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8. ABSTRACT

Friction loss tests were carried out on a 287 mm I.D. cement mortar lined pipeline. The test length was 81 m long, consisting of 9 × 9 m lengths of flanged pipe, preceded by a 36 m long approach section. The steel pipes were centrifugally lined.

The results showed that over the range of Reynolds numbers from 4.7×10^4 to 8.1×10^5 achieved during the tests, Darcy friction factors for the pipe followed the Colebrook-White transition function with a roughness k of .01 mm.

9. DISTRIBUTION STATEMENT

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1. Introduction

1.1 Aim of Tests

Tests were carried out at the State Rivers and Water Supply Commission hydraulic laboratory at Werribee, Victoria, to ascertain the hydraulic roughness of steel pipes centrifugally lined with a cement mortar manufactured by Steel Mains Pty.Ltd. The Company felt that pipes of this type were unfairly disadvantaged by the use by pipeline designers of unduly high roughness values. Data from laboratory testing of pipes were sought to allow flow resistance charts to be based on the results of tests on new pipes as is done by the manufacturers of other types of pipe.

The author of this report collaborated in the testing using the facilities of the State Rivers and Water Supply laboratory as an alternative to setting up a sufficiently long test bed at the University of New South Wales Water Research Laboratory.

1.2 Friction Equations and Pipe Roughness

The hydraulic design of a pipeline requires the selection of a valid pipe friction formula and a measure of pipe roughness to insert in that formula. Both theoretical and experimental work has demonstrated that for circular pipes flowing full of water the general relation which covers the full range of flow conditions from laminar to rough-wall turbulent is the equation

$$S = H_{f/1} = \frac{f\,V^2}{2gd} = \frac{8fQ^2}{\pi^2 g d^5} \qquad (1)$$

where

S is the slope of the energy line

H_f is the friction head loss (m)

l is the pipe length (m)

V is the mean flow velocity ($m.s^{-1}$)

g is gravitational acceleration ($m.s^{-2}$)

d is the pipe diameter (m)

f is the "friction factor" which is a function of Reynolds number $\mathbb{R}(= \frac{Vd}{\nu}$ where ν is kinematic viscosity ($m^2.s^{-1}$)) and relative roughness ($\frac{k}{d}$ where k is a linear measure of wall roughness, usually expressed as an "equivalent sand grain roughness". This is the diameter of uniform sand grain roughening which gives the same friction factor as the actual roughness for fully rough tubulent flow).

Equation (1) is generally known in Australia as the Darcy equation.

The function relating f, $\mathbb{R}$ and $\frac{k}{d}$ is complex. It varies with the type of roughness. For pipes which have a relatively random distribution of size and spacing of roughness projections the Colebrook-White equation

$$\frac{1}{\sqrt{f}} = -2 \log \left(\frac{k}{3.7d} + \frac{2.51}{\mathbf{R}\sqrt{f}}\right) \qquad (2)$$

has been found to give a satisfactory fit to both laboratory and field experimental data. Friction factors for pipes which have regular roughness projections such as sand grains or corrugations do not fit this equation.

Figure 1 shows typical f - $\mathbf{R}$ relations for turbulent flow in smooth pipes, pipes with random roughness and pipes which have been roughened artificially by uniform sand grains. The rough pipes have the same friction factor at high Reynolds numbers (fully rough turbulent flow) but different transition curves for lower Reynolds numbers. It is in this transition range that most water supply pipes operate in service.

Prior to the introduction of the Darcy and Colebrook-White equations, several empirical equations relating velocity V, hydraulic radius R (= $\frac{d}{4}$ for a circular pipe) and friction slope $\frac{H_f}{l}$ were proposed by Hazen and Williams, Manning, Scobey and others. These equations were founded on limited experimental data and had no theoretical foundation. Early difficulty in solving the Colebrook-White equation for f and entrenched conservative attitudes of some engineers has led to the retention of the empirical equations despite the fact that electronic calculators and computers have now simplified the iterative solution of the Colebrook White equation and the preparation of chart solutions.

In this report the friction factor-Reynolds number relation has been chosen as the basic frame of reference for analysis of the experimental results. It is recommended that application of the results should also be based on this relation. However, at the client's request, equivalent values of coefficients in the empirical exponential equations of Manning, Hazen-Williams and Scobey have been provided and a graphical comparison of the fit of the equations is presented.

