Heat transfer in insulation and insulated structure

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by

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HEAT TRANSFER IN INSULATION AND INSULATED STRUCTURE

CLAES G BANKVALL

FOREWORD

This thesis comprises the following reports:

/1/ Bankvall, C.G., Heat transfer in fibrous materials.
National Swedish Building Research, Document D:4, 1972.

/2/ Bankvall, C.G., Natural convective heat transfer in insulated structures.
Lund Institute of Technology, Building Technology, Report 38, 1972.

Detailed information on the theoretical and experimental investigations, as well as information on references etc. can be found in the two original reports.

Lund, November, 1972.

Claes Bankvall

CONTENTS

NOMENCLATURE

d	thickness	m
h	height	m
B_o	specific permeability	m^2
ρ	density	kg/m^3
ε	porosity	-
T	temperature	K
λ	thermal conductivity	W/m K
p	pressure	N/m^2

SI-units have been used unless clearly stated otherwise in the context.

1 INTRODUCTION

Effective thermal insulations are of considerable importance in the field of building physics, due to increasing requirements of comfort, and the necessity of reducing costs. A more effective utilization of the insulating materials, requires further knowledge of their properties. This is especially true for many types of high-performing thermal insulations with complicated mechanisms of heat transfer. It is not possible to judge the behaviour of an insulation inside a structure without knowledge of the different ways of heat transfer in the material itself.

Most highly insulating materials are porous, i.e. they usually contain large amounts of air or other gas. The pore system can be closed, as in many cellular plastics, or open, as in mineral wool. In an open-pore material the transport of heat due to conduction and radiation can be further increased by natural of free convection, i.e. heat transported by a gas-flow due to temperature-induced differences in gas density.

Permeable materials with complex mechanisms of heat transfer, like mineral wool, have often presented difficulties when evaluating building structures where these materials have been used for insulation. The first part of the present report discusses the basic mechanisms of heat transfer in a fibrous material, the second part deals with the natural convective heat transfer in the insulated space or structure. This basis of division corresponds to the original reports (cf. Foreword). The problem of forced convection and protection from wind is not taken up in this study.

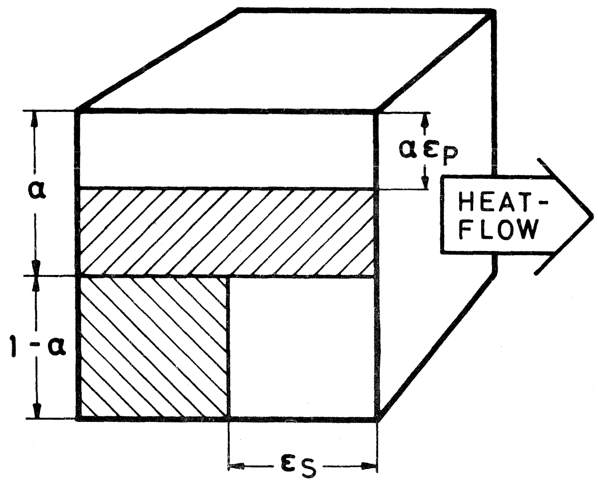


FIG. 1. Model for conduction due to solids and gas in porous material with open pore system, unit volume.

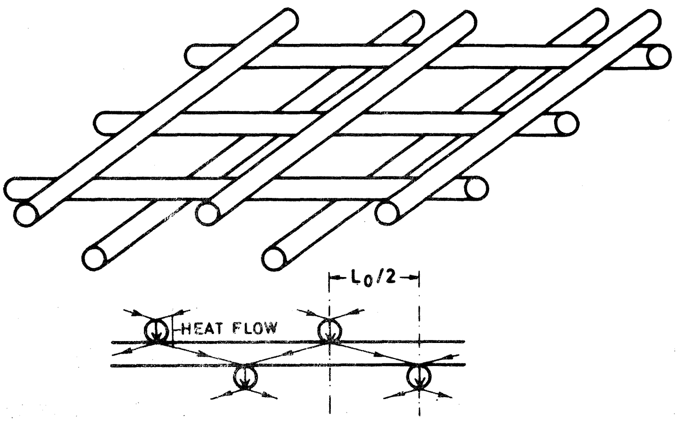


FIG. 2. Model for conduction in fibers and fiber contacts (solids).

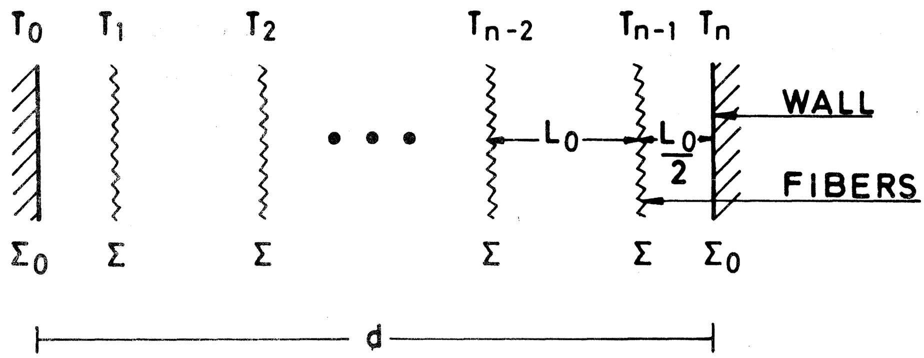


FIG. 3. Model for radiation in fibrous material.

2 MECHANISMS OF HEAT TRANSFER IN (FIBROUS) INSULATION

The basic mechanisms of heat transfer in a porous, in this case fibrous, material are: conduction in solid phase constituting the insulation, radiation within the material and conduction due to the gas confined in the insulation.

