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Josua P. Meyer[a]; Gideon P. Greyvenstein[a]
[a] Department of Mechanical Engineering, Potchefstroom University for CHE, Potchefstroom, South Africa

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The Influence of Height Above Sea Level on the COP of Air-Source Heat Pumps Used for Water Heating

JOSUA P. MEYER and GIDEON P. GREYVENSTEIN

Department of Mechanical Engineering, Potchefstroom University for CHE, Private Bag X6001, Potchefstroom, 2520, South Africa

In some countries (e.g., South Africa), important market areas for air-source heat pumps are situated high above sea level, and it is therefore necessary to know what the influence of height is on the performance of heat pumps for water heating. Therefore, a study was conducted that predicts the influence of height on the coefficient of performance (COP) of air-source heat pumps. An expression was derived that gives the heat transfer to the evaporator of a heat pump as a function of air density and enthalpy as well as the outside heat transfer coefficient. This expression was used to predict the influence of height above sea level on COP. The results were verified by means of a simulation study. It was concluded that height has little influence on COP if the evaporator of the heat pump is wet on the air side. A more significant weakening in the performance occurs, however, when the air side of the evaporator is dry.

INTRODUCTION

Hot water for residential building in South Africa is heated mostly by means of direct electrical resistance heaters. This is so because the burning of natural gas or coal is not a viable alternative to direct electrical heating of water. South Africa does not have sources of natural gas near its most populous areas. The burning of coal is inconvenient, and it also poses a pollution hazard. An attractive alternative to direct electrical heating is to heat water by means of air-source heat pumps. Meyer and Greyvenstein [1] have recently shown that heat pumps are in many instances more economical than direct electrical heating. On the basis of this finding, it was concluded that a significant market exists for water-heating heat pumps in South Africa. Despite their economic advantage, heat pumps are not commonly used in South Africa.

The financial contribution made by the National Energy Council made this study possible and is hereby gratefully acknowledged.

The reasons are the following. Heat pumps are relatively unknown in South Africa, and the principles on which they operate are not as easily understandable as other methods of heating. Heat pumps were only recently marketed in South Africa for the first time, and a lack of technoeconomic information exists [2].

The basic heat pump cycle is identical to the vapor-compression refrigeration cycle, i.e., saturated vapor at low pressure and temperature enters the compressor and undergoes adiabatic compression. The hot, high-pressure vapor condenses in the condenser at a constant pressure while giving off an amount of heat to the surroundings, which are at a lower temperature. The refrigerant leaves the condenser as saturated liquid. An adiabatic throttling process follows, and the refrigerant is then evaporated at constant pressure to complete the cycle. The basic heat pump cycle is identical to the refrigeration cycle, the only difference between a heat pump and a refrigerator being their basic function. The refrigeration system cools the external fluid flowing through the evaporator, whereas in a heat pump the external fluid flowing through the condenser is heated. For the heating of hot water with a heat pump, the condenser heats the water; while air is used as a source which is cooled by the evaporator.

The heat transfer in the evaporator on an air-source heat pump is a function of the psychometric condition of air, which in its turn is a function of the height above sea level. It can therefore be expected that the performance of air-source heat pumps is influenced by the height above sea level. Seeing that important marketing areas may be situated high above sea level (i.e., the most important marketing areas in South Africa, namely, Johannesburg and Pretoria, are approximately 1500 m above sea level), it is important to determine to what extent performance is influenced by height above sea level. If the performance is known, technoeconomic information can be compiled that can be used to determine the life-cycle cost of heat pumps.

The aim of this article is to determine the influence of height above sea level on the coefficient of performance (COP) of air-source heat pumps. This is done by deriving an expression for the heat transfer in the evaporator as a function of the density and enthalpy of air as well as the outside convection heat transfer coefficient. By determining what influence the height above sea level has on these variables the expression for heat transfer can be used to determine to what extent the heat transfer in the evaporator and thus also the performance are affected by height above sea level. The foregoing influence of height on the coefficient of performance of air-source heat pumps is then verified by means of a simulation model.

