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# Calculations of the temperature and flow field in a room ventilated by a radial air distributor

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**Keywords: temperature, radial air distributor, ventilation**


## Calculs de la température et du champ d'écoulement dans un local ventilé par un distributeur d'air radial

*Pour obtenir un confort maximal dans les locaux ventilés ou climatisés il est nécessaire d'optimiser les conditions de température, d'humidité et de vitesse. Cette étude se rapporte à un local de 4,7 m de long sur 3 m de large et de 2 m de haut, ventilé par un distributeur à plaques radiales au centre du plafond, l'air étant éliminé par des sorties au niveau du sol aux deux extrémités du local. On procède à une analyse numérique détaillée en s'appuyant sur les équations de quantité de mouvement et d'énergie, comprenant l'énergie kinetic et la dissipation de la turbulence. Les équations ont été codées pour un système de grille à 50×50 points et résolues à l'aide d'un ordinateur Cyber 175. L'écoulement dans le distributeur est indiqué à la Fig. 3 et le champ de vitesse totale est présenté à la Fig. 7. La comparaison des prévisions et des résultats des expériences de Schäfers est présentée à la Fig. 8: la concordance est acceptable.*

**In ventilated or air-conditioned rooms optimal conditions of temperature, humidity and air velocity are required. In the present study the behaviour of a jet emerging from a radial plate distributor and the resulting air flow in the room were investigated. To predict the behaviour of the air flow a numerical scheme was used to solve the conservation equations for mass, momentum and energy with the $k/\varepsilon$-turbulence model. The numerical results are compared with available experimental data.**


## Nomenclature

| | |
|---|---|
| $B$ | width of the room |
| $d$ | diameter |
| $H$ | height of the room |
| $h$ | slot-height, step-height |
| $k$ | turbulent energy |
| $L$ | length of the room |
| $p$ | pressure |
| $Pr$ | Prandtl-number |
| $r$ | coordinate in radial direction |
| $Re$ | Reynolds-number |
| $S_u, S_v$ | Source terms of momentum equation |
| $u$ | velocity in $z$-direction |
| $v$ | velocity in $r$-direction |
| $\bar{v}_R$ | mean air velocity in the room |
| $\bar{v}_z$ | mean air velocity in level $z$ |
| $z$ | coordinate in axial direction |
| $z_R$ | length of the recirculation region |
| $z_2$ | half width |
| $\varepsilon$ | dissipation of turbulent energy |
| $\eta$ | dynamic viscosity |
| $\theta$ | temperature |
| $\Delta\theta$ | difference of temperature |
| $\bar{\theta}_R$ | mean temperature of the room |
| $\rho$ | density |
| $\psi$ | stream function |

### Indices

| | |
|---|---|
| a | outlet |
| eff | effective |
| o | throat |

In ventilated or air-conditioned rooms optimal conditions of temperature, humidity and air velocity are required. Therefore for the design of ventilation systems the basic fluid dynamic relations of interaction between injected air, room air, heat sources, and heat sinks must be known. When new outlets are developed or when known air distribution systems are used under unusual conditions, tests in small-scale experimental rooms must be performed.

The disadvantages of such experiments are:

1. It is often impossible to fulfill all similarity conditions. Therefore, experimental results may only approximate full-scale room conditions.
2. Such tests mean expenditure of a considerable amount of manpower as well as material.
3. Because many parameters are of importance the test facility must be modified very often.

Therefore, for several years numerical schemes

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have been under development to predict the temperature and velocity distributions. For simple room geometries and air inlet systems, simulated by two-dimensional configurations, successful work has been published e.g. by Nielsen[1] or Hanel and Scholz[2]. For more complex geometries for which these approximations are no longer valid, numerical predictions are often not possible, because computer time and available computer storage in existing computers are usually exceeded. But for these complex geometries the two-dimensional numerical methods may still be employed, in many cases, to estimate, for instance, the effects of varying parameters.

