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# Investigation of a combined ventilation and heating system for resedential buildings

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# Investigation of a combined ventilation and heating system for resedential buildings

## Synopsis

Combined ventilation and heating systems in floors demand extensive investigations about the heat transfer before they could be installed in residential buildings. For basic investigation about the heat transfer two experimental plants with different duct geometries are build in a laboratory of the University of Essen. Especially the measurements of temperature on different places of the plants are taken to determine the heat transfer at the two floors. The evaluation of the test data show that a floor with a cavity which is built with cones is unqualified for a combined ventilation and heating system. The difference of the temperature between the air inlet and the room is too low, because the heat loss at the floor surface is too high. A good combination is the air duct system which is ending in an area of 1000 mm lenght containing a cavity which is built with cones. This system guarantees sufficient high temperature at the air inlet to obtain a comfortable room temperature.

## List of symbols

| | |
|---|---|
| k | coefficient of heat transmission [$W/m^2K$] |
| L | lenght [m] |
| q | heat flux [$W/m^2$] |
| Q | heat flow [W] |
| t | temperature [°C] |
| $\Delta T$ | difference of temperature [K] |

## Subscrips

| | |
|---|---|
| c | convective |
| log | logarithm |
| RL | indoor air |
| tot | total |
| 0 | reference level |

## 1. Introduction

Residential buildings are now better insulated than ever before because of the energy saving issues. Consequently the tightness of the buildings is increasing, but this requires the application of mechanical ventilation systems. On the European market today a lot of ventilation systems combined with conventional heating systems are already available. Another possibility is the use of a warm air heating system which combines ventilation and heating in the building. Basic investigations about the heat transfer of these systems are accomplished on two experimental plants with a great variation of working parameters. Hereby floors with different duct geometries are put in operation. To assess the working quality the splitting of heat flows into a part which release heat through the floor surface and a part which is transported by air is of particular interest.

## 2. Description of testing plants

The testing plants (Fig. 1) consist of two materials, the lower layer is heat insulating polyurethan and the upper layer is finish floor.

The difference between the two plants is the intermediate layer, which is in plant 1 a cavity built with cones and in plant 2 ducts which are cut in an additional layer of polyurethan. One of the ducts is also equipped with cones. Both ducts are ending in a cavity of one meter length built with cones and the supply air flows through the air inlet into the room. The outlet grilles should be installed in front of the external wall.

The supply air is heated up by a water/air heat exchanger which is admitted with electrical heated water. The temperature of the supply air at the entrance of the testing area variies between 29°C and 39°C. The range of variation of the supply air temperature during one serie of experiments was always lower than 3K. The difference between the supply air and the internal temperature was only 10K because of an insufficient cooling of the room.

The temperature sensors are installed at the entrance in the floor, the beginnig ( axis A) and the end (axis B) of the testing area and also at the inlet grille, see Fig. 1. For the determination of the air volume flow a device basing on the thermoanemometrical principle is installed over the inlet grilles.

