Heat transfer influenced by turbulent airflow inside an axially rotating diffuser

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Abstract

The paper presents a study of heat transfer between the turbulent airflow and the inner wall surface of an axial diffuser rotating around its longitudinal axis. Heat transfer was assessed through the measurement of a time-dependent temperature field of the diffuser inner wall surface. Measurements of the instantaneous flow velocity components were performed by a laser-Doppler anemometry system, which delivered information on mean velocity components as well as on the turbulence intensity. A significant increase of all three mean velocity components was observed near the rotating diffuser wall in comparison with a non-rotating diffuser.

Temperature field measurements were carried out by means of infrared thermography. The experiment showed a significant dependence of the temperature field on the turbulent flowfield induced by diffuser rotation. A strong influence of the flow separation and reattachment on the temperature distribution was observed, while rotation was found to suppress the occurrence of flow separation from the diffuser wall.

Properties of the velocity field such as turbulence intensity were directly coupled with the temperature distribution in order to gain the information on how to enhance or reduce heat transfer by changing the integral parameters of the diffuser (e.g. rotation frequency or amount of flow).

1. INTRODUCTION

A velocity field in an axial diffuser that rotates around its longitudinal axis is quite different than the velocity field in the non-rotating (stationary) diffuser because of the swirl introduced to the axial flow. The swirl itself originates in the diffuser wall rotating around the diffuser longitudinal axis – this gives the airflow in the near-wall region a tangential velocity component. The well-known property of the swirl flow is the centrifugal force, which is a consequence of the radial pressure gradient $\partial p/\partial r$, expressed in the simplest form as [1]:

$$\frac{\partial p}{\partial r} = \rho \cdot \frac{u_t^2}{r} \tag{1}$$

Centrifugal force helps the flow to spread radially outwards towards the rotating diffuser wall. The unique flow kinematics in the rotating diffuser mentioned above leads to the assumption that such a distinctive flow field in the near-wall region may significantly affect heat transfer in the diffuser. Namely, the temperature field on the rotating inner wall surface may be noticeably different from that of the stationary diffuser. Our goal is therefore to establish the properties of heat transfer in the rotating diffuser through the observation of temperature field of the inner diffuser wall surface as a function of the velocity field of the turbulent flow inside the diffuser. Results of such an experiment could enhance the existing knowledge of the heat transfer and heat exchangers in general. The performed thermodynamic analysis should give more indepth view of the entire flow field in the rotating diffuser.

2. EXPERIMENTAL SET-UP

2.1. Velocity field measurements

Analysis of the velocity field in the rotating diffuser was carried out by the experimental station shown in Fig. 1.

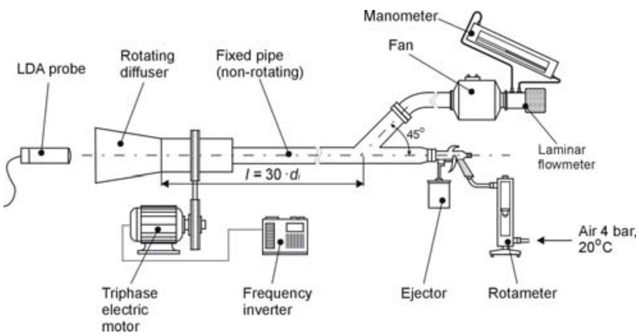

Figure 1: *Layout of the experiment for velocity field measurements.*

Length L of the conical part of the diffuser was 165 mm with the inner divergence angle θ = 18°. Inlet (d_i) and outlet (d_o) diffuser diameters were 60 mm and 112.27 mm, respectively. Measurements of the velocity components were performed by a TSI two-component back-scatter LDA system with a Spectra Physics Model 2016 argon-ion laser. Velocity components were measured within the 15 mm wide band in the x-z plane in six equidistant transverse sections (Outlet, A, B, C, D and Inlet, Fig. 2). There were 15 equidistant measuring points at each transverse section, which were 1 mm apart (in the radial direction of the diffuser) from each other.

