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Heat transfer characteristics of a bayonet tube using air under laminar conditions

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Abstract—This paper considers the heat transfer performance of a bayonet tube under laminar conditions when the fluid is air. The results of a series of laboratory experiments on a small bore rig are reported. Particular attention is given to the effect of the mass flow rate, as represented through the Reynolds number, and the effects of system geometry, as represented through the end clearance ratio, the flow area ratio and the length–diameter ratio. After considering each of these separately, their relative importance is explored in an empirical heat transfer correlation. Practical performance levels are demonstrated.

1. INTRODUCTION

To MAINTAIN soil beneath engineering structures in a frozen state, various devices have been suggested. Among these is the bayonet tube. This consists of a tube which penetrates the soil from an access surface, often horizontal, but which may also be inclined, e.g. on a berm or embankment. Concentrically located within this tube is another which supplies cold fluid, forcing it to return along the annular space created between the two tubes. Heat transfer to the fluid, especially in the annular return, thus provides the means by which the soil may be frozen and subcooled.

This device has been studied both theoretically [1] and experimentally [2–6]. It may be used with any nonfreezing fluid, depending upon the economic penalty and the possibility of environmental degradation in the event of a leakage. Only air will be considered here, partly because it is ecologically benign and partly because it raises the possibility of the flow being driven by the local wind in winter.

Exploratory experiments on a wind-driven system have been conducted previously on a small-bore, laboratory apparatus [7, 8]. These indicated that the flow may be laminar or turbulent, depending on local conditions. A detailed study of the hydraulic characteristics of the bayonet tube under laminar conditions has also been completed [9]. The present work is a complementary study of the heat transfer characteristics under laminar conditions. In particular, it investigates the effect on heat transfer of the main parameters of the system: the air flow rate (prescribed through the Reynolds number) and the tube geometry (defined by the length–diameter ratio, the end clearance ratio and the flow area ratio).

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2. THE EXPERIMENTS

The experiments were conducted on the small bore rig illustrated schematically in Fig. 1. This consisted of a 2.57 cm bore copper tube with a length varying from 25.7 to 102.8 cm; the overall length–diameter ratio (L/D) thus ranged from 10 : 1 to 40 : 1. Inside this tube was inserted another made of acrylic. The inner diameter of this insert tube was varied over the range 1.07 cm $\leqslant d_i \leqslant$ 1.99 cm; its wall thickness also varied slightly (from 1.06 to 1.2 mm). The insert tube was aligned concentrically with the main tube and extended to a distance H (from the closed lower end) which was varied such that $0.034 \leqslant H/D \leqslant 2.01$.

The air entering the insert tube came directly from a settling chamber which in turn was supplied from the building mains. Prior to entering the settling chamber, however, the air was depressurized in a flow regulator positioned upstream of a control valve. Located well downstream of this valve was a laminar mass flow meter (Meriam Instruments 50 MJ 10). No attempt was made to control the air leaving the apparatus; it was allowed to discharge into quiescent room air.

The pressure difference across the flow meter was measured with a Validyne transducer (DP 15-30), the signal from which was read on a digital multimeter after demodulation. The mass flow rate thus determined was used to calculate mean velocities by invoking continuity; density was calculated from the local temperature and pressure using the ideal gas equation of state. Static pressure was measured at two points having the same elevation, both corresponding to the mouth of the bayonet tube.

Temperatures were measured throughout using copper–constantan thermocouples. As indicated, these were located in the settling chamber, the insert tube and at various locations (every 4 cm) in the main tube wall. A thermocouple was also located in the room air for calibration purposes. Thermocouple signals were fed through a switching box and read on a digital multimeter.

NOMENCLATURE

d	inner tube diameter	Re	Reynolds number
D	outer tube diameter	T	temperature.
F	flow area		
h	heat transfer coefficient	Subscripts	
H	end clearance	a	annulus
L	overall tube length	f	fluid mean
Nu	Nusselt number	i	inside
$\dot{Q}$	heat flux	o	outer
Ra	Rayleigh number	w	tube wall.

The tube was heated by means of a heating tape wrapped around the outside after the thermocouples had been installed. Electrically insulating tape separated the heating tape from the tube wall. A thick layer of thermal insulation then wrapped over the heating tape served to limit the heat lost to the atmosphere. The latter was estimated from a calibration test in which the insert tube was withdrawn and replaced by thermal insulation. In this condition, the power supplied to the heating tape may be plotted against the temperature difference between the tube wall and the surrounding air thus providing an estimate of heat loss during the tests. The heating tape was installed over four equal (25.7 cm) lengths with the windings being connected in parallel through a multi-channel temperature controller. This was fed from the main building power supply and served to adjust the power supplied to individual windings so as to maintain a close approximation to an isothermal tube wall (± 2 K).