HEAT TRANSFER IN THE EVAPORATOR

The change in psychometric condition of air through the evaporator of a heat pump is shown in Fig. 1. The air is cooled from a dry-bulb temperature t_1 to t_2. At the same time the humidity ratio lowers from w_1 to w_2 as a result of the condensation of moisture on the outside surface of the evaporator. The extension of the line that links points 1 and 2 bisects the 100% relative humidity line at point A, which is the apparatus dew-point temperature t_A. The apparatus dew-point temperature is approximately equal to the mean coil (or fin) surface temperature, t^*, of the evaporator.

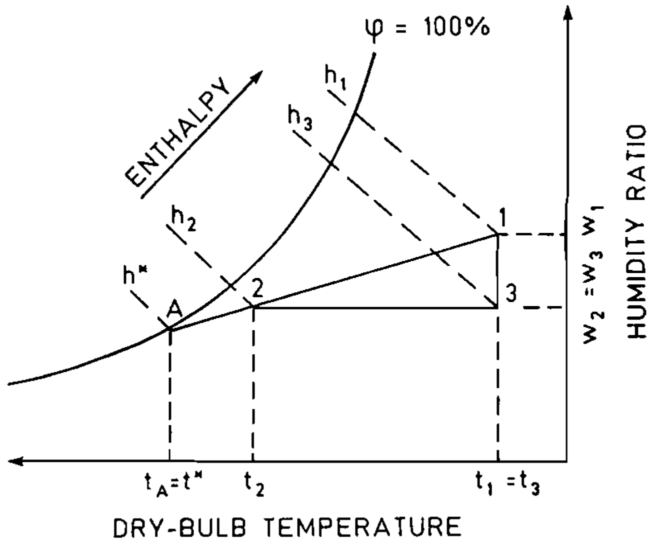

Figure 1 Change in psychometric condition of air through the evaporator of a heat pump.

The coil ratio of the evaporator is defined as the ratio of the sensible heat removed by a coil to the total heat removed by a coil:

$$S = \frac{h_3 - h_2}{h_1 - h_2} \tag{1}$$

The sensible heat transfer in Eq. (1) can be described as a function of the outside area of the evaporator and the convection heat transfer coefficient on the outside of the evaporator [3]:

$$E_w = \frac{A\alpha(t_1 - t_2)}{\ln [(t_1 - t_2^*)/(t_2 - t^*)]} \tag{2}$$

The sensible heat can also be given in terms of the following equation [4]:

$$E_w = \dot{m}c(t_1 - t_2) \tag{3}$$

By equating Eqs. (2) and (3) and by further simplification, it follows that

$$\frac{t_1 - t^*}{t_2 - t^*} = e^{A\alpha/\dot{m}c} \tag{4}$$

The temperature ratio on the left-hand side of Eq. (4) is by approximation equal to the same enthalpy ratio (see Fig. 1),

$$\frac{h_1 - h^*}{h_2 - h^*} = e^{A\alpha/\dot{m}c} \tag{5}$$

From the above equation it follows that

$$h^* = \frac{h_1 - h_2 e^{A\alpha/\dot{m}c}}{1 - e^{A\alpha/\dot{m}c}} \tag{6}$$

The total heat transfer is given by

$$\dot{E} = \dot{m}(h_1 - h_2) \tag{7}$$

By substitution of h_2 from Eq. (7) in (6) and by replacing the mass flow rate of the air through the evaporator with the product of air fluid density and the air volume flow rate, it can be stated that

$$h^* = h_1 - \frac{E/\rho Q}{1 - e^{-A\alpha/\rho Qc}} \tag{8}$$

The mean fin temperature of the evaporator t^* is simply a function of h^* and the height above sea level. It can be determined by using psychometric relations obtained from basic psychometric formulas and constants taken from well-known handbooks [5, 6], or it may be read from psychometric charts for moist air for any barometric pressure, or it can be calculated by computer programs (i.e., PSYCHART) [7].

It can be concluded from Eq. (8) that the mean fin temperature is influenced by air intake conditions, the volume flow rate through the evaporator, and the convection heat transfer coefficient. These variables are discussed subsequently.