In the present study the behaviour of a jet emerging from a radial plate distributor and the resulting air flow in the room are investigated. The temperature of the jet may be lower or higher than the average room temperature.

The numerical results of the flow and temperature field in the jet region and within the room are compared with the experimental results by Schäfers[3] in a testroom with a radial plate distributor. This work continued a research programme of the Institute for Heat Transfer in Aachen on ventilation systems[4,5]. The testroom, Fig. 1, has a floor area of 3×4.7 m and a height of 2 m. The radial jet is generated by a plate diffusor mounted in the centre of the ceiling. The return air is sucked out through outlets placed at floor level on both sides of the room. Because of these boundary conditions the flow is approximately axi-symmetric and therefore the measurements and the calculations may be restricted to one half of a lateral plane.

## Numerical analysis

### Conservation equations

To predict the velocity and temperature field the two-dimensional, steady state conservation equations for mass, momentum and energy must be solved. These equations are used along with two additional equations describing the air turbulence. The $k,\varepsilon$-turbulence model proposed by Jones and Launder[6] is employed. For non-isothermal calculations buoyancy forces are considered in the momentum equation. As shown by Schmitz[7] the buoyancy effects within the turbulence equations may be neglected.

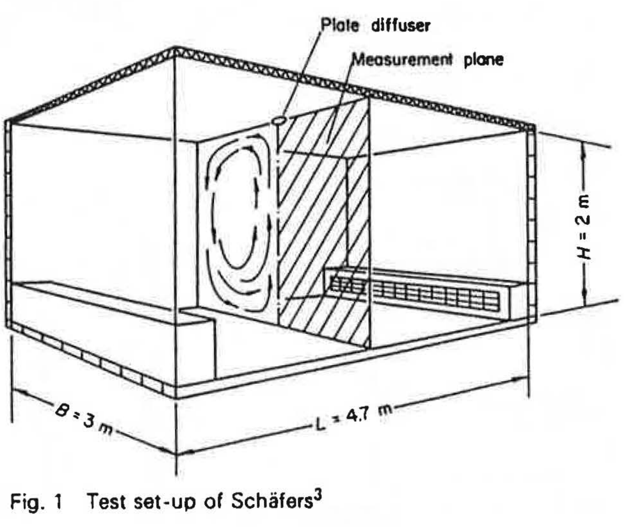


Fig. 1 Test set-up of Schäfers[3]

*Fig. 1 Installation d'essai de Schäfers[3]*

Continuity:

$$\frac{\partial}{\partial z}(\rho r u)+\frac{\partial}{\partial r}(\rho r v)=0 \tag{1}$$

Momentum, axial direction:

$$\frac{1}{r}\left[\frac{\partial}{\partial z}(\rho r u^2)+\frac{\partial}{\partial r}(\rho r u v)-\frac{\partial}{\partial z}\left(r\eta_{eff}\frac{\partial u}{\partial z}\right)-\frac{\partial}{\partial r}\left(r\eta_{eff}\frac{\partial u}{\partial r}\right)\right]=-\frac{\partial p}{\partial z}+S_u \tag{2}$$

Momentum, radial direction:

$$\frac{1}{r}\left[\frac{\partial}{\partial z}(\rho r u v)+\frac{\partial}{\partial r}(\rho r v^2)-\frac{\partial}{\partial z}\left(r\eta_{eff}\frac{\partial v}{\partial z}\right)-\frac{\partial}{\partial r}\left(r\cdot 2\eta_{eff}\frac{\partial v}{\partial r}\right)-\frac{\partial}{\partial z}\left(r\eta_{eff}\frac{\partial u}{\partial r}\right)\right]+2\eta_{eff}\frac{v}{r^2}=-\frac{\partial p}{\partial r}+S_v \tag{3}$$

Energy equation:

$$\frac{\partial}{\partial z}(\rho r u\theta)+\frac{\partial}{\partial r}(\rho r v\theta)|-\partial\partial z\left(r\frac{\eta_{eff}\partial\theta}{Pr_{eff}\partial z}\right)-\frac{\partial}{\partial r}\left(r\frac{\eta_{eff}\partial\theta}{Pr_{eff}\partial r}\right)=0 \tag{4}$$

