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Dynamic Modeling of Bulk Milk Cooler

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ABSTRACT

It is well known fact that operation of many refrigeration systems is dynamic in nature and bulk milk coolers are no exception to this. It is necessary to study bulk milk cooling systems and improve them to reduce chilling time and energy consumption within desired limits. Bulk milk coolers are analyzed and diagnosed with a simple thermodynamic model. A global approach is required whereby detailed component models are linked together to develop an accurate and complete simulation model package of vapour compression bulk milk cooler. These component models include mass, momentum and energy balance equation along with thermo physical property data and the appropriate heat or work transfer relationships. Solving these equations by method of substitution yields a prediction of refrigeration system behaviour. This helps in large reduction in time and cost spent in designing a system for given application, leading to the opportunity of achieving a more optimal design through evaluation of different design configurations. This dynamic model helps in adopting better control strategies to improve energy efficiency. The authors have attempted to develop such a global model for bulk milk coolers using vapour compression system with R22 as refrigerant.

Introduction

This paper presents the philosophy and simulation model of Bulk Milk Cooler(BMC). Bulk milk cooler uses the vapour compression system. These are used to cool the milk from its harvesting temperature of 35^{o}C to storage temperature of 4^{o}C. Existing bulk milk coolers consume relatively more energy and more time to complete the operation. The operation and performance of BMC depends upon the behaviour of vapour compression system consisting of reciprocating compressor, air cooled condenser, thermostatic expansion valve and jacketed plate type evaporator. behaviour of the system with respect to time.

The vapour compression system using R22 as refrigerant is most commonly used. This paper is limited to refrigeration machinery that utilizes this cycle. Other devices that use vapour compression system are domestic refrigerator, window air conditioner, heat pump and industrial refrigeration systems. However large differences exist among all these devices resulting in very different operating characteristics. These differences may be in the components or component configurations, the cooling capacity, secondary fluid, heating or cooling and control aspects. For the bulk milk cooler following are generally accepted as norms in the industry:

1. Large cooling requirements are coupled with large energy demands.
2. There is large refrigerant mass.
3. Reciprocating compressors are employed.
4. Direct expansion plate type jacketed evaporators with air-cooled plate fin type condensers are employed.
5. There shall be no sharp corners, Projecting parts inside the BMC.
6. The milk shall be cooled in 2 to 3 hours to desired temperature.

In the following section the philosophy behind developing simulation models for vapour compression direct expansion type BMC is discussed. To accomplish this task, literature for all forms of vapour compression equipment for both steady state and dynamic situations is reviewed. Paper describes operational behaviour of bulk milk cooler, assumptions in modeling and benefits that modeling provides and basic philosophy for developing a model for any physical system.

Literature Survey:

Of the simulations models relating to chillers that have been reviewed, all have been steady state situations. Gordon et al. [11] proposed a simple thermodynamic model that relates the coefficient of performance to the cooling capacity and no physical modeling is performed. Bendapaudi and Braun [2] have documented more detailed literature review wherein it is found that none of the papers included numerical study of the models developed in terms of discretization or integration algorithm. Braun et al. [3] developed a simulation model for variable speed centrifugal chiller where the main focus was on compressor. S. Wellsandt and Vamling [4] have investigated plate type evaporator for pressure drop and heat transfer. They have considered vertical configuration with falling film of water for analysis. W. Zhang et al. [5] have studied the flow boiling phenomena and have suggested generalized correlation for low Reynolds number flow through micro channels. Browne M.W. and Bansal [1] have reviewed about 60 research papers and revealed that detailed scientific investigation of Vapour compression system needs to done for both steady as well as transient states. Open literature directly related to bulk milk coolers is not found in the survey hence detailed scientific investigation bulk milk cooling system needs to be done. It will help in studying the dynamic behaviour of the system with respect to time. Due to the apparent lack of open literature it is needed to develop a complete bulk milk cooler model, which incorporates the detailed model of each component in the system.

