COMPRESSOR MODEL FOR HEAT PUMP DRYER SIMULATION

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ABSTRACT

Heat pump dryer (HPD) is a system that provides a controllable drying environment so that the best quality of the dried products is attained. Because electricity is consumed, the application of HPD is debatable on the ground of the possibility of noncompetitive drying cost. Studies of HPD by many researchers were done by simulation. Since compressor is the major electricity consumer component in the HPD system, it is needed to be simulated with a correct model, especially when the cooling effect of the refrigerant-cooled compressor is significant. In this study, two compressor models were investigated, namely modified model and efficiency model. The results of simulation were verified by experiment. The simulation revealed that the modified model, which comprised of the polytropic-adiabatic compression and refrigerant cooling processes, was the most suitable model for HPD compressor simulation.

INTRODUCTION

Heat pump dryer (HPD) is an energy efficient drying system. It is simply a dryer equipped with a heat pump, normally a vapor compression type. The thermal energy of the dryer exhaust air is recovered by the change of refrigerant phase in the condenser and evaporator. The HPD is generally divided into two groups depending on the roles of the heat pump. The dehumidifying and heat recovering role, as shown in Fig. 1(a), provides low absolute humidity working air to the dryer at a relatively low temperature. The heat pump can also supply high temperature air at high absolute humidity, which is known as a heating and heat recovering HPD as depicted in Fig. 1(b). An extensive study has demonstrated that the heating and heat recovering HPD possessed higher performance (Prasertsan *et al.*, 1996b). Drying capability of the air from ambient is increased by raising its temperature at the

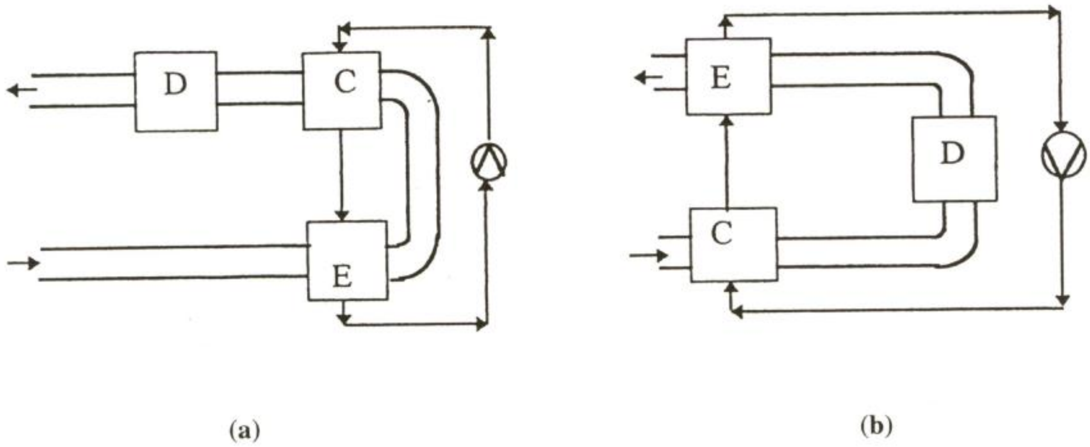

Fig. 1 Heat pump dryer system (E = evaporator, C = condenser, D = dryer)
(a) Dehumidifying-heat recovering HPD
(b) Heating-heat recovering HPD

