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Claude-A. Roulet and Jakobus van der Maas,
Solar Energy and Building Physics Research Laboratory
Swiss Federal Institute of Technology
Lausanne, Switzerland

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SUMMARY

A study was performed in an industrial hall heated by several propane radiant heaters located all around the hall at mid-height. The purpose of the study was to determine the advantages of this system when compared to convective heaters, and to determine its limits of application.

For that purpose, a minimum instrumentation was installed in the building. This includes an automatic data logger, recording indoor temperatures, moisture content, and meteorological data on site. The energy consumption was manually recorded each week.

CO_2 concentrations were also measured on a single day during and after working hours, and with heaters on and off. The interpretation of these measurements with an appropriate two zone model provided an estimate of internal and external air flow rates.

Other measured data were used together with various simplified models to quantify various unknown quantities. In particular, the specific heat loss, the required peak power and important dynamic thermal characteristics of the hall are be directly deduced from the measurements. It is also shown that an importnat advantage of radiant heaters is that their heat is mainly transmitted to the floor during morning boost heating after night set back. This allows the comfort temperature to be reached earlier, with a lower air temperature, hence lower ventilation heat loss.

Introduction

The purpose of the study, performed in 1991 and yet only published in a report [1], was to determine the performance of a radiative heating system for large industrial halls. In particular, the comfort condition in the hall was examined and the actual energy use was compared to that of the same hall equipped with other heating systems. Therefore, temperature fields, air change rate and indoor air quality were measured.

Description of the building

The measured industrial hall is a metallic building 120 m long, 38 m wide and 15 m high (Fig. 1). Most of the volume is a large open space in which metallic building elements are prepared by blowtorch cutting, arc welding and painting. The gallery is used for storage. A relatively small enclosure, located under this gallery, separately ventilated and heated, is used for office work.

The hall is naturally ventilated and heated by 70 gas radiant heaters located at 5 m height all around the hall. These heaters are catalytic propane burners, 570 mm diameter, providing 10 kW each. Sixty percent of the heat is radiated to surfaces in front of the burner and 40% is transmitted to the air, in a plume rising over the heater.

Measurement strategy

The heating system, used by intermittence, interacts with the building, and the thermal behaviour of the former depends on the other. Therefore, the study of the heating system should include a survey of the building. Measurements were performed on the building to reach three goals:

- to complete the data base required to make a simplified model of the energetic behaviour of the hall,
- to fit the free parameters of this model,
- to examine the temperature gradients and air movements within the hall.

For that purpose, a minimum instrumentation was installed in the building. This includes an automatic data logger, recording as well indoor temperatures and moisture as meteorological data on site; and a few instruments manually recorded each week. The weekly measurement include:

3 electrical energy counters, for power, lighting and office area,
3 propane flow meters, for heating, blowtorch cutting machine and other blowtorches, and
1 degree-hour meter, giving the integrated indoor-outdoor temperature difference.

The long time measurement, performed between January and April 1991, included half hour average values of:

- relative humidity of air outside and in the hall, at 1 and 12 m height,
- global horizontal solar radiation,
- wind velocity and direction,
- air temperature (protected from radiation) outside, in the hall at 10 locations and in an office room.

