Int. J. Heat Mass Transfer. Vol. 7, pp. 1207–1216. Pergamon Press 1964. Printed in Great Britain

HEAT TRANSFER AND PRESSURE LOSS IN HELICALLY COILED TUBES WITH TURBULENT FLOW

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(*Received* 14 *February* 1964 *and in revised form* 17 *March* 1964)

Abstract—Experimental results for forced convection heat transfer and friction factors, obtained with water flowing through steam heated coils, are reported and compared with the limited results available to date. Existing equations for isothermal friction factors in smooth coils are deemed satisfactory. Non-isothermal friction factors and heat-transfer coefficients can be estimated from proposed equations for design purposes, but results cannot yet claim the same validity as those for straight pipes.

NOMENCLATURE

A, inner surface area of coil;
c, specific heat;
D, mean diameter of coil;
d, bore of coiled tube;
f, friction factor $(\tau_w/\frac{1}{2}\rho u_b^2)$;
h, convection heat-transfer coefficient;
k, thermal conductivity;
l, length of coil measured along tube axis;
P, pitch of coil;
p, pressure;
Q, rate of heat transfer;
t, temperature;
U, overall heat-transfer coefficient;
u, flow velocity;
μ, dynamic viscosity;
ρ, density;
τ, shear stress;
Nu, Nusselt number $(= hd/k)$;
Pr, Prandtl number $(= c_p \mu/k)$;
Re, Reynolds number $(= \rho u_b d/\mu)$.

Suffixes

B, Blasius value $(f_B = 0{\cdot}0791/(Re)^{0{\cdot}25})$;
b, bulk, or weighted mean, value;
f, property at film temperature;
g, saturation value;
p, at constant pressure;
r, at axis of tube;
s, straight tube value;
sup, superheat value;
w, at wall;
1, 2, 3, see Fig. 1.

1. INTRODUCTION

IN SPITE of the frequent use of helically coiled pipes in heating and refrigerating plant, surprisingly little information is available on the inside heat-transfer coefficients in such coils, although adequate data are available on isothermal pressure losses. It is of course plausible that secondary flow, caused by the centripetal forces acting on the fluid, should produce an increase in friction factor and heat-transfer coefficient over that obtained in an equivalent straight pipe, and this has been confirmed by a number of experimenters whose results are summarized below. All equations quoted from other authors have been translated into the nomenclature of the paper. Particular attention is drawn to the friction factor $f = \tau_w/\frac{1}{2}\rho u_b^2$ for which some authors use a factor equal to $2f$ or $4f$.

1.1 *Pressure loss data*

The work on pressure losses in smooth coiled tubes with isothermal turbulent flow was summarized by Ito [1], who proposed two equations, based on the results of several workers, namely

$$f = 0{\cdot}076\,(Re)^{-0{\cdot}25} + 0{\cdot}00725\left(\frac{d}{D}\right)^{0{\cdot}5} \quad (1)$$

for $0{\cdot}034 < (Re)\,(d/D)^2 < 300$, and

$$\frac{f}{f_B} = \left[(Re)\left(\frac{d}{D}\right)^2\right]^{0{\cdot}05} \qquad (2)$$

for $(Re)\,(d/D)^2 > 6$, where f_B is the Blasius value. Transition from laminar to turbulent flow in coils occurs at higher values of Re than in straight tubes, and Ito proposes

$$(Re)_{\text{crit}} = 2 \times 10^4 \left[\frac{d}{D}\right]^{0{\cdot}32}. \qquad (3)$$

White in earlier work [2] proposed an equation similar to (1) for the turbulent range, but with somewhat different constants.

For the laminar range, White [3] proposed that

$$f = C\left(\frac{16}{Re}\right), \qquad (4)$$

where

$$\frac{1}{C} = 1 - \left[1 - \left\{11{\cdot}6\,\frac{(D/d)^{0{\cdot}5}}{Re}\right\}^{0{\cdot}45}\right]^{1/0{\cdot}45}.$$

This equation is stated to hold for a range of Dean number $(Re)\,(d/D)^{0{\cdot}5}$ between 11·6 to 2000; below 11·6, coils and straight tubes give identical results, i.e. $C = 1$.

Isothermal pressure loss tests were also conducted by the present authors, partly as a check on the instrumentation and quality of the coils (e.g. smoothness of inner surface). It has also been found possible to correlate the non-isothermal friction data. The correlation obtained using properties at the film temperature, found satisfactory by Seban and McLaughlin [4], was not adequate for the higher heat fluxes used by the present authors.

