The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

A Design Guide for Thermally Induced Ventilation

C Filleux*, S Krummenacher*, P Kofoed**

* Basler & Hofmann Consulting Engineers, CH 8029 Zürich, Switzerland

** Sulzer Infra Laboratory, CH 8401 Winterhur, Switzerland

Synopsis

A design guide for displacement ventilation (thermally induced ventilation) has been prepared. It is based on quasi stationary experiments carried out in the Sulzer Infra laboratory in Winterthur. The significant design parameters identified by factorial analysis are the air flow rate, the internal load, the convective part of the internal load and to a lesser extent the room height. Using a linearized polynom representation for the temperature increase near the floor as well as for the vertical temperature gradient in the occupied zone a design nomogram has been obtained. Within its range of application the design nomogram also applies for displacement ventilation systems combined with cooled ceilings. The design guide is published in german and french language and is one of the major outputs of the Swiss research program on «Energy Relevant Air Movements in Buildings (ERL)».

Background

Detailed knowledge on displacement ventilation has tremendously increased in the early nineties. International cooperation has encouraged the refinement of the technique of Computational Fluid Dynamics and numerous full scale experiments and field tests have produced data on room air flow. These activities mainly took place in the scientific world.

In the consulting branch we observe a strong market penetration of low velocity inlets for displacement ventilation. Manufacturers have produced video films to demonstrate room air movements and have made their air supply units more attractive for architects.

It is only very recently, that P.V. Nielsen (Displacement ventilation - theory and design, 1993) and H. Skistad (Displacement ventilation, 1994) have published comprehensive books in english language. Until then, the system designer only had the choice to either carefully study the limited literature or to directly discuss the design with manufacturers. In the latter case one has no tools to verify the design parameters proposed by the manufacturers and becomes very quickly product-dependent . It is in the view of giving better support to system designers, that the elaboration of a design guide has been started 1992 in Switzerland.

We here give a brief overview of the experimental procedure, show some aspects of the results obtained and finally demonstrate the nomogram developped for system design. For more details the design guide itself as well as the various scientific reports must be consulted.

In the following we will use the term «Thermally Induced Ventilation» to describe a ventilation system where the air is driven by heat sources (apparatus, persons) or sinks (cold windows). In the scandinavian literature such a system is often referred as displacement ventilation. We feel that the term displacement ventilation is to closely related to piston flow type systems and therefore is not the appropriate term to describe the physical processes which are driving the room air movement.

Planning of the Experiments

From our own practical experience in designing ventilation systems and from the experimental data available we identified a set of parameters likely to be relevant in the design of thermally induced ventilation systems (Table 1).

Design parameter	Influenced quantity
Room height	Temperature profile, range of applications
Air supply rate	Thermal comfort, air flow pattern, cooling capacity
Supply air temperature	Thermal comfort, air flow pattern, cooling capacity
Internal load (source strength, location of the source, convective part)	Air flow pattern, temperature profile
Radiative cooling	Cooling capacity, range of applications

Table 1: Relevant parameters in the design of thermally induced ventilation systems

The range of value of the design parameters to be chosen depends on the specific application. Here we limit ourselves to single cell office rooms and meeting rooms. The range of the above parameters for such rooms was therefore chosen as following (Table 2):

Design parameter	"low value"	"high value"
Specific air flow rate	10 $m^3/h/m^2$	20 $m^3/h/m^2$
Internal load	10 W/m^2	30 W/m^2
Convective part of internal load	20 %	80 %
Room height	2.4 m	3.6 m

Table 2: Range of value of design parameters investigated

The following comments have to be made: The supply air temperature was not considered as an independent parameter in the experimental set-up, since it is connected to the supply air velocity through the Archimedes number. It was kept at a constant temperature (18° C) for both summer and winter conditions. The "low value" for the specific air flow rate has been chosen relatively high compared to the swiss recommendations for mininum air flow rates in non-smoking areas of office buldings. The values for internal load apply to thermally induced ventilation without cooled ceilings.

The factorial design method was used to plan the number of trials (experiments) in the laboratory. This method has the advantage to significantly reduce the number of trials and identifies those variables (design parameters) which have the largest influence on a control quantity. Control quantities are e.g. the temperature gradient in the occupied zone or the room air temperature at floor level. They can be represented in the following form:

$$Y=b_0+b_1x_1+b_2x_2+b_3x_3+b_4x_4+....$$

were x_1, x_2, etc are the variables (design parameters). With four variables the number of trials is $2^4=16$. All variables were either set to the "low value" or to the "high value" of table 2. Three "Zero-trials" with intermediate values were added in order to test the linearity.

