DEVELOPMENT OF A DETAILED THERMAL MODEL FOR DESIGNING HEAT PIPE HEAT EXCHANGERS

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Abstract. *The conductance of heat pipes is different from line to line inside a heat exchanger bundle. For the purpose of pre-designing heat pipe heat exchangers, it is possible to assume that the heat pipe conductance is homogeneous along the bundle. For a detailed design, it is recommended to split the bundle modeling into sub-bundles, for assessing heat pipe conductance in several temperature levels.*

Another application of sub-bundle models is the designing of heat exchangers that are constructed with two or more sub-bundles that have different physical characteristics, such different fins, geometric arrangement, working fluids, heat pipe diameter, etc.

This work proposes a multi-bundle model for heat pipe heat exchangers, implemented in Engineering Equation Solver (EES) using a matrix structure. This structure allows the user to change the model complexity in a fast way, by only changing the size of the matrix, and consequently the number of sub-bundles that are being considered.

The model showed itself an useful tool for designing heat pipe heat exchangers. The model was also linked to a genetic algorithm routine for obtaining optimal synthesis of heat exchangers.

Keywords*: mathematical modeling, heat exchanger, thermosyphon*

1. INTRODUCTION

Increasing the energy efficiency is today an important challenge in all kinds of process industry - energy, oil, chemical, siderurgy, food and others. The heat exchanger is the key equipment for energy saving and reducing thermal pollution. Energy savings can be increased by using Heat Pipe Heat Exchangers (HPHEs). When properly designed, they may have reduced life cycle costs in comparison with conventional heat exchangers.

Proper thermal design of a Heat Pipe Heat Exchanger (HPHE) is essential to ensure economic viability. Poor design can lead to oversized or subsized equipments. Sometimes a technically adequated thermal design can be improved by means of finding a cheaper design solution that fits to the same thermal load.

On designing thermosyphon heat exchangers, the large number of design parameters result in several degrees of freedom. Several configurations that satisfy one same set of design restrictions can coexist.

Heat Pipe Heat Exchangers can be cheaper than conventional heat exchangers, upon a life-cycle cost analysis depending on the application. On designing thermosyphon heat exchangers, the large number of design parameters result in several degrees of freedom. Several configurations that satisfy one same set of design restrictions can coexist.

For instance, lower price and/or compact and/or low weight equipment configurations can be found by allowing the use of higher pressure losses. Using dense staggered bundles geometry can result in cheaper equipments in comparison with disperse and aligned bundles, both of them supplying the same thermal load.

1.1. Heat Pipe Heat exchanger design

The importance of a multi-parameter model for the design of HPHE's is recognized by literature. Silverstein (1992) proposed a simplified design method where the set of design parameters is modified manually and the interdependent variables are recalculated in an iterative process until a technically satisfactory solution is found. In a review of HPHE design optimization Faghri (1995) found only multivariable design simulation models, one of them coupled to an orthogonal regression analysis method for post-optimization. Bejan (1996) suggests a proper complexity for the thermal simulation model to be used on multivariable optimization, in order to avoid excessive computational work. Borges & Mantelli (2007b) proposed and implemented a single-bundle model (written in Visual Basic and Ms-Excel) coupled to a mathematical programming algorithm for prescribing optimal HPHE design solutions. Borges, Mantelli et al (2007b) proposed a Multi-objective programming model for dealing with conflicting objectives in HPHE design, such as equipment weight and flow head loss.

1.2. The need for a multi-bundle model

One of the use possibilities for the multi-bundle model is the design of heat exchanger with uncommon geometry, as different fin density for each row in the bundle, as well as different transversal pitch, HP diameter, work fluid on heat pipes (HPs), etc. In this way it is possible to create a heat exchanger more compact and cheaper.

For example, it is known that water is a largely used working fluid for thermosyphons, because it is a great and cheap working fluid but the working temperature cannot exceed, or even approach, the critic temperature of the water. This occurs because when this limit is reached, the water do not condenses, acting like a non-condensable gas, decreasing the equipment performance. So, to contour this problem we could take some actions, among them, change the working fluid on heat pipes or reduce the steam temperature inside the heat pipes. As no other working fluid is so cheaper than water, selecting another one could be economically impracticable, so reducing the steam temperature is the alternative. This can be done through reduction of the external tube convection coefficient, in other words, changing the heat pipe geometry. Because this, we would have bare tubes with a big transversal pitch, small external diameter, etc. In this case we use the multi-bundle model, so we could select different bundle settings for each row, ensuring the maximum performance for a heat exchanger, as well as the costs reduction. Another design strategy for the same problem is to use different finning density in each row of heat pipes. For this, a multi-bundle model is also very necessary.

In this work, a multi-bundle model is proposed and implemented. As will be seen further on, the performance of each bundle is calculated using a single-bundle model, and the multi-bundle model is a matrix linkage of single-bundles.

