Transient Behaviour of Economizers for Multi-Stage Refrigeration Plant

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TRANSIENT BEHAVIOUR OF ECONOMIZERS
FOR MULTI-STAGE REFRIGERATION PLANT

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Abstract

When a refrigeration plant operates with a large difference between condensing and evaporating pressures, multistage operation is required to obtain an acceptable performance. In two-stage systems, vessels that combine flash-gas removal and intercooling are commonly used. The disadvantage of these vessels is the large amount of refrigerant required for proper operation. An alternative for these vessels is the use of "economizers", dry evaporator heat exchangers. The advantages of this solution are: smaller volume requirements, lower costs and much smaller refrigerant contents.

This paper describes a dynamic simulation model for economizers of multi-stage refrigerating plants. A finite difference method has been used to solve the mass and combined energy and momentum equations. In the two-phase flow region, the local void fraction is used to combine the separate equations for the two-phases. Both the void fraction models of Levy and Hughmark have been implemented. The two-phase flow heat transfer relationship of Gungor and Winterton has been used to predict the heat transfer rate in the two-phase flow region. The two-phase flow pressure drop relationship of Lockhart and Martinelli has been used to predict two-phase flow pressure drop.

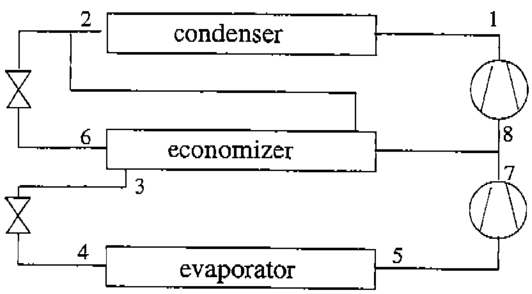
The model has been implemented in Fortran. The main objective of the model was to predict the effect of design criteria on the (dynamic) performance and refrigerant contents of compact economizers. The model results have been compared with experimental data obtained with a shell-and-tube economizer. The model predicts the performance of the economizer within 10%. Hughmark's void fraction model leads to refrigerant contents which are twice as large as obtained with Levy's model.

INTRODUCTION

The current phase-out of CFC- and HCFC-refrigerants due to their ozone depleting potential (ODP) and the global warming potential (GWP) of the HFC-refrigerants has led to a growing scepticism towards the use of large quantities of chemicals unfamiliar to the biosphere as refrigerants. A conversion to ecologically safe refrigeration systems is required. The new systems must satisfy certain criteria:

- energy efficiency equal to or better than existing systems
- acceptable system cost
- safe system designs with respect to the local environment

The first point requires the use of system designs showing high performance results. The last point requires system designs which lead to as low as possible refrigerant mass contents.



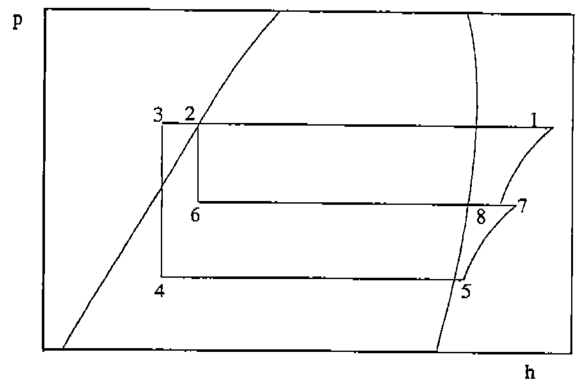


Fig. 1. - Two-stage refrigeration system including an economizer.