1.2 *Heat-transfer data*

Jeschke [5] tested two coils of 6·1 and 18·2 D/d ratio, both having an l/d ratio of about 1140. He cooled air in turbulent flow up to $Re = 150\,000$. Assuming the D/d effect to be linear in the range covered, he proposed the following empirical equation.

$$Nu = \left(0{\cdot}039 + \frac{0{\cdot}138}{D/d}\right)(Re)^{0{\cdot}76}. \qquad (5)$$

which, taking a value of 0·7 for Pr, can be translated into

$$(Nu)\,(Pr)^{-0{\cdot}4} = 0{\cdot}045\left(1 + \frac{3{\cdot}54}{D/d}\right)(Re)^{0{\cdot}76} \qquad (6)$$

However, his experimental technique, rather sketchily described, is suspect in several respects. Being the only work on this subject for many years, Jeschke's results have often been quoted (sometimes incorrectly), but they are best forgotten.

Kirpikov [6] tested four coils with D/d ratios of 10, 13 and 18, two coils of $D/d = 10$ having markedly different l/d ratios (viz. 208 and 115). Steam heating was used on the outside with water as the cooling fluid. Kirpikov made some unspecified allowances for the entry and exit sections; these involved a length coaxial with and a length perpendicular to the coil axis, and two bends, at each end of the coil. The heat-transfer coefficients were obtained using the wall to bulk temperature difference. The present work showed that with steam heating there are marked peripheral variations of wall temperature, due to the way the condensate runs off the coil, and it is difficult to see how Kirpikov could have obtained valid data by his method. The final relation Kirpikov proposed for the range $10^4 < Re < 4{\cdot}5 \times 10^4$ was

$$(Nu)\,(Pr)^{-0{\cdot}4} = 0{\cdot}0456\,(Re)^{0{\cdot}8}\left(\frac{d}{D}\right)^{0{\cdot}21}. \qquad (7)$$

The properties were evaluated at the arithmetic mean of the bulk temperature of the fluid at inlet and outlet. Kirpikov, in his paper, also quoted earlier Russian work by Aronov and Pinajev, which gave results respectively about 10 per cent higher and 8 per cent lower than his own.

Seban and McLaughlin [4] tested two coils of D/d ratio 17 and 104, heating the fluid by passing a current through the tube wall. Water was the fluid used for the turbulent range, and the results can be expressed by

$$(Nu)\,(Pr)^{-0{\cdot}4} = \left(\frac{f}{2}\right)(Re), \qquad (8)$$

with properties evaluated at the mean film temperature. In this equation the friction factor is given by Ito's correlation, equation (2), although instead of f_B Seban and McLaughlin take the friction factor for a straight pipe as

$$f_s = 0{\cdot}046\,(Re)^{-0{\cdot}2}. \tag{9}$$

Combining equations (8), (2) and (9), the result can be put in the following form

$$(Nu)\,(Pr)^{-0{\cdot}4} = 0{\cdot}023\,(Re)^{0{\cdot}85}\left(\frac{d}{D}\right)^{0{\cdot}1}. \tag{10}$$

The authors consider Seban and McLaughlin's results as the most carefully obtained to date, but nevertheless their values scatter considerably about this equation. Incidentally, in drawing support from Jeschke's result, they use an equation which has been incorrectly quoted by McAdams [7].

In view of the different temperatures at which the fluid properties were evaluated, a direct comparison between Kirpikov's results and those of Seban and McLaughlin cannot be made from the data provided. It is clear, however, that they differ widely in the recommended exponent of d/D. The present work with steam heated coils, evaluated in turn with properties at the bulk and film temperature, enables a proper comparison to be made of all the results.

2. DESCRIPTION OF APPARATUS

Figure 1 depicts the essentials of the apparatus. The coil was fixed to the demountable cover (B) of a steam chamber (A) in such a manner that all thermocouples could be checked, and the coil be pressure tested, before the cover was mounted on the chamber. Thermocouple wires passed through a water filled U-tube (C), which also acted as an indicator of the steam pressure in the chamber, to a multi-point two-pole switch, and thence to an ice-junction and Tinsley potentiometer. Both chamber and cover were steam jacketed and lagged, the steam pressure being maintained a few inches of water above atmospheric to avoid air leakage, and the temperature a few degrees above the saturation value. The rate of water flow was measured using weigh tanks, and the condensate flow by collecting and weighing on an accurate balance. The water was fed to the coil via a water softening plant, and isothermal pressure loss tests were carried out both before and after hot runs to check that there had been no deposit.

Considerable trouble was taken to produce coils with a minimum ovality of bore, and to determine the mean diameter of the bore accurately. The coils were constructed of copper tube, nominally 0·5 in. o.d. × 16 swg, annealed in an inert atmosphere to avoid internal scale. They were filled with "Cerrobend" before bending, and this was washed out with hot water and steam—not acid—to preserve the smoothness of the inner surface. Samples cut from each end of the tube were measured for outer and inner diameter, and