In order to evaluate the thermal conduction the material may be treated as a combination of a solid phase and a gas phase. The simplest method is to combine the two extreme limits of the thermal conductivity of a two-phase mixture. This is shown in FIG. 1 for a material with an open pore system. In the part α of the unit volume the two phases are parallel in respect to the heat flow, and in $(1 - \alpha)$ they are in series. The porosity, ε of the material is given by

$$\varepsilon = (1 - \alpha)\, \varepsilon_S + \alpha\, \varepsilon_P \tag{1}$$

The thermal conductivity due to conduction in solids (fibers) λ_F and conduction due to gas λ_G, in FIG. 1, is

$$\lambda_F + \lambda_G = \alpha \cdot (1-\varepsilon_p)\lambda_s + \alpha \cdot \varepsilon_P\, \lambda_g + (1-\alpha) \cdot \frac{\lambda_s\, \lambda_g}{\varepsilon_S \cdot \lambda_s + (1-\varepsilon_S) \cdot \lambda_g} \tag{2}$$

λ_s is the thermal conductivity of the solid phase and λ_g of the gas. If the gas pressure in the material is reduced, the thermal conductivity of the gas, in that case, λ_{ge} is given by

$$\lambda_{ge} = \lambda_g \cdot \frac{pL_o}{pL_o + E\,T} \tag{3}$$

p is the pressure, T the temperature and E a constant depending upon the gas. L_o the "effective pore diameter" or the mean distance between fibers can be calculated from

$$L_o = \frac{\pi}{4} \cdot \frac{D}{1 - \varepsilon} \tag{4}$$

where D is the mean diameter of the fibers.

The conduction in solids, i.e. in fibers and fiber contacts, λ_F, can be calculated separately if the structure of the material is

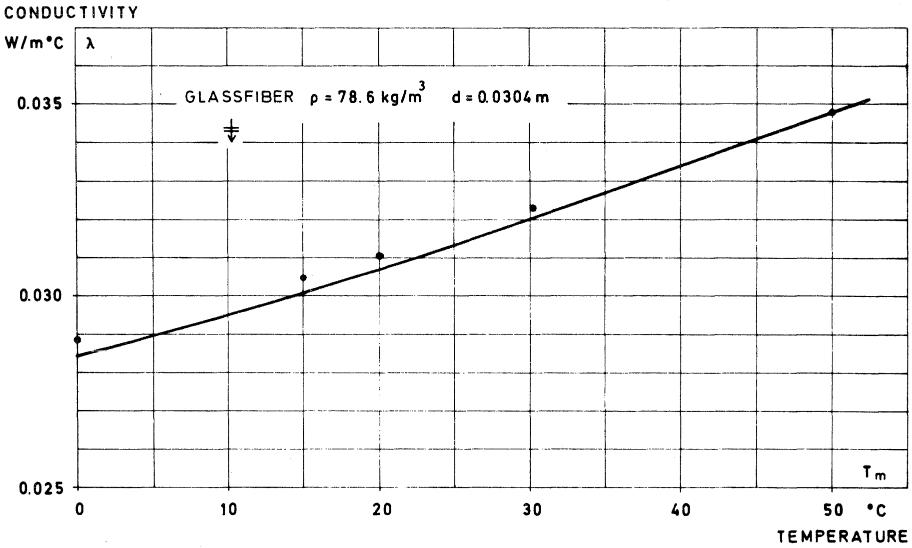


FIG. 4. The total thermal conductivity of a fibrous material as a function of mean temperature (— calculated values, ● measured values).

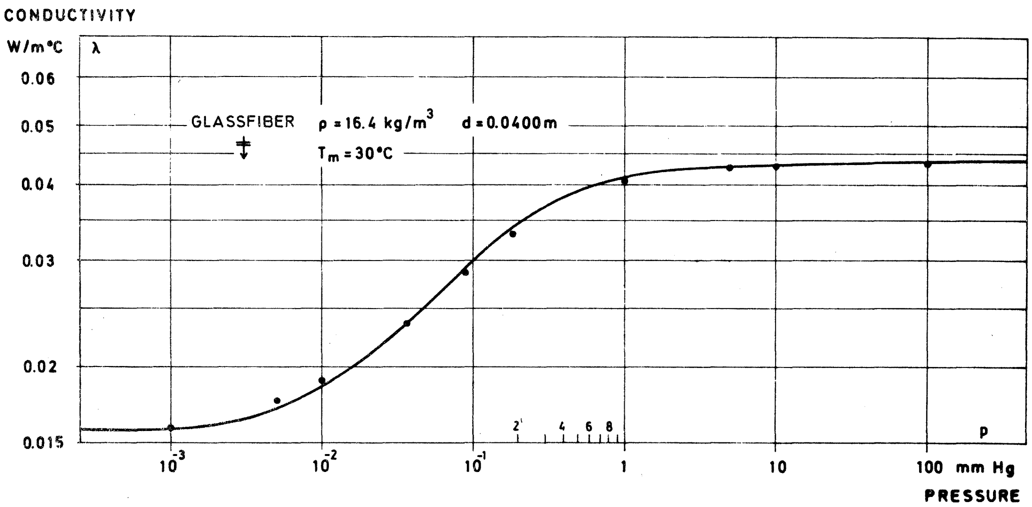


FIG. 5. The influence of air pressure on the thermal conductivity of a fibrous material (— calculated values, ● measured values).

known or can be assumed, for example as in FIG. 2. Such a model will, however, only give a very approximate value for λ_F.

The radiation in the fibrous material will be absorbed, transmitted, reflected and scattered by the fibers. Thus it is very difficult to give a physically complete and correct picture of this mechanism of heat transfer. If the fibrous material is considered as consisting of disoriented fibers in layers at right angle to the heat flow, the model in FIG. 3 may be considered. In the figure, d is the thickness of the material, L_o the distance between fiber layers and T_o and T_n the wall temperatures. It is also assumed that the walls and the fiber layers behave as grey, non-transparent bodies with emissivity Σ_o and Σ. If the temperature difference is moderate in comparison to the absolute temperature, then the effective thermal conductivity due to radiation λ_R can be calculated from

$$\lambda_R = \frac{4\sigma_s \cdot L_o \cdot T_m^3}{\left(\frac{1}{\beta} + \frac{L_o}{d} \cdot \left(\frac{2}{\Sigma_o} - 1\right)\right)} \tag{5}$$

where T_m is the mean temperature of the material and the factor $\beta = f(\Sigma)$ decribes the radiational properties of the fibers and fiber layers ($\sigma_s = 5{\cdot}7{\cdot}10^{-8}$ W/m^2 K^4). If $d > L_o$ and $\Sigma_o \simeq 1$ then

$$\lambda_R = 4\sigma_s \cdot L_o \cdot \beta\, T_m^3 \tag{6}$$

The total effective thermal conductivity of the fibrous material is given by

$$\lambda = \lambda_G + \lambda_F + \lambda_R \tag{7}$$

When the material is evacuated, the remaining effective thermal conductivity is $\lambda_F + \lambda_R$, and λ_G is, consequently, the maximal decrease in thermal conductivity that can be expected. Three unknown factors have been introduced in the models, β and an arbitrary pairing of ε_S, ε_P and α.