Fig. 1 shows a two-stage compression refrigeration arrangement illustrating the application of economizers as discussed in this paper. In these systems, a part of the condensate flow is used to subcool the main condensate flow so that a larger enthalpy change is attained in the evaporator. The refrigerant evaporated in the economizer enters the compressor as a side stream. This is a system configuration frequently encountered in twin-screw compressor refrigeration plants, where the side stream enters the compressor through the "economizer port". Fleming et al. [1995] compared the performance of a single stage system with the performance of a two-stage system with an open flash tank serving the economizer port and concluded that such

arrangement increases the refrigerating capacity considerably but the COP is only slightly improved. With the economizer arrangement shown in fig. 1 the system COP can be increased up to 10% in comparison with the single stage arrangement, depending on the operating conditions. This system is a low cost two-stage arrangement which fulfills to the first two criteria stated above. The economizers conventionally used in these systems are shell-and-tube evaporators with relatively large amounts of liquid refrigerant in the shell side. When compact economizers could be implemented, the system would comply to the three criteria stated above.
The COP of the system increases when the temperature difference between the subcooled liquid (θ_3) and the interstage evaporation temperature (θ_6) decreases. To obtain acceptable values of the COP, this temperature difference should be 5 K or lower. This implies that the dry expansion evaporation control strategy directly influences the system performance. The study of the dynamic behaviour of the economizer will allow the development of the control strategy.
During transient and part load conditions the refrigerant mass contents may drastically increase.

MODEL FORMULATION

From the literature (Wang and Touber [1991], MacArthur and Grald [1989]) it appears that a distributed model of the economizer will be required to be able to obtain the results we are interested in for different economizer arrangements.

Basic Equations

The economizer consists of three different regions: the evaporating and superheating refrigerant, the heat exchanger wall and the subcooled refrigerant liquid. Fig. 2 illustrates these regions for a control volume in the evaporating refrigerant zone. The flow in the different regions is assumed to be one-dimensional. In the evaporating region the conservation equations are applied to both phases. When the assumptions as made by Wang and Touber [1991] for the evaporating refrigerant side are considered, a set of equations is obtained for both phases similar to the set of equations obtained by these researchers. The following continuity equation is, for instance, derived for the vapor phase:

$$\frac{\partial}{\partial t}(\langle\alpha\rangle\rho_v) + \frac{\partial}{\partial x}(\langle\alpha\rangle\rho_v v_v) = \dot{M}_{lv} \qquad (1)$$

where $\langle\alpha\rangle$ is the void fraction, with $\langle\alpha\rangle = A_v/A$ and $A = A_v+A_l$ and v_v is the vapor velocity.
The energy equation for the two phase flow is:

$$\frac{\partial}{\partial t}[\langle\alpha\rangle\rho_v h_v+(1-\langle\alpha\rangle)\rho_l h_l]+\frac{\partial}{\partial x}[\langle\alpha\rangle\rho_v v_v h_v+(1-\langle\alpha\rangle)\rho_l v_l h_l]=(\pi d/A)\phi \qquad (2)$$

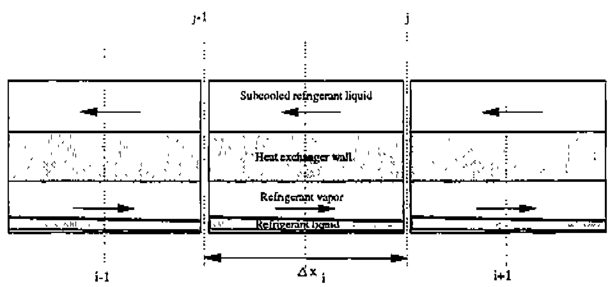


Fig. 2. - Control volume of economizer.

with ϕ the heat flux. The continuity equation for the liquid phase and the momentum equation for the vapor and liquid phases are similar to the corresponding equations presented by Wang and Touber [1991].
The energy equation for the heat exchanger wall is:

$$(\rho V c_p)_{hx}\frac{\partial\theta_{hx}}{\partial t} - \alpha_r A_r(\theta_r-\theta_{hx}) + \alpha_c A_c(\theta_{hx}-\theta_c) = 0 \qquad (3)$$

The energy equation for the subcooled refrigerant liquid and for the superheated vapor on the evaporating refrigerant side is as follows (P_n is the perimeter of the n^{th} wall):

$$(\rho A c_p)_{hx}\frac{\partial\theta_c}{\partial t} + (\dot{m}c_p)_c\frac{\partial\theta}{\partial x} + \sum_{n=1}^{m}\alpha_c P_n(\theta_c-\theta_{hx(n)}) = 0 \qquad (4)$$

