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CONVECTIVE EXCHANGES IN A CLIMATE ROOM AND IN A COMMERCIAL OPEN PLAN OFFICE

Summary. A general overview of some experimental investigations in the field of air diffusion in occupied dwellings is presented. The communication is structured into two distinct parts. The first one describes the study of jets behaviours inside a laboratory climate room and the influence of various heating sources, including mechanical ventilation, on energy consumption estimates. The second part presents results of a short measurement campaign performed in a large open plan administrative office aiming to provide a better appraisal of convective heat transfers occurring between central and peripheral zones which experience significant spatial temperature differences.

PART 1 : CLIMATE ROOM INVESTIGATION OF AIR DIFFUSION PROCESSES AND REFINED HEAT BALANCE CLOSURE

I. INTRODUCTION

Air movements and temperature stratification inside heated or ventilated volumes may significanlty affect the heat balance of the room and yield sometimes discomfort for occupants. Usual calculations have no possibilities to ascertain the importance of such phenomena. The investigations presented here tend to estimate the errors arising in classical heat balances and influencing the expectations of energy consumption.

II. BRIEF DESCRIPTION OF THE CLIMATE ROOM

The climate room used for comparisons between effects of various heating and ventilation systems is pictured on figure 1. It consists of an internal parallelipipedic volume of 4.74 m X 3.45 m X 2.70 m where a ceiling plenum may be mounted. Its internal walls are constituted of wood fiber (18 mm) insulated with 25 mm of polyurethane and 50 mm of polystyrene. Its backside may be easily moved to modify the room's internal length. The whole volume is contained in a well insulated envelope. All internal walls are equipped with compensation compartments where ventilo-convectors and eventually electric heaters allow a perfect temperature control.

The front wall of the climate room simulates an outdoor wall. It is constituted of an equivalent window of epoxy and PVC, which acts as sets of heat flux meters. Polystyrene is added to represent the opaque part of the wall (counting for 64% of the total surface). Outdoor conditions are simulated in a nearby compartment where a plates heat exchanger together with an electric heating maintain a typical temperature of - 3°C. Four ventilators blow air from this compartment into a 36 cm wide air layer along the external wall, controlling a velocity of air between 3 and 4 m/s along the wall, a typical range for wind effects in real conditions.

Air infiltration is simulated by a controlled ventilation with linear outlets all along the window frame.

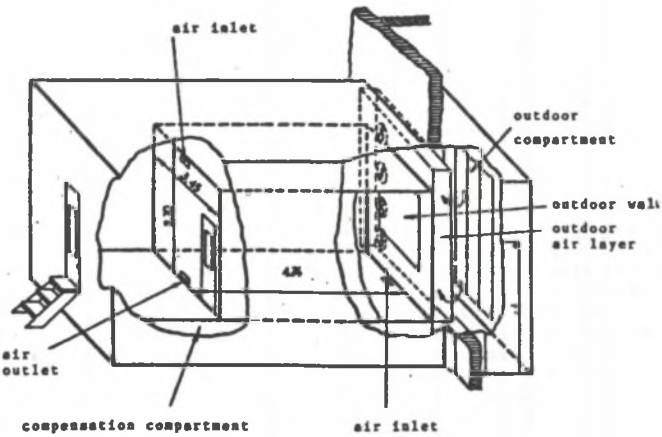


Fig. 1. View of the climate room
Rys. 1. Przekrój komory klimatycznej

The climate room is instrumented with approximately 300 points of measurements. Heat flux meters, consisting of very sensitive thermopiles with a reference superficial temperature sensor in the center of each are distributed over the internal surfaces of the interior walls. The outdoor wall and window are particularly instrumented. Actually, we may refer to those as fluxmetrical surfaces (fig. 2 and 3).

The occupied zone, defined as an internal volume of air limited at 0.70 m of the vertical walls and of 1.8 m high, is equipped with 6 locations of air and resultant temperature measurements, at 0.08, 0.75, 1.50 and 1.80 m above the floor level.

The compensation boxes behaviour is also controlled through regularly distributed air and surface temperature sensors.

A three-directional chariot is mounted inside the room and enables the displacement of additional sensors with minimum perturbations of the indoor climate. It is notably used for air velocity measurements. Those are performed by thermistor omnidirectional anemometers. All temperature recordings are performed using Copper-Constantan thermocouples.

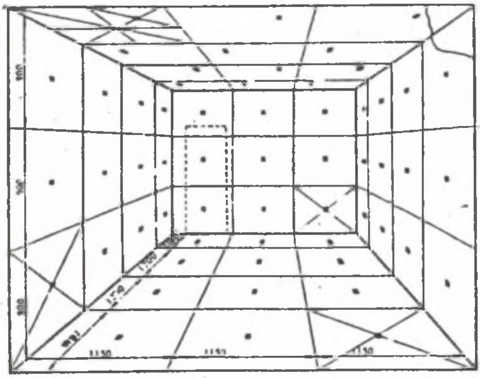


Fig. 2. Inside of the climate room seen from outdoor wall. Locations os superficial temperatures measurements

Rys. 2. Wnętrze komory klimatycznej widziane od ściany zewnętrznej. Rozmieszczenie punktów pomiaru temperatur powierzchniowych

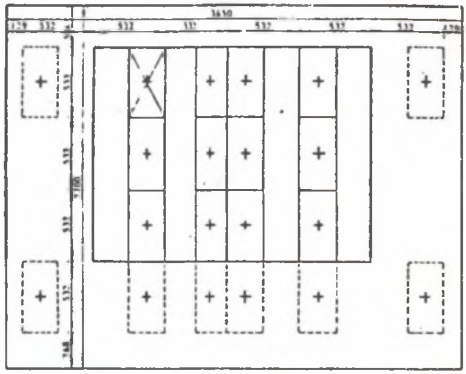


Fig. 3. Outdoor wall and window. Locations of heat fluxmeters

Rys. 3. Ściana zewnętrzna oraz okno. Rozmieszczenie mierników strumienia ciepła

III. CALIBRATION OF THE CLIMATE ROOM

Using a reference resultant temperature inside the room, a characteristic global heat transfer coefficient at the internal surfaces of the walls was experimentally determined with a value of 8.1 $W/m^2 °K$. From this point of view, convective and radiative heat exchanges at surfaces are merged, the latter effect resulting from a linearization of Stefan's law. In the air layer along the external wall, a similar procedure was applied and yielded a transfer coefficient of 21.0 $W/m^2 °K$. The airtightness of the room was investigated by pressurization tests for two different insulation levels of the external wall. Results produces the following regression law :

$$n = a\,\Delta p^{k} \quad (1)$$

where n is the airflow rate introduced in the room during the tests (vol/h) and Δp is the difference between the static pressure in the room and its value in absence of air production. It is to be noted that the presence of ventilators in the simulator of outdoor climate creates a slight overpressure of the environment with respect to the room in normal conditions.

