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Thermoeconomic Optimization of an Air-Cooled Tube-Bank Condenser

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A model of an air-cooled condenser consisting of a tube bank in different flow conditions is analyzed. The phase change of the condensing fluid inside the tubes, in this case refrigerant R-134a, is explicitly considered in the heat transfer process. Use is also made of existing correlations to estimate the pressure drop at the interior and the exterior of the tube bank and the heat transfer coefficients both between the fluid and the tube wall and between the tube wall and the surrounding air. The entropy generation minimization method is applied to this system leading to an optimal thermodynamic design. This involves the air speed, the tube bank configuration (either in line or staggered), the spacing between adjacent tubes, the tube diameters and the area required for a specific condensation duty. The previous design is subsequently combined with a cost analysis to construct an objective function for the final thermoeconomic optimization of the condenser. The results suggest that the major source of irreversibility is in the air at the exterior of the tubes and that the most influential variables on the performance of the system are the air speed and the configuration of the bank.

1. Introduction

Good engineering design must rely on solid knowledge about the physical principles behind processes involved in the performance of particular devices. Amongst the most ubiquitous devices in many practical applications, compact heat exchangers stand on their own right. Therefore it is not surprising that they have received a lot of attention in the specialized literature. Most devices make use of water as the cooling fluid. However, many important chilling applications such as air conditioning systems, refrigeration, etc. also use air for that purpose. Although it has the limitation of a low thermal conductivity, and hence a poorer performance in heat transfer than water, a particular advantage of air is its abundance and low cost. A major aim of this paper is to get some insight with respect to its likely performance in a condenser constituted by a horizontal tube bank. Of particular interest is to conduct a detailed analysis of the local entropy generation in the whole system depending on the operating and design conditions. Specifically, we will present the case in which the condensing fluid is the well known refrigerant 134a.

Previous studies along similar lines have already been reported. In particular Lin et al. (2001) considered the condensation process of saturated FC-22 vapor flowing through horizontal cooling tubes. They found an optimum Reynolds number that minimizes the entropy generation rate in the case of a single tube and, for the multi-tube case and under certain constraints such as a fixed area, an optimal cooling temperature that also yields a minimum of entropy for a given condensation duty. This temperature was found to depend strongly upon many process parameters such as mass flow rate and tube geometry. On the other hand, Khan et al. (2007) analyzed the heat transfer from the tube bank to the cooling air for two configurations: staggered and in line. They obtained an expression for the generalized entropy production of the process which was then used to derive the conditions for a minimum. However, they used a fixed size for the equipment and neglected the condensing fluid inside the tubes, thus assuming that the controlling irreversibility is the heat transfer between the external tube wall and the air. Here, taking as starting point a simple mathematical model inspired by the

work of Khan et al. (2007), we perform a similar analysis but also include the consideration of condensation of the working fluid inside the tubes. Furthermore, the required area for heat transfer is not fixed but rather determined according to the operating conditions. Temperature and pressure profiles are determined and the location where the main irreversibilities occur may also be obtained from the model. As shown below, this allows us to evaluate the contribution due to the phase change of the fluid to the heat transfer across the tube walls and from the tube walls to the surrounding cooling air. Once the transport problem has been solved, costs are also included so that a thermoeconomic analysis is also feasible and carried out.
The paper is organized as follows. In the next Section we introduce the model for the tube bank and derive the governing equations for heat transfer corresponding to such model. This is followed in Section 3 by the derivation of the thermodynamic and thermoeconomic target functions to be optimized and by a brief description of the optimization method. Section 4 contains the results stemming out of our formulation in terms of the chosen relevant variables both for the temperature and pressure profiles and the ensuing analysis of the entropy generation and optimal economic performance. The paper is closed in the final section with further discussion and some concluding remarks.

2. Mathematical model

The model developed corresponds to a crossflow air-cooled condenser, with air flowing outside the tube bank and a pure substance condensing inside the tubes. The balance equations for the volume element shown in Figure 1 are presented in three groups corresponding to condensing fluid, air and tube walls, respectively.

