Distributed and Non-Steady State Modelling of an Air Cooler

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Abstract

This paper presents the equation derivation as well as the simulation results of a non-steady state distributed model for a commonly used type of evaporators — air coolers. In contrast to others, the model puts the stress on the refrigerant mass transportation occurring in the two-phase flow region inside an air cooler. The so-called slip-effect between the vapour and liquid is tackled by using an advanced computer package PHOENICS (see [27]). Because the model is purely distributed, it is applicable to various kinds of tube circuit arrangements of air coolers. The purpose of the model is studying and optimization of non-steady state behaviour of refrigerating systems with capacity control.

Nomenclature

Symbols | Physical Meanings | Units
--- | --- | ---
A | Cross-section area | [m²]
B | Constant | |
C | Constant | [kJ/(kg · K)]
c | Specific heat | |
D | Outer diameter of pipes | [m]
d | Inner diameter of pipes | [m]
F | Friction force | [N/m²]
f | Interfacial momentum-transport coefficient | [kg/(m² · s)]
h | Specific enthalpy | [kJ/kg]
L | Length | [m]
M | Mass | [kg]
M | Mass flow rate | [kg/s]
m | Interfacial mass-transport rate | [kg/(m² · s)]
P | Pressure | [N/m²]
Q | Heat transfer | [kw]
q | Heat flux | [kw/m²]
S | Surface area | [m²]
T | Absolute temperature | [K]
| Time | [s]
u | Velocity | [m/s]
V | Volume | [m³]
w | Water-vapour concentration in air | [kg/kg]
x | Quality | [kg/(m³ · s)]
z | Coordinate | [m]

Greek Symbols

α | Heat transfer coefficient | [kw/(K · m²)]
θ | Void fraction | |
γ | Latent heat of water | [kJ/kg]
η | Fin efficiency | |
ς | Temperature | [° C]
ξ | Friction factor | |
ρ | Density | [kg/m³]
1. Introduction

Since almost every refrigerating system operates under variable conditions, a pure steady-state in fact does not exist. Specially if capacity controllers are equipped on the system, investigations of the system should not be limited only to steady-state analysis, but also attention has to be paid to non-steady state or dynamic analysis. Dynamic modelling is a mathematical description of the system in time domain. Mass and energy are supposed to accumulate or dissipate in the system due to differences between the input and output of mass and energy to the system.

This paper will introduce a distributed dynamic model for a commonly used type of evaporators — air coolers. Often this type of evaporators are accompanied with thermostatic expansion valves; the refrigerants in them need to be usually evaporated before they leave. Such air coolers are also called dry-expansion evaporators. Compared to the other components of refrigerating systems or other types of evaporators, dry-expansion evaporators are relatively more complicated to be modelled, due to the complexity of the two-phase flows involved in the evaporators. For example, in case of a shell-and-tube evaporator, lumped models are fully successful in describing the mechanism. However, lumped models would be inadequate for dry-expansion evaporators. In the past years, many models have been developed for compressors, expansion devices, condensers, as well as evaporators. But air coolers are still an obstacle for the model-makers. Therefore, it is still necessary to investigate the problem.

A complete dynamic model of dry-expansion evaporators should include two parts: on-period model and off-period model. The first is to describe the dynamic behaviour of evaporators during on-periods and the second is to depict what occurs when the refrigeration system is switched off. The reason to consider these two aspects is because most of the used refrigeration systems today are still equipped with on-off control elements. In fact, dynamics of an evaporator is strongly influenced by turn-on and turn-off of the compressor. Therefore, ignoring of the off-period modelling is irrational.

Dynamic behaviour of an evaporator results from both energy and mass accumulations. The heat capacity of the pipes and refrigerant acts as a reservoir of heat energy. The two-phase flow taking place inside the evaporator pipes is the reason of mass accumulation. Thus the success of a dynamic modelling for dry-expansion evaporators to large extent depends upon the degree of understanding of the two-phase flow mechanism.

Usually, the heat exchange process between the refrigerant and coolant in a dry-expansion evaporator takes place in the form of cross-flow. However, in order to improve the heat transfer efficiency or meet the special requirements of...
customers, manufacturers design air coolers with various pipe circuiting methods. This diversity makes the modelling of air coolers even more difficult. Accordingly, distributed models are increasingly required to maintain the flexibility and applicability.

