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Cell-based dynamic heat exchanger models—Direct determination of the cell number and size

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ABSTRACT

Large amounts of thermal energy are transferred between fluids for heating or cooling in industry as well as in the residential and service sectors. Typical examples are crude oil preheating, ethylene plants, pulp and paper plants, breweries, plants with exothermic and endothermic reactions, space heating, and cooling or refrigeration of food and beverages. Heat exchangers frequently operate under varying conditions. Their appropriate use in flexible heat exchanger networks as well as maintenance/reliability related calculations requires adequate models for estimating their dynamic behaviour. Cell-based dynamic models are very often used to represent heat exchangers with varying arrangements. The current paper describes a direct method and a visualisation technique for determining the number of the modelling cells and their size.

1. Introduction

The variability as well as the uncertainty of operating conditions of heat exchangers have been generally modelled in the framework of the concepts of flexibility, controllability, reliability and operability (Oliveira, Liporace, Araújo, & Queiroz, 2001; Skogestad & Postlethwaite, 1996; Sikos & Klemeš, 2010). Recent work in the field of dynamic operation and controller tuning of heat exchanger networks (Dobos & Abonyi, 2010; Dobos, Jäschke, Abonyi, & Skogestad, 2009) has illustrated the importance of using adequate and computationally efficient dynamic heat exchanger models. A very important issue is that heat exchangers are usually used in networks rather than standalone (Klemeš, Friedler, Bulatov, & Varbanov, 2010), which identify the computational efficiency as a key model property.

The appropriate use of heat exchangers under varying conditions requires adequate dynamic models. There are two general approaches to modelling the dynamics of a heat exchanger – distributed and lumped. These two model types have a number of features, which make them suitable for different applications. A comparison of the main features of the two approaches is given in Table 1.

The lumped cell-based models are more popular (Mathisen, Morari, & Skogestad, 1994; Roetzel & Xuan, 1999; Varga, Hangos, & Szigeti, 1995). There have been noticeable advances in the field of dynamic simulation of heat exchangers. Recent examples include: Luo, Guan, Li, and Roetzel (2003) model the dynamic behaviour of multi-stream heat exchangers; Konukman and Akman (2005) heat integrated plant; Ansari and Mortazavi (2006) present a distributed heat exchanger model; and Díaz, Sen, Yang, and McClain (2001), Varshney and Panigrahi (2005), and more recently Peng and Ling (2009) and Vasičkaninová, Bakošová, Mészáros, and Klemeš (2010, 2011) featuring a neural network based model. A prominent example of dynamic heat exchanger modelling from the food industry is presented by Georgiadis and Macchietto (2000) on the case of plate heat exchangers under fouling with milk. These models are quite complex and a little difficult to understand by process engineers. Most importantly, applied to heat exchanger networks, they feature high computational loads. The current paper is a step in direction of alleviating this problem.

Cell models can result in a potentially large number of equations, but the equations are very simple and the approach offers a uniform framework and modelling flexibility to accommodate any type of surface heat exchanger with any flow arrangement. The model complexity can be controlled by the number of cells, allowing a trade-off between the accuracy and the ability of the model to tackle large and complex process systems such as heat exchanger networks. Usually dynamic heat exchanger models (Roetzel & Xuan, 1999) are based on certain assumptions:

