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PERFORMANCE STUDY OF FINNED TUBE EVAPORATORS IN A HUMID ENVIRONMENT

BREAK

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ABSTRACT

Finned tube evaporators are commonly used as cross flow heat exchangers in air conditioning and refrigeration systems for cooling and dehumidification. A simulation model for the steady-state performance of evaporator coils is developed in this paper. The coils are simulated as equivalent, parallel refrigerant circuits. The governing equations for heat and mass balance across the tube are presented. Local heat and mass balances are applied to a control volume of infinitesimal tube length along the surface of each row for different heat transfer zones, and the uneven distribution of air temperature inside the coils and the local flow characteristics of the refrigerant are simulated. The effects of evaporating temperature, coil face velocity, inlet air temperature and relative humidity on the evaporator performance are presented. The simulation results related to the humid environment with a fixed coil geometry are discussed.

NOMENCLATURE

A	effective heat transfer area (m^2)
C_p	specific heat (kJ/kg K)
d	tube diameter (m)
h	specific enthalpy (kJ/kg)
k	thermal conductivity (W/mK)
L	coil tube length (m)
m	mass flow rate (kg/s)
N_p	No. of parallel refrigerant circuits
N_O	No. of divided sections of a tube pass
Q	heat transfer rate
S_f	air-side fin pitch (m)
S_L	longitudinal tube spacing (m)
S_T	transverse tube spacing (m)
T	temperature (K)
ΔT_m	mean temperature difference (C)
U	overall heat transfer coeff. (W/m^2K)
W	humidity ratio of moist air (kg/kg)
y	half of fin thickness

Greek

α	heat or mass transfer coeff.(W/m^2K or kg/m^2s)
η_f	fin efficiency
ρ	densiiy (kg/m^3)
δ	incremental element

Subscripts

a = air; p = primary surface
e = extended surface; r = refrigerant
i = inside; s =saturated air
o =outside or outlet; 1, 2=inlet, outlet of an element

Superscripts

h = heat transfer; m = mass transfer

INTRODUCTION

Evaporator is one of the main components of refrigerating and air conditioning systems, in which refrigerant evaporates in tubes for the purpose of cooling and/or dehumidifying surrounding air or other substances. The increasing interest in heat recovery from refrigeration systems and the application of new environmental friendly working fluids have created a need for a detailed analysis of the individual components of the refrigeration system, especially in a humid environment where it is considered that the performance of a refrigeration cycle is dependent on the humidity of air.

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In the modelling of the evaporator coil, two approaches are commonly used. The ORNL model (Fischer, 1983) is developed using the parallel refrigerant circuits for a lumped heat exchanger. The other method (Ellison et al 1981, Domanski 1986, Oskarsson et al 1990) adopted a tube-by-tube computational approach. The thermal and fluid flow characteristic of each tube are computed individually using the local temperature and heat transfer coefficient. In the evaporator coil design, it is well known that design parameters such as evaporating temperature, coil face velocity and inlet air temperature affect the evaporator performance. However, performance study related to a humid environment has been scarce.

MODEL SIMULATION

In this study, cross-flow evaporator coils are simulated as equivalent, parallel refrigerant circuits with unmixed flow on both the air and the refrigerant sides. Consider a single refrigerant circuit as shown in Fig 1a. An element along a tube pass is taken as the control volume for analysis, as shown in Fig 1b. Mass flow rates of refrigerant inside each parallel circuit and moist air through each control volume are:

$$\delta m_r = m_r / N_p \qquad (1a)$$

$$\delta m_a = m_a / (N_p\, N_o) \qquad (1b)$$

The primary and the extended surface areas of the infinitesimal length are:

$$\delta A_p = \pi d_o \left[\delta L - 2y \left(\frac{\delta L}{S_f} + 1 \right) \right] \qquad (2a)$$

$$\delta A_e = \left(2 S_T S_L - \frac{\pi d_o^2}{2} \right) \left(\frac{\delta L}{S_f} + 1 \right) \qquad (2b)$$

The amount of heat transferred is governed by

$$Q = U_o A_o \Delta T_m = U_i A_i \Delta T_m = m_a \Delta h_a = m_r \Delta h_r \qquad (3)$$

Ideal Coefficient of Performance for Cooling (C.O.P.)

