Convection Flows from an Overhead Projector and a Data Projector

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SUMMARY

When room ventilation is based on the stratification or zoning strategy, the ventilation air flow rate is determined on the grounds of the convection flows of the heat sources. Thus, in the design phase of office ventilation, the convection flows of common office devices should be known. Especially, to correctly dimension the ventilation air flow, it is important to know the volume flow rates. However, even if the convective power of an office appliance is known, the cooling fan makes the prediction of the characteristics of the convection flow very difficult. Therefore, direct measurement data is needed. In this study, the convection flows of an overhead projector and a data projector were examined. Cross sections of the velocity and temperature fields were measured and analyzed to obtain the volume and momentum flow rates.

INTRODUCTION

In the stratification and zoning strategies of ventilation, the ventilation air flow has to overrun the combined convective flows everywhere in the occupied zone to prevent them from returning. Then, when dimensioning the ventilation in a space with numerous (human or non-human) heat sources, the designer must know the characteristics of the flow generated by each source. Mere knowledge of, *e.g.*, the power consumption of a source is not sufficient. The relative proportions of convective and radiative heat transfers have to be known. (Heat conduction is not significant.) In addition, a warm electric device usually has a fan, generating forced convection with properties completely different from those of natural convection. Therefore, experimental information of the convection flows of typical heat sources in ventilated rooms is needed. This paper reports the results obtained from two common office appliances; an overhead projector and its modern counterpart, a data projector. Both devices have a fan, making the prediction of their convection flows very challenging. Velocity and temperature distributions on different cross sections of the flows were measured, and the results were used to compute the air flow rates, momentum flow rates, and convective powers. The results were compared to a basic heat source, which was a heated, black metal cylinder with 40 % convection.

METHODS

The electric power consumed by an electric device dissipates by two basic mechanisms; convection and radiation. Since the devices usually have a fan, convection is further subdivided in two separate portions. In forced convection heat is carried away by a jet generated by the fan, and in natural convection warm surfaces generate a rising plume, also carrying heat with it. This division is illustrated in Fig. 1. The blow from the fan is most often directed horizontally, whereupon the forced and natural convection flows remain separate.

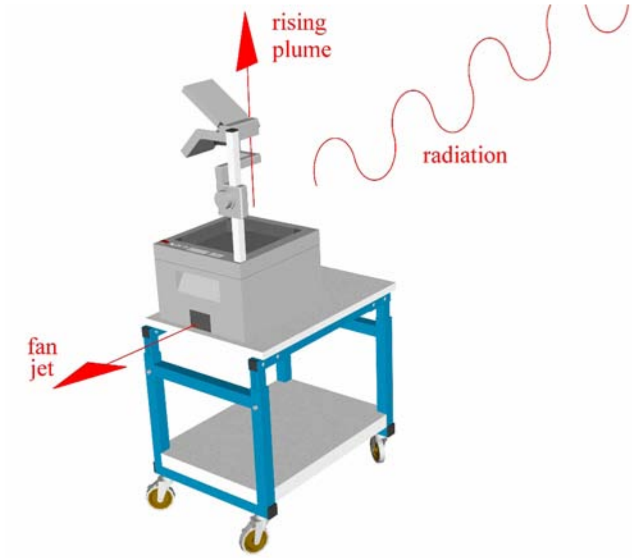

Figure 1. Mechanisms of heat loss from an office device.

The device under examination was placed in a test room shown in Fig. 2. The supply air was delivered through six perforated pipes on the floor to minimize the disturbances inflicted to the plumes, and the exhaust was dispersed at the ceiling. Moreover, the ventilation air current was set as low as possible (~245 l/s) without causing return flow of the plume air. The flow field was measured by two directional sensitive ultrasonic anemometers, attached in a traversing robot, also displayed in Fig. 2. The robot was able to move in all three directions and was programmable, so that measurements could be performed without human presence. In each point of measurement, the anemometers stayed one minute gathering data.