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Nomenclature

Latin symbols

A_{CELL}	heat transfer area for a modelling cell wall
$C_{p,C}$	specific heat capacity of the fluid in the cold cell tank
$C_{p,FLUID}$	specific heat capacity of a fluid
$C_{p,H}$	specific heat capacity of the fluid in the hot cell tank
$C_{p,W}$	specific heat capacity of the wall material
h_{HI}	enthalpy of the hot inlet to a cell
h_{HO}	enthalpy of the hot outlet from a cell
h_{CI}	enthalpy of the cold inlet to a cell
h_{CO}	enthalpy of the cold outlet from a cell
$h_{IN,HOT}$	enthalpy of the hot inlet to a heat exchanger
$h_{OUT,HOT}$	enthalpy of the hot outlet from a heat exchanger
$h_{IN,COLD}$	enthalpy of the cold inlet to a heat exchanger
$h_{OUT,COLD}$	enthalpy of the cold outlet from a heat exchanger
m_{COLD}	mass flowrate of the fluid flowing through a cold cell tank and a heat exchanger
m_{FLUID}	mass flowrate of the fluid flowing through a modelling cell tank
m_{HOT}	mass flowrate of the fluid flowing through a hot cell tank and a heat exchanger
$mh_{C,CELL}$	mass holdup of the fluid in a cold modelling cell tank
$mh_{H,CELL}$	mass holdup of the fluid in a hot modelling cell tank
mh_{TANK}	mass holdup of the fluid in a modelling cell tank
mh_W	mass of the wall in a modelling cell
$N_{CELL,MIN}$	thermodynamically possible minimum number of modelling cells
Q_{CELL}	the rate of heat transfer through the cell wall into or out from a modelling cell tank
$Q_{CELL,C}$	heat transfer rate for the cold tank in a modelling cell
$Q_{CELL,H}$	heat transfer rate for the hot tank in a modelling cell
t	time
$T_{0,MIN}$, $T_{N,MIN}$	temperatures of the stream with the smaller heat capacity flow-rate in a heat exchanger, respectively, at the hot and the cold ends
T_{CI}	temperature of the fluid at the inlet of a cold modelling cell tank
T_{CO}	temperature of the fluid at the outlet of a cold modelling cell tank
$T_{FLUID,I}$	temperature of a fluid at the inlet of a modelling cell tank
$T_{FLUID,O}$	temperature of a fluid at the outlet of a modelling cell tank
T_{HI}	temperature of the fluid at the inlet of a hot modelling cell tank
T_{HO}	temperature of the fluid at the outlet of a hot modelling cell tank
T_W	temperature of a modelling cell wall
U_{CELL}	overall heat transfer coefficient for a modelling cell
v_C	volumetric flowrate of the fluid in the cold cell tank
$V_{C,CELL}$	volume of the cold cell tank
v_H	volumetric flowrate of the fluid in the hot cell tank
$V_{H,CELL}$	volume of the hot cell tank
V_W	volume of the wall in a modelling cell
ΔT_{LM}	logarithmic mean temperature difference for a heat exchanger

Greek symbols

$\alpha_{H,CELL}$	film heat transfer coefficient for the hot tank in a modelling cell
$\alpha_{C,CELL}$	film heat transfer coefficient for the cold tank in a modelling cell
ρ_H	density of the fluid in the hot cell tank
ρ_C	density of the fluid in the cold cell tank
ρ_W	density of the wall material

(1) The heat transfer area is uniformly distributed throughout the heat exchanger unit.
(2) All thermal properties (film heat transfer coefficients, specific heat capacities) of the fluids and the exchanger wall are constant. The stream temperatures are considered to vary.
(3) The heat conduction along the axial direction (i.e. direction of the fluid flow) is negligible both within the fluids and within the wall.
(4) The wall thermal resistance to heat transfer is negligible. The effect of this assumption is equivalent to reducing the overall heat transfer coefficient. Therefore the imprecision resulting from this assumption can be compensated by an equivalent increase in the values of the film transfer coefficients.
(5) No heat is lost to the ambient through the exchanger casing.

The distributed models are derived from the general differential equations for heat transfer in a material medium. They are based on the consideration of an infinitely small differential element of the fluid stream or the wall. The resulting model is a set of few partial differential equations (one for the shell pass, two equations per tube pass) with differentiation with respect to time and the considered spatial coordinates (e.g. length). The basic model considers single pass apparatuses (one shell and one tube passes) with co-current and counter-current flows. Technically, it is possible to be extended for multi-pass heat exchangers and different flow configurations – including cross-flow (Roetzel & Xuan, 1999). However, the model becomes too complex and difficult to comprehend and solve.

