Proceedings of the 18th UK Heat Transfer Conference
9-11 Semptember 2024, Birmingham
UKHTC2024

STEADY-STATE MODELLING OF HEAT EXCHANGERS FOR REFRIGERATION APPLICATIONS

Giovanni Roberti[1], Michael Giovannini[1], Luca Molinaroli[2], Marco Lorenzini[1,*]

[1]Department of Industrial Engineering, University of Bologna, via Fontanelle 40, I-47121 Forlì (FC), Italy
[2]Department of Energy, Politecnico di Milano, via Lambruschini 4, I-20156, Milano (MI), Italy

ABSTRACT

This work deals with the steady state modelling of counterflow brazed plate heat exchangers for refrigeration applications. The main challenges of the stationary approach are the definition and tuning of the iterative algorithm required for the non linear system of equations. The model is experimentally validated, and a variable step size gradient descend algorithm is evaluated to reduce the number of iterations carried out. Some tuning parameters are defined in the framework of an optimization procedure to enhance the accuracy of the model.

1 INTRODUCTION

This work focuses on steady state modelling of counterflow brazed plate heat exchangers (BPHX) employed in vapour compression systems (VCSs) for refrigeration, with the aim providing a useful and simple tool to evaluate the performance of this component, supporting the reduction of the ever-growing environmental impact and energy consumption of the refrigeration sector, [1]. Recent developments in control theory led to the development of real time systems employing numerical models as digital twins; both state-state and dynamical modelling have proved their usefulness in this context. Yet, a steady state model can be suitable for rating the performance of a plant, or for automatic fault detection; they even can be use as a benchmark for the assessment of a dynamical model and for its initialization. Many researches have dealt with this topic, [2], some issues need to be addressed to develop a suitable model for the real-life application. The multi-region approach is often preferred because it allows the definition of average values for the physical properties and for the heat transfer coefficients (HTCs) in each region preserving good accuracy, even though the refrigerant is undergoing a change of phase. Usually the energy balance and the suitable ε-NTU relation are applied in each region. Given the non linearity of the latter, an iterative procedure is needed to solve the system of equations. Software like Matlab offers suitable tools to automatic solve non linear system of equations which ensues, yet this often results in non physical solutions if no accurate supervision is exerted. Moreover, the available models are often design-oriented, so that the control parameters are fixed in the initial stage of the algorithm, [2]. In this work, the steady state models of BPHX is set up and validated comparing the numerical results with the experimental data obtained from a heat pump employing R450A as refrigerant. The system consists of two counterflow BPHXs, a semi-hermetic reciprocating compressor and an electronic expansion valve. Particular attention is devoted to solution algorithm, following the approach proposed by [3], different solver options, like the starting point or the scaling factor of the step size, have been tested to evaluate the execution speed and the accuracy of the models.

2 MODEL OF THE HEAT EXCHANGERS

The model of the BPHX is based on some hypothesis: the cross-sectional area is assumed constant along the heat exchanger for both flows; the pressure losses are considered negligible; the thermal conductive resistance of the plate and thermal conduction in the fluid are neglected; the heat transfer area is the same for both fluid flows. During stationary operations in rated conditions, the refrigerant enters the condenser as superheated vapour and exits as subcooled liquid; so, a three-zone representation is obtained. Each of them are characterized by a different refrigerant conditions: superheated vapour,

*Corresponding author: marco.lorenzini@unibo.it

two-phase mixture and subcooled liquid. At the other end, in stationary conditions, the refrigerant enters the evaporator as two-phase mixture, and exits with a certain degree of superheating; the model of the evaporator consist therefore of a two-region representation. The procedure proposed employs the energy balance equations between the refrigerant and the secondary fluid and the $\varepsilon - NTU$ equation for each zone (eq. (1)-eq. (2)).