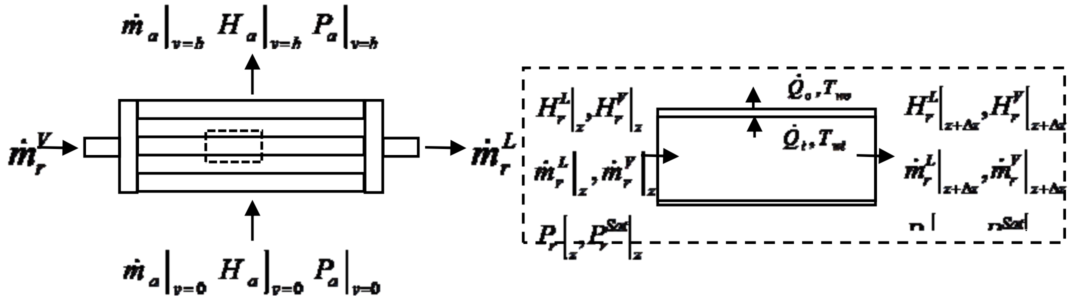

Figure 1. Schematic representation of the condenser.

The equations that describe the behavior of the condensing fluid inside the tubes are the mass and energy balances (Eqs. 1 to 3), the pressure drop equation (Eq. 4) and the phase equilibrium condition (Eq. 5) given by

$$\dot{m}_r^L\Big|_z + \dot{m}_r^V\Big|_z - \dot{m}_r = 0 \tag{1}$$

$$\dot{m}_r^L\Big|_{z+\Delta z} + \dot{m}_r^V\Big|_{z+\Delta z} - \dot{m}_r = 0 \tag{2}$$

$$\dot{m}_r^L H_r^L\Big|_{z+\Delta z} - \dot{m}_r^L H_r^L\Big|_z + \dot{m}_r^V H_r^V\Big|_{z+\Delta z} - \dot{m}_r^V H_r^V\Big|_z + \dot{Q}_i = 0 \tag{3}$$

$$P_r\Big|_{z+\Delta z} + \Delta P_r - P_r\Big|_z = 0 \tag{4}$$

$$P_r\Big|_{z+\Delta z} - P_r^{sat}\Big|_{z+\Delta z} = 0 \tag{5}$$

where $\dot{m}_r^L$, $\dot{m}_r^V$, $\dot{m}_r$ are the liquid, vapor and total mass flow rates; H_r^L, H_r^V the liquid and vapor specific enthalpies, $\dot{Q}_i$ the rate of heat transferred into the inner pipe wall, P_r, ΔP_r are the pressure and pressure drop, and P_r^{sat} is the saturation pressure of the condensing fluid. The mass, energy and pressure drop equations for the condenser air-side are:

$$\dot{m}_a\Big|_{y=B} - \dot{m}_a\Big|_{y=0} = 0 \tag{6}$$

$$\dot{m}_a H_a\Big|_{y=B} - \dot{m}_a H_a\Big|_{y=0} - \dot{Q}_0 = 0 \tag{7}$$

$$P_a\Big|_{y=B} + \Delta P_a - P_a\Big|_{y=0} = 0 \tag{8}$$

where $\dot{m}_a$, H_a are the air mass flow rate and specific enthalpy, $\dot{Q}_0$ is the heat transferred from the outer pipe wall into the air, P_a , ΔP_a are the pressure and pressure drop of the air flowing through the tube bank. The energy balance is the only equation considered in the mathematical modeling of the pipe walls, namely

$$\dot{Q}_i - \dot{Q}_0 = 0 \tag{9}$$

Here $\dot{Q}_i$ and $\dot{Q}_o$ are the heat transfer rates from the condensing fluid into the inner pipe wall and from the outer pipe wall into the air. Eqs. (1) to (9) must be supplemented with correlations to calculate pressure drops (ΔP_r, ΔP_a), heat transfer coefficients to calculate heat transfer rates ($\dot{Q}_i$, $\dot{Q}_0$) and vapor pressure (P_r^{sat}). This set of equations allows the calculation of twelve variables, namely: $P_r\big|_{z+\Delta z}$, $T_r\big|_{z+\Delta z}$, $\dot{m}_r^L\big|_{z+\Delta z}$, $\dot{m}_r^V\big|_{z+\Delta z}$, $\dot{m}_a\big|_{y=B}$, $T_a\big|_{y=B}$ $P_a\big|_{y=B}$, $\dot{Q}_i$, $\dot{Q}_o$, $P_r^{sat}\big|_{z+\Delta z}$, $T_{w,i}$, $T_{w,o}$. The last two correspond to the inner and outer wall temperatures, respectively. The other variables, corresponding to the inlet conditions must be specified.
The model equations are solved in the discretized form presented above using sufficiently small Δz (1x10^{-2} m), determined from a numerical sensitivity analysis, to be a good approximation of a differential model in coordinate z , avoiding the complications of solving a differential algebraic system. The condenser segments are solved sequentially until all vapor is condensed.