EVAPORATION TEMPERATURE AS A FUNCTION OF AIR INTAKE CONDITION AT A GIVEN HEIGHT

For a given evaporator at a given height with a constant total heat transfer E, the extreme right term in Eq. (8) remains constant. This means that the enthalpy difference $(h_1 - h^*)$ will also remain constant. This fact can be used to determine to what extent the evaporation temperature is influenced by the air inlet conditions.

In Fig. 2, two lines of constant enthalpy h^* and h_1 are presented on a psychometric chart. Point B is the intersection of the 100% relative humidity line and the h^* line, while point A is situated on the h_1 line. The moisture content at A is the same as at B. For inlet conditions to the left of point A on the h_1 line—for example, conditions C and D—the mean fin temperature remains constant equal to t_B. In these cases there is a reduction of moisture content, which means that condensation occurs on the outside surface of the evaporator. Lines of constant enthalpy coincide approximately with lines of constant wet-bulb temperature, which means that in cases where condensation occurs on the outside of the evaporator, the evaporation temperature, which is approximately a constant increment lower than t^*, is a function only of the wet-bulb temperature of the inlet air.

Should the inlet condition of point A on line h_1, for example, be at point E, the outlet condition on line h^* at point F will be of the same moisture content as point E. This means that the mean fin temperature is determined by the dry-bulb temperature of the inlet air. In cases where no condensation occurs on the outside surface of the evaporator, the evaporation temperature is largely a function of the dry-bulb temperature of the inlet air.

INFLUENCE OF HEIGHT ABOVE SEA LEVEL ON THE VOLUME FLOW RATE THROUGH THE EVAPORATOR

The air volume flow rate through the evaporator is determined by the fan curve and the system resistance of the evaporator. The pressure increase through the fan as well as the pressure loss through the evaporator is directly proportional to the density, which means that the volume flow rate, Q, is not influenced by density and therefore also not by height above sea level.

INFLUENCE OF HEIGHT ABOVE SEA LEVEL ON THE CONVECTION HEAT TRANSFER COEFFICIENT

For a compact heat exchanger, the product of the Stanton and Prandtl numbers is a function of the Rey-

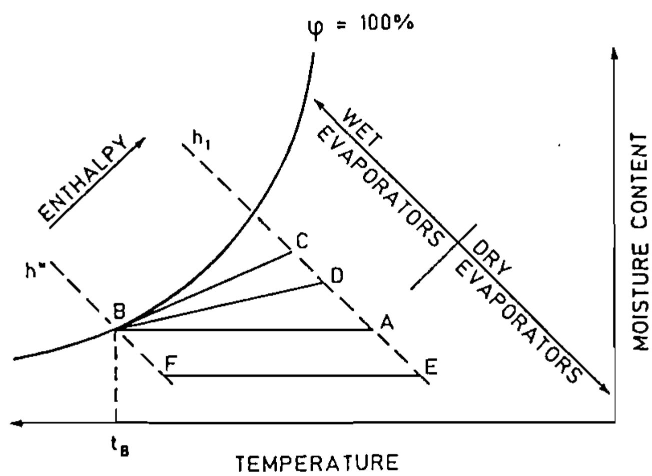

Figure 2 Mean fin temperature as a function of air inlet condition.

nolds number; that is,

$$\mathrm{St}\,\mathrm{Pr}^{2/3} = f(\mathrm{Re}) \tag{9}$$

where the Stanton number is

$$\mathrm{St} = \frac{\alpha}{\rho c V} \tag{10}$$

and the Prandtl number is

$$\mathrm{Pr} = \frac{c\mu}{k} \tag{11}$$

and finally the Reynolds number is

$$\mathrm{Re} = \frac{\rho V D}{\mu} \tag{12}$$

Should one substitute Eqs. (10), (11), and (12) in (9) and should it be accepted that all variables except α and ρ remain constant, it can be written that

$$a_1 \frac{\alpha}{\rho} = f(a_2, \rho) \tag{13}$$

where a_1 and a_2 are constants.