Table I gathers the results of those tests and shows that quite much care should be devoted to the equilibrium of the room with the compensation boxes if uncontrolled heat losses are to be avoided.

Table I

level of insulation of external wall	a	u
weak	1.55	0.71
good	1.20	0.77

IV. HEATING AND COOLING SOURCES

The figure 4 depicts the different heating and cooling possibilities that were mounted in the climate room. They include :

1. two single plate radiators placed under the window ;

2. one multiple plates radiator under the window ($\sim$ 85% of emissivity by natural convection) ;

3. a similar one placed along a wall ;

4. an oil bath electric radiator on the backside of the room ;

5. and 6. two linear air inlets for mechanical ventilation. One is located in the floor, under the window, the other one is placed in the back wall, close to the ceiling ;

7. and 8. radiative distributed heating sources, produced by electrically heated carpetings and located on the floor and on the ceiling.

All devices are completely instrumented in order to perfectly control their heating or cooling contributions. In particular, hot water radiators are controlled following the Belgian norm NBN 236.

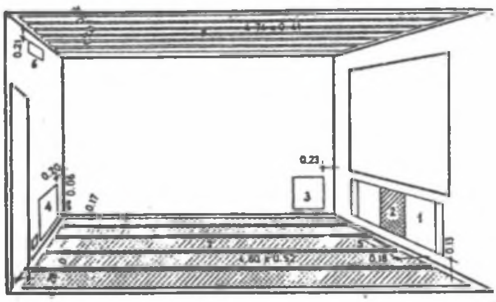


Fig. 4. Heating and cooling possibilities in the climate room

Rys. 4. Możliwości ogrzewania i chłodzenia w komorze klimatycznej

V. THERMAL COMFORT WITH MECHANICAL VENTILATION

A correct design of a mechanical ventilation system must provide a well defined air renewal rate and maintain the occupied zone's temperature within a prescribed range. Airflow rates and inlet air temperatures are generally determined by the knowledge of the heat loads expected in the room and adequate estimate of its heat balance.

In principle, there remains a considerable freedom for the choice of types of air inlets and their spacial locations. However, air movements inside the room should be investigated, if local discomfort due to too high air velocities or too low temperatures in the occupied zone are to be avoided.

Jets behaviour were thus studied in the climate room for various air delivery situations. Successively the supply by a ceiling anemostat, a linear inlet placed in the backside wall, close to the ceiling and an other linear aperture in the floor under the window were considered. *In all cases, heating was supplied by a radiator situated also under the window.*

Tests were performed for 25,50 and 100 m^3/h of air supply and inlet temperature ranging between 10 and 22°C. Heat balances yield overall energy consumptions between 550 and 950 W, depending on ventilation conditions.

Air flow patterns are sketched on figure 5 for typical situations.

For all situations, we may notice that air velocities in the occupied zone (noted on the figure for illustration purpose only because of the inaccuracy inherant to such low recordings) are only influenced when jets fall into that zone. The same observation may be expressed concerning temperatures.

Jets created by the anemostat benefit of the Coanda effect and remain close to the ceiling's surface. The same phenomenon occurs for cold

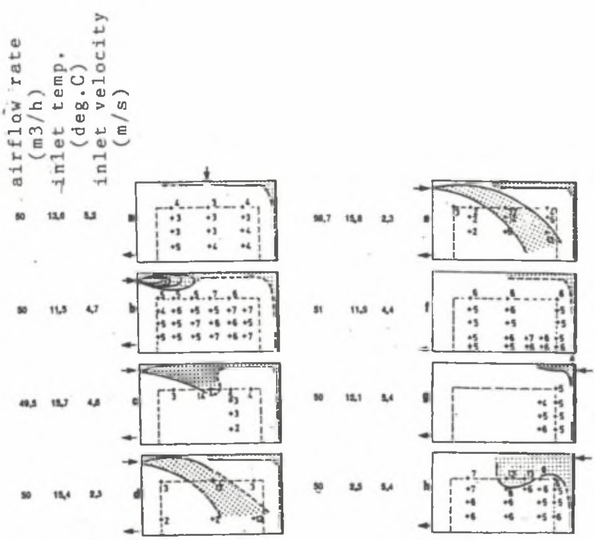


Fig. 5. Air flow patterns for various inlet conditions. Dark zones experience velocities above 10 cm/s

Rys. 5. Przepływ powietrza dla różnych warunków wlotowych. W strefach zaciemnionych obserwuje się prędkość powyżej 10 cm/s

air injected behind the heat source, which furthermore mixes favourably with the natural convection plume rising above the radiator. Initial velocities at the linear aperture of the backwall were modified by changing the width of the outlet. For prescribed inlet temperature and flowrate, this modification may result in a decreasing throw of the jet, which is then likely to enter the occupied zone and yield discomfort. It was tried to represent this effect, using Jackman's experimental law (3) :

$$\frac{z}{\sqrt{A}} = K\, Ar \left(\frac{x}{\sqrt{A}}\right)^3 \qquad (3)$$

where the Archimede number uses the square root of the inlet area as characteristic dimension and the inlet velocity. Z represents the vertical departure of the meanline of the jet from the x horizontal coordinate at the inlet's height. Various values of the factor K can be found in the literature, ranging from 0.04 to 0.1. Good agreements were found here with the initial proposal of Jackman (0.04).

From the above considerations, it was possible to propose a comfort criterium, defining a minimum inlet velocity dependant on the airflow rate (V) and the temperature difference between room and inlet conditions :

$$u_o \geqslant 4.7 \; \frac{\Delta T^{2/3}}{\dot{V}^{1/3}} \quad \begin{matrix} (^\circ K) \\ (m^3/h) \end{matrix}$$

Equation (3) may be used directly for the choice of adequate inlet sizes. The limiting values it provides are most probably too drastic as it does not consider the contraction of the jet just after outlet neither the benefit of streamlines reattachment on the ceiling (Coanda effect). No occupancy heat loads were simulated during the tests. Their effect could eventually advantage comfort too. Finally, the installator may also reduce the distance between the linear inlet and the ceiling and widen its length. He must also consider another (upper) limitation of the air velocity, depending on the acoustic comfort, which was here completely ignored.

VI. CLOSING HEAT BALANCES FOR BETTER ENERGY CONSUMPTION ESTIMATES

The sizing of the energy consumption required for conditionning a building depends on two types of estimates : the heat balance of the dwelling and the heat deperditions and efficacity of the system's components and distribution network. Tests in a climate room enable very refined heat balances and provide orders of magnitude of errors committed by usual procedures in the calculation of this contribution to energy consumptions.

Classical static heat balances represent the zone to be conditionned as a thermally homogeneous volume at a reference resultant temperature t_R, recorded generally in the center of the room at a height of 0.75 or 1.50 m.