Kinetic energy of turbulence:

$$\frac{1}{r}\left[\frac{\partial}{\partial z}(\rho r u k)+\frac{\partial}{\partial r}(\rho r v k)-\frac{\partial}{\partial z}\left(r\frac{\eta_{eff}\partial k}{\sigma_k\partial z}\right)-\frac{\partial}{\partial r}\left(r\frac{\eta_{eff}\partial k}{\sigma_k\partial r}\right)\right]=G-\rho\varepsilon \tag{5}$$

Dissipation rate:

$$\frac{1}{r}\left[\frac{\partial}{\partial z}(\rho r u\varepsilon)+\frac{\partial}{\partial r}(\rho r v\varepsilon)-\frac{\partial}{\partial z}\left(r\frac{\eta_{eff}\partial\varepsilon}{\sigma_\varepsilon\partial z}\right)-\frac{\partial}{\partial r}\left(r\frac{\eta_{eff}\partial\varepsilon}{\sigma_\varepsilon\partial r}\right)\right]=C_1\frac{\varepsilon}{k}G-C_2\rho\frac{\varepsilon^2}{k} \tag{6}$$

with $G=\eta_t\left[2\left[\left(\frac{\partial u}{\partial z}\right)^2+\left(\frac{\partial v}{\partial r}\right)^2+\left(\frac{v}{r}\right)^2\right]+\left(\frac{\partial u}{\partial r}+\frac{\partial V}{\partial z}\right)^2\right]$;

and parameters $C_1=1.43$; $C_2=1.92$; $C_\eta=0.09$; $\sigma_k=1.0$; $\sigma_\varepsilon=1.3$.

### Computer program

A computer code proposed by Pun and Spalding[8] was extended to solve the above differential equations. A grid system of ≈50 by 50 points is used, which is detailed enough to cover both the jet region and the flow field in the room. The time needed for one typical run was about 3000 to 4000 s with a maximum

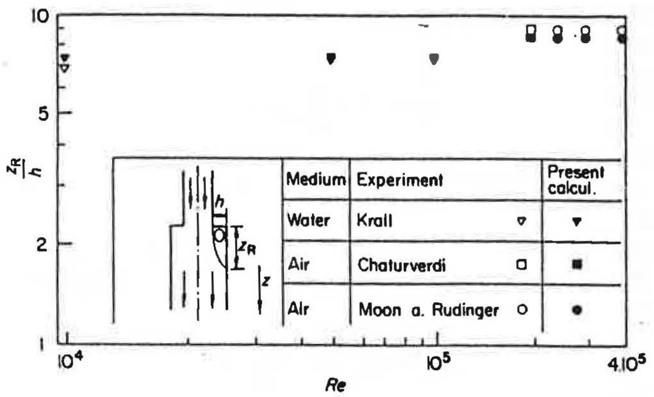


Fig. 2 Measured and calculated reattachment points of a sudden pipe expansion

*Fig. 2 Points de rattachement mesurés et calculés pour une dilatation subite du tuyau*

memory requirement of about 500 k Byte on a Cyber 175.

### Program testing

Because the geometry of a sudden pipe expansion is the basis for the following calculations of a radial outlet, this geometry was used to check the computer program. For the reattachement of a flow in a pipe with a sudden expansion much experimental work has been published. The predicted reattachement points and the corresponding measured values for air and water[9-11] are plotted in Fig. 2 for different Reynolds numbers. As can be seen the experimental and calculated values coincide quite well in the laminar and in the turbulent regime.

## Numerical results

The numerical predictions are discussed separately for the jet region and for the flow within the room although both regions were calculated in one single run.