For a round, tube-type, continuing-fin heat exchanger (i.e., one with a tube outside diameter of 10.2 mm, a fin pitch of 315 per meter, flow passage hydraulic diameter of 3.63 mm, free-flow area/frontal area ration of 0.534, heat transfer area/total volume of 587 m^2/m^3, fin area/total are of 0.913, and a fin thickness of 0.33 mm), it is known that

$$\alpha = a_3 \rho^{0.604} \tag{14}$$

The constant a_3 is a function of ρ, c, V, Pr, and Re. Should the following typical values be accepted, ρ = 1.22 kg/m^3, c = 1.006 kJ/kg K, V = 3.4 m/s, Pr = 0.708, Re = 800 and $\mu = 1.8 \times 10^{-5}$ kg/m s, it follows from data of the Trane Company (compiled in [8]) and Eq. (9) that α = 57.8 $W/m^2\,°C$. From Eq. (14), it then follows that a_3 = 50.88.

INFLUENCE OF HEIGHT ABOVE SEA LEVEL ON THE MEAN FIN TEMPERATURE

The mean fin temperature has been calculated by using Eqs. (8) and (14) as well as from psychometric relations. The proportions E/Q and $A/(cQ)$ used in Eq. (8) were taken as mean values for a series of 12 air-to-water heat pumps marketed in South Africa. The series ranges from units with a heating capacity of 3 kW to units with a heating capacity of 80 kW. The mean values are as follows: E/Q = 5300 W s/m^3 and $A/(cQ)$ = 0.017 kg K/W m. Substitution of these values as well as Eq. (14) in Eq. (8) leads to

$$h^* = h_1 - \frac{5300/\rho}{1 - e^{-0.865/\rho^{0.396}}} \tag{15}$$

The procedure for the calculation of the mean fin temperatures is as follows: Accept values for height above sea level and inlet wet-bulb temperature; with these values known, use psychometric formulas to determine ρ and h_1; calculate h^* with the aid of Eq. (15) and calculate t^* with the aid of psychometric relations.

Figure 3 indicates the evaporator mean fin temperature as a function of the inlet wet-bulb temperature and height above sea level for cases where the evaporator fin temperature is smaller than the dew point, that is, for cases where moisture condenses on the outside surface of the evaporator. Figure 4 shows the evaporator mean fin temperature as a function of the inlet dry-bulb temperature and height above sea level in cases where the fin temperature is greater than the dew point of the inlet air, that is for cases where no condensation occurs on the outside of the evaporator.

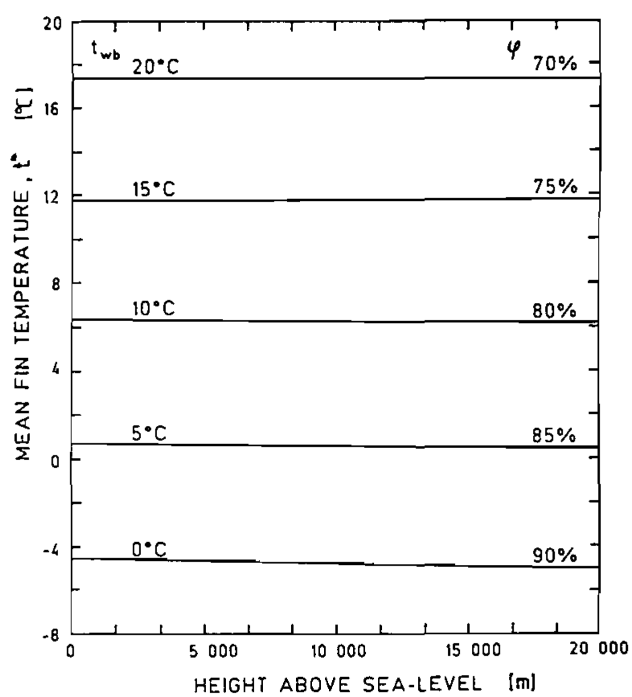

Figure 3 Evaporator mean fin temperature, t^*, as a function of height above sea level and inlet wet-bulb temperature for cases where condensation occurs on the outside of the evaporator.

