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Measurements of Convective Heat Transfer from a Horizontal Cylinder Rotating in a Pool of Water

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MEASUREMENTS OF CONVECTIVE HEAT TRANSFER FROM A HORIZONTAL CYLINDER ROTATING IN A POOL OF WATER.

Kurt M Becker

Summary:

The present paper deals with measurements of heat transfer from a horizontal cylinder rotating in water. The experimental results have been correlated by the equation

$$Nu = 0.11\, Re^{0.68} \cdot Pr^{0.4}$$

for a range of rotating Reynolds numbers from 1000 to 46000, and Prandtl numbers from 2.2 to 6.4. This equation compares very well with the experimental and theoretical information available for air in published works.

The analogy suggested by Anderson and Saunders between natural convection from a horizontal plate and the present type of flow has been used to predict the Nusselt numbers. Analytical and experimental results have been found to compare very well with each other.

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1.0 Introduction

Convective heat transfer from a horizontal cylinder rotating in air has earlier been studied by several investigators. The purpose of the present paper is to present data obtained with a cylinder rotating in a pool of water, and to show the effects of Prandtl number on this type of heat transfer.

The heat transfer coefficient for the cylinder is assumed to be a function of the following independent variables

$$\alpha = f\,(d,\ g,\ c_p,\ \omega,\ \beta,\ \rho,\ \mu,\ \lambda,\ \theta) \tag{1}$$

Using dimensional analysis equation 1 can be reduced to

$$Nu = f_1\,\left(\frac{\omega d^2 \rho}{2\mu},\ \frac{g d^3 \beta \theta \rho^2}{\mu^2},\ \frac{\mu c_p}{\lambda}\right) \tag{2}$$

or

$$Nu = f_2\,(Re,\ Gr,\ Pr) \tag{3}$$

2.0 Literature Review

The significant quantitative studies found in published works are those of Anderson and Saunders (1), Etemad (2), Dropkin and Carmi (3) and Kays and Bjorklund (4). The present section briefly describes these studies and other information available which is of importance to the subject.

Anderson and Saunders (1) investigated the heat transfer from horizontal cylinders, 1.0, 1.8 and 3.9 ins. in diameter, each 2 ft long rotating in still air, and found that up to a critical value of the Reynolds number, based on surface velocity, the Nusselt number is almost independent of the Reynolds number, and the rate of heat transfer is then mainly determined by the free convection. Using theoretical considerations the critical Reynolds number was found to be equal to

$$Re_{Cr} = 1.09\ Gr^{1/2} \tag{4}$$

Above the critical Reynolds number it was found that the Nusselt number increased with the Reynolds number and that the Grashof number had a negligible effect on the rate of heat transfer.

Anderson and Saunders suggested that the flow set up by the rotating cylinder above the critical Reynolds number is analogous in many respects to the irregular flow which occurs in free convection above a heated horizontal plate facing upwards. Using this analogy they derived the following expression for the heat transfer from a cylinder rotating in still air.

$$Nu = 0.10\ Re^{2/3} \tag{5}$$

This equation compared excellently with the measurements.

Etemad (2) studied experimentally the heat transfer and flow around horizontal cylinders, 2 3/8 and 2 1/2 ins. in diameter, rotating in air. A range of Reynolds numbers from 0 to 65.400 was studied. From interferometric observations he found that the laminar Couette motion broke down at a critical Reynolds number of 900 compared with 1080 computed from the relation established by Anderson and Saunders. The interferometric pictures also showed that the secondary flow above the critical Reynolds number bore some resemblance to the secondary flow between two concentric cylinders, when the inner cylinder was rotated. The latter type of flow has been studied by Taylor (5), Kaye and Elgar (6) and others. Etemad found further that up to a Reynolds number of 14500 the secondary flow remained in steady motion. Above this value the secondary flow broke down and the flow became turbulent. The heat transfer results by Etemad compared excellently with the data of Anderson and Saunders. For Reynolds numbers above 8000, the heat transfer rates were independent of the Grashof number and the following equation correlated the experimental data.

$$\mathrm{Nu} = 0.076\ \mathrm{Re}^{0.70} \tag{6}$$

For Reynolds numbers below 1000 the Nusselt numbers depended almost entirely on the Grashof numbers, and in the intermediate range between 1000 and 8000 both the Grashof and the Reynolds numbers influenced the rate of heat transfer and the following correlation was recommended

$$\mathrm{Nu} = 0.11\left[(0.5\ \mathrm{Re}^2 + \mathrm{Gr}) \cdot \mathrm{Pr}\right]^{0.35} \tag{7}$$