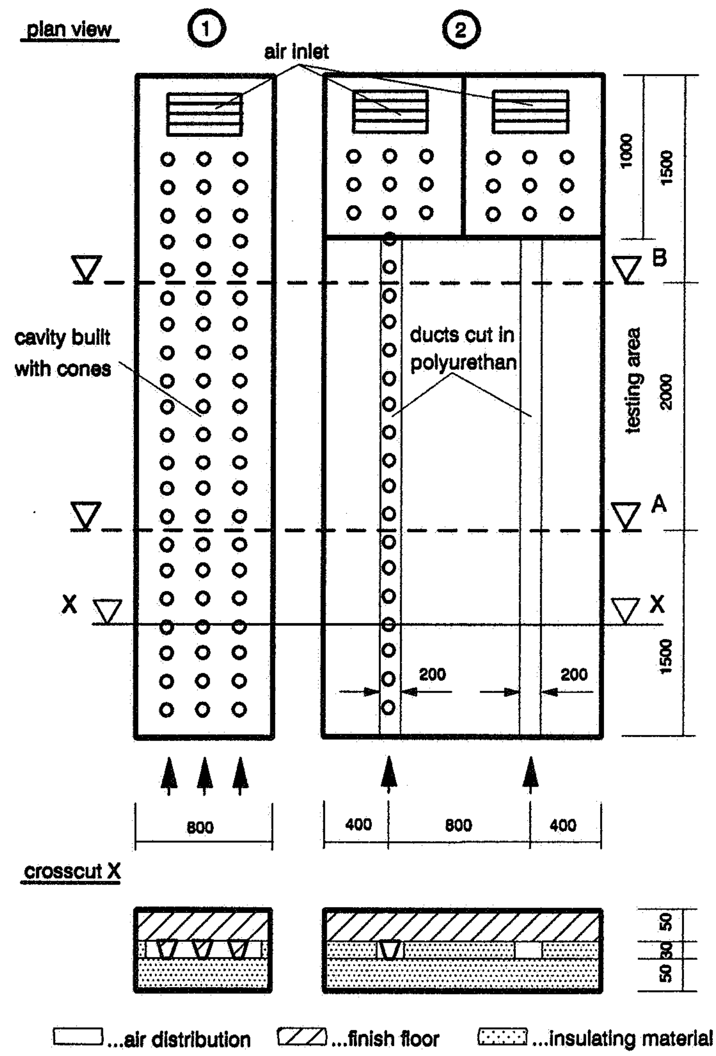


Fig. 1: Plan view and crosscut of the Testing Plant 1 and 2

## 3. Measurements and results

For a better understanding the results of the testing plant 1 and 2 are directly compared.

Fig. 2 and 3 show the temperature difference between the axis A and B as well as the convective heat loss related to the air flow rate. The temperature difference is standardized by the relation to the length of the testing area. As convective heat loss is considered from the supply air in the duct respectively the cavity to the floor surface. For the testing plant 1 the convective heat loss is related to the whole testing area (W/m$^2$), whereas for the plant 2 it is relative to the length of the ducts (W/m). The convective heat loss of the duct system depends only on the air flow rate and the temperature difference; for the cavity system also the width takes influence on the heat loss.

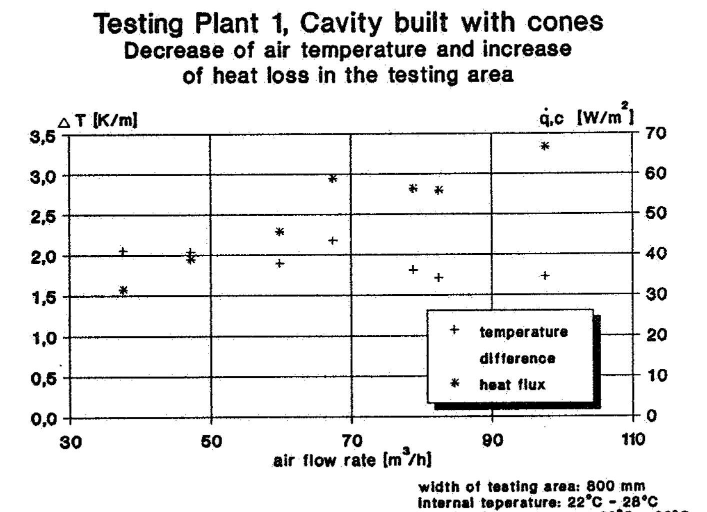


Fig. 2

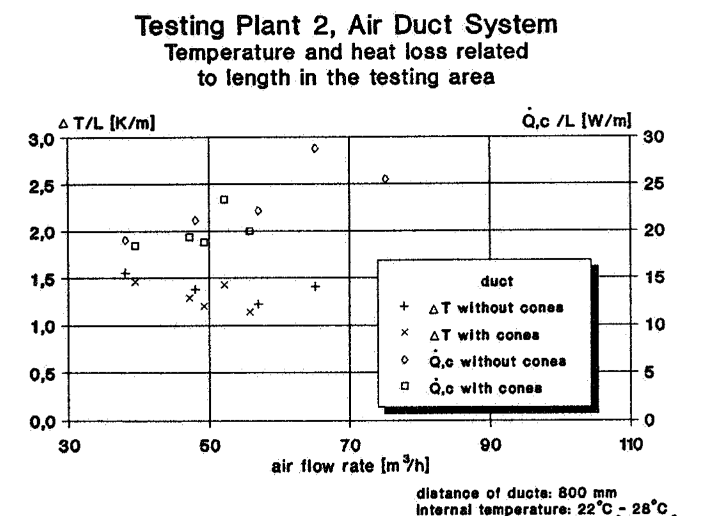


Fig. 3

It is evident that for both systems the temperature difference decreases slightly, whereas the convective heat loss rises strongly with increasing air flow rate. Considering an air flow rate of 50 $m^3/h$ the temperature difference of plant 1 is 2K, of plant 2 only 1,3K.