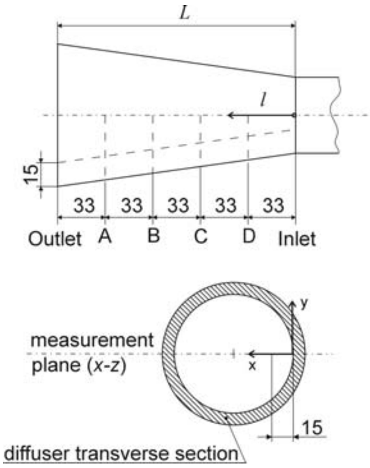

Figure 2: *Positions of the measurement points inside the diffuser cone (all dimensions in milimeters).*

There were 9 different operating conditions of the diffuser (combination of three rotating frequencies f and three different airflow Reynolds numbers Re, measured at the diffuser inlet) at which the velocity field of the diffuser flow was measured and analysed (Table 1).

Table 1: *Operating conditions of the diffuser during velocity field measurements (airflow temperature 20°C).*

f **[Hz]**	**Re [-]**
0	$5.84 \cdot 10^3$
30	$1.17 \cdot 10^4$
52.8	$2.01 \cdot 10^4$

2.2. Temperature field measurements

Experimental arrangement shown in Fig. 3 was used to assess heat transfer between the airflow and inner wall surface of the rotating diffuser.

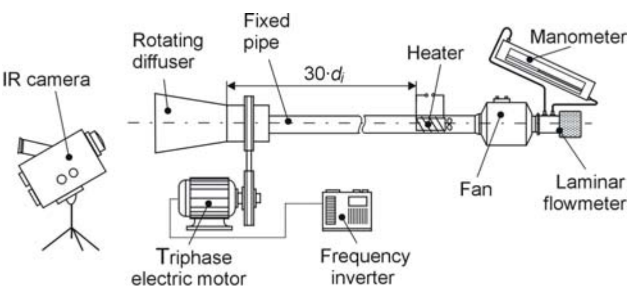

Figure 3: *Layout of the experiment for temperature field measurements.*

Measurements of the inner diffuser wall temperature were performed by an AGEMA 570 infrared (IR) sensitive camera, which operated in a range of wavelengths between 7.5 μm and 13 μm, had a stated thermal sensitivity < 0.15 K and a spatial resolution 1.3 mrad. The diffuser inner wall was heated by the hot air supplied by the fan and the heater for 420 seconds during the experiment at each operating condition; this time range was long enough to capture the appropriate number of images, yet still short enough not to reach the steady temperature state on the inner diffuser wall surface. In this way, the heat transfer of the outer diffuser wall surface practically did not affect the temperature field of the inner wall surface. Nevertheless, it was recognized that the typical shape of the temperature profile did not change much after about 15 seconds of operation. The diffuser operated at the same set of rotating frequencies f and mass flows as in the case of velocity field measurements. However, the airflow inlet Re number was different because of the different airflow temperatures T_i at the diffuser inlet (Table 2).

Table 2: *Operating conditions of the diffuser during temperature field measurements.*

f **[Hz]**	**Re [-]**
0	$4.90 \cdot 10^3$ @ $T_i = 360$ K
30	$9.53 \cdot 10^3$ @ $T_i = 374$ K
52.8	$1.72 \cdot 10^4$ @ $T_i = 353$ K

3. RESULTS

The results show that only a thin layer of the airflow adjacent to the diffuser inner wall surface rotates because of the rotating diffuser wall [2] (Fig. 4).

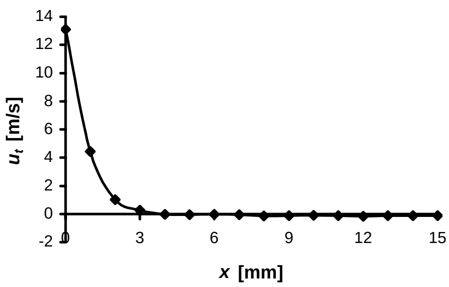

Figure 4: *Distribution of the time averaged mean tangential velocity u_t at the transverse section A (f = 52.8 Hz, Re = $1.17 \cdot 10^4$).*

Apart from that, due to rotation, there is a significant increase not only in values of tangential (u_t) and radial (u_r) velocity components, but also in axial (u_a) velocity components (shown in Fig. 5) in the vicinity of the rotating wall of the diffuser [2].

