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The Influence of the Humidity on Thermal Comfort, Heat Load Calculation and Cooling Capacity

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1. Heat transmission of the human body

The human heat transmission is done by convection, radiation and by evaporation of water to the environment. This physical transmissions cause the following six parameters of thermal comfort:

- activity level
- clothing
- air temperature
- air humidity
- air velocity
- wall temperature

The different heat transmission mechanism take over different parts of the total heat load. The ratio are depending on various parameters. With rising air-temperature the convection is decreasing meanwhile the latent heat by evaporation is increasing. In fig. 1 the influence of the activity level on the different ratios is shown. The total heat losses with an activity level related to 120 W is fairly constant over a wide range of temperature. But the ratio between the latent heat and the sensible heat is very different at various air temperatures.

For a higher activity level related to 250 W or to 350 W shows the same tendency. The sensible heat is always the sum of convection an radiation.

In fig. 2 the different heat transfer parts are shown depending on the total heat loss and the air-temperature. It can be seen very clearly, that at a room air-temperature of 33°C the radiation and the convection is going to be zero. At higher temperatures the evaporation must take over the incoming convection and radiation. That means the latent heat is higher than the heat production in the human body. In the open space or also in cars the incoming solar radiation must be considered already at lower room temperatures because the incoming radiation must be bilanced by additional evaporation.

Because the parameters of the heat convection are very stable because of the constant body temperature and a fixed clothing the control mechanism of the temperature by changing the heat losses can only be done in the latent heat ratio. The rise of the temperature of the body surface can only be shifted in very small limits and the room air temperature and the air-velocity can not be ajusted individually. A change in the radiation is also not possible at a fixed wall temperature and given room configuration. This shows clearly that whole control must be done by variable evaporation.

Out of this considerations we see that the only possible ajustment of different activity levels for the occupants in a room can only be done by variation of the evaporating water. This means that a fairly low relativ humidity is necessary that the different bodies can find their own thermal comfort.

2. Humidity and comfort

The first investigations about the influence of the humidity of thermal comfort have been carried out by Samuel Lewis (3). His diagram is shown in figure 3.

As a result of the work by Lewis a diagram of humidity vs. temperature was established. Figure 4 shows the comfort zone which was used in the sixties and seventies.

The investigations of O. Fanger (5) about the thermal comfort in office buildings shows a much smaller influence of the humidity. The reason of this results is the very small change of activity level, a very simular clothing and a fairly stable air temperature. As shown in figure 1 and 2 the activity level is of great influence. It is not possible, therefore, to use the values for office buildings in a much broader scale. In figure 5 the optimal conditions as shown in figure 3 are compared with the comfort zone of DIN 1946 part 2. This results can also be shown by experiment where we found that all test persons believe 26°C, 40% as definetly cooler than 24°C, 60%. This shows that the influence of temperature can not be discussed without humidity.

3. Calculation of cooling load

The correct equation for this calculation is

$$\dot{Q} = \dot{M}_L \cdot (h_{AL} - h_{ZL}) \qquad (1)$$

By using the enthalpy h the humidity is automatically included. Very often the equations only use the temperature differences which is not correct.

Another very important mistake is done by using the exhaust air temperature as mean room temperature. An optimal design of an air conditioning plant use a higher exhaust temperature. (See figure 6) A ratio of enthalpie differences shows the effectivness of an air conditioning plant (2).

$$\eta_a = \frac{h_{AB} - h_{AZ}}{h_{AB} - h_{ZU}} \tag{2}$$

This ratio is as higher as more heat is going to the air flow after leaving the direct environment of the occupants. Out of this consideration it is clear that the calculation of the cooling load must also consider the design of the air flow.

4. Calculation of the refrigeration capacity

The cooling load is not enough to discribe the refrigeration capacity because it is also necessary to consider the air changes. The air change rate cause especially in summertime a different dehumidification load which effects the refrigeration capacity. The minimum air changes are influenced strongly by the material which is used in the interial design. This can be shown in figure 7 (7). This example shows how great the influence of the material can be to the total energy consumption offer building. This influence is somewhat higher than the influence of the insulation.