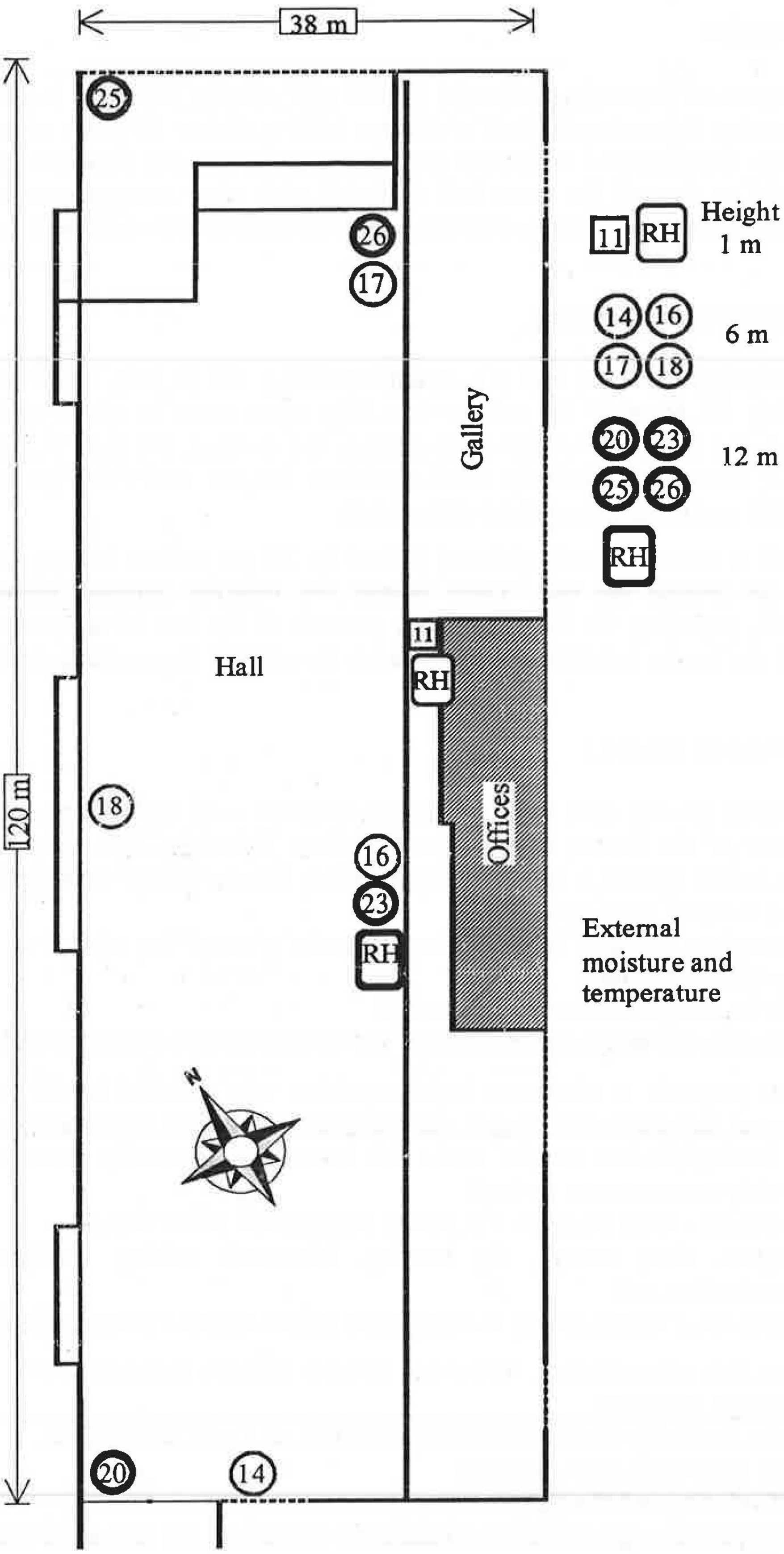

Fig. 1: Plan of the measured industrial hall, showing the location of temperature (numbered) and air relative humidity (RH) sensors.

The locations of temperature sensors in the hall (figure 1) were carefully chosen according to theory of experimental planning [2]. This allowed us to determine the temperature gradients through the hall with a minimum number of measured points.

The measurements were completed by a one day measurement of air flow rates and concentration of some contaminants.

Indoor Air Quality and Air flow Patterns

Most visitors entering the hall would judge that the air quality is very poor. The air is loaded with dust and organic compounds coming from arc welding and painting. The hall is ventilated only by door openings. The painting area is equipped with a ventilation system sucking up the air loaded with solvents. This air is not, however, blown outside, but re-circulated in the hall, since the local work health authority accepted the resulting volatile organic compounds concentration.

Measurements were performed in order to get some information on air change rate and air flow pattern in the hall. In such a large hall (60'000 m³), the tracer gas techniques available in our laboratory could not be used, since these would require too much tracer gas (about 600 l/hour), and a tremendous work to install the necessary piping. The cost of the experiment would be too large with respect to the need for information. PFT techniques could be used, but these were not available by then in Switzerland.

We used therefore the natural tracer gas sources existing in the hall: the propane burners. Their propane flow rate was known by meter readings, and the concentration of carbon dioxide resulting from the combustion of propane was measured by two analysers at 1.5 m height at the centre of the hall and at three heights (1, 6 and 12 m) at 3 m from the facade close to the centre of the hall. There are no other significant CO_2 sources in the hall.

Measurements began at 17:00 on February 20. The open main door of the hall was closed at 18:00. From this time, the heating (hence the CO_2 generation) was reduced stepwise by the thermostat, which shuts down the heaters successively. At 20:30 h, the heating was completely stopped. Figure 2 and tables 1 and 2 show the measured data.

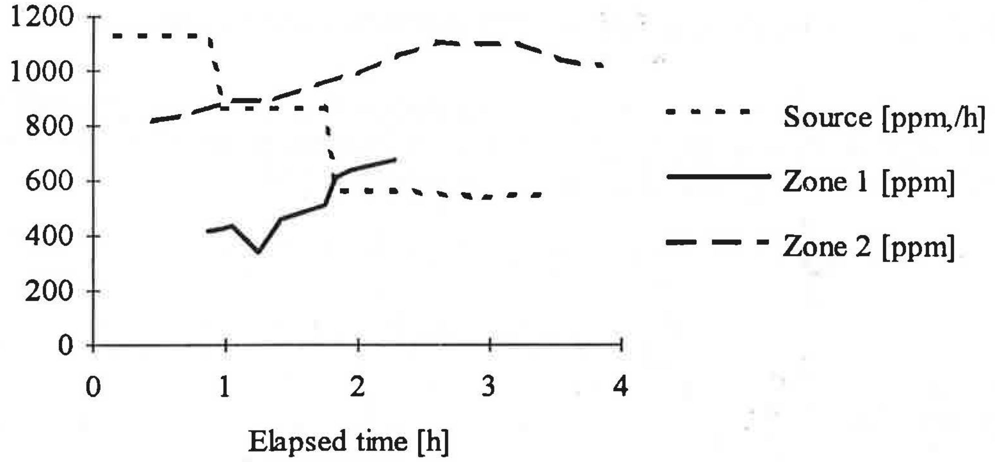

Fig. 2: CO_2 source strength and resulting concentrations in zones 1 and 2 versus time.

