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A USER-FRIENDLY SIMULATION AND OPTIMIZATION TOOL FOR DESIGN OF COILS

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ABSTRACT

Air-cooled cross flow heat exchangers are very popular in the refrigeration and air conditioning industry. The fin type, tube arrangement, flow circuitry, and many geometry parameters, are to be designed and optimized. In this paper, a generic simulation and optimization tool for the design of air-cooled condensers/evaporators is introduced. It is featured with a user-friendly interface, freedom in choice of pure or mixed refrigerants, number of tubes and rows, type of fins, and geometry dimensions including tube diameter, tube length, tube spacing, fin thickness/pitch. There is no restriction on refrigerant flow circuiting, and coupled heat exchangers with multiple fluids inside different subsets of tubes can be modeled and analyzed simultaneously. The heat transfer and pressure drop both on the refrigerant side and air side are calculated according to the appropriate correlations or data provided by the user. A segment-by-segment approach within each tube is adopted. The graphical user interface allows non-uniform air distribution by assigning individual temperature, flow rate and humidity value to each segment of the frontal face area of the coil. The calculated refrigerant and air properties at each segment can be automatically exported to a spreadsheet. Example simulation results are presented. A sample optimization case using a genetic algorithm is shown at the end of the paper, where a combination of the tube diameter and number of rows is optimized to yield the smallest heat transfer area subject to the given heat load and allowable pressure drops.

NOMENCLATURE

A: area
c: constant
C_p: specific heat capacity
D: tube diameter
DP: pressure drop
f: fitness value
h: specific enthaply
J: Junction
T: tube, Temperature
l: length
JTA: junction-tube matrix
$\dot{m}$: mass flow rate
P: pressure
ρ: density
HTC: heat trasnfer coefficient
q: heat duty
R: penalty parameter
Q: total heat duty
$N_{circuit}$: Circuitry index
N_{row}: Number of rows
$N_{tube,row:}$ Number of tubes at each row

Subscript:
a: air
i: tube i
in: inlet
j: junction
k: segment k
out: outlet
ref: refrigerant

INTRODUCTION

Plate-fin-tube heat exchangers are widely used in the air conditioning and refrigeration industry. The air passes between the fin plates while the refrigerants or coolants flow through the tubes. Due to their complexity in the geometry, tube arrangement, circuitry, non-uniformity of airflow, and variety of working fluids, it is quite difficult to accurately predict the performance of these coils by analytical/graphic design approaches as described in many heat exchanger and HVAC handbooks. In this paper, HXDesigner, a simulation and optimization tool for design of air-cooled plate-fin-tube heat exchangers, is introduced. It distinguishes itself by adopting network concept for the computational approach, providing a user-friendly graphical interface, and integrating genetic algorithm for optimization design.

Many heat exchangers models have been presented in the open literature, with increasing complexity of the calculation procedure, detail of the coil parameter input, and range of working fluids.

Domanski (1991) developed a simulation model for evaporator in residential air conditioning. It accounted for the misdistribution of the air flow and refrigerant flow and made simplified assumption about the heat transfer and pressure drop on both refrigerant and air side. A tube-by-tube approach is used to analyze the performance of each tube separately. Individual refrigerant property and mass flow rate as well as air property and mass flow rate are assigned to each tube and calculated in a proper order depending on the refrigerant circuitry and air stream.

The computational model presented by Bensafi (1997) discretizes plate-fin-tube coil into tube elements and solves the associated governing equations of each element with local values of temperature, pressure and heat transfer coefficient. The working fluids include water, R22, R134a, and some refrigerant mixtures. The coil geometry and circuitry, and operating parameters are given thorough an input file. The computation algorithm starts at the inlet tube and tracks the refrigerant flow to the exit. The outlet air temperature/humidity and refrigerant temperature/quality of each element are repeatedly calculated and updated until the difference of the successive values of these properties are within a pre-specified tolerance. When there are multiple circuits in a coil, the refrigerant flows in each branch are calculated iteratively to yield the same pressure drop.