Experimental Set-up

All experiments were performed under quasi stationary conditions at the Sulzer Infra Laboratory in Winterthur. The measurements took place in a full scale furnished room of 7.25 x 7.25 m. A thermally induced ventilation system with wall mounted diffusers, through one entire sidelength, and two exhaust openings under the ceiling were installed. Air temperatures and air velocities were measured at three different positions, at two working places and free placed, in different heights. In addition surface temperatures were registered. The velocity measurements were carried out using a Dantec low velocity multiflow analyzer with temperature compensated sphere probes. The temperatures were measured with thermocouples connected with a HP precision Voltmeter.

Example Results

The measured air temperature distribution (Figure 1) corresponds to the pattern known from measurements in other laboratories. At higher (30 W/m2) thermal load the room air temperature is shifted towards higher temperatures and its distribution is much more inhomogeneous than with a relatively low thermal load (10 W/m2). In addition a larger convective part of the thermal load increases significantly the temperature gradient in the occupied zone.

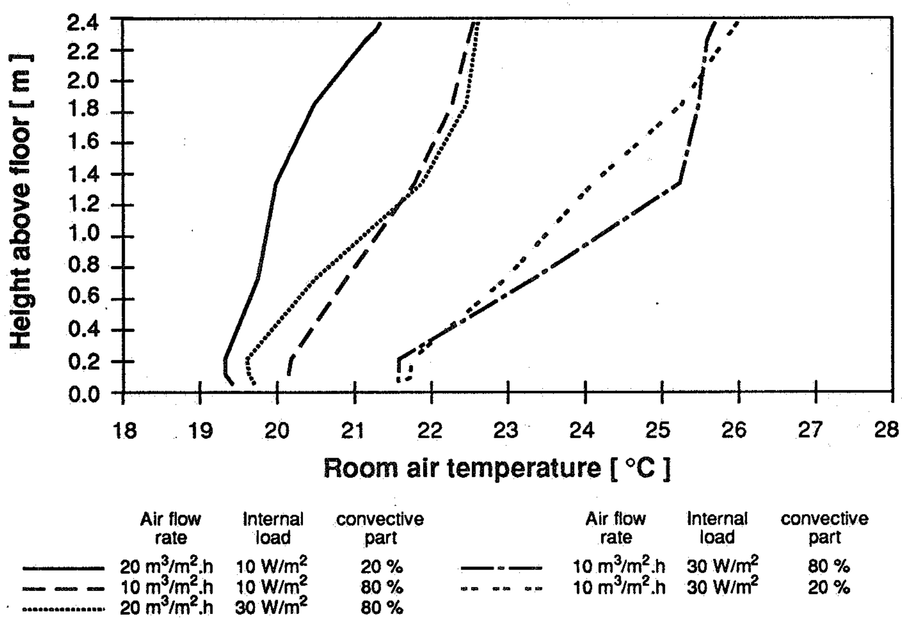

Figure 1: Air temperature distribution for thermally induced ventilation

The air velocity profile has also been measured but is not shown here. In the occupied zone, outside the rising thermal air flows, typical values for air velocities are 0.03 to 0.07 m/s, turbulence indices range from 5 to 50%. The corresponding PPD (pecentage of persons dissatisfied) with 5% is at it's minimum. The highest velocities are found between 5 and 10 centimeters above the floor.

Model assumptions

In many cases the temperature distribution in a room is nearly homogeneous and the temperature measured close to the floor appears to be approximately half way between the supply air temperature and the extract air temperature. Based on these findings the 50/50 model was stipulated several years ago in scandinavian countries (see figure 2). This model assumes that 50% of the temperature difference in the room is taken up in the floor. This applies to rooms of conventional height (2.4m to 3.6m) and to"normal" heat loads.