The Operational Behaviour of BMC

It is well known that operation of many refrigeration systems is dynamic in nature. Bulk milk coolers are basically designed to operate at milk farms or dairy cooperative societies and are built in capacities ranging from 500 liters to 5000 liters. Small sized BMCs consists of one vapour compression system and medium to large sized BMCs consists of two vapour compression systems both operating simultaneously. These are designed to chill the given quantity of milk to 4°C within three hours. The chillers operate under two conditions

1. First Milking condition
2. Second Milking condition.

First milking condition requires three hours to cool the milk from 35°C to 4°C wherein tank is filled to its half capacity. Second milking condition requires three hours to cool the milk from 19°C to 4°C when the tank is completely filled.

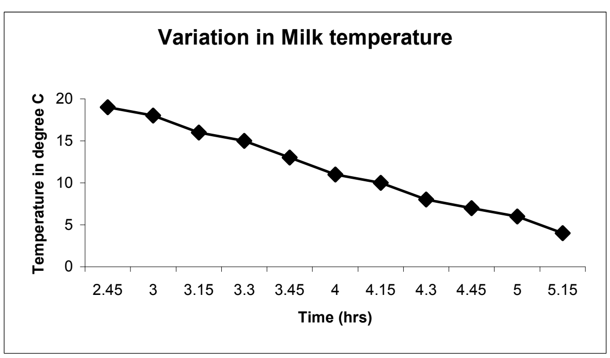

Figure 1 shows the temperature variation of 3000 liter. capacity BMC.

Operation of BMC is shut down when the milk temperature reaches 4°C. The temperature is further maintained by insulating the tank by polyurethane foam of sufficient thickness. Small amount of heat leak takes place from top cover since it is single walled and not insulated. Milk is continuously churned to maintain the homogeneity and avoid fatting. Agitator is directly attached to low rpm motor supported on top cover. Stirring also contributes somewhat to increase the heat transfer and maintain the uniform temperature throughout the BMC. The temperature gradient from top to bottom and from front to back side should be less than 0.5°C.

The experimental data on a 2000 liter and 3000 liter capacity BMC has been recorded at the time of its performance testing in industry. The cooling load varies reasonably along with compressor input power, which varies by 15% with 30% variation in cooing load.

Modeling and its benefits:

The goal of modeling is to gain an improved understanding of characteristics of the system. To do this it is necessary to describe the important aspects of the system in terms of mathematical relationships. For a refrigeration system this means defining relationships for each of the component in the system. Generally these component models include mass, momentum and energy balance equations along with thermo physical property data and appropriate heat and work transfer relationships. Solving these relations simultaneously would yield a prediction of refrigeration systems operation. Ideally if the model is detailed enough the predicted performance will closely match that of the actual system. The model will give following advantages:

1. Large reduction in time and cost spent on designing a system for given application.
2. Achieving more optimal design through evaluation of different design configuration.
3. Variation in component configuration that can be easily evaluated through a computer simulation to identify significant opportunities for improvement.
4. Better designs to improve energy efficiency.

There are no standard rules that ensure development of correct simulation model of a particular system. It may therefore be classed as individual discipline.

The objectives of dynamic modeling of BMC are:

1. To predict response of the system and its components to changes in operating conditions.
2. To predict the system parameters at any instant during operation.
3. To determine the necessary improvements to reduce the chilling time to the lowest extent possible.

A refrigeration system as with any other physical system is infinitely complex. It will not be possible nor desirable to describe all of the phenomena in the system. It has been found from various studies that existing steady state models of liquid chillers may disagree with experimental data by up to 25%.

Popular Approaches to Modeling Vapour Compression Refrigeration Systems

A significant body of literature exists on modeling of refrigeration equipment of various configurations. In terms of approaches to modeling the most important differences are in the way the refrigerant in the heat exchanger are treated. Fig 2 gives brief review of the methods and techniques that might be used.