The cell-based models combine a sufficient number of perfectly mixed model tanks, called cells, which makes the simulation results equivalent to those from a distributed model. Two mass and three energy balances are formulated for the elements of each heat exchange cell. All they take the form of ordinary differential equations with respect to the time.

Both described modelling approaches have their strong sides and associated problems. As a result, they are usually suitable for different applications. The distributed models recognise the continuous nature of the heat transfer both in time and in physical space. They can be solved relatively easy for simpler flow configurations such as single-pass co- and counter-current devices. Thus, they may be the preferred means to investigate the dynamics of heat transfer in general and for detailed studies of single heat exchangers.

However, applying distributed models to more complex heat exchangers and heat exchanger networks usually results in rather high computational burden. This is where cell-based models are generally stronger. Although cell models can result in potentially large numbers of modelling equations, these equations are very simple and offer a uniform modelling framework for any type of surface heat exchanger with any flow arrangement. Several authors (Mathisen et al., 1994; Varga et al., 1995) working in the field of process control and controllability prefer the cell modelling approach because of the modelling and computational simplicity. The main advantage of the model is that its complexity can be controlled by the user by adjusting the number of modelling cells. This allows exploiting the trade-off between the accuracy and the ability of the model to tackle large and complex process systems such as heat exchanger networks.

The computational advantages of cell-based models become clearer after considering the known techniques for solving the distributed models. The latter approach resorts to intensive numer-

Table 1
Qualitative comparison of heat exchanger model types.

Property	Distributed model	Lumped model
Basic modelling element	Differential element	Heat exchange cell
Continuity	Continuous in both space and time	Continuous in time and discrete in space
Differentiation	Differential with regard to both space and time	Differential with regard to time only
Simplifying assumptions	Only two-stream heat exchangers are considered	Perfect mixing is assumed in the fluid compartments of each modelling cell
Solution methods	The approaches used vary from direct numerical integration using finite differences to hybrid methods using analytical solution of the Laplace-transformed models and numerical approximation of the reverse Laplace transformation back into the time domain	Mostly direct numerical integration using finite differences, approximating the time derivatives. Note, that the physical space is already explicitly discretised by introducing the division of the exchanger into cells

ical computations either for finite difference integration, or for the reverse Laplace transformation.

2. Heat exchange cell

2.1. Definition

A simple heat exchange cell is defined as two perfectly stirred tanks, exchanging heat only with each other through a dividing wall. This type of arrangement is illustrated in Fig. 1.

2.2. Assumptions for the modelling cells

The following modelling assumptions are employed to derive the dynamic cell model:

(i) Both tanks in the heat exchanger cell feature perfect mixing. This means that the temperature in each tank can be considered constant with regard to space.
(ii) The fluid densities are constant. This holds completely for liquids. For gases, this would be true as long as the pressures are kept approximately constant.
(iii) The tanks are completely full with the corresponding fluids.
(iv) As the model aims mainly at controlling the fluid temperatures, the streams are assumed to have finite constant specific heat capacities, effectively excluding process streams with pure vaporization or condensation. This hypothesis can be also used to reflect a gradual phase change (as in crude oil preheat), which has been piecewise linearised. The latter case means that the phase change has been represented with one or more process stream segments with finite constant specific heat capacities.
(v) The wall resistance to heat transfer is neglected; its temperature is considered uniform within the cell volume. The main reason for adopting this assumption is the complexity of the dynamics of the heat transfer through the wall. The wall heat capacity (kJ/°C), i.e. the metal heating and heat buffering, could cause significant delay in the heat flow and – influence the temperature distribution in time. For this reason the wall heat capacity is taken into account.

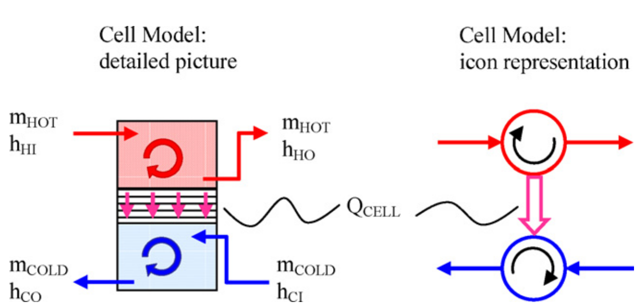

Fig. 1. Representation of a modelling cell.