condenser, hence reducing the relative humidity. Upon leaving the dryer, the inherent energy of the working air (both sensible and latent heat) is captured by the evaporator and released to the incoming working air at the condenser (via the compressor). The previous study was carried out by computer simulation of the HPD components, which were later on linked together to simulate the HPD system (Prasertsan *et al.*, 1996a, b). The system simulation was iterated until the convergence of the assumed states of the two working fluids (air and refrigerant) was achieved. Heat and mass transfer principle was employed for the condenser and evaporator simulations. The compressor model followed the polytropic compression process proposed by Threlkeld (1972) for which an adiabatic condition was assumed. A prototype HPD was built for both experiment verification of the simulation (Prasertsan *et al.*, 1997) and drying of agricultural materials (Prasertsan and Saen-Saby, 1997). Simulation of the system revealed the interaction of the two working fluids and the effects of the drying rate and the ambient conditions on the HPD performance (Prasertsan *et al.*, 1996b, c; Prasertsan *et al.*, 1995). In general, the simulation results agreed well with the experiment. However, the predicted compressor works were, to a certain extent, lower than that of the actual values. Such discrepancy affected the performance prediction of the HPD, especially the specific moisture extraction rate (SMER, kg water evaporated per kWh electricity). It was believed that the assumed adiabatic condition and the efficiency of the compressor were responsible for the discrepancy. Since the compressor is an important component determining the performance of the HPD system, there is a need to modify or adjust the compressor model to give result correctly. This paper presents a study to establish an appropriate compressor model for the HPD simulation.

COMPONENT MODELING AND HPD SIMULATION

The main components of the HPD are the dryer, the condenser, the evaporator and the compressor. Since dryers appear in many types and the drying performance is material and operation dependent, it is appropriate to be represented by an universal model. The proposed model is the dryer efficiency (DE), which is based on the drying efficiency of the working air, Equation (1).

$$DE = \frac{\omega_{a,out} - \omega_{a,in}}{\omega_{a,sat} - \omega_{a,in}} \tag{1}$$

The condenser and the evaporator models; -which were developed based on heat and mass transfer concept- are physical component dependent and complicated, but believed to be accurate. This allows the simulation of the HPD system to be verified by experimentation. The complexity of the model is resulted from the phase changing of refrigerant (single and two phase flow which affects heat transfer characteristic and pressure drop), moisture condensation on the evaporator surface (wet and dry region heat transfer).

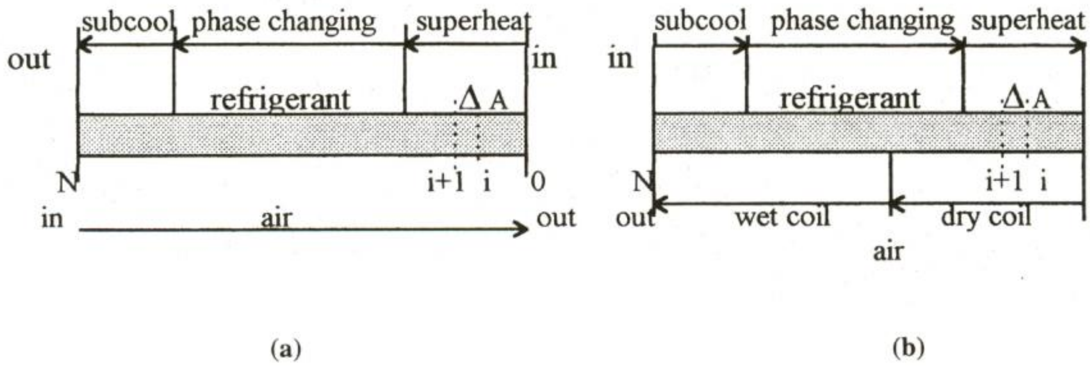

Fig. 2 Change of working fluids in heat exchangers
(a) Condenser
(b) Evaporator

The condenser model comprises of 3 regions as shown in Fig. 2(a). The simulation advances from the superheat side by infinitesimal section △A. The approaching temperature of the working air is known. The heat transfer in the condenser is analysed based on the effectiveness-NTU method (Kays and London, 1964). Within the single phase region, the leaving temperatures (of the working fluids) from section $\triangle A$ are determined from the approaching temperatures as given in Equations (2) and (3)

$$T_{a,o} = T_{a,i} + \frac{\varepsilon C_{min}(T_{r,i} - T_{a,i})}{C_a} \quad (2)$$

$$T_{r,o} = T_{r,i} - \frac{\varepsilon C_{min}(T_{r,i} - T_{a,i})}{C_r} \quad (3)$$