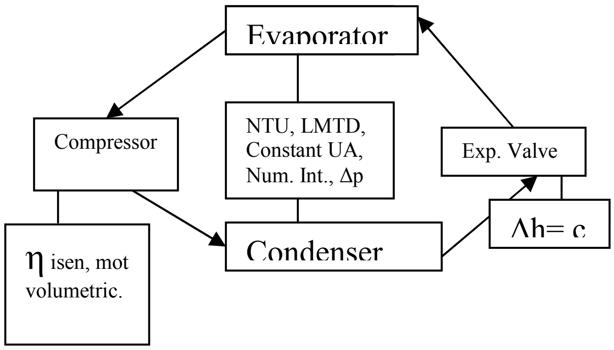

Fig. 2 Dynamic Modeling Techniques

A more detailed review of literature on dynamic modeling of refrigeration systems is documented by Bendapaudi and Braun[2] (2002a,2002b). Several other researchers have documented dynamic system models based on essentially one of the above approaches.

Model Formulation:

A dynamic model of bulk Milk cooler consists of dynamic component models of Compressor, Condenser, Expansion Valve and Evaporator. These component models and overall system models are described in this section.

Compressor:

An exact analytical model of reciprocating compressor is extremely complex. Complicated refrigerant flow through valves, flow through compressor inlet and outlet, a complicated heat transfer mechanism, the presence of lubricant within compressor shell, refrigerant charge and refrigerant oil mixture concentration and properties are all phenomena that are almost analytically indescribable. Two parameters that are commonly used to quantify reciprocating compressor operation are volumetric efficiency and compressor power. Volumetric efficiency is directly proportional to refrigerant mass flow rate. So mass flow as directly measurable value is utilized in the model. Popovic and Shapiro [3] proposed semi empirical model with following assumptions into modeling procedure:

1. The modeled compressor cycle is the approximation of real compressor cycle.
2. Compressor mass flow losses are accounted for in the model.
3. Compression and expansion in compression cycle are polytropic processes with equal polytropic exponents.
4. The oil has negligible effects on refrigerant properties and compressor operation.
5. There are pressure drops at suction and discharge compressor lines.
6. The pressure drop processes are assumed to be processes with constant enthalpy.

Piston displacement (PD) is the volume actually swept out during one cycle. Compressor manufacturers usually provide rate of piston displacement (RPD) in volume per compressor shaft revolution rather than piston displacement volume

$$PD = Vc - Va = RPD.\ RPM.\ Time \text{ --------- } (1)$$

by substituting this expression into volumetric efficiency definition, the following expression for volumetric efficiency is obtained

$$\eta_v = \frac{m}{RPD.RPM.Time} \bullet v_{in} \text{ --------------- } (2)$$

In equation (2) m stands for mass intake refrigerant vapour. The intake mass divided by time gives mass flow rate through the compressor. Combining equation (2) with definition of volumetric efficiency the final equation determining the refrigerant mass flow rate through the compressor

$$m = \frac{RPD.RPM}{v_{in}} \bullet \left[1 + C - C\left(\frac{P_{dis}}{P_{suc}}\right)^{1/n}\right] \text{----}(3)$$

Popovic and Shapiro have given the correlation for heat transfer loss coefficient and compressor cylinder heat loss expressed as

$$Q_{loss} = 1 - \frac{Q_{cyl}}{Q_{mot}} \text{ --------- } (4)$$

Input work to the compressor is given by equation

$$W = \frac{m(h_{out} - h_{in}) - W_{cal}\left[1 - \left(Q_{cyl}/Q_{mot}\right)\right]}{1 - \left[1 - \left(Q_{cyl}/Q_{mot}\right)\right]} \text{ ------- } (5)$$

The postulated model needs eight input quantities to determine the mass flow rate, refrigerant outlet rate and required compressor power. The input information includes refrigerant inlet state, outlet refrigerant pressure, clearance volume, motor speed, polytropic exponent and two compressor performance characteristics to be determined using experimental data.

Condenser:

Condenser used is of plate fin air cooled type. Refrigerant flows through the tubes and air flows across the tubes between the fins. The standing assumptions made are

1. Flow is one-dimensional.
2. Negligible heat conduction along the axial direction of heat exchangers.
3. Invariant mean void fraction in two phase section during short transient.
4. Negligible refrigerant pressure drop along the heat exchanger.
5. storage capacitance of mass and thermal energy of all single phase sections is negligible compared to two phase section that dominates heat exchanger dynamics.