The experimental procedure was as follows. With the calibration test completed, a particular insert tube was installed and the control valve carefully opened to establish the flow rate at a particular level. The

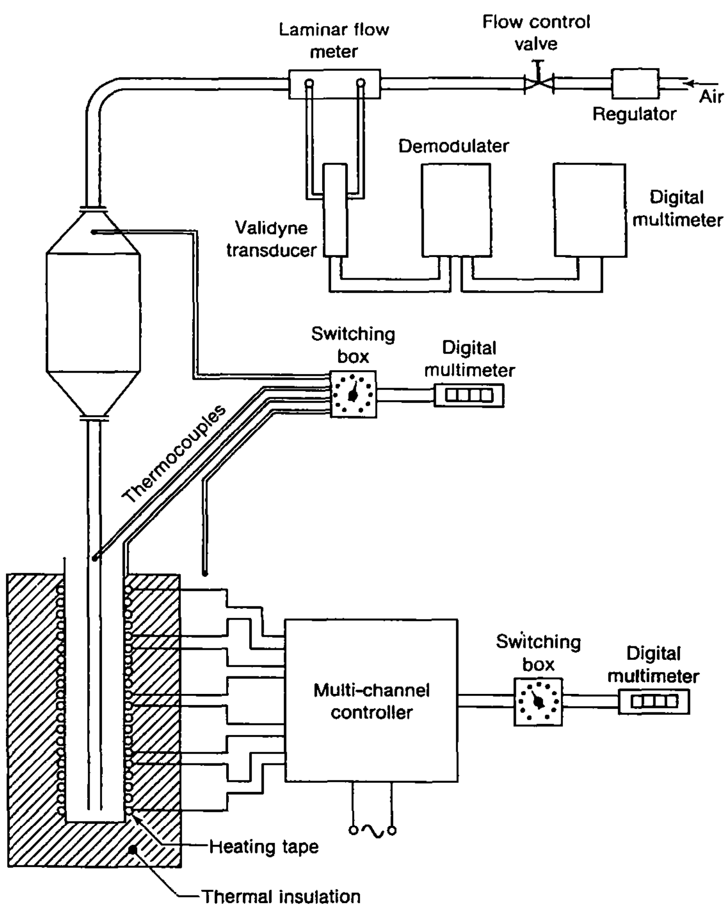

FIG. 1. Schematic of bayonet tube rig.

temperature controller was then switched on and the heat supply rate increased until the wall temperature was set well in excess of the inlet air temperature (e.g. 100°C). After a period of about 5 h, a steady state was reached. Readings were then taken of flow rate, temperature and pressure. The flow rate was then altered and the procedure repeated with the same wall temperature after a re-adjustment period of about half an hour. In this way, alterations in the heat flux produced no change in the external temperature field; the heat leakage thus remained the same.

A formal error analysis of each of the variables was undertaken. Uncertainty bars are shown on most of the figures discussed below.

3. DISCUSSION OF RESULTS

3.1. *Definitions*

The data obtained above were all converted into nondimensional groups describing various effects on thermal behavior. In this process fluid properties were calculated at the mean fluid temperature T_f between inlet and outlet. The temperature difference employed in the calculations was then defined by $T_w - T_f$. This was used in the definition of the Nusselt number

$$Nu = \frac{\dot{Q}(D-d_o)}{\pi k D L (T_w - T_f)} \tag{1}$$

and the Rayleigh number

$$Ra = \beta g \frac{(D-d_o)^3 (T_w - T_f)}{\nu \kappa} \tag{2}$$

both of which are based on the gap in the annulus, where most of the heat transfer takes place. The Reynolds number was defined by

$$Re = \frac{4\dot{m}}{\mu \pi d_i}\left(\frac{D}{D+d_o}\right) \tag{3}$$

following Lock and Wu [9] who noted that this expression using the mass flow rate $\dot{m}$ takes appropriate limiting forms when $d_o/D \to 0$ or 1.