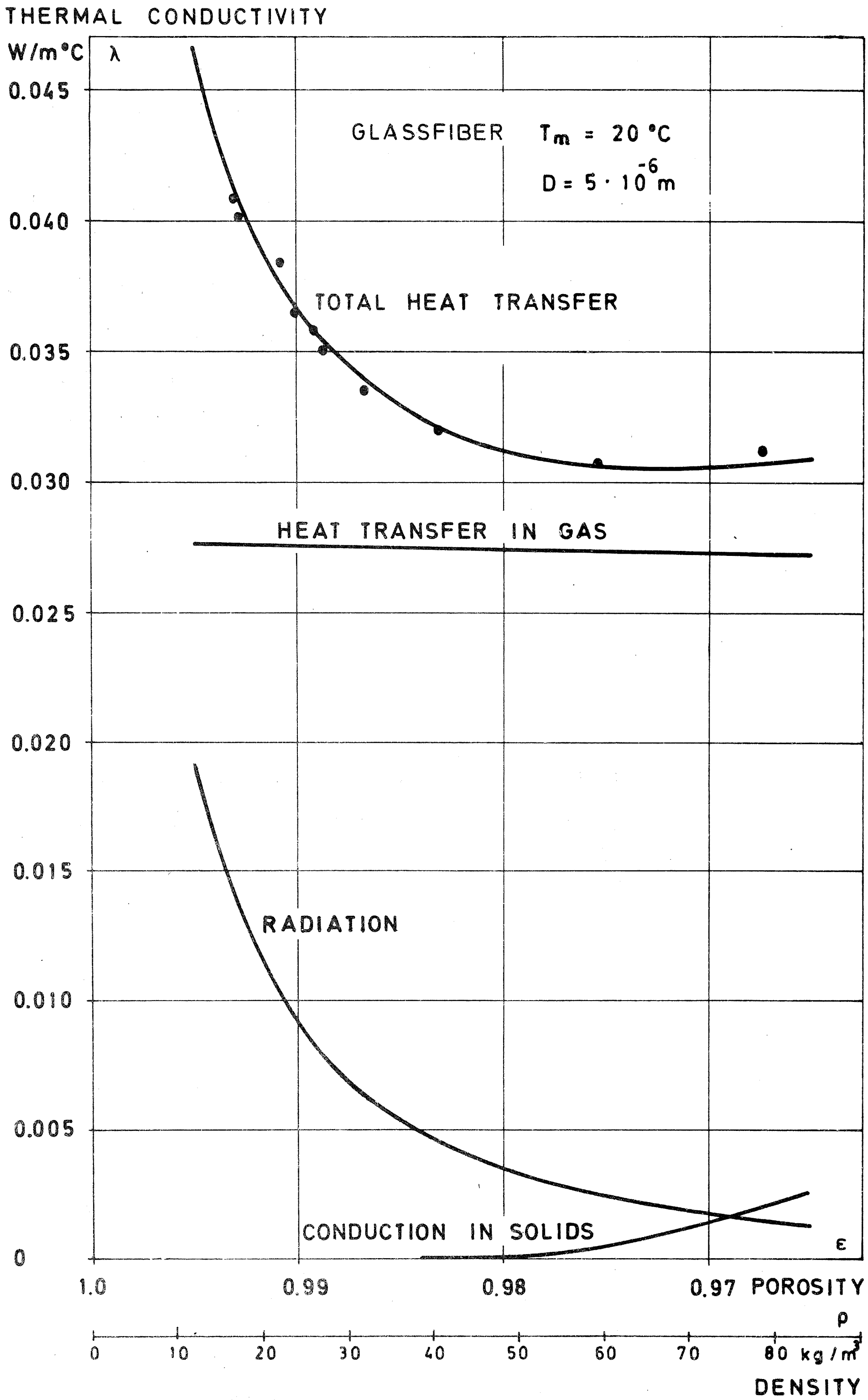


FIG. 6. The mechanisms of heat transfer in a fibrous material (— calculated values, ● measured values).

A detailed experimental investigation of a fiber glass material of different densities was made in a one sided guarded hot plate, with the heat flow directed downwards so as to avoid convection. The radiation coefficient was found to vary linearly with the temperature (0-50 ^{o}C) and the density (15-80 kg/m^3). The parameters ε_S, ε_P and α varied linearly with the density. FIG. 4-6 show results from these measurements.

The influence of the mechanisms of heat transfer on the effective thermal conductivity of the fibrous material can be summarized as follows

- conduction due to gas contributes the largest part of the thermal conductivity in the range of density studied (15-80 kg/m^3)

- radiation is of greatest importance for low density materials and leads to high values of thermal conductivity in these cases.

- conduction in solids is important in high density materials where it can lead to an increase in the thermal conductivity value.

- increasing mean temperature of a material gives an increase in its thermal conductivity value. This is especially noticeable at low densities due to radiation.