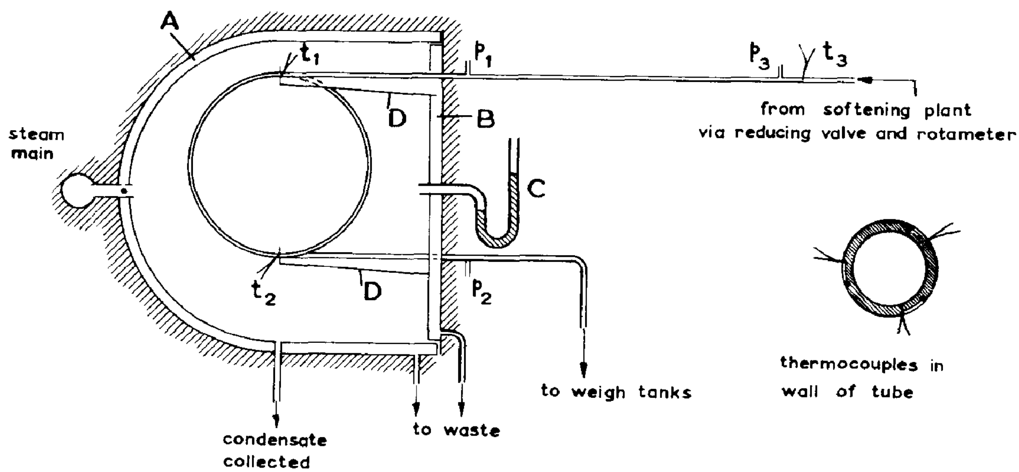

FIG. 1. Apparatus.

wall thickness, the results being compared for consistency. The manufactured coil was then checked along its length for ovality by measuring outer diameters for several diametral positions; the ratio of major to minor axis was always less than 1·006. Comparison of the calculated volume of the coil with the measured quantity of water contained gave agreement to better than 0·5 per cent of the diameter, with the water volume always being less than the calculated value as would be expected. The accurate dimensions of the three coils are given in Table 1.

Table 1

Coil no.		1	2	3
Number of turns n		8·5	6·5	4·5
Mean diameter D	in	4·005	4·927	7·474
Bore d	in	0·3734	0·3723	0·3723
D/d*		10·80	13·30	20·12
length $l = \pi Dn$*	ft	8·975	8·425	8·823
l/d		288	272	284
pitch P	in	1·5	1·5	1·5
inner surface area A	ft^2	0·8773	0·8212	0·8599

* When calculating D/d and l, D was taken, not as the true mean coil diameter, but as $(1/\pi)\sqrt{[P^2 + (\pi D)^2]}$, thus allowing for the obliquity of the helix.

A straight approach length of over 180 tube diameters was used. After 50 diameters, to obtain a fully developed velocity profile, pressure tappings p_3 and p_1 (Fig. 1) were incorporated to provide a straight test length of 133 diameters. The friction factor in this test length always agreed with the Blasius value within 1 per cent. Since the pressure drop is inversely proportional to $d^{5\cdot25}$, this gives added confidence in the determination of the bore and in the assumption that the copper tube had a smooth inner surface.

Allowance had to be made for additional short straight lengths between p_1 and t_1, and t_2 and p_2, when calculating the net friction factor for the coil between the thermocouple stations t_1 and t_2. No allowance was necessary for the condensate over these sections because the shutes (D) carried the condensate into the steam jacket and thence to waste.

All thermocouples were of calibrated copper–constantan wire, enamelled and glass insulated. For the water temperatures, t_1 and t_2, a butt-jointed thermocouple of 30 swg was placed diametrically across the flow at inlet and outlet, and the thermocouple circuit was such that the temperature difference was measured directly, as well as the temperature t_2 at exit. The temperature of the water t_3 at inlet to the straight entrance length was also measured by a thermocouple. Three 36 swg thermocouples were soldered to the wall of the tube at several stations along the coil in the manner shown in Fig. 1, to obtain an idea of the circumferential variation of temperature in the tube wall, and also to enable a correction to be applied to t_1 and t_2 as will be explained later. The variation was in fact so large that a mean wall temperature would have little meaning for the purpose of determining the internal heat-transfer coefficient (see criticism of Kirpikov's work in the Introduction).

3. EVALUATION OF RESULTS

The results obtained from the authors' experiments are discussed in the following subsections, dealing respectively with isothermal pressure losses, heat-transfer data, and non-isothermal pressure losses.

3.1 *Friction factor in isothermal flow*

The experiments on isothermal pressure losses were carried out mainly to verify that the coils were smooth and well formed; for this purpose pressure losses are much more sensitive than heat-transfer data, and even moderate agreement with other pressure loss data would have made the coils suitable for the heat-transfer work. Nevertheless, much of the previous work on pressure loss was carried out with relatively large D/d ratios and often with single turns, so that the authors' results can be regarded as adding something useful to existing data.