With regard to the system boundaries, any quantity – material or energy, entering the system is considered positive (+). Also, any quantity – material or energy leaving the system is considered negative (−).

2.3. Equations

Regarding the material balances, assumptions (ii) and (iii) above make them trivial, eliminating any change in the amount of fluid holdup. The energy balance of a tank in a cell is as follows:

$$mh_{\mathrm{TANK}} \cdot c_{\mathrm{p,FLUID}} \cdot \frac{dt_{\mathrm{FLUID,O}}}{dt} = m_{\mathrm{FLUID}} \cdot C_{\mathrm{p,FLUID}} \cdot T_{\mathrm{FLUID,I}} - m_{\mathrm{FLUID}} \cdot C_{\mathrm{p,FLUID}} \cdot T_{\mathrm{FLUID,O}} \pm Q_{\mathrm{CELL}} \tag{1}$$

The term Q_{CELL} represents the rate of heat transfer through the cell wall. It is subtracted for hot side tank and added for cold side tank. The heat transfer rate for the hot and the cold tanks are:

$$Q_{\mathrm{CELL,H}} = \alpha_{\mathrm{H,CELL}} \cdot A_{\mathrm{CELL}} \cdot (T_{\mathrm{HO}} - T_{\mathrm{W}}) \tag{2}$$

$$Q_{\mathrm{CELL,C}} = \alpha_{\mathrm{C,CELL}} \cdot A_{\mathrm{CELL}} \cdot (T_{\mathrm{W}} - T_{\mathrm{CO}}) \tag{3}$$

According to the adopted assumptions the overall heat transfer coefficient U_{CELL} would be calculated using only the film transfer coefficients of the fluids in the two tanks:

$$U_{\mathrm{CELL}} = \left(\frac{1}{\alpha_{\mathrm{H,CELL}}} + \frac{1}{\alpha_{\mathrm{C,CELL}}} \right)^{-1} \tag{4}$$

As a result, the following related energy balances are obtained for the hot tank, the cold tank and the wall:

$$mh_{\mathrm{H,CELL}} \cdot C_{\mathrm{p,H}} \cdot \frac{dT_{\mathrm{HO}}}{dt} = m_{\mathrm{HOT}} \cdot C_{\mathrm{p,H}} \cdot T_{\mathrm{HI}} - m_{\mathrm{HOT}} \cdot C_{\mathrm{p,H}} \cdot T_{\mathrm{HO}} - \alpha_{\mathrm{H,CELL}} \cdot A_{\mathrm{CELL}} \cdot (T_{\mathrm{HO}} - T_{\mathrm{W}}) \tag{5}$$

$$mh_{\mathrm{C,CELL}} \cdot C_{\mathrm{p,C}} \cdot \frac{dT_{\mathrm{CO}}}{dt} = m_{\mathrm{COLD}} \cdot C_{\mathrm{p,C}} \cdot T_{\mathrm{CI}} - m_{\mathrm{COLD}} \cdot C_{\mathrm{p,C}} \cdot T_{\mathrm{CO}} + \alpha_{\mathrm{C,CELL}} \cdot A_{\mathrm{CELL}} \cdot (T_{\mathrm{W}} - T_{\mathrm{CO}}) \tag{6}$$

$$mh_{\mathrm{W}} \cdot C_{\mathrm{p,W}} \cdot \frac{dT_{\mathrm{W}}}{dt} = \alpha_{\mathrm{H,CELL}} \cdot A_{\mathrm{CELL}} \cdot (T_{\mathrm{HO}} - T_{\mathrm{W}}) - \alpha_{\mathrm{C,CELL}} \cdot A_{\mathrm{CELL}} \cdot (T_{\mathrm{W}} - T_{\mathrm{CO}}) \tag{7}$$