C.O.P. of evaporator coils is defined as the refrigerating capacity divided by the sum of energy required to compress a given amount of refrigerant from a suction pressure to a fixed discharge pressure isentropically.

Governing Equations

There are two heat transfer zones inside the tube corresponding to evaporation and superheat of refrigerants, and there are two zones outside the tube due to unsaturated (dry) and saturated (wet) air. For simultaneous heat and mass transfer on the wet zone, the rate of heat gain due to mass transfer associated with the dehumidification is calculated using a wet or latent air-side convective heat transfer coefficient (HTC). The effect of air dehumidification is to increase the heat transfer. The driving potential for the simultaneous heat and mass transfer is taken as the temperature difference between air in the main stream and the saturated water film temperature at the outer surface of the coil.

Mass balance on air-side

In the dry zone, air is cooled at a constant humidity ratio.

$$W_{a2} = W_{a1} \qquad (4)$$

In the wet zone, air is cooled with decreasing humidity ratio and dry-bulb temperature.

$$\delta m_a W_{a2} = \delta m_a W_{a1} - \alpha_a^m (\delta A_p + \eta_f \delta A_e)(W_a - W_s) \qquad (5)$$

where W_a : average humidity ratio of air and W_s : saturated humidity ratio at the tube outer surface temperature.

Heat balance on air-side

In the wet zone, heat and mass transfer process occur simultaneously with the condensation of moisture. It is assumed that moisture condensed in each control volume is drained at the local coil surface temperature and the thermal resistance is neglected.

$$\delta m_a h_{a2} = \delta m_a h_{a1} - [\alpha_a^h (\delta A_p + \eta_f \delta A_e)(T_a - T_{s,a}) + C_{p,w} T_{s,o} \delta m_a (W_{a1} - W_{a2})] \qquad (6)$$

Heat balance on refrigerant-side

At steady state, the heat balance between the air side and the refrigerant inside the tube is:

$$\alpha_a^h (\delta A_p + \eta_f \delta A_e)(T_o - T_{s,o}) = \alpha_c \delta A_i (T_{s,o} - T_r) = \delta m_r (h_{r2} - h_{r1}) \qquad (7)$$

Where α_c is the HTC from the tube outer surface to the refrigerant (W/m^2K).

The outer surface temperature can be computed by:

$$T_{s,a} = \frac{\alpha_a^h (\delta A_p + \eta_f \delta A_e) T_o + \alpha_c \delta A_i T_r}{\alpha_a^h (\delta A_p + \eta_f \delta A_e) + \alpha_c \delta A_i} \qquad (8)$$

The overall HTC is expressed as:

$$\frac{1}{U_o A_o} = \frac{1}{\alpha_a^h (A_p + \eta_f A_c)} + \frac{d_a - d_i}{\pi (d_a + d_i) L k_t} + \frac{1}{A_i \alpha_r^h} \qquad (9)$$

Heat and mass transfer coefficient

In the calculation of HTC in the air- side, correlation presented by Fischer et al (1983) is used for the dry zone, and the HTC for wet zone is computed considering both the latent and sensible heat using Threlkeld (1970) correlation. The HTC on refrigerant-side depends on the flow conditions. For superheated single-phase refrigerant and the region of evaporating two-phase flow, correlation proposed by Perry et al (1973) and Jung et al (1991) respectively are used.

Computer simulation

Governing equations are numerically solved and applied to a set of coil geometric parameters given in Table 1. In this study inlet air temperature is kept constant, and the relative humidity, evaporating temperature and air flow velocity are varied. Effects of these parameters on the performance of evaporating coil are investigated. The elemental length of 5 mm was used in the simulation.