Table 1: Measured gas and CO_2 flow rates and CO_2 concentrations.

Time [h]	Flow rates		CO_2 [ppm] indoors-outdoors		
from 17:10	Propane [m³/h]	CO_2 [ppm/h]*	1 m height	6 m height	12 m height
0.75	7.86	1131	417	836	799
0.87	7.86	1131	427	871	825
0.98	6.00	864	436	903	879
1.05	6.00	864	340	923	871
1.25	6.00	864	458	981	920
1.42	6.00	864	510	1030	960
1.75	6.00	864	612	1070	1040
1.83	3.91	563	637	1105	
1.95	3.91	563	674	1102	
2.28	3.91	563		1100	
2.92	3.75	540		1041	
3.45	3.8	547		1019	

* the CO_2 concentration in ml/h is divided by the hall volume in m³ to obtain a flow rate in ppm/h.

As long as the heaters are on, carbon dioxide concentrations are not the same at lower and higher levels. Therefore, measured data are interpreted using a two zone model of the hall, which is shown in figure 3. This model assumes that both zones have an homogeneous concentration of tracer gas. In fact, measurement of CO_2 concentration at 6 m and 12 m height show that the heater plumes mix quite well the air volume above their level, but the homogeneity was not controlled in the lower zone.

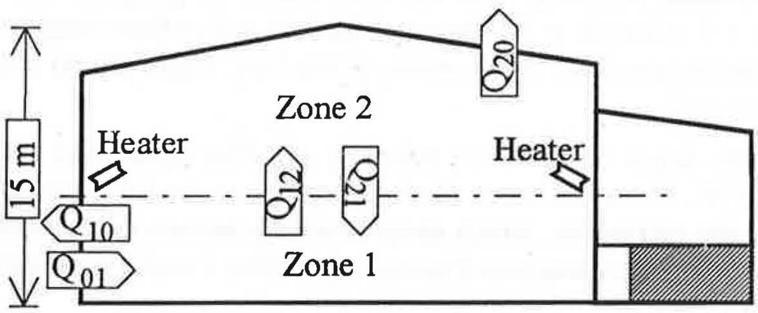

Fig. 3: The two zone model used for interpretation of tracer gas measurements.

Since this high hall is submitted to a strong stack effect, it was also assumed that there is no infiltration in the upper zone, but only exfiltration. By conservation of masses of CO_2 and air, the following equations are then obtained [3]:

$$\begin{aligned} \frac{dm_1}{dt} &= Q_{01}(C_0 - C_1) + Q_{21}(C_2 - C_1) \\ \frac{dm_2}{dt} &= q + (Q_{21} + Q_{20})(C_1 - C_2) \end{aligned} \tag{1}$$

together with

$$Q_{10} + Q_{20} = Q_{01} \qquad \text{and} \qquad Q_{12} = Q_{21} + Q_{20} \tag{2}$$

where:

Q_{ij} are the mass air flow rates from zone i to zone j, 0 being for outdoors, 1 for the lower zone and 2 for the upper zone.

m_i are the masses of tracer gas (CO_2) present in both zones

C_i are the concentrations of tracer gas in each zone

q is the mass flow rate of tracer gas generated by the burners.

From these equations, the rate at which air rises in the hall, Q_{12}, can be calculated.

$$Q_{12} = \frac{\frac{dm_2}{dt} - q}{C_1 - C_2} \tag{3}$$

However, one more assumption is necessary to solve the four equations (1) and (2) which contain 5 unknowns. If we assume that the lower zone has no exfiltration, (which is true when all doors, which are 4 m high, are closed),

$$Q_{10} = 0 \qquad \text{then} \qquad Q_{20} = Q_{01} \tag{4}$$

Then the infiltration rate can be calculated:

$$Q_{01} = \frac{\frac{dm_1}{dt} + \frac{dm_2}{dt} - q}{C_0 - C_2} \tag{5}$$

Air flow rates Q_{01}, Q_{12} and Q_{21} are deduced from measurements using the above equations, during the period of time while C_1 and C_2 are both measured. The concentration in zone 2, C_2, is taken as the average of the measurements taken at 6 and 12 m. To avoid large numerical scattering, the interpretation is based on the average of two successive measurements. For example, at time t_j, equation (5) is interpreted as:

$$Q_{01}^j = \frac{\frac{C_1^{j+1} - C_1^{j-1} + C_2^{j+1} - C_2^{j-1}}{t_{j+1} - t_{j-1}} - q^j + q^{j+1}}{C_2^{j+1} - C_2^j - \left(C_0^{j+1} - C_0^j\right)} \tag{6}$$

The result is shown on figure 4.