The governing equations for condenser can be formulated as

$$\rho_l h_{fg} A(1-\gamma)\frac{\partial l}{\partial t} = m(h_l - h_i) + \alpha_i \pi D_i l(T_r - T_w) \text{-- } (6)$$

$$AL\frac{\rho_g}{\partial P_c} \bullet \frac{\partial P_c}{\partial t} = m - \frac{\alpha_i \pi D_i l(T_r - T_w)}{h_{fg}} \text{ -------}(7)$$

$$(C_p \rho A)_w \frac{\partial T_w}{\partial t} = \alpha_i \pi D_i l(T_r - T_w) + \alpha_0 \pi D_0 l(T_a - T_w) \text{ ---------- } (8)$$

Expansion Valve

Thermostatic expansion valve is used to regulate the refrigerant flow rate such that a constant amount of superheat is maintained at the evaporator exit. During the expansion process the refrigerant liquid passes through the valve and a portion of it flashes in the reduced pressure of the evaporator. This expansion is universally modeled as an isenthalpic process on all refrigeration systems regardless of the geometry of the expansion valve. Browne and Bansal [ ] proposed a model developed by Armand et al for capillary by changing proportionality constant K . The relationship is given as

$$m = K \sqrt{2 \rho_l \Delta p} \text{ -------------------------- } (9)$$

$$K = K' + G(T_{sup} - T_{ref}) \text{ ---------------------- } (10)$$

$$K' = m / \sqrt{2 \rho_l \Delta p_{i-1}} \text{ ------------------------ } (11)$$

Where G is the proportional gain, T_{sup} is the actual superheat at the evaporator, while T_{ref} is the desired set reference value of the superheat and Δp_{i-1} is the pressure drop at the previous iteration.

Evaporator:

Plate type evaporator is used as the bottom of bulk milk cooler made from stainless steel and is 2.4 m long, 2m wide. The wall thickness is about 1 mm. Two steel sheets are attached by spot welding creating flow channels. Application of spot welding involves the possibility of fluid exchange between different channels. The effective heat transfer area is 4.8 m^2. Zhang and Hibbiki [5] have developed generalized correlation for flow boiling heat transfer at low liquid Reynolds number in small diameter channels. It is given as

$$\alpha\, t_p = S.\, \alpha_{pb} + F.\, \alpha_{sp} \text{ ------------------------ } (12)$$

Where $F = \xi \,.\, \text{Ø}_f = 0.64\, \text{Ø}_f$ ------------------- (13)

For liquid single phase heat transfer coefficient for turbulent flow correlation given by Dittus Boelter is adopted.

$$\alpha_{sp} = 0.023\, Re_f^{0.8}\, Pr_f^{0.4}\, (k_f/D_h) \text{ ----------------}(14)$$

Since the refrigerant enters the evaporator subcooled the calculation method also accounts for subcooled boiling. This is done by comparing the heat flux for onset of nucleate boiling and actual heat flux.

$$\phi_{onb} = \frac{2\sigma . T_{sat} \alpha_{fo}}{r_{cr} \rho_g h_{lg}} \text{ ------------ } (15)$$

Results and Discussions

The mathematical description of refrigeration process outlined above leads to system of nonlinear algebraic equations. These are solved by successive substitution method. Figure 3 shows the variation in the measured work input to the compressor and the predicted work input. It is found that the predicted work input to the compressor closely matches with actual. Actual value of work input being larger than the predicted. The variation is found to be 25%. Some other correction factor needs to be introduced with more detailed analysis.

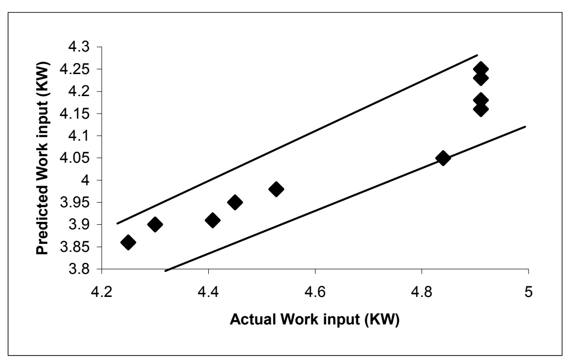

Fig.3 Actual Compressor work input verses predicted work input.