2. Pipes Tested

The pipes tested were 324 mm O.D. x 6mm wall thickness steel pipes lined with cement mortar to give a mean inside diameter between 285mm and 287mm. A diameter of 287mm has been used in the calculation of friction factors and Reynolds numbers. Diameter variations measured in the pipes were up to ± 4mm from the mean. Pipe lengths were 9m.

The lining process used involves the deposition of mortar inside the pipe from the end of a tubular "spear" supplied by a positive displacement mortar pump. The "spear" is moved at a constant rate through the pipe being lined to give uniform deposition along its length. The pipe is then rotated on rubber belts to distribute the mortar around the circumference. A higher rotational speed compacts the mortar to form a lining of uniform thickness. The mortar is compacted by the radial pressure gradient developed and the vibration of the pipe on the belts. The lining is trimmed by hand for a short distance at the ends of the pipe.

A large number of lined pipes, including the tested pipes, was inspected at the client's factory to check the surface finish of the lining. The surface was found to be generally smooth with a cement-rich skin. Smooth corrugations of variable wave length and amplitude were evident in all pipes. There were also well distributed sharp dry mortar particles sticking out of the surface. The corrugations probably result from resonant vibrations set up by the rubber belts while the mortar particles come from previous mixes.

The pipes tested were of fair average surface finish and were considered to be representative of all the pipes inspected.

3. Test Pipeline

3.1 General Layout

A 117m long straight pipeline consisting of 13 x 9m lengths of flanged pipe was set up by the State Rivers and Water Supply Commission on concrete blocks on level ground. Bends were situated at both ends of the line, the upstream one leading to the pumps and the downstream one diverting water from a control valve at the end of the pipeline into a channel leading to the measuring tanks. An approach length of 35.5m preceded the test length of 81m between upstream and downstream pressure tappings. An intermediate pressure tapping point was established 45m from the upstream tapping point to allow linearity of the head loss with length to be checked.

Provision was made for a flow straightener made of a bank of parallel tubes to be inserted at the upstream end of the approach length. However, it was not used as there was no problem with pressure fluctuations in the pipeline and its use would have reduced the maximum flow rate available.

3.2 Pressure Tappings

At each end of the upstream, intermediate and downstream tapping points, four tappings were located at uniform spacing around the circumference of the pipe.

Prior to the lining of the pipes, ½ inch B.S.P. steel sockets were welded to the pipes and greased threaded plugs, each with a 3 mm diameter tapping hole, were screwed in to protrude into the pipes an amount equal to the design thickness of the lining. After lining, the plugs were adjusted to bring the tapping holes flush with the lining surface. Plastic tubes of equal length (approximately 1m) were led from tails screwed into the tapping sockets to small averaging manifolds (one at each tapping point) from which single tubes were led to a three tube vertical water manometer. The manometer thus registered the average pressure at each of the tapping points.

4. Pumping and Flow Measuring Equipment

The upstream end of the pipeline was connected to the rising main from the laboratory's pumps which drew water from a large sump. A balancing tank connected to the pump discharge manifold by a separate pipeline helped reduce pressure fluctuations in the test pipeline.

The return channel from the discharge end of the pipeline terminated in a Y branch which allowed the water to be diverted into the sump or measuring tanks as required.

The laboratory's volume measuring tanks have been accredited by N.A.T.A. They consist of two open tanks below ground level connected by a gate. Each tank has a calibrated tape linked to a float to measure the water level which is common when the connecting gate is open.

5. Test Procedure

With the pumps running, the control valve at the discharge end of the test pipeline was adjusted to set the flow and water was returned via the channel to the sump until steady conditions had been achieved. Water was then diverted to the measuring tank until a sufficient volume had been collected to give an accurate measure of the flow rate. The time was measured by stopwatch. The change in water level was measured on the float tapes in both of the interconnected measuring tanks, thus allowing a check on gross error to be made. Readings on all three columns of the manometer were taken several times during the test and an average value determined. The proportionality of head loss with length between the upstream and intermediate and intermediate and downstream tapping points was checked by calculating the head loss per unit length over the two sections. The uniformity of pressure at the four tappings at each tapping point was checked by injecting dye into the tubes connecting the tappings to the averaging manifold and observing the flow velocities in the tubes.

Tests were carried out from the maximum discharge available to the smallest discharge at which a reading could be obtained from the manometer.

Two tests were generally run at each discharge setting to ensure that flow in the return channel had stabilised before the first test was started. Since the channel was relatively steep there was no significant problem with channel storage effects caused by operating the diversion gates at the sump and measuring tank.