In summer everybody is speaking about cooling in air conditioning plants when he is going to degreese the air temperature. For the dehumidification we have to calculate in central europe an enthalpie difference of about 25 kJ/kg dry air. It is very important to see that the highest enthalpie is not in the area of the highest temperature. Figure 8 shows the areas where temperature degreese is important (zone A) and where the dehumidification is necessary (zone B). In this figure 8 the area 1 shows 90% of all outdoor air conditions in Germany and area 2 shows 99% of the outdoor air conditions. The refrigeration capacity is therefore influence by the outdoor air humidity more evidently than by the outdoor air temperature.

5. Conclusion

This paper shows the great influence of the humidity on comfort, cooling load and refrigeration capacity. A lot of knowledge is available since a long time. Very often simple calculation methods do not use this knowledge. The computer programs of nowadays allow the consideration of humidity without any problems.

Literature

/1/ Schweizer Kühllastregeln, 1969, Schweizer Verein für Heizung und Lüftung

/2/ Steimle, F., Klimakursus, C.F. Müller-Verlag Karlsruhe, 1969

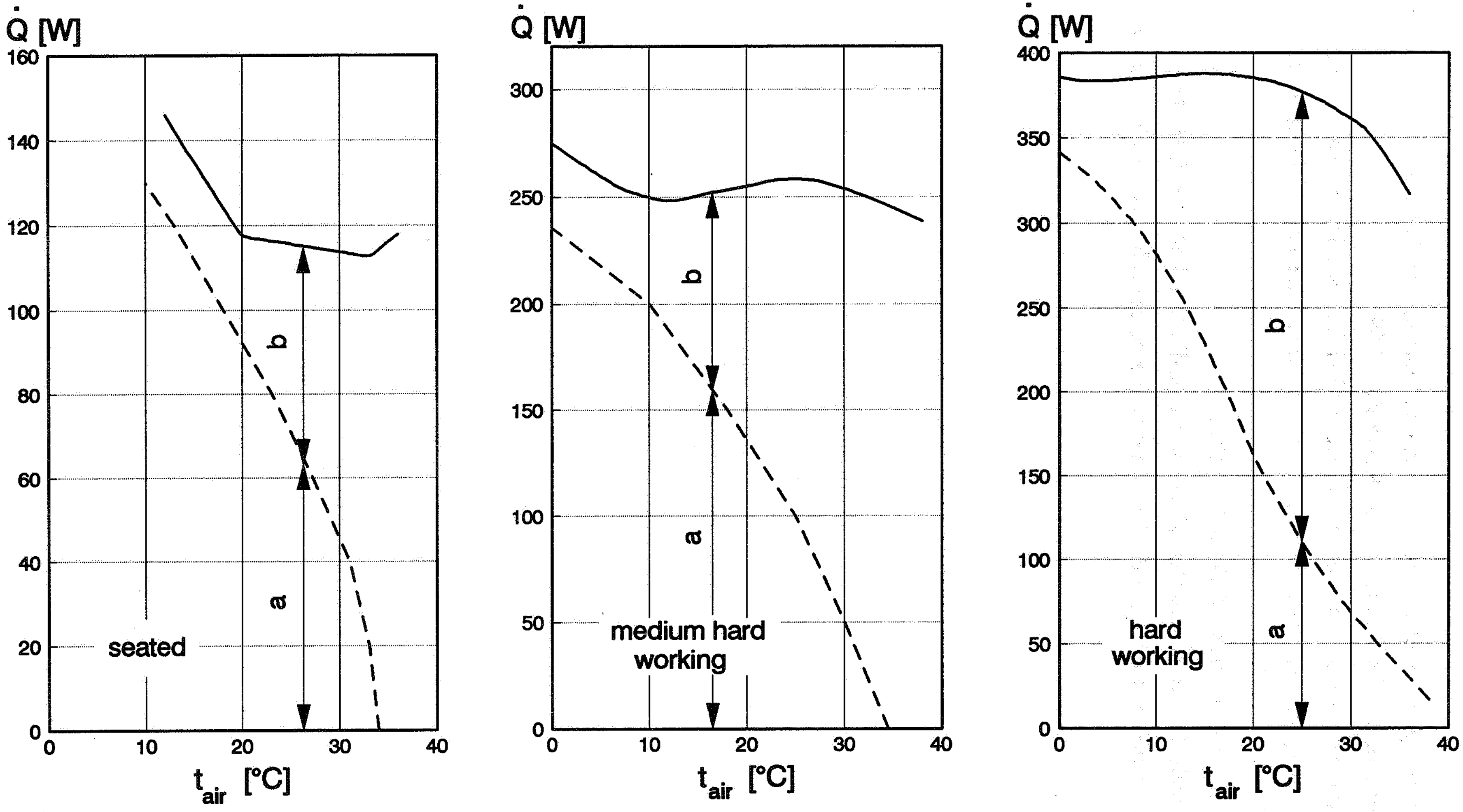
/3/ Lewis, S.R., Air conditioning for Comfort Engineering Publications, Inc, Chicago 1932