Conductive heat losses are expressed in terms of this temperature and an outdoor air temperature, t_o, using a global transfer coefficient (W/m^{2}OK) :

$$Q''_{cond} = U\,(t_R - t_o) \quad (W/m^2) \tag{4}$$

$$\frac{1}{U} = \frac{1}{h_o} + \sum_j \frac{e_j}{k_j} + \frac{1}{h_i} \tag{5}$$

where h_i and h_o are global exchange coefficients along surfaces with prescribed constant values (in our case, respectively 8.1. and 21.0 W/m^2°K). e_j and k_j represent the thickness (m) and the thermal conductivity (W/m°K) of each layer constituting the external wall. For composite walls (for example, an opaque wall with a window), an overall transfer coefficient may be composed, using respective surfaces as weighting factors :

$$\bar{U} = \frac{\sum_j A_j U_j}{\sum_j A_j} \tag{6}$$

The use of global coefficients, as mentionned earlier, merges convective and radiative exchanges along the surfaces. The linearization of Stefan's law for the latter term is indeed a very reasonable approximation for the range of superficial temperature differences encountered in rooms in standard occupancy conditions and this term should not be considered as an important cause of error.

However, due to complex movements of air masses at quite different temperatures in the room, significant discrepancies between reality and assumptions can occur in the estimate of convective heat transfers. In particular, the use of a unique constant exchange coefficient may yield important errors.

Losses due to air infiltrations are generally unaccurately sized, due to the lack of knowledge of the amount of outdoor air entering the room. They are expressed as an enthalpy flux :

$$Q_{infilt.} = \rho \, C_p \, \dot{V} \, (t_a - t_o) \qquad (W) \qquad (7)$$

where ρ stands for air density (kg/m^3)

C_p is the air heat capacity (J/kg°K)

$\dot{V}$ the volumic air flow rate (m^3/s)

t_a is the room air temperature (°C).

Finally, let us mention that mechanically extracted air is assumed to leave the room with the reference temperature t_R. However, again air movements may introduce non negligeable temperature gradients which will affect this term of the heat balance.

A detailed study of all contributors to the heat balance was performed for different types of heat sources and ventilation supplies, varying the air infiltration rate around the window (0,0.7 and 1.2 vol/h).

A resultant temperature at mid-height (1.5 m above the ground) is maintained throughout the tests, together with an outdoor climate at t_o = - 3°C. Results are however normalized for t_R = 22°C at 0.75 m and a unit infiltration rate (1 vol/h). Two types of insulation of the external wall are also considered. The various situations analyzed are summarized in table II.

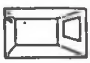
Measurements by all heat flux meters and correlation with temperatures recorded in the climate room and the compensation compartments enabled the estimate of heat budgets with an accuracy ranging between 1.5 and 5%.

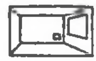
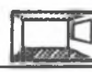
First, global transfer coefficients were experimentally determined, using :

$$\bar{U} = \frac{\bar{Q}''}{t_R - t_o} \qquad (8)$$

and compared to usual values deduced from the combined use of equations (5) and (6). Differences range between 4 and 16%, depending on the type of heating, for a good insulation of the external wall and reach 34% in some cases for a weak insulation. Worse situations occur with radiators heating in the first case but are overgraded by mechanical ventilation in the case of a weak ventilation. The minimum discrepancies appear for radiative heating on horizontal surfaces. The general trend of classical calculations underestimates the transfer coefficient. Those observations can be explained by the degree of mixing of the air, and the occurence of vertical temperature gradients in the room (stratification). Indeed, the use of mechanical ventilation increases significantly the convective exchanges on the inside surface of the external wall, whose effect on the heat transfer through the wall is mainly important when only a weak insulation is used. On the other end, convective movements are minimum for distributed radiative heating, while temperature variations along the height of the room are maximum when heating surfaces operate on the ceiling.

Table II

MECHANICAL VENTILATION		
From floor	$= 2\ m.s^{-1}$	
	$= 4\ m.s^{-1}$	
From opposite wall	$= 1.5\ m.s^{-1}$	
	$= 4\ m.s^{-1}$	

RADIATORS	
Single panel	
Multiple panels	
Multiple perp. panels	
Multiple panels	
RADIATIVE SURFACES	
Heating floor	
Heating ceiling	
33 % floor 67 % ceiling	

Temperature stratification has a definite influence on the heat losses by air extraction from the room, depending on the location of the aperture through which this air is leaving. Vertical temperature profiles recorded along the central vertical of the room are presented on figure 6 for various situations of heating. Maximum differences occur in the case of radiative heating from the ceiling, where low convective movements are observed or when primary plumes or jets reach this ceiling without having been influenced by the cold external wall or by mixing with the infiltration air.

The effect on heat losses due to air extraction is due to the fact that air will leave the room at a temperature t extract different from the assumed reference resultant temperature t_R. The error inherent to this above assumption may be sized by the definition of an "anisothermy factor" :

$$f_a = \frac{t\ extract - t_R}{t_R - t_o} \qquad (9)$$

The probable error of classical calculations ranges around 10%.

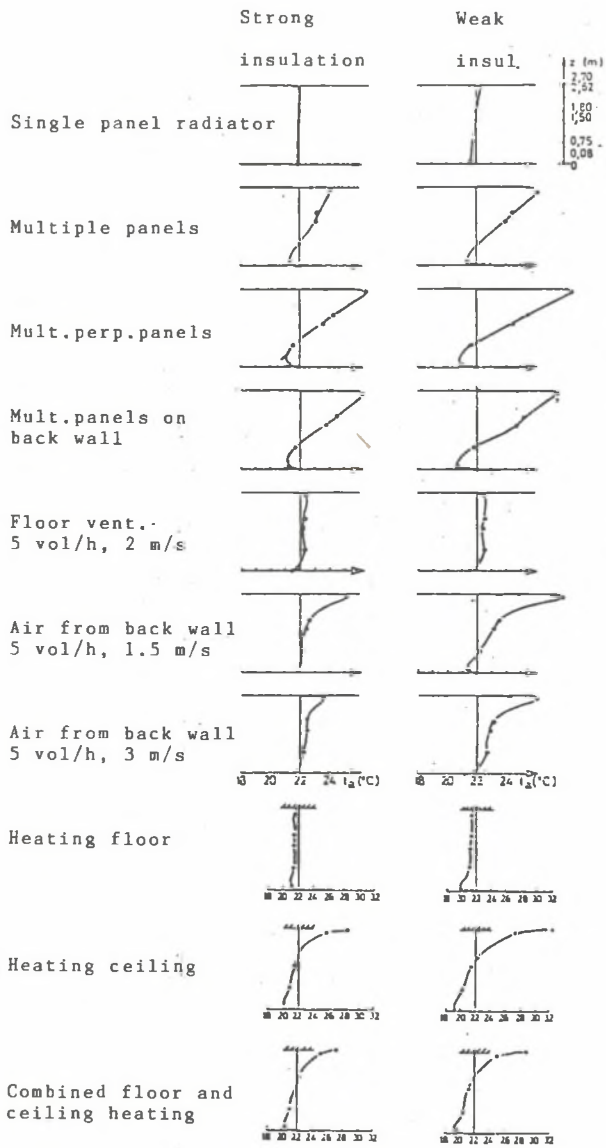


Fig. 6. Vertical temperature profiles in the center of the climate room
Rys. 6. Pionowe profile temperatur w środku komory klimatycznej