6. Results

Table 1 gives the measured head loss and discharge values together with calculated Darcy friction factors and Reynolds numbers. Where the discharge is marked with a * it is the mean of two volumetric discharge measurements at the same control valve setting. The maximum variation between two such readings was 2% at a discharge of 0.022 m^3s^{-1}. At higher discharges the variation was less than 1% and averaged approximately ½%.

No problem was encountered with excessive variation of pressure between tappings at any tapping point. Nor was there any problem with difference between hydraulic gradients between the upstream and intermediate and intermediate and downstream tapping points. For the higher flow rates the gradient in the downstream portion of the test length was 5% greater than that in the upstream portion. For lower flows the difference decreased until, at the lowest flow, measurement error prevented a meaningful comparison. Friction factors are plotted against Reynolds numbers in Figure 2 to give a picture of the degree of scatter and the conformity to the Colebrook-White equation. Curves derived from this equation for $\frac{k}{d}$ values of .000 01, .000 05, .000 1, .000 2 and for a k value of .010 mm for a 287 mm pipe (k/d = .000 035) are plotted on the figure for comparison.

7. Comparison of Results with Predictions of Exponential Equations

Figure 3 shows a plot of hydraulic gradients versus velocities calculated from the test results and predicted by the exponential equations of Manning, Hazen-Williams and Scobey. The equations, converted to S.I. units, are given in Appendix A. It can be seen that none of the recommended coefficients for concrete used in these equations gives a fit comparable with that of the Colebrook-White equation. Table 2, which gives the values of the coefficients required to fit the exponential equations to test results at the extremes and an intermediate point in the range of test discharges, demonstrates that no one coefficient covers the range. The coefficients required also fall outside the range of recommended values except for the very low velocity of .2 ms^{-1}.

8. Discussion of Results

8.1 Accuracy of Test Results

8.11 Flow Rate Measurement

The areas of the volume tanks are known to an accuracy of ±.3%. Depths of measurement ranged from 164±.5 mm for the lowest flow rate to 1093±.5 mm. Times ranged from 600±.5 seconds for the lowest flow rate to 150±.5 seconds for the highest. The accuracy of discharge measurement thus varied from ±.6% for the lowest flow to ±.4% for the highest.

8.12 Head Loss Measurement

Fluctuating water levels in the manometer tubes were averaged visually with the aid of a transparent plastic T-square which allowed the mean of high and low water levels to be estimated on the steel tape scale. The three manometer columns were located in parallel grooves in the vertical manometer board as close as possible to the common scale.

The accuracy of reading the difference in water levels in the upstream and downstream manometer tubes was estimated to be ±.5mm at the lowest difference of 12mm (±5%), increasing to ±1mm at 30mm, and ± 10 mm at the highest difference of 2078 mm (± .5%). At the lowest flow no surging occurred and the accuracy of the measurement of the manometer level difference should represent the accuracy of measurement of the head difference in the pipeline. However, as the flow rate increased the effects of surging and pressure difference between individual tappings at each tapping point became apparent so the error in estimating the head difference in the pipeline would be greater than the error in measuring the manometer level difference.

8.13 Friction Factors and Reynolds Numbers

Reynolds Number $\mathbb{R} = \frac{Vd}{\nu} = \frac{4Q}{\pi d \nu}$

Error in ν for error in temperature of ± 1°C	=	± 2.5%
Error in d estimated at ± 1.5 mm in 287 mm	=	± .5%
Error in Q	=	± .5%
∴ Error in $\mathbb{R}$	=	± 3.5%

Friction Factor $f = \frac{\pi^2 g d^5 H_f}{8 l Q^2}$

Error in d^5	=	± 2.5%
Error in H_f in pipe (estimated from manometer reading error)	=	± 1% to ± 5%

Error in 1	= < .1%
Error in Q^2	= ± 1%
Error in f	= ±4.5% to ± 8.5%

In rounded figures the errors are:

Reynolds Number	± 4%
Friction Factor	< ± 5% for $\mathbf{R}$ 2×10^5
	± 5% for $\mathbf{R} = 2\text{x}10^5$ increasing to
	± 9% for $\mathbf{R} = 5\text{x}10^4$