/4/ Steimle, F., Spegele, H., Die Behaglichkeit in klimatisierten Räumen, Kältetechnik-Klimakursus 22 (1970) S. 81/82

/5/ Fanger, O., On thermal comfort, McGraw Hill, 1972

/6/ Steimle, F., Spegele, H., Klima und Behaglichkeit, Klima-Kältetechnik, 1973, H 4/5

/7/ Steimle, F., Internationale Konferenz der IEA über neue Energietechniken, April 1992, Dortmund



a: sensible heat b: latent heat

Heat-transmission of persons in normal clothing

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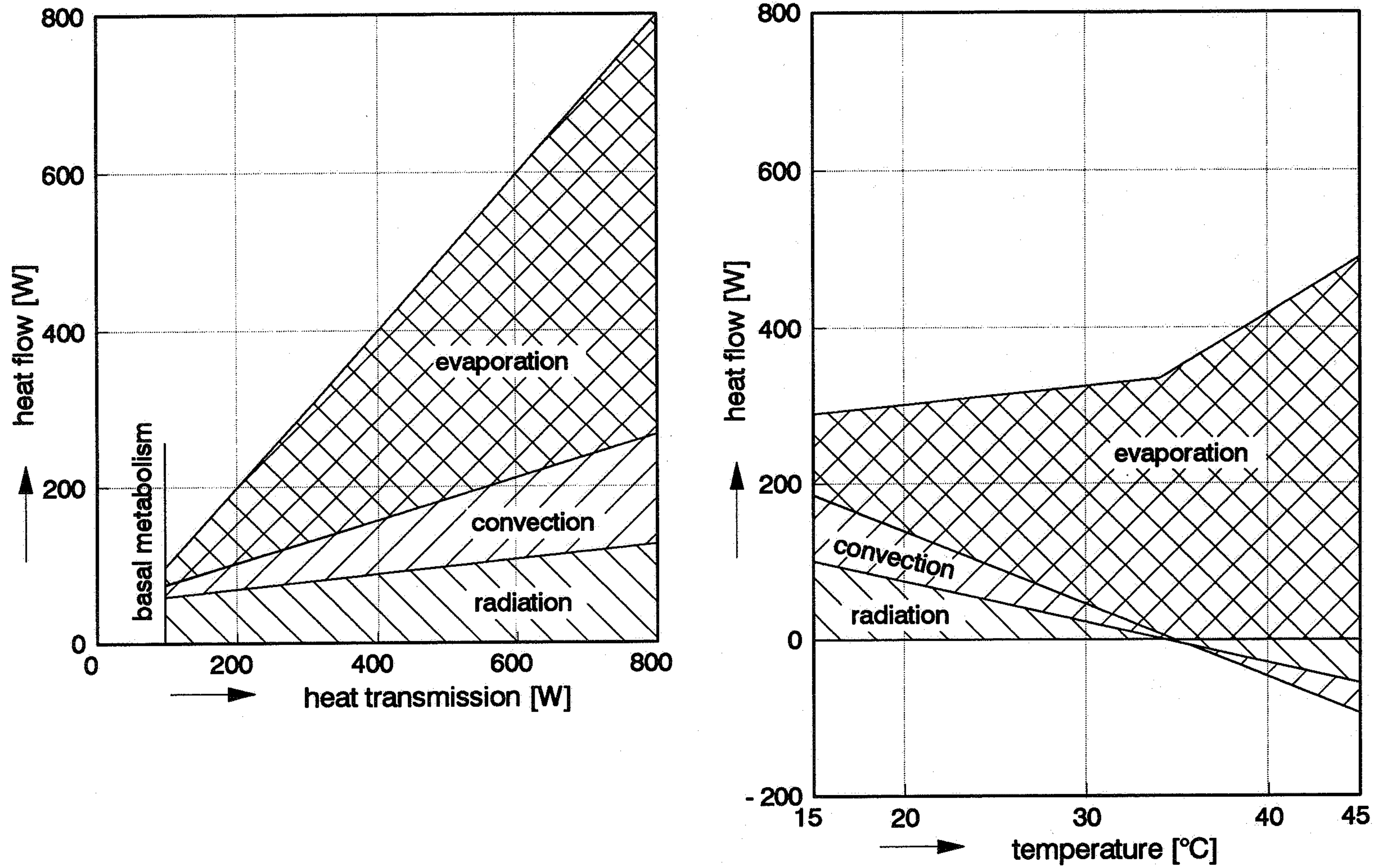

