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DEMAND FOR HEAT FROM ROCK MASS AND BOREHOLE HEAT EXCHANGERS FOR A VENTILATION SYSTEM IN CASE OF THE AUDITORIUM AT THE FACULTY OF DRILLING, OIL AND GAS AT AGH UST IN KRAKOW****

1. INTRODUCTION

The analysis of energetic and economic efficiency of heating and cooling systems based on borehole heat exchangers is currently a subject of research at the Faculty of Drilling, Oil and Gas at AGH University of Science and Technology in Krakow. For this purpose the Laboratory of Geo-energetics was created from the very beginning and equipped with five 78-meter-deep borehole heat exchangers, each of which has a different construction [17, 18]. Two heat pumps are used for heating the auditorium, basing on low-temperature heat from the rock mass. In summer they produce coolness for the air conditioning, which is simultaneously a process of regeneration of heat resources in the rock mass. The heat produced during the production of coolness is inserted into the rock mass. Systems of this kind are being increasingly utilised because of the ability to rationalise energy management at facilities requiring cooling and heating.

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The calculations of heat demand for the ventilation system resulted in disparate results, depending on the assumed external temperature. Variants of hourly, daily and monthly average temperatures measured with a dry-bulb thermometer were considered.

2. CHARACTERISTICS OF THE FACILITY

The auditorium, which the ventilation system was designed for, is located in an adapted former storage room in a connector of A3 and A4 pavilions on the premises of AGH UST in Krakow. The facility is used by the Faculty of Drilling, Oil and Gas at AGH. The amphitheatrical hall (Fig. 1), descending towards the speaker, has 150 seats. The main exit leads through a corridor, which is also included in the ventilation system, to the A4 pavilion. The three fourth of the room has a basement. The total cubature of the auditorium amounts to 1,000.72 m^3, out of which the part located underneath the amphitheatrical set of seats is not a subject of the ventilation project and, simultaneously, is unavailable. Consequently, the cubature of the auditorium including the vestibule amounts to 814 m^3. The air change rate for the hall amounts to 7 air changes per hour, whereas for the vestibule it is 1.5.

Fig. 1. The interior of the auditorium, heated and cooled using the rock mass

It needs to be highlighted that the auditorium is greatly fitted with windows, which significantly influences its thermal balance. The total window area amounts to 74.6 m^2, whereas the exterior wall area is 210 m^2 [1].

Fresh air for the auditorium is provided by an air supply and exhaust central unit by CLIMA-PRODUKT company, placed in the basement (Fig. 2).

Fig. 2. Fragment of the ventilation central unit with a rotary exchanger for heat recuperation

3. ORIENTATION OF THE FACILITY

Figures 3 and 4 present the orientation of the auditorium in space. The contour of the room was marked with the continuous line. The area of northward transparent walls amounts to A_0 = 24 m^2. The area of southward transparent walls amounts to A_0 = 50.6 m^2. The heat transfer coefficient for transparent walls is U_0 = 2.0 W/(m^2·K).

Fig. 3. Northern elevation

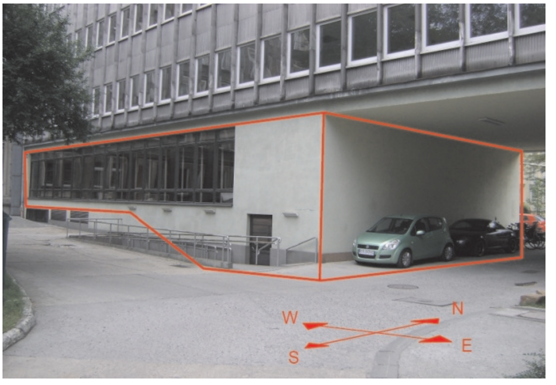

Fig. 4. Southern and eastern elevation

4. CALCULATION PRINCIPLES

The following calculation principles were assumed. They result from the construction of the auditorium, its manner of use and its location.

4.1. External walls

28 m^2 of the floor in the auditorium is based on ground of total transfer coefficient of U = 0.417 W/(m^2·K). Under the remaining part of the auditorium and above the ceiling there are rooms of regulated temperature, therefore heat transfer between those rooms were not considered in the calculations.