In the last few years a lot of publications are related to dynamic modelling of dry-expansion evaporators. They can be classified into four categories: black box models, 1-zone models, 2-zone models and distributed models. For instance, [11], [2], [3] are connected with black box models. [4], [5], [6] are 1-zone models. 2-zone evaporator models have been very popular and [7],...,[16] are all in this category. Distributed models are gaining the field in the recent years [17], [18], [19], [20]. However, one of the key problems in the distributed models is the description of void fraction and two-phase flow. The momentum exchange between refrigerant liquid and vapour causes slip-effect which influences the mass distribution of refrigerant. A better void fraction model could make the distributed models more realistic.

2. Derivation of the Equations

Above it has been made clear that several possibilities exist for non-steady state modelling of dry-expansion evaporators. The choice normally depends on the compromise between the accuracy and complexity of the model to be made. However, for a relatively flexible model of dry-expansion evaporators, a basic demand may be that the model has to be applicable to as many types of evaporator pipe configurations as possible. This requires the model to be distributed. Moreover, the mass transportation process gradually takes place inside the evaporator tube. To predict a correct refrigerant mass distribution, the model should be also distributed. Starting from this point of view, the following modelling strategies can be made: Strategy 1. The model to be set up will be distributed in structure and the mass balance equation will be solved in the form of partial differential equations. At the meantime, the heat transfer between refrigerant and tubes, tubes and other tubes, tubes and air will be calculated locally; Strategy 2. Because the refrigerant temperature in the two-phase flow region is almost constant except the effect of pressure drop, the energy balance equation can be easily integrated in this region so that it will be solved lumped. The temperature decrease caused by pressure drop will be added later on. This strategy avoids simultaneous solution of the mass balance and energy balance equations which are otherwise all partial differential equations. In the single-phase region, the refrigerant has little mass and heat contents and it is possible to consider the refrigerant temperature as a zero-order parameter (no heat and mass accumulations in the region), while the pipe wall of this region still plays a dynamic role; Strategy 3. The momentum equation in the two-phase flow region is the most difficult equation compared to the other two. This equation determines the pressure drop-mass flow rate correlation as well as the slip factor: void fraction model. Due to the quick equilibrium of the momentum transport, the momentum equation can be assumed as time-independent. The overall momentum equation for both liquid and vapour together will be replaced by a pressure drop-mass flow rate correlation which is derived semi-empirically. The separated equations for each of the two phases will be solved beforehand by using standard computer packages like PHOENICS. The results will be fitted into a void fraction-slip factor function

2.1 Two-phase flow region on the refrigerant side

If we exclude the possibility that subcooling exists in the evaporator, normally a dry-expansion evaporator can always be divided into two completely different regions: two-phase flow region (evaporation region) and single-phase region (superheat region). Most of the length of the evaporator tube passed by the refrigerant belongs to the two-phase flow region. One-dimensional two-phase flows can mathematically be described with the so-called separated flow model which takes account of the fact that the two phases can have different properties and velocities. The separated flow model consists of 7 equations: continuity, momentum and energy equations for both phases and an interphase transfer rate equation describing how the phases interact with each other. However, a detailed description of the interphase energy transfer mechanism is by far impossible. The energy equation is usually written for the combined flow of the mixture. A simplified one-dimensional two-phase flow model can be made by considering the system shown in Fig. 1 (see [21]).

Figure 1 Control volume for the refrigerant inside the evaporator pipes.
Continuity equation for the vapour phase:

\[ \frac{\partial}{\partial t} \left( \langle \alpha > \rho \right)_{v} + \frac{\partial}{\partial z} \left( \langle \alpha > \rho \ uu \right)_{v} = m_{v} \]  

(1)

where \( \langle \alpha > \) = \( A_{v} / A \) and \( \Lambda = A_{v} + \Lambda_{i} \).

