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FUNDAMENTALS OF THERMAL-FLUID SCIENCES

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s the constant *average* con-
onential of both sides and
on for the determination of

c_p (17-35)

an fluid temperature $T_m(x)$

the fluid and the surface
rate of decay depends on
own in Fig. 17-21. This
ransfer units, denoted by
heat transfer systems. For
s almost equal to the sur-
the fluid temperature can
t, an NTU of about 5 indi-
e heat transfer will not in-
the tube. A small value of
s for heat transfer, and the
ngth is increased. A large
hich means a large tube)
, but it may be unaccept-
of heat transfer equipment
performance and cost.

(17-36)

(17-37)

$\frac{T_i}{T_i)}$ (17-38)

Note that $\Delta T_i = T_s - T_i$
between the surface and
tively. The ΔT_{ln} relation
ally failsafe, since using
or the denominator will,
n be used for both heat-
fluid in a tube.

using the arithmetic mean temperature difference is less than 1 percent. But the error increases to undesirable levels when ΔT_e differs from ΔT_i by greater amounts. Therefore, we should always use the logarithmic mean temperature difference when determining the convection heat transfer in a tube whose surface is maintained at a constant temperature T_s.

17-6 THERMAL ENTRY LENGTH

The Reynolds number for flow inside pipes of diameter D was defined in Chap. 12 as

$$\mathrm{Re} = \frac{\mathcal{V}_m D}{\nu} \tag{17-39}$$

where $\mathcal{V}_m$ is mean fluid velocity and $\nu = \mu/\rho$ is the kinematic viscosity of the fluid. Under most practical conditions, the flow in a pipe was said to be laminar for $\mathrm{Re} < 2300$, turbulent for $\mathrm{Re} > 4000$, and transitional in between.

Relations for hydrodynamic entry length were developed in Chap. 12 by considering a fluid entering a circular pipe at a uniform velocity. We now consider a fluid at a uniform temperature entering a circular tube that is at a different temperature. This time, the fluid particles in the layer in contact with the surface of the tube will assume the surface temperature. This will initiate convection heat transfer in the tube and the development of a **thermal boundary layer** along the tube. The thickness of this boundary layer also increases in the flow direction until the boundary layer reaches the tube center and thus fills the entire tube, as shown in Fig. 17-23. The region of flow over which the thermal boundary layer develops and reaches the tube center is called the **thermal entry region,** and the length of this region is called the **thermal entry length** L_t. The region beyond the thermal entry region in which the dimensionless temperature profile expressed as $(T - T_s)/(T_m - T_s)$ remains unchanged is called the **thermally developed region.** The region in which the flow is both hydrodynamically and thermally developed is called the **fully developed flow.**

Note that the *temperature profile* in the thermally developed region may *vary* with x in the flow direction. That is, unlike the velocity profile, the temperature profile can be different at different cross-sections of the tube in the developed region, and it usually is. However, it can be shown that the

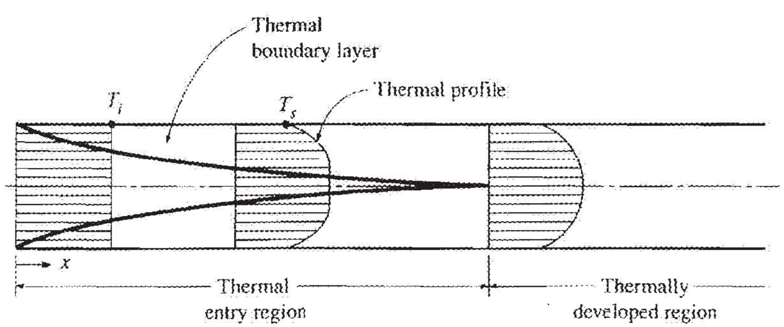

FIGURE 17-23
The development of the thermal boundary layer in a tube. (The fluid in the tube is being cooled.)

dimensionless temperature profile defined above remains unchanged in the thermally developed region when the temperature or heat flux at the tube surface remains constant.

In laminar flow in a tube, the magnitude of the dimensionless Prandtl number Pr is a measure of the relative growth of the velocity and thermal boundary layers. For fluids with $\mathrm{Pr} \approx 1$, such as gases, the two boundary layers essentially coincide with each other. For fluids with $\mathrm{Pr} \gg 1$, such as oils, the velocity boundary layer outgrows the thermal boundary layer. As a result, the hydrodynamic entry length is smaller than the thermal entry length. The opposite is true for fluids with $\mathrm{Pr} \ll 1$ such as liquid metals.

