, of the heat source and sink (secondary fluids).
- A data bank of usually employed compressors, and heat exchangers can be defined and stored in such a way that the analysis and optimization just requires the variation of the considered design parameter, becoming a very comfortable tool for equipment selection and optimization. For reversible units a single click allow to automatically exchange the corresponding heat exchangers. Built in libraries allow for an adequate local evaluation of the heat transfer coefficients and friction factors in the heat exchangers and piping. Enhancement/suppression factors allow for an easy adjustment to experimental results.
- Parametric studies with single or combined input variables can be easily performed. The user interface has been designed in order to make data input as comfortable as possible and to allow a fast visual analysis of the obtained results.

MODELING

The global model of the system is divided in submodels: compressor, heat exchangers, expansion valve, accessories, and piping. A basic description of each submodel is given below.

Calculation of the refrigerant thermodynamic and transport properties is performed by REFPROP subroutines from NIST [1]. The corresponding properties are then conveniently stored in a data file. The required properties are then calculated by interpolation from the data file. Additionally, built-in tables allow the calculation of the properties of any usual secondary fluid, i.e. water, air and common brines.

Each submodel involves a series of non-linear equations and in the case of the heat exchangers, maybe also the solution of a system of ODEs, which is discretized with a finite volume technique. The global set of equations forms a complex system of non-linear equations AEs or DAEs which is solved through a Newton-like solver.

The independent variables chosen for the global set of equations are pressure and enthalpy in each inlet and outlet point. This choice assures a smooth variation of the variables, not given by other choices like temperature or quality. The main characteristics of the employed models are described in the following sections. Then, these models are coupled to form a global model of the heat pump, which takes into account the matching problem of the global set of equations with special care.

COMPRESSOR

For the compressor, three equations are required to characterize its behavior: one for the mass flow rate, one for the power input and one for the outlet enthalpy.

The mass flow rate can be calculated as:

$$\dot{m} = \rho_i \dot{V}_s \eta_v \tag{1}$$

The second governing equation of a compressor involves compressor efficiency:

$$\dot{E} = \frac{\dot{m}\left(i_{is} - i_i\right)}{\eta_c} \tag{2}$$

Finally, it is necessary to calculate the outlet conditions of the compressor introducing the heat transfer to the environment:

$$h_o = i_i + \frac{i_{is} - i_i}{\eta_c}(1-\xi) \tag{3}$$

The empirical parameters involved in those equations: volumetric efficiency, compressor efficiency, and heat losses to the environment can be easily input. ART accept 4 different ways to define them: constant values, a polynomial or data table as a function of the pressure ratio, default functions depending on compressor type (built in correlations for compressors of different types have been introduced in the program), the coefficients of the corresponding ARI polynomials. All this information can be conveniently stored in a data bank in such a way that the user is able to reuse the supplied information again or modifying it.

HEAT EXCHANGERS

HEs can be considered in ART at two basic different levels of detail depending on the required accuracy on the HE results. The simplest way is to treat them as a HE with total area and known Overall Heat Transfer Coefficient (OHTC). Then, the program uses an ε-NTU characterization to model the heat exchanger. This simple model allows closing the refrigeration cycle taking into account the major effect of the HE on the operating cycle without the need of a more detailed definition. The OHTC can be prescribed either as a constant or as a function of the heat flux or the secondary fluid mass flow rate. This strategy is employed whenever one whishes to include the HE in the estimation of the performance of the refrigeration unit but the main interest of the optimization study is placed on another component.

When the interest of the study is placed on the optimization of the heat exchanger or a very accurate estimation of the unit performance is required, a detailed definition and accurate modeling of the HE must be used. ART includes a HE model based on the discretization of the HE in cells along the refrigerant and secondary fluid paths, assuming one-dimensional flow. The model is able to take into account both heat transfer and pressure drop, with local evaluation of the heat transfer coefficient and friction factor, by built in correlations, as well as of the fluid properties. This model is able to take into consideration most of the geometrical and operation parameters of current evaporators and condensers.