Continuity equation for the liquid phase:

\[ \frac{\partial}{\partial t} \left( (1-\langle \alpha >) \rho \right)_{l} + \frac{\partial}{\partial z} \left( (1-\langle \alpha >) \rho \ uu \right)_{l} = -m_{l} \]  

(2)

Addition of (1) and (2) with rearrangement can results in the following equation,

\[ \frac{\partial}{\partial t} \left( \langle \alpha > \rho \right)_{v} + \frac{\partial}{\partial t} \left( (1-\langle \alpha >) \rho \right)_{l} + \frac{\partial}{\partial z} \left( \langle \alpha > \rho \ uu \right)_{v} + \frac{\partial}{\partial z} \left( (1-\langle \alpha >) \rho \ uu \right)_{l} = \frac{\partial}{\partial t} \left( \langle \alpha > \right)_{v} + \frac{\partial}{\partial t} \left( (1-\langle \alpha >) \right)_{l} \]  

(3)

(3) has the form of a propagation equation. \( \langle \alpha > \) and \( \Omega \) are respectively corresponding to the velocity and source of the void fraction propagation.

Energy equation for the mixture:

\[ \frac{\partial}{\partial t} \left( \langle \alpha > \rho \ h + (1-\langle \alpha >) \rho \ h \right)_{l} + \frac{\partial}{\partial z} \left( \langle \alpha > \rho \ uu \ h + (1-\langle \alpha >) \rho \ uu \ h \right)_{l} = \left( \langle \alpha > \right)_{v} \]  

= \left( \langle \alpha > \right)_{v} \]  

(4)

Integration of (4) in the domain of \( z = (0, L_{e}) \) gives

\[ \frac{\partial}{\partial t} \frac{Q}{ho} \frac{(h - h_{T})}{h_{T}} M_{f} = \frac{\partial}{\partial t} \frac{dM}{dT} + M_{T} \frac{dT}{dM} \]  

(5)

where

\[ Q = \int_{L_{e}}^{L_{e}} \pi d \alpha \langle \alpha > \frac{dQ}{dT} \ dz \]  

\[ M_{f} = \int_{L_{e}}^{L_{e}} \alpha \langle \alpha > \rho \ h \ dz \]  

\[ M_{T} = \int_{L_{e}}^{L_{e}} \alpha \ h_{T} \ dz \]  

\[ \bar{M}_{f} = \int_{0}^{L_{e}} \frac{\partial M}{\partial t} + M_{T} \frac{dM}{dT} \]  

The heat transfer coefficient in the two-phase flow region is calculated by using the correlation of [29].

Momentum equation for the vapour phase:

Since the equilibrium process of momentum transport is rapid, the momentum equation will be considered as time-independent.

\[ \frac{\partial}{\partial z} \left( \langle \alpha > \rho \ uu \right)_{v} = \frac{\partial}{\partial t} \left( \langle \alpha > \left( F_{w} + F_{w} \right) \right) \]  

(6)

In case of annular flows, \( F_{w} \) is zero since there is no contact surface between the pipe wall and the vapour refrigerant.

Momentum equation for the liquid phase:

\[ \frac{\partial}{\partial z} \left( (1-\langle \alpha >) \rho \ uu \right)_{l} = \left( (1-\langle \alpha >) \left( F_{w} + F_{w} \right) \right) \]  

(7)

By definition: \( M = \langle \alpha > \rho \ u ; M_{l} = (1-\langle \alpha >) \rho \ u \), addition of (6) and (7) gives
\[
\frac{\partial}{\partial z}(M \mathbf{u} - M \mathbf{u}^e) = \frac{\partial P}{\partial z} - F_{pw,\ell p} \tag{8}
\]

which is the momentum equation for the vapour-liquid mixture and will be replaced later on by an empirical correlation for calculating the pressure drop in the two-phase flow region (see [33,34]).

Interfacial friction equation:

The interfacial friction force in (4) and (5) is calculated as follows (see [22], [27]),

\[ F_i = \left( -p_i \xi_i (u_i - u) \right) \tag{9} \]

where the interfacial momentum-transport coefficient \( f_i \) is dependent on the interface conditions. A correlation for the case of wavy-interfaces has been given in [22].

