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Experimental study of the convective heat transfer from a rotating disc subjected to forced air streams

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ABSTRACT

The convective heat transfer from a rotating disc subjected to forced streams of air has been investigated experimentally by means of an electrically heated disc in a wind tunnel. Mean heat transfer coefficients have been obtained for a wide range of air crossflow velocities and rotational speeds. The influence of finite disc thickness and incidence (angle of attack) on heat transfer has been investigated. The extreme conditions of a stationary disc in a crossflow and a rotating disc in still air have been considered, too. Concerning the onset of rotational heat transfer augmentation it is found that a universal ratio exists between the rotational and the crossflow Reynolds numbers.

1. Introduction

Convective heat transfer from a rotating disc has been studied extensively in the literature (see, for instance, the reviews by Dorfman [1], Owen and Rogers [2,3], or Shevchuk [4]), because this configuration represents an obvious starting point for analysing a lot of technical applications occurring in the areas of turbo machinery, computer hard discs, CVD reactors, train wheel or disc brake design. The majority of the investigations considers enclosed rotating discs or free rotating discs with or without an outer forced flow perpendicular to the disc plane. Even the latter basic case is still subject of actual research, as demonstrated for instance by the work of Elkins and Eaton [5] or Shevchuk [6]. Configurations based on a rotating disc subjected to an outer crossflow parallel to the plane of rotation have found much less attention, and the heat transfer from an inclined rotating disc (i.e. with an angle of attack to the uniform stream) was obviously not investigated so far.

Dennis et al. [7] have studied in 1970 experimentally the mean heat transfer coefficient for a large range of rotational and crossflow Reynolds numbers by means of an electrically heated disc placed in a parallel air stream of a wind tunnel. They have reported "that the freestream turbulence in the tunnel must have been high", perhaps yielding systematically higher heat transfer rates. Booth and de Vere [8] have conducted in 1974 a comprehensive set of measurements of the radial variation of the heat transfer coefficients for the same configuration but obviously without knowledge of the prior work [7]. The both authors have concluded that "in any situation the level of heat transfer coefficients is determined in the main by the speed of the transverse air flow". Such a statement is at least partially in contradiction to the findings of Dennis et al. [7]. He et al. [9] have employed in 2005 the naphthalene sublimation technique for obtaining the local Sherwood number of a rotating disc for a limited range of a local Reynolds number, but unfortunately they have not made a distinction between the rotational and the crossflow Reynolds numbers. Recently, aus der Wiesche has presented results of an extensive large-eddy-simulation (LES) study [10]. His numerically data have indicated much lower heat transfer coefficients as measured experimentally by Dennis et al. [7].

Since the deviations between the available data are substantial, an independent re-investigation has been performed. This present study considers in addition to the classical axisymmetric configuration also the influence of finite disc thickness and incidence (angle of attack) on the heat transfer. The basic flow fields caused by the crossflow are illustrated by means of Fig. 1 for a stationary disc. Whereas the forced flow perpendicular to the disc leads to an axisymmetric stagnation flow, Fig. 1a, the combination of rotation and transverse crossflow leads to a non-axisymmetric stagnation flow in case of an inclined disc, Fig. 1b. The parallel flow over a disc with finite thickness is characterized by flow separation at the leading disc edge followed by reattachment of a turbulent boundary

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Nomenclature

a	coefficient, –
A	order-parameter (Landau model), –
C	correlation constant, –
C_W	drag coefficient, –
d	disc thickness, m
I_H	electric current, A
h	mean heat transfer coefficient, W m^{-2} K^{-1}
L	length in flow direction, m
m	correlation exponent, –
Nu	mean Nusselt number, $\text{Nu} = hR/\lambda$, –
p	pressure, Pa
P_H	electric heat input, W
Pr	Prandtl number, –
R	disc radius, m
Re_u	crossflow Reynolds number, $\text{Re}_u = u_\infty R/\nu$, –
Re_ω	rotational Reynolds number, $\text{Re}_\omega = \omega R^2/\nu$, –
T	temperature, K
TF	turbulence factor, –
u	crossflow velocity, m s^{-1}
U_H	electric voltage, V
x	co-ordinate (in flow direction), m
y	co-ordinate (normal to flow direction), m