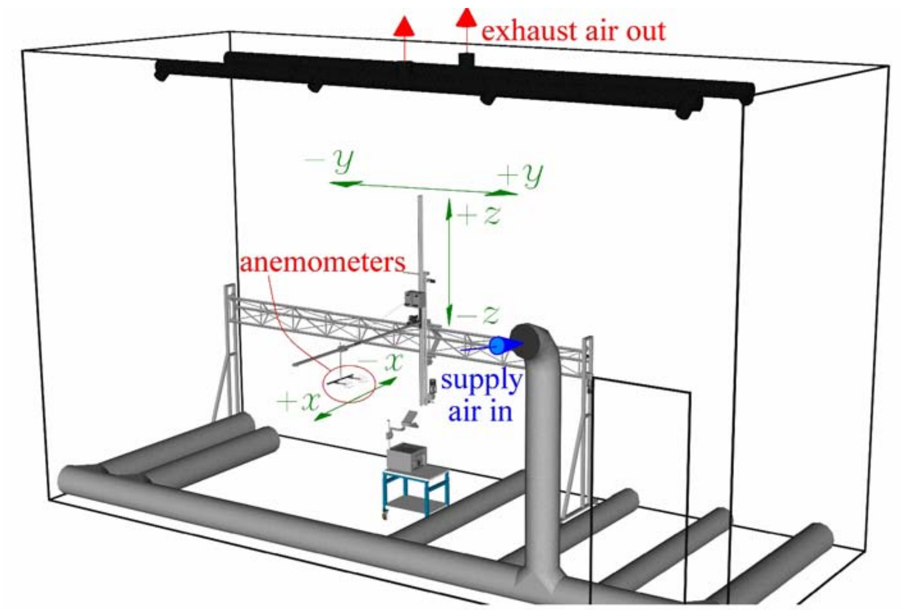

Figure 2. Test room and the measurement robot, carrying the anemometers.

The anemometers were used to measure horizontal and vertical cross sections of the flow fields of the plumes (jets). Figure 3 illustrates one such cross section (the near field in front of the exit opening of the data projector). These cross sections were used to integrate the volume, thermal energy (*i.e.* heat), and momentum flow rates through them. This was possible, since there was a thermistor attached to both anemometers, giving the temperature distributions also.

Figure 3. Measurement grid for the near field in front of the exit opening of the data projector fan. Points of measurement are marked by blue balls. An illustration of the outcome of this measurement is displayed in Fig. 6 (bottom row).

When integrating the air volume flow rate through a plane of measurement, the background velocity has to be subtracted from the velocity measurements. Let $\boldsymbol{v}$ be the velocity vector, and $\boldsymbol{w}$ its background component. Further, let subindex $\perp$ denote the direction perpendicular to the plane of measurement. (This plane can be either horizontal or vertical.) Then the volume flow rate through the plane is

$$q_{\mathrm{pl}} = \int_{\substack{\text{plane of} \\ \text{measurement}}} (v_{\perp} - w_{\perp}) \mathrm{d}A \ . \qquad (1)$$

The background component $w_{\perp}$ was deduced from the fringe area of the measurement plane.

The heat power P_{c}, conveyed away from the device by the convection flow (plume or fan jet), can be calculated, if at every point we know the flow velocity and temperature rise ΔT of the air. Here ΔT is the increase in temperature, experienced by the air after entering the plume (*i.e.*, the part of the temperature originating from the device). The heat power conveyed through an area element dA is then $\mathrm{d}P_{\mathrm{c}} = c_{\mathrm{p}}\, \rho\, \Delta T\, v_{\perp} \mathrm{d}A$, and the entire power conveyed through the plane of measurement is

$$P_{\mathrm{c}} = \int\limits_{\substack{\text{interior of}\\\text{conv. flow}}} c_{\mathrm{p}}\, \rho\, \Delta T\, v_{\perp} \mathrm{d}A \quad . \qquad (2)$$

Note that the background velocity $\boldsymbol{w}$ is not subtracted now, since it is the total air flow flushing the device surfaces that is conveying the heat. Also note that the area of integration covers merely the interior of the flow region. This may be difficult to delimit in practice, and if the integration is extended over the entire plane of measurement, it may be necessary to replace $v_{\perp}$ by $v_{\perp} - w_{\perp}$ to reduce the errors accruing from integration outside of the convection flow. Therefore, and for the results to be of general use, the background component of the flow velocity should be insignificant. As for ΔT, its calculation is complicated by temperature stratification. New air is entrained into the flow from all heights, and ΔT should be the present temperature minus the average original temperature of all the air in the flow, which we denote by T_0. When the flow is an upward travelling, natural plume, the effect of stratification can be approximated in a simple way. If at height z above the virtual origin of the plume the ambient temperature is $T_{\mathrm{amb}}(z)$ and plume volume flow rate is $q_{\mathrm{pl}}(z)$, then at an arbitrary height h (above the virtual origin)