Heat trasmission by the human body

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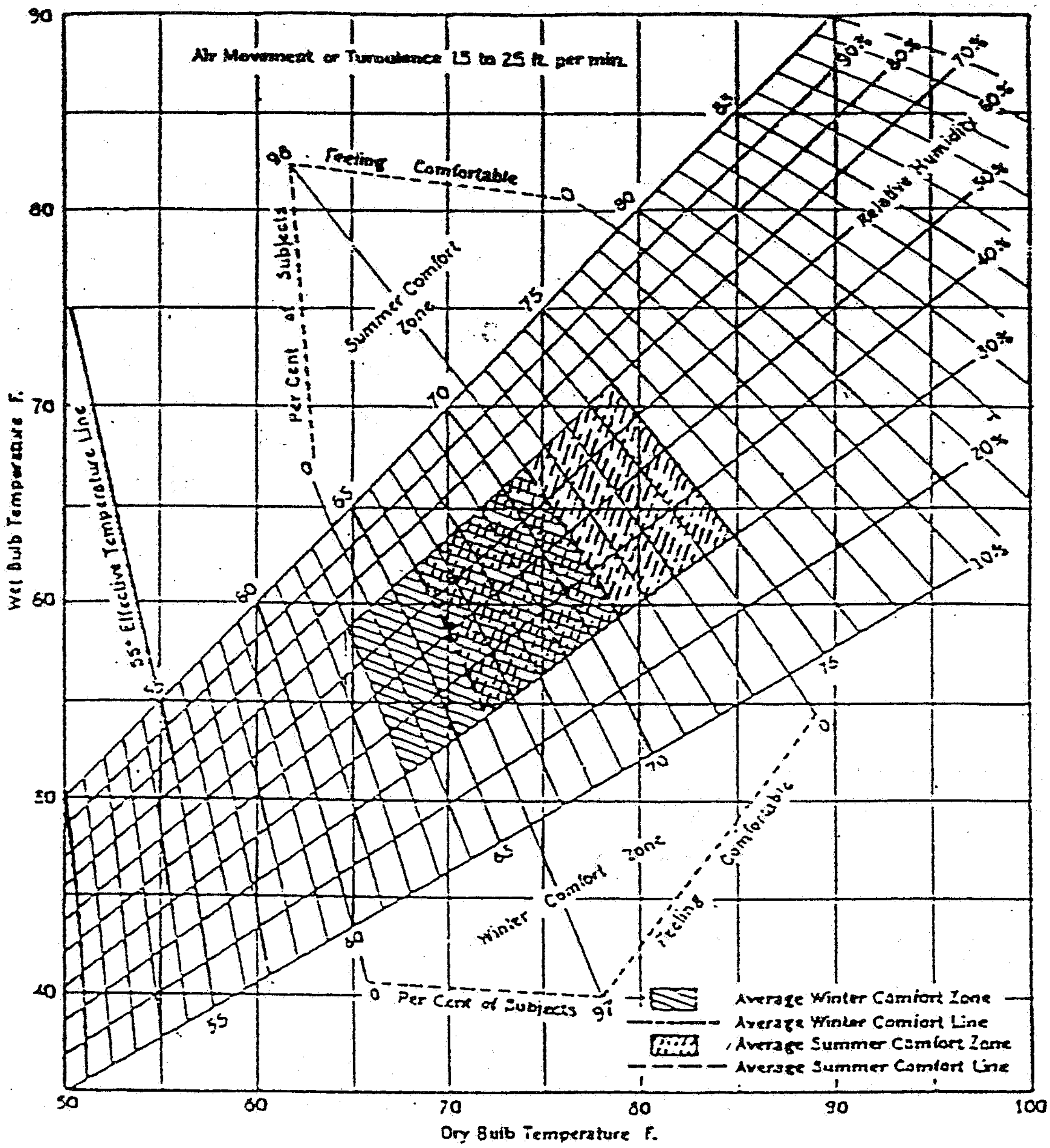


