Crossflow heat transfer
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Publisher's version / Version de l'éditeur:

International encyclopedia of heat and mass transfer, 1997

CROSSFLOW HEAT TRANSFER.

Following from: Crossflow

When a fluid flows across a solid-object or ensemble of solids at a different temperature, *crossflow heat transfer* results. Heat transfer is a function of Reynolds number,

$$Re = \frac{\rho u D}{\eta} \tag{1}$$

and Prandtl number,

$$Pr = \frac{\eta c_p}{\lambda} \tag{2}$$

where ρ is density, u is a bulk velocity, η is viscosity, λ is the fluid conductivity, and c_p is specific heat. D is a characteristic length, such as a diameter. Another popular choice for crossflow heat exchangers is a hydraulic diameter, D_h.

At very low flow-rates, heat transfer is by *conduction* alone. If the objects are inclined to the vertical, *natural or mixed-convection* may be important. Under these circumstances, the engineer may also be interested in maintaining *thermal stratification.* As *Re* is increased *forced-convection* becomes the dominant mode of heat transfer. The *local heat transfer coefficient*, α, is defined by,

$$\dot{q} = \alpha (T_w - T_b) \tag{3}$$

where q is the local wall heat flux, T_w is the wall temperature, and T_b is a reference bulk temperature. For single objects T_b is chosen as the free-stream temperature. For banks of objects, a variety of references are used. α is non-dimensionalized in terms of a *local Nusselt number*, *Nu*,

$$Nu equiv \frac{\alpha D}{\lambda} \tag{4}$$

Figure 1(a) and (b) show *Nu* distributions around a circular cylinder in the *Re* ranges $4x10^3$-$5x10^4$ and $3.98x10^4$-$4.26x10^5$ respectively. It can be seen that at low *Re, Nu* is a maximum, at $\phi = 0°$, where skin-friction is zero. In this region, the flow is similar to that occurring at the stagnation point for flow normal to a flat-plate. *Nu* decreases to a minimum near to the separation point, at the side of the cylinder, and then increases in the wake. Most heat transfer occurs through the front-half of the cylinder. As *Re* increases, heat transfer in the latter-half increases due to wake-turbulence. At $Re = 2\text{x}10^5$, *a turbulent thermal boundary layer* is established and the *Nu* distribution is quite complex, with twin minima occurring at around 90° and 140° probably due to laminar-to-turbulent transition in the boundary-layer, and boundary-layer separation, respectively. At high *Re,* there is a large *Nu* maximum in the turbulent boundary-layer around $\phi = 110°$. Heat transfer at the rear stagnation-point is by now as much as at the front stagnation point.

Overall heat transfer is expressed in terms of an *average heat transfer coefficient*, , defined by means of a *rate equation*,

$$\dot{Q} = \bar{\alpha} A \Delta T_M \qquad \textbf{(5)}$$

where Q is the total rate of heat transfer, *A* is the total heat transfer area, and ΔT_M is an average or effective temperature difference between the solid-wall and the bulk of the fluid. For situations involving the use of extended surfaces or *fins* a modified rate equation is employed. (*See: Tube banks, Crossflow Over*). is non-dimensionalized either as an *average Nusselt number*, N*u*, according to Eq. 4 or as an *average Stanton number*, S*t*

$$\overline{St} = \frac{\bar{\alpha}}{\rho c_p u} \qquad \textbf{(6)}$$

Re and *Pr* are often evaluated at the bulk temperature, T_b, though some authors advocate the use of a *mean film temperature* $(T_w + T_b)/2$. N*u* and S*t* are frequently correlated according to a power-law relationship,

$$\overline{Nu} = \left(a + c\,Re^m\right) Pr^n \qquad \textbf{(7)}$$

where *c* and *m* are *Re*-dependent, and *a* accounts for natural convection, if present. Use of *n* = _, based on the Colburn-Chilton analogy is widespread, though empirical correlations may involve different values of *m*. Figure 2 shows N*u* and S*t* as a function of *Re* and *Pr.* A *Pr*-independent *heat transfer factor*, *j*, is defined as,

$$j = \overline{St}\, Pr^{1-n} \qquad \textbf{(8)}$$

and may be considered the heat transfer analogue of the friction factor, *f*/2.