$$T_0(h) = \frac{1}{q_{\mathrm{pl}}(h)} \int_0^h T_{\mathrm{amb}} \frac{\mathrm{d}}{\mathrm{d}\,z} q_{\mathrm{pl}} \mathrm{d}\,z$$

and $\Delta T = T - T_0(h)$. If we insert the point source formula $q_{pl}(z) = AP_c^{1/3} z^{5/3}$ and assume linear temperature gradient, *i.e.* $T_{\mathrm{amb}}(z) = T_{\mathrm{amb}}(0) + \beta\, z$, we obtain
$T_0(h) = T_{\mathrm{amb}}(0) + \frac{5}{8}\beta\, h = T_{\mathrm{amb}}(h) - \frac{3}{8}\beta\, h$ and

$$\Delta T = T - T_{\mathrm{amb}}(h) + \frac{3}{8}\beta\, h \, . \qquad (3)$$

Stratification correction (3) was not used when integrating over a vertical cross section of an exhaust jet (*i.e.*, β was set to 0). It was applied after the jet had veered almost vertical and a horizontal cross section was measured. This procedure somewhat underestimates the convection power far from the device, since it underestimates the proportion of cool air entrained into the flow from low heights. However, P_{c} was only used to judge whether the convection flow has been captured well enough by the measurement. If far away from the source the measured values of P_{c} have diminished too much, then the flow has probably become too wide or ragged to be analyzed, and the other results are not reliable either.

In addition to volume and energy flow rates, momentum flow rate through each plane of measurement was also calculated. Momentum flow rate $\boldsymbol{M}$ tells the total momentum vector $\boldsymbol{p}$ conveyed per unit time across the piece of plane measured, *i.e.*

$$\boldsymbol{M} = \dot{\boldsymbol{p}} = \mathrm{d}\,\boldsymbol{p} / \mathrm{d}\,t = \int\limits_{\substack{\text{plane of}\\\text{measurement}}} \boldsymbol{v}\, \mathrm{d}\dot{m} = \int\limits_{\substack{\text{plane of}\\\text{measurement}}} \rho\, \boldsymbol{v}\, v_{\perp} \mathrm{d}A \quad .$$

This quantity is useful in CFD modelling. The near field momentum flux is a necessary boundary value, and momentum flow rate elsewhere in the flow is useful in validating CFD solutions. The most interesting component of $\boldsymbol{M}$ is usually that perpendicular to the plane. It gives the perpendicular force that would be impinged by the flow on the plane in case the plane absorbed the flow. It is given by

$$M_z = \int_{\substack{\text{plane of}\\ \text{measurement}}} \rho v_\perp^2 \,\mathrm{d}A\,.$$

The measurement robot only provided one minute time averages $\langle v_\perp \rangle$ and the corresponding standard deviations $s_\perp$. These were then used to calculate the squared average of $v_\perp$ as $\langle v_\perp^2 \rangle = \langle v_\perp \rangle^2 + s_\perp^2$. The deviation originates from the turbulence and fluctuations of the flow. The background flow, which was assumed constant and laminar, was subtracted, so that the final formula for the flow of $p_\perp$ was

$$\langle M_\perp \rangle = \int_{\substack{\text{plane of}\\ \text{measurement}}} \rho \left[\left(\langle v_\perp \rangle - w_\perp \right)^2 + s_\perp^2 \right] \mathrm{d}A\ , \qquad (4)$$

where angle brackets stand for (one minute) time averaging. When computing the above integral, however, the term $s_\perp^2$ was added only in points where $s_\perp < \left(\langle v_\perp \rangle - w_\perp \right)$ to avoid inclusion of mere turbulence in quiescent air.

Finally, the radiative powers of the overhead projector and the data projector were estimated. This was done by measuring the surface temperatures in several points, taking the means of their fourth powers, and applying the formula

$$P = \varepsilon \sigma\, A \left(\overline{T_\mathrm{d}^4} - \overline{T_\mathrm{r}^4} \right), \qquad (5)$$

where subindices d and r stand for device surfaces and room surfaces, respectively, overline means spatial average, and emissivity of the device surfaces $\varepsilon = 0.95$. The radiative power of the black cylinder, used as a reference heat load, was calculated more accurately by dividing surfaces into smaller sections and calculating the view factors.