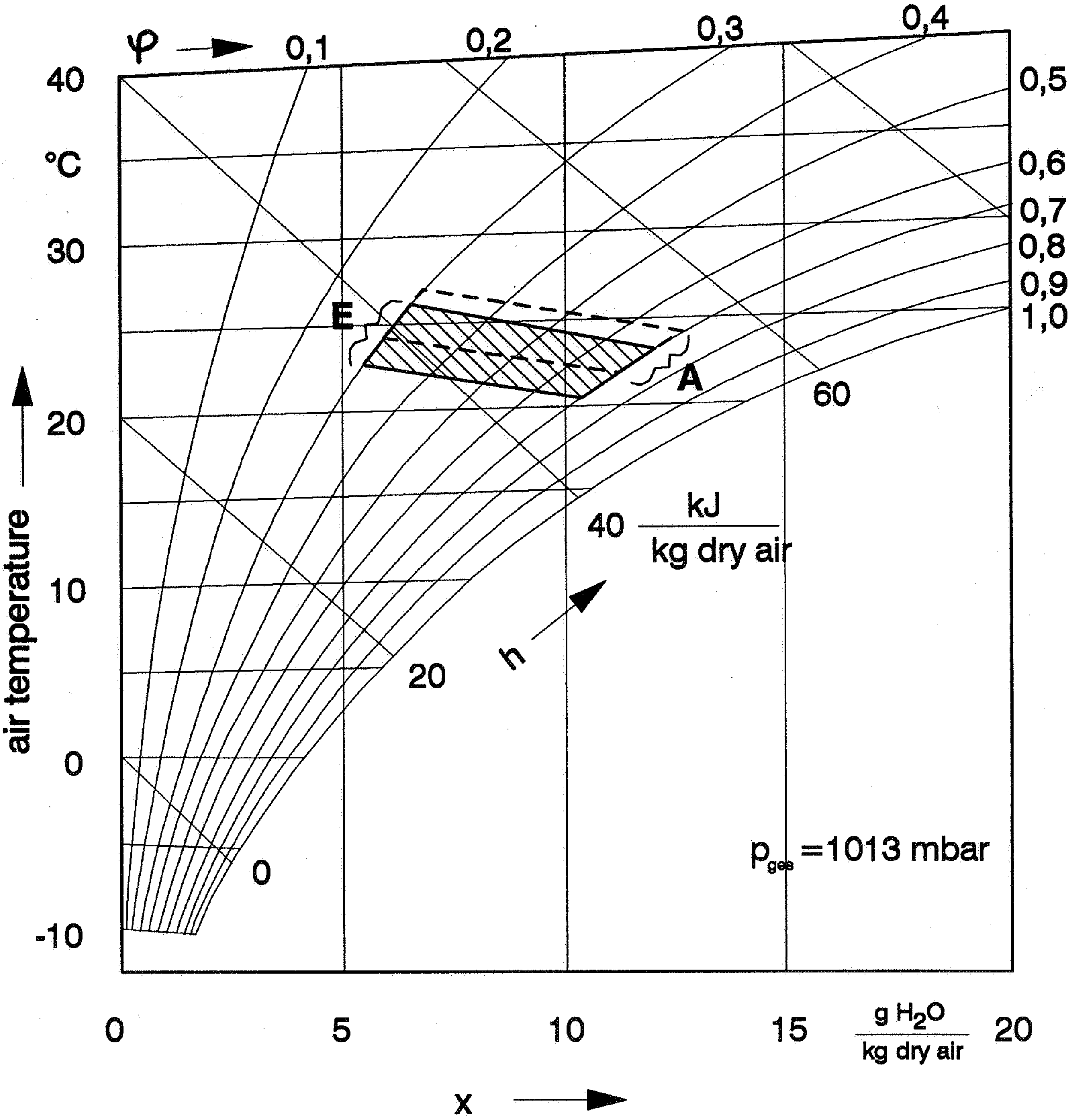
Fig. 16

PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURE LINES FOR STILL AIR

Shaded area indicates the comfort zone.