The range of Reynolds number covered was 3×10^3 to 5×10^4, with water at mains temperature. Table 2 sets out the corresponding ranges of $(Re)(d/D)^{0\cdot5}$ and $(Re)(d/D)^2$; the first group, called the Dean number, emerges from Dean's theoretical work on laminar flow, and the latter is found to be characteristic of turbulent flow according to White and Ito. It is

Table 2

Coil no.	1	2	3
D/d	10·8	13·3	20·1
$(Re)_{crit}$ from (3)	9300	8700	7600
$(Re)\ (d/D)^{0.5}$ up to transition	910–2800	820–2400	670–1700
$(Re)\,(d/D)^2$ above transition	80–430	50–280	20–120

apparent that coils 1 and 2 yield values of the Dean number slightly outside the range for which equation (4) is quoted. The results are shown in Fig. 2. Most are for turbulent flow, but a few results were obtained in the laminar regime for the purpose of determining the critical Reynolds number. Curves labelled White and Ito have been plotted for the three D/d ratios used, and the accepted curves for a straight pipe have been included for comparison.

In the turbulent regime it will be seen that for coils 2 and 3 the results agree with Ito's equations within 1·5 per cent, and that for the tightest coil Ito's equations appear to underestimate the friction factor by about 3 per cent. In assessing the results, and the non-isothermal pressure loss data in Section 3.3, the following point should be borne in mind. Allowance was made for the pressure loss in the short inlet and outlet lengths and due to the presence of the thermocouples t_1 and t_2 as a result of separate experiments on a straight tube. It is to be expected that the allowance at the exit will be somewhat too low because of secondary flow effects carried over from the coil. In view of this, Ito's correlations can be regarded as very satisfactory.

A sharp discontinuity at transition is obtained when the results (including those for laminar flow) are expressed by plotting the ratio of coil friction factor f to the Blasius factor f_B, against Reynolds number, as in Fig. 3. The values of $(Re)_{crit}$ obtained from equation (3) are shown by short vertical lines, and it can be seen that the experimental results support Ito's criterion.

3.2 *Heat-transfer results*

The rate of heat transfer Q was obtained in two ways: (i) from the product of mass flow of water and temperature rise, and (ii) from the product of mass flow of condensate and the specific enthalpy change $h_{fg} + c_{p\mathrm{sup}}\,(t_{\mathrm{sup}} - t_g)$. The resulting energy balance was never in error by more than ± 3 per cent and for most of the tests in the turbulent range it was better than $\pm 1{\cdot}5$ per cent. The average value of Q was

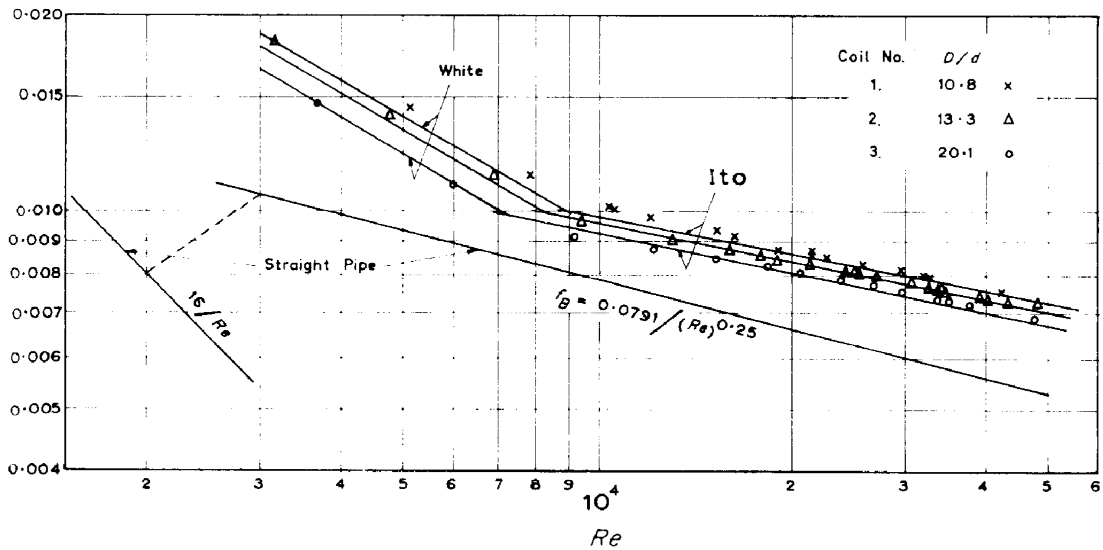

FIG. 2. Isothermal friction factors.