The northern window area is 24 m^2, and the area of windows and glass door on the southern side amounts to 50.6 m^2. The transfer coefficient for windows and the front door was set at the level of U = 2.0 W/(m^2·K). The remaining area of external walls amounts to 210 m^2 with the transfer coefficient of U = 0.397 W/(m^2·K).

4.2. Internal walls

The auditorium shares the western wall with the main pavilion of the A4 building, where the temperature is regulated. The wall was not included in the energy calculations.

4.3. Central heating system

In heating season (t_z < 13°C) the auditorium is heated by a central heating system of 28 kW powered by the MPEC City Heating Energetics Company.

4.4. Volumetric flowrate of air for the auditorium and the vestibule

Within the project of the installation [2, 3] the required (peak) volumetric flowrate of air was determined by two methods. The first, calculated on the basis of the cubature of the room and the vestibule, assumes a sevenfold change of air per hour for the auditorium and 1.5 change for the vestibule. The second – assumes a supply of 30 m^3/h of fresh air per user. The results of the calculations were presented in Table 1.

Table 1

The calculated volumetric flow of air [2, 3]

	Determined volumetric flowrate, m^3/h
Method I	5320
Method II	4320

Eventually, the bigger flowrate of 5320 m^3/h was used for further calculations.

4.5. Ventilation working time

It was assumed, that the ventilation shall be working daily from 8 am to 6 pm, including summer (vacation) time.

4.6. Average temperature of surroundings

Having used data put on the Ministry of Infrastructure website for Krakow-Balice meteorological station, three variants were analysed:

a) calculations were carried out in 1 hour interval, assuming the 30-year average temperature for every hour of the year as the external temperature,
b) the temperature on a given day was calculated as a daily average,
c) a dry-bulb average monthly temperature [16] was used for calculations.

4.7. Radiation intensity

Values of the solar radiation intensity, depending on the space orientation of the wall and its location, were taken from data put on the Ministry of Infrastructure's website for Krakow-Balice meteorological station. For every variant, gains from the radiation were calculated in an hourly interval. The auditorium location between A3 and A4 pavilions, which limits exposure to sun, was taken into consideration.

4.8. Other conditions

1. Calculations of heating and cooling power demand were prepared in compliance with norms [6–13, 15].
2. Calculations of the energy demand exclude energy recovery from the rotary recuperator.
3. The number of people in the auditorium was set at 150 and the cubature of the room at 814 m^3.

5. ENERGY BALANCE OF THE AIR FLOW IN THE VENTILATION SYSTEM

PN-EN ISO 13790 norm [14] introduces an hourly method of calculating the heat demand for heating and cooling of a building, based on an electric analogy of a lumped heat capacity method. The calculation procedure, the detailed characteristics of the used model and modifications of the method were numerously described in the trade literature [4, 5].

To estimate the amount of heat for the ventilation system for the auditorium, being the basis of the economic analysis of borehole heat exchangers made at the Faculty of Drilling, Oil and Gas at AGH UST, an original, simplified calculation method was applied.

It was assumed, that the temperature inside the auditorium is 20°C. To simplify the calculation, gains from people were also calculated for the final temperature inside the auditorium. Heating gains of 28 kW in the heating season were added to the balance when the temperature outside is not higher than 13°C.

5.1. Balance of the energy flow irrespective of the internal temperature

Gains from lighting:

$$\dot{Q}_{const1} = 840 \text{ W} \tag{1}$$

Heating gains from MPEC:

$$\dot{Q}_{const2} = 28 \text{ kW} \tag{2}$$

Gains from people (overt and covert):

$$\dot{Q}_{const3} = \varphi \cdot n \cdot \left(q_j + w_j \cdot c_{pp} \cdot t\right), \text{ W} \tag{3}$$

where:

- φ – people presence coefficient, assumed $\varphi = 1$, –,
- n – amount of people, assumed 150, –,
- q_j – unit overt heat flowrate delivered to the surroundings by one person for the temperature of 20°C, assumed according to tables $q_j = 85$ W,
- w_j – unit steam flowrate delivered to the surroundings by a human depending on activity and the temperature of the surroundings, assumed according to tables $w_j = 4.1$ g/h,
- t – internal temperature, $t = 20$°C,
- c_{pp} – specific heat of steam at constant pressure, $c_{pp} = 41840$ J/(kg·K).