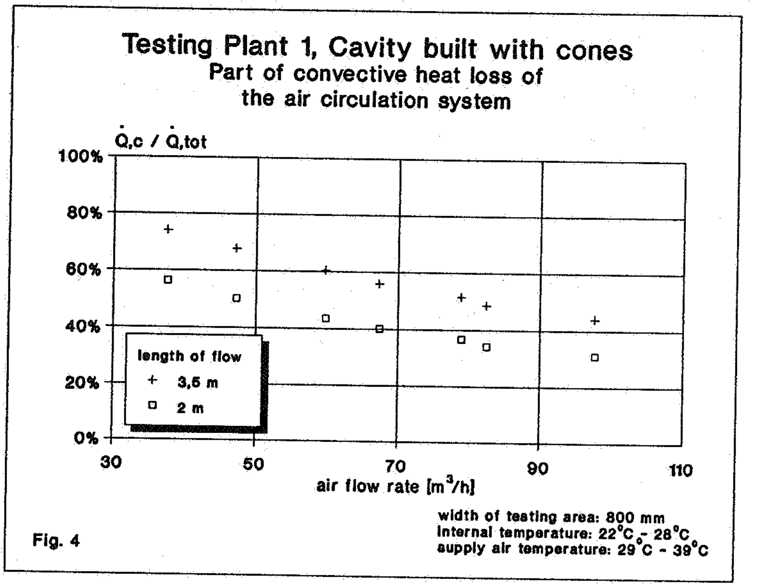


Fig. 4

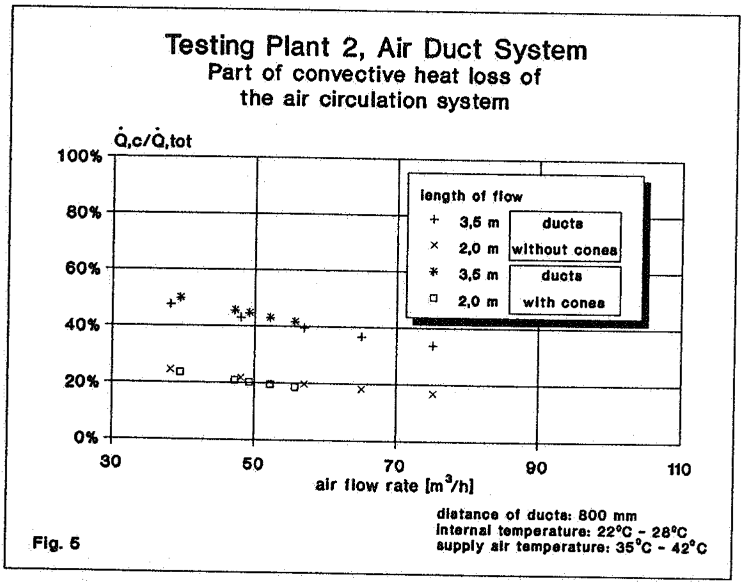


Fig. 5

Fig. 4 and 5 show the part of convective heat loss of the air circulation system related to the air flow rate. For an air flow rate like in practice of 40 $m^3/h$ the part of the surface heating in the testing plant 2 is 40%, whereas in plant 1 it's greater than 70% because of the great storing capacity of the floor. This effects low temperatures at the air inlet so that the system 1 is not qualified for the transport of warm air over a long way. On the other hand the temperature at the air inlet of the system 2 is high enough to guarantee a good performance of the combined ventilation and heating system. The controllability is also better as in system 1.