Since this region of high velocities near the inner surface of the diffuser wall was suspected to influence significantly the temperature field on the diffuser wall, an effort was made to investigate the region more closely. The flow in the region near the diffuser wall was turbulent at each operating condition of the diffuser. This was confirmed by the power spectrum of measured velocity components, which corresponds to the Kolmogorov's law of turbulent decay [3] (Fig. 6).

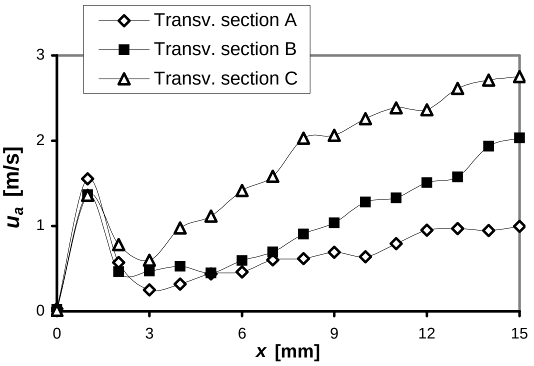

Figure 5: *Distribution of the time averaged mean axial velocity u_a at transverse sections A, B and C (f = 52.8 Hz, Re = $2.01 \cdot 10^4$).*

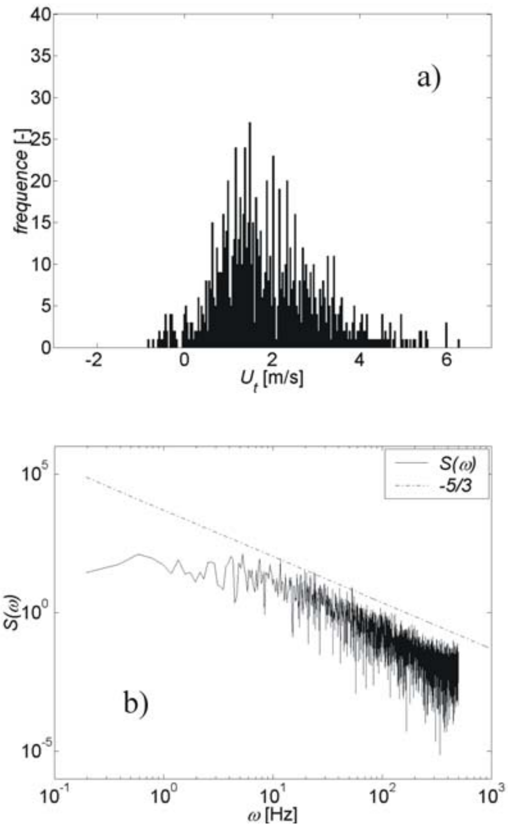

Figure 6: *Distribution (a) and power spectrum S(ω) (b) of instantaneous tangential velocity component at the diffuser outlet at a radial distance x = 1 mm from the diffuser inner wall surface (f = 52.8 H,z Re = $1.17 \cdot 10^4$). The dashed line in diagram (b) represents the Kolmogorov's –5/3 cascade law of turbulent decay.*

In the measuring points that were close to the inner diffuser wall surface, turbulent kinetic energy (*tke*) was calculated using the Eq. 2 [4]:

$$tke = \frac{1}{2}\left(\overline{u_t'^2} + \overline{u_r'^2} + \overline{u_a'^2}\right), \qquad (2)$$

where $\overline{u'^2}$ denotes the mean square of the appropriate velocity fluctuation component. Values of *tke* at different diffuser transverse sections and operating conditions are shown in Fig. 7.