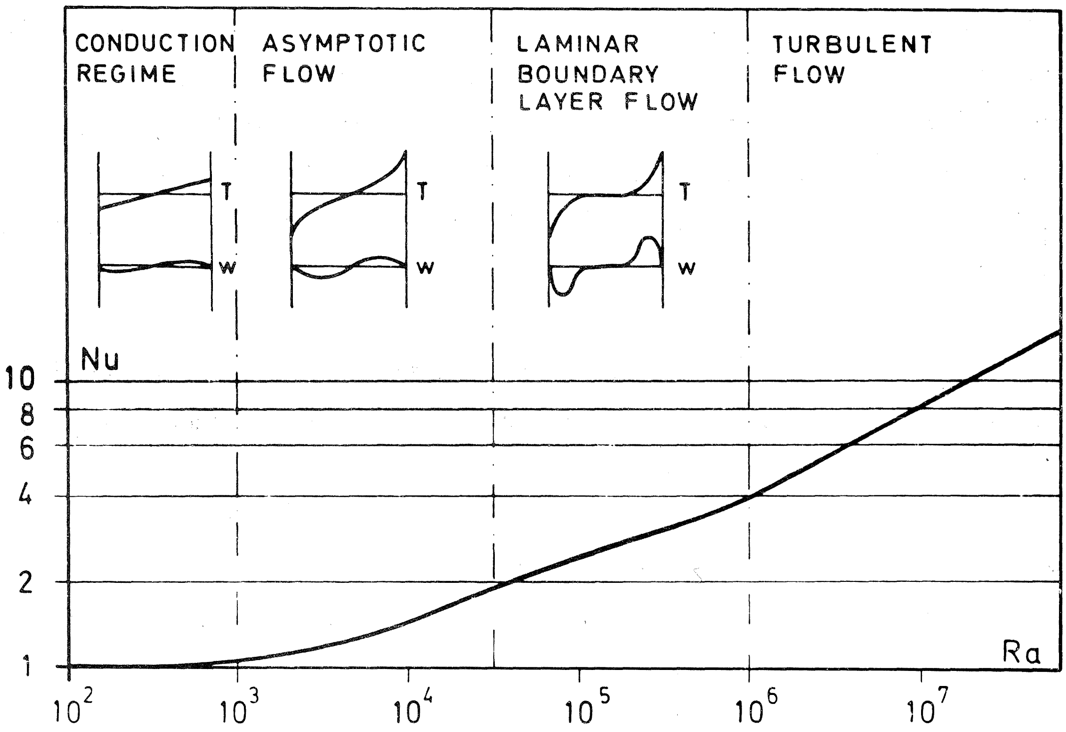


FIG. 7. Schematic temperature and flow fields in natural convective heat transfer in a vertical air space (h/d ≃ 10).

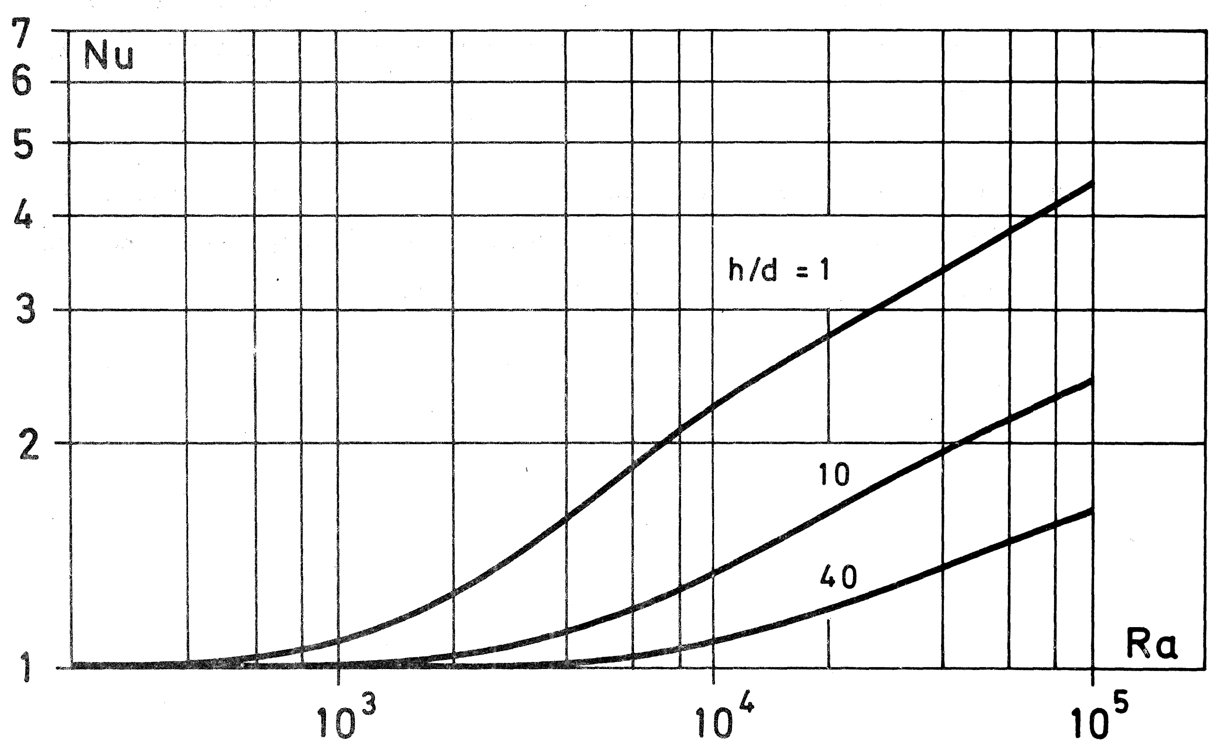


FIG. 8. Natural convective heat transfer in vertical air space with isothermal vertical and insulated horizontal boundaries.

3 NATURAL CONVECTIVE HEAT TRANSFER IN INSULATED STRUCTURES.

Natural or free convection can be considered as the heat transfer by flow of fluid due to the interaction between the field of gravity and temperature-induced density variations in the fluid. This is a fairly well known phenomenon in the air space and in order to understand the heat transfer due to convection in an insulation it is suitable to observe the two extreme limits from the point of view of heat transfer.

One extreme limit of the porous insulation is the solid structure which is reached as the porosity of the material decreases. The other extreme limit is reached as the porosity increases and is the air space (or uninsulated structure). When the basic physical mechanisms of heat transfer are known in a material without convective gas flow, the question is under what circumstances natural convection will be of importance to the total effective thermal conductivity of the insulation, for example when installed in a wall. In order to understand the phenomenology of natural convection, valuable information can be gained from the available knowledge of the behaviour of the air space.

It is possible to show theoretically that the convective heat transfer through a vertical air space, with height h and thickness d, can be expressed dimensionlessly as the ratio between the thermal conductivity with convective flow in the space, λ_{cv} and the conductivity in the stagnant air, λ.

$$\frac{\lambda_{cv}}{\lambda} = Nu = f(Ra, h/d) \tag{8}$$

When no convective flow is present the Nusselt number Nu, will be equal to one. The Rayleigh number is

$$Ra = \frac{g \cdot \Delta T \cdot d^3}{a \cdot \nu \cdot T_m} \tag{9}$$

g is the gravitational acceleration, ΔT the temperature difference over the space, T_m its mean temperature, a the thermal diffusivity of the air and ν its kinematic viscosity.

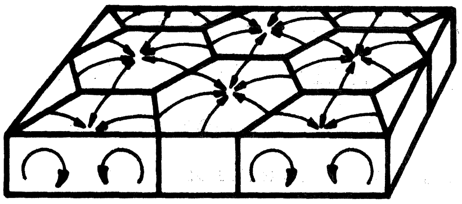


FIG. 9. Convection cells in horizontal air space.