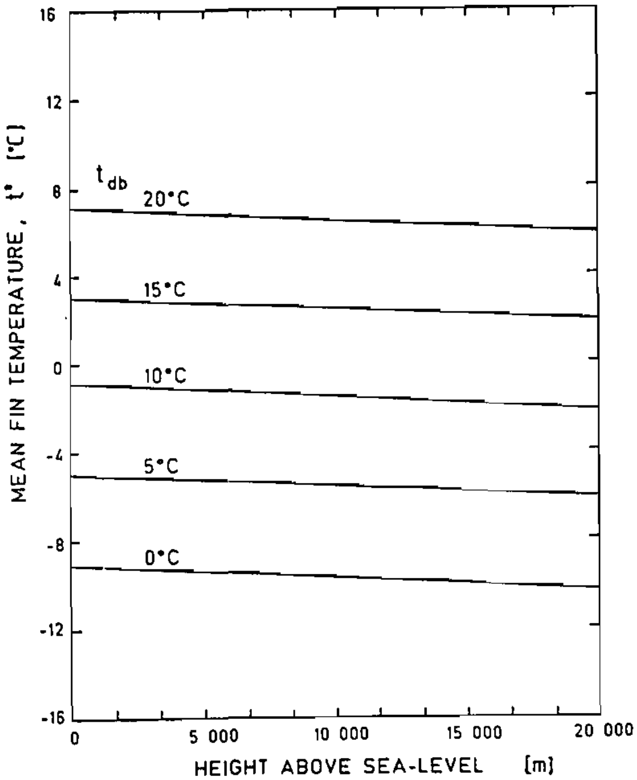

Figure 4 Evaporator mean fin temperature, t^*, as a function of height above sea level and inlet wet-bulb temperature at a relative humidity of 30% for cases where no condensation occurs on the outside of the evaporator.

From Fig. 3 it can be concluded that in the case of wet evaporators, height above sea level has almost no influence on the evaporator fin temperature at high wet-bulb temperatures. At low wet-bulb temperatures, an increase in height causes a small decrease in t^*. From Fig. 4 it can be deduced that in the case of dry evaporators, an increase in height causes a decrease in the fin temperature. The extent to which the fin temperature is influenced by height above sea level is not a strong function of the inlet dry-bulb temperature.

Since temperatures less than zero occur in South Africa only occasionally, ice formation for $t^* < 0\,°C$ is not considered in this study.

INFLUENCE OF HEIGHT ABOVE SEA LEVEL ON THE PERFORMANCE OF HEAT PUMPS

In the preceding paragraphs the mean fin temperature of the evaporator was calculated as a function of the air inlet conditions and height above sea level. Seeing that the evaporation temperature is almost a constant increment lower than the mean fin temperature, the evaporator temperature is influenced to the same extent as t^* by height above sea level and air inlet conditions.

It should be kept in mind that the heat output at the condenser and coefficient of performance of an air-source heat pump decrease with a decrease in evaporation temperature. Then the following two conclusions can be drawn: First, in the case of wet evaporators, the heat output and coefficient of performance are strong functions of the wet-bulb temperature of the inlet air. Performance increases with an increase in wet-bulb temperature and vice versa. At high wet-bulb temperatures, height above sea level has almost no influence on performance. At lower wet-bulb temperatures, the influence of height above sea level increases, but is never very great. An increase in height causes a small decrease in heat output and coefficient of performance.

Second, in the case of dry evaporators, the heat output and coefficient of performance are strong functions of the dry-bulb temperature of the inlet air. The performance is influenced to a lesser extent by the height above sea level. An increase in height causes decreases in heat output and coefficient of performance. The extent to which the coefficient of performance is influenced by height above sea level is not a strong function of the inlet dry-bulb temperature.

SIMULATION STUDY

The effect of height above sea level on the performance of air-source heat pumps was also studied with the aid of a simulation model discussed in [2] and [9]. The model describes the performance of a vapor-compression heat pump and refrigeration system with a thermostatically controlled expansion valve, giving its design, the fan curve, the compressor characteristics, the design of the heat exchangers, and the condition of the external fluids through the heat exchangers.

The coefficient of performance and heat output of an example system were calculated for a series of conditions at sea level and for a height of 1500 m. The calculations were done in cases where moisture is withdrawn from the air (wet evaporators) and where no moisture was withdrawn from the air (dry evaporators). The results of the different cases are given in Figs. 5 and 6.