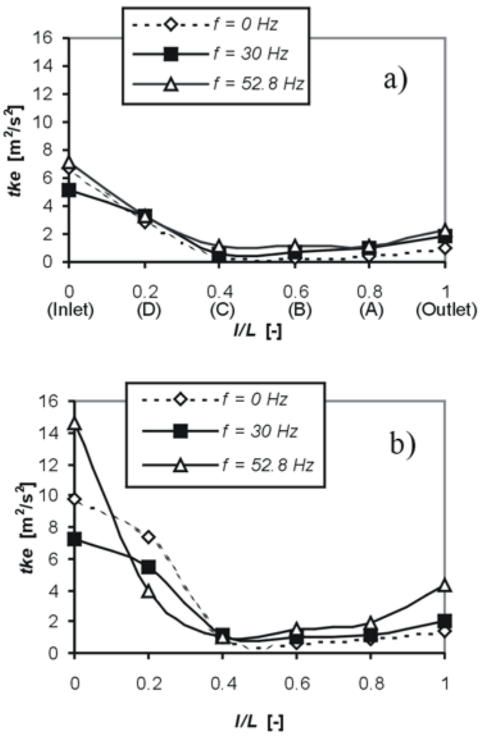

Figure 7: *Distribution of turbulent kinetic energy along the diffuser length at a radial distance x = 1 mm from the diffuser inner wall surface; a) – Re = $5.84 \cdot 10^3$, b) – Re = $2.01 \cdot 10^4$).*

It can be seen from Fig. 7 that the values of *tke* change significantly with Re number as well as with the rotation frequency *f*. While rotation seems to generally increase the values of turbulent kinetic energy at lower Re numbers along the whole length of the diffuser (Fig. 7a), this obviously does not apply to higher Re numbers, especially in the region between diffuser inlet and transverse section C (Fig. 7b). It is interesting that the results of temperature measurements on the inner surface of the diffuser wall show similar dependence of the temperature field on operating conditions of the diffuser (Fig. 8). At low inlet Re numbers, initial values of the temperature gradient $\partial T/\partial t$ in the specified region (transverse section D) are the highest in the case of the diffuser that rotates at f = 52.8 Hz and the lowest in the case where f = 0 (Fig. 8a). The situation at higher inlet Re numbers in the same region of the inner diffuser wall surface is quite the opposite (Fig. 8b). This is also evident from temperature distributions on the inner wall surface along the diffuser length at the beginning of the experiment and after 20 seconds of operation (Fig. 9). The area between two curves (t = 0 s and t = 20 s) at fixed rotation frequency of the diffuser (f = 0 Hz or f = 52.8 Hz) could be used as a measure for heat transfer rate between the airflow and the diffuser wall. It can be seen from Fig. 9a that the area between two corresponding curves at f = 52.8 is much larger than the area betwen curves at f = 0 Hz. At higher Re numbers (Re = $1.72 \cdot 10^4$, Fig. 9b), the area between two curves at f = 0 Hz is already greater than the area between curves at f = 52.8 Hz., which implies that heat transfer is better in the case of non-rotating diffuser.

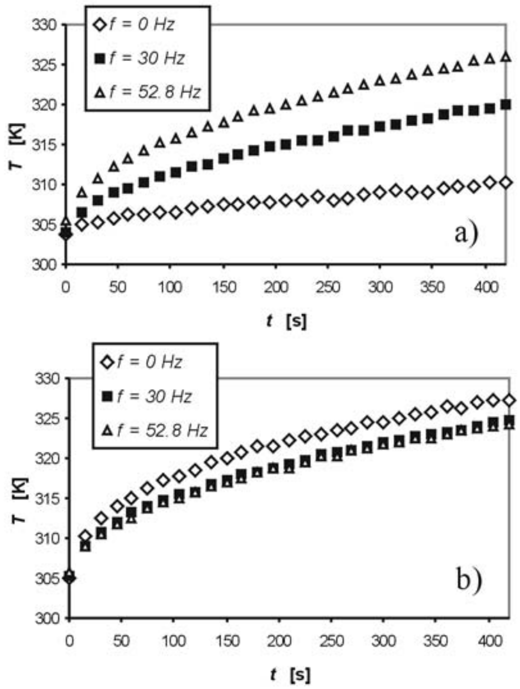

Figure 8: *Temperature time series of the inner diffuser wall surface at the transverse section D (l/L = 0.2); a) – Re = 4.90·10^3, b) – Re = 1.72·10^4).*