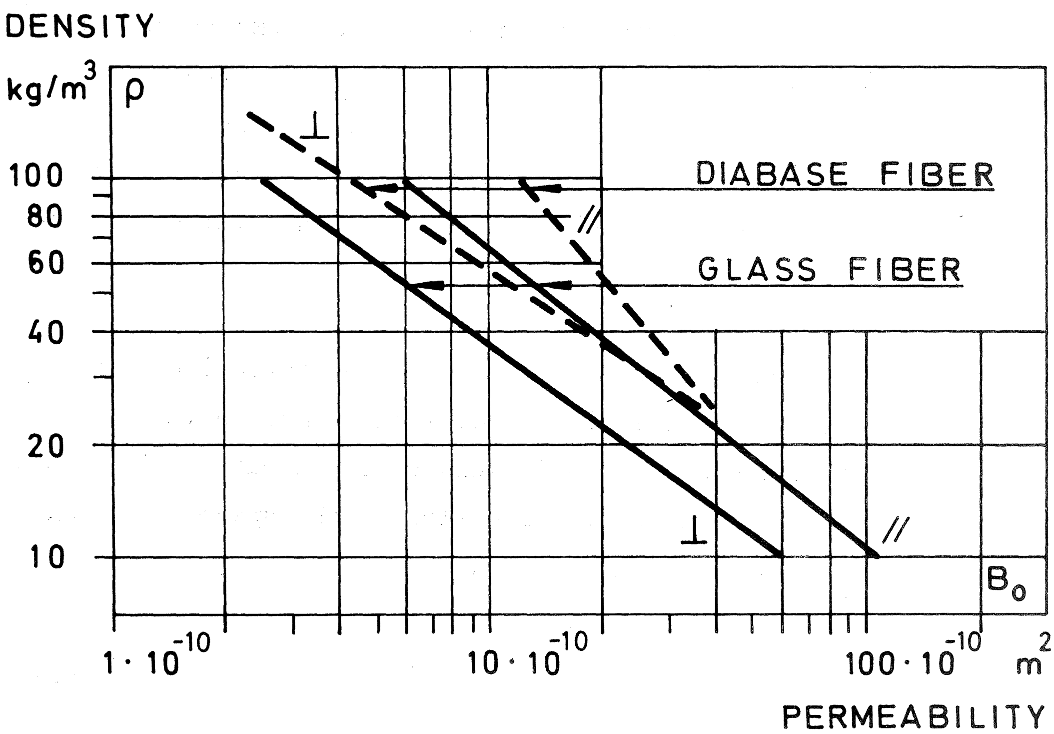


FIG. 10. Experimental specific permeability values for different mineral wool densities.

FIG. 7 schematically shows the behaviour of the convective flow and heat transfer in the vertical space. The flow is directed upwards at the warm side and downwards at the cold side. The air flow leads to deformations in the temperature field, as compared to stagnant air. The temperature gradients at the boundary surfaces are changed and the total heat transfer through the space is increased. This is due to the warming and cooling, respectively, of the air as it flows along the surfaces and at top and bottom passes from one side to the other. The heat flow over the height of the wall is correspondingly deformed. FIG. 8 shows the natural convective heat transfer in a space of different aspect ratio (h/d).

The heat transfer due to natural convection in a horizontal air space heated from below, which is the interesting case, can be shown to set in at a critical Ra-value (cf. FIG. 9) of 1700, i.e. for natural convection

$$Ra = \frac{g \cdot \Delta T \cdot h^3}{a \cdot \nu \cdot T_m} \geq 1700 \qquad (10)$$

where h is the height of the space.

When a permeable material is introduced as insulation in an uninsulated (air) space the resistance to air flow in the space will increase. This can be expressed by the specific permeability coefficient

$$B_o = - \frac{Q}{A} \cdot \frac{\eta}{\text{grad } p} \qquad (11)$$

where Q is the volume flow across the cross-sectional area A, and η the dynamic viscosity of the air.

Generally speaking the permeability is the fluid conductivity of the porous material and the value of B_o is determined by the structure of the material. FIG. 10 shows experimentally measured permeability values parallel and at right angle to the fiber layers in different mineral wool insulations.

The natural convective flow and heat transfer in an insulated space can now be calculated in much the same way as in the case of the

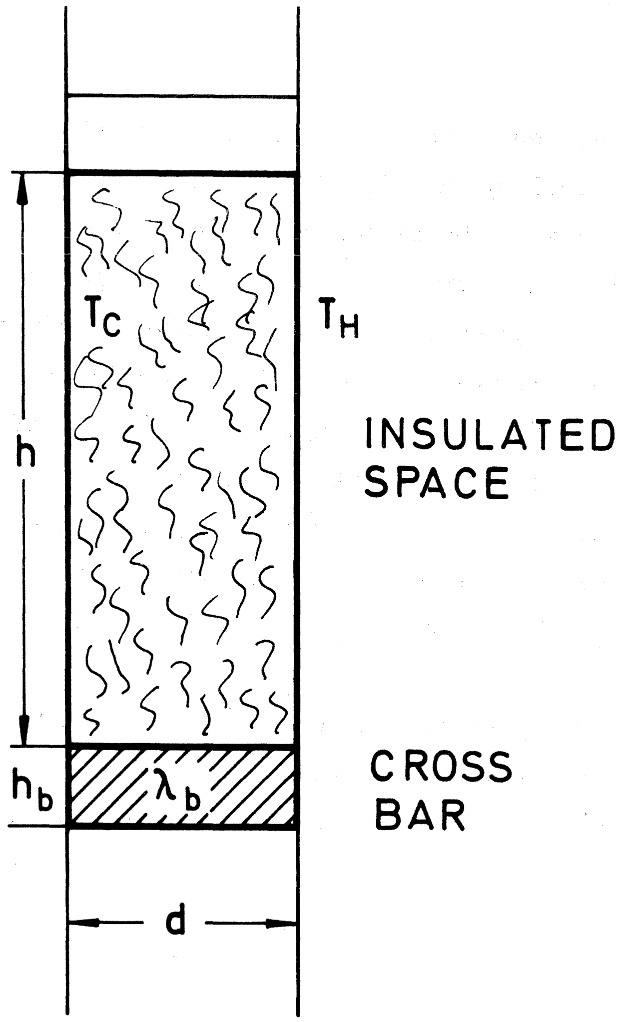


FIG. 11. Vertical permeable space.