2. SINGLE BUNDLE ALGORITHM

The model considers the heat exchange between a hot stream and a cold stream through heat pipes that are organized in regular bundles, i.e., in a row-and-column rectangular geometry. Each heat pipe works as a heat superconductor. This modeling is based on a counter flow heat exchanger, as it is known this type of configuration is more effective than parallel flow.

2.1. The Heat Pipe Bundle

For a heat exchanger with thermosyphons the HP bundle is modeled as counter-flow (Figure 2) through round tubes.

In this case four types of setting are possible, 30°, 45°, 60° and 90°. This last is called aligned tube rows, while the others are called staggered tube rows ($\theta \neq 90°$).

The Figure 1 illustrates the heat exchanger setting, as well the work fluid flow, transversal pitch (xt) and longitudinal pitch (xl).

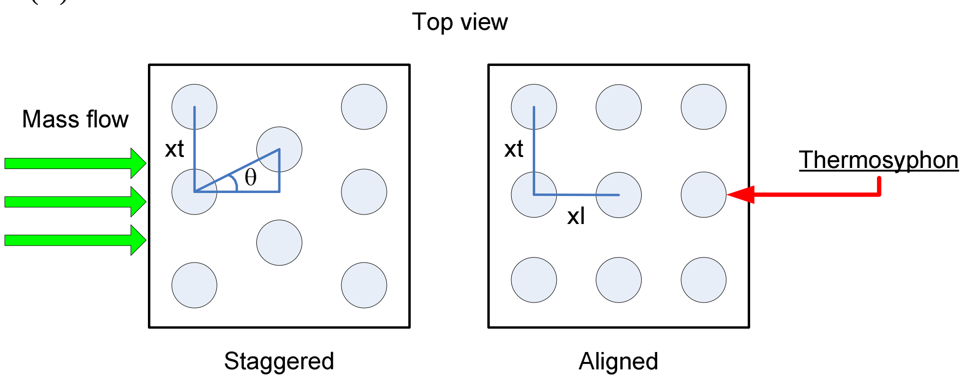

Figure 1 - Schematic view of a heat exchanger section.

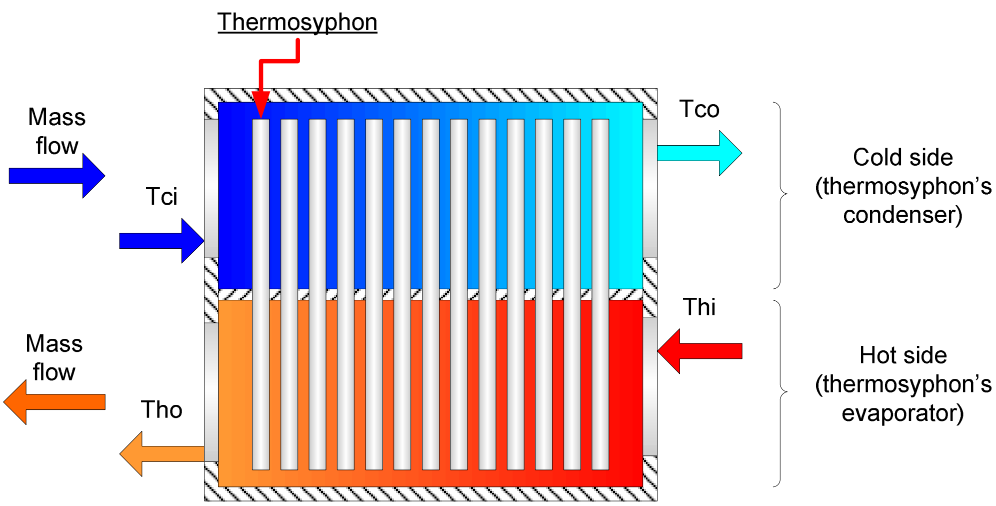

Figure 2 - Schematic view of a heat exchanger with thermosyphons.

For the proposed single-bundle model the assumptions adopted were:

- Turbulent boundary layer
- Steady state
- Incompressible and viscous external flow
- Thermal losses through heat exchanger walls negligible

2.2. Heat pipe bundle external heat transfer coefficient

External convection correlations for plane and finned bundle tubes were used, with option for staggered and aligned bundles. The external convection coefficient of one tube depends on geometrical configuration of the entire bundle, and some correction coefficients are applied when it has few rows. These correlations were obtained from Zukauskas (apud Incropera 2003).