The abovementionned observations enable the calculation of an experimental overall deperdition factor of the room. It is defined as :

$$G = \frac{Q}{V(t_R - t_o)} \quad (W/m^3 °K) \tag{10}$$

where V is the room's volume (m^3) and Q stands for :

$$Q = Q_{cond} + Q_{air} \quad (W) \tag{11}$$

Q_{cond} represents the actual conductive heat losses through the external wall :

$$Q_{cond} = \overline{U}\, A\, (t_R - t_o) \tag{12}$$

with $\overline{U}$ defined by equation (8), and Q_{air} is the amount of heat dissipated by exfiltration :

$$Q_{air} = \rho C_p \dot{V}(t_{extract} - t_o) \tag{13}$$

where $t_{extract}$ is the air temperature at the location of the outlet aperture.

Combining (8) through (13) we obtain :

$$G = \frac{\overline{U}}{L} + n \rho C_p\, (1 + f_a) \tag{14}$$

where L stands for the length of the room.

Classical calculations would use a theoretical deperdition coefficient G_{th}, which may be deduced from (4) by setting the anisotherny factor f_a to zero and using (5) and (6) to define the transfer coefficient $\overline{U}$ Some comparisons are presented on table III.

Table III

DIFFERENCES (in %) BETWEEN THEORETICAL AND EXPERIMENTAL DEPERDITION COEFFICIENT

Type of heating	Insulation level	
	strong	weak
Single panel radiator	5	8
Multiple panels	8	13
Mult.perp.panels	10	10
Mult.panels on back wall	10	7
Floor ventilation (5 vol/h, 2 m/s)	6	22
Back wall ventilation (5 vol/h, 1.5 m/s)	5	18
Back wall ventilation (5 vol/h, 3 m/s)	6	19
Floor heating	0	-5
Ceiling heating	5	-1
Combined floor and ceiling heating	5	-1

We may conclude that classical calculations present sufficient accuracy for distributed radiative heat sources (within 5%), but may lead to errors up to 20% in some situations, particularly for mechanical ventilation in a weakly insulated dwelling.

Classical heat balances assume that the internal exchanges along surfaces are characterized by a unique global exchange coefficient h_i, which is in our case of 8.1 $W/m^2 °K$. More refined calculations will require the analysis of this assumption. On each wall of the climate room, linear correlations were performed to link measured heat transfers through the surface with the temperature differences observed between the surface temperature t_s and the reference t_R:

$$Q'' = C_o + C_1 (t_R - t_s) \quad (W/m^2) \tag{15}$$

Such a law does not actually represent the real physical exchanges, which are in fact driven by the difference between the temperature of each surface and a reference temperature t_{ref}, adequately chosen and characteristic of that surface :

$$Q'' = h_i (t_{ref} - t_s) \qquad (W/m^2) \tag{16}$$

t_{ref} does not represent a true air temperature, because a global heat coefficient is still considered in (16) and may be interpreted as a particular resultant temperature.

Equation (16) may be written as :

$$Q'' = h_i (t_{ref} - t_R) + h_i (t_R - t_s) \tag{17}$$

Comparing (15) and (17), we observe that the determination of C_1 will directly provide the exchange coefficient, while C_o will yield the definition of the reference temperature.

Correlations were not obtainable in all cases. Results are summarized on table IV. Main differences in h_i with the assumed 8.1 $W/m^2 °K$ occur essentially on the window, as expected, and especially when strong air movements occur along its surface. In some situations, the reference temperature t_{ref} falls below t_R, measured at the center of the room. So doing, it reverses the direction of heat transfer as stated by equation (15) which becomes incredible.

In fact, the correlation (16) becomes obsolete in thoses cases.

Now, considering that the heat flux transferred along the surface is actually transmitted by conduction through the wall, we may express :

$$Q'' = \bar{U} (t_{ref} - t_o) \tag{18}$$

where $\bar{U}$ is the experimental global transfer coefficient. It may be written as :

$$\frac{1}{\bar{U}} = \frac{1}{h_i} + R + \frac{1}{h_o} \tag{19}$$

if R stands for the overall resistance from surface to surface of the wall (i.e. $\sum_j e_j / k_j$ for a multiple layers unidimensional wall)

and h_o the global exchange coefficient experimentally determined on the opposite side of the wall.

Noting that the conventional transfer coefficient is deduced from

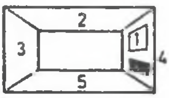
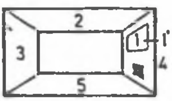
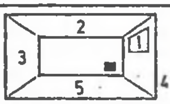
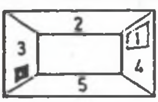
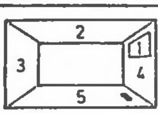
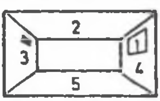
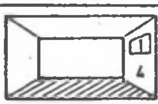
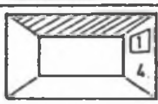
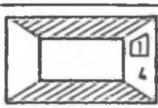
$$\frac{1}{U_{th}} = \frac{1}{h_{ith}} + R + \frac{1}{h_o} \qquad (20); (h_{ith} = 8.1\ W/m^2 °K)$$

we obtain

$$Q'' = \frac{t_{ref} - t_o}{\frac{1}{h_i} + \frac{1}{\bar{U}_{th}} - \frac{1}{h_{ith}}} \qquad (21)$$

Table IV

CONVECTION COEFFICIENTS CORRELATIONS

Heating device	surface	h_i $(W\ m^{-2}\ K^{-1})$	$t_{ref} - t_R$ (K)	r	f
	1	7.7	3.5	0.98	0.14
	3	3.1	0.3	0.64	0.01
	4	4.1	1.9	0.93	0.08
	5	3.6	0.3	0.83	0.01
	1	10.5	1.7	0.93	0.07
	1'	6.8	4.2	0.87	0.17
	4	4.0	3.2	0.96	0.13
	1	7.6	3.6	0.997	0.14
	3 et 4	5.6	2.0	0.82	0.08
	1	7.4	3.3	0.98	0.13
	3 et 4	5.5	1.7	0.79	0.07
	5	2.2	0.3	0.86	0.01
	1	12.5	1.3	0.98	0.05
	2	11.4	-0.1	0.90	
	3 et 4	6.6	0.5	0.83	0.02
	5	2.6	-0.2	0.72	
	1	11.7	0.4	0.99	0.02
	1	5.9	4.3	0.98	0.17
	4	6.6	0.6	0.83	0.02
	1	6.1	5.1	0.95	0.20
	4	6.7	1.5	0.99	0.06
	1	5.5	6.3	0.99	0.25
	4	4.0	4.3	0.94	0.17