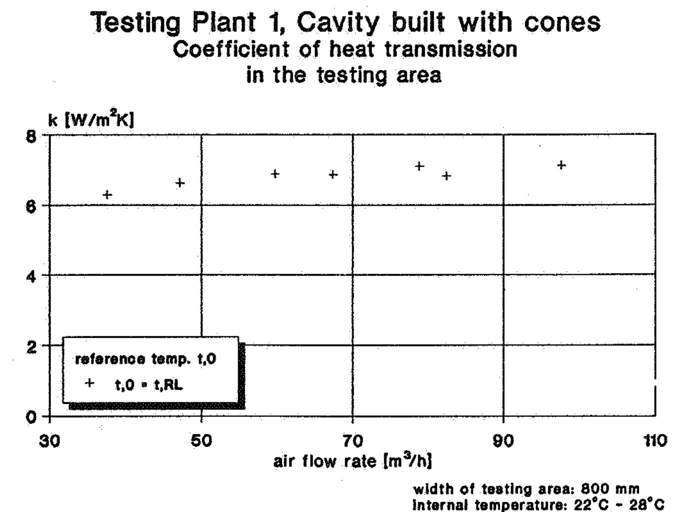

Fig. 6

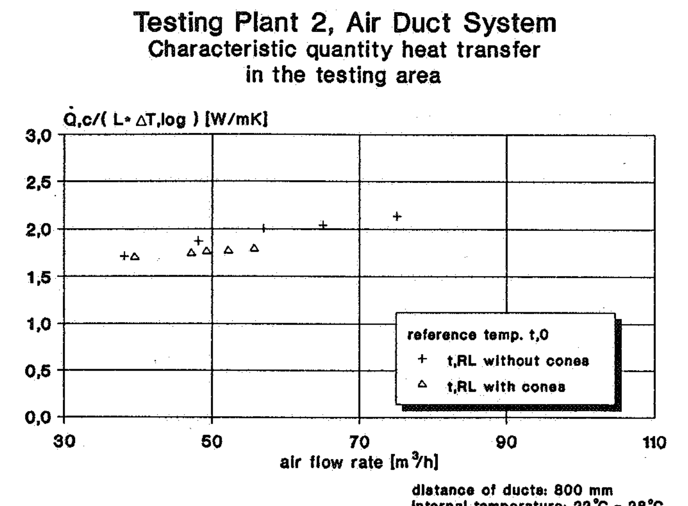

Fig. 7

The coefficient of heat transmission related to the air flow rate is shown in Fig. 6 and the characteristic quantity heat transfer in Fig. 7. These two figures are compared directly, because the distance of the ducts is the same as the width of the testing area (800 mm). Only for a distance lower than 500 mm the characteristic quantity is depending on the distance of the ducts. The coefficient of heat transmission in Fig. 6 with approximately 7 $W/m^2K$ is too high for a combined ventilation and heating system (see also Fig. 2). Fig. 7 shows evidently lower values.