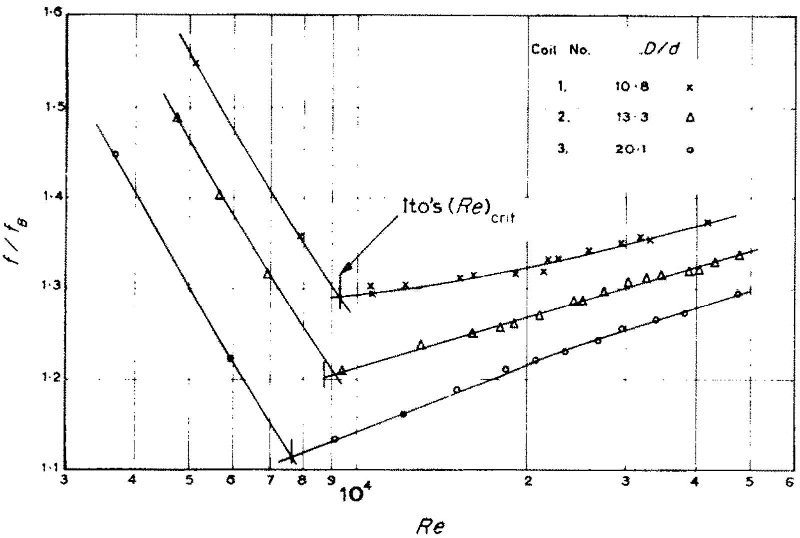

FIG. 3. Critical Reynolds number.

used when determining the heat-transfer coefficient.

The energy balance provided an adequate check on the flow measuring equipment (e.g. on the means of dealing with evaporation during the period of collection of hot water or condensate), and on the method of evaluating the bulk temperature rise of the water. This latter point needs amplification.

The water temperature measured by a thermocouple located at the axis of the pipe will be less than the bulk temperature due to the temperature profile. An estimate of the bulk value can be obtained, however, using the curves in reference 8 which express $(t_w - t_b)/(t_w - t_r)$ as a function of Re and Pr. Although the bulk temperature found in this way will not be accurate when there is a circumferential temperature variation in the pipe wall or when the velocity and temperature profiles have been distorted by secondary flow, the energy balance showed that the resultant error was small.

Having obtained a value for the rate of heat transfer Q, it might be supposed that it would be a simple matter to calculate the internal heat-transfer coefficient h directly from the equation

$$Q = hA\,(t_w - t_b).$$

However, the non-uniformity of t_w invalidates this approach. The authors adopted the following method, due to Wilson [9], which was used successfully by Kreith and Margolis [10]. The overall heat-transfer coefficient U was first found from

$$Q = U \log \text{mean}\, \Delta t,$$

where

$$\log \text{mean}\, \Delta t = \frac{(t_g - t_{b1}) - (t_g - t_{b2})}{\ln \dfrac{(t_g - t_{b1})}{(t_g - t_{b2})}}.$$

The reciprocal of U, i.e. the overall thermal resistance for the whole coil, was then plotted against the reciprocal of the mean fluid velocity u_b raised to the power of 0·8, to yield a straight line whose zero intercept can be interpreted as the thermal resistance of the wall and steam-side taken together. The resistance on the water-side was found by difference. Finally, its reciprocal, hA, when divided by the internal surface area of the coil, yielded the internal heat-transfer coefficient h.

The results are presented in the usual way, by plotting $(Nu)\,(Pr)^{-0\cdot4}$ against Re on logarithmic scales. For comparison with Kirpikov's results, the dimensionless groups have been

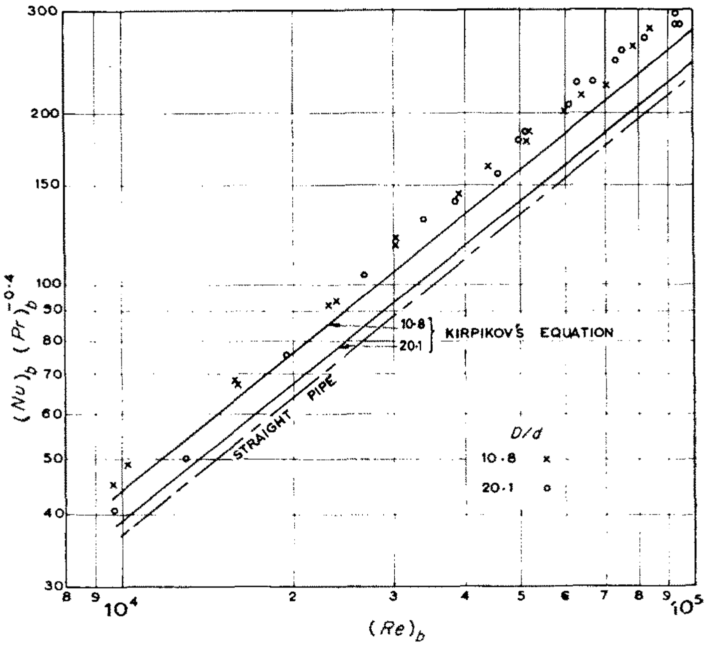

FIG. 4. Heat-transfer results—properties at bulk temperature.

calculated with the properties evaluated at the arithmetic mean of the bulk temperatures of the fluid at inlet and outlet; the resulting graph is shown in Fig. 4. Figure 5 shows the results recalculated with properties taken at the mean film temperature for comparison with the results of Seban and McLaughlin. The suffixes b and f denote bulk and film temperature respectively. Since there was no discernible difference between the results for the two coils having D/d values of 10·8 and 20·1, the heat-transfer experiments with the coil $D/d = 13{\cdot}3$ were curtailed.