We may now define a second anisothermy factor relevant for each wall :

$$f = \frac{t_{ref} - t_R}{t_R - t_o} \qquad (22)$$

Values of f are listed in table IV.
Using (8), it is possible to express :

$$\bar{U} = \frac{1 + f}{\frac{1}{h_i} - \frac{1}{h_{ith}} + \frac{1}{\bar{U}_{th}}} \qquad (W/m^2 °K) \qquad (23)$$

allowing the estimate of the actual global transfer coefficient from quantities used by classical calculations (h_{ith} and U_{th}) and the knowledge of table IV. The comparison with usual assumptions may be performed on :

$$\Delta = \frac{\bar{U} - \bar{U}_{th}}{\bar{U}_{th}} = \frac{f + \bar{U}_{th} \left(\frac{1}{h_{ith}} - \frac{1}{h_i} \right)}{1 - \bar{U}_{th} \left(\frac{1}{h_{ith}} - \frac{1}{h_i} \right)} \qquad (24)$$

In cases where h_i differs only slightly from h_{ith}, Δ may be sized by the anisothermy factor f. It is the case when static radiators are used as heat sources. On the other hand, when the mechanical ventilation is used, the anisothermy practically disappears ($f \simeq 0$) and table IV yields :

$$\Delta \simeq \frac{1}{\frac{25}{\bar{U}_{th}} - 1} \qquad (25)$$

which amounts to 30% for a single glazing and 11% for double pane window.

As a conclusion , table V proposes approximate values of the internal exchange coefficient h_i and the anisothermy factor f for standard situations.

Table V

Type of heat source	h_i ($W/m^2 °K$)	f
radiators	8.0	0.15
behind radiators	8.0	1.00
in the wake of radiators	10.0	0.15
mechanical ventilation	12.0	0.00
radiative heating on floor or ceiling	6.00	0.20
along heated floor	13.00	0.04
along heated ceiling	7.00	0.10

VII. CONCLUSIONS

The influence of very different types of heat sources on the indoor climate has been studied in a completely controlled climate room. The investigation of jets behaviour, when mechanical ventilation is used yields interesting suggestions for the design of air inlets to the conditionned spaces. A detailed study of the various heat losses experienced by the room for different types of heat sources has led to the expression of a very refined heat balance and proposed corrections to usual calculation procedures. In terms of energy consumption, it could be shown that usual estimates tend to underestimate heat losses and may result in errors on the needed energy supply that can reach 30% in some situations. Two effects have a particular importance : air movements introducing large dispersions in the actual heat exchanges along the walls and anisothermy of the indoor ambiance, especially affecting the actual temperature of the air leaving the room. It is noted that the effect of the former on the deperditions of the room may be reduced by a careful insulation of the external walls.

A complete understanding of the various effects on heat transfers may only be performed by splitting convective and radiative exchanges with all sections of the walls. This part of the work was reported in references (1) and (4).

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PART 2 : AIR DIFFUSION AND CONVECTIVE EXCHANGES IN A LARGE OPEN PLAN OFFICE

INTRODUCTION

Computer simulations of the dynamic behaviour of buildings often meet significant discrepancies between predicted and observed indoor temperatures when representing large office areas /3/. Quite important temperature differences indeed occur between the central part and peripheral locations close to outside walls due to heat accumulation from lights and occupancy. An eventual improvement of the simulations consists in splitting the room into a central core and peripheral zones. It is then assumed that the occurence of large air movements produces a certain amount of convective heat transfer that can be represented by conduction through fictitious walls separating zones with appropriate heat transfer coefficients.

In the frame of the International Energy Agency (Annex IV), the monitoring of an administrative building situated in the suburbs of Glasgow (UK) was carried out. Within this continuous data acquisition period, mainly concerned with airflow rates and temperatures in the heating and ventilation systems, indoor air temperatures in the building and meteorological parameters, a one-week measurement campaign was devoted to the recording of air velocities and associated temperatures in an unoccupied part of the building's second floor (figure 1). It took place at the end of April 1983, a rather chilly but sunny period of the year.

The instrumented storey is pictured on figure 1. Its largest dimensions is aligned on an east-west axis. It is an open plan office accomodation of about 4500 m^2 and 3.3 m high in the occupied zone. Its perimeter consists of double glazed tinted glass from floor to ceiling. Plaster tiles separate it from a ceiling void where ventilation supply ducts are running. The floor is covered with carpeting and separates the office area from the first floor ceiling void. A central partition running north-south separates the actually occupied zone from an empty space where measurements were performed.

On figure 1, 5 m deep peripheral zones are shown in the way they are defined by computer simulations. They are actually determined by the way heating and ventilating air is supplied. Separated variable-air-volume systems (VAV) serve the central core and the peripheries. Air is delivered through linear openings running in the ceiling along outside walls and blown towards the center. Is is also distributed in the central core through a regular network of square grids. Openings located close to the limits between zones are designed in order to blow air inwards. Air exhausts consist of pyramidal ventilated light fittings distributed regularly throughout the ceiling. Finally, heating is provided on the periphery by a constant-air-volume delivery (CAV), whose openings are facing glazings in order to cast off cooling by heat exchange with the outside and infiltrations.

INSTRUMENTATION AND EXPERIMENTAL PROCEDURES

Air velocity measurements were performed with 16 low velocity thermal anemometers, thermistors and TNO Delft heat compensated probes (see /5/ through /7/ and /12/). Their range lies from 5 to 10 cm/s and above 100 cm/s, however very low velocity recordings should be expected only with a relatively poor accuracy. Air temperatures were measured with 5 aspirated thermocouples columns recording data from 25 cm to 250 cm of height. Additional temperatures were associated to thermistors measurements.

The IEA monitoring data acquisition provided global air flow rates and temperatures in the HVAC system. A calibrated reference air supply allowed to characterize outlets, when compatible. Static pressures in outlets and jets velocity fields in their vicinity were investigated and provided a detailed distribution of airflow rates throughout the delivery networks.