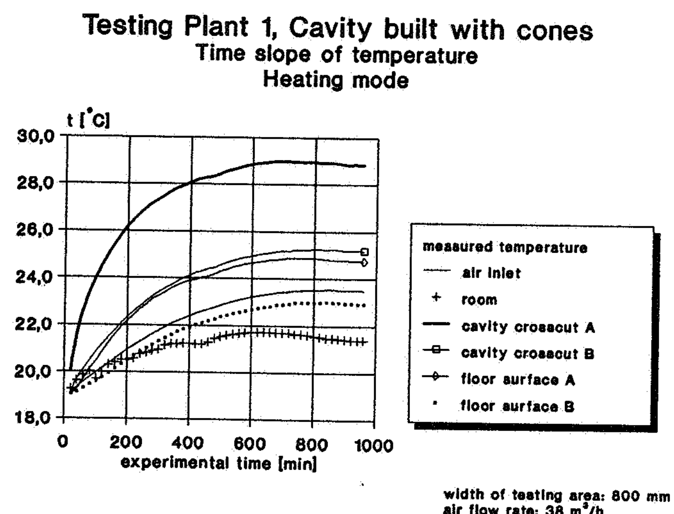


Fig. 8

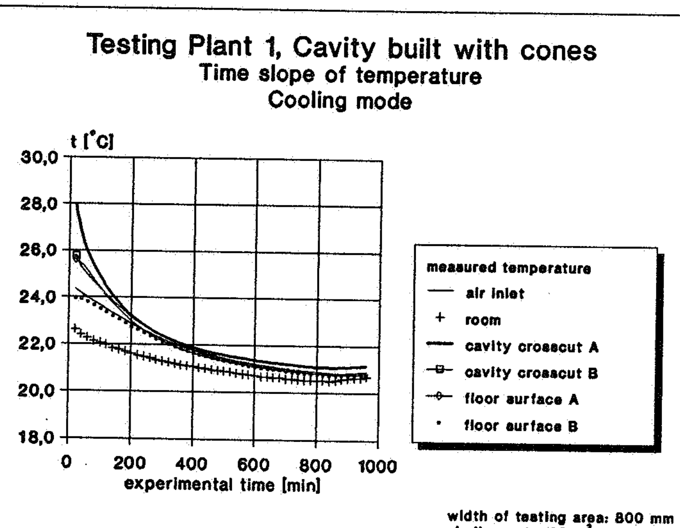


Fig. 9

The influence of the thermal inertia of the cavity floor construction for the heating respectively the cooling mode is recognizable in Fig. 8 and 9. Only after 10h working time the storage process in the floor was finished and the difference between the temperature at the air inlet and the room temperature is too low. Fig. 10 and 11 show a lower thermal inertia because the storing capacity is not so high. At the air inlet of the cavity floor built with cones the temperature increase about 2K after 300 min whereas the temperature at the air duct system increase about 2K after 30 min.

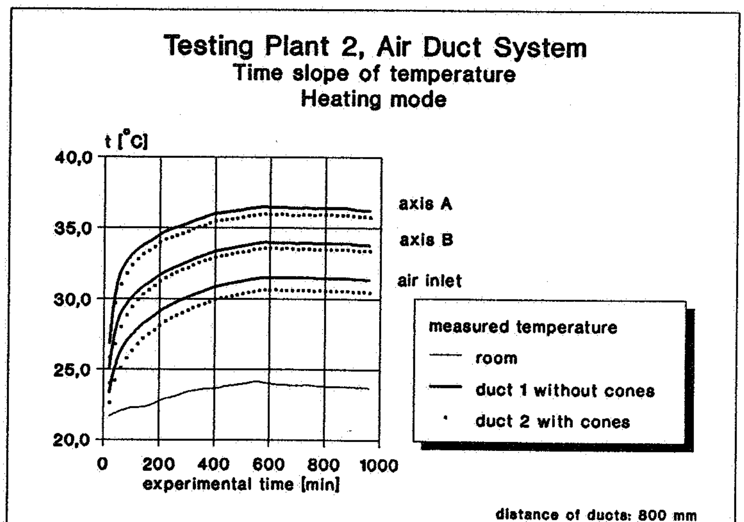


Fig. 10

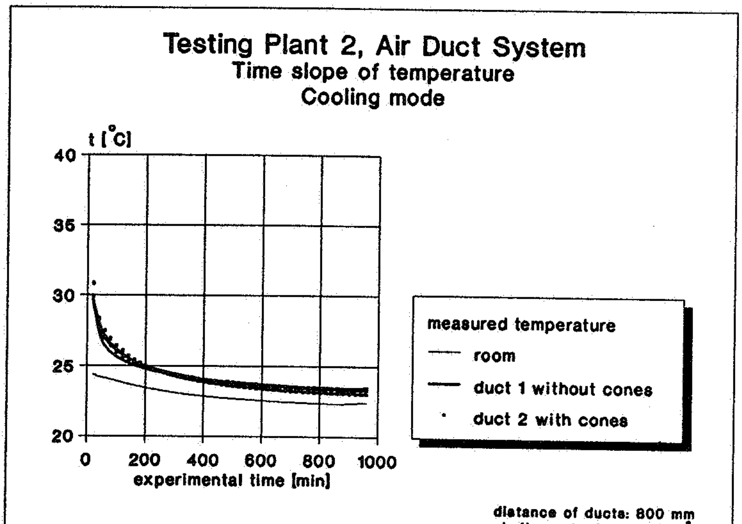


Fig. 11

The investigation showed that the pressure drop of the duct equipped with cones is much to high for practical applications. For this reasons this construction is not further considered.

## 4. Summary

The results of the measurements have shown that the floor with a cavity built with cones is unsuitable for a combined ventilation and heating system. The heat loss at the floor surface is too high so that the supply air temperature at the air inlet is not sufficient for room heating. There is, therefore, also a risk of air draughts in the occupied zone.

The advantages of the air duct system with a cavity area at the end are the compensation of the low temperature at the cold external walls or windows by the veil of warm air. Because of the high temperatures at the inlet grille the system guarantees a good performance combined with a controllability.

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