2.3. The Thermal Resistances of a Thermosyphon Heat Pipe

Heat pipes considered in this work, are thermosiphons. They don't have a porous structure so the condensed fluid returns to evaporator by gravity. Due to this simplification, thermosyphons have a better performance than heat pipes.
For a thermosyphon, the thermal resistances can be seen in the Figure 3:

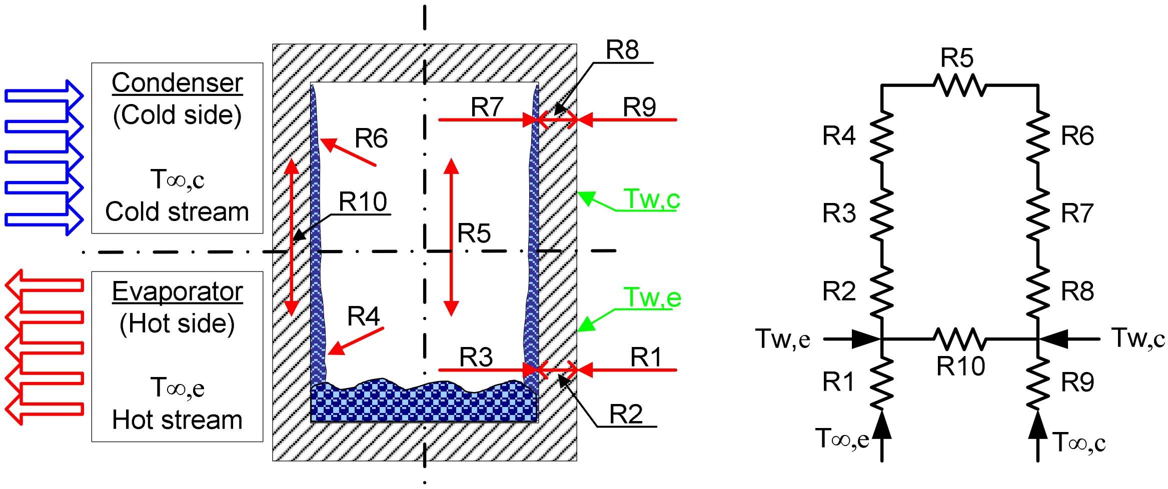

Figure 3 - Thermosyphon resistances diagram

Were the thermal resistances are:

R_1, R_9 convection outside the evaporator and condenser, respectively.

R_2, R_8 conduction through the evaporator and condenser walls, respectively.

R_3, R_7 boiling and condensation, respectively.

R_4, R_6 interface liquid-vapor in the evaporator and condenser. According to Brost (1996) these resistances are small and can be neglected.

R_5 pressure drop in the working fluid flow. This resistance is small and will be neglected.

R_{10} axial conduction, considering steam temperature constant along the heat pipe, this resistance is too big and can be neglected.

Empiric correlations found in lecture were used for this model. The implemented convective correlations for evaporator are from Kaminaga (1992a), Kaminaga (1992b), Stefan & Abdelsalam (1980) and Borishanski (1969). For the condenser, were used correlations from Kaminaga (1992b) and Groll & Rösler (1992).

Radial thermal conduction for round tubes here we use the thermal conduction equation found in Incropera (2003).

2.4. The Overall Heat Transfer Coefficient

For the determination of overall heat transfer coefficient are necessary the determination of the thermal resistances, see Figure 3. Then we can calculate the overall heat transfer coefficient as an inverse of the resistances sum.

2.5. The Adopted Algorithm

The algorithm considers the four tube rows setting for thermosyphons, as well the external convective resistances, the radial conduction resistances in heat pipe walls and the boiling and condensation thermal resistances in evaporator and condenser, respectively.

As one of the assumptions was turbulent boundary layer, the used equations are not adequate to laminar regimen. In finned heat pipe case, it was implemented the fin efficiency calculus, as well as its interference in the reduction of the transversal section in the exchanger, resulting in an increase in maximum speed of the work fluid between the heat pipe.

On this algorithm were also included several correlations for the calculus of the thermosyphons thermal resistances, which can be selected by the designer, as necessary.

In Figure 4 is shown the algorithm implemented on EES package.