The HE detailed model included in ART has been developed to be able to be applied to any kind of HE and flow arrangement. However, for space reasons, in the following, only a fin and tube HE will be considered to illustrate the basis and capabilities of the model, especially for the calculation of the dehumidification process taking place at the air side of tube and fin evaporators. A HE is basically formed by two fluids that exchange heat throughout a series of wall pieces, formed in this case by a combination of tube and fins.

Refrigerant side

In the case of an evaporator or a condenser, a 2-phase flow with phase change occurs. A steady 2-phase flow following an annular pattern is considered to occur inside the tubes. Therefore, the separated fluid model can be considered as adequate, for which the governing equations are:

$$G = \rho u = \text{constant} \tag{4}$$

$$-\frac{dp}{dz} = \frac{2f \cdot G^2 (1-x)^2}{D_h \rho_f} \Phi_f^2 + G^2 \frac{d}{dz}\left(\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_f (1-\alpha)} \right) + \left(\alpha \rho_g + (1-\alpha)\rho_f \right) g \sin\theta \tag{5}$$

$$AG \frac{\partial}{\partial z}\left[x\left(i_g + \frac{G^2 x^2}{2\rho_g^2 \alpha^2} \right) + (1-x)\left(i_f + \frac{G^2 (1-x)^2}{2\rho_f^2 (1-\alpha)^2} \right) \right] + AG \frac{\partial}{\partial z}(zg\sin\theta) = Ph(T_w - T) \tag{6}$$

The continuity equation states the conservation of the mass flow rate and the mass velocity all along a refrigerant circuit. Its value is known from the inlet conditions, so that G can be considered as a known constant in the analysis.

The most employed correlations for heat transfer coefficients and friction factor for annular flow in pipes have been studied by the authors in [2]. In ART, for the case of two-phase flow inside the tubes of a coil, the VDI correlation is used for evaporation, while the Traviss correlation is used for condensation. For the friction factor coefficient, the Chisholm correlation is employed, for both condensation and evaporation.

Air side

For the air, the governing equations are those stated for the mass, energy and momentum conservation. In the case of the differential surface shown in figure 1, the following approximated equations are stated. See [3]:

$$-\dot{m}_a di = dQ - \dot{m}_a \cdot dW \cdot i_{f,wat} \tag{7}$$

$$dQ = \left[h_c\left(T - T_{wat}\right) + h_D\left(W - W_{s,wat}\right)\left(i_{g,T} - i_{f,wat}\right)\right]Pdz \tag{8}$$

$$-\dot{m}_a dW = h_D Pdz\left(W - W_{s,wat}\right) \tag{9}$$

$$\frac{dp}{dz} = -\frac{d\left(\rho u^2\right)}{dz} - f\frac{1}{2D_h}\rho u^2 \tag{10}$$

where $i_{s,wat}$ is the enthalpy of the saturated air at the water surface temperature.

The approach followed to treat the dehumidification process is the one proposed by Threlkeld [3].

Figure 1 shows an air cell in the more general case in which dehumidification of humid air takes place when the humid air is in contact with a cold surface. A water film is formed over the surface. There is a limit boundary layer of air next to the water surface. The hypothesis that the air in contact with the water film is saturated at the temperature of the water surface, T_{wat}, is assumed.

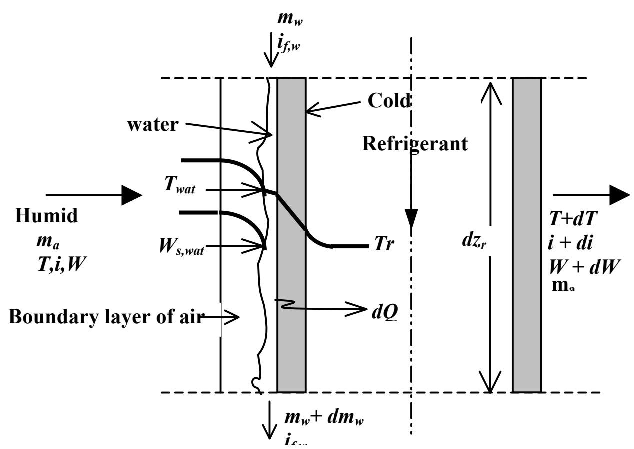

Figure 1. Cooling and dehumidification of humid air around a tube

Then, the heat transfer can be expressed as a function of the enthalpy difference as the driving potential in the following way:

$$dQ = \frac{h_w}{b_w}\left(i - i_{s,w}\right)Pdz \tag{11}$$

where $i_{s,w}$ is the enthalpy of the saturated air at the wall temperature, and:

$$h_w = \frac{1}{C_{p,a}/\left(b_w h_C\right) + y_w/k_w} \tag{12}$$

where k_w is the water thermal conductivity and y_w is the thickness the water film.

The value for h_C comes from semi-empirical correlations. In ART, the ones proposed by Chi Chuan Wang [4] and [5], are used as default correlations for either sensible heat transfer or dehumidifying conditions.

Global solution strategy

The global solution method employed is called SEWTLE (for Semi Explicit method for Wall Temperature Linked Equations) and is outlined in [6]. Basically, this method is based on an iterative solution procedure. First a guess is made about the wall temperature distribution, then the governing equations for the fluid flows are solved in an explicit manner, getting the outlet conditions at any fluid cell, from the values at the inlet of the HE and the assumed values of the wall temperature field. Once the solution of the fluid properties are got at any fluid cell, then the wall temperature at every wall cell is estimated from the balance of the heat transferred across it. This procedure is repeated until convergence is reached. The numerical scheme developed for the calculation of the temperature at every wall cell is also explicit, so that the global strategy consists in an iterative series of explicit calculation steps. The method is of application to any flow arrangement and geometrical configuration, and offers excellent computational speed. Moreover, it can be used, as it is the case of the present paper, for combined single, 2-phase flow and air.

In two-phase flow, the energy (Eq. (6)) and momentum (Eq. (5)) equations are coupled by the pressure through its influence on the temperature. Fortunately the dependence is weak due to the usual small pressure drop inside the heat exchanger. Since all the variables mainly depend on the pressure, the momentum equation can be integrated first. Then, once the pressure at the outlet of the fluid cell is known, the energy equation can be integrated, leading to the evaluation of the enthalpy at the outlet, and of the quality and the rest of variables. The discretized equations for 2-phase flow becomes:

$$i_o = i_i + \frac{h}{\dot{m}}\left(T_w - \frac{T_i + T_o}{2}\right)P \cdot \Delta z - g \cdot \sin\theta \cdot \Delta z \qquad (13)$$

At the airside, for the case in which there is dehumidification of the air, the energy equation (11) is integrated, calculating first the outlet enthalpy, and then the outlet temperature.

$$i_o = i_i e^{-\frac{h_w P \Delta z}{\dot{m}_a b_w}} + i_{s,w}\left(1 - e^{-\frac{h_w P \Delta z}{\dot{m}_a b_w}}\right) \qquad (14)$$

$$T_o = T_i e^{-\frac{h_C P \Delta z}{\dot{m}_a C_{p,a}}} + T_w\left(1 - e^{-\frac{h_C P \Delta z}{\dot{m}_a C_{p,a}}}\right) \qquad (15)$$

The latent heat is calculated as the difference between the total heat and the sensible heat. From the latent heat, the outlet humidity is calculated. The momentum equation is then integrated, getting the outlet pressure of the air. Finally, the outlet properties of the air can be calculated.

Once the refrigerant temperature and the air temperature are known at every cell, the wall temperature can be determined at every wall cell by the balance of heat transfer between the fluids and the wall.

$$P_a \Delta z_a h_{a,eq}(T_w)\left(T_{a,i} - T_w\right) = P_r \Delta z_r h_r \left(T_w - \bar{T}_r\right) \qquad (16)$$

where h_r and $h_{a,eq}$ have been defined in such a way that provide the heat exchanged with the fluids through the equivalent temperature difference in such a way that the equation becomes linear.

Typically, the number of required iterations to get the solution ranges from 10 to 20.