In a procedure for the performance prediction of chilled water coils, Vardhan (1998) calculated the local heat transfer at each tube segment with effectiveness-NTU method. Under wet conditions, parallel flow is assumed instead of cross flow as the coolant heat capacity is much larger than that of air.

Corberan (1998) made a comparative study of a number of correlations for both heat transfer and pressure drop on the refrigerant side, in modeling of the plate finned tube of evaporators and condensers working with R134a. An experimental study was made to validate the model. The pressure drop of the two-phase flow is expressed as the sum of the frictional, momentum, gravitational, and local (at the 180^o bends) pressure drops.

Liang (2001) used a hierarchical system consisting of branch, tube, and control volume to develop a general program that can simulate evaporator coils with splitting and joining. To balance the increase of the heat transfer coefficient and pressure drop in the high vapor quality region and reduce the pressure gradient in the superheating region, the author suggested using a suitable complex refrigerant flow circuitry to improve the coil performance.

Optimization of heat exchanger design has been a long-existing research topic since 1950's, especially in the chemical processing industry (Bulck, 1991, Fax, 1957, Hedderich, 1982, Jegede, 1992, Kovarik, 1989), where analytical solutions for the performance of the heat exchanger are adopted and conventional optimization methods are used. Ragazzi (1995) conducted thermodynamic optimization of evaporators with zeotropic refrigerant mixtures, based on computer simulation model. The entropy generation associated with the heat transfer and pressure drop of both the refrigerant side and the air side is the objective function to be minimized. The effect of number of coil rows and tube diameter on the overall heat exchanger performance is investigated. Reneaume (2000) used a sizing procedure to evaluate the objective function and the constraints, and HSQP algorithm to optimize plate fin heat exchangers.

Recently there is an increasing interest in applying genetic algorithms (GA) to the heat exchanger optimization. Schmit et al. (1996) used GA to improve both the thermal and hydraulic performance of a high intensity cooler by optimizing a mix of discrete and continuous design variables. Aimed at minimizing heat transfer area required for a given heat duty, Tayal et al. (1999) adopted GA to solve a large-scale, combinatorial and discrete optimization problem involving a black-box shell-and-tube heat exchanger model. The tube length, number of shells and baffles, tube and shell orientation, and other variables are optimized with considerable computational savings.

In this paper the model for simulating cross-flow plate-fin-tube heat exchangers with any possible configuration is first presented, followed by some predicted performance results. Then an optimization case study is shown.

PROPOSED MODEL

In order to simulate a heat exchanger without restriction on the tube connection and flow circuitry, an analogy of coil to an electric circuit network can be made. The tube in a coil is like the resistor in an electric circuit, the mass flow rate through a tube is like electric current, and the pressure is like the electric potential. While the electric current via a resistor can be considered as a linear function of the potentials at the two ends of the resistors, the mass flow rate through a tube is a highly nonlinear function and decided by more variables including inlet pressure, inlet enthalpy and outlet pressure of the refrigerant and the surrounding air condition. The inlet enthalpy of the refrigerant is decided by the heat transfer of upstream fluids and increases the complexity of the problem.

Input and output

For the entire heat exchanger, the input parameters for the internal fluid are the inlet pressure P_{in}, inlet enthalpy h_{in}, and outlet pressure P_{out} at each inlet and outlet tube. The input for the air side is the environmental temperature T_{env}, environmental relative humidity ϕ and air flow rate $\dot{m}_a$ at each tube segment of the frontal row.

For the tube or the tube segment, the input for the internal fluid is the inlet pressure P_{in}, inlet enthalpy h_{in}, and outlet pressure P_{out}. The input for the air side is the facing air temperature T_{airin}, facing relative air humidity ϕ_{in}, and air flow rate $\dot{m}_a$.