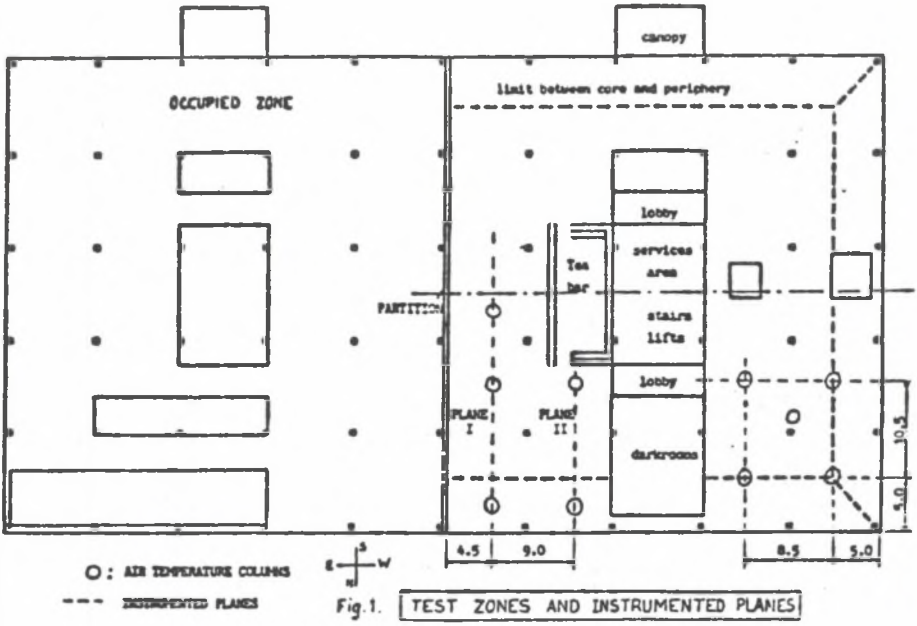

Fig. 1. Test zones and instrumented planes

Rys. 1. Strefy pomiarowe i płaszczyzny w których umieszczono urządzenia pomiarowe

Figure 1 shows the chosen experimental planes and depicts the location of aspirated columns which remained fixed during the test periods. The southern area was not considered, due to its too large dimensions for the amount of available sensors but mostly in order to avoid the unsteady influence of direct solar heat gains on recordings, as routine tests-assuming steadiness-lasted approximately one hour. First tests investigated a northern site between the partition wall and the services area because one-directional effects would be expected. Plane I was extensively instrumented while plane II served as a reference and notably noticed transverse airflows and temperature variations. Detailed velocity profiles (using 8 sensors per vertical) were recorded at fixed locations along plane I, namely at the limit between periphery and central core and at some 10 meters deep position. Airflow rates could then be deduced. A general picture of the flow was provided by verticals carrying only 4 sensors placed at heights of most significant air movements (i.e. in an upper air layer along the ceiling and in the return flux along the floor). They were moved along plane I and collected data every one or two meters. The air flow pattern was specified by intensive smoke visualizations. Selected wall temperatures were simutaneously measured on floor and ceiling at the location of the aspirated columns.

Once the air diffusion process on the northern site was clearly determined, the north-west corner was investigated. Quite complex air movements were observed, due to crossing flows interactions creating strongly vortical and turbulent regions. Those later tests mainly confirmed or modulated the conclusions drawn from the previous experiments.

The IEA exercise brings to the fore horizontal temperature differences between the storey's center and the peripheries that can reach 1 to 2°C. They prove to remain rather symmetrical with respect to a median plane running east-west. On the other hand, vertical variations seem to seldom exceed 0.5°C on the 3.3. m. height as far as the central core is concerned. Those differences are caused by a concentration of heat loads in the central area due to lighting but mostly occupancy during working hours. The heat accumulated in walls, floor and furniture is then slowly discharged to the room during night or week-end.

This behaviour was absolutely impossible to simulate with accuracy with only simple devices. The only requirement that the measurement campaign was able to fulfill was to create and maintain significant horizontal temperature differences. It was realized with the help of 10 blowing heaters for a total load of approximately 20 kW. They were distributed over rather large regions of the central core respecting observed dissymmetries. The use of convective instead of radiative heaters created a better load in volume and provided its horizontal dispersion. However, it was ensured by rising plumes of warm air inducing rather important natural convection movements and overemphasizing vertical temperature variations. Heaters were consequently kept at some distance of the regions where meaningful measurements were performed in order to minimize the influence of such internal air movements.

Another drawback inherent to the use of such convective heaters consists in the fact that they only load a certain air volume and are absolutely inefficient in loading surrounding walls. As a result, when they are put out of operation, the heated zone is quite immediately discharged. Consequently, they had to be kept in operation even when simulating week-ends and nights conditions, in the absence of ventilation. In those cases, they did not represent any actual load anymore.

Those remarks might raise questions about our simulating correctly reality. However, errors introduced by this loading procedure certainly range within the order of magnitude of uncertainties linked to air movements due to people moving around as well as the presence of very many complex obstacles, namely furniture, in actual occupancy conditions.

NATURAL CONVECTION

Tests of natural convection situations were performed during night, after the central part of the instrumented zone had been loaded for several hours. Figure 2 presents a sectional view along plane I showing measured velocities and air flow tendancies. A typical double S-shaped velocity profile is represented suggesting a division of the room's height into four air layers. Heat exchange results mainly from warm air flowing along the ceiling from core to periphery and cooler return air creeping along the floor. Intermediate layers experience very slow movements and indeed lami-

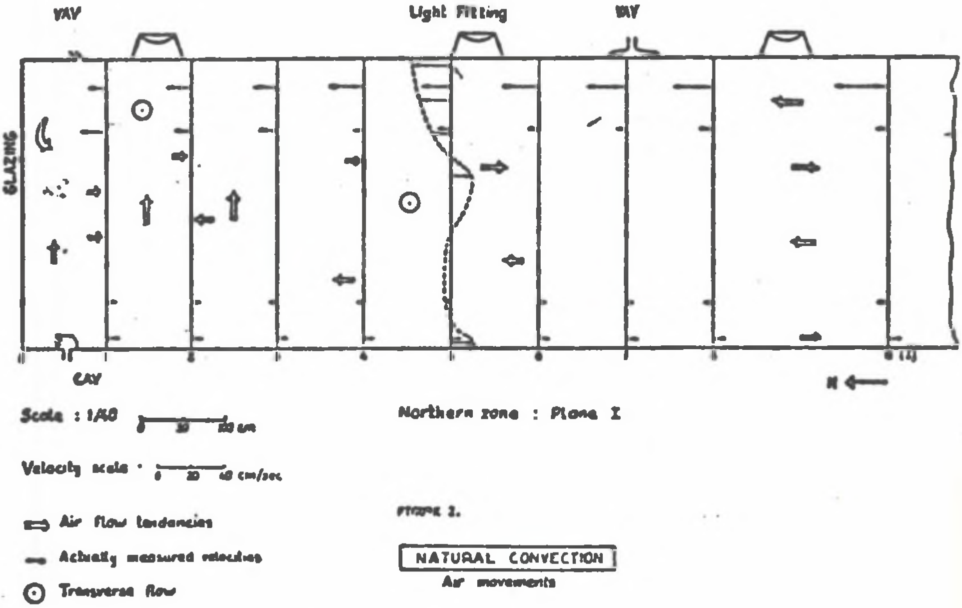

Fig. 2. Natural convection

Rys. 2. Konwekcja naturalna

nar stratification occasionnaly occurs. A mean horizontal temperature difference of 0.9°C is observed between zones, with a maximum reaching 1.5°C at mid-height. Strong vertical variations occur : 3.5°C in the center but still 3.1°C in periphery. Transversally, i.e. from plane I to plane II, very slow movements are barely noticed and temperature gradients are negligeable.