Heating gains from sunlight:

$$\dot{Q}_{const4} = \sum_i A_{oi} \cdot I_{sun_i}, \text{ W} \tag{4}$$

where:

A_i – area of a transparent wall of i, m^2 orientation,

I_{sun_i} – solar radiation intensity, taken from Krakow-Balice meteorological station, on the wall of i orientation under a 90° angle, W/m^2,

i – direction of orientations of transparent walls, for the described facility only: North and South.

In the further part of the paper, specific parameters dependent on the internal temperature were assumed.

5.2. Balance of the energy flow dependent on the internal temperature

Heat gains (losses) caused by the transfer through the ground:

$$\dot{Q}_{dt1} = A_{pod} \cdot U_{pod} \cdot (8°\text{C} - t_w), \text{ W} \tag{5}$$

where:

A_{pod} – area of the floor on the ground, m^2,

U_{pod} – transfer coefficient for floor on the ground, W/(m^2·K),

t_w – internal temperature, °C.

Heat gains (losses) caused by the transfer through non-transparent walls:

$$\dot{Q}_{dt2} = A_p \cdot U_p \cdot (t_z - t_w), \text{ W} \tag{6}$$

where:

A_p – area of vertical non-transparent walls, m^2,

U_p – heat transfer coefficient for non-transparent walls, W/(m^2·K),

t_z – external temperature, °C.

Heat gains (losses) caused by the transfer through transparent walls:

$$\dot{Q}_{dt3} = \sum_i A_{oi} \cdot U_o \cdot (t_z - t_w), \text{ W} \tag{7}$$

where U_o – heat transfer coefficient for transparent walls, 2 W/(m^2·K).

Thermal balance of the air flowrate pumped through the ventilation system:

$$\dot{Q}_{dt4} = \dot{m} \cdot c_p \cdot (t_z - t_w), \text{ W} \tag{8}$$

where:

$\dot{m}$ – mass flowrate of pumped air of density of 1.2 kg/m^3,

c_p – specific heat of dry air, 1020 J/(kg·K).

6. ENERGY DEMAND FOR THE VENTILATION SYSTEM

The internal temperature inside the room with no heating (cooling) of the pumped air was calculated iteratively, using the schematics given in Figure 5. The TOLL error tolerance was set at 0.5°C.

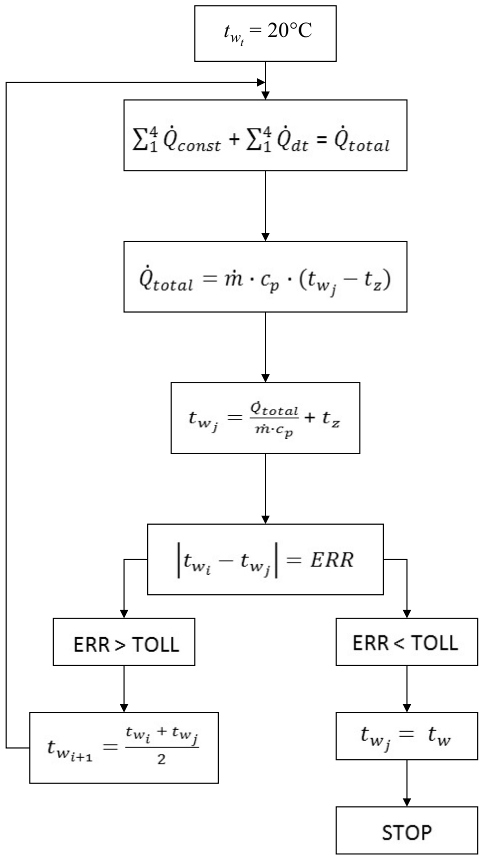

Fig. 5. Block schematics of the method of iterative calculation of the internal temperature

Having the iteratively calculated internal temperature, the heat demand was calculated with formula (8), and the coolness demand with formula (10):

$$\dot{Q}_{heat} = \dot{m} \cdot c_p \cdot (20°C - t_w), \text{ W} \tag{9}$$

$$\dot{Q}_{cool} = \dot{m} \cdot c_p \cdot (t_w - 20°C), \text{ W} \tag{10}$$

After increasing (decreasing) the temperature outside to 20°C, additional losses (gains) of energy occur, presented in equations (11) and (12):

$$\dot{Q}_{Add.heat} = \sum_{1}^{3} \dot{Q}_{dt(20°-t_z)} - \sum_{1}^{3} \dot{Q}_{d(t_w-t_z)},\ \mathrm{W} \tag{11}$$

$$\dot{Q}_{Add.cool} = \sum_{1}^{3} \dot{Q}_{dt(t_z-20°)} - \sum_{1}^{3} \dot{Q}_{d(t_z-t_w)},\ \mathrm{W} \tag{12}$$