A detailed description of the modelling of PHE and coils working as evaporators and condensers, based on the same numerical techniques employed in ART, can be found in [7] and [8].

EXPANSION DEVICE

The expansion device considered at the moment in ART is a thermostatic expansion valve so it involves two governing equations, the first one is the assumption of isoenthalpic flow through the valve and the second one is the statement of the specified superheat

A more complex modeling of the expansion valve, taking into account the flashing process occurring through it is currently under development. This model would allow the consideration of short pipes and orifices, and capillary tubes, as alternative expansion devices.

PIPING

Three piping lines are considered to exist in ART: discharge, liquid, and suction. These lines are regarded as pipes with pressure drop and heat transfer. The model assumes an ε-NTU for the heat transfer calculation, with consideration of the corresponding insulation and surrounding air temperature. The HTC corresponding to the surrounding air is calculated depending on the type of assumed convection process: forced or natural.

Pressure drop is calculated taking into account both friction and acceleration terms. Pressure losses produced by accessories can be specified. They are considered through the classical approach of increasing the length of the pipe.

GLOBAL SYSTEM OF EQUATIONS

A balance between the above stated equations and the list of unknowns easily shows that the system of equations requires an additional closure equation. The refrigerant mass inventory provides this equation in general. However, as discussed in [9], the existence of a liquid receiver at the condenser outlet imposes saturated or slightly subcooled conditions at the outlet of the receiver, so that this new condition allows the closure of the system. Mass inventory then becomes an auxiliary equation, which would only be necessary for the calculation of the liquid level in the receiver. For systems that do not contain a liquid receiver, the statement of the specified subcooling at the condenser outlet would alternatively allow the closure of the system of equations.

The governing equations of each component described above can be basically posed in the form:

$$f(p_i, i_i, p_o, i_o, \dot{m}) = 0 \qquad (17)$$

The input values to each sub-model are the pressure, enthalpy and mass flow rate at the inlet and the outlet. Then, using the thermodynamic functions, any other thermodynamic property (temperature, density, …) can be known and the output value of the function (17) can be calculated. Therefore, the problem is reduced to calculate the solution to a non-linear system of equations f(**x**)=0.

The system of equations is solved using a standard solver based on the MINPACK subroutine HYBRD1, which uses a modification of M.J.D. Powell's hybrid algorithm. This algorithm is a variation of Newton's method, which uses a finite-difference approximation to the Jacobian and takes precautions to avoid large step sizes or increasing residuals. For further description, see [10].

The program continuously surveys the convergence of the method and a special strategy to find appropriate initial solutions and also to carefully bound variables and functions have been worked out. The result of all this has been an extremely robust algorithm.

RESULTS AND DISCUSSION

One of the main objectives of ART is to reduce at maximum the calculation time. The calculation of the performance of a complete refrigeration system with detailed modeling of the evaporator and condenser is performed typically in less than 20s. This allows performing the broadest parametric studies in just a very few minutes. All the main design data of the equipment and the operation conditions have been defined as possible variables for performing single or combined parametric studies. This makes ART a very comfortable analysis and optimization tool.

When desired, the results of ART can be easily adjusted to measured results through a series of Enhancement-Suppression factors which have been included among the inlet data in order to provide a way to adjust the heat transferred and the pressure drop across the different components of the system: HEs, piping or accessories.

The global model and the submodels included in ART have been under development during the past 5 years. They have been individually and globally tested and contrasted with other programs and/or experimental results (see for instance [7][8][11]). Reasonable accuracy, fast computation and, above all, high robustness were the targeted objectives and have been the final result.

ART includes a graphical user interface which has been specially elaborated to make easy the analysis and optimization of refrigeration systems. The user interface has been programmed on C++ and has been entirely targeted to its professional use.

As an example of the accuracy of the model, the following figure shows the comparison between calculated and measured results for an air-to-water reversible heat pump of approx. 18 kW refrigeration capacity under both modes of operation: heating and cooling.