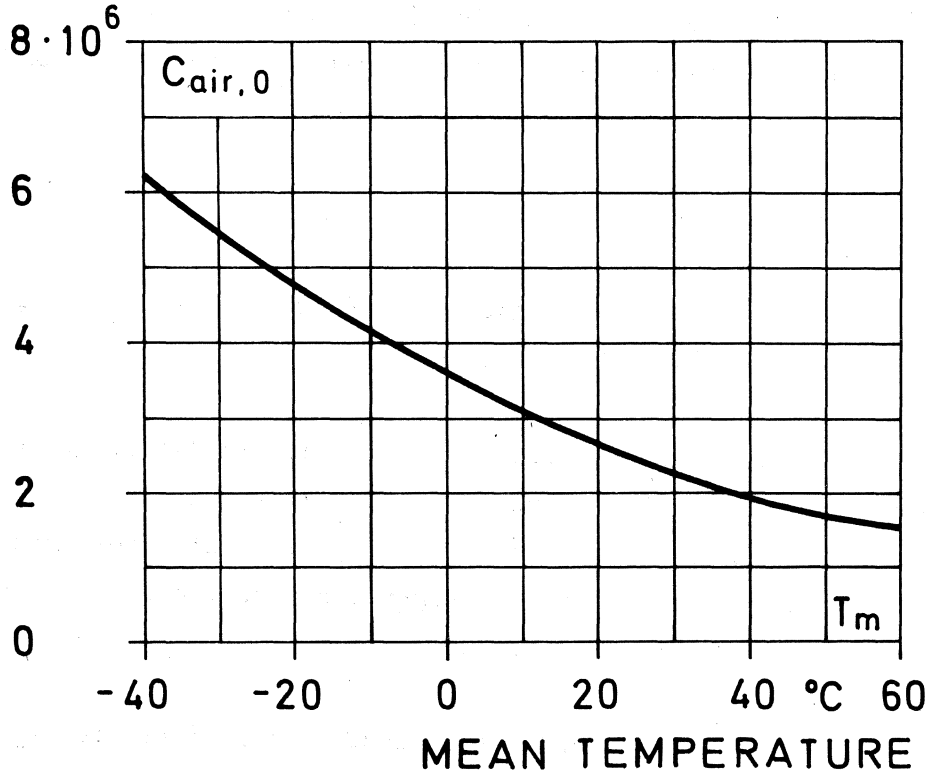


FIG. 12. Coefficient for calculation of Ra_o-value for a permeable space.

air space.

The convective heat transfer in the vertical space (cf. FIG 11) is given by

$$Nu = f(Ra_o, h/d) \tag{12}$$

together with the appropriate boundary conditions.

The modified Rayleigh number, Ra_o, describes the influence from specific permeability B_o, and is defined by

$$Ra_o = C_{air,o} \frac{d \cdot \Delta T \cdot B_o}{\lambda_o} \tag{13}$$

where λ_o is the thermal conductivity of the material at stagnant air. The coefficient $C_{air,o}$ depends solely upon the air, and varies with the mean temperature, as is shown in FIG. 12.

In the same way as for the air space the air flow in the insulated space deforms the temperature field and for this reason alters the heat flow patterns at the vertical isothermal boundaries. The changes in the velocity, temperature and heat flow fields, as well as the convective heat transfer can be calculated numerically for different boundary conditions and aspect ratio. FIG. 13 shows an example of this.

As in the case of the horizontal air space, it can be found that there exists a critical modified Rayleigh number value above which natural convection will be present in the permeable space. The critical Ra_o-value is

$$Ra_o = C_{air,o} \frac{h \cdot \Delta T \cdot B_o}{\lambda_o} = 4\pi^2 \tag{14}$$

The boundary conditions and the aspect ratio ($\simeq 0$) are of little interest in this case.

The theoretical results have been verified by experimental inves-

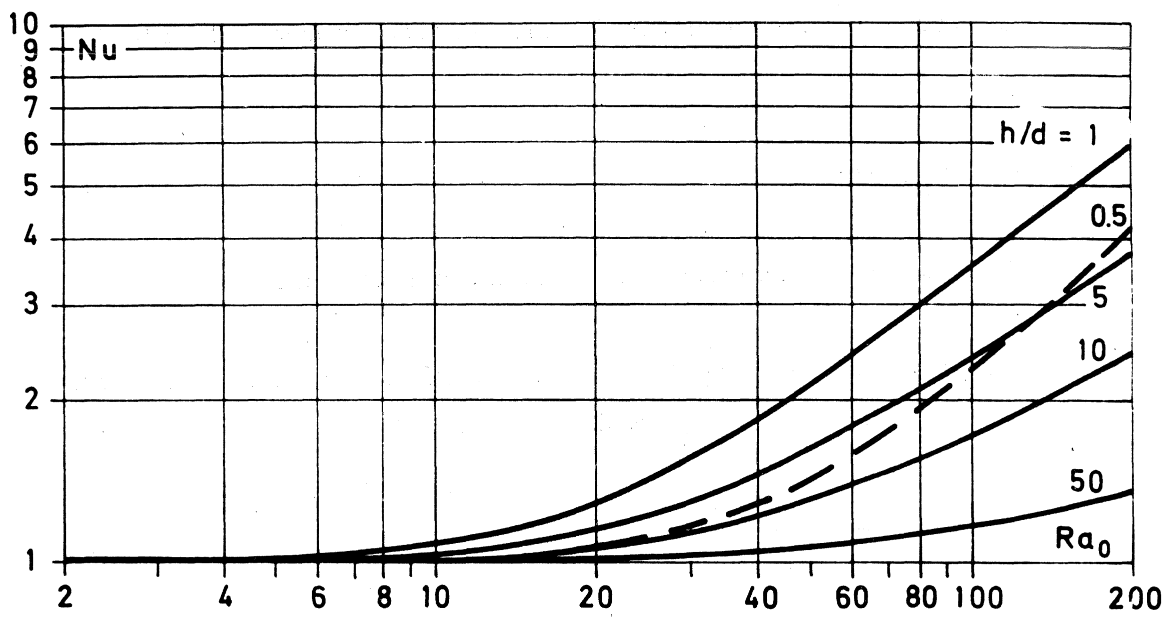


FIG. 13. Natural convective heat transfer in permeable space with isothermal vertical and insulated horizontal boundaries.