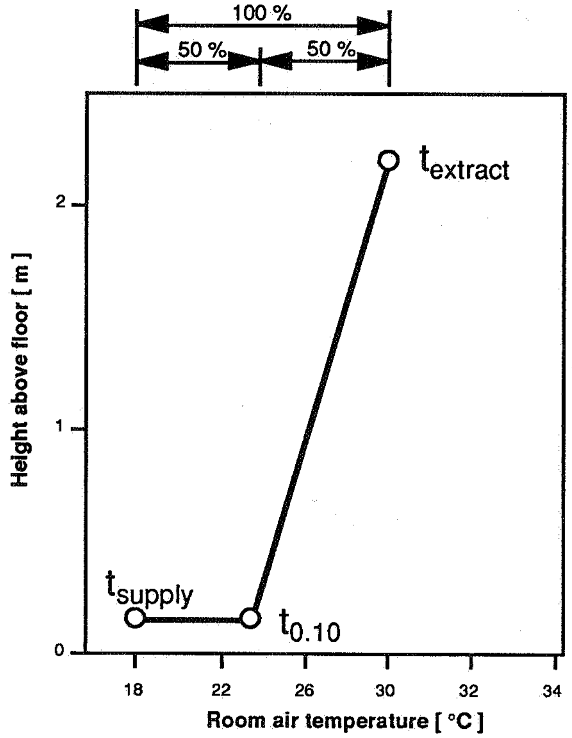

Figure 2: The 50/50 model

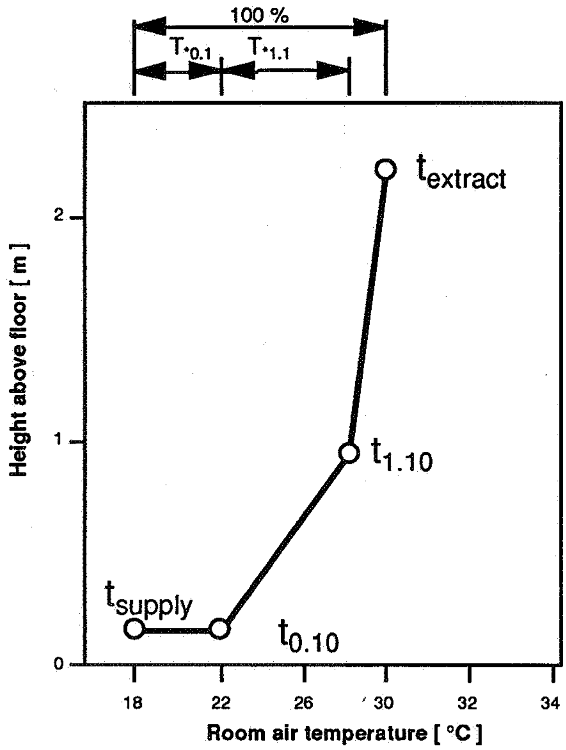

Figure 3: The ERL model

Our own observations show, that a homogeneous distribution is only observed under conditions of very low heat load (10W/m2) and for supply air flow rates in the range of 10 to 20 m3/h/m2 (see figure 1). When the total heat load reaches higher values (which is normally the case) and/or when solar radiation hits the floor, the vertical temperature distribution is no more homogenous. It shows a clear bend at a height between 0.6m and 1.4m depending on the heat load. This occurs because the convection currents become larger than the supply air flow and a mixing of the upper room air is created. As a consequence the measured temperature gradient in the occupied zone is often observed to be larger than the one predicted by the simple 50/50 model. Therefore we looked for a model which better fits the measured data (see figure 3). Since in Switzerland the comfort conditions must be fulfilled in a zone reaching from 0.1 to 1.1m above floor, these marks were taken to calculate the temperature rise near the floor and the vertical temperature gradient in the occupied zone. Note that the difference in height is exactly 1m, so that the temperature rise equals the vertical gradient in the occupied zone.

Data reduction

The analysis of the measured data according to the factorial design method gave the following equations for the dimensionless control quantities:

Temperature rise near floor: $T^*_{0.1}=\{45.5-2.2x_1-2.4x_2-2.1x_4\}\cdot 10^{-2}$ [eqn.1]

Temperature gradient in occupied zone: $T^*_{1.1}=\{57.3-6.6x_1+10.5x_3\}\cdot 10^{-2}$ [eqn.2]

In the dimensionless form the variables x_1,...x_4 take either the value 1 or -1 (see appendix for the case of the temperature gradient). For numerical interpretation the dimensionless variables can be transformed into the design parameters using the following relations:

$x_1=0.2v-3$; $x_2=0.1q-2$; $x_3=0.33q_c-1.67$ and $x_4=1.67r_h-5$

where	v	=specific air flow rate	[$m^3/h/m^2$]
	q	=total heat load	[W/m^2]
	q_c	=convective part of load	[%]
	r_h	=room height	[m]

Graphical representation of the temperature rise near the floor and the temperature gradient can be obtainied from equation 1 and equation 2 respectively using the above transformations. Finally, both representations can be combined in a nomogram. The following simplification were made:

- The relation between temperature gradient in the occupied zone (1.1m-0.1m) and the load has been linearized,
  and
- the nomogram applies for all room heights between 2.4m and 3.6m.

