MODELLING OF THERMAL PROCESSES IN INDOOR ICERINKS

by

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January 1986
Thermal Insulation Laboratory
Technical University of Denmark
Report no 171

Printed at the
Thermal Insulation Laboratory,
Technical University
of Denmark

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PREFACE

This report forms a part of the reporting of the R and D project: Udvikling af varmestråleskærm til skøjtehaller (Development of a radiant heat shield for icerinks). The project is financed by a grant from the Danish Ministry of Energy to the company A/S IKAS-isolering under the act: "Support to develop and utilize new energy technologies".

Part of the project has been carried out at the Thermal Insulation Laboratory of the Technical University of Denmark. This report has been prepared by Teresa Forowicz while in residence at the laboratory as a guest researcher, in collaboration with prof. V. Korsgaard. T. Forowicz has graduated from Warsaw Technical University and holds a Ph.D. in mechanical engineering.

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April 1986

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ABSTRACT

Heat transfer by radiation between ceiling and ice and high humidity of air in indoor icerinks very often cause heavy condensation on the ceiling or roof construction, which has some bad effects. To check how often condensation will occur and possible ways of preventing condensation, a simple computer model of the thermal processes taking place in an indoor icerink was elaborated. The assumptions being made concerning the model system geometry as well as the mathematical problem formulation are described. Next, the mathematical model of the problem being considered, the method of solution and the short description of the simulation program are presented. The report shows further the results obtained from several executions of the program using different data regarding changes in the model itself as well as the influence of different ventilation rates, heat input by radiation and convection etc. These results have allowed for general comparison between four base cases, i.e. between the model icerink with a ceiling made from ordinary building material, with a bright aluminium foil glued to the ceiling surface and with a suspended shield of corrugated bright aluminium plates installed below the roof construction, which surface facing the roof is unpainted or painted to increase its absorptivity.

1. INTRODUCTION

The heat exchange by radiation between the ice surface and the ceiling or roof construction in an indoor icerink cools the ceiling or roof surface facing the ice. This process together with high humidity of the air inside such a building causes very often a heavy water vapour condensation. The condensation will occur when the ceiling or roof is cooled below the saturation temperature. Results following this phenomenon such as greater maintenance or even deterioration of the ceiling or roof construction, water dripping from the ceiling or roof upon the skaters and upon the ice surface causing unpleasant feelings and more frequent need for ice resurfacing - should be prevented.

On the other hand, the radiant heat transfer rate between the ceiling or roof and the ice surface has a great influence on the energy consumption; it considerably increases the cooling load necessary to keep the ice at the desired temperature.

The conclusions from the above statements are not in agreement - for preventing condensation the ceiling temperature should be increased, for cutting down on energy it just should not. Prevention of condensation on the ceiling or roof can also be achieved by lowering the dew point by dehumidification. But this attempt as well as ceiling heating results in a considerable increase of the energy consumption, hence non of these methods can be regarded and recommended as a good solution of the problem of condensation in indoor icerinks.

The next possibility of increasing the ceiling or roof temperature is gluing a material with low emissivity to the ceiling surface or mounting a suspended shield of such a material below the roof construction [1], [2]. This solution has been realized in Rødovre and later in Hørsholm icerinks where a suspended shield of corrugated bright aluminium plates was mounted in a certain distance from the roof, below the supporting construction.

However, under certain weather conditions condensation occurred on the aluminium shield in Rødovre where the roof is uninsulated, but not in the heated icerink in Hørsholm. Condensation only occurred when the outdoor humidity was close to 100% and the air temperature higher than indoor. Due to a high infiltration rate the indoor air would be saturated with a dew point higher than the temperature of the aluminium shield, causing condensation, thereby destroying the low emissivity of the shield, with increased condensation as a result. Since the temperature of the shield in the critical situation is lower than the uninsulated roof its temperature could perhaps be increased sufficiently by radiation from the warmer roof to prevent condensation, if the top side of the shield was painted to increase its absorptivity.

In the present work a simplified mathematical model is set up which simulates the complex thermal processes in an indoor icerink. A computer program for the model has been developed [3] which calculates air and surface temperatures and heat fluxes. Weather data are used as an input. The influence of a shield with varying surface emissivities and its influence on cooling load and risk for condensation are investigated.

2. GENERAL

The exact modelling of the systems continuous in space and time, is impossible. As a result of mathematical modelling we will always have only approximation of the real process being a solution of the model problem consisting of model geometry of the system and its model mathematical description.

Firstly, it is necessary to replace all the continuous variables by the discrete ones; secondly, to make many other simplifying assumptions. Of course to obtain an accurate and predictive model all aspects of the process would have to be integrated into the model.

In the first phase of model preparation it always comes into question what kind of model it should be, complex or simple? etc. Does this problem solution require a sophisticated model, or maybe the simple one is reasonable?

The modelling of the heat transfer processes in an indoor icerink is very difficult, because there are so many different problems which need to be considered in accurately representing an entire process. These are for example condensation and evaporation, sublimation and resublimation of water vapour, the ice melting and freezing, the air movement, convection and radiation inside the building and in the building surroundings, conduction in the structure, etc. On the other hand, modelling is limited by lack of available data regarding values of physical constants and variables as well as appropriate descriptions of some partial processes which are involved in the whole process being considered.

Taking into account above considerations as well as the aim of the model mentioned in the introduction, it was decided that the model icerink should be a simple one which will allow, however, to look for response of the system to some changes in its maintenance.

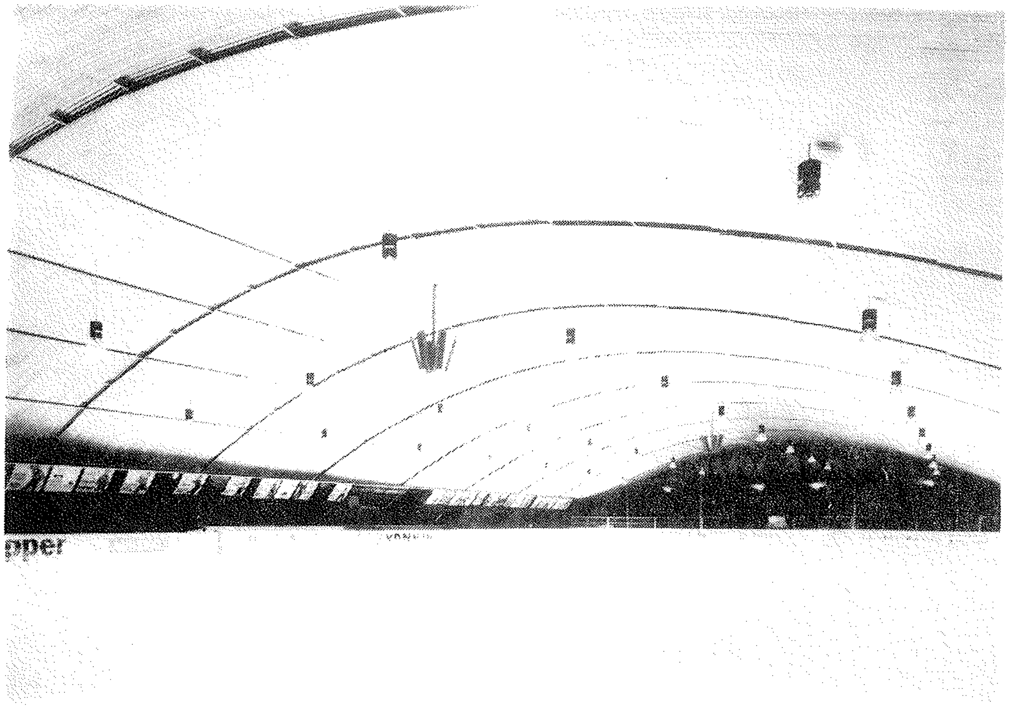

Fig. 1. The icerink at Rødovre.

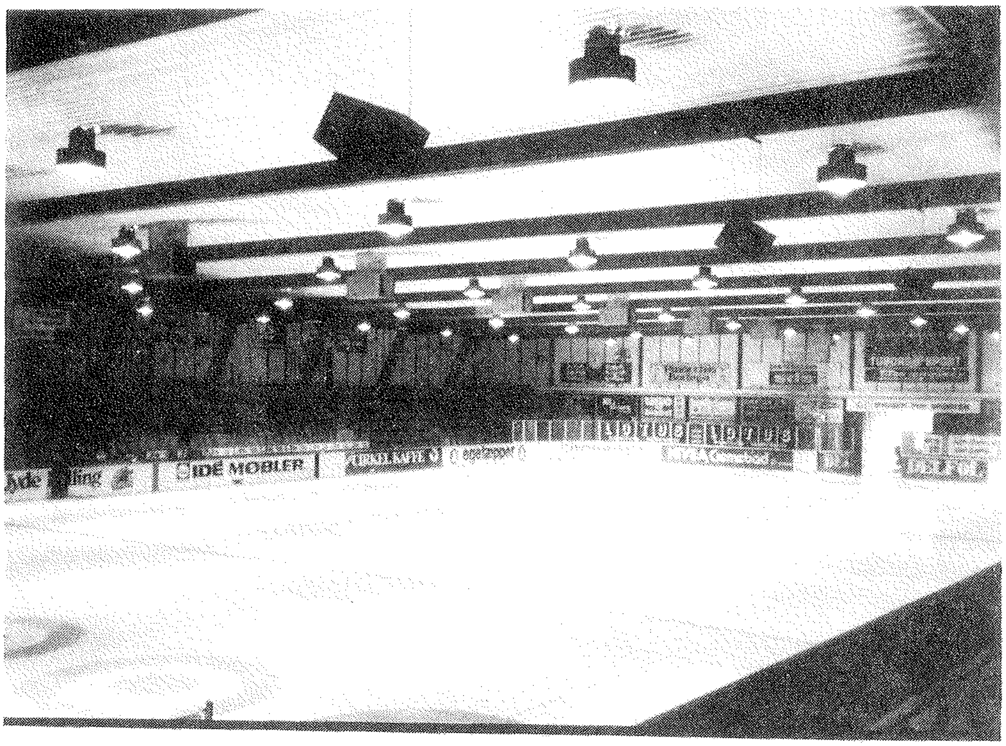

Fig. 2. The icerink at Hørsholm.