In the two phase region, the refrigerant temperature is constant at saturation point. The existing air temperature is given by

$$T_{a,o} = T_{a,i} + \varepsilon_{tp} (T_{r,i} - T_{a,i}) \quad (4)$$

According to the states of the refrigerant, the evaporator model consists of 3 regions (Fig. 2(b)). Normally the liquid phase does not appear in the evaporator, but it is included to generalise the model. On the air side, two regions appear, namely the dry region-where only heat is transferred- and the wet region- where heat transfer and mass transfer take place simultaneously.

In the dry region - single phase refrigerant, the existing temperatures of the refrigerant and air are determined from

$$T_{r,o} = T_{r,i} + \frac{\varepsilon C_{min}(T_{a,i} - T_{r,i})}{C_r} \quad (5)$$

$$T_{a,o} = T_{a,i} - \frac{\varepsilon C_{min}(T_{a,i} - T_{r,i})}{C_a} \quad (6)$$

and in the dry region-two phase refrigerant,

$$T_{a,o} = T_{a,i} - \varepsilon_{tp}(T_{a,i} - T_{r,i}) \quad (7)$$

In the wet region, the leaving air temperature and absolute humidity are given by Equations (8) and (9).

$$M_a C_{pa} (T_{a,i} - T_{a,o}) = h_a \left(\frac{T_{a,i} + T_{a,o}}{2} - T_s\right)\triangle A_0 \quad (8)$$

$$M_a (\omega_{a,i} - \omega_{a,o}) = \left(\frac{\omega_{a,i} + \omega_{a,o}}{2} - \omega_s\right) \left(\frac{h_a}{C_{pa} + \omega_{a,i} C_{ps}}\right) \triangle A_0 \quad (9)$$

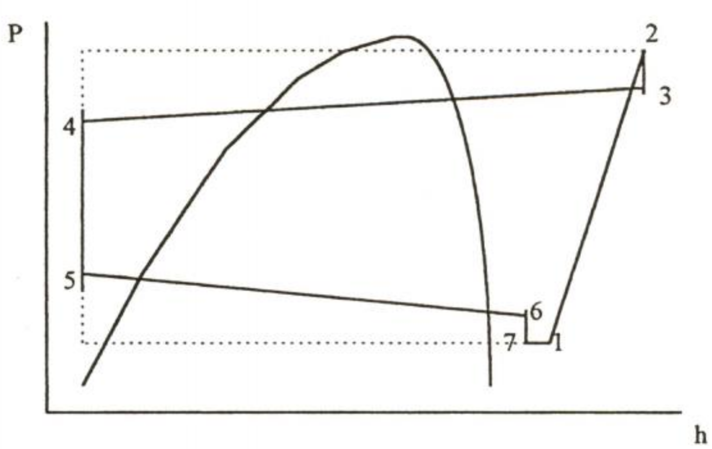

Fig. 3 P-h diagram of HPD refrigerant

P-h diagram of the refrigerant is illustrated in Fig. 3 which includes pressure drop in the condenser, the evaporator and the compressor valves. Detail of pressure drop calculation was given in Prasertsan *et al.*, (1996a). The models predict the change of states of air and refrigerant. The calculated air temperature and humidity were used for simulation verification, the results were given elsewhere (Prasertsan and Saen-Saby, 1997).

The refrigerant enters the compressor at state 6 and leaves at state 3 of Fig. 3. Upon entering the compressor, the refrigerant gains heat from the hot cylinder (process 7-1). A mathematical model for a reciprocating compressor, which was used by many workers (Taylor, 1987; Pendyala *et al.,* 1990; Vargas and Parise, 1995), was described by Threlkeld (1972) and here is referred as the original model (OM).