2.2 Single-phase flow region on the refrigerant side

The situation in this region is relatively simple, because there is no phase change. And since the heat capacity of superheated vapour is small, the refrigerant can be considered as a system without mass and energy accumulations as well as incompressible.

Continuity equation: \[ \frac{dM}{dz} = 0 \tag{10} \]

Energy equation: \[ \frac{d(Mh)}{dz} = (\pi d)q_j \tag{11} \]

\[ \frac{dh}{dz} = \frac{\partial h}{\partial T} \frac{dT}{dz} + \frac{\partial h}{\partial P} \frac{dP}{dz} \]

Momentum equation: \[ \rho_i u_i \frac{d}{dz} = -\frac{dP}{dz} - F_{pw,i} \tag{12} \]

2.3 The pipe wall

The metal pipes of the evaporator are the very important elements which possess considerable mass and heat capacities. Heat transfer process in the pipes is 3-dimensional. Usually the evaporator pipes are assembled with extended fins in order to intensify the outside heat transfer. Thus, hereafter the concept of pipe wall always involves the evaporator pipes as well as the fins. Fig. 2 shows one nodal point of the plate-finned pipes of an evaporator. The basic equations will be subjected to this point.

\[
\frac{\partial T}{\partial t} = \frac{c_p M}{p_{pw} \rho_{pw}} \sum_{i=1}^{n} \left[ \frac{\dot{Q}_i}{T_{i,\ell}} \right] + \dot{Q}_f + \dot{Q}_d \tag{13}
\]
The heat conduction between the pipe walls is calculated according to the method given in [30]. Because the pipes and fins normally consist of different materials, the specific heat should be averaged according to the masses of the materials:

\[
c = \frac{c_p \text{pipe} M_{\text{pipe}} + c_f \text{fin} M_f}{M_p}
\]

(14)

2.4 The air side

The air flowing through the evaporator usually includes air and water vapour. Thus the problem on the air side is actually a diffusion problem which comprises concentration and energy equations. Again, because of the small heat capacity of air, this side is considered as a system without mass and energy accumulations.

\[
\frac{d(M \rho \omega)}{dz} = (\pi D)\omega
\]

(15)

\[
\frac{d(M h \omega)}{dz} = (\pi D)\omega
\]

(16)

Enthalpy definition:

\[
h = c_p T + c_v \theta w + \gamma w
\]

(17)

The air side heat transfer is calculated by using the McQuiston correlation [31,32].

3. Void Fraction Model (VFM)

In equation (3), \(C_2\) is still not known, and should be calculated from the VFM. The purpose of the VFM is to account for the fact that the vapour and liquid travel at different velocities in the two-phase region. The void fraction distribution is determined by equations (4) and (5). However solution of these two equations is difficult. From the literature, three methods can be found to tackle the problem. 1) empirical method; 2) analytical method; 3) numerical method. For example, Hancock's correlation [24] is an empirical one between \(C_2\) and \(<\alpha>\); Levy's correlation [25] is an analytical one between \(x\) and \(<\alpha>\); Zivi's correlation [26] is also an analytical but between \(u/u_L\) and \(x\). However, all these models are subjected to certain required conditions and cannot cover the whole range encountered in the two-phase flows of refrigeration evaporators. The most accurate method is to solve the governing differential equations numerically with a small number of assumptions. With the development of computational fluid mechanics and computer science, numerical solution of 1-dimensional two-phase flows can be obtained by using advanced computer packages. In this paper, PHOENICS [27] has been made use of. The following is the output of the application of PHOENICS.

![Figure 3](image-url)  
**Figure 3** Computational output from PHOENICS. Left: the vapour and liquid velocity distributions over the evaporator pipe; right: the void fraction distributions in the pipe.

As an example problem, the refrigerant with an inlet velocity of 0.2 m/s and quality of 0.0 flows through a tube of 15 m, under constant even distributed heat flux. Fig. 3 shows the vapour and liquid velocity, as well as the void fraction distributions along the tube. Given a different interfacial transport coefficient \(f\), the distribution is different from
another. Based on the numerical solutions, a function between \( u / u_1 \) and \(< \alpha >\) can be fitted for a certain \( f_i \) value by using the least-square root method. The following equation is in the form of a polynomial:

\[
u_i = \sum_{i=0}^{L_0} B_i < \alpha >^i
\]

The comparison of the obtained correlation to the others introduced above is illustrated in Fig. 4. It can be found that the range of \( f \) = 3 to 300 almost covers all of the models. However, selection of \( f \) value has to be through experiments.