3. PHYSICAL MODEL

Figures 1 and 2 show two Danish indoor icerinks at Rødovre and Hørsholm. The simple model of an indoor icerink is shown in Fig. 3.

Two different cases are taken into account depending on the existence of the suspended shield of corrugated bright aluminium plates mounted below the roof construction, i.e. between the roof and ice surfaces. The model of icerink hall with aluminium shield consists of three plates with five active surfaces, i.e. outside and inside surface of the roof, external (facing roof) and internal (facing ice) surface of the aluminium shield as well as the ice surface. The internal air is present in two spaces: between roof and aluminium shield and between aluminium shield and ice. If the aluminium shield does not exist there are two plates with three active surfaces, i.e. outside and inside surface of the roof as well as the ice surface. The internal air is closed in one space. The geometry of the model icerink hall and the numbering of surfaces and spaces involved in both cases, i.e. with and without aluminium shield, are shown in Fig. 3.

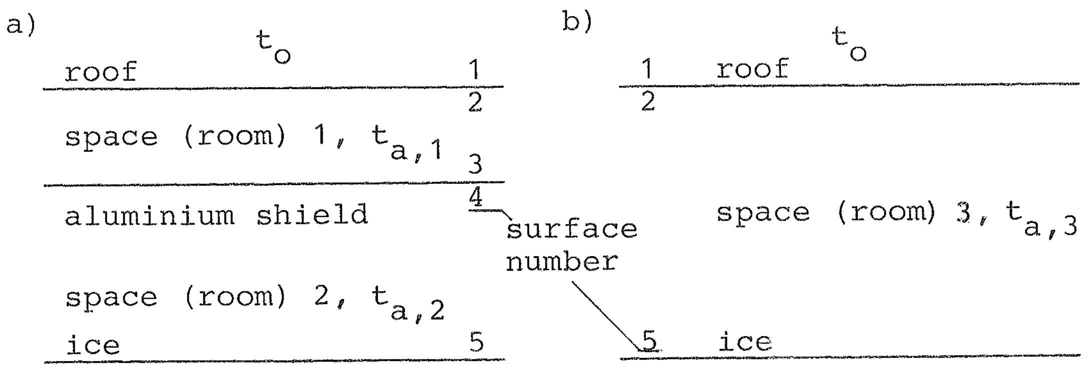

Fig. 3. The geometry of an indoor icerink model
a) with suspended shield of aluminium plates,
b) without aluminium.

It is assumed that roof and aluminium shield have no heat capacity. The roof has a very small heat resistance, the aluminium does not have any. These assumptions hold more or less for the ice rink at Rødovre for which the model was previously defined, since this one is uninsulated, but hardly for Hørsholm ice rink which is better insulated and heated.

The model ice rink hall's thermal system is described by the heat balance equations. For the hall with aluminium shield there are four of them: for surfaces 2, 3 and 4 i.e. inside surface of the roof and aluminium shield as well as for both spaces. For the hall without aluminium there are two heat balance equations: for inside surface of the roof and for the internal air.

These equations form the simple mathematical model of heat transfer in an indoor ice rink. They combine the thermal characteristics, geometry, surfaces and air temperatures, surface excitation components, heat provided to the air and by the air infiltration.

4. MATHEMATICAL MODEL

4.1 Icerink with suspended aluminium shield

4.1.1 Heat balance equation for the inside surface of the roof

Surface 2 exchanges heat convectively with the air flowing between roof and aluminium shield and by radiation with the external surface of aluminium shield. Heat is also conducted to the surface 2 from the roof. At the external surface of the roof heat is exchanged by convection and radiation.

The general form of the heat balance equation for surface 2 at the time τ, is

$$q_{k,2} + q_{r,2} + q_{c,2} + q_{e,2} = 0, \tag{1}$$

where

$q_{k,2}$ - conduction heat flux,
$q_{c,2}$ - convection heat flux,
$q_{r,2}$ - net radiant heat flux due to longwave radiation heat transfer,
$q_{e,2}$ - surface heat flux, e.g. energy input to a unit surface area by radiation from occupants, lighting, etc.

The convection heat transfer flux to the surface 2 is

$$q_{c,2} = h_{c,2}(t_{a,1} - t_2), \tag{2}$$

where

$t_{a,1}$ - temperature of the air in space 1, i.e. between roof and aluminium shield, $^{\circ}C$
t_2 - temperature of the roof inside surface, $^{\circ}C$
$h_{c,2}$ - convective heat transfer coefficient on the surface 2, $W/(m^2 \cdot K)$.

Magnitude of convective heat transfer coefficient h_c depends on the type of convection process. For low air velocities, as these occurring in the icerink hall being analyzed, the heat transfer takes place by free convection, so that h_c is a function of the temperature difference. It is assumed that the value of h_c is given after [2] by

$$h_{c,2} = a\, \Delta t^{0.25}, \tag{3}$$

where $a = 1/0.4$, if the heat is flowing up,
$a = 1/1.7$, if the heat is flowing down.

Heat flux conducted to the surface 2 can be expressed as

$$q_{k,2} = U(t_{sol} - t_2) \tag{4}$$

or

$$q_{k,2} = U\left(t_o + \frac{A_b \cdot I}{h_1} - t_2\right), \tag{5}$$

where

$$U = \left(\frac{1}{h_1} + R_r\right)^{-1}, \tag{6}$$

- overall heat transfer coefficient between outdoor air and roof internal surface, $W/(m^2 \cdot K)$

R_r - roof thermal resistance, $m^2 \cdot K/W$

h_1 - total surface conductance for the roof external surface, $W/(m^2 \cdot K)$

t_{sol} - solar air temperature, ^oC

t_o - outside air temperature, ^oC

A_b - roof absorption coefficient,

I - solar radiation flux on outside surface of the roof, W/m^2.

Outside the building the forced convection is predominant, and convective heat transfer coefficient $h_{c,1}$ is the function of the air velocity. It is assumed that the value of $h_{c,1}$ is described by formulas given in [2]. Furthermore it is assumed that radiation contribution in heat transfer is described in a manner analogous to the convection by using the radiation heat transfer coefficient. The total external surface conductance is defined by one of the two following formulas:

$$h_1 = 10.26 + 4\ v \qquad \text{for } v \leq 5 \text{ m/sec}, \tag{7}$$

$$h_1 = 4.26 + \frac{v^{0.75}}{0.13} \qquad \text{for } v > 5 \text{ m/sec}, \tag{8}$$

where v designates the wind velocity.

The radiant interchange between surfaces in model hall is described by employing the gray body concept. It is assumed that all the surfaces involved are isothermal, uniformly irradiated and have constant radiative properties. As the distances between surfaces are small compared to their size, the radiant heat exchange with the surroundings can be neglected. Then we have

$$q_{r,2} = \frac{\sigma(T_3^{\,4} - T_2^{\,4})}{\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_3} - 1} \quad , \tag{9}$$

where

σ - Stefan-Boltzmann constant, equal to $56{,}697 \cdot 10^{-9}$ $W/(m^2 \cdot K^4)$,

$\varepsilon_2, \varepsilon_3$ - emissivities of the surfaces 2 and 3 respectively,

T_2, T_3 - absolute temperatures of surfaces 2 and 3 respectively, K.

The difference between fourth powers of absolute temperatures T_3 and T_2 can be expressed as follows:

$$T_3^{\,4} - T_2^{\,4} = 2T_{m,2}(t_3 - t_2)\left[2T_{m,2}^{\,2} + \frac{(t_3 - t_2)^2}{2}\right],$$

where $T_{m,2}$ - mean absolute temperature of surfaces 2 and 3, K.

Hence

$$\frac{(t_3 - t_2)^2}{2} << 2T_{m,2}^2$$

is negligible and we have linear approximation

$$T_3^4 - T_2^4 = 4T_{m,2}^3 (t_3 - t_2). \tag{10}$$

Designating

$$W_{2,3} = \frac{4T_{m,2}^3 \sigma}{\frac{1}{\varepsilon_2} + \frac{1}{\varepsilon_3} - 1} , \tag{11}$$

we can write net radiant heat flux due to longwave radiant heat transfer between surfaces 2 and 3 in the form

$$q_{r,2} = W_{2,3}(t_3 - t_2) , \tag{12}$$

where

$W_{2,3}$ - radiative heat transfer coefficient between surfaces 2 and 3 given by equation (11), $W/(m^2 \cdot K)$.

Substitution of (2), (5) and (12) into equation (1) leads to

$$(h_{c,2} + U + W_{2,3}) t_2 - W_{2,3} t_3 - h_{c,2} t_{a,1} = = q_{e,2} + U\ t_o + U \frac{A_b\ I}{h_1} , \tag{13}$$

where

$h_{c,2}$, U, $W_{2,3}$ are given by formulas (3), (6) and (11) respectively,

h_1 is given by (7) or (8) depending on the wind velocity.

4.1.2 Heat balance equation for the aluminium shield

Generally the heat balance for surfaces 3 and 4 at the time τ is given by

$$q_{c,3} + q_{r,3} + q_{e,3} + q_{c,4} + q_{r,4} + q_{e,4} = 0 , \qquad (14)$$

where

$q_{c,3}, q_{c,4}$ - convection heat fluxes,

$q_{r,3}, q_{r,4}$ - net radiant heat fluxes,

$q_{e,3}, q_{e,4}$ - surface excitation heat fluxes.

Using the same approach as previously, we have

$$q_{c,3} = h_{c,3}(t_{a,1} - t_3), \qquad (15)$$

$$q_{c,4} = h_{c,4}(t_{a,2} - t_3), \qquad (16)$$

where

$t_{a,2}$ - temperature of the air in space 2, i.e. between aluminium shield and ice, ^{o}C

t_3 - temperature of the aluminium shield, ^{o}C

$h_{c,3}, h_{c,4}$ - convective heat transfer coefficients at each of the aluminium shield surfaces, i.e. facing roof and facing ice, respectively, given by equation (3), $W/(m^2 \cdot K)$.