(Reprinted by permission from the American Society of Heating and Ventilating Engineers' Guide 1932.)

Lewis -(Numler)



Comfort-Zone in the summer

A....in America
E....in Europe

Comfort-Zone

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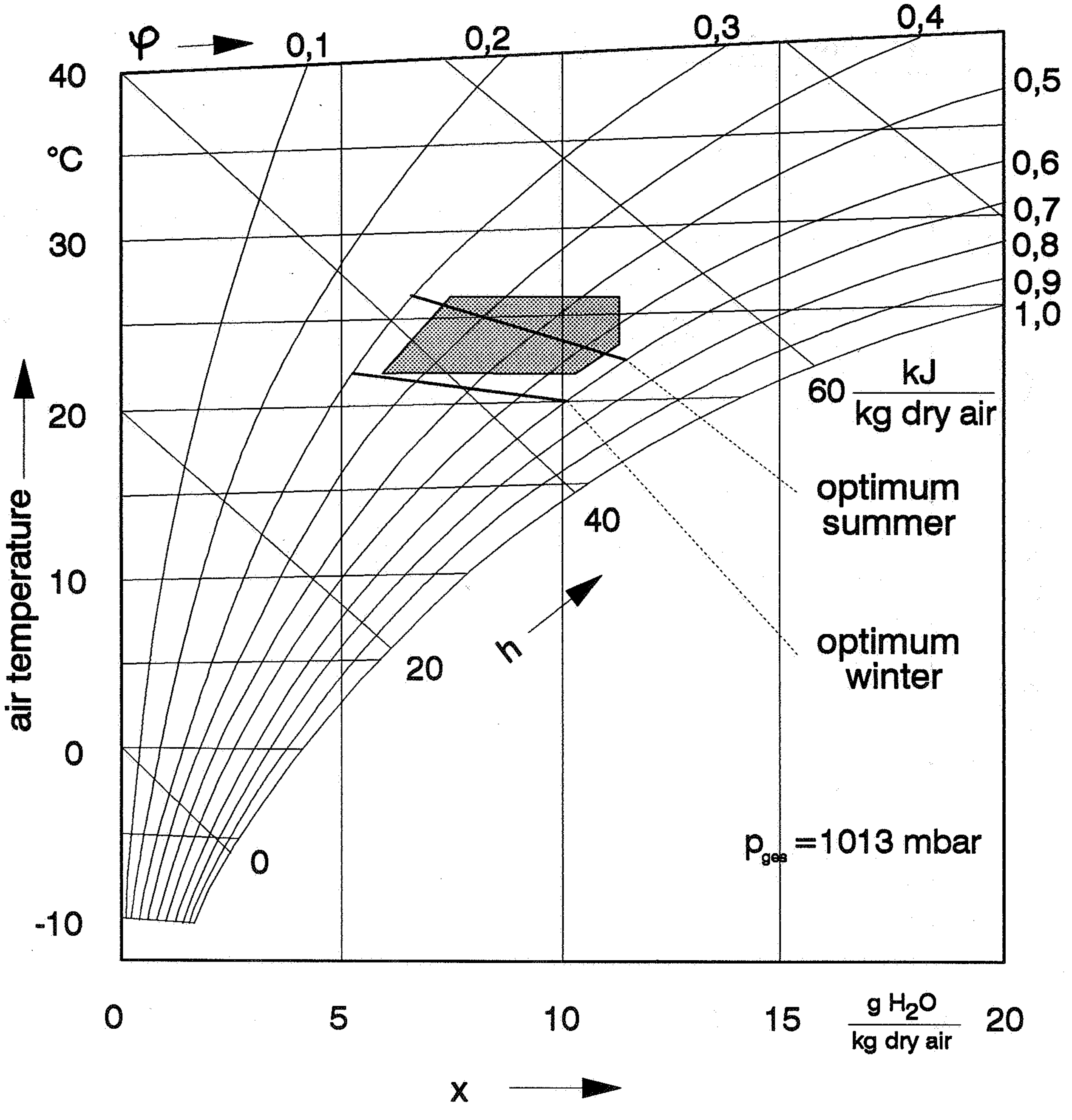