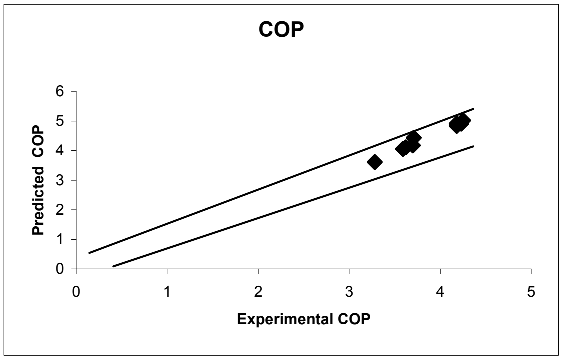

Fig 4. Experimental COP verses Predicted COP

Figure 4 shows the variation in experimental COP with that of the predicted COP of BMC. Predicted and actual figures closely match which show that the overall model give close results with that of experimental. It is observed that model slightly over predicts the COP values. Figure 5 shows variation in condenser effective temperature between the experimental and predicted values. The predicted values are slightly higher than the experimental values but lie within the range of ± 10%.

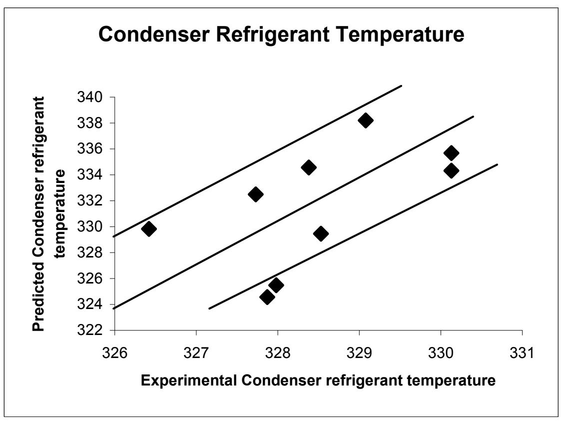

Fig. 5 Experimental Condenser Refrigerant Temperature verses Predicted Condenser refrigerant temperature.

Conclusion:

The predicted and the experimental values of performance parameters such as work input, coefficient of performance, and condenser refrigerant temperature closely match with each other. But it still required to improve these models to give more accurate results with less number of inputs. Detailed model of evaporator employing specific or more accurate heat transfer correlation for channel geometry is required to be developed to accurately predict the evaporator refrigerant temperatures and areas of single phase and two phase flow. Currently the evaporator model gives more variation of evaporator temperatures about 30% as compared to other results. The modular nature of the system model allows the use of other compression and expansion devices. The heat exchanger models are developed from first principle and are thus scalable for any application.

Nomenclature

Alphabets

A	m^2	Area
C	-	Clearance ratio
Cp	J/Kg K	Specific heat at const. Pressure
D	m	Diameter
h	J/Kg	sp. Enthalpy
L	m	Length
l	m	Length of Two Phase Section
m	Kg/s	Refrigerant mass flow rate
P	N/ m^2	Pressure
Q	W	Heat Transfer rate
r	m	radius
T	K	Temperature
v	Kg/m^3	Specific volume

Greek

η	-	Efficiency
ρ	Kg/m^3	Density
γ	-	void fraction
α	W/m K	heat transfer coefficient
ξ	-	adjustment parameter
Ø	W/m^2	heat flux
σ	N/m	surface tension

Subscripts

a	Ambient
cal	calculated
cr	critical
cyl	cylinder
dis	discharge
f	saturated liquid
fg	two phases liquid and gas
g	gas (vapor)
h	hydraulic
i	inside
in	inlet
l	two phase length
mot	motor
o	outside
out	outlet
pb	pool boiling
r	radius
sp	single phase
suc	suction
w	wall

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