8.2 Friction Factors for New Pipes

The results of the tests demonstrate that the Colebrook-White equation accurately predicts friction factors for the pipeline tested if a roughness value k = .01 mm is used. It should be particularly noted that over the range of Reynolds number from $4.7\text{x}10^4$ to $8.1\text{x}10^5$ covered by the tests there was no indication of the type of transition-turbulent curve found by Nikuradse (1) for sand roughened pipes and by Schroder (2) for concrete pipes. It appears that the smooth cement-rich surface caused by the method of distributing and compacting the mortar lining effectively eliminates the sand-grain type roughness which may occur in pipes manufactured by other techniques. When the test pipes were first inspected it was felt that the corrugations on the mortar surface and the projecting particles of dry mortar resulting from previous mortar batching might significantly affect the friction factors. No particular roughness effect which could be attributed to the corrugations or projections was observed over the range of the tests. It appears that the smooth wave form and low amplitude to wave length ratio of the corrugations and the low frequency of occurrence of the projections prevented any significant effect.

For new pipes of the type tested it is very unlikely that any significant deviation from the Colebrook-White transition curve for k = .01 mm will occur in the Reynolds number range up to 10^6 most frequently encountered. However, the Reynolds number at which the type of transition found by Nikuradse and Schroder for "sand roughened" and concrete pipes showed up increased with decreasing pipe roughness. It is therefore possible that for Reynolds numbers greater than 10^6 a transition to a higher relative roughness curve on the f-$\mathbf{R}$ plot could occur. Inspection of Schroder's curves indicates that since no "sand-roughened" type transition has occurred up to a Reynolds number of $8\text{x}10^5$, the maximum value of equivalent sand grain roughness k which the tested pipes would be expected to yield for fully rough turbulent flow would be approximately .06mm, four times the value for Reynolds numbers up to 10^6.

A pipeline designer who wishes to take a conservative approach to predicting head losses for new large diameter pipes or pipes in which velocities are high should either use the modified transition function presented by Schroder or, more simply, adopt a constant Darcy friction factor of 0.014 for Reynolds numbers greater than 5×10^5. The latter procedure would give predicted head losses up to 12% greater than measured in the tests between Reynolds numbers of $5\text{x}10^5$ and $8\text{x}10^5$. Between Reynolds numbers of $8\text{x}10^5$ ($3\ \text{m.s}^{-1}$ in a 2m dia. pipe at 20°C), and $6\text{x}10^6$, calculated head losses would still be greater than those predicted by Schroder's equation for k = .06 mm.

8.3 Friction Factors for Pipes in Service

The roughness of pipes which have been in service for some time may be affected by mechanical or chemical erosion of the surface, biological growths adhering to the surface or encrustation which may be caused by bio-chemical effects.

Growths and encrustations need to be considered regardless of the type of pipe.

Wear of the pipe surface may lead to the removal of the fine surface finish. In the case of the pipes tested, this could lead to a "sand-roughened" type surface. Should this occur, the f-R plot is likely to exhibit the transition characteristics demonstrated by Nikuradse and Schroder at Reynolds numbers above 10^5, with higher ultimate roughness values than those found in the tests described in this report.

Pipeline designers should consider all the factors likely to influence pipe roughness changes, including flow velocity, sediment carried and water quality before adopting "new pipe" roughness for sizing any pipeline.

9. Conclusions

The pipe friction tests described in this report yielded results which followed the Darcy equation with friction factors given by the Colebrook-White equation with a roughness value (k) of .01 mm. The range of Reynolds number covered by the tests was from 4.7×10^4 to 8.1×10^5.

The results do not fit the Manning, Hazen-Williams and Scobey equations using coefficients recommended for concrete. It is recommended that the Darcy equation with friction factors given by the Colebrook-White equation, be used instead of the exponential equations listed above.

The results are for new pipes and allow comparisons to be made with friction head losses in new pipes manufactured from different materials or using different processes. Pipeline designers should assess the possibility of change of roughness in service when estimating friction head losses for design purposes. It is recommended that designers become acquainted with the work of Schroder and his modification of the Colebrook-White transition function if they wish to predict friction losses in pipes exhibiting "sand-roughness" characteristics because of wear of an originally smooth surface.

10. References

1. Nikuradse, J. "Strömungsgesetze in rauhen Rohren". VDI-Forschungsheft Nr.361, Berlin 1933.

2. Schroder, R.C.M., Knauf, D. Über das hydraulische Widerstandsverhalten von Beton-und Stahlbetonrohren im Übergangsbereich.