For the tube or the tube segment, the output is the latent heat load, sensible heat load, charge, mass flow rate, and outlet enthalpy of the internal fluid, and the leaving air temperature and humidity.

For the entire heat exchanger, the output is the total heat load, total charge of the internal fluid, outlet enthalpy and temperature of the internal fluid, and the exit temperature and humidity of the air streams.

Equations

The mass flow entering a junction is equal to the mass flow leaving the junction in steady state. The energy flow entering a junction is equal to the energy flow leaving the junction1 in steady state (Lindsay, 2000). The enthalpy hj at the inlet of each tube downstream to a junction is the weighted average enthalpy of the fluid mixed at the junction from the upstream tubes.

$$\sum_{JTA[j][i]=-1} \dot{m}_i = \sum_{JTA[j][i]=1} \dot{m}_i$$

where, $\dot{m}_i = f_i(P_{i,in}, h_{i,in}, P_{i,out})$

$$\sum_{JTA[j][i]=-1} \dot{m}_i h_{i,out} = \sum_{JTA[j][i]=1} \dot{m}_i h_{i,in}$$

where, $h_{i,out} = \phi_i(P_{in}, h_{in}, P_{out})$, $h_{i,in} = h_j = \sum_{JTA[j][i]=-1} \dot{m}_i h_i / \sum_{JTA[j][i]=-1} \dot{m}_i$

The subscript j and i denotes junction and tube, respectively.

To account for non-uniform air distribution and heterogeneous refrigerant side heat transfer coefficients, it is necessary to discretize the tube into a number of segments, form and solve the hydraulic equation and energy/heat transfer equations for each segment.

The single-phase hydraulic equation of the segment k of tube i can be expressed as:

$$P_{i,k,in} - P_{i,k,out} = c\frac{2l}{\pi\rho D^3}\dot{m}_i^2$$

Appropriate correlations are available to describe the two-phase hydraulic equation, depending whether it is condensation or evaporation process, choice of refrigerants, operating conditions, and tube geometry.

When the average wall/fin temperature of the tube segment is higher than the dew temperature of the air around the segment, no water vapor condensation occurs. The heat transfer between the air and refrigerant is described below:

$$\dot{m}_{air}c_p(T_{i,k,air,in} - T_{i,k,air,out}) = 0.5A \cdot HTC \cdot (T_{i,k,air,in} + T_{i,k,air,out} - T_{i,k,in} - T_{i,k,out})$$

The energy balance between the air and refrigerant is expressed in the following.

$$\dot{m}_{air}c_p(T_{i,k,air,in} - T_{i,k,air,out}) = \dot{m}_i(h_{i,k,out} - h_{i,k,in})$$

Dehumidification is considered when the calculated average wall/fin temperature of the tube segment, under assumption of dry surface condition, is less than the dew point of the air streams. The enthalpy difference method by McQuiston (1994) is used to solve the combined heat and mass transfer problem:

$$q = \rho_a h_m A \eta_{ms} \Delta h_m$$

where h_m is the mass transfer coefficient, which can be obtained with Chilton/Colburn analogy, η_{ms} is the surface effectiveness with combined heat and mass transfer, Δh_m is the mean enthalpy difference.

Solution methodology

On the refrigerant side, a fractional step method is used wherein the hydraulic equation (pressure/mass flow rate relationship) and energy equation (heat transfer between refrigerant and air) are solved alternatively and repeatedly. In this way, the highly nonlinear system of equations is decoupled and non-linearity is reduced so that the solver becomes more robust. At the beginning, the pressure field and mass flow rate of each tube are obtained for the entire heat exchanger assuming no heat transfer between the refrigerant and the air, and the enthalpy of the refrigerant throughout each circuit is assumed to be equal to the inlet enthalpy of each circuit. When solving the heat transfer equation of each tube, the mass flow rate is obtained from the previously solved hydraulic equation, the inlet enthalpy is obtained from the currently solved heat transfer equations of the upstream tubes and energy balance equation at the upstream junction.