### Isothermal and non-isothermal radial jets

The air enters the inlet cross section of the nozzle (AMCA-Standard 210-67) with a given inlet velocity and is directed in the radial direction by the distributor plate. Depending on the distance between the plate and the ceiling the flow separates from the wall and reattaches again due to the Coanda-effect. Because the separation region has a considerable influence on the development of the jet this phenomenon was examined first. For four distances the calculated stream lines are shown in Fig. 3.

For a slot width of $h=10$ mm and $h=15$ mm no flow separation can be observed whereas for a slot width of $h=20$ mm and $h=30$ mm rather extended recirculating zones can be observed. The zero stream line separates the recirculation zone from the actual flow. This flow behaviour was confirmed qualitatively by experiments.

A quantiative comparison between calculation and experiment is possible in the jet region, where measured velocity profiles are available from Schäfers at different distances from the centreline for an inlet velocity of 15 m s$^{-1}$ and isothermal flow (see Fig. 4).

The calculated profile at $r=75$ mm at the outlet of the plate distributor is a result of a flow computation beginning at the nozzle inlet, where the velocity is known from experiments. This is in contrast to the aforementioned calculations by Nielsen, who used measured velocity values of the jet region as boundary values for the computations of the air flow in the rooms.

A similar good agreement can be observed for the other profiles further downstream. A possible reason for the discrepancies in the outer part of the profiles between calculations and experimental results may be due to limited experimental accuracy or deficiencies of the employed turbulence model. At present improvements of the turbulence model are investigated by varying the model constants and reorganizing the numerical scheme.

The region of similar profiles is reached at larger distances from the outlet as shown in Fig. 5, where a dimensionless representation is used. The velocity is divided by the maximum velocity of the profile and the distance from the ceiling by the so called half width, which is the distance from the ceiling where the actual velocity is half the maximum velocity. Again, the calculated values compare well with the measurements. The analytical relationship derived by Glauert[12] is added for comparison.