3.2. *Parametric study*

Figure 2 shows a plot of Nusselt number vs the flow area ratio

$$\frac{F_i}{F_a} = \frac{d_i^2}{D^2 - d_o^2}$$

with the other parameters essentially fixed at set reference values: specifically, $L/D = 20$, $H/D \simeq 1.0$ and $Re \simeq 900$. Also shown is the corresponding plot of the heat transfer coefficient

$$h = \frac{\dot{Q}}{\pi D L (T_w - T_f)}.$$

The latter is seen to be a monotonically increasing function of F_i/F_a, indicating that, for a given heat flux, the mean fluid temperature rises towards the wall temperature as the annular gap decreases; this result

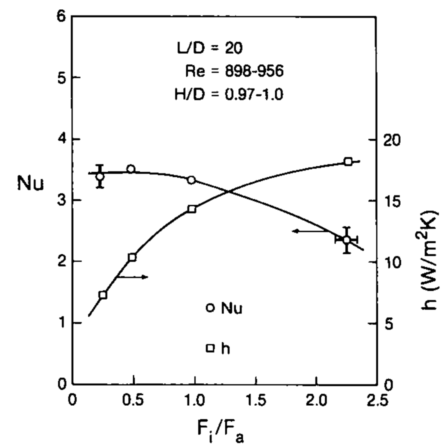

FIG. 2. Effect of flow area on Nusselt number and heat transfer coefficient.

was not unexpected. It is clear that heat transfer coefficients in excess of 10 W m^{-2} K^{-1}, a practical value in cryosyphon design, are readily obtainable under laminar conditions. The Nusselt number may not be a monotonic function of F_i/F_a. The raw data in Fig. 2 suggest a shallow optimum near $F_i/F_a = 0.5$, the location of the pressure drop minimum [9]. However, the uncertainty is too great to confirm this. The observed decrease in Nu with F_i/F_a evidently reflects the fact that h increases as a small positive power of F_i/F_a.

The above-mentioned hydraulic study [9] also revealed that the end clearance ratio had a substantial influence on pressure loss. This influence does not extend to thermal performance, as Fig. 3 indicates. The small spread in the data, particularly for very low values of H/D, may reflect the behavior of the end region vortex, but it is unlikely to have any practical effect on thermal design if H/D is chosen close to 0.5, the value which minimized the hydraulic loss [9].

In contrast, the length–diameter ratio was found to

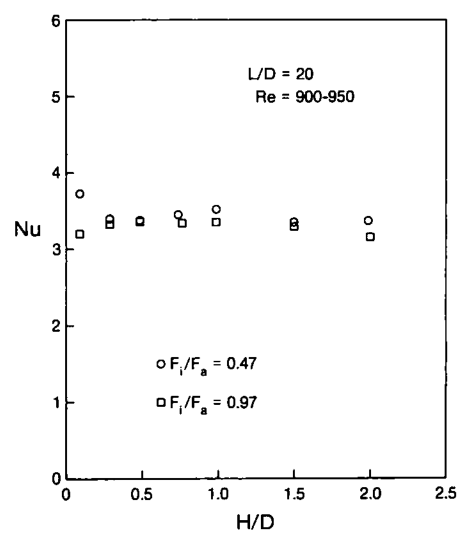

FIG. 3. Effect of end clearance ratio on Nusselt number.

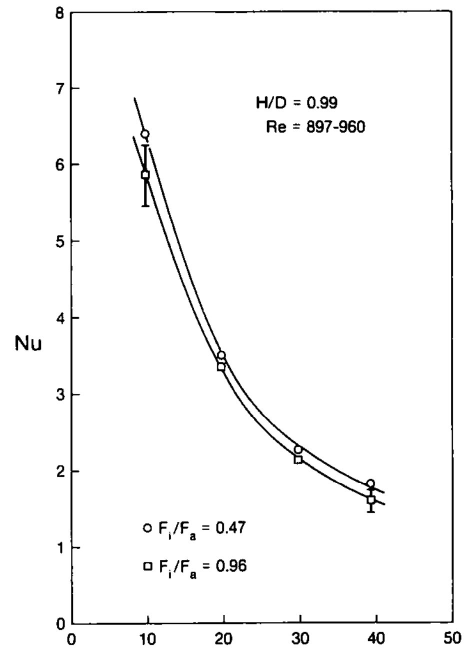

FIG. 4. Effect of length–diameter ratio on Nusselt number.

exert a strong influence on both hydraulic and thermal performance. As would be expected, increasing L/D increases the frictional penalty. Figure 4 reveals that it also has a deleterious effect on heat transfer. The empirical correlation developed below shows that the Nusselt number is very close to being inversely proportional to the length–diameter ratio, thus implying that the overall rate at which heat was supplied ($\dot{Q}$) is independent of the tube length. This suggests that the thermal regime in the annulus was fully developed. An estimate of the appropriate Graetz number, defined by $Gz = Re\,Pr(D-d_o)/L$, indicates that $Gz < 25$, thus supporting the suggestion.

Finally, the effect of Reynolds number was considered. The resultant curves are shown in Fig. 5. Not surprisingly, an increase in Reynolds number, which is directly proportional to the mass flow rate (the system geometry being fixed for each test), leads to an increase in heat transfer. This is attributable to more vigorous forced convection and carries with it a frictional penalty. The data shown are all in the laminar range, but the upper limit of this range is not known precisely for a bayonet tube; the annular flow is downstream of a 180° turn in which the fluid is known to separate. In the event that separation induces turbulence, it is to be expected that the friction loss and heat transfer rate would both increase.