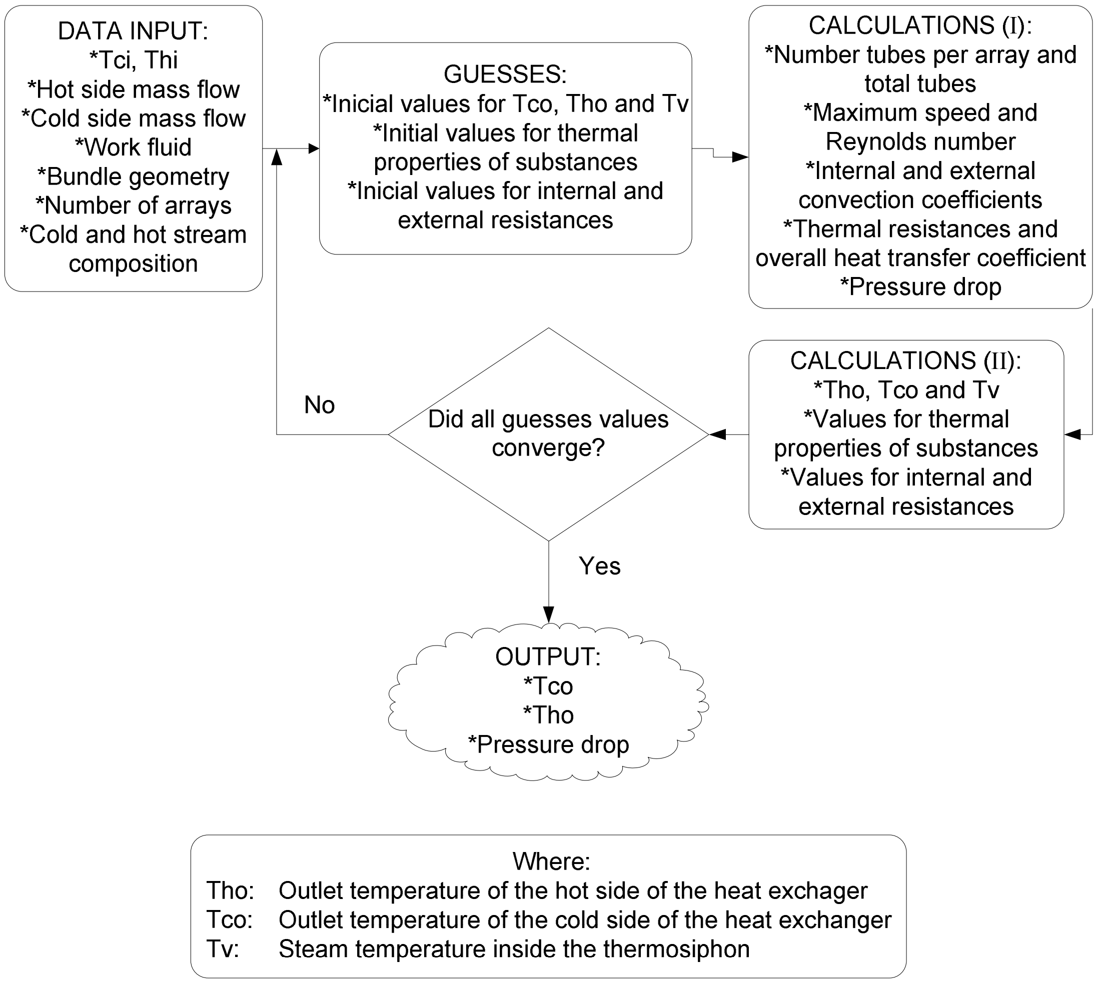

Figure 4 - Algorithm diagram

3. MULTI- BUNDLE ALGORITHM

Each sub-bundle inside the multi-bundle modeling is the same as the model created for the single-bundle, unless some changes in the calculus of the correction factors for external convection, that are based on total row number and total heat pipe number.

Since it is an iterative and recursive problem, with data interchange between the sub-bundles, it was proposed a matrix model implementation, to improve the accuracy in modeling. Also, the matrix model allow one to vary quickly the number of sub-bundles modeled, either improving the data input, because a worksheet could be copied from other programs.

A schematic of this multi-bundle modeling is presented in Figure 5.

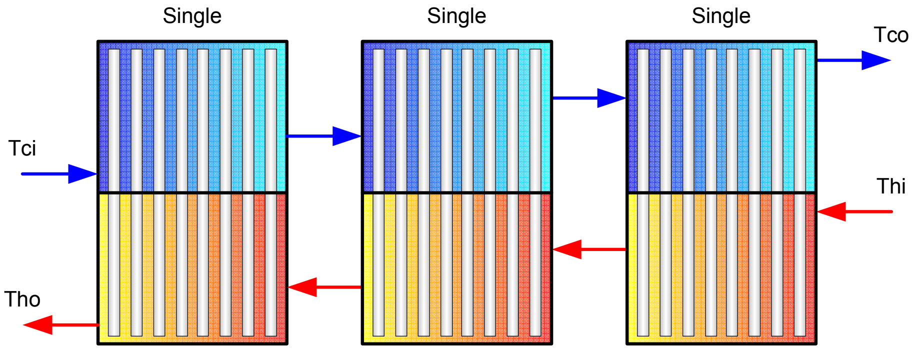

Figure 5 - Flow information diagram for 3 sub-bundles of 8 arrays.

3.1. Matrix Model

The proposed mathematical modeling uses a unique input data for each created sub-bundle, as shown in Figure 6.

$$
\begin{array}{ccc}
 & & \begin{array}{ccccc} Tci & & & Tho & \end{array} \\
\begin{bmatrix} P_1^1 & P_2^1 & \cdots & P_n^1 \end{bmatrix} & \begin{bmatrix} D_1^1 & D_2^1 & \cdots & D_n^1 \end{bmatrix} & \begin{bmatrix} T_{CI}^1 & T_{HI}^1 & T_{CO}^1 & T_{HO}^1 & T_V^1 \end{bmatrix} \\
\begin{bmatrix} P_1^2 & P_2^2 & \cdots & P_n^2 \end{bmatrix} & \begin{bmatrix} D_1^2 & D_2^2 & \cdots & D_n^2 \end{bmatrix} & \begin{bmatrix} T_{CI}^2 & T_{HI}^2 & T_{CO}^2 & T_{HO}^2 & T_V^2 \end{bmatrix} \\
\vdots & \vdots & \vdots \\
\begin{bmatrix} P_1^i & P_2^i & \cdots & P_n^i \end{bmatrix} & \begin{bmatrix} D_1^i & D_2^i & \cdots & D_n^i \end{bmatrix} & \begin{bmatrix} T_{CI}^i & T_{HI}^i & T_{CO}^i & T_{HO}^i & T_V^i \end{bmatrix} \\
\vdots & \vdots & \vdots \\
\begin{bmatrix} P_1^m & P_2^m & \cdots & P_n^m \end{bmatrix} & \begin{bmatrix} D_1^m & D_2^m & \cdots & D_n^m \end{bmatrix} & \begin{bmatrix} T_{CI}^m & T_{HI}^m & T_{CO}^m & T_{HO}^m & T_V^m \end{bmatrix} \\
 & & \begin{array}{ccccc} & Thi & Tco & & \end{array}
\end{array}
$$