Above formulas (1) – (5) were used to analyze the raw data measured along the flow cross sections. In eqs. (2) and (4), air density ρ is left inside the integrations, since it varies locally along with temperature.

RESULTS

The basic characteristics of the convection flows can be seen from the smoke visualizations in Figs. 4 and 5. It is evident that the convection flow from the data projector is purely forced, but the overhead projector also has a weak plume of natural convection. The jets of forced convection turn more and more vertical on their way, beginning to behave more and more like natural plumes. The jet of the data projector is hotter and slower, whereby it turns faster. The figures also show, how its jet is pasted on the table top (Coanda effect). Therefore, when performing the robot measurements, both the devices were set at the edge of the table. Since the forced convection flow of the overhead projector is slow, cool and wide when it finally rises up, measuring its volume, energy, and momentum flow rates is very inaccurate at high altitudes. The near fields of $v_\perp$ and T of both devices are seen in Fig. 6. They are vertical cross sections in front of the exhaust opening, either 0.2 m or 1.3 m apart from it. They show the fields when looking against the flow, towards the device.

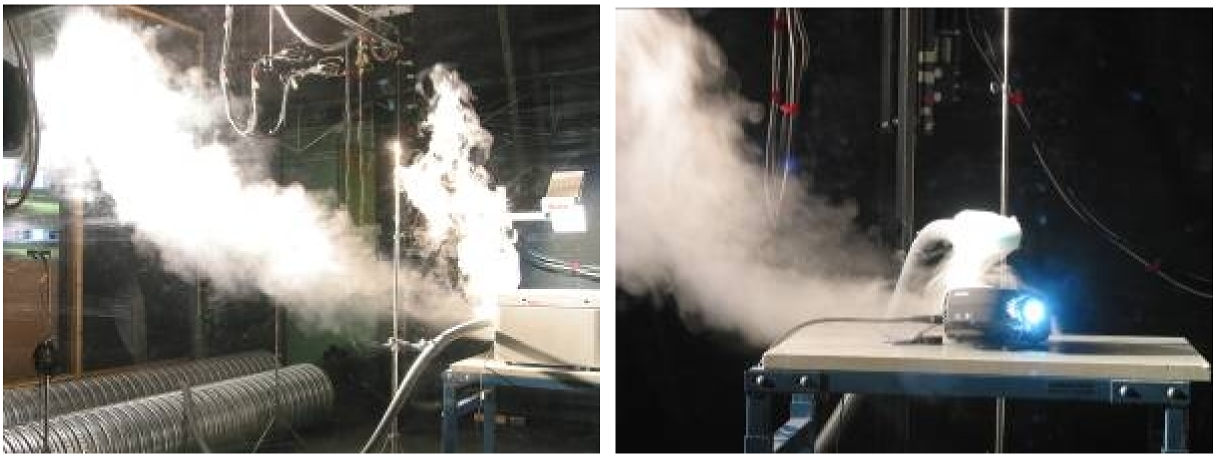

Figure 4. Smoke tests reveal how the convection of the overhead projector (on the left) is divided between forced and natural convection, whereas all convection flow from the data projector (on the right) is forced convection.