The volumetric efficiency is given by

$$\eta_v = [1+c-c\,(\frac{P_2}{P_1})^{1/k}](\frac{v_6}{v_1}) \qquad (10)$$

and the mass flow rate by

$$M_r = \frac{(PD)N\eta_v}{v_6}$$

$$= [1+c-c\,(\frac{P_2}{P_1})^{1/k}]\frac{(PD)N}{v_1} \qquad (11)$$

The power required to drive the compressor is given by

$$W_c = P_1 v_1 M_r\,(\frac{k}{k-1})\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad (12)$$

and the discharge temperature by

$$T_{r,2} = T_{r,1}\,(\frac{P_2}{P_1})^{(k-1)/k} \qquad (13)$$

The simulation was carried out based on Equations (1)–(13). Full details of air and refrigerant circuits simulation were described in Prasertsan *et al.,* (1996a). The results of the simulation explaining the HPD performance characteristics were presented in Prasertsan *et al.,* (1996b).

ALTERNATIVE MODELS FOR COMPRESSOR

It has been generally known that the compressor model is very complicated because of the lack of basic knowledge of fluid flow, heat transfer and thermodynamics of the process occurring in the compressor (Qvale *et al.,* 1972). The experiment verification revealed that the predicted compressor works differed from the measured values (Prasertsan *et al.,* 1997). The disagreement has been explained by the nonadiabatic compression process. The compressor used was a heavy duty and hermetically-sealed type. It was cooled by the refrigerant. Thus, the compression index is neither isentropic nor constant throughout the compression process. Furthermore, the compressor mathematical model depends strongly on the design, build and material used in the manufacturing. A rigorous model taking into account of all parameters will overshadow the HPD system simulation. Less detailed models, like that of Threlkeld (1972) or a model based on effciency, would be appropriate for most purpose and, hence, used as alternative models for this study. Although other models such as the real gas model and the empirical model is not examined, brief discussion is given in the result and discussion section.

Modified Model

The original model appropriately describes an open type compressor when a near-adiabatic condition is assumed. In a closed type compressor, nonadiabatic compression is inevitable because of the cooling effect (by the refrigerant). Consequently, the refrigerant enters the compressor at a higher degree of superheat. In this study, the controlled volume is the outer shell of the compressor, which near adiabatic condition is reasonably assumed. The refrigerant enters and leaves the controlled volume at states 1 and 2, respectively (see Fig. 4; for simplicity, pressure drops were not shown, but included in the simulation). Pressure at 2 was fixed by the pressure limit switch of the compressor, while the temperature was measured by a thermocouple. Alternatively, the modified model can be viewed from Equation (12):- by varying the k value. However, this choice was not proposed for the study simply because in reality the compression index is not easily obtained by experimentation.

Refrigerant leaves the evaporator at state 1. However, due to the cooling effect, it is further heated to state 1', the state at which the compression process starts. The degree of superheat at the suction valve therefore depends on the efficiency of the compressor. The actual process is 1'-2, for which the deviation (from isentropic) of the compression index is not known. For simplicity, the original model is modified by considering that the compression consists of two processes, namely isentropic compression (1'-2'), where the original model is applicable, and cooling process 2'-2. The compressor power is obtained from

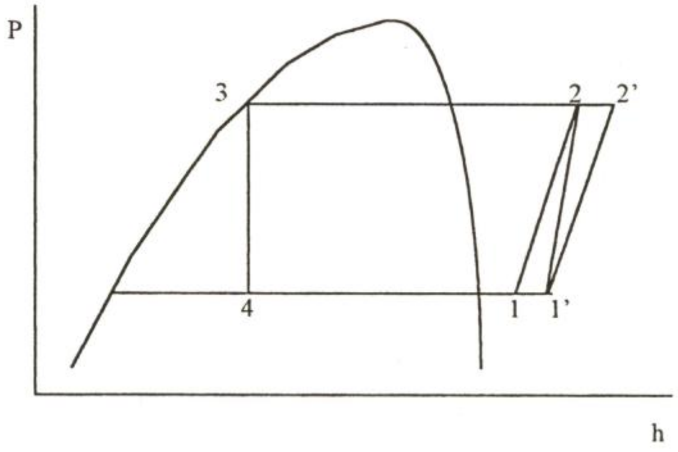

Fig. 4 P-h diagram of refrigerant with modified Threlkeled compressor model

$$W_c = P_1 v_{1'} M_{r1'} \left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] - M_{r1'}(h_{2'} - h_2) \qquad (14)$$