Greek symbols

α	angle of attack, °
λ	thermal conductivity, W m^{-1} K^{-1}
ν	kinematic viscosity, m^2 s^{-1}
ω	rotational speed, rad s^{-1}
Ψ	control-parameter (Landau model), –

Subscripts

1, 2, 3	temperature sensor number 1, 2, 3
cr	critical value
t	turbulent
x	local
∞	reference or ambient value

Mathematical symbols

Δx	difference or uncertainty of variable x
$\langle x \rangle$	averaging of variable x

layer, Fig. 1c, and a substantial different behaviour in respect to heat transfer is expected. In agreement with Kreith [11] it is not only important to know what methods and correlations are available to calculate the heat transfer, but it is also important to appreciate the limits of the existing knowledge and to realise that particular in rotating systems often new and unexpected phenomena may occur.

2. Apparatus and experimental procedure

The mean heat transfer coefficients were found directly by measurements of the electric heat input and the surface and ambient air temperatures after steady state was achieved for given rotational and crossflow speeds and angle of attacks.

2.1. Experimental apparatus

The apparatus used for the present study is shown in Fig. 2. It consists of an electrically heated composite disc with radius $R = 200$ mm carried overhung at one end of a shaft mounted in bearings. Lead wires from the heater layer and thermocouples are brought out through slip rings mounted on the shaft. The shaft can be driven from an electric motor and is capable of rotational speeds up to 3000 r.p.m.. The test rig is placed with the disc central in the neck of an open jet wind tunnel with airspeed from 1.3 m/s up to 40 m/s enabling crossflow Reynolds numbers $1.7 \times 10^4 < \text{Re}_u < 5.2 \times 10^5$. Additionally to the parallel configuration, the test structure with the rotating disc can be inclined to the air stream. To avoid vibrations, the structure is mounted with rubber damping elements to the fixed ground plate.

The disc is a composite structure, as shown in Fig. 2 in detail. An electric heater layer (a Silikon heater designed and manufactured by Horst GmbH, Lorsch, Germany) is glued directly under an aluminium disc with thickness 2 mm and radius 200 mm. The upper surface temperature was continuously measured by means of three PT1000 thermocouples (in accordance to DIN EN 60751, class 1/3 DIN) fitted in grooves flush with the surface at different radial locations. To ensure a sufficient thermal insulation to the bottom surface and to the shaft, an air gap of 1.5 mm between the heater layer and the bottom plastic disc has been designed. However, by preliminary tests a measurable heat loss through the bottom surface has been noticed and a further insulating Styrofoam layer has been added to the bottom disc. The final thickness of the composite disc is 24 mm. By adding further layers, it is possible to increase the aspect ratio d/R of the disc to desired values.

2.2. Preparatory measurements

Prior to the heat transfer measurements the turbulence level and the crossflow velocity profiles of the employed wind tunnel have been analysed. The concept of the turbulence factor TF, for instance presented by Barlow et al. [12], relates the turbulence level in a wind tunnel to the Reynolds number $\text{Re}_{u,\text{cr}}$ at which the drop in drag of a sphere is observed. Based on experiments on perfectly smooth spheres in an atmospheric flow, it is known that this

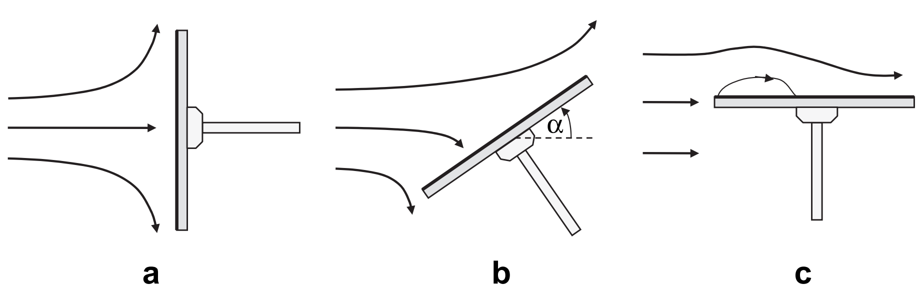

Fig. 1. Schematics of the flow fields caused by the outer forced flow in case of a stationary disc.