The example system considered here consists of a double-tube counterflow-type condenser with a total length of 12 m. The outside diameter of the inner tube is 6.35 mm and the outside diameter of the outer tube is 12.7 mm. The wall thickness of both tubes is 0.7 mm. The evaporator is a compact fin-and-tube

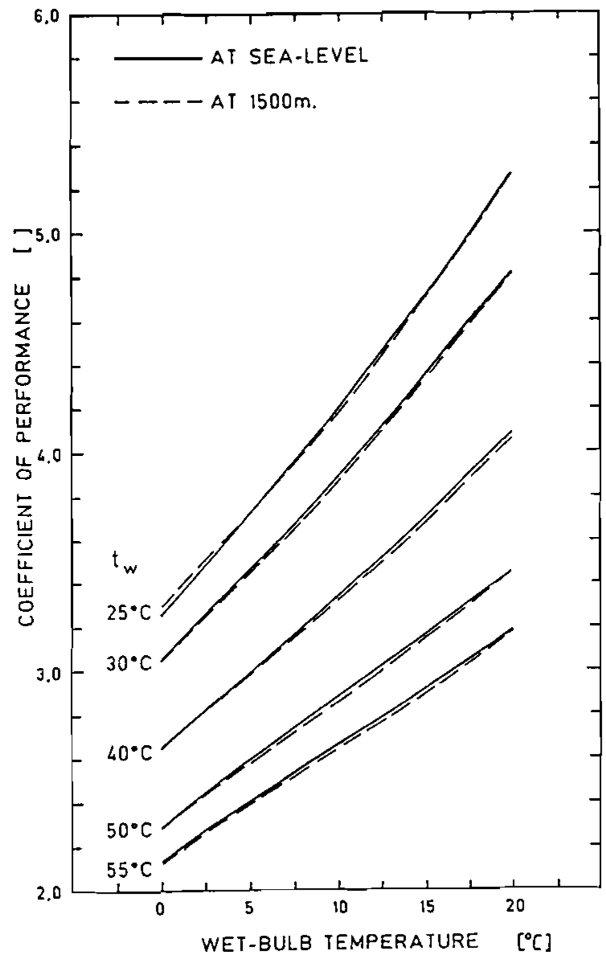

Figure 5 The coefficient of performance as a function of wet-bulb temperature and water outlet temperature for a wet evaporator.

heat exchanger with both a height and width of 406.4 mm. The distance between the tube columns is 25.4 mm and the distance between the rows 22 mm. Two rows were used. The outside diameter of the tubes is 10.02 mm and the wall thickness is 0.4 mm.

The compressor is a JRE4-0075 PAV hermetically sealed R22 unit manufactured by Copeland with a nominal power consumption of 560 W. The performance characteristics (experimental data obtained from the manufacturer) are given in [2]. An axial flow fan with a diameter of 300 mm is used, and the fan curve is also given in [2]. The equivalent length and inside diameter of the connection pipe between the compressor and the condenser are 1.5 m and 4.95 mm, respectively, while the one between the evaporator and the compressor has an equivalent length of 2.0 m and an inside diameter of 8.13 mm. The system has a thermostatically controlled expansion valve with a superheat setting of 8 °C.

Simulations were done for cases where moisture was withdrawn from the air (wet evaporators) and where no moisture was withdrawn from the air (dry evaporators). The results of the simulations are presented in Figs. 5 and 6. From Fig. 5 it can be deduced that in the case of wet evaporators, height above sea level has a very small influence on the coefficient of performance of air-source heat pumps. The weakening of the coefficient of performance with an increase in height of 1500 m is typically smaller than 1% for all conditions. In the case of dry evaporators (Fig. 6), the weakening in performance is more significant, that is, more than 3.6% for all conditions with an increase in height of 1500 m. It can therefore be concluded that the results of this simulation study are in accordance with the conclusions drawn from the foregoing section.