The simulation was carried out by varying the degree of superheat of state 1' (or cooling capacity), which was derived from the combined motor and compressor efficiency ranging from 65-85%.

The compressor modeling began with an assumed degree of superheat at state 1'. Refrigerant mass flow rate and the compressor work for process 1'-2', and $T_{2'}$ were determined from Equations (11)-(13). Deviation from ideal gas property was taken into account by the compressibility factor using generalized chart given in Russell and Adebiyi (1993). Heat loss (or cooling capacity) was calculated from the ideal compressor work and the assumed efficiency. Iteration for compressor was carried on until convergence of the assumed state 1' was achieved, after which simulation of refrigerant in the condenser, expansion valve and evaporator were performed. Air side simulation was undertaken along with the refrigerant in the respective components. Finally, the HPD performance was determined and compared with the experimental results.

Efficiency Model

The efficiency model is simpler than the model using detailed hardware design parameters. It is also very flexible to be modified to simulate a range of compressor types. By treating the compressor as a black box- without knowing the process inside- the model considers only the input and output conditions. Because of its simplicity, the efficiency model is very popular among researchers, for example, Jolly *et al.*, (1990), Poduval and Murthy (1992) and Jia *et al.*, (1993). In this approach, the isentropic efficiency is the key parameters of the compressor performance. The power of the compressor is

$$W_c = M_r (h_2 - h_1) \qquad (15)$$

The discharge enthalpy (h_2) is determined from isentropic efficiency (η_i)

$$h_2 = h_1 + \frac{h_{2,S} - h_1}{\eta_i} \qquad (16)$$

Alternatively, $$W_c = \frac{M_r (h_{2,S} - h_1)}{\eta_i} \qquad (17)$$

The model was simulated for an overall efficiency of 65%

RESULTS AND DISCUSSION

The experimental results were obtained from the experimental HPD constructed and tested at the Prince of Songkla University. Details of the test rig and experimentation were given elsewhere (Prasertsan *et al.*, 1997). In brief, the drying chamber was loaded by layers of wet blankets. The drying rate was determined from the change of the properties of air passing through the drying chamber. There were four HPD configurations undergone experimentation. However, only results of the heating-heat recovering HPD were used for