The simulation process begins with a given set of air inlet conditions i.e. temperature (Ta_i), relative humidity (R.H.), air face velocity (Va) and the refrigerant conditions i.e. evaporating temperature (Te), inlet subcooling and outlet superheating conditions. The total air mass flow rate is calculated from the face velocity and coil face area. Based on an estimated refrigerant mass flow rate, the governing equation together with the correlation for heat and mass transfer coefficients are applied on the control volume to calculate the air and refrigerant properties at the exit of the control volume. The first control volume is selected at the outlet of the refrigerant. The above computations are repeated for each successive control volumes until the inlet enthalpy of refrigerant converges.

RESULTS AND DISCUSSION

Fig. 2 shows the effect of air inlet R.H. on the evaporator performance. It is seen from the figure that the cooling load, air outlet temperature and the water condensing rate increase with increase in relative humidity, except when R.H. is below 40% where all the above parameters remain unchanged because there is zero condensation of moisture. It can also be observed that the increase in cooling load consists of rapidly increasing latent cooling load and slowly decreasing sensible cooling load. The total cooling load increases due to the availability of increased energy in high humidity air, and simultaneous heat and mass transfer occurring along the evaporator coil caused by condensation of moisture. Hence, a higher flow rate of refrigerant and power are required to meet the increased cooling load in a high humid environment.

Fig. 3 shows that the cooling load and the water condensing rate decrease with increase in the evaporating temperature. The cooling load reduces due to a lower temperature differences between the air and the refrigerant. The rate of fall of cooling load with the evaporating temperature is higher for high R.H., but the rate of fall of the water condensing rate remains approximately constant with R.H., except for low R.H. and higher evaporating temperature the rate of fall is zero. The rate of fall of water condensing rate is zero at low R.H. of 30%, for R.H. of 50% it is zero beyond 10°C, when this happens wall temperature of the evaporating coil is higher than the dew point. Also, the figure shows that the C.O.P. increases with evaporating temperature as expected.

Figs 4 and 5 show the effect of coil face velocity on the cooling load, air exit temperature and water condensing rate for different R.H. All three parameters remain constant at low R.H., and increase with increase in R.H. and the face velocity. The cooling load increases due to the increase of air mass flow rate intensifying the heat transfer on the air-side. Fig. 5 shows that the specific condensing rate, which is defined as the water condensing rate per unit mass of air flow rate, although increases with R.H., it is inversely proportional to the velocity. It shows that the cooling load at 25°C air temperature for a humid environment (90% R.H.) is about two times that in a low humid environment (50% R.H.)

CONCLUSIONS

The analytical model and computer simulation for a given configuration of an evaporating coil show that the relative humidity is a very significant parameter in determining the quality of air conditioning in a humid environment. A high humid environment will require much higher cooling load compared to a low humid environment, which is mainly the results of higher latent heat load due to condensation of moisture than sensible cooling. The study has identified that in a high humid environment, the lower cooling load will be obtained with low face velocities and higher evaporating temperatures, but the exit air temperature will be higher.

The approach proposed in this study gives a method to analyze the parameters and to select the best operating points in a humid environment for a given geometry of the evaporating coil. Extending the proposed approach to different geometry of coils and optimization of parameters would give a better tool to design the coils in a humid environment, which is being pursued.