The two full lines in Fig. 4 are the result of evaluating equation (7), Kirpikov's equation, for the D/d ratios of 10·8 and 20·1; the chain dotted line refers to a straight pipe and was obtained from

$$(Nu)_b = 0{\cdot}023\,(Re)_b^{0{\cdot}8}\,(Pr)_b^{0{\cdot}4}.$$

In Fig. 5 the dotted line represents the results for Seban and McLaughlin's coil of $D/d = 17$. It is evident that the present results are more than 10 per cent higher than Kirpikov's and about 10 per cent lower than those of Seban and McLaughlin. It is known from heat transfer in straight pipes, that variations of boundary conditions (e.g. constant heat flux, etc.) have only a small effect on Nusselt number in turbulent flow, with the possible exception of liquid metals. Thus the results obtained with steam heating can be fairly compared with those obtained using electrical dissipation. There seems little doubt that the Russian results are too low, and certainly that the exponent of d/D, viz. 0·21, is much too large. It is obvious that further work on coils of D/d between 30 and 100 is required to establish this exponent precisely, but it will be of doubtful value unless greater accuracy can be assured. Accepting Seban and McLaughlin's exponent of 0·1, the present results can be described by the equation

$$(Nu)_f = 0{\cdot}021\,(Re)_f^{0{\cdot}85}\,(Pr)_f^{0{\cdot}4}\,(d/D)^{0{\cdot}1}. \quad (11)$$

The two full lines in Fig. 5 represent this equation for the D/d ratios of 10·8 and 20·1, and it can be seen that the experimental results are within ± 10 per cent of these curves. The results definitely support a Reynolds number exponent of 0·85 in preference to 0·8; the difference

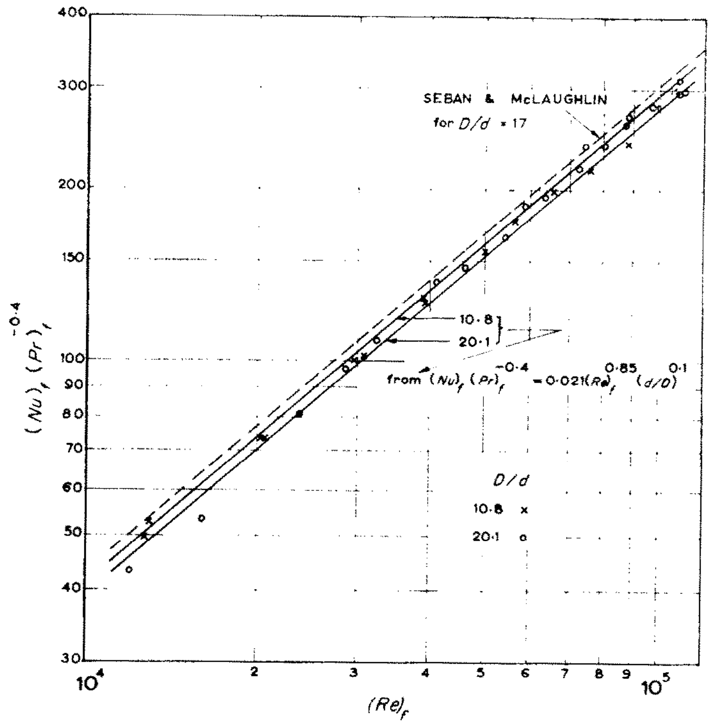

FIG. 5. Heat-transfer results—properties at film temperature.

between the authors' results and those of Seban and McLaughlin lies in the constant 0·021.

In view of the doubtful significance of the mean wall, and hence film, temperature, as well as for convenience of calculation, it is preferable to quote an equation in terms of $(Nu)_b$, $(Re)_b$ and $(Pr)_b$. The results in Fig. 4 might in fact be expressed by

$$(Nu)_b = 0{\cdot}023\,(Re)_b^{0{\cdot}85}\,(Pr)_b^{0{\cdot}4}\,(d/D)^{0{\cdot}1}. \quad (12)$$

In putting forward equations (11) and (12) it must be emphasized that the results presented here make no contribution to the determination of the exponent of Pr for coils.

Maintaining a constant steam-side resistance during a series of tests, upon which the validity of the Wilson method depends, presented considerable difficulty. A drop-wise promoter was used in some of the test runs to reduce the steam-side resistance and hence its significance in the deduction of h, but even so the authors estimate that their results are only accurate to within ± 10 per cent. This includes errors due to the margin of uncertainty in taking the value of the intercept from the $1/U$ vs. $1/u_b^{0{\cdot}8}$ plot and in using 0·8 as the power of u_b in this plot. It is clear from the discussion in Ref. 4, of the results obtained with a comparable coil ($D/d = 17$), that electrical dissipation in the pipe wall provides results which are no more accurate than does steam heating: a reliable method has yet to be found.