![Figure 4](image1)

**Figure 4**  
Comparison of the existing void fraction models to that from PHOENICS.

Equation (3) is the void fraction propagation equation in which the wave speed \( U \) and source \( \Omega \) are dependent upon the void fraction model. The propagation speed is directly related to the position of the transition point between the two-phase and single-phase regions. On the other hand, the mass content in the evaporator is important when the refrigerant charge and inventory are calculated. It also affects the off-period behavior of the evaporator. Different void fraction models can result in different mass content distributions in the evaporator.

4. Off-period Modelling

Since during off-periods no refrigerant mass flow is present, the modeling on the refrigerant side can be treated differently. For the air side as well as the pipe wall there is no difference between on-periods and off-periods, provided that the evaporator's fans keep running.

![Figure 5](image2)

**Figure 5**  
Experimental result to verify the assumption that a saturated equilibrium state is immediately reached throughout the whole evaporator coil, as soon as the compressor is stopped.
The experiment (see [35]) has demonstrated that as soon as the compressor stops, the inlet and outlet refrigerant temperatures in the evaporator immediately reach the saturated temperature corresponding to the measured pressure (see Fig. 5). This phenomenon supports the assumption that a saturated equilibrium state is immediately reached throughout the whole evaporator coil. This assumption allows to consider the refrigerant as homogeneous, that is, lumped model (see Fig. 6).

![Figure 6](image)

**Figure 6** An evaporator "tank" which is assumed to represent the off-period evaporator.

Continuity equation for the vapour:  
\[
\frac{d(\alpha \rho)}{dt} = \dot{M}_v
\]  
(19)

Continuity equation for the liquid:  
\[
\frac{d(1-\alpha)}{dt} = -\dot{M}_v
\]  
(20)

Substituting (20) into (19) and using  
\[
\frac{d\rho}{dt} = \frac{d\rho}{d\rho} \frac{d\rho}{dP} \frac{dP}{dT} \frac{dT}{dt}
\]

and neglecting \( \rho_v/\rho_l \)

\[
\dot{M}_v = V <\alpha> \frac{d\rho}{dT} \frac{dT}{dt}
\]  
(21)

Energy equation for the mixture: is the same as (5)

5. Presentation of the Results

The distributed and non-steady state model for air coolers has been implemented on a SUN workstation (series No. 3/60) at the Lab. for Refrigeration and Indoor Climate Engineering of Delft University of Technology. Simulations have been made for several operations. In order to validate the model, a test stand has been set up on which the experiments have been carried out. Hereafter the comparison between them is described.

The test stand consists of a refrigerating system and a refrigerated room. The evaporator used in the test stand is an air cooler with cooling capacity of 10 kW (\( T_e = 10^6 \) C, \( T_s = 10^6 \) C). Since the experiments were aimed at checking the evaporator model, the condenser and refrigerated room were intended to be excluded. A computer-controlled datalog was used to register the data in time domain. The registered data pertinent to the condenser and refrigerated room were used as input in the computer program. These data include the condensation pressure, the subcooling temperature, the air inlet temperature and humidity of the evaporator. Because of the close interactions among the evaporator, thermostatic expansion valve and reciprocating compressor, it was decided to run the evaporator model together with the models of the other two components which were developed and validated by van der Meer [14] at the same Laboratory. To limit the possible influencing effects, the experiments were made under circumstances without dehumidification. However, the investigation is still going on towards more complicated situations. Three sets of results will be presented below.

The first set is the results for 5 steady-state operations. Using steady-state processes to validate dynamic models is a very economical and short-cut approach. Given a certain steady-state circumstance, the dynamic model should in principle predict the steady-state behaviour of the system, if the simulation time is long enough. The deviation between the simulated results and the reality can be used to evaluate the accuracy of the model. Tab. 1 shows the comparison between the simulations and experiments.