And furthermore

$$q_{r,3} = W_{2,3}(t_2 - t_3), \qquad (17)$$

$$q_{r,4} = W_{4,5}(t_5 - t_4), \qquad (18)$$

where

$W_{2,3}$ is given by term (11),

$$W_{4,5} = \frac{4\ \sigma\ T_{m,4}^{3}}{\frac{1}{\varepsilon_4} + \frac{1}{\varepsilon_5} - 1} \quad , \tag{19}$$

- radiative heat transfer coefficient between surfaces 4 and 5, $W/(m^2 \cdot K)$

$T_{m,4}$ - mean absolute temperature of surfaces 4 and 5, i.e. aluminium shield and ice, K

$\varepsilon_4, \varepsilon_5$ - emissivities of the surfaces 4 and 5.

Substituting equations (15), (16), (17), (18) into (14) and arranging yields

$$-W_{2,3}t_2 + (h_{c,3} + h_{c,4} + W_{2,3} + W_{4,5})t_3 - h_{c,3}t_{a,1} - h_{c,4}\ t_{a,2} = q_{e,3} + q_{e,4} + W_{4,5}\ t_5 \quad , \tag{20}$$

since the ice temperature is assumed to be known.

4.1.3 Heat balance for the air

Heat is transferred to and from internal air by convection depending on the temperature difference between the air and surrounding surfaces, by convection of a part of the heat which is given off from people, lighting, etc., as well as by ventilation and infiltration.

For the air enclosed in space 1 we can write this in the form

$$C_1(t_{a,1}^{\tau} - t_{a,1}^{\tau-d\tau})/d\tau = A\ h_{c,2}(t_2 - t_{a,1}) + A\ h_{c,3}(t_3 - t_{a,1}) + c\ G_1(t_o - t_{a,1}) + Q_{k,1} \quad , \tag{21}$$

where

$C_1 = V_1 \cdot \rho \cdot c$ - heat storage capacity of the air in space 1, J/K

V_1	- volume, m^3
ρ	- the air density, kg/m^3
c	- the specific heat of air, J/(kg·K)
A	- the surface area, m^2
G_1	- the infiltration and ventilation air quantity, kg/s
t_o	- the outdoor air temperature, ^oC
$Q_{k,1}$	- heat transferred to the air from people, lighting, etc., W
$d\tau$	- time step, s
$\tau, \tau - d\tau$	- superscripts designating the level of time when the terms are evaluated. Those quantities without superscripts are evaluated at time level τ.

After arrangement of the above equation we have

$$- A\, h_{c,2}\, t_2 + \left[\frac{C_1}{d\tau} + A(h_{c,2} + h_{c,3}) + c\, G_1\right] t_{a,1}$$

$$- A\, h_{c,3}\, t_3 = c\, G_1\, t_o + \frac{C_1}{d\tau}\, t_{a,1}^{\tau - d\tau} + Q_{k,1}\, . \tag{22}$$

Writing the heat balance for the air in space 2 in a manner analogous to the equation (21) and arranging it, gives

$$- A\, h_{c,4}\, t_3 + \left[\frac{C_2}{d\tau} + A(h_{c,4} + h_{c,5}) + c\, G_2\right] t_{a,2} =$$

$$= c\, G_2\, t_o + \frac{C_2}{d\tau}\, t_{a,2}^{\tau - d\tau} + A\, h_{c,5}\, t_5 + Q_{k,2}\, , \tag{23}$$

where

$h_{c,5}$ is convective heat transfer coefficient on the ice surface, given by formula (3).

4.1.4 Final set of equations

Finally we have a system of four heat balance equations which result at each time step. In matrix form we can express this result as

$$
\begin{bmatrix}
\frac{C_1}{d\tau} + A(h_{c,2} + h_{c,3}) + c\,G_1 & 0 & -A\,h_{c,2} & -A\,h_{c,3} \\
0 & \frac{C_2}{d\tau} + A(h_{c,4} + h_{c,5}) + c\,G_2 & 0 & -A\,h_{c,4} \\
-h_{c,2} & 0 & h_{c,2} + U + W_{2,3} & -W_{2,3} \\
-h_{c,3} & -h_{c,4} & -W_{2,3} & h_{c,3} + h_{c,4} + W_{2,3} + W_{4,5}
\end{bmatrix}
$$

$$
\begin{bmatrix} t_{a,1} \\ t_{a,2} \\ t_2 \\ t_3 \end{bmatrix}
=
\begin{bmatrix}
cG_1 t_o + \frac{C_1}{d\tau}\, t_{a,1}{}^{\tau-d\tau} + Q_{k,1} \\
cG_2 t_o + \frac{C_2}{d\tau}\, t_{a,2}{}^{\tau-d\tau} + Ah_{c,5} t_5 + Q_{k,2} \\
q_{e,2} + Ut_o + U\,\frac{A_b \cdot I}{h_1} \\
q_{e,3} + q_{e,4} + W_{4,5} t_5
\end{bmatrix}
\qquad (24)
$$

or in more compact form as

$$\mathbf{B} \cdot \mathbf{t} = \mathbf{u} \qquad (25)$$

where $\mathbf{t}$ represents the unknown column vector of temperatures, $\mathbf{u}$ represents the column vector of values which are known or which are functions of unknown vector $\mathbf{t}$, and $\mathbf{B}$ is the matrix of coefficients which are dependent on the unknown vector of temperatures.

4.2 Icerink without aluminium shield

For an icerink without aluminium shield we have only two heat balance equations for the inside surface of the roof and for the air. In matrix form we can express this result as

$$\begin{bmatrix} \frac{C_3}{d\tau} + A(h_{c,2} + h_{c,5}) + cG_3 & -Ah_{c,2} \\ -h_{c,2} & h_{c,2} + U + W_{2,5} \end{bmatrix} \begin{bmatrix} t_{a,3} \\ t_2 \end{bmatrix}$$

$$= \begin{bmatrix} cG_3 t_o + \frac{C_3}{d\tau} t_{a,3}^{\tau-d\tau} + Q_{k,3} + Ah_{c,5}\, t_5 \\ q_{e,2} + Ut_o + U\frac{A_b I}{h_1} + W_{2,5}\, t_5 \end{bmatrix} \qquad (26)$$

or as before

$$\mathbf{B} \cdot \mathbf{t} = \mathbf{u}$$

where values C, G, Q_k, t_a with subscript 3 are analogous to the previously defined, evaluated for the whole volume of air in icerink

$$W_{2,5} = \frac{4\delta\, T_{m,5}^3}{\frac{1}{e_2} + \frac{1}{e_5} - 1},$$

- radiative heat transfer coefficient between surfaces 2 and 5, $W/(m^2 \cdot K)$

$T_{m,5}$ - mean absolute temperature of surfaces 2 and 5, K.

5. METHOD OF SOLUTION

The convective heat transfer coefficients h_c are dependent on the unknown vector of temperatures and on the direction of the heat flow (formula (3)), the coefficients W of radiation heat transfer depend on the unknown temperatures. Thus the systems of equations (24) and (26) are nonlinear and cannot be solved directly. On the other hand, the number of equations is very small. In such cases direct techniques are useful and assuming negligible computational round-off error, they give exact solution in a finite number of operations.

Taking above into account, the method using both iterative and direct techniques was chosen. At each time step the convective heat transfer coefficients for all the indoor surfaces involved, total surface conductance for outside surface of the roof and mean absolute temperatures are evaluated as functions of vector of temperatures computed in the previous time step. Then the system of equations is solved directly and all coefficients and mean temperatures are corrected. The above procedure is repeated until each element of the vector of differences between new and old vectors of solution is smaller than a given value.

At each iteration the set of equations is solved directly employing the Gauss-Jordan reduction method with maximum pivot strategy. This method was chosen instead of other methods of solving the systems of linear equations since it involves the smallest computational round-off error.

First approximation of temperatures beginning the computations is evaluated employing total surface conductances for all the surfaces involved instead of using separate values for convective and radiative heat transfer coefficients as it is done in the next iterations. Then, the mean absolute temperatures, convective and radiative heat transfer coefficients are computed, and an ordinary procedure of solution starts.

6. SIMULATION PROGRAM

The problem being considered and its solution method have been programmed for a digital computer using FORTRAN language. The program which simulates thermal processes taking place in the icerink hall is described in detail in a separate report [3]. It can be utilized in some different ways regarding changes in the model itself as well as simulated period of time.

a) Concerning model of icerink it may have or does not have an aluminium shield mounted below roof construction in a certain distance from the roof.

b) Concerning period of time, there are two possibilities:
   - simulation of thermal conditions during the specified period of time. In this case program utilizes the weather data from the Danish Test Reference Year.
   - one day computations of quasistationary conditions. Then the Test Reference Year is not used and program utilizes the weather data specified by the user.

The program calculates temperatures of the air, of the inside surface of the roof and the aluminium shield, if the last one is assumed to exist. Next, it compares the roof or aluminium shield temperature with dew-point temperature of the outside air to check the possibility of water vapour condensation.

The ice surface is assumed to have a constant temperature, its value is written in BLOCK DATA.

The program calculates radiation heat fluxes absorbed by all surfaces involved, convective heat flux on the ice surface and cooling load needed for keeping the temperature of the ice surface on the desired, constant level. Computed cooling load is the one owing to heat transfer by radiation with roof or aluminium shield and convection with the internal air; the other modes of heat transfer connected with condensation and sublimation of water vapour on the ice surface, radiation with other surfaces, etc. are not taken into account. The energy input by radiation from occupants, lighting, heating etc. can be added.

The program consists of a main program and nine subprograms.