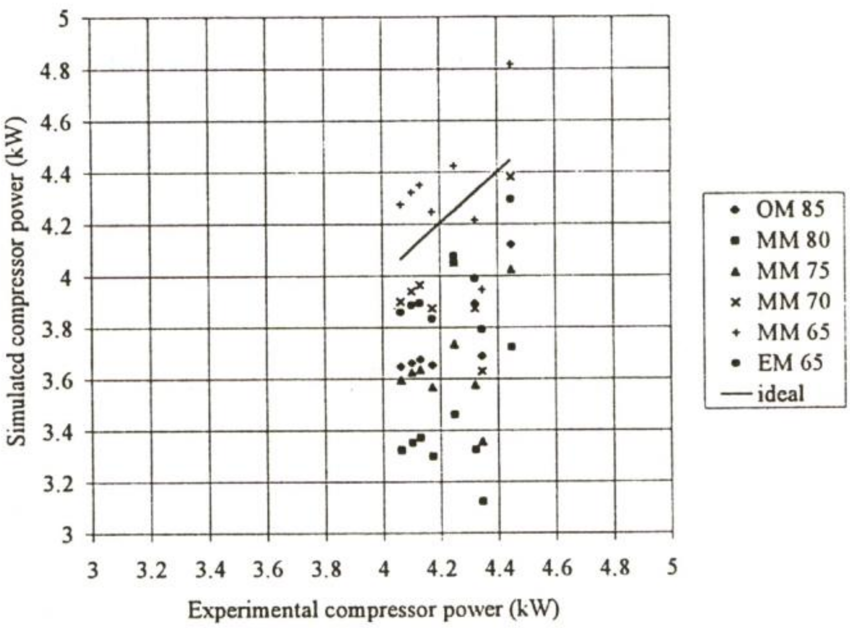

Fig. 5 Verification of compressor models

the compressor model verification. Conditions of experiment were the input data for the simulation study so that comparison can be made as given in Fig. 5. Results of the original model (designated as OM85), which the assumed efficiency was 85%, was included in Fig. 5 (extracted from Prasertsan *et al.*, (1997)).

For the modified model (MM), the simulated compressor work increased as the efficiency decreased from 80% to 65%. It was noticed that the original model, although simulated based on 85% efficiency, predicted a compressor work higher than that of the modified model having 75-80% efficiency. This appreciatively indicates the handicap of the original model when handles the simulation of the refrigerant-cooled compressor. The assumed adiabatic condition gave unrealisically low compressor work and, thus, the model should be modified as proposed. Efficiency of higher than 70% always gives the simulated work lower than that of the measured one. Fig. 5 implies that the modified model is accurate if the overall compressor efficiency of 65-70% is assumed.

The efficiency model (EM in Fig. 5) was preliminarily simulated for 65% efficiency. It was found that the simulation still predicted compressor work far less than the measured result. For the model to predict the compressor work accurately, the overall efficiency could be as low as 55%, which is, perhaps, too low (for the direct-drive compressor) to be acceptable. Apart from the relatively low efficiency, unlike the modified model, the efficiency model did not compensate for the refrigerant cooling effect. Therefore, no improvement or modification on the efficiency model was tried and it was considered that the modified model is more favourable in comparison to the efficiency model.

Alternatively, the compressor model can be derived empirically from experiment (Stoecker, 1989). However, such method has to be implemented with an extensive experiment as was presented by Kiatsiriroat *et al.,* (1994), which is not the scope of this study. Furthermore, the model obtained by this method depends on the type, design, model, and etc. of the compressor; thus lack of generality. Real gas model, which is the model of ideal gas-polytropic-adiabatic compression multiplied by compressibility correction function (Avallone and BaumeisterIII, 1987), was tested as an alternative. It was found that the real gas model predicted the compressor work less than that of the original model (about 6% lower) and was not sensitive to the intake and discharge states of the refrigerant (in the range of this study). Therefore, if the real gas model is modified in such a way as of the modified model (i.e., by incorporating the refrigerant cooling effect), it will simulate correctly for the overall efficiency slightly lower than that of the modified model.

The compressor work is proportional to the refrigerant mass flow rate and the specific volume of the refrigerant at the intake stage (Equation 14). Since the high specific volume is associated with the low mass flow rate, the compressor work is relatively constant (Prasertsan *et al.,* 1997). This was confirmed by the only 10% variation of the measured compressor works as it can be seen in Fig. 5. It should be noted that the measurement of the compressor works was done (and cross checked) by both a calibrated kWh meter and a current recorder; hence the measured data was relatively accurate. On the contrary, the simulation was based on the measured states of the refrigerant, which small error might magnify the deviation from the ideal line of Fig. 5. Therefore, in this study it is possible