Hence, the total energy demand for the ventilation system shall amount to (13) and (14):

$$\dot{Q}_{total.heat} = \dot{Q}_{warm} + \dot{Q}_{Add.heat},\ \mathrm{W} \tag{13}$$

$$\dot{Q}_{total.cool} = \dot{Q}_{cool} + \dot{Q}_{Add.cool},\ \mathrm{W} \tag{14}$$

7. RESULTS

In the paper the following calculations were made: the accumulated demand for coolness, the accumulated demand for heat, and the accumulated total demand for energy as an algebraic sum of the demand for heat and coolness, depending on the assumed external temperature. The calculations were made with an hourly change, between 8 am and 6 pm, from 1 January to 31 December. The results of the calculations were presented in Figures 4–6 and Table 2. The relative yearly demand in percentage expresses the relation between the yearly demand of specific external temperatures and the yearly demand calculated with the use of hourly external temperature.

Table 2

Collation of the yearly demand for coolness, heat, and coolness and heat for the ventilation system

	Yearly energy demand			
Average external temperature	coolness, MWh	heat, MWh	total, MWh	relative, –
Hourly	64.5	37.0	101.4	1.00
Daily	47.4	41.9	89.3	0.88
Monthly	39.1	39.5	78.6	0.78

The obtained results were shown in Figures 6–8, depending on the method of averaging the atmospheric air temperature.

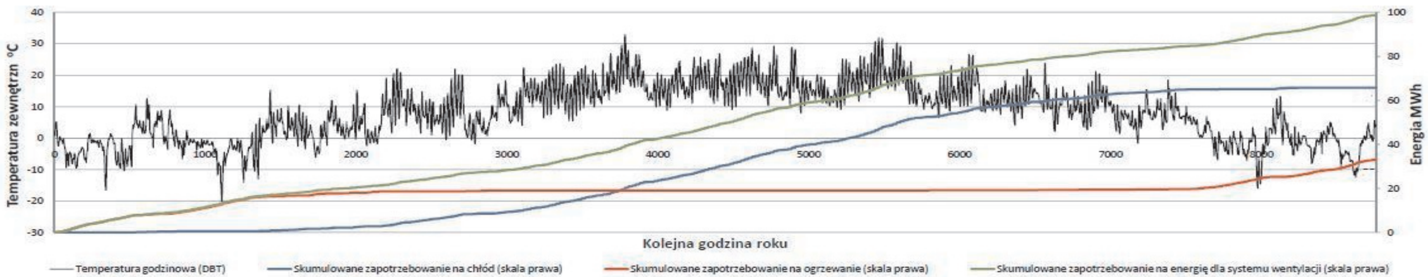

Fig. 6. Accumulated demand for heat, coolness and total energy for a yearly exploitation of the auditorium calculated while taking into account the hourly external temperature

Fig. 7. Accumulated demand for heat, coolness and total energy for a yearly exploitation of the auditorium calculated while taking into account the daily external temperature

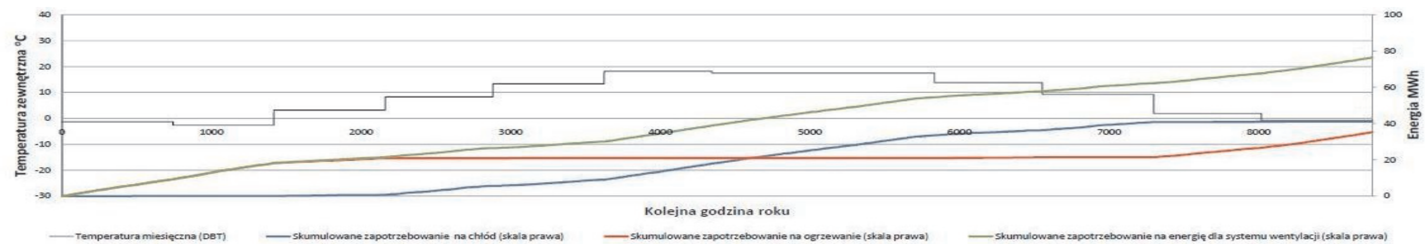

Fig. 8. Accumulated demand for heat, coolness and total energy for a yearly exploitation of the auditorium calculated while taking into account the monthly external temperature