Since one cell is certainly not sufficient for describing a whole heat exchanger, usually several cells are combined together, following the flow arrangement of the actual device. Hence, the sizes of the cell tanks, the film heat transfer coefficients and other parameters are calculated by partitioning the whole heat exchanger. Therefore, it is more convenient to refer to the volumes of the cell tanks rather

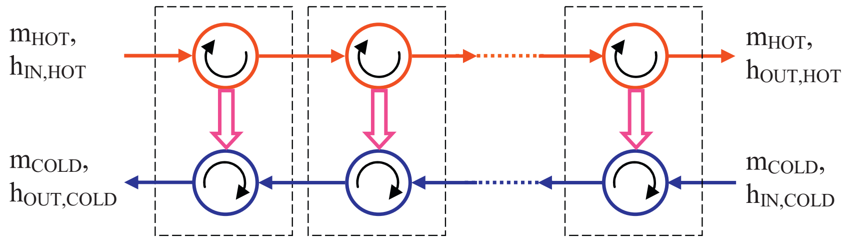

Fig. 2. Cell arrangement for a single-pass shell-and-tube heat exchanger.

than to the mass holdups. The following equations were used by Varga et al. (1995):

$$\frac{dT_{HO}}{dt} = \frac{v_H}{V_{H,CELL}} \cdot (T_{HI} - T_{HO}) - \frac{\alpha_{H,CELL} \cdot A_{CELL}}{V_{H,CELL} \cdot \rho_H \cdot C_{p,H}} \cdot (T_{HO} - T_W) \quad (8)$$

$$\frac{dT_{CO}}{dt} = \frac{v_C}{V_{C,CELL}} \cdot (T_{CI} - T_{CO}) + \frac{\alpha_{C,CELL} \cdot A_{CELL}}{V_{C,CELL} \cdot \rho_C \cdot C_{p,C}} \cdot (T_W - T_{CO}) \quad (9)$$

$$\frac{dT_W}{dt} = \frac{\alpha_{H,CELL} \cdot A_{CELL}}{V_W \cdot \rho_W \cdot C_{p,W}} \cdot (T_{HO} - T_W) - \frac{\alpha_{C,CELL} \cdot A_{CELL}}{V_W \cdot \rho_W \cdot C_{p,W}} \cdot (T_W - T_{CO}) \quad (10)$$

The above equations assume insignificance of the wall capacitance and work with the film heat transfer coefficients directly. While this simplifies the computations, it also has a relatively high probability to distort the dynamics estimates. A comparison made by Mathisen et al. (1994) shows that neglecting the capacitance of the wall leads to estimates of the dynamic response of a heat exchanger that are much less inert than those with accounting for the wall capacitance.

As pointed out by Varga et al. (1995), the cell outlet temperatures are the state variables for the model. They constitute the vector of controlled variables for the cell. This leaves the stream flowrates and the inlet temperatures as variables that influence the outlets. One usual case is (Varga & Hangos, 1993):

(i) The inlet temperatures are disturbances – they are beyond the operator control.
(ii) The stream flow-rates are manipulated variables.

Additionally, there are also other possible degrees of freedom at the level of heat exchanger networks, which can potentially be used as manipulated variables such as the duties on the utility heaters and coolers.

2.4. Cell model of a heat exchanger

The described heat exchange cell model can be used by system and control engineers to construct dynamic models of complete heat exchangers and based on this – models of whole heat exchanger networks. A heat exchanger can be represented by a combination of heat exchange cells, arranged in a way to most accurately reflect the flow patterns in the actual device (Varga et al., 1995). An example for a single-pass heat exchanger is shown in Fig. 2. More complex cell configurations are also possible. The cell numbering is assumed to start at the inlet of a tube-side stream and to follow its path. Usually this is the hot stream.

3. Derivation of the cell parameters

3.1. Driving force effects of cell modelling

The cell model uses the temperature differences between the modelling tanks as estimates of the driving forces. They are smaller than the actual temperature differences (Fig. 3). The effect is stronger for smaller number of cells and weaker for larger number of cells. To compensate for it, the cell heat transfer coefficients must be larger than those for the exchanger as a whole.