Temperature variation of fluid properties affects both *f* and *j*. This may be accounted for by writing,

$$j = j' \left(\frac{Pr}{Pr_w}\right)^p \approx j' \left(\frac{\eta}{\eta_w}\right)^p \qquad \textbf{(9)}$$

where *j* ' denotes an temperature-independent (adiabatic) value, and Pr_w is evaluated at the wall temperature, T_w. *f* is treated in a similar manner. Temperature-independent values of *f* ' and *j* ' appear in the literature for a large number of geometries. These measures of overall performance are of much interest to the heat transfer engineer.

The geometry of the heat transfer surface will substantially alter the mechanisms of fluid flow and heat transfer in crossflow. Other influencing factors include; the effect of thermal boundary conditions (constant q_w vs. T_w), the influence of containing ducts or other blockage, free-stream-turbulence, as well as the use of roughened surfaces or fins to enhance heat transfer.

References

Lohrische, W. (1929) Forschungsarbeiten auf dem Gebeite des Ingenieurwesens. No. 322, pp. 46, 1929. (*In German*)

Schmidt, E., Wenner, K. (1941) Forschung auf dem Gebeite des Ingenieurwesens. Vol. 12, No. 2, pp. 65-73, 1941. (*In German*)

_ukauskas, A., _iug_da, J. (1985) Heat Transfer of a Cylinder in Crossflow. Hemsisphere, New York, 1985. (*Translator E. Bagdonaite, Editor G.F. Hewitt*)

Leading to:

Tube banks, single-phase heat transfer in

Tube (single), single-phase heat transfer to in cross flow

Tube banks, boiling heat transfer in

Tubes (single), boiling on outside of in cross flow

Tube banks, condensation heat transfer in

Tube (single), condensation on outside of in cross flow

Cross reference terms

Cylinder, crossflow heat transfer

Heat transfer factor

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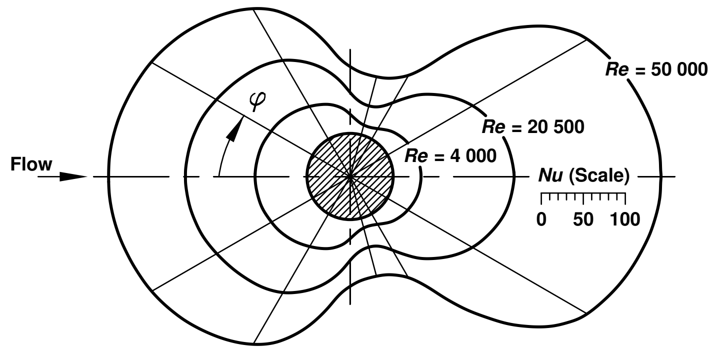

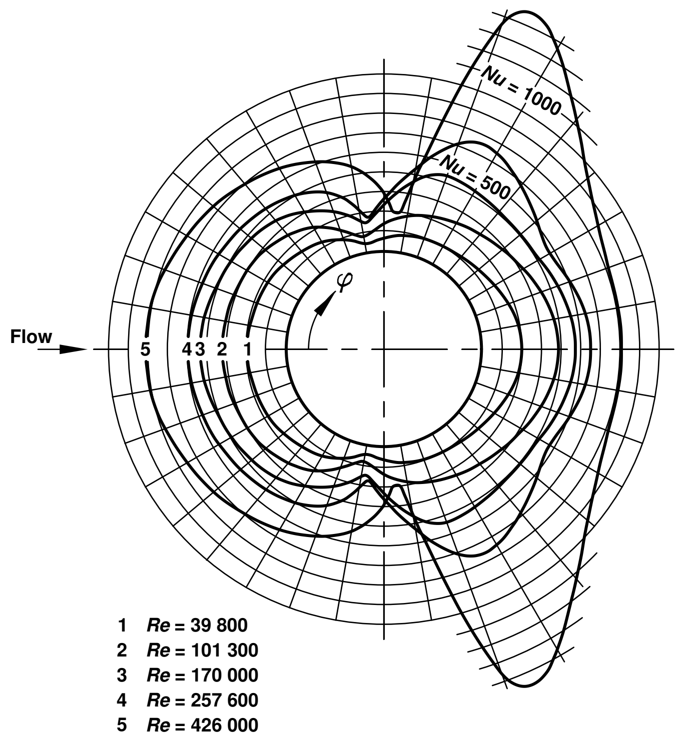

Fig. 1. Distribution of local heat transfer coefficient around a circular cylinder for flow of air. From (a) Lohrisch (1929) and (b) Schmidt and Wenner (1941)

Fig. 2. Average heat transfer for flow around a circular cylinder expressed in the form of Nusselt and Stanton numbers. From _ukauskas et al. (1985)