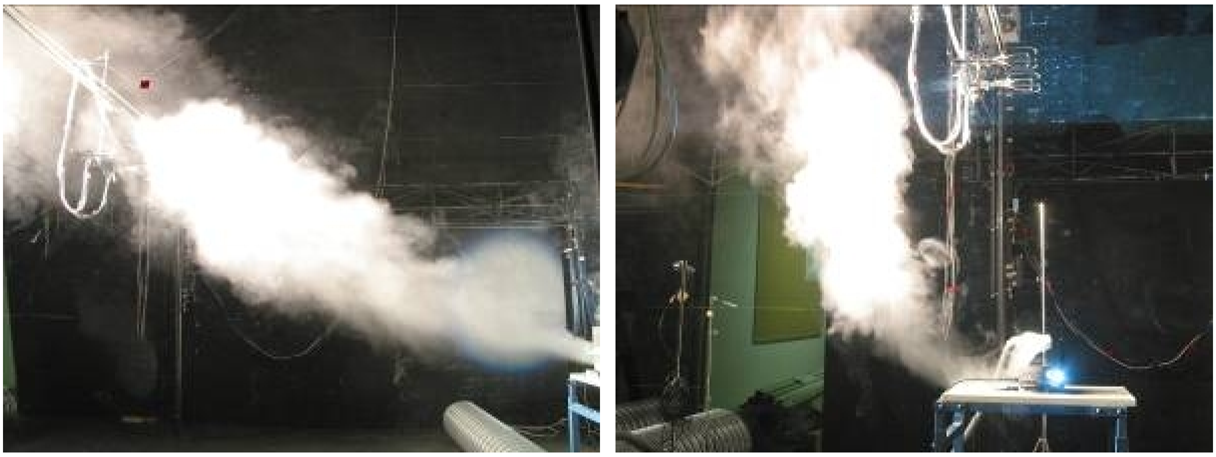

Figure 5. The fan jet from the overhead projector (on the left) is faster and cooler than that from the data projector (on the right), therefore rising up farther away from the device.

The results of the analyses are gathered in Figs 7 – 9. The office devices examined are always compared with a reference plume source, which was a black metal cylinder, heated through its full length. Its properties were known from our earlier measurements, and a measurement with the heating power as close as possible to that of the office appliance was chosen. The height of the cylinder was 1.2 m, and it was standing on a 50 cm high piece of insulator. The height of the table, carrying the projectors, was 0.7 m. The first of the figures, Fig. 7, displays the convection powers (*i.e.*, the heat flow rates) flowing through the cross sections. They should be considered as a test of reliability. For the measurements to be reliable, the sum of the (forced and natural) convection and radiation powers (kinetic energy flow rate is not significant) should be near the heating power used, written in black. Thereby, a reduction in the measured convection power during the rise of the flow tells that the precision is deteriorating.