3. Optimization

In this work the condenser is optimized using a thermodynamic objective function and a thermoeconomic objective function. This approach serves to identify the similarities and differences of the optimal designs depending on the optimization criterion.

3.1 Objective function

Thermodynamic optimization. This approach aims to maximize the thermodynamic efficiency of the condenser. Such an objective can be achieved finding the conditions that minimize the entropy production rate in the condenser $(ds/dt)_{irr}$ given by

$$\left(\frac{ds}{dt}\right)_{irr} = \dot{m}_a s_a\big|_{y=B} - \dot{m}_a s_a\big|_{y=0} + \dot{m}_r s_r\big|_{z=l} - \dot{m}_r s_r\big|_{z=0} \tag{10}$$

where $s_r\big|_{z=0}$, $s_r\big|_{z=l}$, $s_a\big|_{y=0}$, $s_a\big|_{y=B}$, are the inlet and outlet entropies of the condensing fluid and air respectively.

Thermoeconomic optimization. This approach aims to minimize the cost of the condensing refrigerant, taking into account the cost of the heat exchanger (given by its area), and the cost of the cooling air regarded as a function of the pressure drop on the air side. To this end, an optimum configuration of the heat exchanger (number of pipes, pipe diameter, tube spacing, air velocity, and so on) must be found. The thermoeconomic objective function TOF is given by

$$TOF = \dot{\Pi}_1 + \dot{Z}_1 + \dot{Z}_2$$

where $\dot{\Pi}_1$, $\dot{Z}_1$, $\dot{Z}_2$ are the thermoeconomic cost of power consumption in the fan, the depreciation cost of the fan and the depreciation cost of the heat exchanger, respectively. The cost index for heat exchanger and fan were taken from Loh (2002) and Couper et al. (2012).

3.2 Design variables

Both thermodynamic and thermoeconomic optimizations use the same decision variables for staggerd and inline configurations, namely: air speed ($v_{min} \leq v \leq v_{max}$), external pipe diameter ($D_o = [D_1, D_2, \ldots, D_n]$), tube bank row and column combinations ($[Ro,Co] = [(Ro_1,Co_1),(Ro_2,Co_2), \ldots, (Ro_m, Co_m)]$) and dimensionless tube pitch ($St = [St_{min} \leq St \leq St_{max}]$). The nature of the objective functions, the model and design variables lead to a mixed-integer nonlinear optimization problem.

4. Study case

The specific problem analyzed is the condensation of 5 kg/s of R-134a that enters the condenser as saturated vapor at 320 K and exits it as saturated liquid, using air at 280 K as cooling fluid. The allowed values of the decision variables are: $1 \le v \le 6$ m/s, D_o = [12.7, 19.05] mm, [Ro,Co] = [(10,5),(9,6), (8,6), (8,8), (6,8), (6,9), (5,10)] and $1.2 \le St \le 3$.

4.1 Thermodynamic properties, heat transfer coefficients and pressure drop

Thermodynamic properties of R-134a were taken from REFPROP™, while air (regarded as an ideal gas) and tube wall (copper) properties were taken from Cengel et al. (2011). The in tube pressure drop and heat transfer coefficient were estimated from the correlations given in Ould Didi et al. (2002). The pressure drop and the heat transfer coefficient in the condenser air side are estimated using the correlations proposed by Khan et al. (2007).

4.2 Model solution and optimization routines

The model was implemented in MATLAB™ language. The *fsolve* function was used to solve the model equations, and the optimization was carried out using the *ga* function that is a genetic algorithm able to solve MINLP problems (Panjeshahi et al., 2010); further information about these functions is available in the user's guide (Mathworks, 2014).