The hydraulic equations are solved with the Newton-Raphson method, and the energy and heat transfer equations are solved with the successive substitution method

On the air side, initially the air condition (temperature and humidity) facing each segment of each tube in the coil is assumed to be the same as that at the frontal face area. During solution of the energy equation of the refrigerant at each segment, the condition of the air leaving each segment is also calculated. After solving the hydraulic equations of the refrigerant in the entire coil, the air conditions facing each segment are updated, and used in solving the energy equation for the refrigerant side in the next step. These processes are repeated until both the refrigerant and air conditions do not change within a specified tolerance.

SIMULATION RESULTS

The simulation program was run to illustrate the reliability of the conversion and to demonstrate that the results are reasonable. Verification is in progress.

An arbitrary circuitry intended to be more complex than expected in reality was constructed in order to test the capability of the simulation tool to simulate heat exchangers with tubes connected in an arbitrary fashion (Figure 1). It shows the mass flow rate through each tube (represented by tube number on the abscissa) respectively. Similar to an electric circuit, tubes connected in parallel decrease the overall flow resistance, while tubes in series increase the overall flow resistance, and the mass flow rate through each tube varies accordingly.

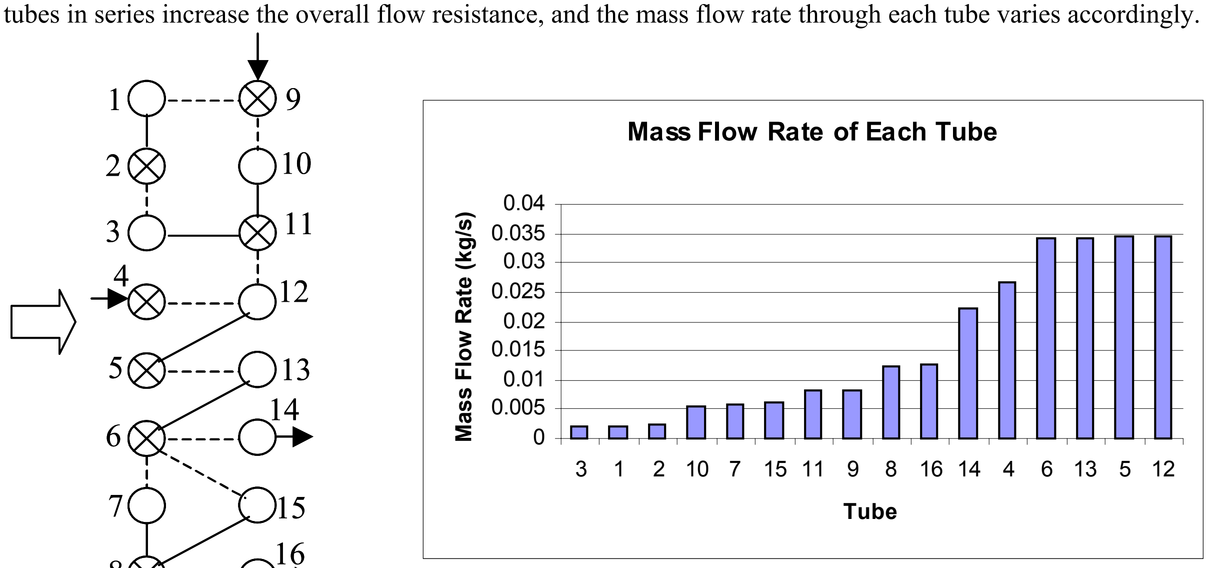


Figure 1 Mass Flow Rate through Each Tube in an Arbitrary Circuitry

Configurations of cross-counter flow and parallel-flow were also compared (Figure 2). It is an absorber using ammonia/water mixture as the working fluid. Figure 3 shows the average refrigerant temperature in each row and the average air temperature between each row. The inlet air temperature is indicated at the left side. The inlet refrigerant temperature is shown at the left side for parallel flow and at the right side for counter flow. As is well established, in a parallel flow configuration, the outlet temperature of the hot streams is never lower than the outlet temperature of the cold streams (air). In counter flow configuration, each tube plays an almost equivalent role in the amount of heat transfer, and the total heat duty of the heat exchanger is considerably larger than that in the parallel flow configuration (Figure 4).