Comfort-Zone

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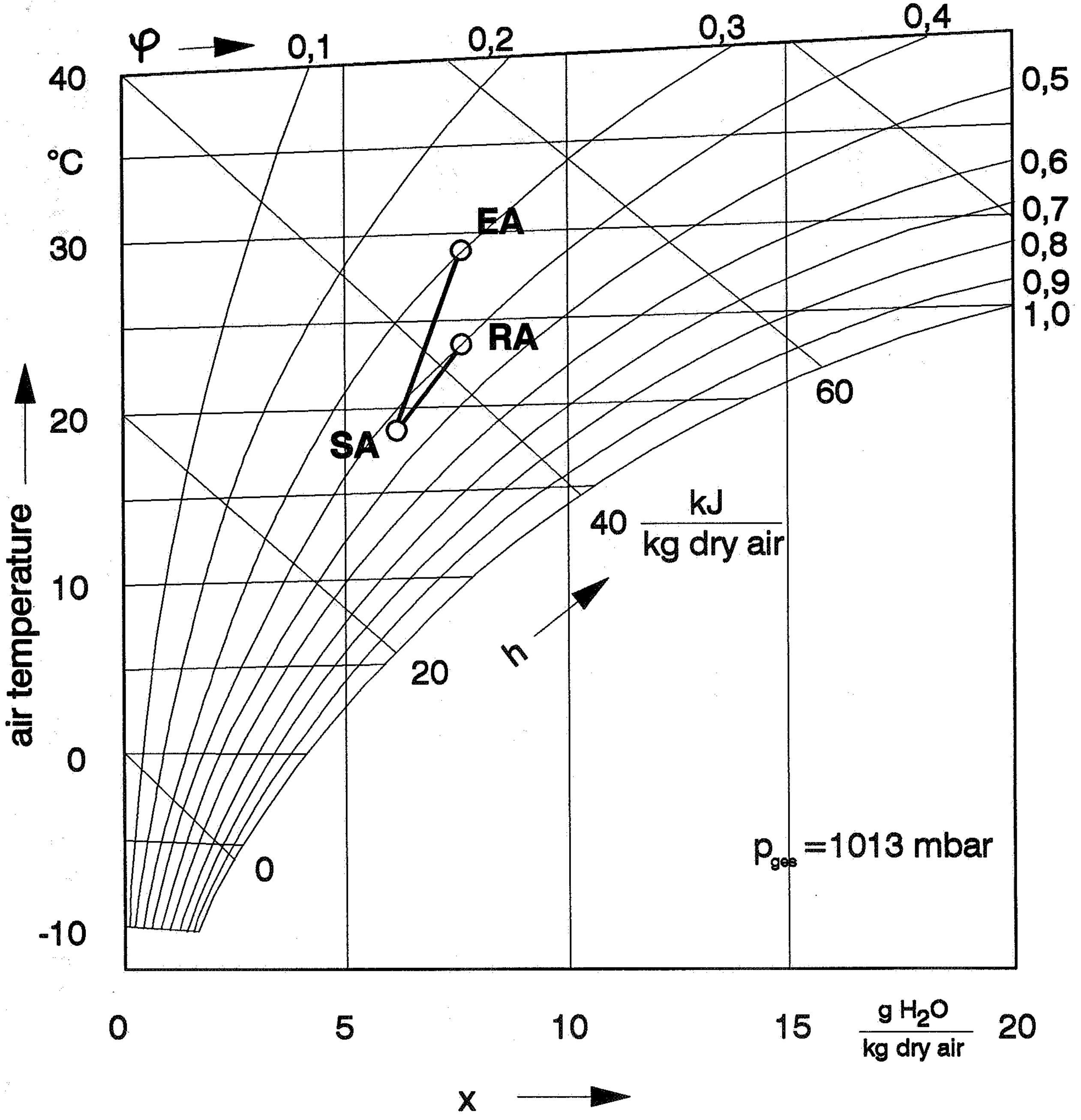