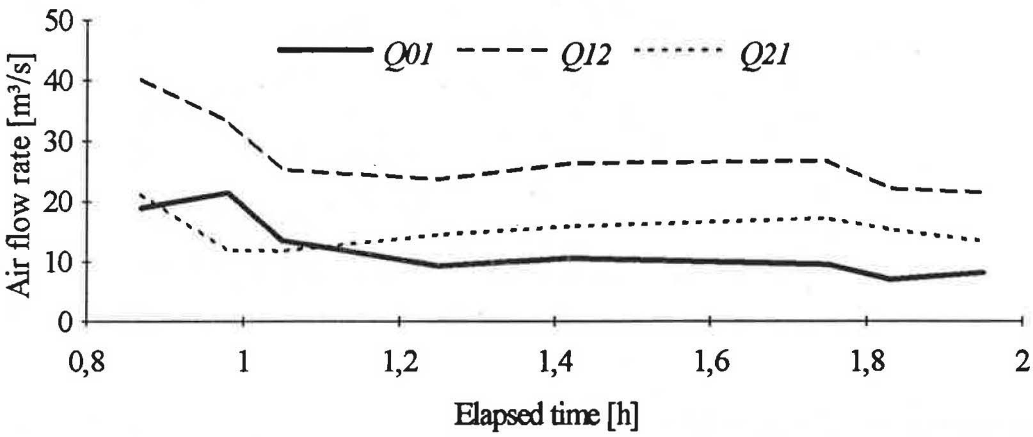

Figure 4: Air flow rates versus time at the end of a working day. Q_{01} is infiltration rate, while Q_{12} and Q_{21} are exchange air flow rates between lower (1) and upper (2) zone in the hall.

Table 2: CO_2 concentrations measured during the decay.

Time	CO_2 int - CO_2 ext	
	1 m	6 m
20.50	1015	1015
22.50	889	889
22.78	716	
23.02		
23.78	777	
23.92	766	
24.00		
24.02	755	
24.05	745	
24.08		765

During the decay experiment, all doors were closed, and the heaters were switched off. As shown on table 2, the carbon dioxide is evenly distributed, and a single zone model can then be used. The air change rate deduced form this decay experiment is 0,08 [h^{-1}], that is 4800 m^3/h or 1,3 m^3/s.

As a conclusion, it is shown that, at least in this case, air flow rates can be estimated on the basis of measurements of the concentration of locally generated contaminants. A condition is that the contaminant source strength is known and strong enough to be significant, and that the air flow pattern can be drawn.

For this building, the fresh air flow rate is about 20 m^3/s (air change rate is about 1) during the day, when main doors are mostly open, but goes down to 1,3 m^3/s (air change rate less than 0,1) when doors are closed.

Flows between the upper and lower halves of the hall are quite large when heating is on. The descending flow rate is equal or larger than the fresh air flow rate, while the rising flow rate is twice as much. In other words, the internal air change rate is twice the fresh air change rate. The pattern shown on figure 5 was also observed during smoke tests.

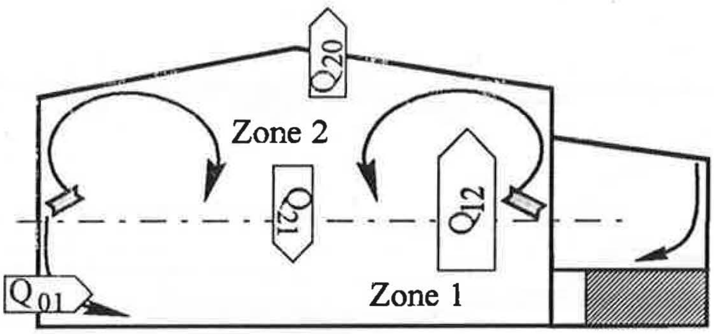

Fig. 5: Air flow pattern in the hall as deduced from measurements and smoke tests.

Measurements of temperatures

Air temperature was measured continuously at 10 locations in the hall, shown on figure 1. Some results are summarised in table 3, and shown on figures 6, 7 and 8.

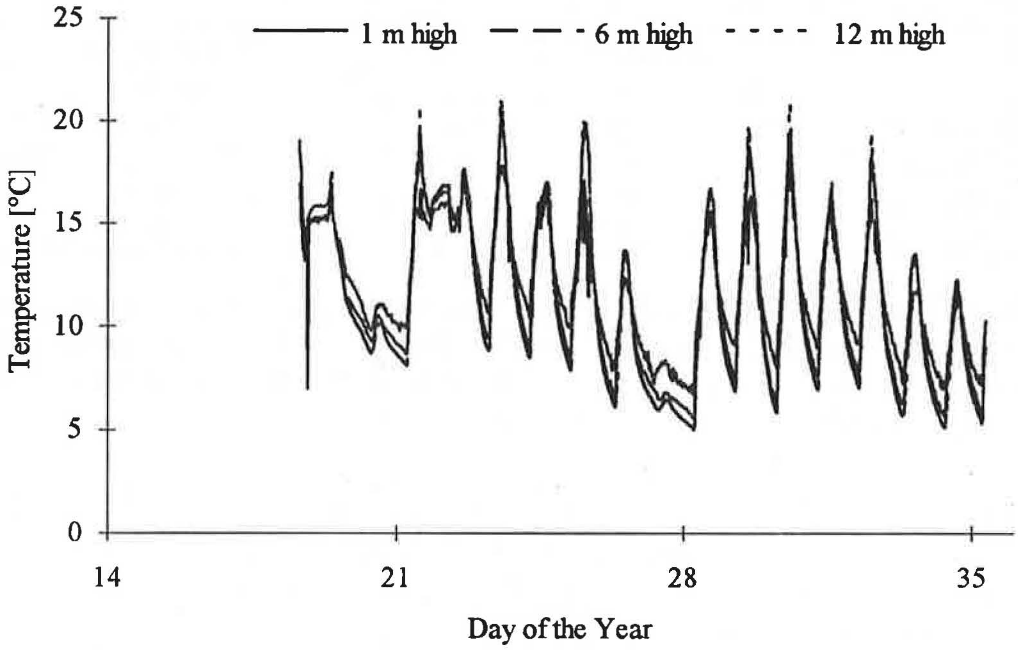

Fig. 6: Air temperature during January, averaged for each height.