Where:
m = array size
P_n^i = Fixed array parameters
D_n^i = Dependent array variables or calculated parameters
Tci = Inlet temperature of the cold side of the heat exchanger
Thi = Inlet temperature of the hot side of the heat exchanger
Tco = Outlet temperature of the cold side of the heat exchanger
Tho = Outlet temperature of the hot side of the heat exchanger
i=[2;m-1]

Figure 6 - Matrix algebra of the model.

In this modeling were assumed that the inlet temperatures in cold side (condenser) and hot side (evaporator), as well as the row number are prescript by user.

The hot side inlet temperature for each bundle is the hot side outlet temperature from the later bundle and the cold side inlet temperature for each bundle is the cold side outlet temperature from previous bundle, as shown in Figure 6.

The proposed model requires unique input data for each created sub-bundle. Thus, one can model a heat exchanger with m-sub-bundles and x-rows in each sub-bundle. In the case that x=1 and n≥1, each sub-bundle modeled is indeed one row of the heat exchanger, and we have the row-to-row calculations.

4. MODEL IMPLEMENTATION

The model was implemented in EES (Engineering Equation Solver) package. This choice was made because it is a Engineering-dedicated equation solver, facilitating the iterative and recursive calculations, with also a reasonable database of thermal properties. The graphic interface of the program is shown in Figure 7 and Figure 8.

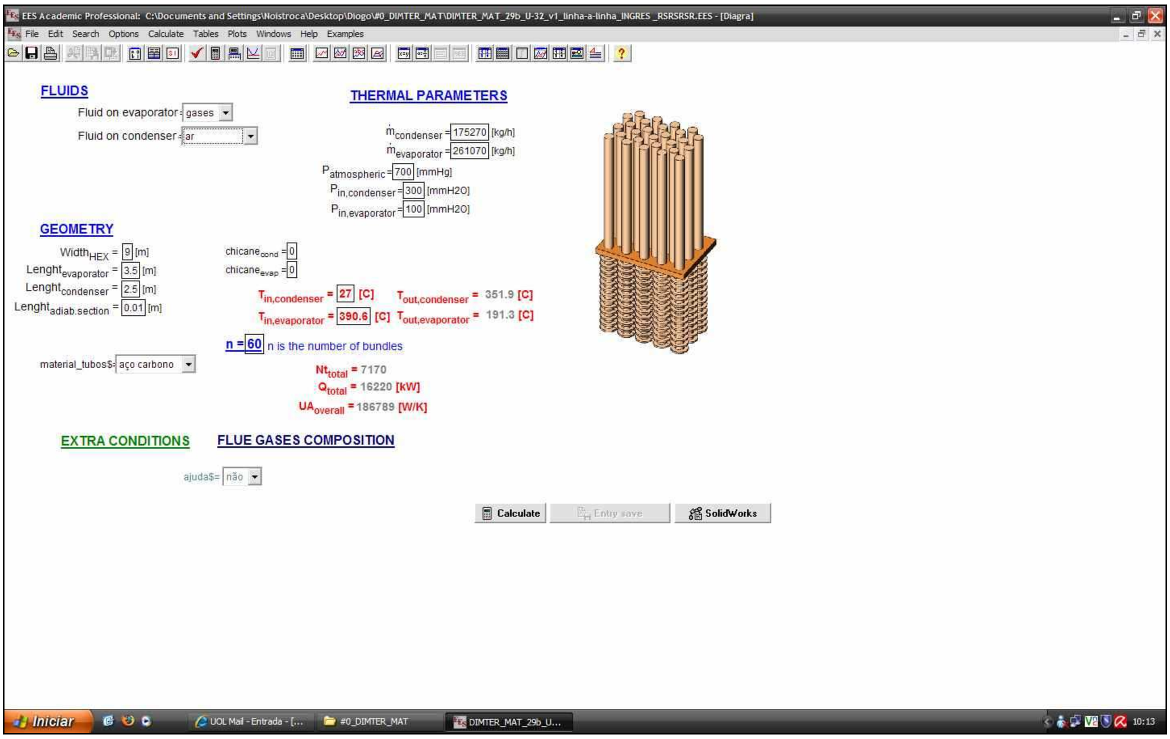

Figure 7 - Input interface for global parameters.