Discretization of the Equations

The discretized equations can be determined by integration of the basic equations shown above. To discretize the different equations, the economizer has been divided into a series of control volumes as shown in fig. 2 and the different equations have been integrated over the appropriate control volumes with respect to time and distance. Liebmann's (Hoogendoorn [1978]) implicit finite difference method has been used to discretize the equations. This implies backward differences in time and central differences in place. The continuity equation for the evaporating refrigerant side becomes

$$\{[\langle\alpha\rangle\rho_v+(1-\langle\alpha\rangle\rho_l]'_i - [\langle\alpha\rangle\rho_v+(1-\langle\alpha\rangle)\rho_l]^o_i\} \frac{\Delta x}{\Delta t} + [\langle\alpha\rangle\rho_v v_v+(1-\langle\alpha\rangle\rho_l v_l]'_j - [\langle\alpha\rangle\rho_v v_v+(1-\langle\alpha\rangle)\rho_l v_l]'_{j-1} = 0 \tag{5}$$

with ' the present time step and ° the previous time step. In a similar way, the energy equation becomes

$$\{[\langle\alpha\rangle\rho_v h_v+(1-\langle\alpha\rangle\rho_l h_l]'_i - [\langle\alpha\rangle\rho_v h_v+(1-\langle\alpha\rangle)\rho_l h_l]^o_i\}\frac{1}{\Delta t} + [\langle\alpha\rangle\rho_v v_v h_v+(1-\langle\alpha\rangle\rho_l v_l h_l]'_j - [\langle\alpha\rangle\rho_v v_v h_v+(1-\langle\alpha\rangle)\rho_l v_l h_l]'_{j-1}\frac{1}{\Delta x}$$

$$- \alpha_r \sum_{n=1}^{m} d_n \left(\theta'_{hx(n)i}-\theta'_{ri}\right) = 0 \tag{6}$$

with d_n the perimeter/surface area ratio of the n^{th} wall. Assuming, for positive refrigerant velocities, that

$$\langle\alpha\rangle'_{j-1} = \langle\alpha\rangle'_{i-1} \quad \text{and} \quad \langle\alpha\rangle'_j = \langle\alpha\rangle'_i \tag{7}$$

than the continuity and energy equations become

$$v'_{lj} = \frac{-\langle\alpha\rangle'_i v'_{vj}\rho_v + \langle\alpha\rangle'_{i-1} v'_{vj-1}\rho_v + (1-\langle\alpha\rangle)'_{i-1} v'_{lj-1}\rho_l}{\rho_l\,(1-\langle\alpha\rangle)'_i} - \frac{\{[\langle\alpha\rangle'_i\rho_v + (1-\langle\alpha\rangle)'_i\rho_l] - [\langle\alpha\rangle^o_i\rho_v+(1-\langle\alpha\rangle)^o_i\rho_l\}(\Delta x/\Delta t)}{\rho_l\,(1-\langle\alpha\rangle)'_i} \tag{8}$$

$$(h_v-h_l)(1-\langle\alpha\rangle'_i = \frac{v'_{lj-1}(1-\langle\alpha\rangle)'_{i-1}\dfrac{\rho_l(h_v-h_l)}{\Delta x} + \alpha_r \sum_{n=1}^{m} d_h \left(\theta'_n-\theta'_{hx(n)i}\right) + (1-\langle\alpha\rangle)^o_i \dfrac{\rho_l}{\Delta t}(h_v-h_l)}{\dfrac{\rho_l v'_{lj}}{\Delta x} + \dfrac{\rho_l}{\Delta t}} \tag{9}$$

In these two equations the void fraction, the vapor velocity and the liquid velocity for the present time step ' and i-th control volume are unknown. A third equation giving the relationship between void fraction and vapor mass fraction (quality, x) couples these two equations:

$$\frac{v_v}{v_l} = \frac{x(1-\langle\alpha\rangle)\rho_l}{(1-x)\langle\alpha\rangle\rho_v} \tag{10}$$