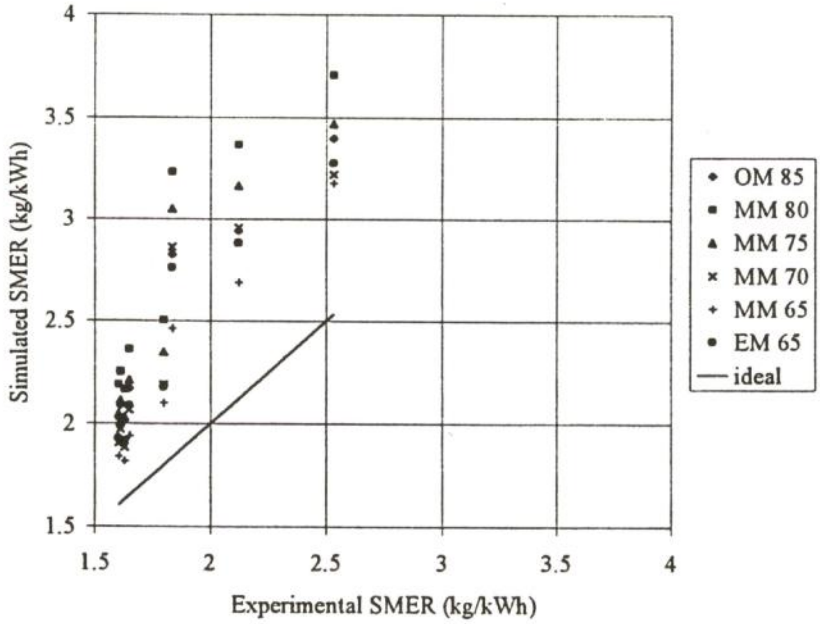

Fig. 6 Specific moisture extraction rate of different compressor models

to draw a conclusion that the modified model appropriately simulates the compressor work and the efficiency is in the vicinity of 65-70%

The specific moisture extraction rate (SMER) is the parameter indicating the HPD performance. The SMER has a special characteristic in the way that it is a function of both dryer (kg of water evaporated) and heat pump (compressor energy). The accuracy of the SMER prediction is, therefore, not easy to obtain. However, as the compressor works were better predicted (by the modified model), the simulated SMERs were improved as shown in Fig. 6. It is likely that the discrepancy between the simulated and experimental results was caused by measurement error, especially the air mass flow rate and the wet bulb temperature which were the most influential factors affecting the moisture extraction rate calculation. Since the air flow rate was as high as 1.5 kg/s, a temperature error of only 0.5 °C was significant.

CONCLUSION

Heat pump drying process is very complicated because of the interaction of the two working fluids and the non-steady state of the batch drying. The operation of the HPD for optimum condition is easier understood by simulation because the parameters could be controlled and manipulated as required. By this approach, the accurate component models are of ultimate importance. The compressor model is the critical component in the system simulation. Simulation based on the thermodynamic property of refrigerant and compressor characteristic is often represented by the polytropic adiabatic compression, which is not adequate for the real situation where the process is nonadiabatic. Alternative models for the refrigerant-cooled compressor were proposed and studied. It is concluded that the modified model, which considers the compression process consisting of two processes, namely polytropic-adiabatic compression and cooling processes, is a suitable model. The model predicted the compressor work correctly when the overall efficiency was 65-70%.

NOMENCLATURE

A	=	heat transfer area (m^2)
C	=	heat capacity rate ratio (decimal)
C_{min}	=	smaller heat capacity rate (kW/K)
Cp_a	=	specific heat of air (kJ/kg)
c	=	clearance ratio of compressor (decimal)
h	=	heat transfer coefficient (kW/m^2K) specific enthalpy (kJ/kg)
k	=	compression index (specific heat ratio)
M	=	mass flow rate (kg/s)
N	=	speed of compressor (r.p.s.)
P	=	pressure (kPa)
PD	=	piston displacement (m^3/rev)
T	=	temperature (K or °C)
W_c	=	compressor power (kW)
v	=	specific volume (m^3/kg)
ε	=	heat exchanger effectiveness
η	=	efficiency
ω	=	absolute humidity ratio

Subscripts

a = air

i = inlet or isentropic

o = outlet

r = refrigerant

s = saturated condition or isentropic

tp = two phase

v = volumetric

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