The second set is about the dynamic behaviour of the evaporator during 3-stage capacity control. The compressor was controlled automatically by a thermostat which had a sensor in the refrigerated room. According to a certain logic setting to the thermostat, the compressor might run at two different rotational speeds or just stop. During off-periods, the thermostatic expansion valve was supposed to close completely and immediately. The operation started with a static period of 150 seconds. Then several actions were imposed or just happened to the system. *Action 1*: the heating load in the room was manually switched from 6.866 kW to 0.866 kW, while the compressor kept running still at 700 rpm. *Action 2*:...
The experimental and simulation results which include 5 steady-state operations.

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Table 1: The experimental and simulation results which include 5 steady-state operations.

The experimental and simulation results (for the simulation, fi = 30). Left: the refrigerant temperatures; right: the air temperatures.

The experimental and simulation results (for the simulation, fi = 30). Left: the refrigerant temperatures; right: the air temperatures.

at the moment time=470 seconds, the compressor stopped automatically due to decreased air temperature. At the same time, the heating load was again manually switched to 9.866 kW. **Action 3:** when time=720 seconds, the compressor automatically operated at 700 rpm, while the heating load remained the same. **Action 4:** at 1300 seconds, the compressor automatically switched to high rotational speed, 1100 rpm, but the heating load was changed to 0.866 kW. **Action 5:** at 1550 seconds, the
compressor resumed back to 700 rpm and the heating load did not change. **Action 6:** at 1900 seconds, the compressor stopped again because of the low air temperature. However, in order to start the compressor, the room was heated up by 9.88 kw of heating. **Action 7:** at 2100 seconds, the compressor was activated at 700 rpm. At 2250 seconds, the operation was quitted. Fig. 7 shows the comparison between the experimental and simulation.

The third set is about an operation with long off-period. The experiment also started with a static period of 150 seconds. There were only two actions in this experiment. **Action 1:** at 150 seconds, the compressor was manually stopped and the heating load remained the same throughout the whole operation, the air temperature in the refrigerated room then resol gradually. So did the refrigerant temperature and pressure in the evaporator. The expansion valve closed because of the flooding of the evaporator. **Action 2:** at 870 seconds, the compressor was manually turned on at 700 rpm. Because the superheat was small, little hunting appeared. At 1500 seconds, the experiment was quitted. Fig. 8 shows the comparison between the experimental and simulated results.

6. Conclusions

From the results presented above, it can concluded that the model is able to describe the evaporator dynamic behaviour caused by capacity controls like on-off or multi-stages. Specially for on-periods, the simulations and experiments are in good agreement. Below the advantages and disadvantages of the model will be pointed out. Following such a judgement, the application areas of the model are outlined.

Advantages

1) The void fraction propagation equation as introduced to the dynamic modelling of a dry-expansion evaporator can intuitionistically depict the transient motion of the refrigerant liquid inside evaporator coils. The two-phase flow slip-effect is included in the parameter C, that may be treated separately by using the standard computer packages to solve two-phase flow problems.
2) By adoption of the distributed modelling methodology, the model is very flexible for various types of evaporator coil configurations. Meanwhile, the heat conduction through the pipes and fins can be computed more accurately.
3) Integration of the energy balance equation in the two-phase flow region avoids simultaneous solution of the energy balance equation and propagation equation. This treatment to large extent makes the model workable, which otherwise would take unacceptable computation time.
4) The heat transfer coefficient in the two-phase flow region can locally be calculated according to the local vapour quality.

Disadvantages

1) The model needs relatively long computation time. The author’s experience is that it costs about 10 hours on a SUN 3/60 series work station to simulate 1 hour of dynamic operation for an evaporator divided into 60 elements, with using 0.1 second time-step. Therefore it is suggested to use variable time-steps in order to save computational costs.
2) Frost-formation is not yet taken into account in the model

Based on the evaluation to the model, it is suggested that the complex distributed non-steady state model be useful only for short-term simulations, such as, 1 hour (rather than days or years). In case of long-term simulations, combination of a steady-state refrigerating machine model with a dynamic refrigerated room model is feasible.

References