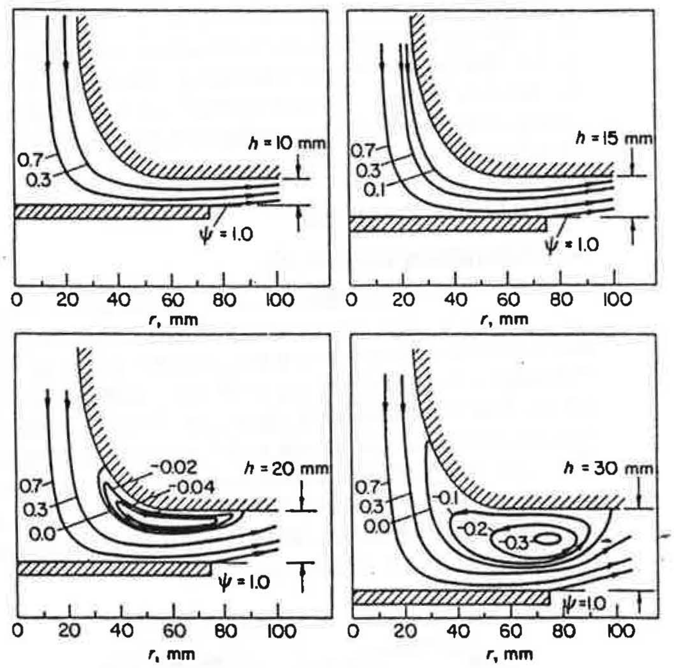


Fig. 3 Stream lines in the outlet

*Fig. 3 Lignes de courant dans l'orifice de sortie*

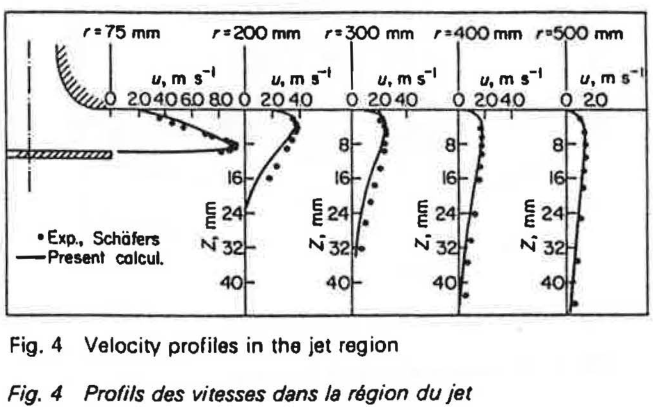


Fig. 4 Velocity profiles in the jet region

*Fig. 4 Profils des vitesses dans la région du jet*

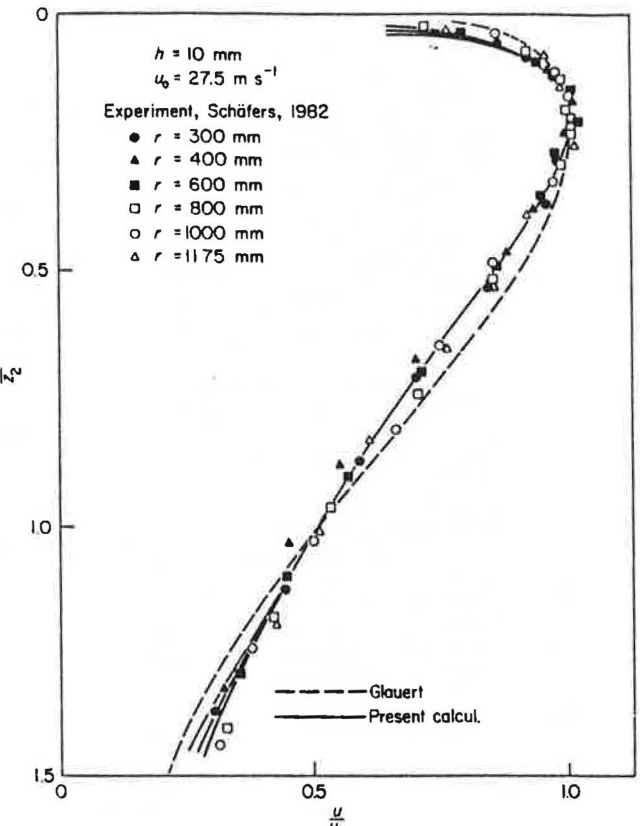


Fig. 5 Velocity profiles in the similarity region

*Fig. 5 Profils des vitesses dans la région de similitude*

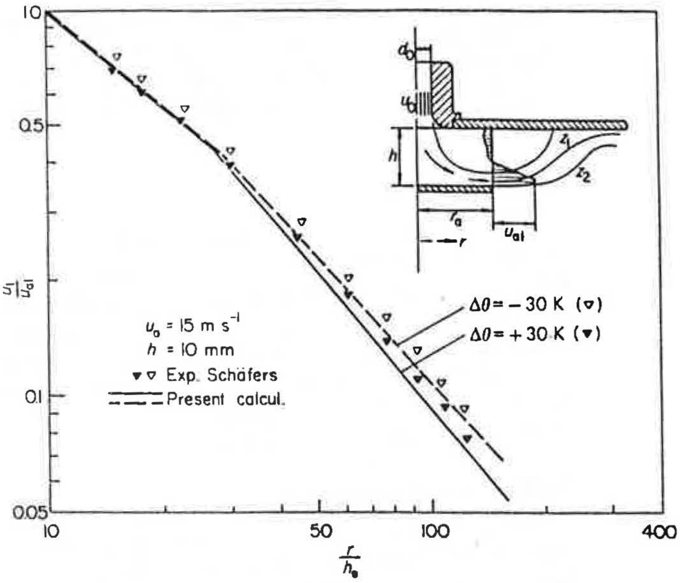


Fig. 6 Decay of maximum velocity

*Fig. 6 Affaiblissement de la vitesse maximale*

Another typical parameter describing the behaviour of the jet is the decay of the maximum velocity along the jet path. These values for a warm and a cold inlet jet are plotted in Fig. 6 in logarithmic scales. The actual path of the jet is divided by an effective distance of the plate distributor from the ceiling according to a definition proposed by Regenscheit[13]. The effective slot width is the distance which would result if the inlet flow emerges from the plate with a constant velocity equal to the measured maximum value.