Equations (3) and (4) can be integrated in a similar way to give, for the heat exchanger wall:

$$\theta'_{hxi} = \frac{(\alpha_r A_r)_i\,\theta'_{ri} + (\alpha_c A_c)_i\,\theta'_{ci} + \dfrac{(mc_p)_{hxi}}{\Delta t}\theta^o_{hxi}}{(\alpha_r A_r)_i + (\alpha_c A_c)_i + \dfrac{(mc_p)_{hxi}}{\Delta t}} \tag{11}$$

Subscripts r and c refer respectively to evaporating and subcooled refrigerant side.

and for the subcooled refrigerant liquid:

$$\theta'_{ci} = \frac{(\dot{m}c_p)_c\, \theta'_{ci-1} + \sum_{n=1}^{m}(\alpha_c A_n)_i\, \theta'_{hx(n)i} + \dfrac{(mc_p)_{ci}}{\Delta t}\, \theta^{o}_{ci}}{(\dot{m}c_p)_c + \sum_{n=1}^{m}(\alpha_c A_n)_i + \dfrac{(mc_p)_{ci}}{\Delta t}} \qquad (12)$$

Void Fraction and Two Phase Heat Transfer Models

Rice [1987] evaluated several void fraction correlations for their effect on refrigerant charge inventory predictions. He concluded that the void fraction model has a major effect on refrigerant inventory predictions for evaporators. Hughmark's [1962] correlation gives the highest predictions for evaporators. The closest agreement to the measured total system charge is obtained when correlations giving the highest predictions are used. Nevertheless, the methods based on analytical solutions (Levy [1960] and Zivi [1964]) are frequently implemented in practice due to their simplicity. Hughmark's method requires an iterative calculation and leads to larger simulation times. Fig. 3 shows a comparison of several void fraction models. Also the homogeneous model line (assuming no slip between the two phases) is shown. The figure applies for R22 and an evaporating temperature of -1°C. Both Hughmark's and Levy's methods have been implemented in the model so that the effect of the void fraction model on economizer performance can be quantified.

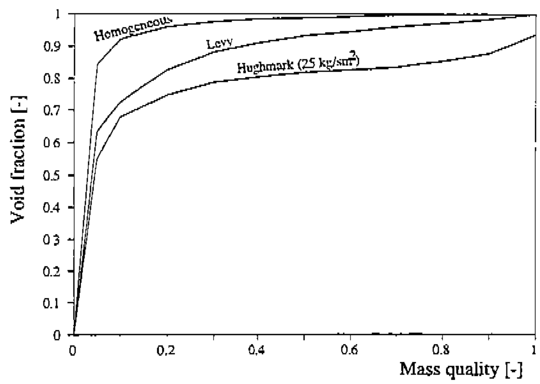


Fig. 3. - Comparison of local R22 void fraction predictions (evaporating temperature = -1°C).

Fig. 4 shows a comparison between the correlations most widely used to predict the heat transfer coefficient in flow boiling of refrigerants inside horizontal tubes. The results for the correlations of Chen [1966], Dhar and Jain [1983], Gungor and Winterton [1987], Jung and Radermacher [1991], Kandlikar [1990] and Shah [1982] are compared in this figure. Specially for small diameters, the results obtained with the different correlations may be quite different. Kandlikar's results are too optimistic. Gungor and Winterton's results are conservative. The two-phase heat transfer relationships of Jung and Radermacher and Shah show a strong dependence of the refrigerant quality; the other correlations show a much smaller effect of quality. Gungor and Winterton's relationship was selected to be implemented in the programme because the experimental data used to determine their relationship include mass fluxes as low as 12.5 kg/sm^2. Some of the relationships are based only on mass fluxes larger than 250 kg/sm^2. These correlations may lead to an over prediction of the heat transfer coefficient, specially in part-load operation.