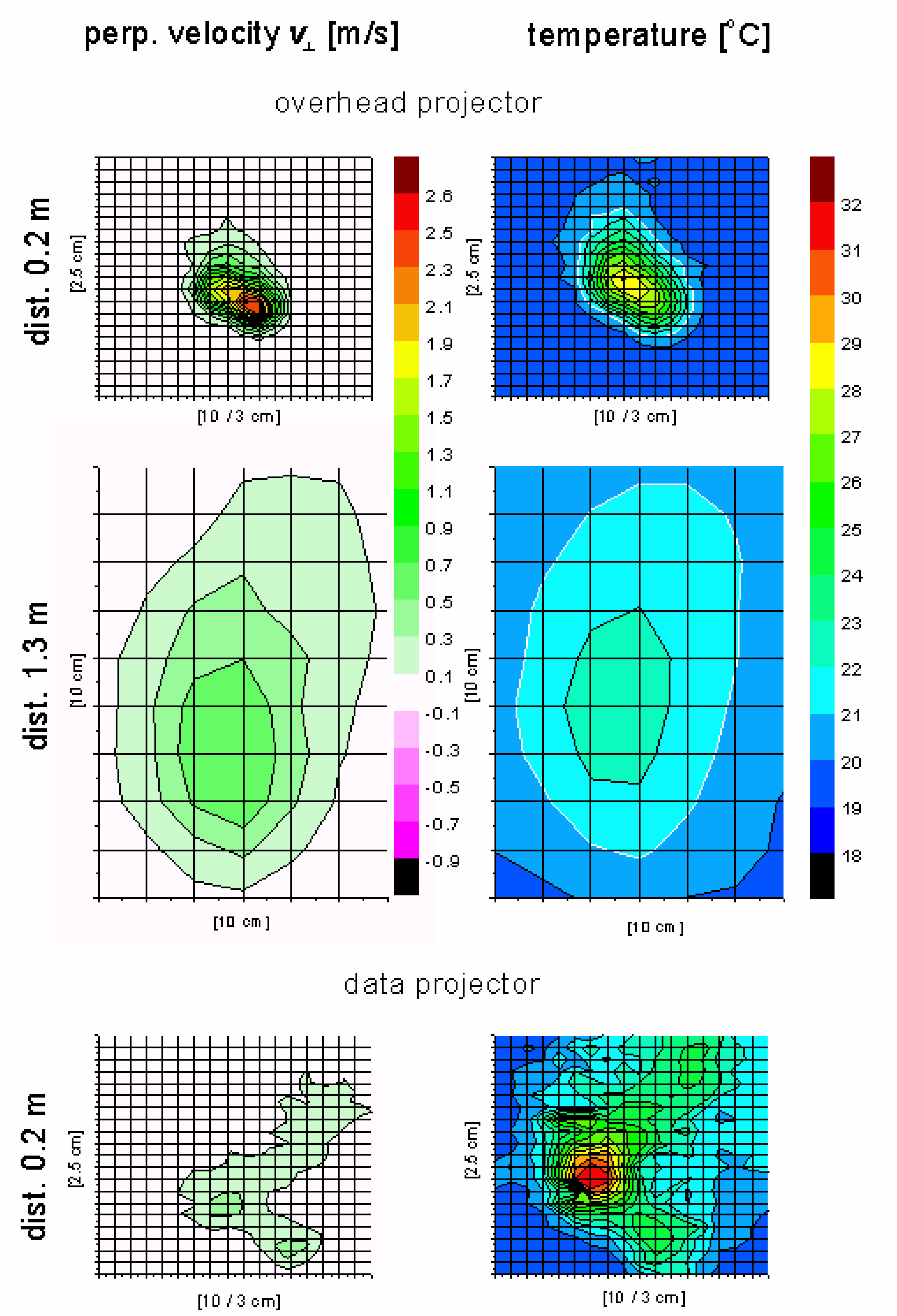

Figure 6. Vertical cross sections of the jets of forced convection. The velocity maps display the component $v_\perp$, parallel with the original direction of the blow and perpendicular to the plane of measurement. All the maps are drawn in the same scale, the grid of measurement points is shown, and the grid size is given in centimeters.

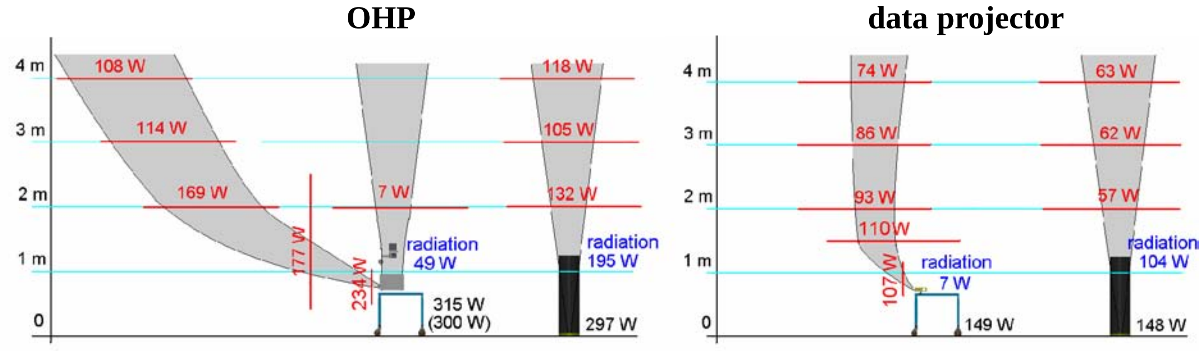

Figure 7. Measured convective and radiative powers of the office devices as compared with those of a black metal cylinder. The power consumption of the overhead projector was normally 315 W, but when measuring the near field 0.2 m apart, a reserve lamp had to be changed, after which the power consumed was 300 W. OHP = overhead projector.