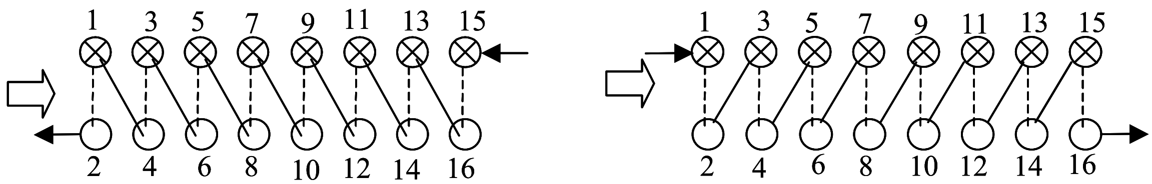
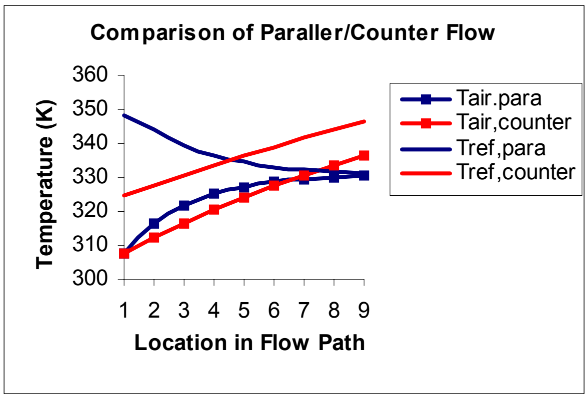


Figure 2 Cross-Counter and Cross-Parallel Flow



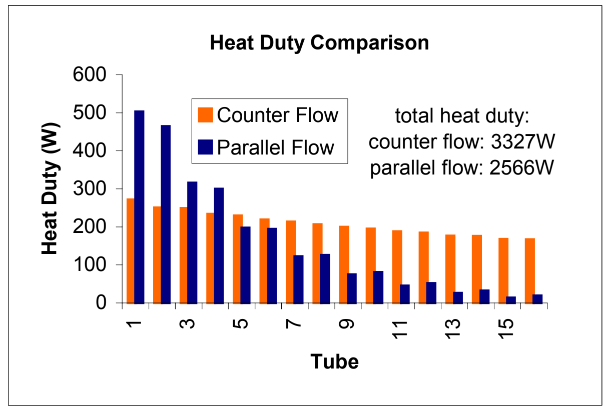


Figure 3 Refrigerant and Air Temperature

Figure 4 Heat Capacity of Each Tube

Figure 5 is a comparison of the sensible heat with latent heat of each tube in a coil configured of 24 tubes, when air dehumidification occurs on the tube surfaces with 70% of inlet air relative humidity. The sensible heat is considerably higher than the latent heat. At the last four tubes in the path of refrigerant flow, no condensation of water vapor took place, due to the relatively high wall/fin temperature of the tubes.