3.2. Number of cells needed for adequate modelling of a heat exchanger

Conceptually, the thermodynamically possible minimum number of modelling cells is given by the number of heat transfer units for the exchanger (Mathisen et al., 1994):

$$N_{CELL,MIN} = \frac{T_{0,MIN} - T_{N,MIN}}{\Delta T_{LM}} \quad (11)$$

where $T_{0,MIN}$ and $T_{N,MIN}$ refer to the temperatures of the stream with the smaller heat capacity flow-rate, respectively, at the hot and the cold ends of the exchanger.

As it has been shown, the number of modelling cells is closely linked to the values of the heat transfer coefficients. This ensures that the overall efficiency of the heat exchanger at steady state will be accurately estimated. Consider the heat exchanger driving forces at steady state. As shown above, fewer cells produce lower estimates of the temperature differences, and vice versa (Fig. 3). Ideally, an infinite number of cells should produce the same temperature differences as in the continuous model.

Reducing the number of cells, below a certain minimum number, the cell temperature differences would become negative, making the model thermodynamically incorrect. This limiting case can be used for estimating the lower bound on the number of cells needed. If the temperature differences in all heat exchange cells are assumed exactly zero, a sequence of steps can be projected starting from one of the exchanger ends, as is shown in Fig. 4.

The procedure can start from any exchanger end. For instance, from the exchanger cold end (option 1 in Fig. 4), a vertical line is drawn from the cold stream to the hot stream. From that point,

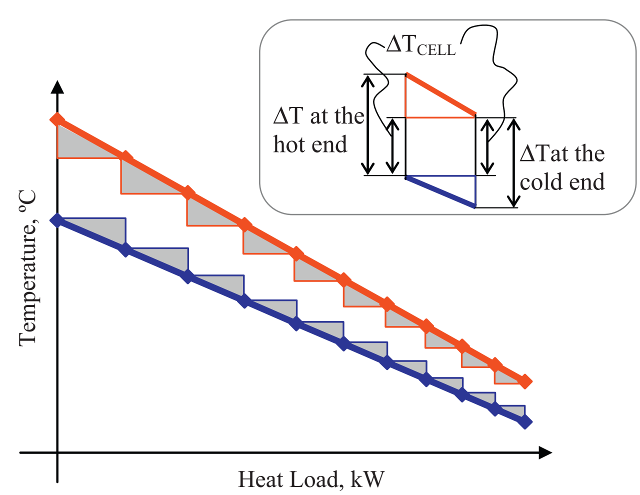

Fig. 3. Driving forces decrease in the cell model.

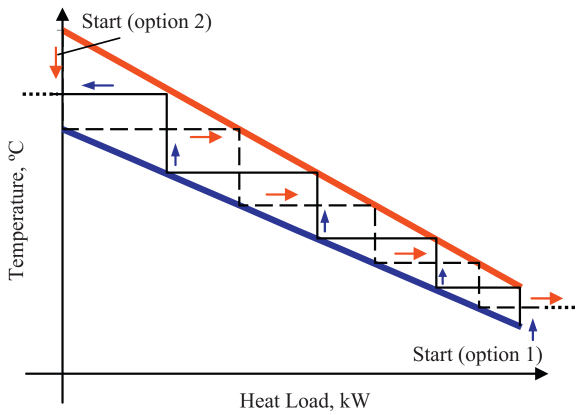

Fig. 4. Lower bound on the number of cells based on temperature differences.

a horizontal line in the left direction is drawn to the cold stream, finishing the construction of the first cell. This geometrical operation is equivalent to assuming zero temperature difference in a heat exchanger cell at the cold end of the device. Minimum temperature difference leads to maximum possible cell size in terms of heat transfer load and in turn – minimum number of cells. Further cells are drawn on the diagram in the same way, until the total heat load reaches or exceeds the one for the heat exchanger at the hot end. For the particular case in Fig. 4 the minimum number of cells is four. The described procedure can be performed in the opposite direction (option 2 in Fig. 4), resulting in the same number of cells. The procedure is analogous to the classical method for determination of the theoretical number of stages of a binary distillation column (McCabe & Thiele, 1925). For obtaining a feasible cell model, the number of cells must be larger than the identified minimum.