Conservation of mass specifies that any amount of air flowing in one direction should be canceled by an equivalent flow rate in the opposite one. Detailed velocity profiles at two fixed locations provide an estimated 0.120 m^3/s per unit width of net air transfer with an overall accuracy of 20% on the mass balance. This error is however equivalent to an air transfer that would flow at a velocity around 2 cm /s, below measurements sensitivity.

One is now able to estimate airflow rates in both upper and lower air layers at any measurement location and imagine compensation flows through intermediate layers in order to respect the above conclusions. Knowing the temperature distribution at any location, air enthalpies can be calculated and the amount of heat transfer finally can be expressed. Repeated calculations at all available positions show a quite low dispersion in the results and conclude to averaged heat transfer of the order of 70 W/m^2. In order to assess the validity of the procedure followed here, a heat balance can now be expressed on elementary volumes of unit width defined by the successive instrumented positions. Convection coefficients along walls are deduced from available velocities and wall temperatures, assuming forced convection along the ceiling and natural convection along the floor. On such subvolumes, energy conservation is respected within 20% of the calculated convective coupling. A subsequent check of the procedure is finally performed by expressing the heat balance of the whole loaded central area between the partition wall and the services area. This last budget is satisfied within 50% of the total load, a quite acceptable figure, nevertheless, when considering the uncertainty interent to such contributors as lights, infiltrations... etc...

Tests in the north-west corner had to be carried out with a heat load distributed on a much larger area in order to ensure symmetries. Locally, temperature gradients were therefore lowered and quite slow air movements were detected.

Dominant flows occured in the north-south direction. Movements towards west were barely visible in the instrumented area and took place only in regions closer to the median symmetry plane running east-west. Consequently the conclusions drawn from observations in the northern site are probable overestimates for the western periphery.

Resultant temperature fields recorded during the night at mid-height suggest even more complex couplings. When the central core is relatively uniformly loaded, temperatures decrease from center to the north-west corner, turning around the service area along the southern site. Circular air movements are thus eventually possible.

CONVECTIVE HEAT TRANSFERS UNDER HVAC CONDITIONS

Figure 3. represents a sectional view along plane I in the northern instrumented site showing measured velocities and airflow tendancies under HVAC conditions. Entrainment by the warm ascending CAV jet along the window is clearly visible, meeting a strong cooling jet issued from the peripheral linear VAV outlet. Cold air falls down significantly at about 4 meters from the glazing and a first cellular pattern is observed, approximately confined to the peripheral zone. On the other hand, warm air flowing from the central core arrives along the ceiling barely influenced by the central VAV supply and meets the diffusing cold jet from the periphery. Along the floor, return air creeps towards the center. In the vicinity of the limit between core and periphery, very turbulent and vortical mixing occurs. The mean temperature difference again reaches 0.8°C, while the maximum appears above mid-height due evidently to the presence of cool air in the upper layer of the periphery. Vertical variations are still important but stay below 3°C and experience an inversion in the periphery. Again transverse effects are negligeable.

Due to the presence of specific injections of air, conservation of mass only applies on well defined volumes. A first approach consists again in considering the whole air volume between the partition wall and the services area (figure 1). It is then divided into zone I from glazing to a 3 meters deep transverse plane. Zone II lies between that limit until a 10 meters deep plane, separating the unloaded core from the region where blowing heaters are distributed. This zone contains the mixing region, where unstable movements do not allow any rigorous local analysis. Zone III,

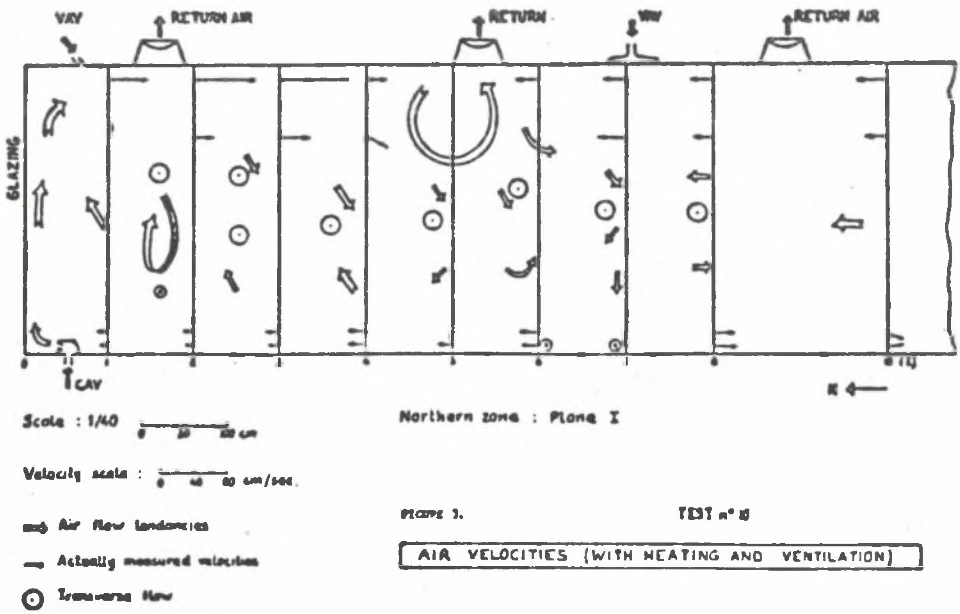

Fig. 3. Air velocities (with heating and ventilation)

Rys. 3. Prędkość powietrza (z ogrzewaniem i wentylacją)

finally, contains the convective heaters. Mass conservation yields the various flowrates mentioned on figure 4, allowing a non negligeable air transfer from zone I to zone II .

Next a further subdivision is performed, taking into account the regularity appearing in the VAV outlets and air exhausts networks. It results in the definition of an elementary slice, symmetrical with respect to the instrumented plane (plane I on figure 1) and 3.65 m wide. Again zones I, II, III are defined and air flow rates determined. On such zones, velocity profiles again provide air rates, compensatory fluxes and air enthalpies. Resulting convective coupling can be deduced, carefully separating the contribution of the peripheral VAV jet flowing accross the 3 meters limit from the actual coupling due to a globally compensated airflow through that plane. Heat balances may then be expressed as sketched on figure 5.. Conservation of energy proves to be reasonably respected.