As shown on Figure 6, the temperature varies strongly during the day. During the night, the heating is stopped and the temperature sets back. Early in the morning, the heaters bring the temperature at a comfortable level for hard work. When the sun rises, the solar gains through the large glazed panes result in a strong increase in temperature, especially in the upper part of the building.

Figure 7 shows that the air temperature averaged during working hours is very homogeneous. Only location 16, at 6 m height, above the office, is significantly higher than the average.

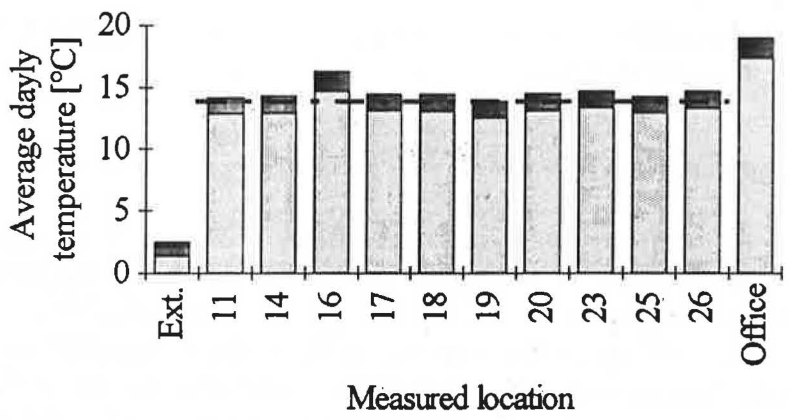

Figure 7: Air temperatures at 10 locations in the hall and in the office room, averaged during working hours between January and April 1991. Dashed part corresponds to ± one standard deviation. The horizontal dashed line shows the average temperature in the hall during the same period.

Table 3: Average internal temperatures and temperature gradients in January 1991.

	Temperatures [°C]			Gradients [K/m]		
	1 m	6 m	12 m	6 - 1 m	12 - 6 m	12 - 1 m
Maximum	17.9	20.1	21.0	1.5	0.2	0.7
Average	11.8	11.5	11.3	-0.056	-0.034	-0.044
Minimum	6.8	5.6	5.0	-0.4	-0.3	-0.2
Std. dev.	2.8	3.7	4.0	0.24	0.069	0.14

Figure 8 and table 3 confirm this result. The thermal gradient is very small, especially in the upper part. In the lower part, which is the occupied space, the gradient is slightly higher, but never gets past 1,5 K/m, and is only 0,06 K/m on average. The largest gradients are observed when solar radiation is high.

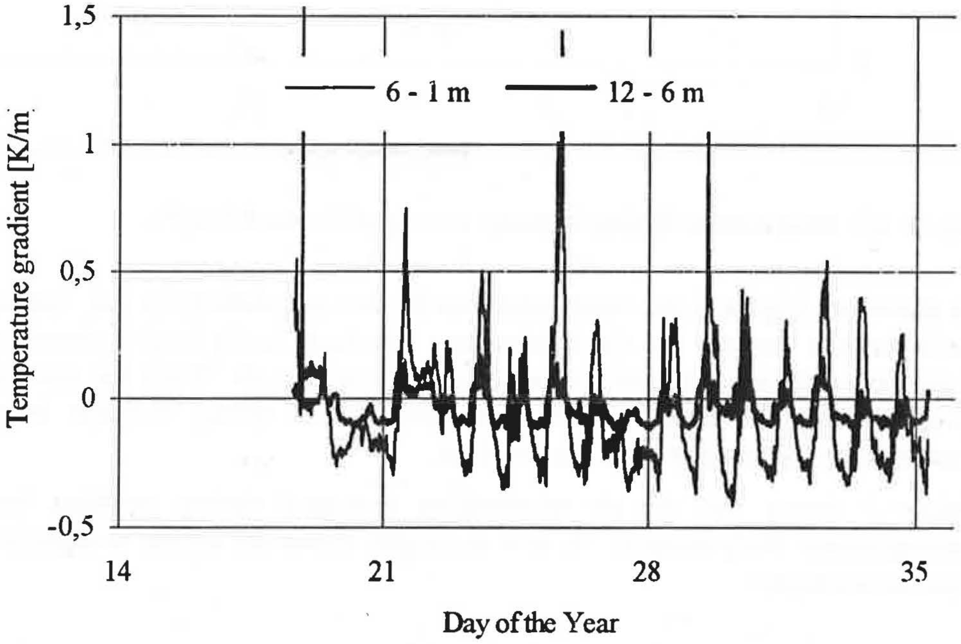

Figure 8: Temperature gradients during January, in the lower part of the hall (thin line) and in the upper part (thick line).