$$m_r\,(h_{r,in} - h_{r,out}) = m_f\, c_{p,f}\,(T_{f,in} - T_{f,out}) \tag{1}$$
$$\varepsilon\, C_{min}(T_{r,in} - T_{f,in}) = m_r\,(h_{r,in} - h_{r,out}) \tag{2}$$

The condenser is be treated as a set of three different heat exchangers connected in series. For each zone, the effectiveness ε_i $(i = sh,\ tp,\ sc)$ is a function of the number of transfer units NTU_i and the heat capacity ratio γ_i. Under the assumption above, the product of the overall heat transfer coefficient times the heat transfer surface area of the i-th zone $(UA)_i$ can be calculated in a simplified way, (eq. (3)).

$$\varepsilon_i = f\ (NTU_i,\ \gamma_i) \qquad NTU_i = \frac{(UA)_i}{C_{min,i}} \qquad \gamma_i = \frac{C_{min,i}}{C_{max,i}} \qquad (UA)_i = A_i \left(\frac{1}{\alpha_{r,i}} + \frac{1}{\alpha_{f,i}}\right)^{-1} \tag{3}$$

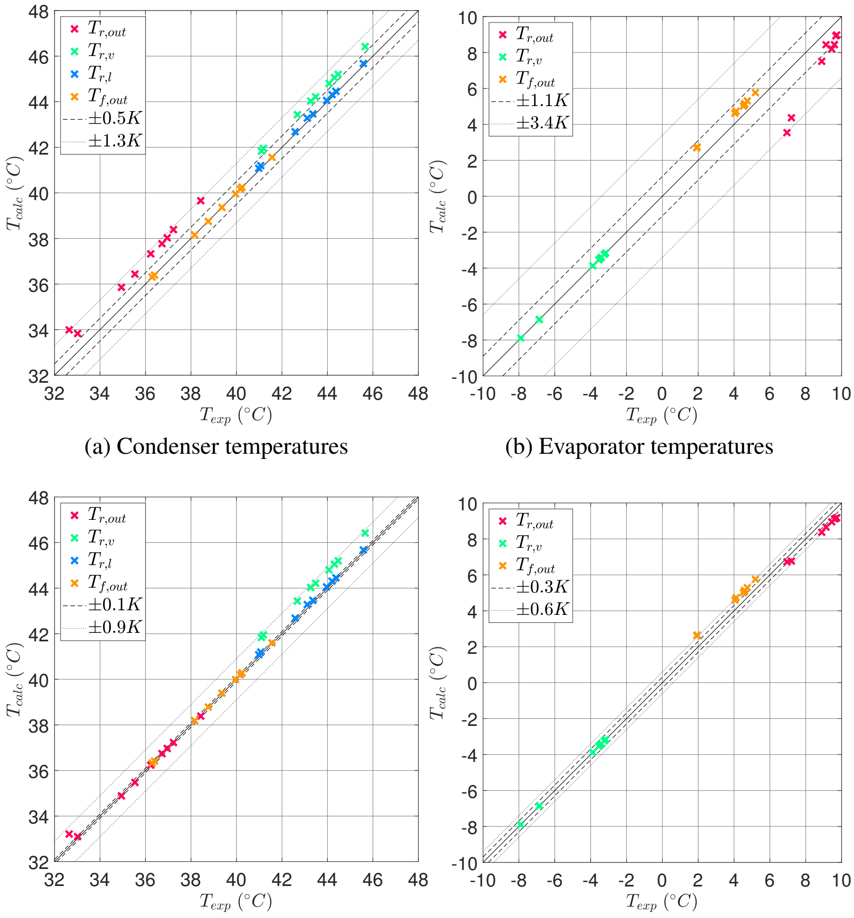
The sum of the heat transfer area for each region must be equal to the total area of the heat exchanger, that can be conveniently expressed in normalized form, $\zeta_{sh} + \zeta_{tp} + \zeta_{sc} = 1$ and $\zeta_{sh} + \zeta_{tp} = 1$.
The pressure glide related to the zeotropic mixture $R450A$ deserves one more consideration. Since this is very small, it can be considered negligible compared with the condensation pressure; however, it becomes significant in the pressure range of the evaporator. So, while the heat transfer in the condenser is considered isobaric, in the evaporator a lumped pressure drop occurring in the two-phase region accounts for the pressure glide.