RA.....room air
SA.....supply air
EA.....exhaust air

Change of condition in the h,x - diagram

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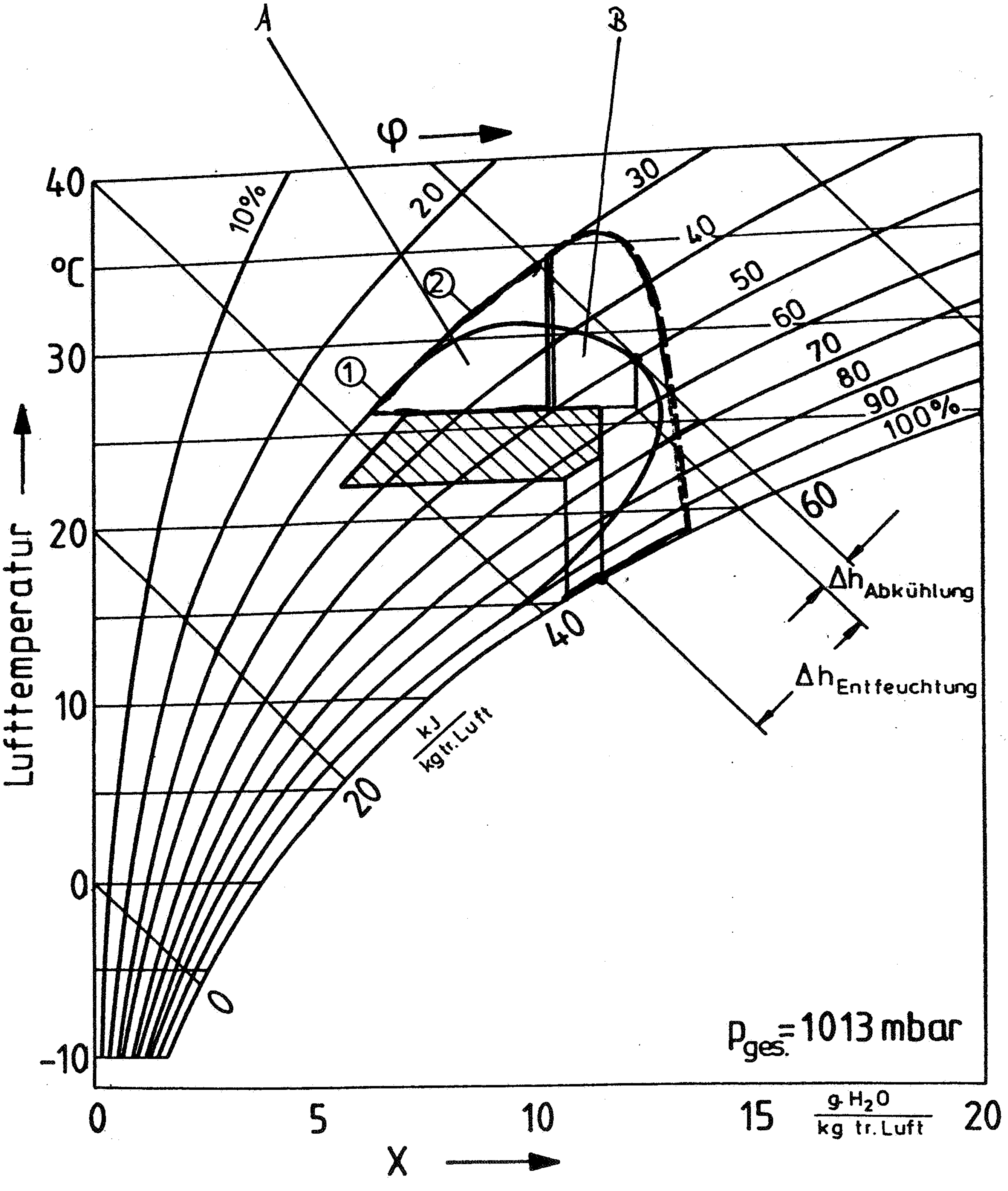
					energy demand		
	ventilation		air changes		heating		dehydration
marble floor:	0,1 m^3/m^2h	≙	0,04 ach	≙	1 W/m^2	≙	2 W/m^2
carpet floor:	2 m^3/m^2h	≙	0,8 ach	≙	25 W/m^2	≙	40 W/m^2
	up to		up to		up to		up to
	8 m^3/m^2h	≙	3,2 ach	≙	100 W/m^2	≙	160 W/m^2

Ventilation rates


A
B
φ
40
°C
30
20
10
0
-10
Lufttemperatur
10%
20
30
40
50
60
70
80
90
100%
①
②
60
Δh Abkühlung
Δh Entfeuchtung
40
kJ / kg tr. Luft
20
0
p ges. = 1013 mbar
0
5
10
15
g H₂O / kg tr. Luft
20
X