Table 1: Results of Friction Head Loss Tests

	Discharge Q $m^3.s^{-1}$	Head Loss H_f m	Temperature ^oC	Velocity $m.s^{-1}$	Gradient S	Darcy Friction Factor f	Reynolds No. **R**
	0.1708	1.298		2.64	.01602	0.0129	6.3×10^5
	0.1424	0.961	13	2.20	.01186	0.0138	5.3×10^5
	0.1018	0.512		1.57	.00632	0.0144	3.8×10^5
	0.0212	0.030		0.328	.00037	0.0194	7.8×10^4
*	0.0302	0.056	12	0.467	.00069	0.0178	1.09×10^5
*	0.0776	0.308	12.5	1.20	.00380	0.0149	2.82×10^5
	0.1177	0.699		1.81	.00863	0.0147	4.3×10^5
*	0.1614	1.180	13	2.49	.01457	0.0132	6.0×10^5
*	0.1722	1.313		2.66	.01621	0.0129	6.4×10^5
	0.1803	1.430		2.79	.01765	0.0128	6.8×10^5
	0.2197	2.078	13.5	3.40	.02565	0.0125	8.1×10^5
	0.2143	1.962		3.31	.02422	0.0124	7.9×10^5
	0.1996	1.809		3.09	.02233	0.0132	7.4×10^5
	0.0128	0.012	14	0.198	.00015	0.0213	4.9×10^4
*	0.0223	0.030		0.345	.00037	0.0175	8.7×10^4
*	0.0460	0.117		0.71	.00144	0.0161	1.79×10^5
*	0.0645	0.215		1.00	.00265	0.0150	2.50×10^5
	0.1219	0.701	15	1.88	.00865	0.0137	4.7×10^5
	0.1249	0.723		1.93	.00893	0.0135	4.9×10^5
	0.1409	0.895		2.18	.01105	0.0131	5.5×10^5

* Indicates that the discharge given is the average of two sequential volumetric measurements at the same control valve setting. All other results are for single volumetric discharge measurements.

Table 2: Coefficients Required in Exponential Formulae

Test Results				Coefficients Required in Exponential Formulae		
Q m^3s^{-1}	H_f m	V	S_f	Manning	Hazen-Williams	Scobey
.0128	.012	.198	1.5×10^{-4}	.011	142	.369
.0776	.308	1.20	3.80×10^{-3}	.009	150	.444
.2197	2.078	3.40	2.57×10^{-2}	.008	152	.484
Recommended coefficients for use in exponential formulae for concrete pipes				Steel formed .012 - .014 Smooth .011 - .012	Very smooth 130 Extremely smooth 140	Steel formed .345 Very smooth .370

APPENDIX A

Exponential Pipe Friction Formulae

V	=	mean velocity in pipe
d	=	internal diameter of pipe
d_{ins}	=	internal diameter of pipe in inches
R	=	hydraulic radius = $\frac{d}{4}$ for full pipe
S	=	friction head loss per unit length of pipe
H	=	head loss due to friction = S x pipe length
C	=	coefficient

Subscripts ft/s, ft and m/s, m indicate that feet per second and feet or metres per second and metres are required for velocity and hydraulic radius (or diameter).

Equations are converted from English to S.I. units retaining coefficients unchanged.

1. Hazen-Williams Formula

$$V_{ft/s} = 1.318\ C_1\ R_{ft}^{0.63}\ S^{0.54}$$

$$3.28\ V_{m/s} = 1.318\ C_1\ 3.28^{0.63}\ \frac{d_m^{0.63}}{4^{0.63}}\ S^{0.54}$$

$$\therefore\ V_{m/s} = 0.355\ C_1\ d_m^{0.63}\ S^{0.54}$$

$$\text{or}\quad S = \frac{6.81}{d_m^{1.17}} \left(\frac{V_{m/s}}{C_1}\right)^{1.85}$$

2. Scobey's Formula(for Concrete Pipes)

$$V_{ft/s} = C_s\ H_{ft}^{0.5} d_{ins}^{\ 0.625} \qquad (H_{ft} = \text{head loss in feet over 1000 feet})$$

$$\therefore\quad 3.28\ V_{m/s} = C_s\ (3.28\ H_m)^{0.5}\ (3.28 \times 12 \times d_m)^{0.625}$$