Thermal model of the hall

In order to understand the temperature variations and to calculate the effects of radiative and convective heaters, a simplified model was applied. This model [4] is a nodal model illustrated on figure 9. The heating system is modelled by two heat sources. The radiative source, Q_r, is directly coupled to the surface, while the convective part, Q_c, heats the air. The air node, at temperature T_a, is coupled to the surface (area S) by the heat transfer coefficient h, and has a relatively small capacity including the furniture, machines and manufactured parts located in the hall. The surface node, at temperature T_s is coupled to the external environment by the steady state resistance, R_{stat}, and by a dynamic resistance, R_{dyn}. This resistance is found by solving the equation of heat for a semi-infinite medium submitted to a step of heat flow density. For a single step:

$$R_{dyn} = \frac{2\sqrt{t}}{\sqrt{\pi\lambda\rho c}} = \frac{2\sqrt{t}}{b\sqrt{\pi}} \tag{9}$$

where:

- t is the time elapsed since the step started,
- λ is the thermal conductivity of the material,
- ρ is the density of the material,
- c is the heat capacity of the material,
- b $=\sqrt{\lambda\rho c}$ is the thermal effusivity of the material.

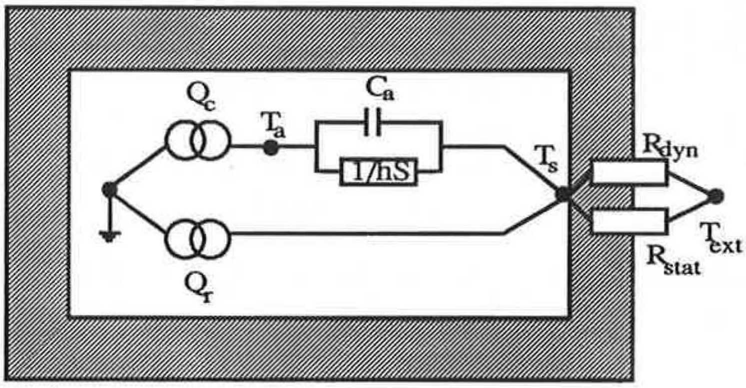

Fig. 9: Simplified thermal model of the hall.

This model can also be applied to a series of steps, as those shown on figure 10 to predict the thermal behaviour of the hall.

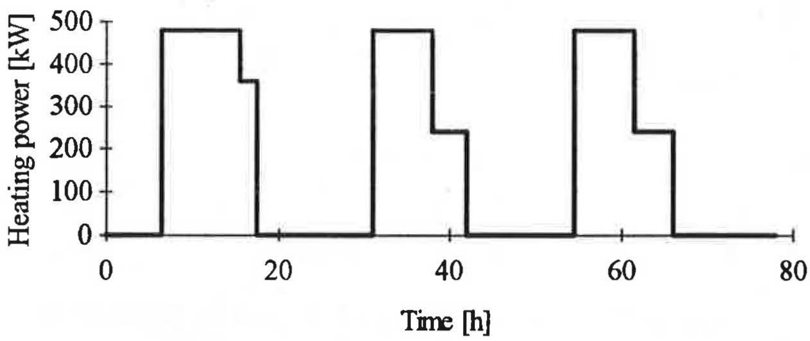

Fig. 10: Heating power schedule used as input to the model for three consecutive days.

The inputs to the model are the following:

- h_c convective heat transfer coefficient = 6 W/m²K
- S area of hall ground and envelope = 14 400 m²
- τ time constant of the heat capacity of air = $V\rho c_a/(h_c S)$ = 3 600 s
- R_{stat} = 1 m²K/W. The average U-value of the hall is 0,8 W/m²K, and 0,2 W/m²K takes account of the ventilation loss.
- b = 400 W$\sqrt{s}$/m²K. This value is obtained by best fit of the model on the measured data.

Calculation are first performed with a 60% radiative heating system, as is the installed system. The results of the calculation, together with the corresponding measurements are shown on figure 11. The correspondence is surprisingly good for such a simple model. It should be said that, in this hall, most of the mass is in the floor. The light walls are largely glazed (double pane) and roof is made of light, but well insulated metallic fabric.

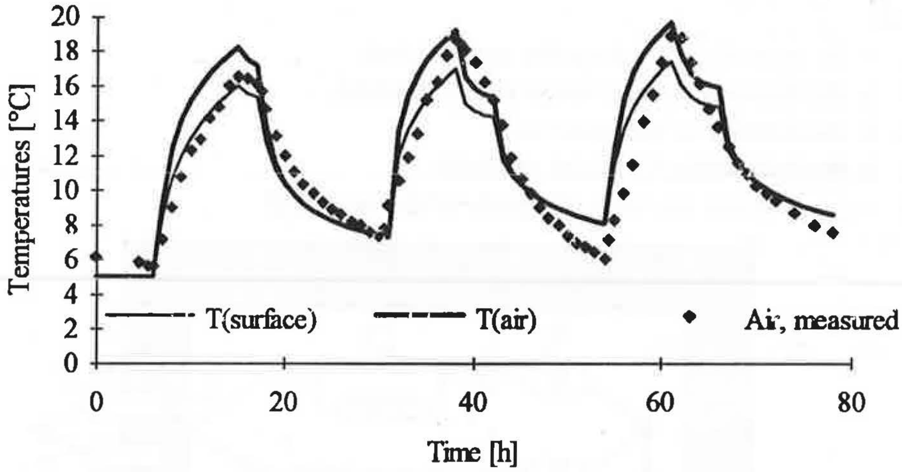

Fig. 11: Surface and air temperatures calculated with the model for radiative heating, compared to measured air temperature.