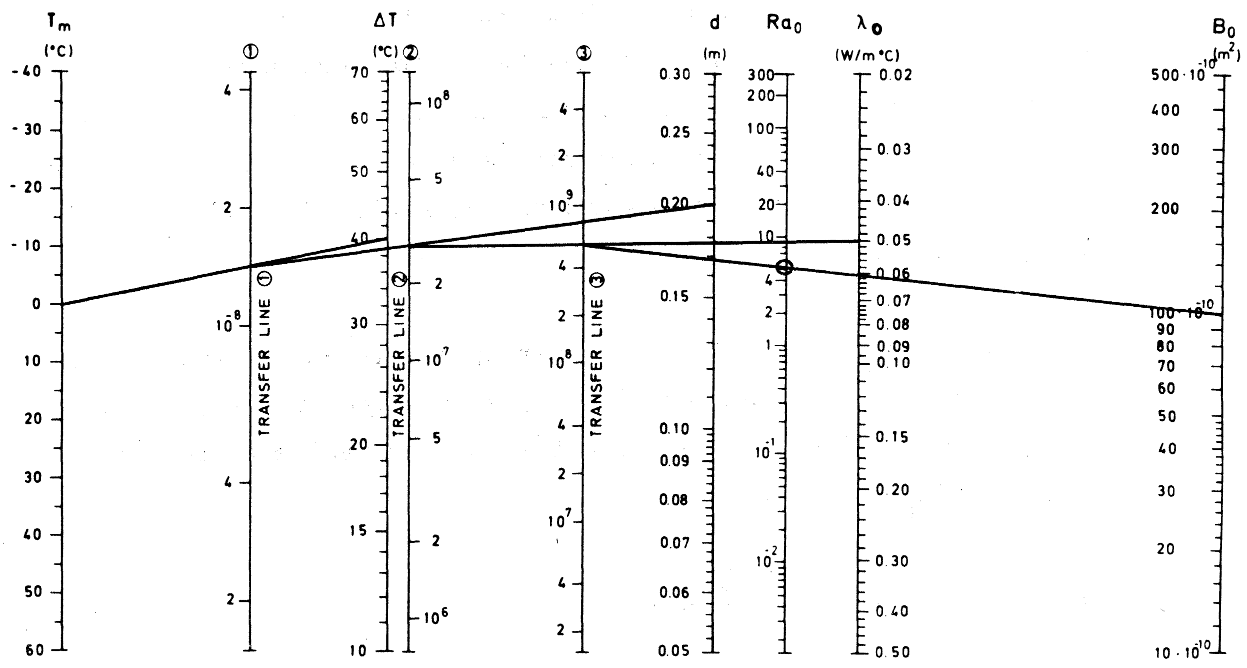


FIG. 14. Diagram for calculation of Ra_o-value in insulated space (cf. last page). In the diagram calculation is shown for T_m = 0 °C, ΔT = 40 °C, d = 0.2 m, λ_o = 0.05 W/m K and $B_o = 100 \cdot 10^{-10}$ m^2. This gives Ra_o = 5.5.

investigations in guarded hot plates and by measurements on insulated wall structures. These investigations have also shown the influence of small air spaces and openings within the insulated space. This permits the following conclusions to be drawn:

The natural convection in a fully insulated space is governed by the modified Rayleigh number, Ra_o the aspect ratio, h/d and the boundary conditions. The Ra_o-value can be calculated as in FIG. 14 (cf. last page) or from the previous equations. Full information on the influence of boundary conditions and aspect ratio can be found in the original report (cf. Foreword).

- In the case of the horizontal space a critical Ra_o-value = $4\ \pi^2$ exists. If only normal building physical applications are considered, the temperature of the warm side can be assumed to be 20 °C and on the cold side it is seldom below -20 °C. In FIG. 15 the minimal specific permeablility-values necessary to exceed the critical Ra_o-value at $\Delta T = 40$ °C and $T_m = 0$ °C are given as a solid line for different λ_o-values and h = 0.30 m. This figure gives an indication of what the conditions have to be to induce natural convective heat transfer in the horizontal space.

- In the vertical permeable space, the situation is slightly more complicated, since both aspect ratio and boundary conditions often influence the amount of natural convective heat transfer in the space. In order to investigate the implications of this under normal applications, assumptions are made similar to those in FIG. 15, $\Delta T = 40$ °C, $T_m = 0$ °C and d = 0.20 m. Unlike the horizontal case no critical Ra_o-value exists in the vertical case. The condition for 5 % natural convective heat transfer is therefore illustrated in FIG. 16 for different specific permeabilities, B_o and thermal conductivities, λ_o. The Ra_o-value when calculating this figure was taken from FIG. 13 and is valid for those boundary conditions. The aspect ratio was chosen as one, since this represents a situation with approximately maximal convective heat transfer.

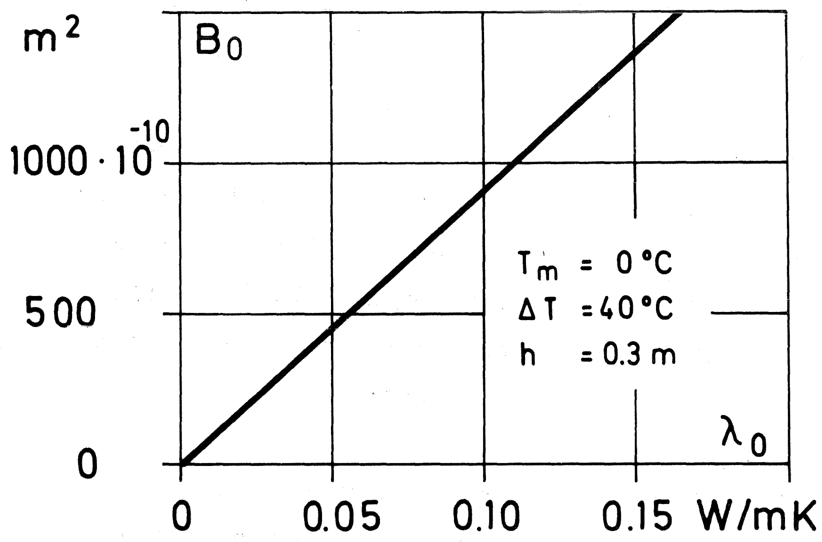


FIG. 15. Minimal specific permeability for natural convective heat flow in horizontal insulations of different thermal conductivity.