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Geometric Parameters	Values
Number of tube rows in air flow direction, N_r	4
Number of parallel refrigerant circuits, N_p	20
Length of stright tube, L	1000 mm
Transverse tube spacing, S_T	25 mm
Longitudinal tube spacing, S_L	25 mm
Outer tube diameter, d_o	9.53 mm
Inner tube diameter, d_i	7.78 mm
Fin thickness, 2*y	0.12 mm
Fin density, fins/meter	391
Fin type	flat plate
Coil frontal area	0.5 m^2

Table 1 Evaporator coil geometric parameters

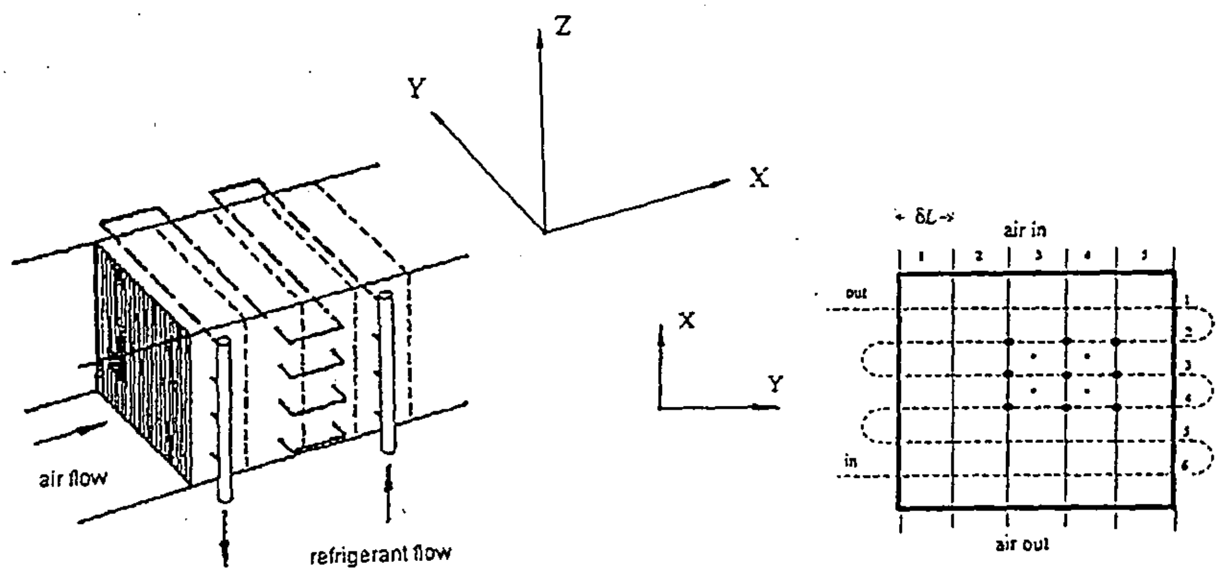