7. RESULTS OF SIMULATION

The results of simulation of the real heat transfer processes taking place in the icerink at Rødovre are presented below. An inside view of this icerink is shown in fig. 1. The dimensions of it are:

- The length and width of the ice surface are 60 m and 30 m respectively.
- The height of the icerink varies from 2 m at the sides to 9 m at the center.
- The aluminium shield is mounted below the roof construction at a distance which varies from approx. 0.9 m to 1.5 m.
- The length and width of the aluminium shield are 64 m and 30 m respectively.
- The distance between the aluminium shield and the surface of the ice varies from 4.1 m to 7.8 m due to the aluminium shield being cylindrical.

According to the physical model of the icerink shown in fig. 3 it is assumed that the roof, the aluminium shield and the ice are parallel plates and all with the same dimensions, namely 62 x 30 m equal to 1860 m^2. They are placed at the following distances:

- The distance between the ice surface and the aluminium shield is 4.1 m and the distance between the aluminium shield and the ceiling is 1 m.

If there is lack of further information, standard data for any of the computations are as follows:

- The ventilation or/and infiltration rate is the same in both spaces, i.e. between the roof and the aluminium shield (space 1) and between the aluminium shield and the ice (space 2); it is equal to one change of air per hour.
- No heat is transferred to the air from persons, lighting, etc.

- No long-wave radiation flow from persons, lighting, heating, etc. is absorbed by any of the surfaces in question.
- The roof absorption coefficient $A_b = 0.85$.
- The emissivity of the bright aluminium is equal to 0.05, the emissivity of the ice, of the painted surface of the aluminium shield and the ceiling is equal to 0.95 [3].
- The heat resistance of the roof is equal to 0.01 $m^2 \cdot K/W$ as it is assumed to be uninsulated.

The four most important basic cases were taken into account and compared:

- The model of an icerink with a suspended shield of aluminium plates where the surface facing the roof was unpainted.
- The same model but where the surface facing the roof was painted to increase its absorptivity.
- The model of an icerink without the aluminium shield and with a ceiling made from ordinary building material.
- The same model with a bright aluminium foil glued to the ceiling surface.

Furthermore the influence of the quantity of ventilation air, the emissivity of the aluminium, the different modes of heat input to the model icerink and the heat load on the ice surface were checked.

Several executions of the simulation program are summarized below.

I. The icerink model with aluminium shield:

1. Aluminium shield unpainted.
2. Aluminium shield painted.
3. Aluminium shield unpainted, different ventilation and/or infiltration rates.
4. Aluminium shield unpainted, different intensities of the radiant heat flow absorbed by the aluminium.

5. Aluminium shield unpainted, the heat transferred to the air enclosed in both spaces of the icerink.

6. Aluminium shield unpainted, different values of the aluminium emissivity.

II. The icerink model without aluminium shield:

7. Standard data.

8. Aluminium foil glued to the ceiling surface.

The results of the computations outlined above are shown in figs. 4-16 and in table 1. Figs. 4 and 7-12 illustrate the computed number of hours during which the water-vapour condensation occurred for any of the ten-day periods during the whole ice-skating season, that means between September 1 and March 31 for different cases of an icerink maintenance. Figs. 13 and 14 indicate the heat transfer between the surface of the ice and the surroundings due to the radiation heat exchange with the aluminium shield or with the ceiling, as well as convection with the air enclosed in space 2 or in space 3 depending on the icerink model for four basic model cases. This heat transfer is equal to the part of the cooling load needed to keep the temperature of the ice at the desired, constant level of $-5^{o}C$.

Table 1 gives the rounded-off values of these parts of cooling loads as well as their sums for any of the months in question and for the whole season.

Figs. 15 and 16 are illustrating the general comparison between the number of hours with water-vapour condensation and comparison between the cooling loads needed for keeping the temperature of the ice at the level of $-5^{o}C$ - due to the heat transfer, as before - during the whole season for the four basic model cases in question as well as for different values of the aluminium shield emissivity, the ventilation air quantity and the different modes of heat input to the icerink.

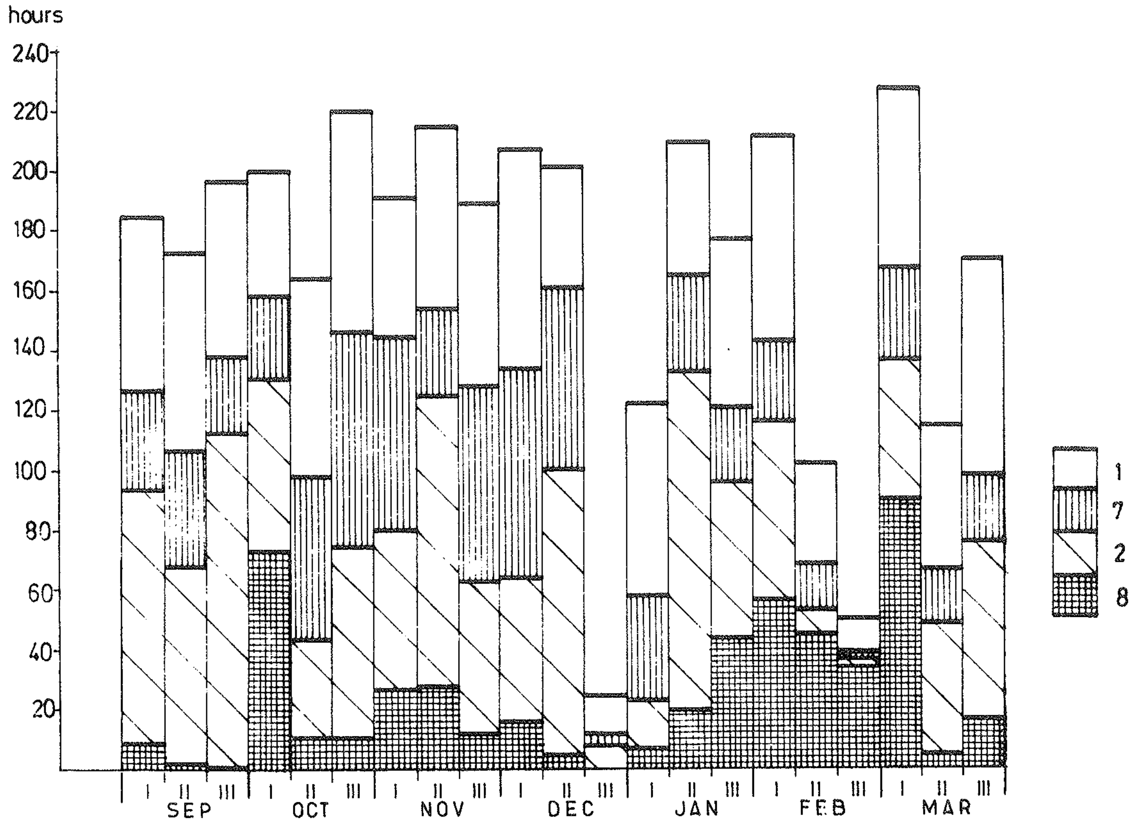

Fig. 4. The number of hours during which the water-vapour condensation occurred for four model cases of an ice rink fabric:

1 - with aluminium shield unpainted,

2 - with aluminium shield painted (top side),

7 - without aluminium shield,

8 - with aluminium foil glued to the ceiling surface.

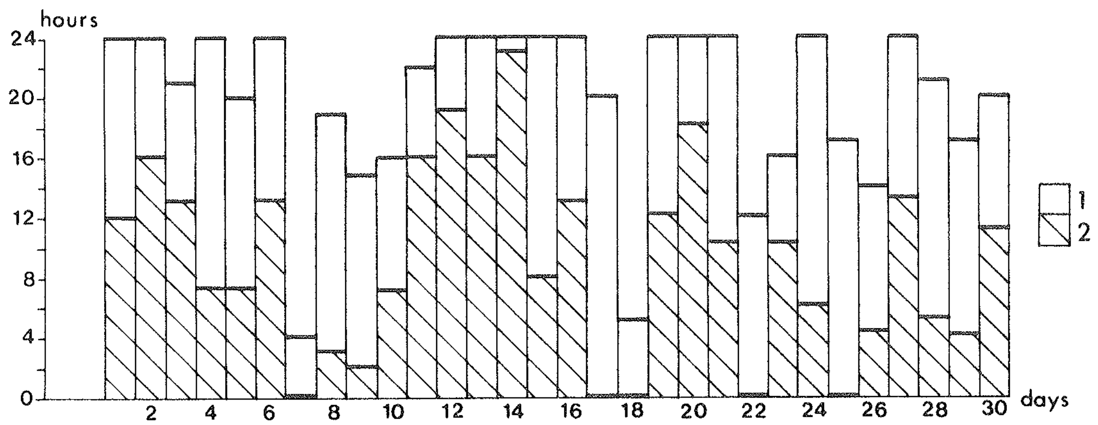

Fig. 5. The number of hours with water vapour condensation for any of the days in November and for:
1 - aluminium shield unpainted,
2 - aluminium shield painted (top side).

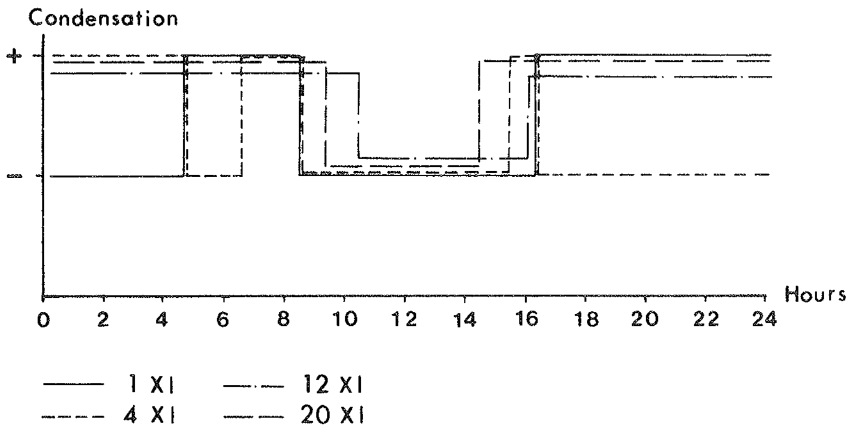

Fig. 6. The water-vapour condensation occurrence during few days in November for the model icerink with aluminium shield painted.
+ condensation occurs,
- it does not.