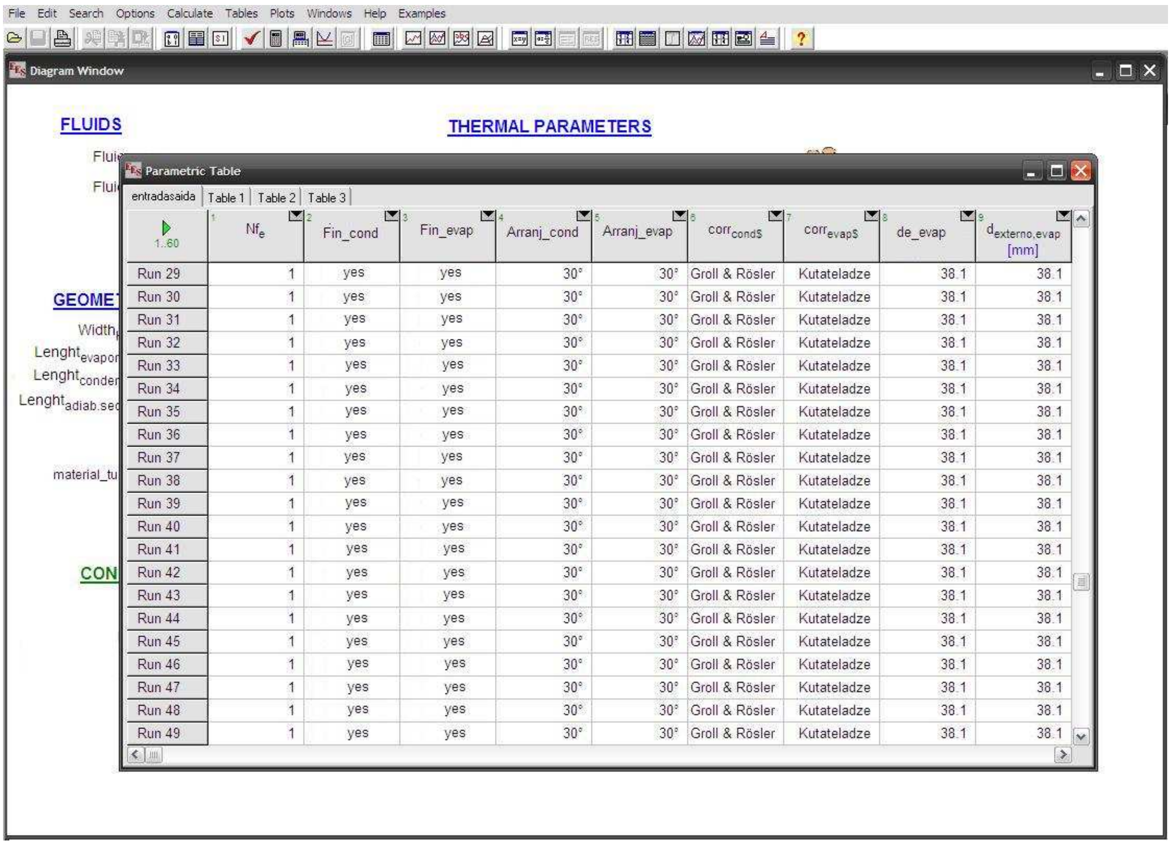

Figure 8 - Input interface for matrix parameters.

One should notice that due to model implementation inside an equation solver, it's possible to quickly change which are the input and which are the output variables. EES can fast exchange variables among the dependent and the independent variable groups. This is a good feature for post-design analysis, since is possible to do some sensitivity analysis of heat exchanger performance with respect to inlet streams variations.

There are some implementation details for linking sub-bundles. For instance, when staggered bundles are used, it is necessary to follow the j, j-1, j, j-1 … sequence of heat pipes per row. The implemented code automatically does this sequence.

5. CASE STUDY: AIR PREHEATER

A case study of designing a heat exchanger for an oil refinery was proposed. The design data are shown in Table 1.

Table 1 - Design data.

	Units	Cold side	Hot side
Fluids	-	Air	Flue gas
Flow rate	kg/h	175,270	261,070
Inlet temperature	°C	27	390.6
Max. Pressure drop	mmWC	120	120
Allowable width	m	9	9

A base-case design was proposed, using the model as a design tool. For this base-case proposition, several design parameters was choose by relying in the author's technical feeling. Further in this article, this design configuration will be optimized.

As the steam temperature inside the thermosyphons in hottest side of the exchanger were close to critical temperature of water, the designing needed to be split into two bundles, because it was necessary to use another work fluid inside the heat pipes apart from water. This division was made to reduce the cost of the heat exchanger. So, it would be two bundles, one operating until a secure temperature (Tv) for the water as an working fluid (Tv<300°C) and another operating with naphthalene (Tv>300°C).

Executing the program, we have the results in Table 2.

Table 2 - Design results from the multi-bundle model.