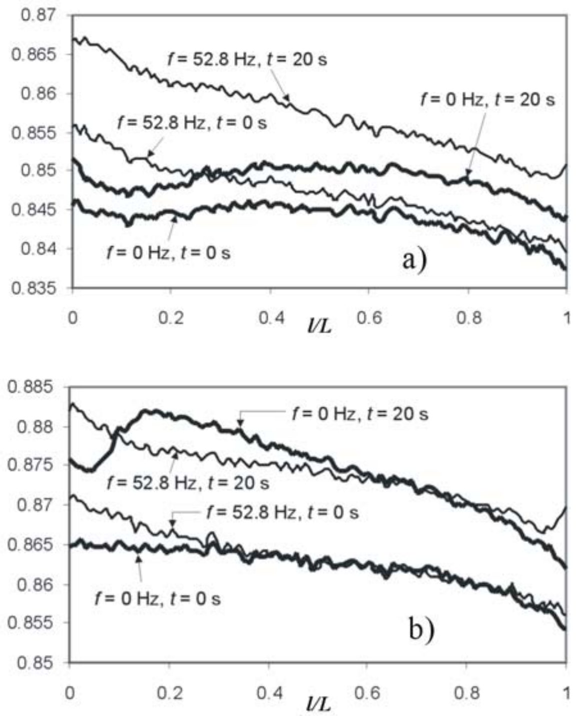

Figure 9: *Temperature of the inner diffuser wall surface at f = 0 Hz and f = 52.8 Hz; a) – Re = 4.90·10^3, b) – Re = 1.72·10^4).*

Important features here are the occurence of flow separation and consequential flow reattachment. Both are present in the case where f = 0 Hz, but are suppressed by the rotation. Flow separation decreases the rate of heat transfer and can be detected by the temperature minimum close to the diffuser inlet in Fig. 9 at f = 0 Hz. This region can be seen in IR images as the dark area near the diffuser inlet (marked by an arrow in Fig. 10a) On the other hand, the flow reattachment, which follows downstream of the flow separation, greatly increases the rate of heat transfer [5], therefore the region of flow reattachment can be traced via the position of the temperature maximum (e.g. at $l/L \approx 0.2$ in Fig. 9b). Due to rotation, no such regions were observed in the case of a diffuser, rotating at f = 52.8 Hz (Fig. 10b).

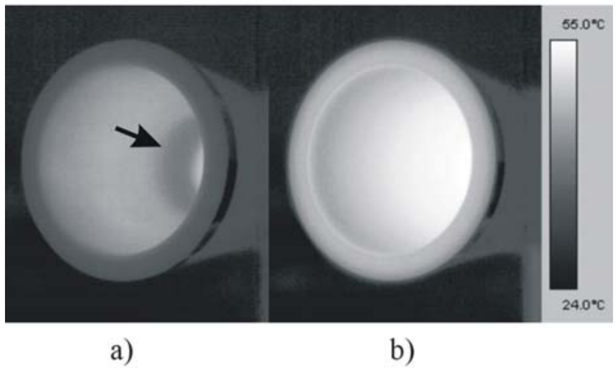

Figure 10: *Infrared (IR) images of the diffuser at Re = 4.90·10^3 after t = 420 s of operation; a) f = 0 Hz; b f = 52.8 Hz.*

4. CONCLUSIONS

Experimental results show that heat transfer inside the rotating diffuser is strongly influenced by turbulent flow structures, which are the consequence of the diffuser geometry, difusser rotation and Reynolds number of the flow. Both integral parameters, rotation frequency and inlet Reynolds number, can increase or decrease the heat transfer rate between the airflow and the inner wall surface of the diffuser. Thus, these two parameters cannot be taken separately in the assessment of heat transfer inside the rotating diffuser. There is an important role of phenomena such as flow separation from the inner diffuser wall surface and its reattachment, both of which influence the heat transfer rate. Rotation of the diffuser was found to suppress the occurrence of these phenomena to some extent.

BIBLIOGRAPHY

[1] Gupta, A.K., Lilley, D.G., Syred, N. *"Swirl Flows"*, Abacus Press, Tunbridge Wells, 1984

[2] Bajcar, T., Širok, B., Trenc, F. *"Flow kinematics in a rotating axial diffuser"*, Experimental Thermal and Fluid Science 27 (7), pp.769-780, 2003

[3] McComb W. D. *"The Physics of Fluid Turbulence"*, Oxford University Press, Oxford, 1996

[4] Bradshaw, P. *"Turbulence, Topics in Applied Physics"*, Springer-Verlag, Berlin, 1976

[5] Terekhov, V.I., Yarygina, N.I., Zhdanov, R.F. *"Heat transfer in turbulent separated flows in the presence of high free-stream turbulence"*, International Journal of Heat and Mass Transfer, Vol. 46 (23), pp. 4535-4551, 2003