The hydrodynamic and thermal entry lengths in laminar and turbulent flows are given approximately as

$$L_{h,\,\text{laminar}} \approx 0.06\,\mathrm{Re}\,D \tag{17-40}$$

$$L_{t,\,\text{laminar}} \approx 0.06\,\mathrm{Re}\,\mathrm{Pr}\,D = \mathrm{Pr}\,L_{h,\,\text{laminar}} \tag{17-41}$$

$$L_{h,\,\text{turbulent}} \approx L_{t,\,\text{turbulent}} \approx 4.4D\,(\mathrm{Re})^{1/6} \tag{17-42}$$

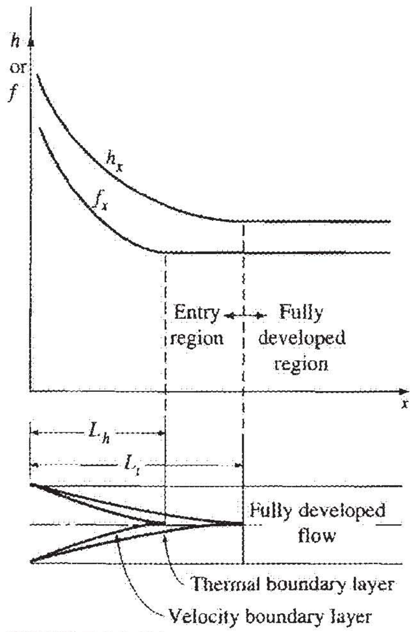

FIGURE 17-24

Variation of the friction factor and the convection heat transfer coefficient in the flow direction for flow in a tube ($\mathrm{Pr} > 1$).

In *turbulent flow*, the intense mixing during random fluctuations usually overshadows the effects of momentum and heat diffusion, and therefore the hydrodynamic and thermal entry lengths are of about the same size. Also, *the friction factor and the heat transfer coefficient remain constant in fully developed laminar or turbulent flow* since the velocity and normalized temperature profiles do not vary in the flow direction.

Consider a fluid that is being heated (or cooled) in a pipe as it flows through it. The friction factor and the heat transfer coefficient are *highest* at the pipe inlet where the thickness of the boundary layers is zero, and decrease gradually to the fully developed values, as shown in Fig. 17-24. Therefore, the pressure drop and heat flux are *higher* in the entry regions of a tube, and the effect of the entry region is always to *enhance* the average friction and heat transfer coefficients for the entire tube. This enhancement can be significant for short tubes but negligible for long ones.

Precise correlations for the heat transfer coefficient for the entry regions are available in the literature. However, the tubes used in practice in forced convection are usually many times the length of either entry region, and thus the flow through the tubes is assumed to be fully developed for the entire length of the tube. This approach, which we will also use for simplicity, gives *reasonable* results for long tubes and *conservative* results for short ones.

17-7 ■ FORCED CONVECTION IN PIPES

We mentioned earlier that flow in smooth tubes is laminar for $\mathrm{Re} < 2300$. The theory for laminar flow is well developed, and both the friction and heat transfer coefficients for fully developed laminar flow in smooth circular tubes can be determined analytically by solving the governing differential equations. The Nusselt number in the fully developed laminar flow region in a circular tube is determined from the energy equation to be (Fig. 17-25)

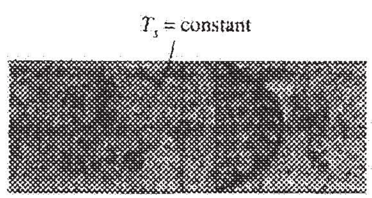

FIGURE 17-25

$\mathrm{Nu} = 1.8$

All properties are evalua
which is evaluated at the
The Nusselt number
laminar flow in tubes o
numbers for flow in the
defined as

where A_c is the cross-sect
Nusselt number is availa
mined from $h = k\,\mathrm{Nu}/D_h$

TABLE 17-3

Nusselt number for fully d
sections ($D_h = 4A_c/p$, Re

Cross-section of tub
Circle
Rectangle
Ellipse