It is also necessary to account for the estimated responsiveness of the heat exchanger, which can be expressed through the apparent dead time in reaction of the outlet temperatures to inlet temperature variations. The general trend is in favour of increasing the number of cells. Usually, the best trade-off for a given heat exchanger can be found by varying the number of cells and registering the resulting apparent dead time – Fig. 5. The latter features a typical pattern of asymptotic approach of the dead time estimate to the real one. Therefore, one could start with a number of cells one or two above the lower bound – equal number of cells per heat exchanger pass. The cell number can be gradually increased with one cell per tube pass at each step. The increase should stop when the estimates of the apparent dead time approach the actual ones closely enough.

3.3. Cell-based film heat transfer coefficients

As it has become apparent, the heat transfer coefficients used in the cell model must be larger than those used for calculations

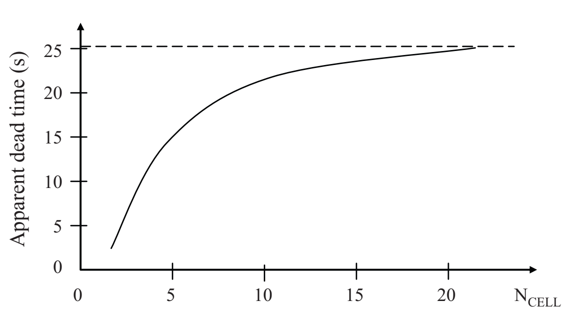

Fig. 5. Trend of the apparent dead time prediction of the hot stream outlet response to inlet temperature changes (after Mathisen et al., 1994).

of the exchanger as a whole. One option for obtaining their values is solving the heat transfer equations for the cells in steady state as part of an optimization formulation. The procedure description follows.

Initialisation

A number of parameters has to be known, including the stream heat capacity flow-rates, their film transfer coefficients, the tube wall parameters: wall thickness and heat conductivity, the number of cells, the inlet and outlet temperatures of one of the streams, say the cold one; and the inlet or outlet temperature of the other stream, say the inlet temperature of the hot stream; the exchanger heat transfer area.

Step 1. The following heat exchanger properties are calculated directly (only once): duty, the outlet temperature of the second stream, the temperature differences at the hot and the cold ends, the average driving force, the average area per cell, the overall heat transfer coefficient of the heat exchanger (this calculation uses the wall conductivity).

Step 2. The steady-state energy balances of the cell tanks plus the equations for heat transfer across the cell wall segments are formulated.

Step 3. The cell number (thus also the temperature differences) and the overall cell heat transfer coefficients are used as optimization variables. The calculation of the heat transfer flows in each cell and of the cell tank temperatures are also added to the problem. These calculations follow the assumptions for the cell model above, neglecting the wall conductivity and compensating for it with larger values of the film heat transfer coefficients according to Eqs. (2)–(4).

Step 4. Finally, an objective function involving the squared error for the total heat exchanger load to be minimized is set. The formulation uses a set of additional inequalities on the cell overall heat transfer coefficient that help in coping with the non-linearity.

4. Conclusions

A method for direct identification of the number of cells in the model, mostly applicable to shell-and-tube heat exchangers has been developed. A useful visualisation of the procedure is provided. Although from previous work there are analytical formulae for calculating the number of modelling cells and the cell heat transfer coefficients, they are limited by certain assumptions – for instance equal heat capacity flow-rates of the hot and the cold streams. On the other hand, the method presented in the current paper allows the computation of the cell number and heat transfer coefficient for a more general case, also providing thermodynamic reasoning, which is invaluable for performing engineering tasks. The method can be readily extended further to the other kinds of heat exchangers. The detailed case studies performed by the authors have shown promising results and lead to the suggestions for the future work as relaxation of the simplifying assumptions. The case studies are going to be published in a dedicated paper.

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