The conclusion of the above analysis consists in an observed heat transfer of the order of 150 W/m² from the loaded core zone to the unloaded part. This figure may be considered as an overestimate, because of the vicinity of the heaters. Most of this heat is diffused in the mixing region surrounding the limit between core and periphery and evacuated through air exhausts. Only remains a 30 W/m² coupling on the 3 meters limit, however still oriented towards the glazing. We should mention here that those values are probably less accurate than the coupling proposed for natural convection, though an error estimate is difficult to assess.

From the aforementioned results, it may seem hazardous to deduce a unique convective coupling between core and periphery without changing any definition of those regions. It would

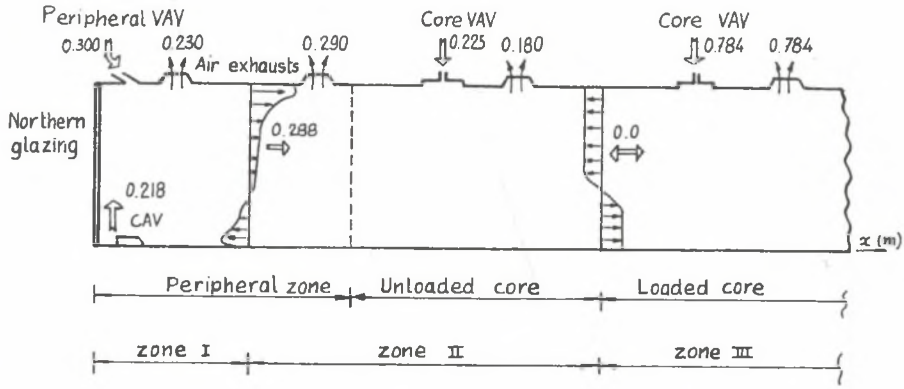

Fig. 4. Sketch of air transfers in HVAC conditions

Fig. 4. Szkic wymiany powietrza w warunkach ogrzewania, wentylacji i klimatyzacji

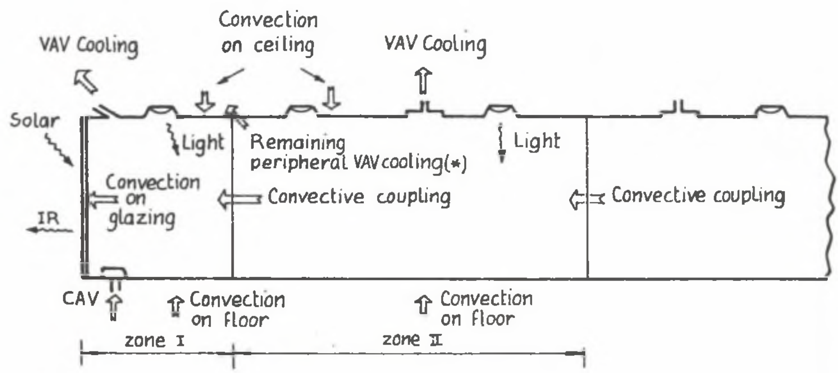

Fig. 5. Sketch of heat balances in HVAC conditions

Rys. 5. Szkic bilansów cieplnych w warunkach ogrzewania, wentylacji i klimatyzacji

probably range closer to 30 W/m² and a figure around 50 W/m² could eventually be envisaged. It is quite interesting that a heat budget expressed on the whole loaded central core is exactly balanced if an 85 W/m² coupling between zones is assumed.

In the north-west corner, the air movements become essentially three-dimensional. Along the northern glazing, the characteristic cellular pattern is still observed. When reaching the limit between core and periphery, the cool VAV jet has fallen to an height of approximately 2.5 m. and allows warmer air to worm in between, along the ceiling. Those observations are still valid right in the corner due to the absence of VAV outlet on the western wall at the corner's edge.

However, at floor's level, air shows a tendancy to flow west, more and more obvious as one gets near the corner. The various flow crossings evidently create strong shear stresses and complex turbulent movements result. Quite some mixing should here be expected.

Along the western glazing, the cool VAV air flows east, while along the floor returning air takes a south, then south-west direction. The characteristic exchange perpendicularly to the core-periphery limit occurs only at 15 m from the edge.

No definite conclusions can be drawn from measurements in the corner, due to lack of some crucial information. However the general picture of the flow suggests that corners should be considered as square buffer zones -10 to 15 m deep-in which quite strong mixing should occur. Globally their effect probably weakens the convective couplings as described on the northern site. This influence is certainly more important for the exchange between center and western periphery because this side of the building presents the shortest dimensions.

CONCLUSIONS

Air flow movements and temperature fields in both natural convection and HVAC conditions in a large office building have been investigated. They show a quite definite coupling between a warm central core and cooler peripheral zones. Heat exchanges are estimated and prove non negligeable. During HVAC conditions, regular cellular airflow movements present physical evidence that peripheral zones behave in a significantly independant way, but that still heat exchanges between zones occur. Investigation in one of the office's corner suggests that strong mixing subzones in the angles tend to reduce the global heat exchanges by convection between zones. An interesting question raised by this work concerns the limitations of the eventual application of the present results to real life situations.

ACKNOWLDGEMENTS

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WYMIANA KONWEKCYJNA W KOMORZE KLIMATYCZNEJ ORAZ W HALI BIUROWEJ O DUŻEJ POWIERZCHNI

S t r e s z c z e n i e

Przedstawiono ogólny przegląd badań eksperymentalnych w dziedzinie dyfuzji powietrza w zamieszkałych budynkach. Badania podzielić można na dwie odrębne części, z których pierwsza opisuje bad nia zachowania się strug wewnątrz komory klimatycznej (badania laboratoryjne) oraz wpływ różnych źródeł ciepła, łącznie z wentylacją mechaniczną, na oszacowanie zużycia energii.

Część druga przedstawia wyniki krótkiej sesji pomiarowej przeprowadzonej w dużej hali biurowej w celu otrzymania lepszego oszacowania konwekcyjnej wymiany ciepła występującej między strefą centralną i obrzeżną, w których obserwuje się znaczne przestrzenne różnice temperatury.

КОНВЕКЦИОННЫЙ ОБМЕН В КЛИМАТИЧЕСКОЙ КАМЕРЕ И В КОНТОРСКОМ ПОМЕЩЕНИИ С БОЛЬШОЙ ПЛОЩАДЬЮ

В работе даётся общий просмотр экспериментальных исследований в области диффузии воздуха в жилых зданиях.

Исследования можно подразделить на 2 отдельные части . Первая занимается исследованиями струй внутри климатической камеры (лабораторные исследования) и влиянием разных источников тепла, вместе с механической вентиляцией, на оценку расхода энергии. Вторая часть представляет результаты короткой измерительной сессии, проведённой в большом конторском помещении, с целью получения лучшей оценки конвенционного обмена тепла между центральной и береговой зонами, в которых выступает значительный пространственный температурный напор.