H_m = metres head loss over 1000 feet (1000/3.28 metres)

$$\therefore \quad V_{m/s} = 5.48\ C_s\ H_m^{0.5}\ \ d_m^{.625}$$

$$\therefore \quad H_m = \frac{V_{m/s}^{\ 2}}{30.06 C_s^{\ 2} d_m^{\ 1.25}}$$

$$\therefore \quad S = \frac{V_{m/s}^2}{30.06\ C_s^{\ 2}\ d_m^{\ 1.25}} \times \frac{3.28}{1000}$$

$$\text{i.e.} \quad S = \frac{V_{m/s}^{\ 2}}{9164\ C_s^{\ 2}\ d_m^{\ 1.25}}$$

3. Manning's Equation

$$V_{ft/s} = \frac{1.486\ R_{ft}^{\ 2/3}\ S^{\frac{1}{2}}}{n}$$

$$V_{m/s} = \frac{R_m^{\ 2/3}\ S^{\frac{1}{2}}}{n}$$

$$= \frac{\left(\frac{d_m}{4}\right)^{2/3} S^{\frac{1}{2}}}{n}$$

$$\text{or} \quad S = \frac{6.35\ n^2 V_{m/s}^{\ 2}}{d_m^{\ 4/3}}$$

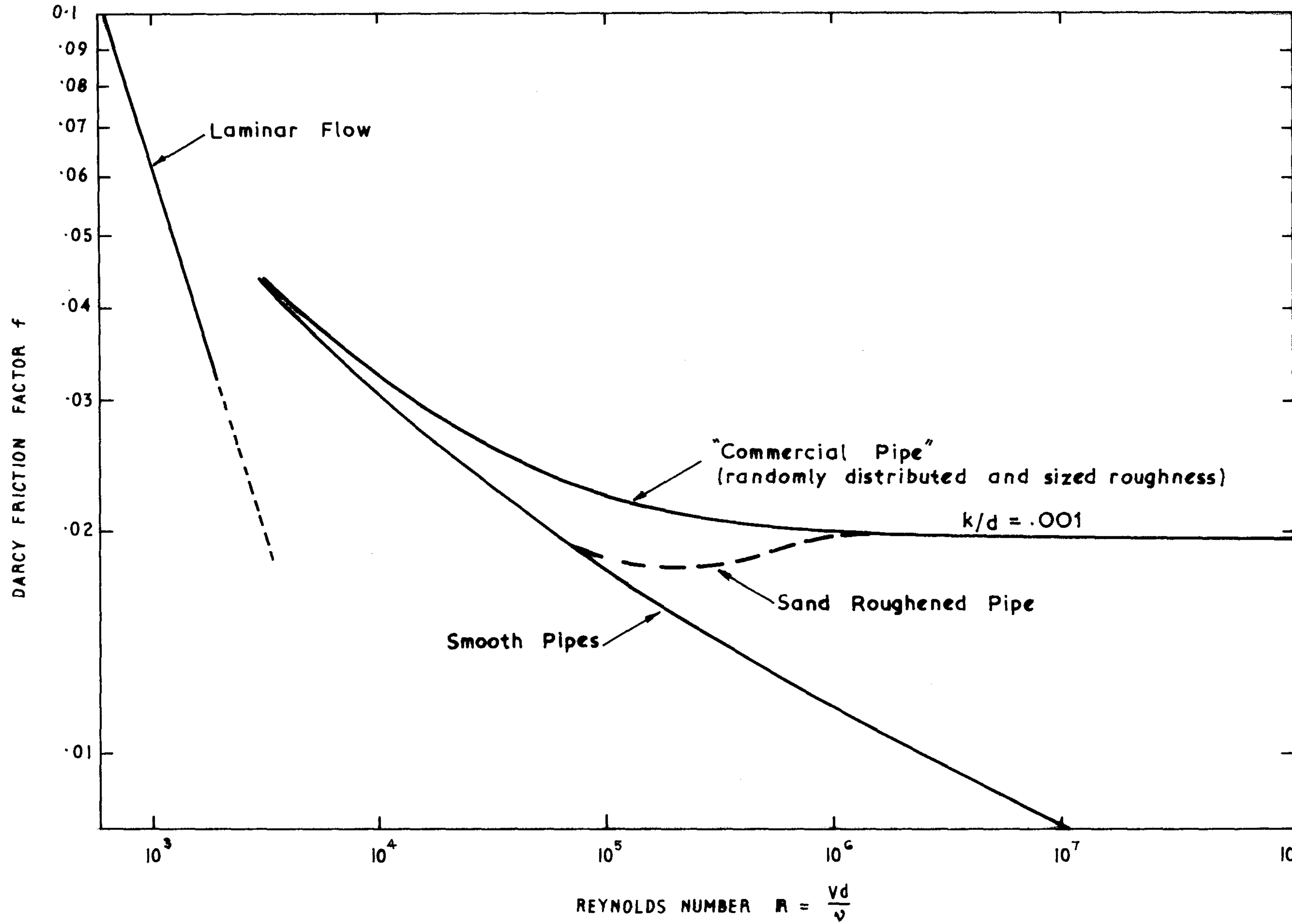