5. Results and discussion

5.1 Entropy production distribution

Figure 2 shows the temperature and entropy production profiles for an staggered condenser with 64 tubes (8 rows and 8 columns) of 12.7 mm (external diameter, BWG: 12), an air inlet speed of 3 m/s and a tube pitch of 1.3. The main entropy production source is located in the condenser air-side with 96.2 % of the total entropy production (279.86 W/K), while the entropy production at the tube side and wall account for 3.7 % and 0.1 %, respectively. The same case, using the inline configuration, generates a similar entropy production distribution than in the staggered configuration (air-side: 96.3 %, tube-side: 3.6 % and pipe wall: 0.1 %) though the total entropy production is different (290.49 W/K). The greater entropy production in the inline case is directly associated with differences in the condenser size required in the operation (staggered: 154.6 m^2, inline: 179.3 m^2) since the air outlet temperature and pressure drop are greater for the staggered case.

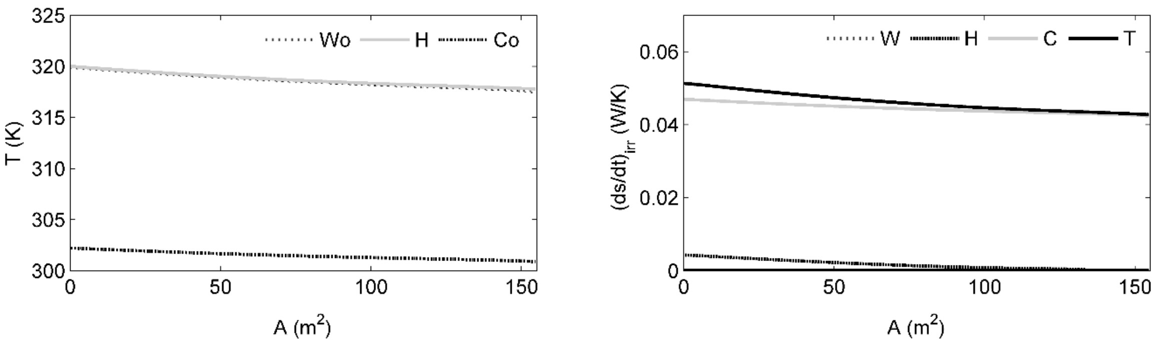

Figure 2. Staggered configuration. A. Temperature profiles: external tube wall (Wo), R134a (H), outlet air temperature (Co). B. Entropy production profiles: tube Wall (W), R-134a (H), air (C), total (T).

The influence of inlet air speed and tube pitch on the entropy production (Figure 3) shows that for air speed below 3.4 m/s the entropy production decreases monotonously with tube pitch for both, inline and staggered arrangements. For speeds greater than 3.4 m/s the lowest entropy production rates correspond to tube pitches from 1.4 to 2.4 (inline) and from 1.4 to 1.6 (staggered). The difference in this behavior is the irreversibility caused by air-side pressure drop, especially above 5 m/s, in using a staggered arrangement instead of the inline one (data not shown). These observations qualitatively agree with the analysis of an air-cooled cross-flow heat exchanger with constant tube wall temperature performed by Khan et al. (2007). The minimum condenser area, both for inline and staggered arrangements, is given by conditions of maximum air speed (10 m/s) and minimum tube pitch (1.25), 71.6 m^2 (inline) and 60.9 m^2 (staggered). A sensitivity analysis of air speed and tube pitch shows that the former has the biggest influence on condenser size, especially for velocities lower than 4 m/s.

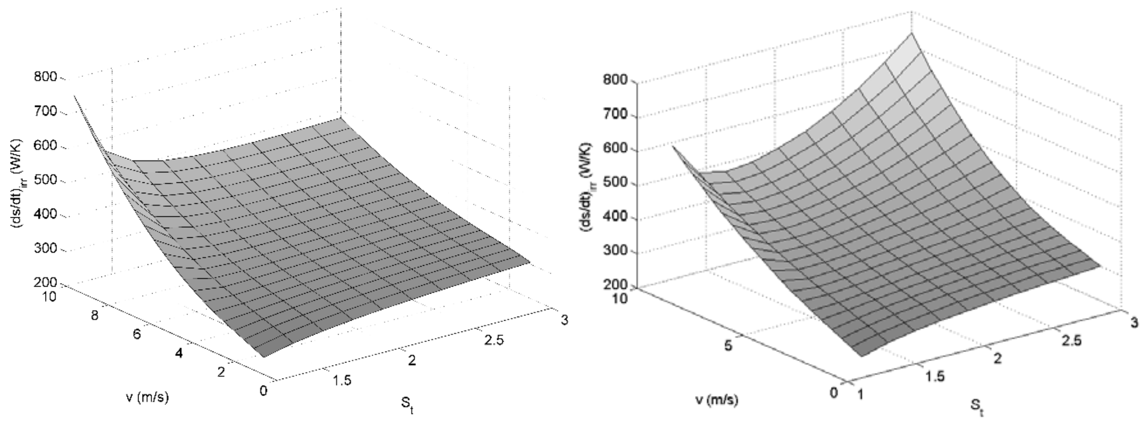

Figure 3. Entropy production rate as a function of air-inlet speed and tube pitch. A. Inline B. Staggered.