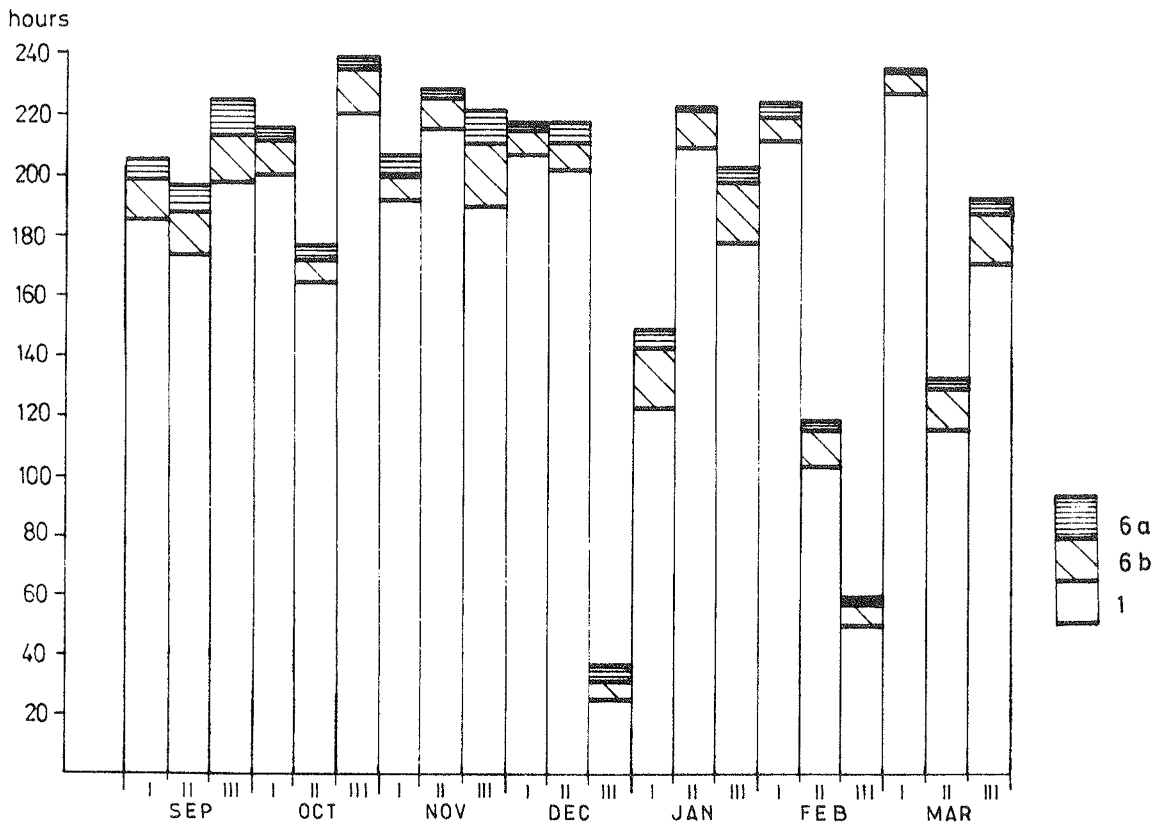

Fig. 7. The influence of the aluminium shield emissivity on the number of hours with water-vapour condensation for the model icerink with aluminium shield unpainted.

1 - emissivity equal to 0.05,

6a- emissivity equal to 0.25,

6b- emissivity equal to 0.15.

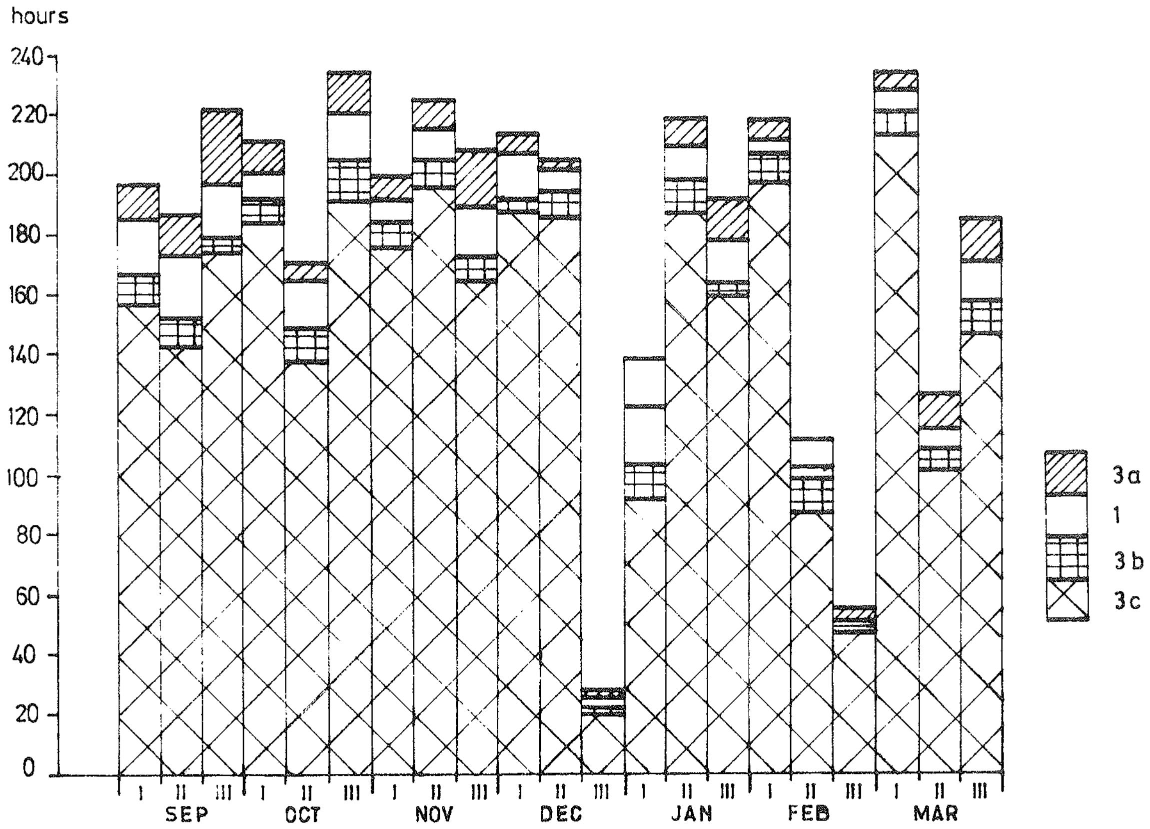

Fig. 8. The influence of the ventilation rate on the number of hours with water-vapour condensation for the model icerink with the aluminium shield unpainted. Ventilation at the rate of:

1 - 0.7 kg/s in the space 1 (between the roof and aluminium shield) and 2.7 kg/s in the space 2 (between the aluminium shield and ice) i.e. 1 change of the air per hour in both spaces,

3a- 0.3 kg/s in the space 1 and 1.4 kg/s in the space 2 i.e. 0.5 change per hour in both spaces,

3b- 0.7 kg/s in the space 1 and 5.5 kg/s in the space 2 i.e. 1 and 2 changes of the air in spaces 1 and 2, respectively,

3c- 0.7 kg/s in the space 1 and 8.2 kg/s in the space 2, i.e. 1 and 3 changes per hour, respectively.

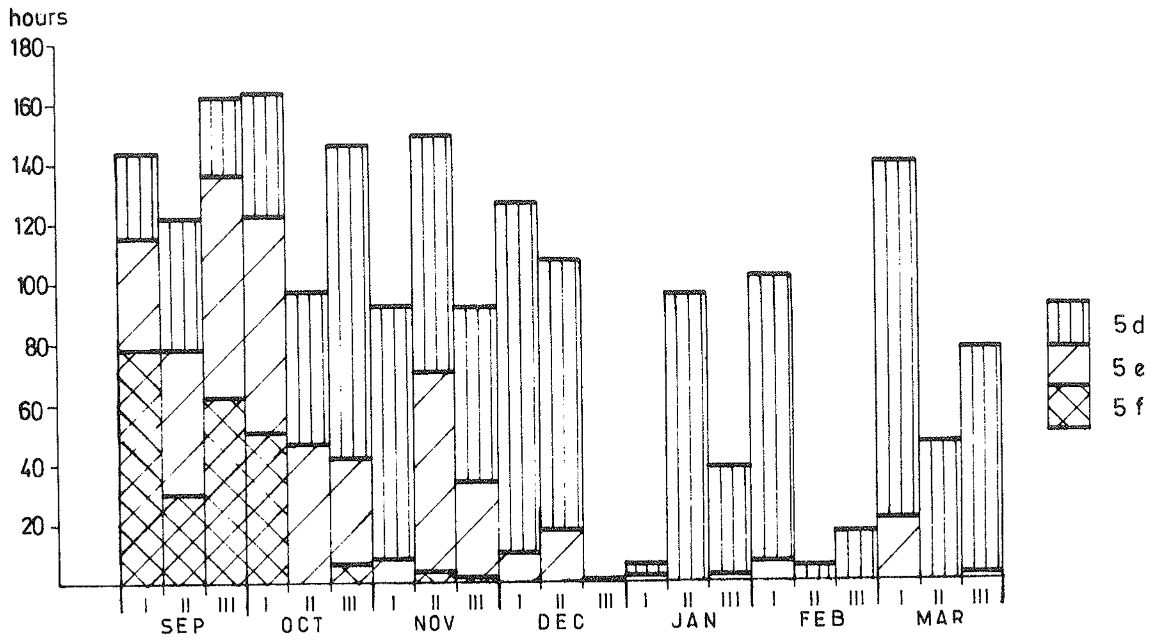

Fig. 9. The number of hours with the occurring condensation as a function of the heat energy transferred to the air between roof and aluminium shield (i.e. to the space 1) for the model icerink with the aluminium shield unpainted.