Design procedure

When designing a ventilation system, the designer must answer the questions "What is the purpose of the ventilation system ?" and "Is the priority to remove heat or to provide good air quality ?". In many cases there will be no clear answer to the second question, and consequently no "best" system. It is however important to have in mind, that displacement ventilation is a system for the best possible air quality, but without cooled ceilings is not appropriate if one deals with a large heat load.

The various steps used in the design procedure are as usual. The design nomogram (see figure 4) is a useful tool to determine the supply air flow rate and to check the temperature gradient in the occupied zone as well as the temperature rise near the floor in one single step. It also offers the possibility to vary the convective part of the heat load.

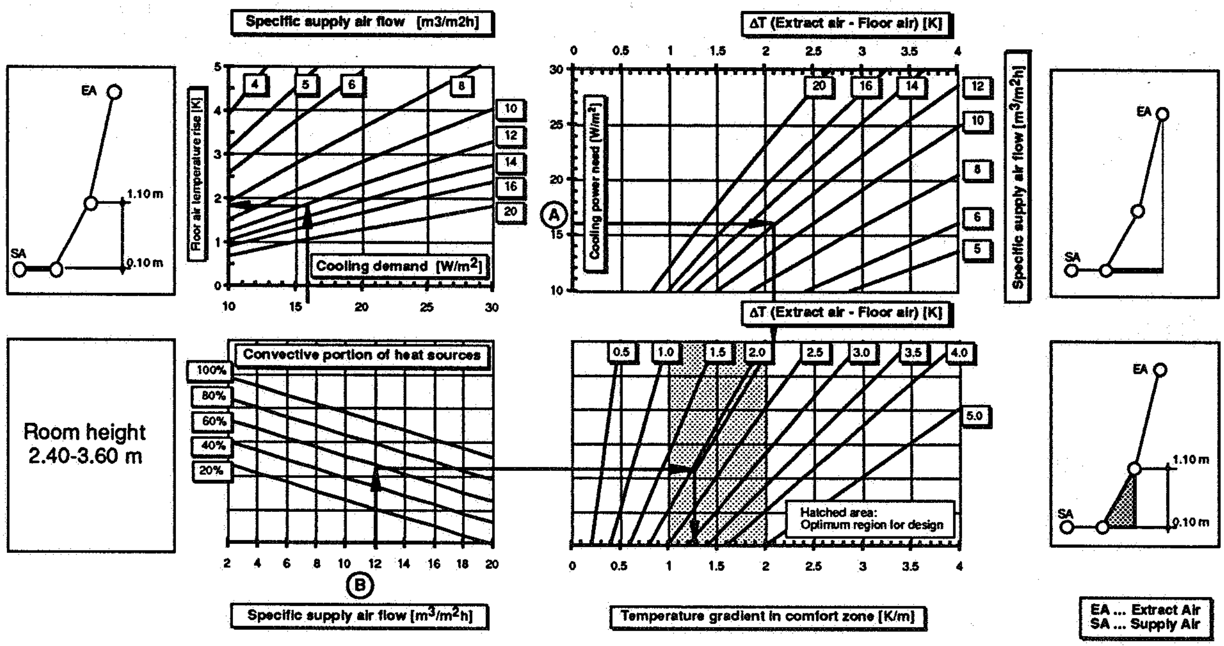

Figure 4: Design nomogram for thermally induced ventilation (displacement ventilation)

In order to illustrate the use of the design nomogram, an example calculation for a single cell office is shown in table 3. The results are compatible with those obtained from other design procedures (see e.g. H. Skistad).

Dimensions and data			*Determine air flow rate*	
Single cell office	Floor area	25 m2		
	Room height	2.8 m	Cooling load*	$q = 16\ W/m^2$
Internal load	2 Persons	240 W	Convective part of internal load	$q_c = 57\%$
	2 PC	200 W		
	1 Laser-Printer	100 W	Specific air flow rate	$v = 12\ m^3/m^2/h$
	Lighting	250 W		
			Tempcrure gradient	$T_{1.1} = 1.3$ K/m
External load		0 W		
			Temperature rise near floor	$T_{0.1} = 1.8$ K
	TOTAL	790 W		

* The cooling load is smaller than the total of internal and external load due to heat storage in the building fabric and non-simultaneity of partial loads

Table 3: Example calculation for a single cell office room

Acknowledgements

The authors wish to thank all scientist in research and industry who have participated in the task. They are to numerous to be listed here.

Our special thanks go to our scandinavian friends Professor Eystein Rodahl and Hans Martin Mathisen from Trondheim as well as Professor Peter V. Nielsen from Aalborg who encouraged the work and gave us advice in setting up the experiments.