5.2 Thermodynamic optimization

The thermodynamic optimum designs for the case presented in Section 4 favors an external tube diameter of 19.05 mm and a staggered configuration in all cases (see Figure 4). This result differs from the one reported by Khan et al. (2007) which found the inline configuration thermodynamically more favorable than the staggered configuration. This apparent discrepancy is explained by the fact that they analyze a heat exchanger performance with fixed heat transfer area (rating problem). In contrast, the present work treats the heat transfer area as variable to fulfill a specified heat duty (design problem). In fact, our results show that the inline configuration generates less entropy locally. Nonetheless, it is possible to find designs with lower overall entropy production for every inlet velocity using a staggered configuration, mainly because the lower areas required in staggered designs makes the overall entropy generation to be lower than the overall entropy produced in the inline configuration.

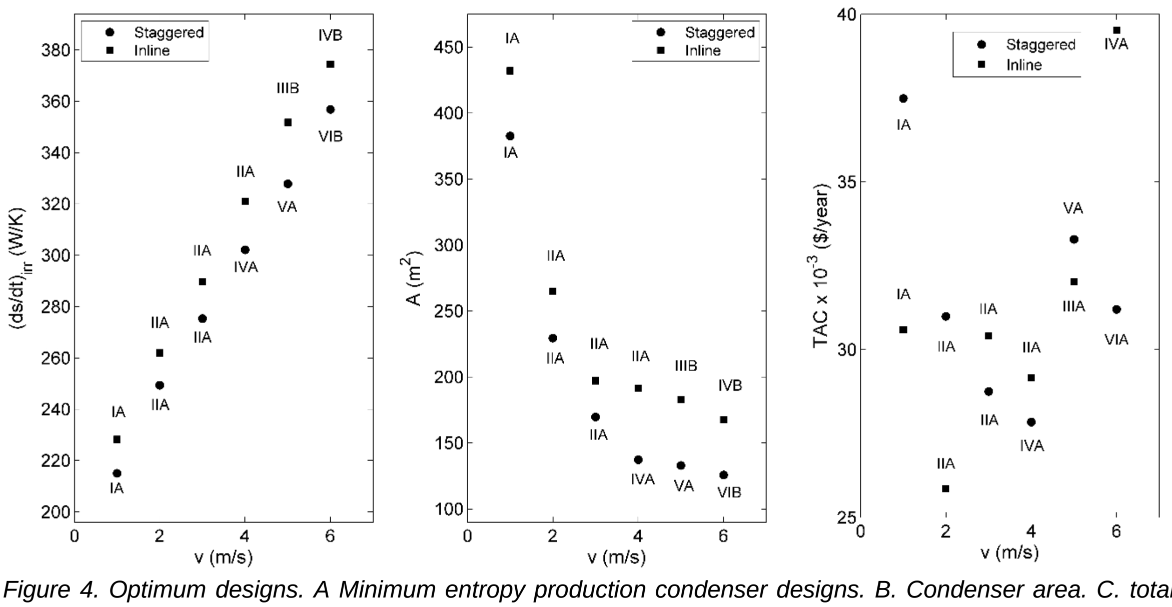

Figure 4. Optimum designs. A Minimum entropy production condenser designs. B. Condenser area. C. total annualized cost. I: (S_t =1.25), II: (S_t =1.30), III: (S_t =1.51), IV: (S_t =1.72), V: (S_t =2.57), VI: (S_t =2.99); A: ([Ro Co]=(10,5)), B: ([Ro Co]=(8,8))