5d - heat input $Q_{k,1}$ = 18600 W, i.e. 10 W/m^2,

5e - $Q_{k,1}$ = 37200 W, 20 W/m^2,

5f - $Q_{k,1}$ = 55800 W, 30 W/m^2.

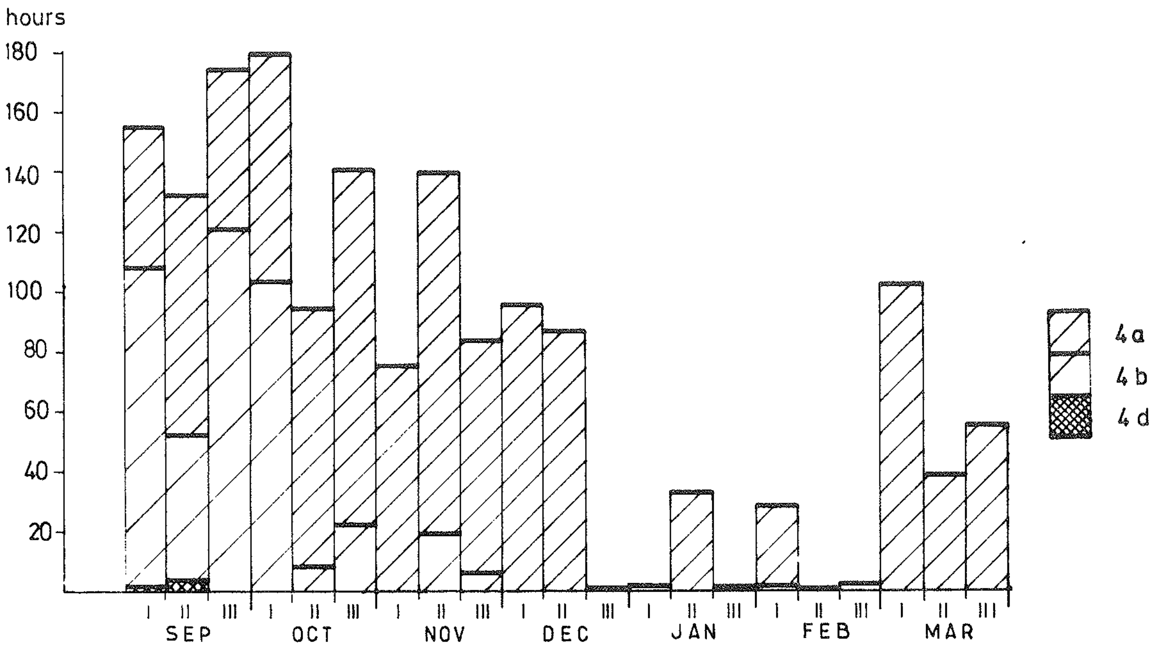

Fig.10. The number of hours with the occurring condensation as a function of the intensity of the radiant heat flux, $q_{e,3}$, absorbed by the aluminium shield.

4a - $q_{e,3} = 3$ W/m^2,

4b - $q_{e,3} = 6$ W/m^2,

4d - $q_{e,3} = 10$ W/m^2.

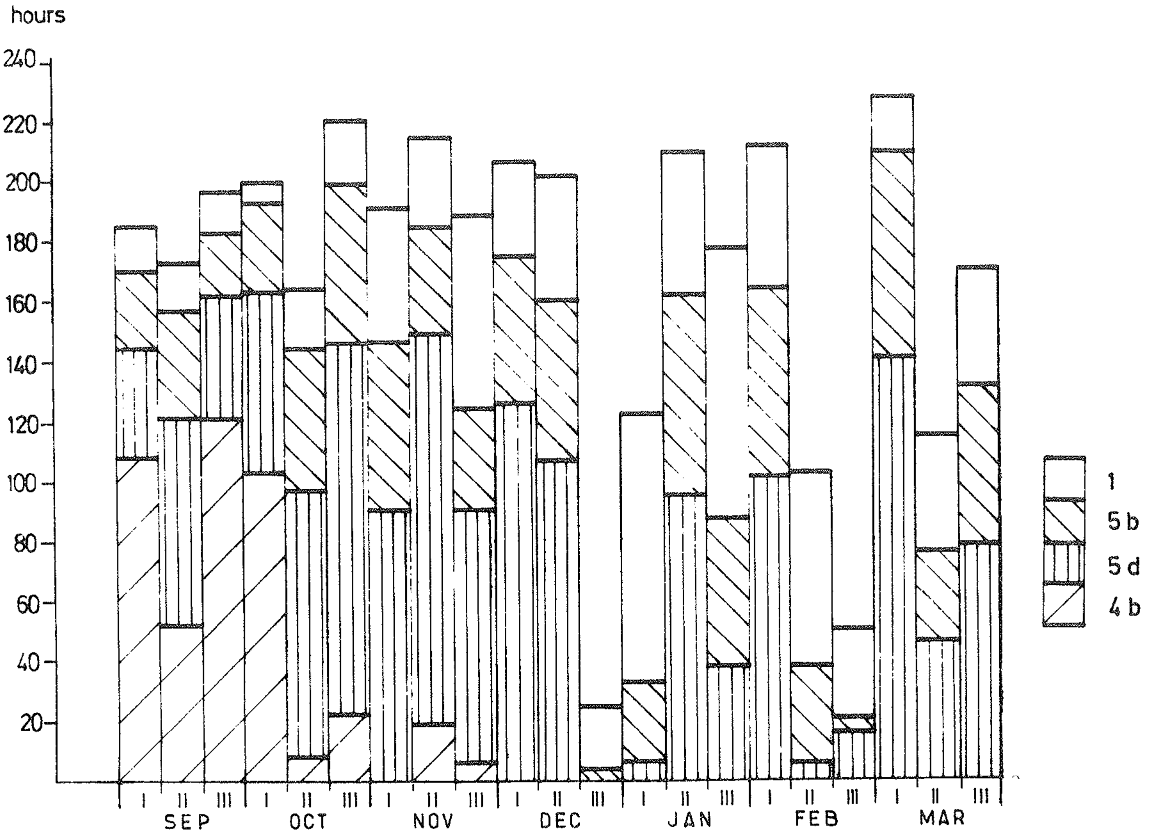

Fig. 11. The influence of different modes of heat input to the model icerink with an aluminium shield on the number of hours during which condensation occurred.

1 - no heat input,

4b- radiant heat flux absorbed by the aluminium shield $q_{e,3} = 6\ W/m^2$,

5b- heat transferred to the air enclosed in space 2, the heat input $Q_{k,2} = 11160\ W$, i.e. $6\ W/m^2$,

5d- heat transferred to the air in space 1, $Q_{k,1} = 18600\ W$, i.e. $10\ W/m^2$.

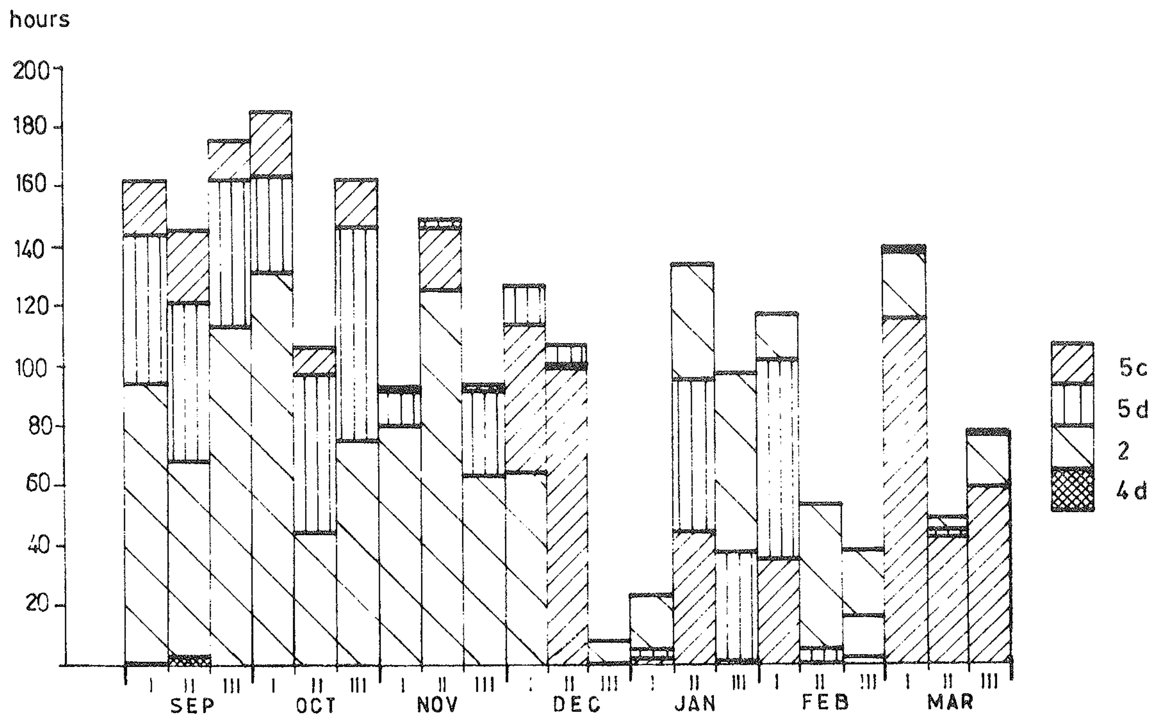

Fig.12. Comparison of the effect of painting the top of the aluminium shield with the effect of different modes of the heat input to an icerink on the number of hours with water-vapour condensation. Heat input to a unit surface area equal to 10 W/m^2.

2 - aluminium shield painted,

4d- radiation on the aluminium shield,

5c- heat transferred to the air enclosed in the space 2 (between the ice and aluminium shield surfaces),

5d- heat transferred to the air in the space 1 (between the roof and aluminium shield).