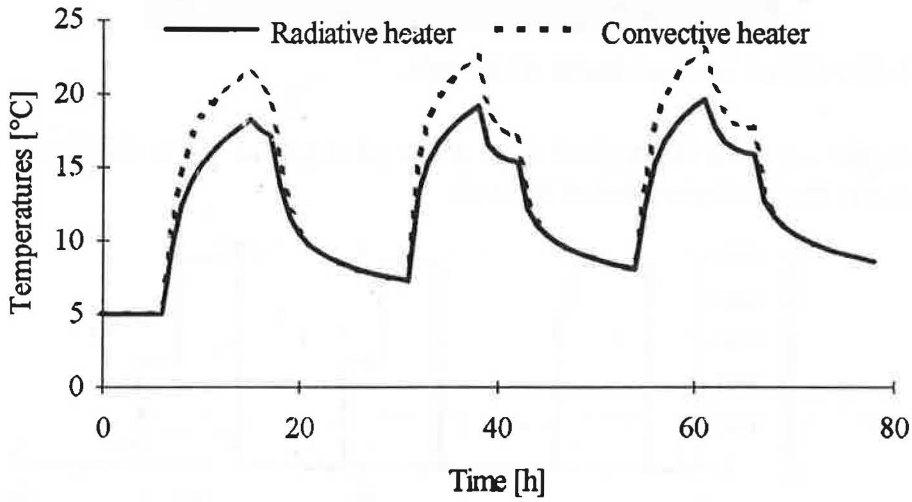

Fig. 12: Air temperature for three consecutive days with night set back, when the hall is heated either by a 60% radiative heater or by a 100% convective heater. Surface temperature is the same in both cases.

The effect of replacing the radiative heaters by pure convective one (such as fan heaters) can be seen on figure 12. For the same surface temperature, the air is much warmer. The main difference in dynamic behaviour is as follows.

In the morning, during boost heating, a convective heater provides its heat to the air, which transmits it to the various surfaces. This transfer is however limited by thermal surface resistance to about 6 W/m²K, and is relatively uniform at the beginning, when all surfaces are cold. The surfaces are then heated and their temperature rise proportionally to the dynaic resistance R_{dyn} (equation 9). Building elements with the lower effusivity, such as the roof and walls in the studied hall, would reach a comfortable temperature much faster than heavier surfaces. The heat flow rate then diminishes on these light surfaces and eventually concentrates on the heavier surfaces, which will be warm only later.

With a radiant heater, most of the heat is directed to the ground. It then receives a heat flow rate which is more in accordance with its effusivity, and will reach its comfort temperature earlier. Most of the workers are more in thermal contact with the floor than with the walls, and feel then more comfortable. In addition, they receive directly a part of the thermal radiation of the heater.

Energy balance

The energy balance of the building was obtained in two ways. The first one comes from the comparison of the measured weekly energy use to the weekly average outdoor temperature. This provides the so-called energy signature [5]. Figure 13 shows the weekly average heating power, P, versus average outdoor temperature, $\overline{\theta}$.

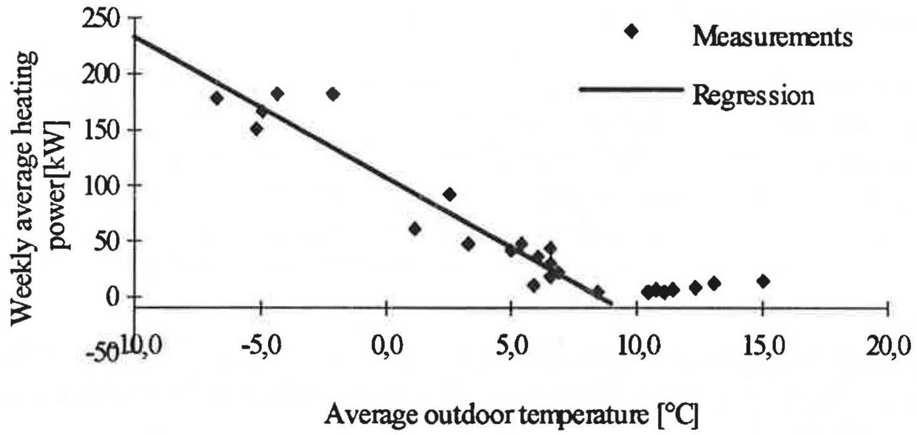

Fig. 13: Energetic signature of the measured building.

The regression line during the heating season follows equation:

$$P = P_o - H\overline{\theta} \tag{8}$$

where:
P_0 is the value of this power for 0°C outdoors
H is the slope of the signature

H is also the specific heat loss of the building, since the heater efficiency is 100%. From measurements, values of

$$P_o = 110 \pm 10 \text{ kW} \qquad \text{and} \qquad H = 12 \pm 2 \text{ kW/K} \tag{9}$$

were determined. Confidence intervals are based on the dispersion of the measured points and are given for 95% probability. Equation (8) also allows us to determine a limit temperature for heating equal to 8°C and a minimum installed power of 240 kW for -11 °C outdoors. The actual installed power is 700 kW, which allows the night set back and a morning heating in a reasonable time. From the local yearly meteorological data (degree-days and length of heating season) a seasonal heating energy requirement of 1500±150 GJ can be calculated.