Optimal pitch and row-column arrangement tend to be *similar for both* inline and staggered configurations when the inlet-air velocity is below 3 m/s. These solutions are characterized by a relative narrow pitch and a greater number of tubes in the flow direction (row). These designs tend to increase the air-side heat transfer coefficient, while solutions for velocities seek diminishing the pressure drop by increasing the tube pitch and the number of tubes in a row. Heat transfer areas tend to decrease as the inlet-air velocity increases, see Figure 4. A sharp decrease in heat transfer areas is noticeable for velocities below 3 m/s in the inline configuration and 4 m/s in the staggered configuration. Heat transfer areas using the inline configuration are from 12 % (1 m/s) to 33 % (4 m/s) greater than the ones required for staggered optimal design for the same air-inlet velocity. Despite this fact, entropy production differences between inline and staggered configurations tend to be similar in the velocity interval analysed.

5.3 Thermoeconomic optimization

The thermoeconomic cost of the condensation process is a function of the capital cost: heat exchanger area and auxilliary devices (fan and compressor) and also of the operating cost: energy demanded by the operation. Since the irreversibilities generated in the tube side are a small part of the total of irreversibilities (see Section 5.1), the variation of the compressor capacity and power consumption to satisfy the pressure drop is neglected in this study. Therefore the present analysis will focus on what occurs on the air-side.

According to the results shown in Figure 4, a lower TAC would be expected for designs using staggered configurations in all cases, since they demand lower heat transfer area and generate lower entropy than inline optimal designs. However, optimal thermodynamic designs using an inline configuration are cheaper for velocities of 1, 2 and 5 m/s. A closer analysis on fan power consumption indicates that it depends both on air volumetric flow and on air pressure drop. In this regard, the air pressure drop in a staggered configuration is at least one order of magnitude bigger than in its inline counterpart which in turn increases fan power consumption an order of magnitude. This difference along an operation year makes a huge economic difference in this operation, thus indicating that low entropy production by itself does not guaranty a process with low operational costs. It is necessary to keep in mind that the economic value of the irreversibility depends on the source of irreversibility. In this case, irreversibilties associated with fan pressure drop tend to be more expensive than the one related to condenser thermal performance.

6. Conclusions

This work has focused on the thermodynamic optimization of a crossflow air-cooled condenser. The main findings are the following: (i) The main irreversibility in this process is located in the condenser air-side. Therefore, design and operation strategies should focus on improving this part of the condenser. (ii) In-tube pressure drop can be regarded as the main source of irreversibility in the condenser tube-side, since heat transfer irreversibility is concentrated in the condenser air-side. Therefore, the tube diameter is the most important decision variable in order to regulate the entropy production in the condenser tube-side. (iii) The inlet air speed has the biggest influence on the entropy production and condenser size. Therefore, it is the main decision variable to consider in the condenser thermodynamic optimization. (iv) Shorter condensers with greater entropy production may be globally more efficient than large condensers with lower local entropy production. Nonetheless, the economic cost of the entropy production source can be different and so the lower operational cost is not necessarily associated to the process with least entropy production.

Finally, it is worth pointing out that the previous analysis confirms the fruitful outcome of combining thermodynamic performance and economic analysis in engineering design.

References

Cengel Y. A., Ghajar A. J., 2011, Heat and mass transfer, McGraw Hill, New York, USA.

Couper R. J., Penney R. W., Fair. R. J., Walas M. S., 2012, Chemical Process Equipment: Selection and Design, Butterworth – Heinemann, USA.

Khan W. A., Culham J. R., Yovanovich M. M., 2007, Optimal design of tube banks in crossflow using Entropy Generation Minimization Method, Journal of Thermophysics and Heat Transfer, 21, 372-378.

Lee D.J., Peng X.F., 2001, Condensation performance in horizontal tubes with second-law consideration, International Journal of Energy Research, 25, 1005-1018.

Loh, H.P., U.S., 2002, Department of Energy National Energy Technology Center, Final Report Process Equipment Cost Estimation, DOE/NETL-2002/1169.

Mathworks, 2014, Optimization toolbox^{TM} User's guide, Mathworks, US.

Ould Didi M.B., Kattan N., Thome J. R., 2002, Prediction of two-phase pressure gradients of refrigerants in horizontal tubes, International Journal of Refrigeration, 25, 935-947.

Panjeshahi M. H, Joda F. and Tahouni N., (2010), Pressure drop optimization in multi-stream heat exchanger using genetic algorithms., Chemical Engineering Transactions, 21, 247-252 DOI: 10.3303/CET1021042