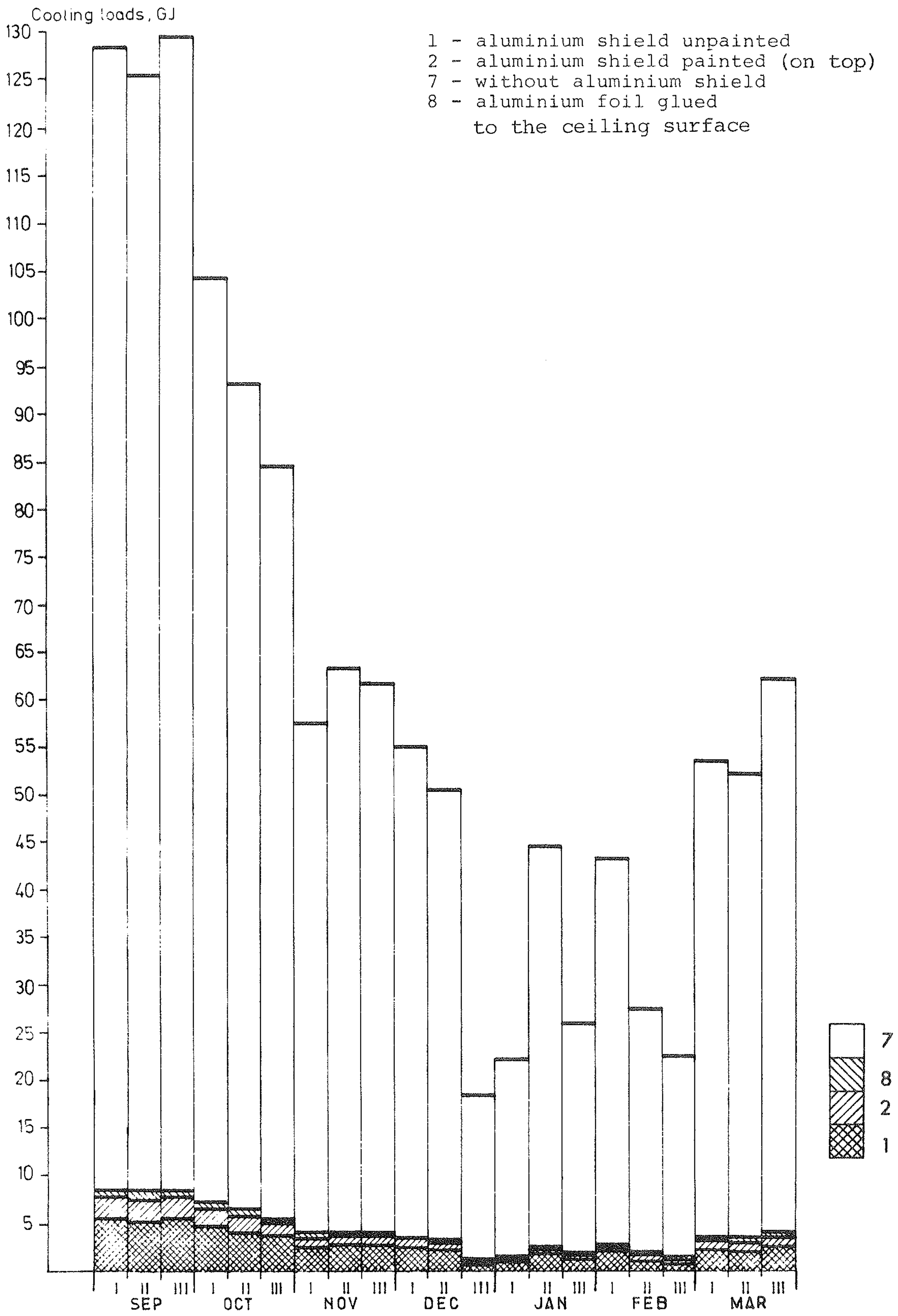

Fig. 13. Heat load on the ice surface due to the heat exchange by radiation between the aluminium shield or the inside surface of the roof, for four model cases of an icerink.

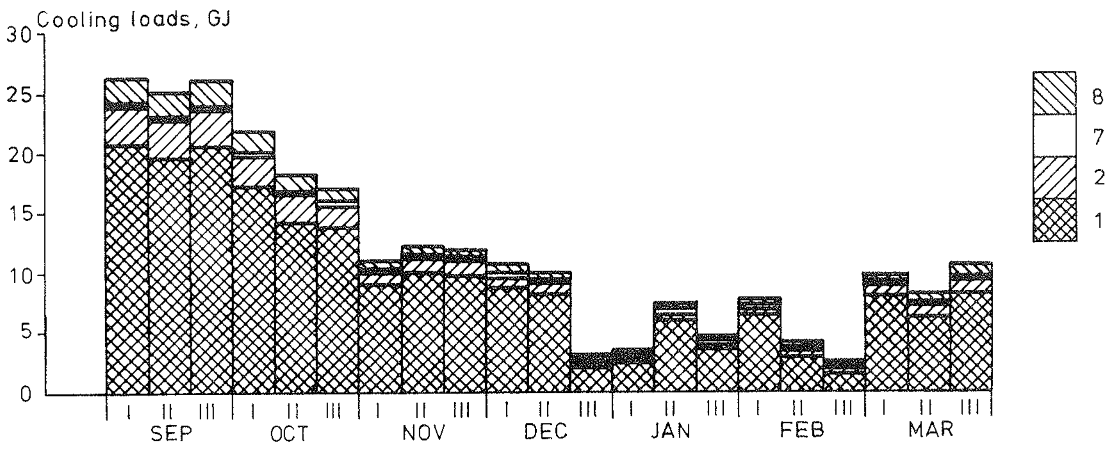

Fig.14. Heat gainings on the ice surface due to the heat transfer by convection with the surrounding air, for four model cases of an icerink:

1 - with aluminium shield unpainted,

2 - with aluminium shield painted (on top)

7 - without aluminium shield,

8 - with aluminium foil glued to the ceiling.

MONTH	ALUMINIUM SHIELD unpainted, (1)			ALUMINIUM SHIELD painted, (2)			ALUMINIUM GLUED to the roof, (8)			WITHOUT ALUMINIUM, (7)		
	RAD	CONV	TOT	RAD	CONV	TOT	RAD	CONV	TOT	RAD	CONV	TOT
SEP	16	61	77	23	70	93	26	77	103	384	71	455
OCT	12	45	58	17	52	69	19	57	76	281	52	334
NOV	8	29	37	11	32	43	12	35	47	182	33	216
DEC	5	19	25	7	22	29	8	23	32	124	22	146
JAN	4	13	16	5	14	19	6	15	21	89	14	103
FEB	4	11	15	5	13	19	6	14	20	93	13	106
MAR	7	23	29	10	26	36	11	28	39	168	27	194
TOTAL	56	201	257	79	229	308	88	251	339	1321	232	1554

Table 1. Cooling load needed to keep the ice temperature at the desired level of $-5^{\circ}C$ owing to the heat transfer with aluminium shield or roof by radiation, convection with the surrounding air and totally, GJ.

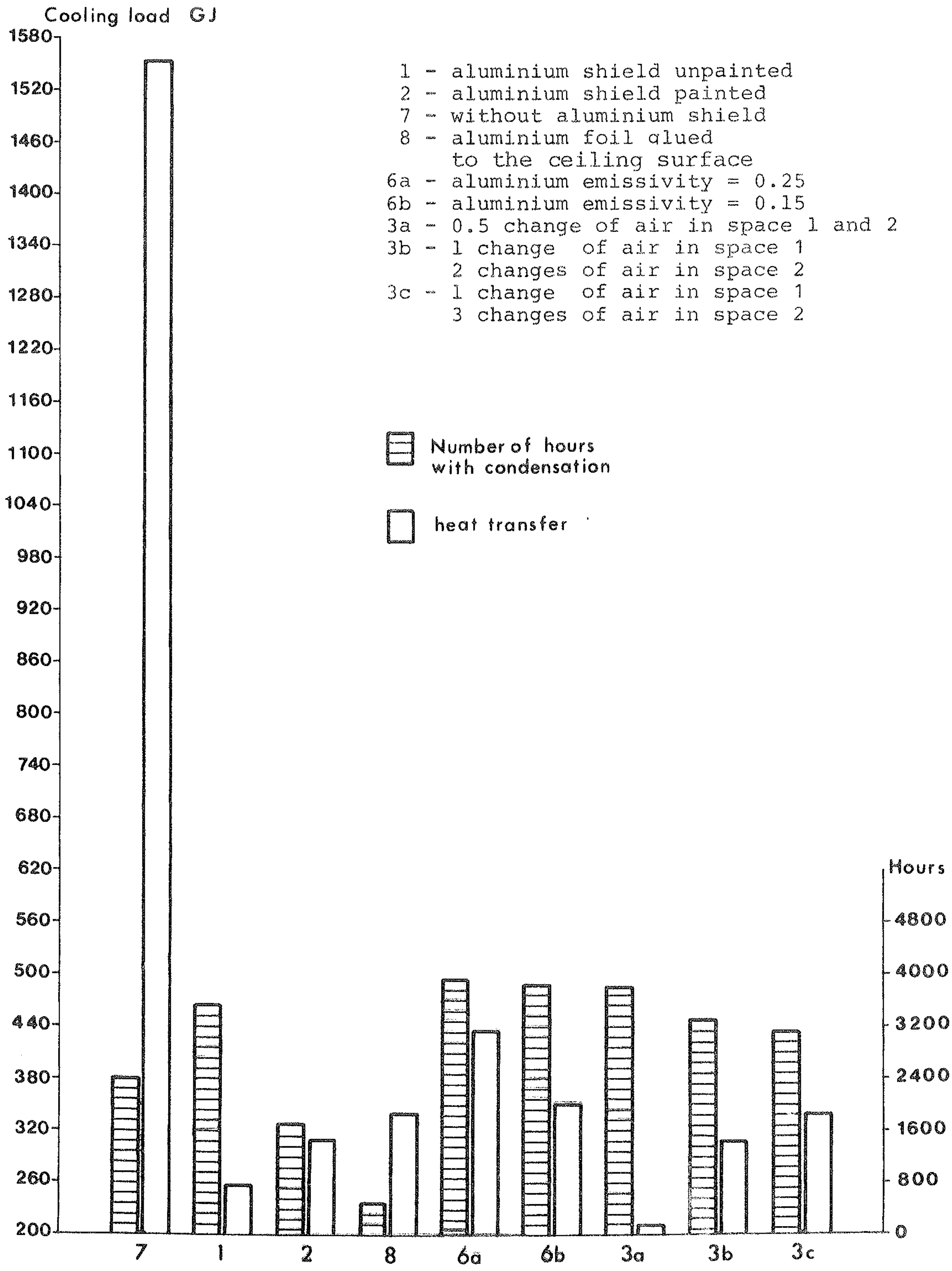

Fig.15. The number of hours with condensation as well as the cooling loads needed for keeping the ice temperature at the desired, constant level -5°C due to the heat transfer with surrounding air and with the ceiling or aluminium shield during the whole season.