CONCLUSIONS

If condensation occurs on the air side of an evaporator, the evaporation temperature is a function only

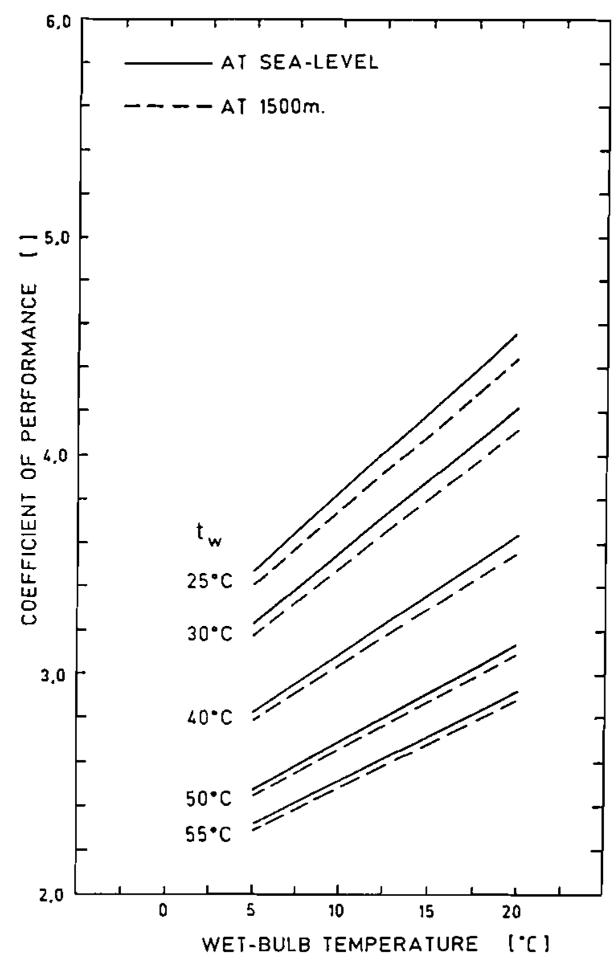

Figure 6 The coefficient of performance as a function of wet-bulb temperature and water outlet temperature for a dry evaporator.

of the wet-bulb temperature of the inlet air, while for cases where no condensation occurs, the evaporation temperature is largely a function of the dry-bulb temperature of the inlet air. This leads to a very small influence on the coefficient of performance with height for air-source heat pumps with wet evaporators. A more significant weakening in the coefficient of performance occurs, however, when the evaporator is dry.

NOMENCLATURE

a_1, a_2, a_3	constants
A	evaporator outside area
c	humid specific heat of air stream
D	hydraulic diameter
E	total heat transfer
E_w	sensible heat transfer
f	function
h	enthalpy
k	conduction coefficient
$\dot{m}$	mass flow rate
Q	volume flow rate
S	coil ratio
t	temperature (dry-bulb)
t_A	apparatus dew-point temperature
t_w	water outlet temperature
V	velocity
w	humidity ratio
α	convection heat transfer coefficient
μ	viscosity
ρ	density
ϕ	relative humidity

Subscripts

1	evaporator inlet
2	evaporator outlet
3	refer to Fig. 1
wb	wet bulb
db	dry bulb

Superscripts

*	evaporator mean value on the air-side surface

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Josua Petrus Meyer was born in Pretoria, South Africa, and obtained the B.Eng. (Mech) degree with distinction at the University of Pretoria in 1984. He was employed as an Associate at the Laboratory for Advanced Engineering from January 1984 to January 1988, during which time he obtained the M.Eng. degree (with distinction) in 1986 and the Ph.D. degree in 1988. He completed National Service from February 1988 to December 1989 at the Military Academy, Saldanha, and then was appointed to the Potchefstroom University where he is currently a Professor in Mechanical Engineering.

Gideon P. Greyvenstein was born in Hofmeyr, South Africa, and matriculated at the Hendrik Verwoerd High School in Pretoria in 1969. After obtaining a B.Sc.Eng (Mechanical) degree at the University of Pretoria in 1974, he joined industry for seven years, where he worked in heat transfer, fluid mechanics, computational fluid mechanics, and manufacturing. At the same time he commenced part-time studies and obtained a Ph.D. in computational fluid mechanics from the University of Pretoria in 1981 and a Masters degree in Business Leadership from the University of South Africa in 1987. At the beginning of 1983 he joined the Potchefstroom University for Christian Higher Education as Professor in Mechanical Engineering. His present position is Head of the Department of Mechanical Engineering and Vice-Dean of the Faculty of Engineering. He is actively involved in heat pump, refrigeration system, and computational fluid dynamics research.