			First bundle (water)		Second bundle (naphthalene)	
		Units	Cold side	Hot side	Cold side	Hot side
PARAMETERS	Fluids	-	Air	Flue gas	Air	Flue gas
	Flow rate	kg/h	175,270	261,070	175,270	261,070
	Inlet temperature	°C	27	NA	NA	390.6
	Outlet temperature	°C	NA	NA	NA	NA
	Staggered	-	30°		30°	
	Number of HPs	-	2271		4899	
	Length	m	2.5	3.5	2.5	3.5
	Tube outer diameter	mm	38.1	38.1	38.1	38.1
	Transversal pitch	mm	75	75	75	75
	Fin step	mm	10	10	10	10
	Fin step thickness	mm	1	1	1	1
	Fin height	mm	15	15	15	15
RESULTS	Inlet temperature	°C	27	322	245	390.6
	Outlet temperature	°C	245	188	357	322
	Pressure drop	mmWC	23	32.42	68	81
	HPHE mass	ton	182.0			

In the Table 3 we can see the results for row-to-row design. As it would be too many values, we will put only the values for first and last row temperatures.

Table 3 - Output data from multi-bundle model with row-to-row calculations.

		Multi bundle model (row to row)	
Parameters	Units	Cold side	Hot side
Fluids	-	Air	Flue gas
Flow rate	kg/h	175,270	261,070
Inlet temperature (first row)	°C	27	200
Outlet temperature (first row)	°C	43	191
Inlet temperature (last row)	°C	350	390.6
Outlet temperature (last row)	°C	352	389.5
Pressure drop	mmWC	91	113.42
Staggered		30°	
Tube number	-	2271 (water) + 4899 (naphthalene)	
Length	m	2.5	3.6
Tube outer diameter	mm	38.1	38.1
Transversal pitch	mm	75	75
Fin step	mm	10	10
Fin step thickness	mm	1	1
Fin hight	mm	15	15

Graphically we can see the results for row-to-row calculus in Figure 9 where, Tci is the cold side inlet temperature, Tco is the cold side outlet temperature, Thi is the hot site inlet temperature and Tho is the hot side outlet temperature.

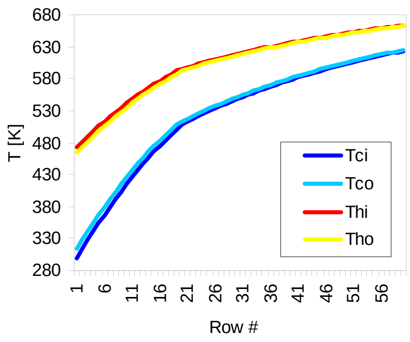

Figure 9 - Inlet and outlet temperatures for each row (the bundle is representative).

In Figure 10 we can see a plot of overall heat transfer coefficient (UA) as a function of cold and hot side mass flows, respectively m_c and m_e. These flow rates through evaporator and condenser varies between 50% above and under the design specification flow rates, so the variation of the overall heat transfer coefficient is shown as a surface. This kind of variation could be seen in transient calculations, where the flow rate could change with time. The design UA calculated is 201.825 kW, the flow rate through the evaporator is 261,070 kg/s and the flow rate through condenser is 175,270 kg/s.

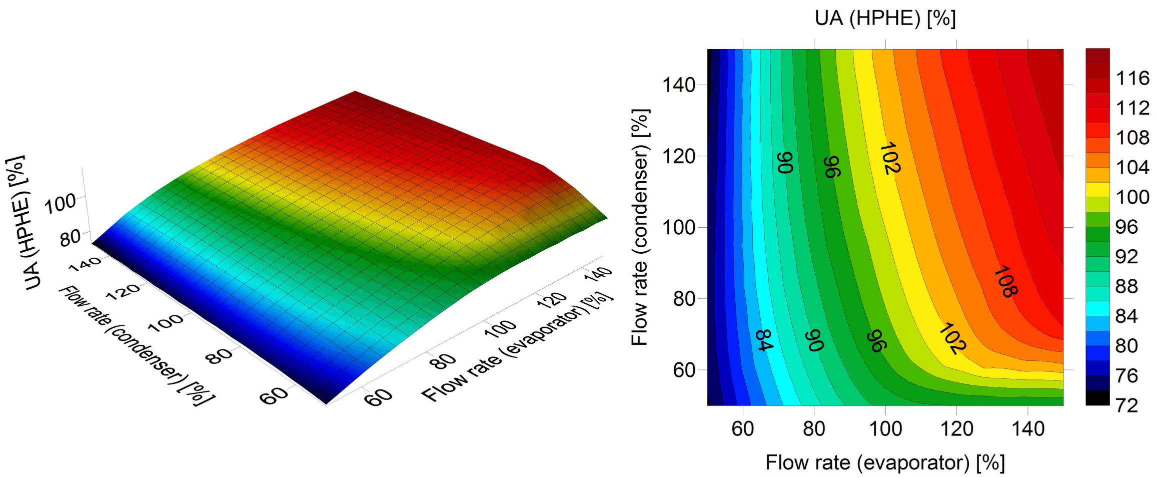

Figure 10 – Sensitivity of HPHE conductance with respect to flow rate perturbations.