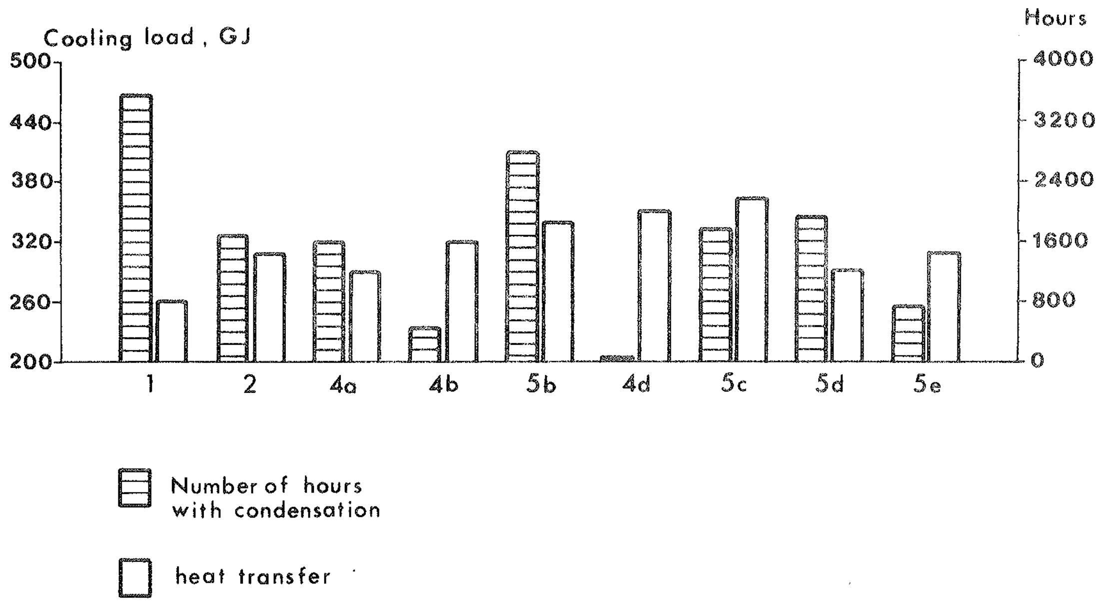

Fig. 16. The number of hours with condensation as well as the cooling loads needed for keeping the ice temperature at the desired, constant level of $-5^{o}C$ due to the heat transfer with surrounding air and with the ceiling or aluminium shield during the whole season for:

1 - an icerink with aluminium shield unpainted,

2 - an icerink with aluminium shield painted,

4a - radiant heat flux absorbed by aluminium shield $q_{e,3} = 3\ W/m^2$,

4b - $q_{e,3} = 6\ W/m^2$,

4d - $q_{e,3} = 10\ W/m^2$,

5b - heat transferred to the air in the space 2 $Q_{k,2} = 11\ 160\ W$ or $6\ W/m^2$,

5c - $Q_{k,2} = 18\ 600\ W$ or $10\ W/m^2$,

5d - heat transferred to the air in the space 1 $Q_{k,1} = 18\ 600\ W$ or $10\ W/m^2$,

5e - $Q_{k,1} = 37\ 200\ W$ or $20\ W/m^2$.

8. CONCLUSIONS

The results of computations presented above allow for comparison of the influence of different ways of decreasing water-vapour condensation in the icerink on the final effect. Assuming that the heat transfer between the ice surface and the walls, as well as the other surfaces not involved in the icerink model, does not change considerably, depending on different cases in question, the results of simulation allow, simultaneously, for comparison between the cooling loads needed for keeping the temperature of the ice at the desired, constant level.

The conclusions from such a comparison are summarized below.

The emmissivity of the aluminium shield has influence on the number of hours with water-vapour condensation and on the cooling load; both of them are increasing as a function of the increase in the emissivity value (see figs. 7 and 15). However, the increase in the cooling load is more pronounced. It grows to 169% of the initial value when the emissivity changes from 0.05 to 0.25, whereas the number of hours with occurring condensation grows to 110%.

Because of the simplicity of the model used one should bear in mind that the number of hours with condensation only can be taken as an index for the probability of condensation, as the emissivity of the bright aluminium changes drastically when condensation starts, which will further the condensation.

The ventilation rate effect on the number of hours with condensation as well as on the cooling load depends on the fact which space is ventilated. The quantity of air flowing through space 1 has negligible influence on the thermal conditions in the icerink. Growth in the ventilation rate from 1 to 3 in space 2 causes an increase in the cooling load to 130% of its initial value. Simultaneously, the number of hours with water-vapour condensation decreases to 88%, (see figs. 8 and 15).

The influence of the heat input to the icerink on its thermal parameters in question depends considerably on the way heat is delivered inside, as it can be readily seen from figs. 9-12 and 16. The most effective mode of heat transfer to the ice rink is irradiation of the aluminium shield. However, this, as well as the other modes of heat input, results in a significant increase in the energy consumption. Firstly, by using energy themselves and secondly, by increasing the cooling load needed to keep the ice surface temperature at the desired level.

The aluminium shield mounting has a big influence on the energy consumption. It reduces the cooling load due to the heat transfer by radiation between the ice surface and roof to 4% of its initial value (see fig. 13 and table 1) and heat transfer by convection to 86% (see fig. 14), totally to 17% of the initial cooling load (see fig. 15). However, it simultaneously causes increase in the condensation problem. The number of hours with water-vapour condensation becomes higher; it reaches 146% of the initial value.

Painting of the aluminium shield surface facing the roof construction reduces the number of hours with condensation to 47% of that for the icerink with aluminium shield unpainted and to 69% of the number of hours with condensation for the icerink model without aluminium shield. Unfortunately, the effect of the aluminium shield painting on the cooling load is the opposite one; it grows to 120% of the cooling load needed in the icerink model with aluminium shield unpainted. Since this value is still only 20% of the cooling load needed in the case of the aluminium shield non-existent, this model can be regarded as a good solution of the problem under consideration.

Another possible way to solve the problem is to apply a bright aluminium foil to the ceiling surface.
This possibility gives a slightly bigger value of the cooling load than an icerink model with the aluminium shield painted (110% of this value), but decreases considerably the number of hours with water-vapour condensation. This number for the whole ice-skating season is 21% of that for the model without aluminium shield, 31% of that for the model with the aluminium shield painted and 15% of that for the model with aluminium shield unpainted.

Having in mind all the uncertainties connected with the physical processes going on in the ice rink, as well as the simplicity of the models considered, the results of the computations presented above and conclusions from these results can still indicate the main tendencies in the thermal behaviour of the ice rink in question. They show clearly the high profits from using the highly polished aluminium surfaces in indoor ice rinks.

Both of the checked possibilities of applying such surfaces, i.e. mounting a suspended shield of aluminium plates with painted surface facing the roof, or applying aluminium foil to the ceiling can be recommended as good, practical and very economical solutions for a correct maintenance of an indoor ice rink.

It should be pointed out that in practice the aluminium shield will prevent the often heavy condensation and dripping from the roof supporting system. Condensation on the aluminium shield can be avoided by utilizing the ice surface as a dehumidifier by a slightly periodic stirring of the air by means of a few ceiling fans. The fans are started when the dew point of the air in the ice rink is close to the temperature of the shield. The fans can also be utilized to prevent or remove fog over the ice surface. Ceiling fans have been installed with good results in the uninsulated and unheated ice rink in Rødovre.

As it can be readily seen from the values of the cooling loads in table 1, they allow for a saving of 1246 GJ (346 MWh) (the model with aluminium shield) or 1215 GJ (338 MWh) (aluminium applied on the ceiling surface) of energy during the whole ice-skating season lasting seven months. These numbers state a significant cutting down on the energy consumption.

One should notice that in a heated ice rink the heating load to keep a certain temperature will be reduced with the same quantity as the cooling load.

LIST OF FIGURES

REFERENCES

[1] Korsgaard, V.: Dansk pat.ans. nr. 194/82.

[2] Korsgaard, V.: "Energibesparende loft med lavt emissionstal til skøjtehaller. Low emissivity ceilings for energy savings in ice skating rinks". Thermal Insulation Laboratory, Technical University of Denmark. July 1984.

[3] Forowicz, T.: "Program ICEH for calculation of temperatures and cooling loads in an indoor ice rink". Users Guide. Thermal Insulation Laboratory, Technical University of Denmark. Report no. 85-13, December 1984.

[4] Korsgaard, V.: "Varmetransmission", Varmeisolering 1. Noter til kursus 6401. Thermal Insulation Laboratory, Technical University of Denmark. January 1981.

[5] Whitaker, S.: "Fundamental Principles of Heat Transfer". Pergamon Press, Inc., New York, 1977.