Fig 1a The arrangement of a single parallel circuit

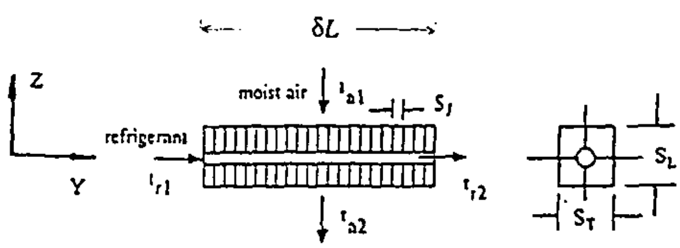

Fig 1b An element along a tube pass with fins

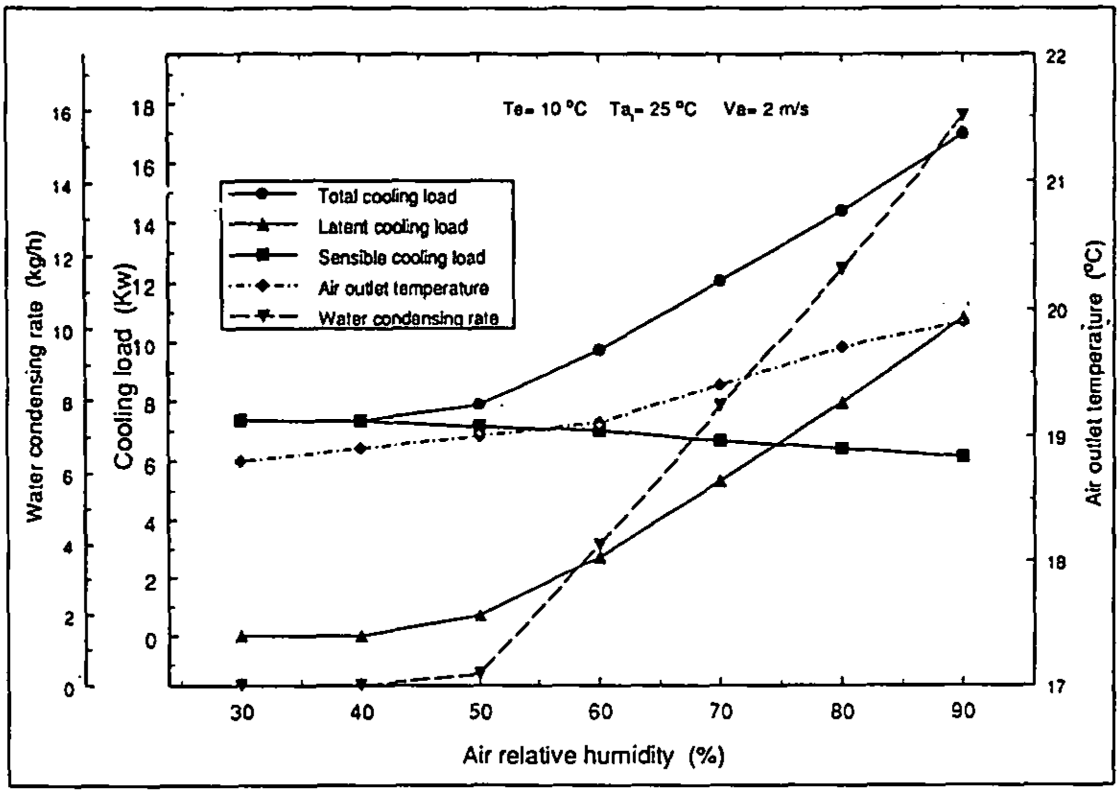

Fig 2 Effect of air inlet relative humidity on coil performance

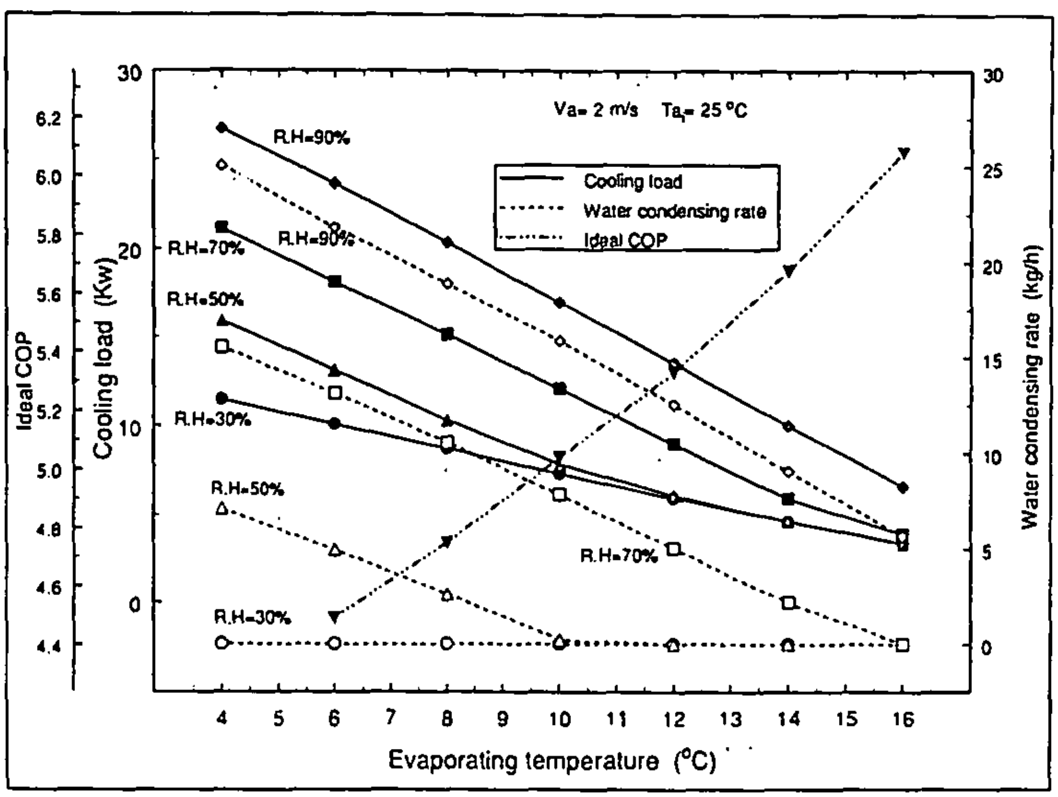