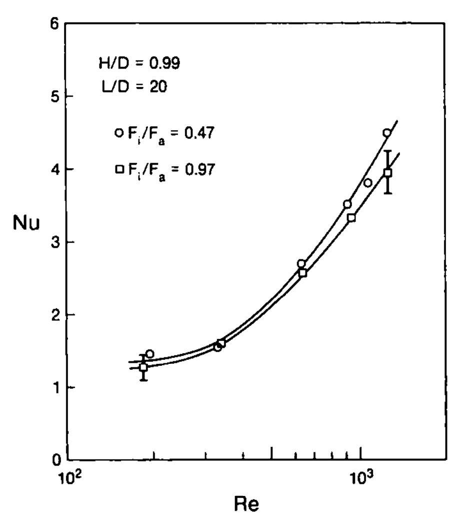

FIG. 5. Effect of Reynolds number on Nusselt number.

3.3. *Empirical correlation*

In searching for an empirical correlation of the data, it was noted from Figs. 3–5 that power law forms might be suitable. The exception was provided by Fig. 2. It was decided that the convexity in Fig. 2 was small enough to be represented by a parabola. With this approximation, various correlating forms were tried from which it rapidly became apparent that the effect of end clearance ratio could be neglected, as suggested by the data in Fig. 3. Each of these attempts eventually reduced to the form

$$Nu = A\left(\frac{D}{L}\right)^{m_1}\left[0.05+\left(0.7-\frac{F_i}{F_a}\right)^2\right]^{m_2} Re^{m_3}\, Ra^{m_4} \tag{4}$$

in which the characteristic length in Nu, Re and Ra was used consistently. Three different lengths were tried: $D-d_o$, $(D^2-d_o^2)/D$ and $2d_i(D-d_o)/D$. The choice was found to have little effect on the coefficient of correlation. A representative correlation based on the definitions given by equations (1)–(3) is shown in Fig. 6. The empirical equation is then given by

$$Nu = 0.60X$$

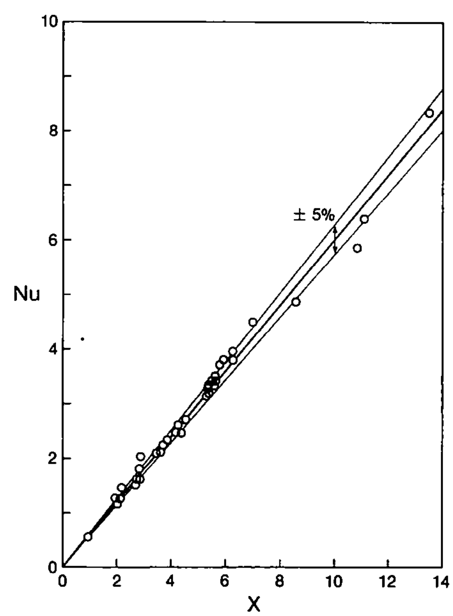

FIG. 6. Empirical heat transfer correlation.

where

$$X = \left(\frac{D}{L}\right)^{0.98}\left[0.05+\left(0.7-\frac{F_i}{F_a}\right)^2\right]^{-0.07} Re^{0.61}\, Ra^{0.05}$$

which emphasizes the strong influence of L/D and Re. The shallow convexity with respect to F_i/F_a is reflected in the small power -0.07. The influence of natural convection is also evident. Although no separate study of the effect of Ra was made, it is worth noting that $40 < Ra < 3000$, well within the laminar range.

4. CONCLUSIONS

This paper presents the results of an experimental study of laminar heat transfer in a bayonet tube using air. The data obtained have been used to interpret the effect of the principal parameters on heat transfer; specific attention has been paid to flow rate and system geometry as defined by the flow area ratio, the end clearance ratio and the length–diameter ratio.

The influence of the flow area ratio on heat transfer coefficient was found to be monotonic. Little, if any, evidence of extremal behavior in the Nusselt number was found. In contrast to the local minima found in a previous study of the frictional penalty, the heat transfer data varied only slightly with flow area ratio and end clearance ratio.

Reynolds number and length–diameter ratio both exerted a strong influence on heat transfer. This dominance was further demonstrated in an empirical correlation which also confirmed the relatively weak influence of end clearance ratio and flow area ratio. The influence of natural convection was also seen in conditions which suggested a fully developed thermal regime. The data indicate that such laminar conditions produce levels of heat transfer which permit practical cryosyphon design using cold winter air.

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