This energy requirement was also calculated from the thermal characteristics of the building, using a simplified method similar to that described in prEN 832 [6]. Assuming an internal temperature of 12°C, close to that measured, and an average air change rate of 0,2/h, the energy requirement for heating was calculated to be 1500 GJ, 27% of it being for compensation of ventilation heat loss. The assumptions used in this model are then validated by the measurement of the signature.

These results can then be used to calculate the effect of an increase in internal air temperature required by convective heaters to attain the same comfort temperature. An average internal temperature 1 K higher will increase the yearly energy requirement by 18%. Other heating systems may also cause larger thermal gradient, which has also a negative effect on energy consumption. A temperature increase of 1 K in the upper zone only would increase the energy requirement for heating by 8%.

Minimum ventilation rate

For each kg of burned propane, the heaters generate 12,8 kWh, 3 kg CO_2 and 1,63 kg water vapour. A minimum air flow rate is then required to dilute CO_2 and water vapour at acceptable levels. The required power from the burners, hence the contaminant source strength, depends on the outdoor temperature and on the ventilation rate itself. The amount of air necessary to dilute water vapour depends also on the outdoor air moisture content. A very simple model, which is described below, can take account of these elements and predict the required air flow rate.

The necessary power is provided by the energetic signature, given in equation (8), and the corresponding mass flow rates of propane and contaminants, q_{H_2O} and q_{CO_2} are easily deduced.

On the other hand, the air moisture β [kg/m³] is linked to the relative humidity, φ [%], and to the saturated vapour pressure, p_s, by:

$$\beta(\theta,\varphi)=\frac{\varphi\, p_s M}{100RT} \quad [\text{kg/m}^3] \tag{10}$$

where:

M is the molar mass of water (0.018 kg/mole)
R is the constant of perfect gases, 8.31396 J/mole K,
T is the absolute temperature.

The saturation vapour pressure can also be related to the air temperature by the empirical relations:

$$\begin{aligned} p_s &= 1.40974\cdot 10^{10}\exp\left[\frac{-3928.5}{\theta+231.67}\right] \quad \text{if } \theta>0 \\ p_s &= 3.61633\cdot 10^{12}\exp\left[\frac{-6150.6}{\theta+273.33}\right] \quad \text{if } \theta\le 0 \end{aligned} \tag{11}$$

The air flow rate for maintaining the required internal air moisture, φ_i, is then

$$Q_{a,\,H_2O} = \frac{q_{H_2O}}{\beta(\theta_i,\varphi_i)-\beta(\theta_e,\varphi_e)} \tag{12}$$

and the air flow rate required to dilute the CO_2 below a limit concentration, C_{lim} [ppm] is:

$$Q_{a,CO_2} = \frac{1000 \cdot q_{CO_2}}{C_{lim} - 340} \quad (13)$$

Results from equations 12 and 13, valid for the considered industrial hall, are shown on figure 14. In this case, $C_{lim} = 5000$ ppm and φ_e=75%.

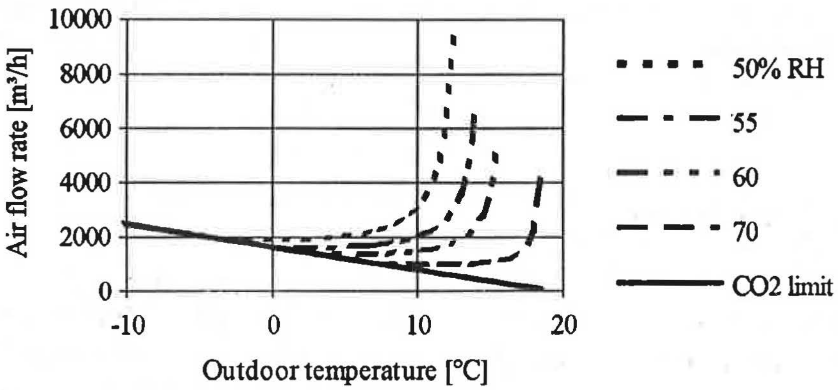

Fig. 14: Minimum air flow rate for dilution of contaminants generated by propane heaters in the industrial hall.

In this case, the minimum air flow rate (less than 0,1 volume/hour) is low when compared to the ventilation required to evacuate dust and gases caused by arc welding and painting. This is because the hall is well insulated and thus requires not much power, hence not much propane.

Conclusions

The purpose of this contribution is to show that much information on air flow patterns and comfort conditions can be obtained from very few measurements. In the present case, we have answered the following questions:

- "How large are the main internal and fresh air flow rates?" using concentration measurements of a locally generated tracer gas and a two zone interpretation model adapted to the considered hall.
- "How are the temperature and contaminants distributed throughout the hall?" from recordings of temperatures at only 10 locations thoroughly chosen.
- "How efficient is a radiative heater when compared to a convective one?" by interpreting temperature records with a simple dynamic nodal model.
- "Is the air change large enough in any circumstances to remove combustion gases?" from energy consumption measurements (or energy requirement modelling), classical hygrometry relations and steady state mass conservation.

In reference [1], a study of the thermal comfort conditions under the radiant heaters, based on the Fanger model, is also presented.

Acknowledgements

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