The work has been supported by the Swiss Federal Office of Energy (BEW) and by the National Energy Foundation (NEFF).

References

1. Nielsen, P.V.
"Displacement ventilation - theory and design, University of Aalborg, August 1993
ISSN 090Z-8006 V9306

2. Skistad H.
"Displacement Ventilation"
Research Studies Press LTD
John Wiley & Sons Inc., 1994

3. Filleux Ch., Krummenacher St., Aiulfi D. and Chuard P.
"Zeitgemässe Lüftungssysteme - Aktuelles Wissen und Leitfaden für die Projektierung" (Design guide)
Verband Schweizerischer Heizungs- und Lüftungsfirmen, 1994

4. Widder F.
"Overview of the Programm ERL"
Verband Schweizerischer Heizungs- und Lüftungsfirmen, 1994

Appendix

Auswertung 2^4 Versuche

Versuch	T** 1.10-0.10 [-]	0	1	2	12	3	13	23	123	4	14	24	124	34	134	234	1234
A	0.536	1	-1	-1	1	-1	1	1	-1	-1	1	1	-1	1	-1	-1	1
B	0.549	1	1	-1	-1	-1	-1	1	1	-1	-1	1	1	1	1	-1	-1
C	0.576	1	-1	1	-1	-1	1	-1	1	-1	1	-1	1	1	-1	1	-1
D	0.365	1	1	1	1	-1	-1	-1	-1	-1	-1	-1	-1	1	1	1	1
E	0.689	1	-1	-1	1	1	-1	-1	1	-1	1	1	-1	-1	1	1	-1
F	0.537	1	1	-1	-1	1	1	-1	-1	-1	-1	1	1	-1	-1	1	1
G	0.816	1	-1	1	-1	1	-1	1	-1	-1	1	-1	1	-1	1	-1	1
H	0.731	1	1	1	1	1	1	1	1	-1	-1	-1	-1	-1	-1	-1	-1
I	0.462	1	-1	-1	1	-1	1	1	-1	1	-1	-1	1	-1	1	1	-1
K	0.327	1	1	-1	-1	-1	-1	1	1	1	1	-1	-1	-1	-1	1	1
L	0.528	1	-1	1	-1	-1	1	-1	1	1	-1	1	-1	-1	1	-1	1
M	0.363	1	1	1	1	-1	-1	-1	-1	1	1	1	1	-1	-1	-1	-1
N	0.771	1	-1	-1	1	1	-1	-1	1	1	-1	-1	1	1	-1	-1	1
O	0.514	1	1	-1	-1	1	1	-1	-1	1	1	-1	-1	1	1	-1	-1
P	0.740	1	-1	1	-1	1	-1	1	-1	1	-1	1	-1	1	-1	1	-1
Q	0.630	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Wirkung		b0	b1	b2	b12	b3	b13	b23	b123	b4	b14	b24	b124	b34	b134	b234	b1234
Summe		9.1748	-1.063	0.4037	0.0007	1.6825	-0.148	0.4091	0.4256	-0.464	-0.231	0.011	0.2744	0.2267	-0.028	-0.482	-0.114
Mittel		0.573	-0.066	0.025	0.000	0.105	-0.009	0.026	0.027	-0.029	-0.014	0.001	0.017	0.014	-0.002	-0.030	-0.007

Table A: Factorial analysis for the temperature gradient

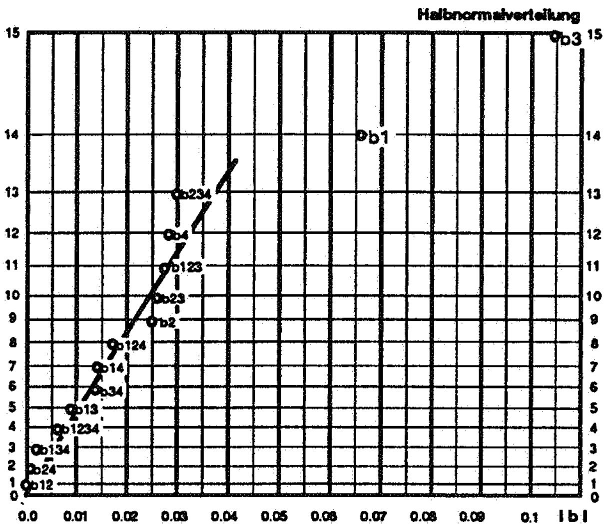

Figure A: Half-normal distribution showing significant coefficients for the temperature gradient