3.3 *Friction factor in non-isothermal flow*

The friction factor f has been plotted against $(Re)_b$ in Fig. 6, the values for $D/d = 10{\cdot}8$ being only slightly greater than those for $D/d = 20{\cdot}1$. The heat fluxes were different for the two coils, however, and it is known that the friction factor is sensitive to changes in the velocity profile caused by a temperature gradient. The heat fluxes in the present tests ranged from 40 000 to 380 000 Btu/h ft² for the coil of $D/d = 10{\cdot}8$, and from 35 000 to 310 000 Btu/h ft² for $D/d = 20{\cdot}1$. The corresponding ranges of mean wall to fluid bulk temperature difference were 52 to 66 degF and 37 to 54 degF, and the values of $(Pr)_b/(Pr)_w$ were therefore different for the two coils. Mikheev [11] suggested that non-isothermal friction results for water in a straight pipe can be predicted from isothermal results by multiplying the isothermal friction factor by $[(Pr)_b/(Pr)_w]^{-1/3}$. The two full lines in Fig. 6, labelled

$$f_{\text{iso}} = f\,[(Pr)_b/(Pr)_w]^{1/3},$$

are the result of applying the reverse procedure to the non-isothermal coil friction factors. Not only does this separate the results for the two coils, but, except at the higher Reynolds numbers, it provides a good correlation with the cold loss as calculated from Ito's equations and shown by the dotted lines.

There is of course no reason why the exponent of 1/3 should apply to a coil—nor that it should be independent of Reynolds number. For practical purposes, however, it is clear that a reasonable estimate of the coil friction factor in the range $10^4 < (Re)_b < 6 \times 10^4$ can be obtained by using

$$f = f_{\text{Ito}}\,[(Pr)_b/(Pr)_w]^{-1/3}, \qquad (13)$$

where f_{Ito} is given by equation (1) or (2).

Seban and McLaughlin's results, obtained

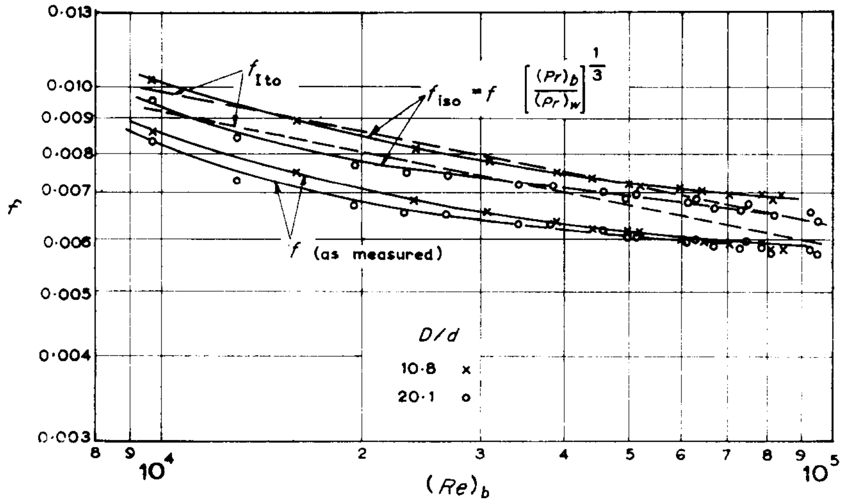

FIG. 6. Friction factors for non-isothermal flow versus $(Re)_b$.

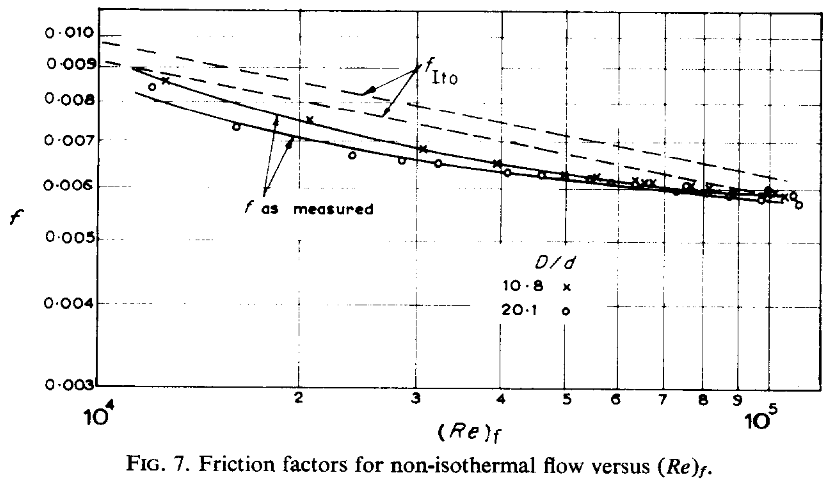

FIG. 7. Friction factors for non-isothermal flow versus $(Re)_f$.

with relatively low heat fluxes, were plotted against $(Re)_f$. Although slightly lower than the cold-loss results, they were deemed sufficiently near for these authors to be satisfied with the correlation on the basis of properties at the film temperature. It is evident from Fig. 7, which shows the present results plotted in the same way, that when higher heat fluxes are involved the use of a film temperature does not take sufficient account of the effect of the temperature gradient.