Fig 3 Effect of evaporating temperature on coil performance

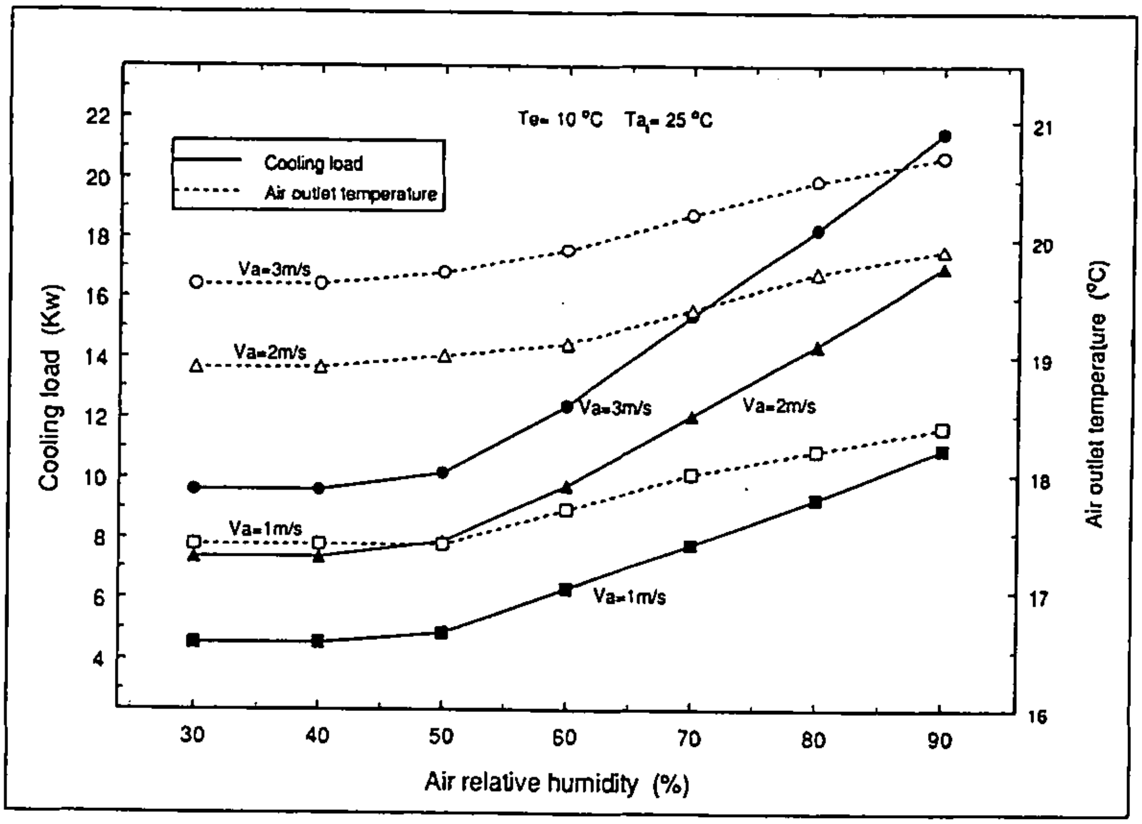

Fig 4 Effect of coil face velocity on coil performance

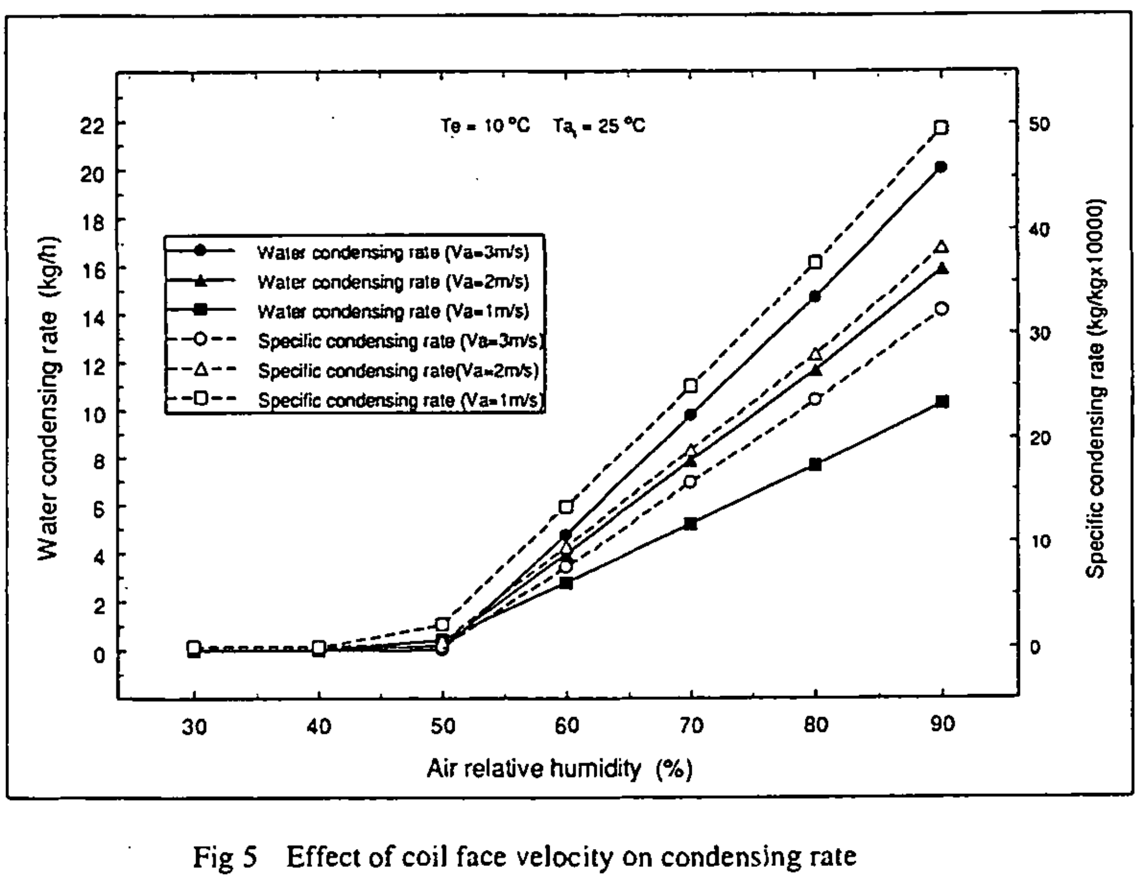

Fig 5 Effect of coil face velocity on condensing rate