FIGURE 1: TYPICAL FRICTION FACTOR VERSUS REYNOLDS No. CURVES FOR SMOOTH AND ROUGH PIPES

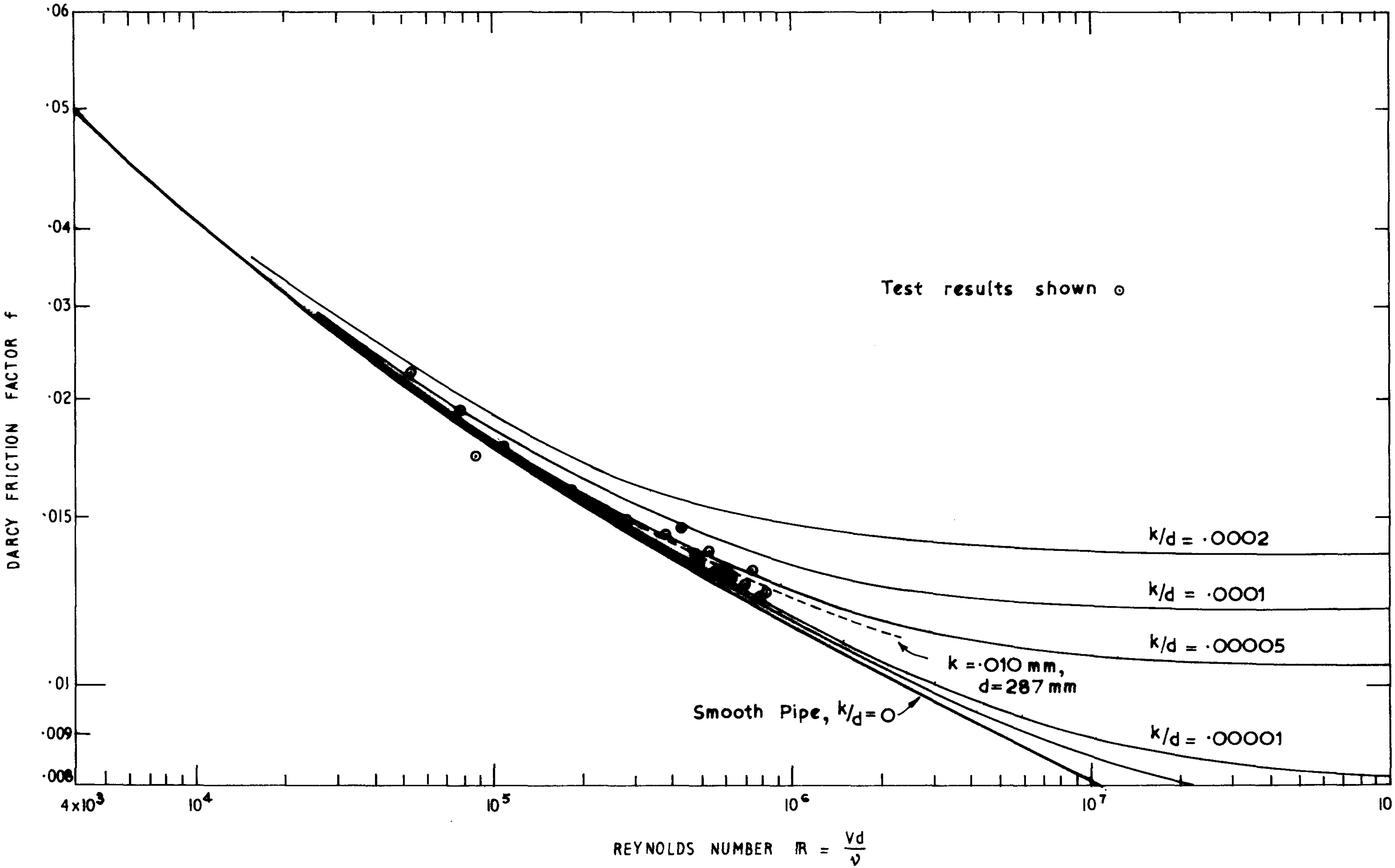

FIGURE 2: FRICTION FACTOR VERSUS REYNOLDS No. PLOT OF TEST RESULTS AND COLEBROOK-WHITE CURVES

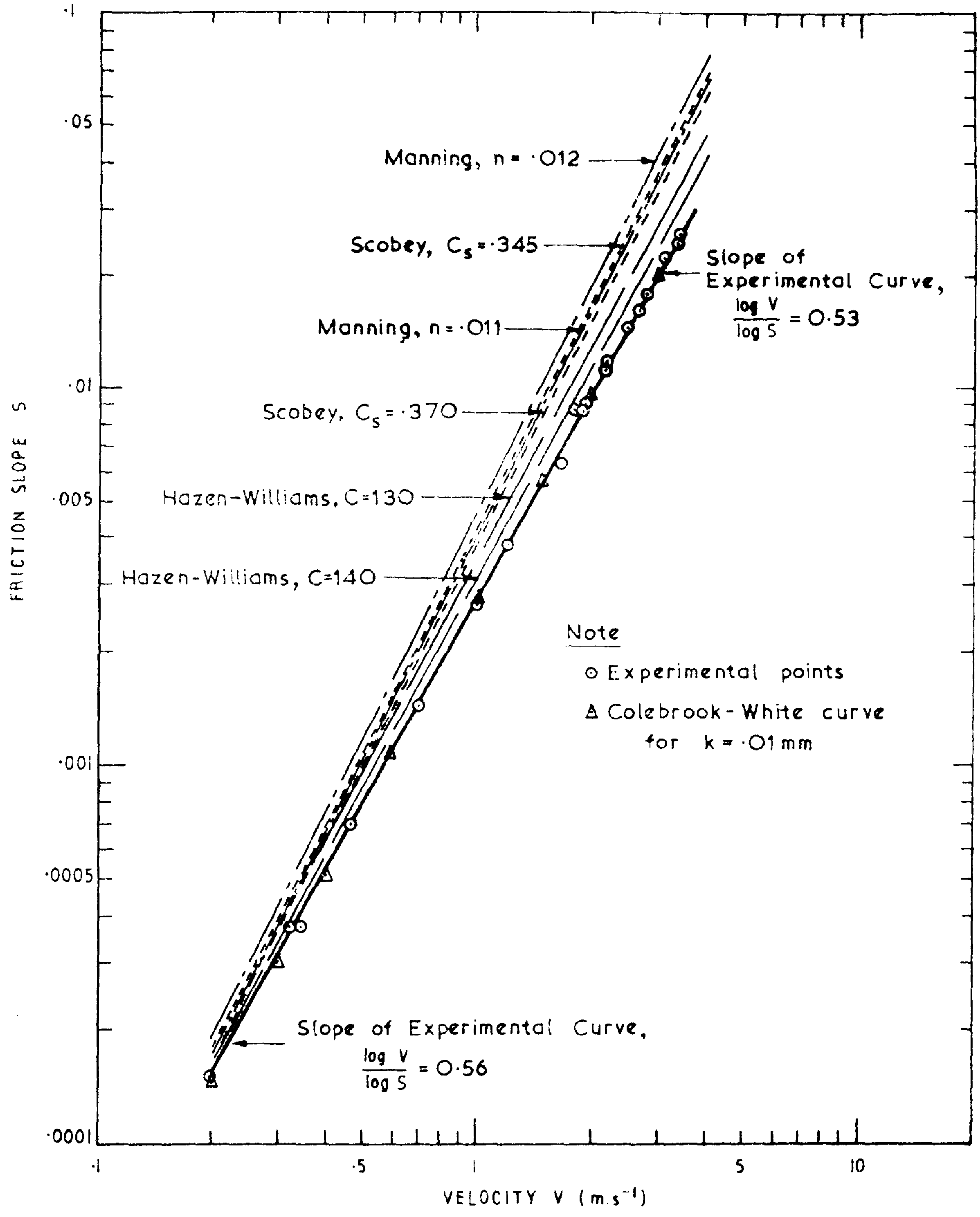

FIGURE 3: COMPARISON BETWEEN ACTUAL FRICTION SLOPES AND PREDICTIONS FROM EXPONENTIAL EQUATIONS