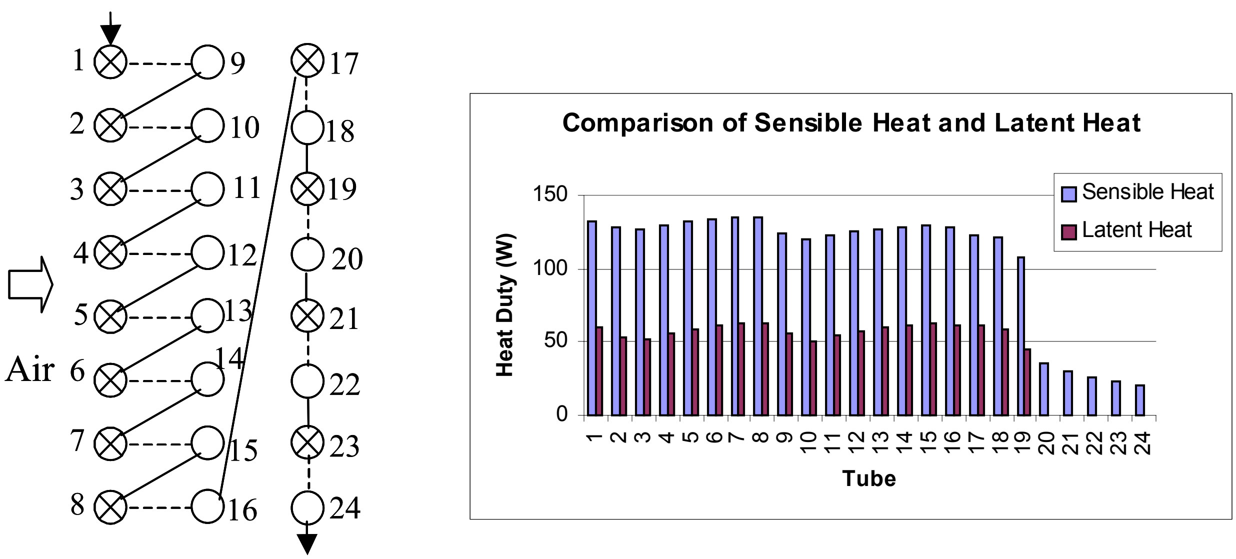


Figure 5 Sensible Heat and Latent Heat of Each Tube

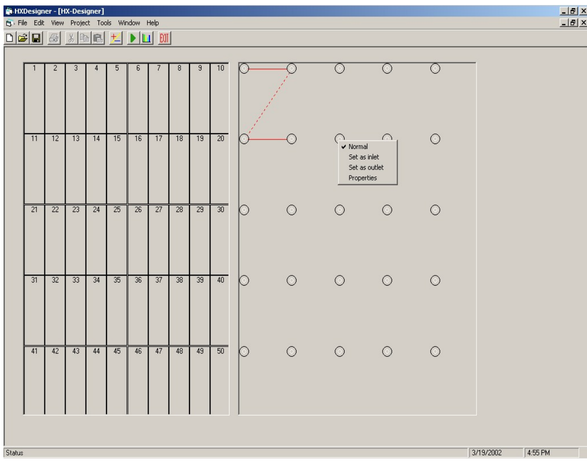
FLUID PROPERTY LIBRARY

A refrigerant fluid property library "CEEEProperties" that is customized from NIST REFPROP Version 6.01(1998), and an ammonia-water mixture library written by R.Tillner-Roth and G.Roth (1998) are integrated with the simulation tool.

GUI DESCRIPTION

The HXDesigner user interface was developed with the objective of providing a user-friendly interface for using the network solver mentioned in this text. The user-interface was developed using Microsoft Visual Studio 6.0. The Visual Basic interface provides the front-end for the solver developed using C++.

The main components of the user-interface are the project window, which guides the user through the different steps in setting up the heat exchanger, and the heat exchanger window that provides a lateral view of the tubes so that the user can connect them with the help of pointing device such as a mouse.