The calculations show the known behaviour of radial jets in the transient as well as in the similarity region. No significant difference between a warm and cold jet is observed, which is in contrast to the flow pattern of flat jets in air-conditioning systems.

## Room air movement

The result of the computations of room air movement, generated by the radial jet at the ceiling will again be compared with the experiments by Schäfers[3].

Because the measuring technique is far more complex and its accuracy is limited in the ventilated room, it was not possible to determine the air velocity field especially for low velocities. Therefore, only a qualitative comparison can be demonstrated in Fig. 7, where a measured stream line pattern is compared with numerical results calculated for the same boundary conditions and plotted only for one half of the plane.

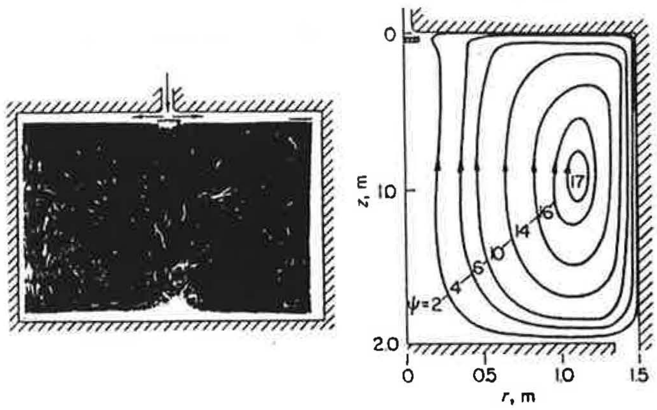


Fig. 7 Velocity field of the room

*Fig. 7 Champ de vitesse dans le local*

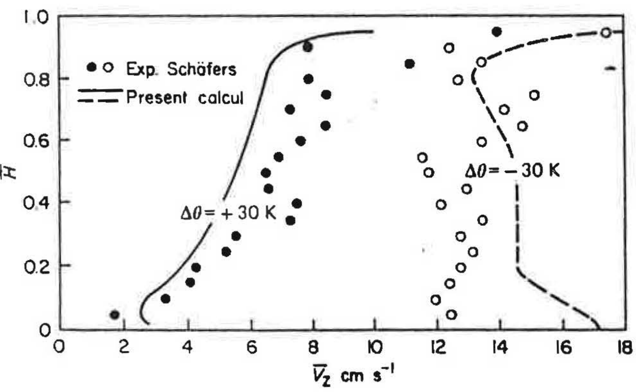


Fig. 8 Vertical velocity distribution in the room

*Fig. 8 Répartition verticale de la vitesse dans le local*

A quantitative comparison is represented in Fig. 8. The averaged velocities are plotted at different horizontal planes of the room *versus* the height of the room for a warm and cold jet. Each experimental point is an average of eight measured values of one plane, whereas the reference value, the room mean velocity, is an average of 150 measured values. In the numerical scheme all grid points of a horizontal line and of the complete grid, respectively, are used for averaging, except grid points within the jet region. This Figure demonstrates that the calculated results agree well with the experiments for non-isothermal conditions.

For a warm inlet air the velocity varies from $< 2\ cm\ s^{-1}$ near the floor to $8\ cm\ s^{-1}$ in the upper zone of the room. The cold jet with an identical inlet velocity induces good mixing with an average room velocity of $13\ cm\ s^{-1}$.

## Conclusions

These few examples show that the behaviour of a radial jet as well as the room air movement induced by a jet can be determined with acceptable accuracy by numerical solution of the conservation equations. The results indicate that at least a part of the time- and money-intensive experimental work in air-conditioning industry could be supplemented or even replaced by numerical computations.


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