4. CONCLUSIONS

It appears that expressions for the pressure loss in coils with isothermal turbulent flow are well established, and that over a limited range of Reynolds number the loss with non-isothermal flow of water can be predicted with the aid of the multiplying factor $[(Pr)_b/(Pr)_w]^{-1/3}$.

The present heat-transfer results fall between Kirpikov's and those of Seban and McLaughlin but, as indicated in the proposed equation (12), they support the Reynolds number exponent 0·85 and d/D exponent 0·1 suggested by the latter. More work is required before our knowledge of heat transfer in coils approaches that provided by the extensive data relating to flow in a straight pipe. For this work to be worthwhile, however, it is essential that a better experimental technique be devised.

REFERENCES

1. H. Ito, Friction factors for turbulent flow in curved pipes, *Trans. Amer. Soc. Mech. Engrs* D **81**, 123–132 (1959).
2. C. M. White, Fluid friction and its relation to heat transfer, *Trans. Inst. chem. Engrs, Lond.* **10**, 66–80 (1932).
3. C. M. White, Streamline flow through curved pipes, *Proc. Roy. Soc.* A **123**, 645–663 (1929).
4. R. A. Seban and E. F. McLaughlin, Heat transfer in tube coils with laminar and turbulent flow, *Int. J. Heat Mass Transfer* **6**, 387–395 (1963).
5. D. Jeschke, Heat transfer and pressure loss in coiled pipes, *Ergaenzungsheft Z. Ver. dtsch. Ing.* **69**, 24–28 (1925).
6. A. V. Kirpikov, Heat transfer in helically coiled pipes, *Trudi Moskov. Inst. Khim. Mashinojtrojenija* **12**, 43–56 (1957).
7. W. H. McAdams, *Heat Transmission*, p. 228. McGraw-Hill, New York (1954).
8. G. F. C. Rogers and Y. R. Mayhew, Evaluation of bulk velocity and temperature for turbulent flow in tubes, *Int. J. Heat Mass Transfer* **1**, 55–67 (1960).
9. E. E. Wilson, A basic for rational design of heat transfer apparatus, *Trans. Amer. Soc. Mech. Engrs* **37**, 47–82 (1915).
10. F. Kreith and D. Margolis, Heat transfer and friction in swirling turbulent flow, *Heat Transfer and Fluid Mechanics Institute, California*, 126–142 (1958).
11. M. A. Mikheev, *Elements of Heat Transfer*, p. 292. Gosudarstvennoje. Energetitcheskoje Izdatelstvo, Moscow (1947).

Résumé—Des résultats expérimentaux pour le transport de chaleur par convection forcée et les coefficients de perte de charge, obtenus avec un écoulement d'eau à travers des serpentins chauffés par le da vapeur d'eau, sont rapportés et comparés avec les résultats partiels disponibles actuellement. Les équations existantes pour less coefficients de perte de charge isothermes dans des serpentins à parois lisses sont jugées satisfaisantes. Les coefficients de perte de charge non-isothermes et les coefficients de transport de chaleur peuvent être estimés à partir des équations proposées pour l'établissement de projets, mais les résultats ne peuvent pas encore prétendre à la même validité que ceux pour les tuyaux rectilignes.

Zusammenfassung—Es werden Versuchsergebnisse für den Wärmeübergang und die Reibungsbeiwerte bei Zwangskonvektion von Wasser in dampfbeheizten Spiralrohren angegeben und mit den wenigen, bis jetzt bekannten Ergebnissen verglichen. Bestehende Gleichungen für isotherme Reibungsbeiwerte erscheinen genügend genau. Nichtisotherme Reibungsbeiwerte und Wärmeübergangskoeffizienten können mit den vorgeschlagenen Gleichungen für Konstruktionszwecke abgeschätzt werden. Sie besitzen aber nicht dieselbe Genauigkeit wie diejenigen gerader Rohre.

Аннотация—Излагаются результаты опытов, а также коэффициенты трения и теплообмена, полученные при течении воды в нагреваемых паром спиральных трубах. Приводится сопоставление с имеющимся ограниченным числом уже известных результатов. Оказывается, что существующие уравнения для коэффициентов трения при изотермическом течении в спиралях с гладкими стенками являются вполне удовлетворительными.

Для расчёта коэффициентов трения и теплообмена в неизотермических условиях можно использовать предложенные уравнения, но нельзя считать, что результаты будут настолько же справедливыми, как и для прямых трубок.