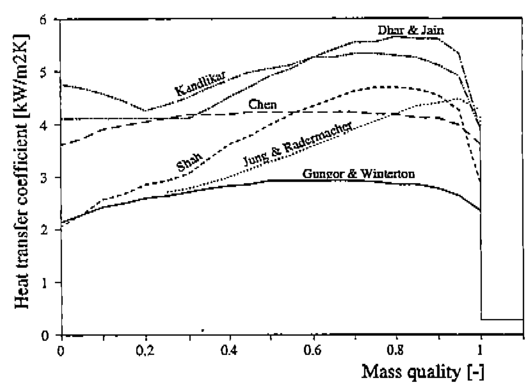


Fig. 4. - Comparison of local R22 heat transfer predictions(evaporating temperature = -1°C; 200 kg/sm²; 15 kW/m²; D=0.01 m).

Solution Methodology

Although the set of equations to be solved is the same for the different types of flow (cocurrent-, countercurrent- or crossflow), the solution methodology depends on the type of flow encountered in the heat exchanger under consideration. Shell-and-tube heat exchangers, for example, can be subdivided in cross flow elements. For cross flow elements an iteration procedure is required to evaluate the local void fraction and refrigerant liquid velocity in the evaporating refrigerant side. The solution methodology shall be illustrated for this type of heat exchanger. Fig. 5 shows schematically the subdivision of the economizer into elements. In this particular case the economizer is a 4-pass shell-and-tube evaporator with 5 baffles leading to the subdivision of the economizer into a matrix of 6 (number of baffles + 1) x 4 (number of passes) elements. Each of these elements is subdivided in several control volumes for which the set of equations apply. The subcooled refrigerant liquid flows from bottom to top (or vice-versa) and the evaporating refrigerant from left to right (or vice-versa). For the different control volumes in heat exchanger element (1,1) the subcooled liquid refrigerant has the same inlet temperature while in the evaporating refrigerant side the mass quality increases from left to right. The heat exchanger is calculated as follows. For the control volumes in element (1,1) both inlet conditions are known so that the outlet conditions can be predicted. The calculation follows the evaporating refrigerant flow. The next element to be calculated is element (2,1). An initial value is required for the subcooled refrigerant liquid at the outlet of this element, which is also the inlet temperature of element (3,1). With this estimated value the inlet of element (2,1) and the outlet of element (3,1) can be predicted. In a similar way estimates have to be made for elements (4,1), (5,1) and (6,1). In the next passes both inlets are known so that the set of equations can be solved. In the last pass the temperature estimates can be checked: for example, the subcooled refrigerant outlet from element (2,4) must be equal to the inlet of element (1,4). If this is not the case, the heat exchanger must be recalculated with new values for the temperature estimates. These values are obtained by dividing the difference between, for example, the outlet temperature of element (1,4) and the inlet temperature of element (2,4) by 2 and adding the result to the previous estimate. The same procedure is applied to correct the previous estimates for all the baffles of the heat exchanger.

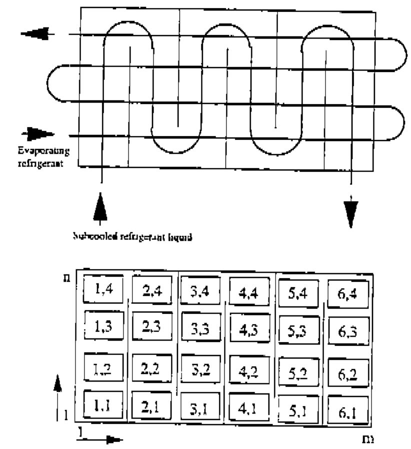


Fig. 5. - Schematic of shell-and-tube heat exchanger.

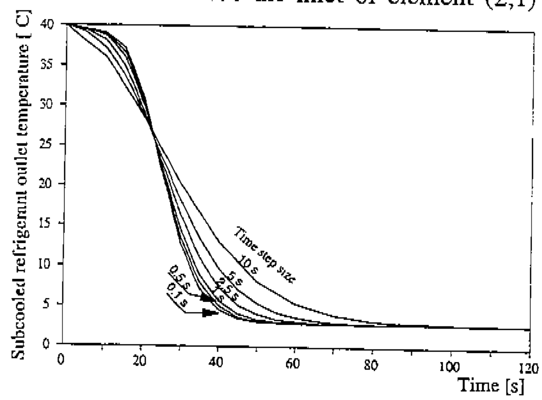


Fig. 6. - Effect of time step size on convergence.