The main components of the studied heat pump are: Danfoss Maneurop compressor SM110, AlfaLaval CB52X BPHE with 46 plates, Alfa Laval coil with 11 circuits, 3 tube rows and a 3/8” tube, and Danfoss TDEX 6 thermostatic expansion valve.

Figure 2 shows the comparison between calculated and measured results in terms of capacity (cooling or heating capacity depending on the operating mode) and the corresponding COP. As can be observed the model predicts the performance of the heat pump within a 5% of relative error without any specific adjustment.

Two different groups of points can be distinguished in the graphs, corresponding to heating and cooling (lower capacities values) operation modes. The variation range of the heat pump operation conditions covers: different air temperatures (35ºC and 40ºC in cooling mode, and 12ºC and 15 ºC in heating mode), air humidity was varied from 60% and 80%, and fan speed, from 35Hz to 50 Hz.

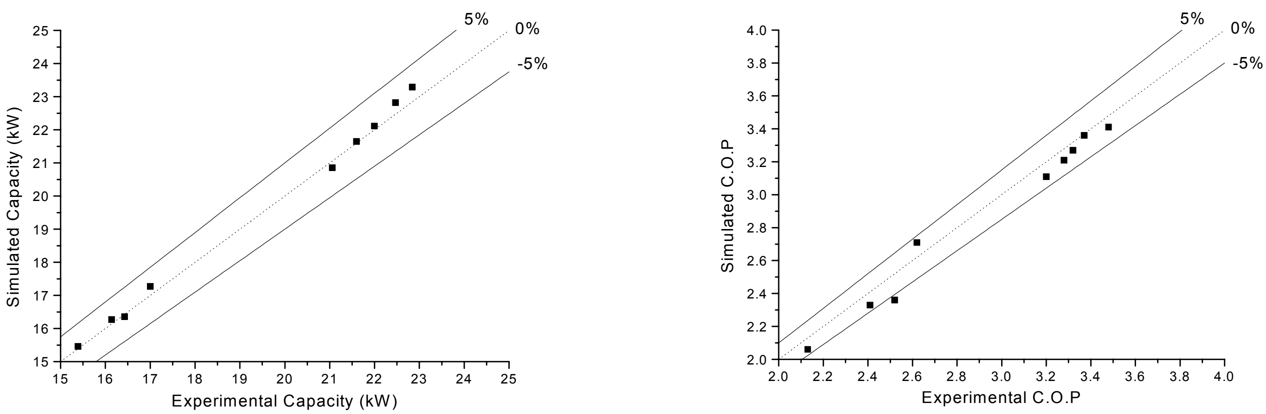

Figure 2 Comparison between measured and calculated results for an air-to-water reversible heat pump of approx. 18 kW refrigeration capacity.

CONCLUSIONS

The following main conclusions can be drawn:

- ♦ A model for refrigeration equipment based on the vapor compression cycle has been presented. The presented model is the heart of a computer code called “ART”.
- ♦ ART has been conceived as a computer code to assist the design of refrigeration and A/C equipment.
- ♦ The global model includes independent submodels for all the components of the system: compressor, HEs, piping and accessories. For the main components different submodels with different levels of accuracy are included in order to allow the user to concentrate in the optimization of the desired component.
- ♦ The detailed model for HEs is based on a one-dimensional discretization of the fluid flows and the wall and takes into account the local estimation of the heat transfer coefficient and friction factor, all along the HE fluid paths. The solution strategy is iterative and consists of a series of successive explicit evaluations of the fluid temperatures and then of the wall temperatures. The authors call the method: SEWTLE, for Semi Explicit method for Wall Temperature Linked Equations. This model takes into account the main design parameters of a HE in such a way that the user is able to select the optimum

component from the available list of components or to optimize its design of a HE to get for instance maximum COP of the unit.

- ♦ The global system of equations is solved using a standard solver based on the MINPACK subroutine HYBRD1, combined with a special strategy to find appropriate initial solutions and also to carefully bound variables and functions.
- ♦ ART is able to calculate a complete refrigeration system, including the detailed modeling of the HEs, in less than 20s. Once, minimally adjusted, typical predicted results show a difference between measured and predicted values lower than 5%.

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