3 SOLUTION ALGORITHM

As proposed in [3], an iterative algorithm is employed for the solution of the non-linear system of equations for each heat exchanger. Starting from a guess value of the outlet refrigerant temperature corresponding to the maximum heat transferred rate ($T_{r,out} = T_{f,in}$), the governing equation are solved for each zone, starting from the last one crossed by the working fluid. The extension of each zone is calculated iteratively. The whole procedure for the condenser (evaporator) is repeated reducing (increasing) progressively the $T_{r,out}$ until the total area of the exchanger is matched within a certain tolerance. This algorithm is implemented both with fixed step correction on the $\Delta T_{t,out}$ and with a variable steps correction, which is evaluated on the amplitude of the deviation.

4 RESULTS AND DISCUSSION

Validation of the steady state models of the condenser and of the evaporator demonstrates a good accuracy in the prediction of the temperature of the refrigerant and of the secondary fluid; in general, temperatures are predicted with a maximum deviation of $1.3\,K$. In the condenser (fig. 1a), $T_{r,v}$ is overrated because the model is based on the hypothesis that the pressure drop can be neglected: the pressure drop in the real condenser decreases the saturation temperature and is responsible of the overestimation of $T_{r,v}$. $T_{r,out}$ is always overrated: this can be due to the small value of the HTC given by the correlation by Martin, which is generally more conservative than others for single phase flow in plate heat exchangers [4]. Also, for the evaporator (fig. 1b), the overestimation of $T_{f,out}$ and the underestimation of $T_{r,out}$ may be related to the Martin correlation for single phase HTC. It is remarkable that the model accounting for the pressure glide is less accurate than the isobaric approach [4]. An optimization procedure is carried out employing the *fmincon* tools in Matlab; as reported in fig. 1c and fig. 1d, use of tuning parameters calculated with the optimization procedure brings about a significant enhancement of the accuracy of the predicted temperature. Finally, the evaluation of a variable-step gradient descent algorithm allows a six-fold reduction in the number of iterations needed to match the requested tolerance on $T_{r,out}$, for both the condenser and the evaporator models.



(a) Condenser temperatures

(b) Evaporator temperatures

(c) Condenser temperatures with optimization

(d) Evaporators temperatures with optimization

Figure 1: Validation and optimization of the condenser and evaporator models.

5 CONCLUSIONS

This work shows that a steady state model of a counterflow BPHX can be accurate and fast enough to be used for reliable design and tuning of new VCS; furthermore, it may be employed also in an advanced control system or in automatic fault detection applications, comparing the actual stationary behaviour with the corresponding values provided by the numerical model.

REFERENCES

[1] J.-L. Dupont, P. Domanski, P. Lebrun, & F. Ziegler. 38th Note on Refrigeration Technologies: The Role of Refrigeration in the Global Economy (2019).

[2] D. L. Blanco, K. Nagano, & M. Morimoto. Steady state vapor compression refrigeration cycle simulation for a monovalent inverter-driven water-to-water heat pump with a desuperheater for low energy houses. *International Journal of Refrigeration*, **35** (2012) 1833–1847.

[3] G. Zsembinszki, A. de Gracia, P. Moreno, R. Rovira, M. Ángel González, & L. F. Cabeza. A novel numerical methodology for modelling simple vapour compression refrigeration system. *Applied Thermal Engineering*, **115** (2017) 188–200.

[4] G. Roberti. *Steady-state Modelling of a Vapor Compression Refrigeration Cycle*. Master degrre thesis, University of Boogna (2020).