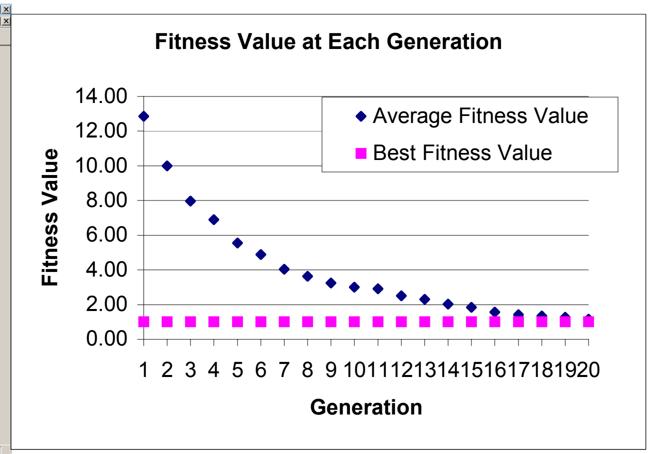


Figure 6 GUI Project Wizards

Figure 7 Fitness Value at Each Generation

Figure 6 shows a screen shot of the application after the steps in the project window have been completed. The left-hand side of the window displays rectangles that indicate the segments of the tubes. Clicking each of the segments provides a window where the user can input the values of air temperature, humidity and velocity. On the right-hand side of the window, the corresponding tube-ends are displayed. A right-mouse click on the tube-end pops-up a menu where the user can select whether the tube is an inlet to the heat exchanger or an outlet. Consecutive left clicks on two different tubes connects them with a red line. The user can also use the tube connection tool to connect or disconnect tubes.

After performing a calculation, the user is presented with the heat duty value and refrigerant charge, and has an option of exporting the output to a spreadsheet.

The GUI features are listed below: User-friendly navigation menus; Ability to select different heat-transfer and pressure-drop correlations from an available list of correlations; Exporting output data (temperature/pressure at segments) to spreadsheet (Microsoft Excel); Support for SI as well as English unit systems; Saving a heat exchanger configuration to a file and retrieving it. The file format is such that it can be read by any other application or platform that supports XML (Extended Markup language); Modular coding approach facilitates the use of the solver in other applications or system level solvers.

OPTIMIZATION WITH GENETIC ALGORITHM

Genetic algorithm (GA) is a search procedure based on the principles of genetics and selection. The variables are represented with genes in a string (chromosome). There are three basic operators operating on strings: reproduction, crossover, and mutation, which provide the power of searching complex and large solution spaces with relative ease and high efficiency. GA is especially applicable to combinatorial mixed discrete/continuous variable optimization problems.

To demonstrate the application of GA to the heat exchanger optimization, a sample case was studied. Variables to be optimized include the number of rows, number of tubes of each row, option of parallel-cross flow and counter-cross flow, and tube diameter D. The objective is to minimize the total heat transfer area (lowest cost), at a given heat load, subject to maximum allowable pressure drops of the refrigerant flow and air flow. The external penalty method is adopted to handle the constraints. The fitness value is defined depending on the calculated constraint variables. Figure 7 shows the best and average fitness values of each generation. Since the population size of 50 is quite large for a 7-bit string, the best string is immediately caught at the beginning. The average fitness value of the 50 strings at each generation is monotonically decreasing as the GA operates.

CONCLUSION

A generalized simulation and optimization tool for coil design has been developed. It can simulate heat exchanger performance for single-phase or phase-change fluids including a variety of pure refrigerants, mixed refrigerants, and ammonia-water mixtures. The tube circuitry/geometry, number of inlet/outlet streams, and correlations for heat transfer coefficients and pressure drops of both the refrigerant and air side, are at the user's choice. Mal-distribution of air flow can be accounted for by assigning individual property and mass flow rate to the segments of the frontal area tubes. A genetic optimization program can be integrated with the simulation model. Both discrete and continuous variables including number of tubes, number of rows, tube diameter, tube length, tube spacing, fin thickness, fin spacing, and refrigerant flow circuitry, can be optimized, in order to minimize heat transfer area, volume, capital cost, and operating cost. Validation of simulation results with experimental data and more optimization case studies are to be reported in the next paper of this continuing program.

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