8. CONCLUSIONS

1. The article clearly shows significant differences in the demand for heat and coolness for the ventilation system in summer and winter, and the total demand for heat, depending on the assumed external temperature. The total heat demand, estimated according to the hourly average is 22% greater than the one calculated according to the monthly average temperature.
2. As a result of exploiting the auditorium, the central heating convector radiator system was adjusted with thermostats. Lack of automatic control over the central heating system during heating season (t_z < 13°C), with 150 people in the hall, at hours of peak radiation intensity lead to a high internal temperature and consequently to the air conditioning turning on. The situation where the central heating and air conditioning systems were working simultaneously was incompatible with the power conservation rule and generated additional exploitation costs, which was resolved.
3. The detailed energy balance is used to determine profitability of using a rotary recuperator and covering the demand for heat and coolness through its use and by heat pumps and borehole heat exchangers.
4. The heat and the coolness are delivered firstly from the rotary exchanger (heat and coolness recuperation), the heater/cooler powered by heat pumps from the Laboratory of Geo-energetics and from the peak (emergency) source in the form of a heater using city heating.
5. The next step towards further energy saving will be a system of regulating the fresh air flowrate dependent on the number of people in the auditorium. The measurement will be a CO_2 sensor in the used (exhaled) air flow. Currently, the multiplicity of air changes does not rely on the amount of the users in the room.

NOMENCLATURE

$\dot{Q}_{const1}$ – heat gains from lighting, W
$\dot{Q}_{const2}$ – heating from municipal district heating, W
$\dot{Q}_{const3}$ – heating gains from people, W
$\dot{Q}_{const4}$ – heating gains from sunlight, W
$\dot{Q}_{dt1}$ – heat gains (losses) caused by the transfer through the ground, W
$\dot{Q}_{dt2}$ – heat gains (losses) caused by the transfer through non-transparent walls, W
$\dot{Q}_{dt3}$ – heat gains (losses) caused by the transfer through transparent walls, W
$\dot{Q}_{dt4}$ – thermal balance of the air flowrate pumped through the ventilation system, W
$\dot{Q}_{heat}$ – heat demand, W
$\dot{Q}_{cool}$ – cool demand, W

$\dot{Q}_{Add.heat}$ – additional losses of energy, W

$\dot{Q}_{Add.cool}$ – additional gains of energy, W

$\dot{Q}_{total.heat}$ – total heat demand for the ventilation system, W

$\dot{Q}_{total.cool}$ – total cool demand for the ventilation system, W

φ – people presence coefficient, assumed $\varphi = 1$, –

n – amount of people, assumed 150, –

q_j – unit overt heat flowrate delivered to the surroundings by one person for the temperature of 20°C, assumed according to tables q_j = 85 W

w_j – unit steam flowrate delivered to the surroundings by a human depending on activity and the temperature of the surroundings, assumed according to tables w_j = 4.1 g/h,

t – internal temperature, t = 20°C

c_{pp} – specific heat of steam at constant pressure, c_{pp} = 1840 J/(kg·K)

A_i – area of a transparent wall of i, m^2 orientation

I_{sun_i} – solar radiation intensity, taken from Krakow-Balice meteorological station, on the wall of i orientation under a 90° angle, W/m^2

i – direction of orientations of transparent walls, for the described facility only: North and South

A_{pod} – area of the floor on the ground, m^2

U_{pod} – transfer coefficient for floor on the ground, W/(m^2·K)

t_w – internal temperature, °C

A_p – area of vertical non-transparent walls, m^2

U_p – heat transfer coefficient for non-transparent walls, W/(m^2·K)

t_z – external temperature, °C

U_o – heat transfer coefficient for transparent walls, 2 W/(m^2·K)

$\dot{m}$ – mass flowrate of pumped air of density of 1.2 kg/m^3

c_p – specific heat of dry air, 1020 J/(kg·K)

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