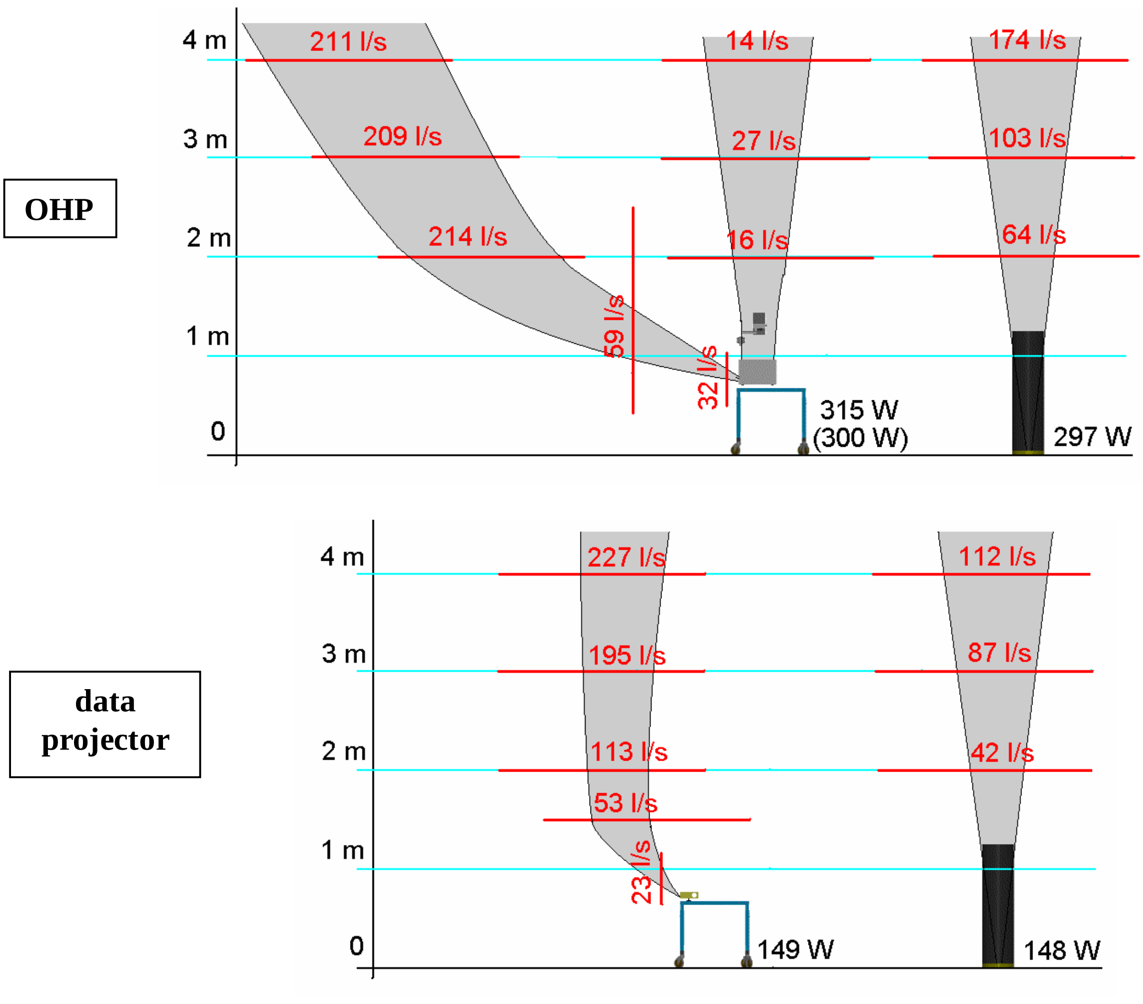

Figure 8. Measured convective volume flow rates of the projectors as compared with those of the black cylinder.