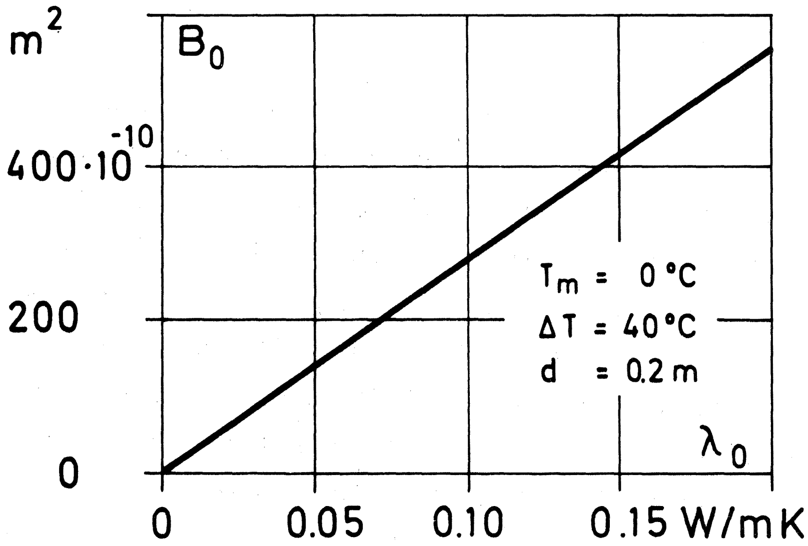
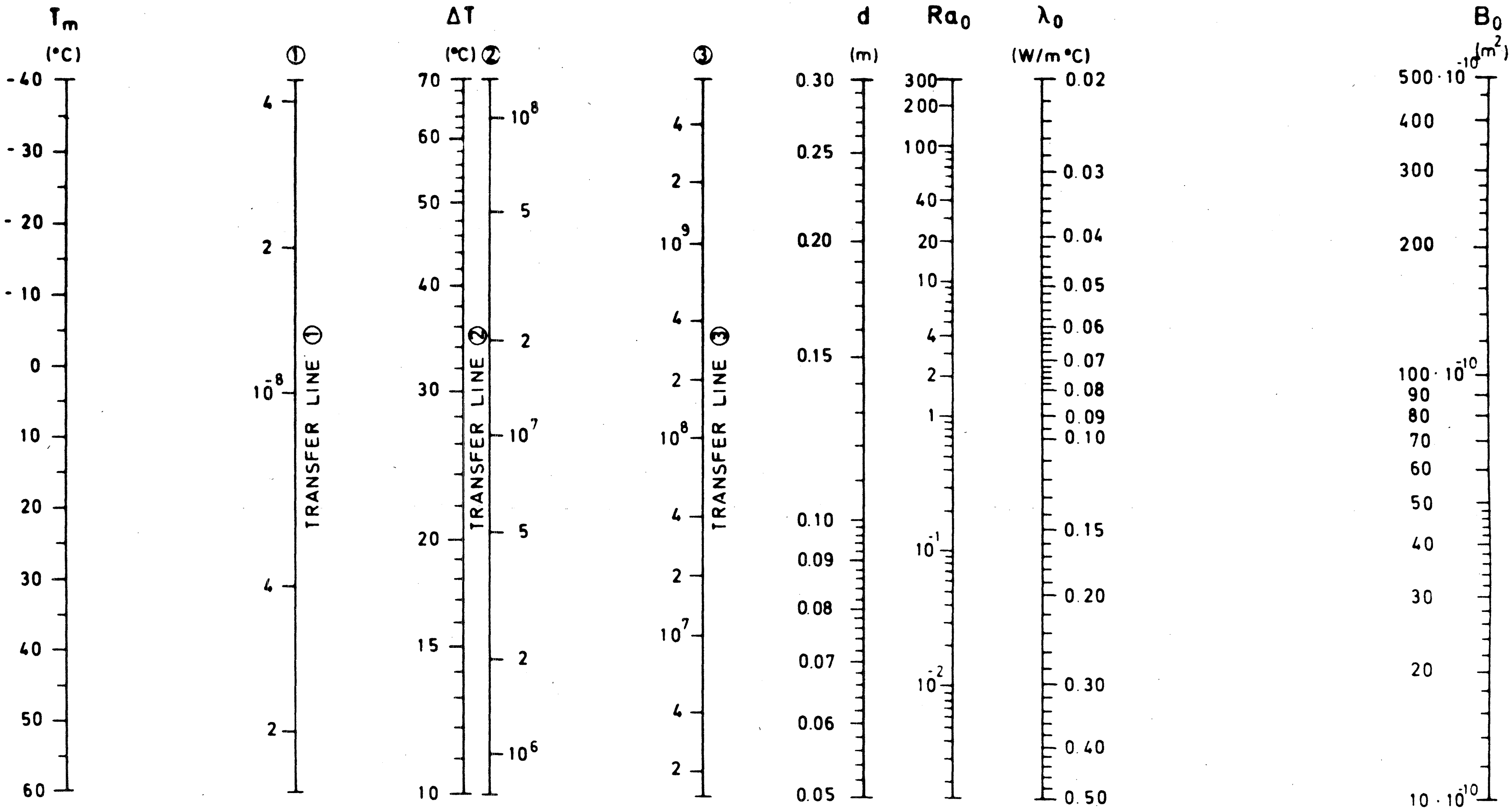


FIG. 16. Minimal specific permeability for 5 % natural convective heat flow in vertical insulations of different thermal conductivity (h/d = 1).

- In general, it can be concluded that in normal temperature conditions, the natural convective heat transfer in an insulated space takes very high permeability values to be of any importance, and that if uncontrolled air spaces and slits are introduced into the insulating structure, any amount of convective heat transfer can be expected. In this case, knowledge about the behaviour of the air space is useful to estimate the ultimate result.



T_m (°C)
-40 -30 -20 -10 0 10 20 30 40 50 60
①
4 2 10^{-8} 4 2
TRANSFER LINE ①
ΔT (°C)
70 60 50 40 30 20 15 10
②
10^8 5 2 10^7 5 2 10^6
TRANSFER LINE ②
③
4 2 10^9 4 2 10^8 4 2 10^7 4 2
TRANSFER LINE ③
d (m)
0.30 0.25 0.20 0.15 0.10 0.09 0.08 0.07 0.06 0.05
Ra_0
300 200 100 40 20 10 4 2 1 10^{-1} 10^{-2}
λ_0 (W/m °C)
0.02 0.03 0.04 0.05 0.06 0.07 0.08 0.09 0.10 0.15 0.20 0.30 0.40 0.50
B_0 (m^2)
$500 \cdot 10^{-10}$ 400 300 200 $100 \cdot 10^{-10}$ 90 80 70 60 50 40 30 20 $10 \cdot 10^{-10}$