Fig. 6 illustrates the effect of time step size on convergence after the start-up of the economizer. At time is zero the liquid injection into the intermediate pressure side of the economizer is initiated. The figure illustrates that after the start-up a time step size of 0.5 s is required. In this figure the axial position step size was 0.05 m. This is 0.5% of the heat exchanger length. When this step size is taken too large than the prediction of the transition point between two-phase flow and superheated flow is not accurate enough, resulting in a poor prediction of the total heat transfer.

RESULTS

Refrigerant Contents

The effect of part load on the refrigerant contents of the evaporating refrigerant side of the economizer has been investigated. Fig. 7 shows some results. From this figure it appears that the amount of liquid in the economizer increases as the temperature driving force reduces. In part load operation the economizer can accumulate a large amount of liquid. Also the time constant of the economizer is strongly dependent on the temperature driving force.

Shell-and-Tube Economizer

Experimental data obtained with a shell-and-tube economizer have been compared with the model predictions. The shell-and-tube economizer had a 12" shell with a heat exchanging length of 3.0 m. The evaporating refrigerant side had 6 passes and there were 27 baffles on the subcooled liquid refrigerant side. The economizer was a component of a refrigeration plant. During the experiments, the economizer was operating under part load conditions. Fig. 8 shows a comparison between experiments and model results. The figure illustrates the capability of the model in predicting the subcooled and evaporating refrigerant outlet temperatures as a function of time. For the conditions of these experiments, the heat flow from subcooled to evaporating refrigerant was predicted within 10%.

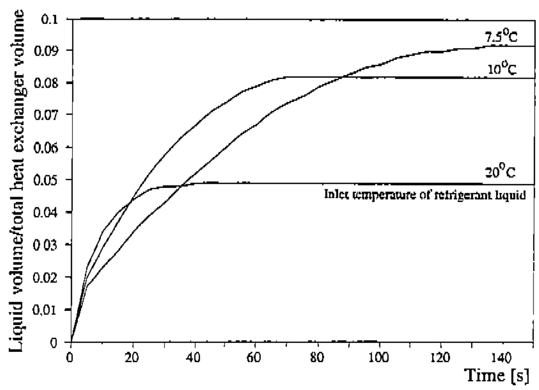


Fig. 7. - Effect of inlet temperature driving force on the quantity of liquid refrigerant in the economizer (R22; evaporating temperature = -5°C).

CONCLUSIONS

A model has been developed describing the behaviour of economizers in multi-stage refrigeration systems. The model allows for the prediction of the transient behaviour of economizers. With this model control strategies can be developed for these heat exchangers. Also the (time dependent) refrigerant contents can be predicted.

Since the time constants of economizers appear to be relatively small, the study of the effect of design criteria doesn't necessarily requires transient model studies. Large time step sizes will allow for the prediction of steady state conditions while the computation times will remain within acceptable values.

A comparison with experimental data shows that the model predicts the economizer performance within 10%.

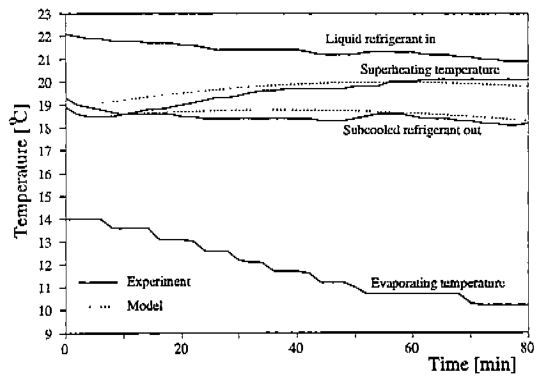


Fig. 8. - Comparison between model and experimental data for a shell-and-tube economizer.

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