OHP

data projector

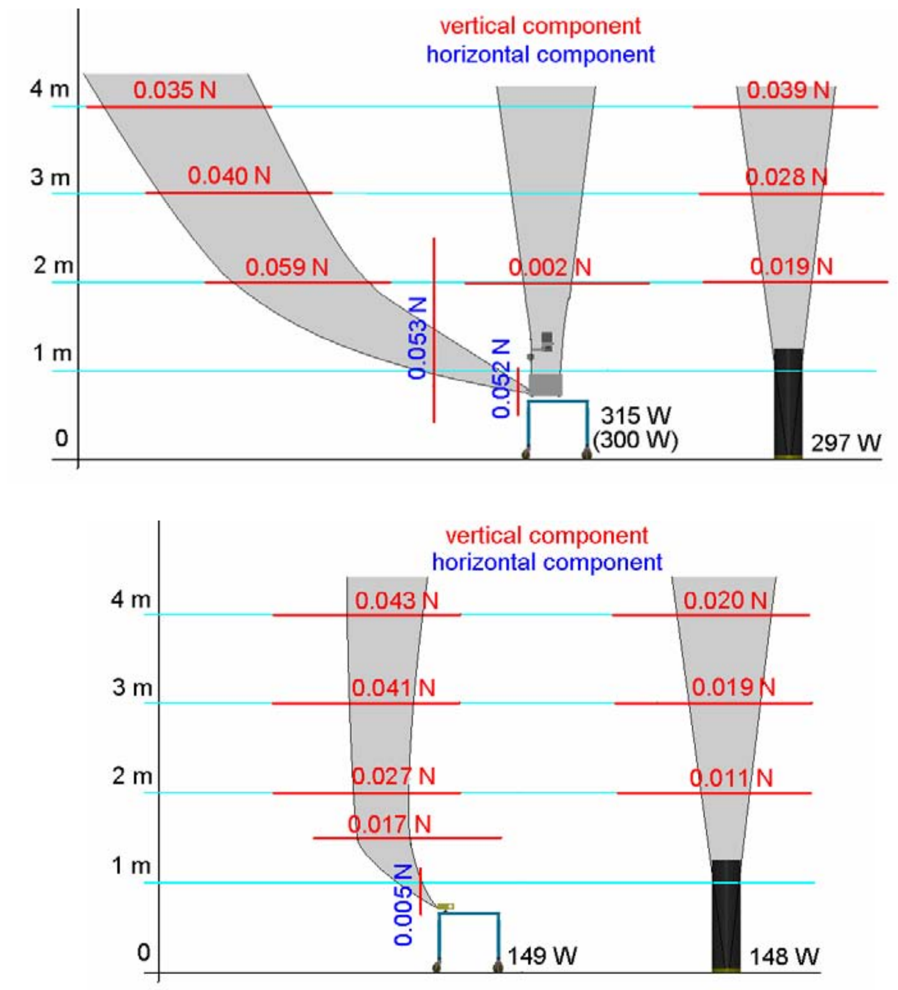

Figure 9. Measured convective momentum flow rates of the projectors v. the black cylinder.

According to the results, the most important of which are shown above, we can say the following:

- In the presence of device fan, a great majority of thermal losses takes place through convection.
- The fan and the increased buoyancy due to increased convection power multiply the convection flow (volume flow rate) by a factor between 2 and 4 as compared with a black cylinder with the same heating power. Therefore, it is a wise idea to attach a data projector to the ceiling.
- The vertical momentum flow rate is increased by a factor between 2 and 3. This can partly be explained by the increased buoyancy, but the most important reason to this are probably the pressure forces at the top surface of the reference cylinder, hindering the growth of the momentum flow rate. These kinds of forces do not confine the exhaust jets of the projectors. A totally new feature is the horizontal momentum flow rate due to the horizontal fan jet (blue numbers in Fig. 9).
- The temperatures of the plume air are not much different between the projectors and the cylinder with the same heating power. At the height of 2 m the maximal temperature in the plume of the overhead projector is 23.9 ºC, when for the cylinder heated by the power of 297 W it is 23.5 ºC. For the data projector and the cylinder heated by 148 W the corresponding values are 21.3 ºC and 22.2 ºC.

DISCUSSION

Common electric office appliances generally have a cooling fan producing a powerful flow of forced convection, which easily absorbs the natural convection flow along. This flow is a horizontal buoyant jet, which first behaves like a jet, and gradually changes to a buoyancy-driven plume. Prediction of the volume and momentum flow rates of such a flow is problematic, and in any case the near field has to be measured to obtain the initial values. In this study the behaviour of the (forced) convection flow in the jet, transition, and plume regions has been measured for two common office devices. The results were compared to results measured for a black cylinder, generating only a natural, buoyant plume.

The farther away the flow has traveled from its origin, the more challenging is its measurement. The reason to this is that the flow becomes cooler, slower, wider, and more disassembled on its way, making it more difficult to distinguish from the background. To provide a criterion to decide, how reliable the results are, the (thermal) energy flow rate (or the convection power) was always calculated together with the volume and momentum flow rates, see Fig. 7. Comparison of the electric power with the sum of the measured radiation and near field convection powers tells us, how the measurement of the initial values has succeeded. After this, the conservation of the measured convection power during the course of the flow gives a conception of how fast the measuring accuracy is deteriorating. Note, however, that when calculating the convection power, the convective temperature rise ΔT is also needed. It is obtained in poorer accuracy than velocity, and therefore the accuracy of momentum flow rate and, especially, volume flow rate is higher than that of the convection power.

As Fig. 8 shows, at the height of 2 m the volume flow rate of the overhead projector is 230 l/s. This is more than ten times the air flow above the head of a person (20 l/s, see [1]). The data projector, despite its relatively small power, also produces a convection flow rate corresponding to several persons. In addition, each person usually has a computer, producing a larger convection flow than the person himself [1]. Therefore, when dimensioning the ventilation air flow rate in stratification or zoning air distribution strategy, the office appliances should definitely be taken into account. Nonetheless, the ventilation air flow rate required can be reduced by attaching the data projector to the ceiling, above the occupied zone.

REFERENCES

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