5.1. Optimization

A synthesis model was implemented for proposing optimal design of HPHEs. This model was set using a built-in genetic algorithm routine from EES package. The objective was to find the design configuration that attends to the load at minimal cost. In HPHEs, the cost is proportional to the weight of equipment (that reflects the use of steel) and to the number of HPs, that reflects the intensity of manufacturing operations, such as drilling, welding and manpower used. A very rough cost function taken from Borges et al (2007a), was used in this work. The design parameters choose for optimization and its box constraints are shown in Table 4.

Table 4 - Variables and its box constrains.

Variable	Guess	Lower Value	Upper Value	Unit
Fin Height (condenser)	0.02244	5.00E-03	max height	m
Fin Width (condenser)	0.00144	5.00E-04	0.005	m
Fin Step (condenser)	0.01	5.00E-03	0.015	m
Fin Height (evaporator)	0.02244	5.00E-03	max height	m
Fin Width (evaporator)	0.00144	5.00E-04	0.005	m
Fin Step (evaporator)	0.01	5.00E-03	0.015	m
Transversal pitch	0.05	4.00E-02	infinity	m
Number of tubes	2000	1.00E+00	infinity	-

The proposed synthesis model was to minimize equipment cost subject to the following restrictions:

- Total pressure drop at the hot side of equipment less than 120mmWC
- Total pressure drop at the cold side of the equipment less than 120mmWC
- Outlet stream temperature equal to 357°C
- Temperature of water (as a working fluid of heat pipes) under 300°C

The optimization routine had to perform 562 simulation calls, since the population was 16 and the number of generations was 32. The resulting optimal design is shown in Table 5.

Table 5 – Configuration changes in HPHE optimization.

		Base-case		Optimized		Units
		1st Bundle (Water)	2nd Bundle (Naphthalene)	1st Bundle (Water)	2nd Bundle (Naphthalene)	
PARAMETERS	* Transversal pitch	75	75	94.84	94.84	mm
	Diameter	38.1	38.1	38.1	38.1	mm
	Condenser lenght	2.5	2.5	2.5	2.5	m
	Evaporator lenght	3.5	3.5	3.5	3.5	m
	* Number of tubes	2271	4899	851	3402	-
	* Fin Heigth (cold side)	15	15	24.21	20.15	mm
	* Fin Width (cold side)	1	1	1.59	1.10	mm
	* Fin Step (cold side)	10	10	12.34	9.94	mm
	* Fin Heigth (hot side)	15	15	5.22	12.98	mm
	* Fin Width (hot side)	1	1	1.31	1.91	mm
	* Fin Step (hot side)	10	10	1.62	7.20	mm
	HPHE width	9	9	9	9	m
	HPHE height	6	6	6	6	m
RESULTS	Inlet temperature (cold side)	300	518	300	401	K
	Outlet temperature (cold side)	518	630	401	626	K
	Inlet temperature (hot side)	595	664	527	664	K
	Outlet temperature (hot side)	461	595	466	527	K
	HPHE lenght	1.4	2.1	0.8	3.4	m
	Pressure drop in cold side	22.8	67.9	7.52	45.3	mmWC
	Pressure drop in hot side	32.4	81.1	9.71	38.3	mmWC
	HPHE Mass	182.0		141.8		ton
	HPHE Cost	2.54E+06		1.52E+06		US$

In the Table 5 the parameters marked with * are the selected variables for optimization. With the Table 5 it's possible to see the monetary economy as well as the reduction of pressure drop. The reduction of number of tubes is 40.7%; the reduction of 22.1% of the HPHE mass, the cost reduction is 39.9%, the reduction of the pressure drop in cold side is 41.8% and the reduction of pressure drop in hot side is 57.7%.

6. CONCLUSIONS

In this work, a multi-bundle model for heat pipe heat exchangers was implemented in Engineering Equation Solver (EES) using a matrix structure. The implemented program allows changes to the model complexity in a fast way, by only changing the size of the matrix, and consequently the number of sub-bundles that are being considered. The model considers several physical phenomena involved in the heat exchange and uses correlations and equations for external convection, internal convection, pool boiling, film condensing, thermal conduction on heat pipes, and calculates the pressure drop, outlet temperatures of the heat exchanger, thermal resistances and many other parameters necessary to analysis of the heat exchanger. Another feature is the possibility of an optimization of the HPHE, because it was implemented in a solver.

Multi bundle modeling of Heat Pipe Heat Exchangers (HPHEs) showed itself useful for designing HPHEs. By dividing the HPHE into sub-bundles, it is possible to design HPHES with heterogeneous bundles. When each row is modeled as a sub-bundle, it is possible to analyze HPHE behavior row-per-row. With this program is possible to design compact, cheap and very competitive HPHEs, because several working fluids, types of heat pipes and geometrical characteristics can be selected individually for each row according to cost of materials and particular needs of the customer.

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