Dropkin and Carmi (3) measured the heat transfer rates from horizontal rotating cylinders to ambient air for Reynolds numbers up to 433.000. The diameters employed were 3.25 and 4.50 in. For Reynolds numbers larger than 15000 they recommended the following equation

$$\mathrm{Nu} = 0.073\ \mathrm{Re}^{0.7}$$

which compares extremely well with the results mentioned earlier. In the region where both rotation and natural convection influenced the heat transfer their data were correlated by the equation

$$\mathrm{Nu} = 0.095\left[0.5\ \mathrm{Re}^2 + \mathrm{Gr}\right]^{0.35}$$

Kays and Bjorklund (4) measured the heat transfer from a horizontal cylinder rotating in air with and without crossflow. In the case of zero crossflow their results compared very well with the investigations previously mentioned. This case was also investigated theoretically by means of the momentum and heat transfer analogy, and it was found that the Nusselt number could be predicted by the equation

$$\mathrm{Nu} = \frac{\mathrm{Re} \cdot \mathrm{Pr} \cdot \sqrt{f/2}}{5\ \mathrm{Pr} + 5\ \ln\left(3\ \mathrm{Pr} + 1\right) + \frac{1}{\sqrt{f/2}} - 12} \tag{9}$$

For estimating the friction coefficient, f, the use of the data by Theodersen and Regier (7) was recommended. In the case of air where Pr = 0.72, the analogy solution agreed very well with the experimental results.

3.0 Description of Apparatus

The details of the rotor is shown in figure 1, and a schematic view of the apparatus is reproduced in figure 2. The electric resistance heated test section consisted of a polished stainless steel tube, 300 mm in length, 10.05 mm in outer diameter and with wall thickness of 0.5 mm. At both ends the tube was silver soldered to copper rods which penetrated to the exterior through seals mounted in the walls of the stainless steel water container. In order to avoid electrolytic exchange of copper ions, the copper rods were covered by stainless steel tubes. On the outside of the water container, the copper rods were bolted to heavy copper cylinders 60 mm in diameter. Sixtheen graphite brushes with a 20 x 20 mm cross-section rested against each of the copper cylinders. This arrangement permitted 3000 amps or approximately 100 kW to be supplied to the test section. The power came from a direct current generator which delivered currents up to 6000 amps in the range between 0 and 140 volts.

The rotor was mounted in four ball bearings so that an axial elongation of a few millimetres was possible. The housings of the ball bearings were electrically insulated from the heavy steel frame on which the apparatus rested.

The test section was rotated by a vee-belt drive from a direct current motor. The speed was controlled by regulating the motor field current and by changing the wheels of the belt drive. By means of this arrangement steady operation of the rotor was obtained for rotating speeds between 100 and 4000 rpm. Below 100 rpm fluctuations in the rotating speed occurred, and no measurements were therefore carried out below this value.

The water container was made from 5 mm thick stainless steel plates and was provided with two windows for visual observation of the flow around the rotating test section. In order to control the water temperature, two water coolers consisting of chromium-plated copper tubes with an outer diameter of 12 mm were placed in the container.

In order to determine the nondimensional numbers governing the heat transfer rates for this type of flow, the following quantities had to be measured

1. Outside wall temperature of the test section
2. Surface heat flux of the test section
3. Water bulk temperature
4. Rotating speed of test section

The outside wall temperature was obtained by measuring the temperature in the interior of the test section. This was achieved by means of a stationary thermocouple mounted inside a steel tube, 3 mm in diameter, which was inserted into a cavity of the rotor so that the thermocouple junction was located in the middle of the test section as shown in figure 1. The steel tube was supported by teflon bearings mounted in the rotating part of the system. It should be emphasized that the thermocouple system is stationary, the test section rotating around it. In order to check the effects of axial conduction, the thermocouple was moved axially, during a few runs, and we found that isothermal conditions within $\pm$ 0.1 $^{\circ}$C existed in the test section along almost its entire length. Axial conduction effects were only observed in approximately 10 mm long stretches at the ends of the test section. From the thermocouple reading, which was indentical with the inside wall temperature, t_{wi}, the outside wall temperature, t_{wo}, was evaluated by means of the equation

$$t_{wo} = t_{wi} - \frac{q/A \cdot d}{2\lambda} \left[\frac{d_i^2}{d^2 - d_i^2} \ln \frac{d}{d_i} - \frac{1}{2} \right] \qquad (10)$$

The water bulk temperature was measured by means of 16 thermocouples placed inside stainless steel tubes located in the water container as shown in figure 3. The bulk temperature was taken as the average value of the thermocouple readings. Since all thermocouples except those two located just over the test section showed the same temperatures within $\pm$ 1 ^{o}C, we found it necessary only to read the eight thermocouples which were nearest to the test section. For the measurement of the thermocouple voltages a precision Cambridge potentiometer was used. The thermocouple readings were also checked during a few runs by inserting a mercury thermometer in the pool of water. The two sets of readings agreed within $\pm$ 0.1 ^{o}C.

The surface heat flux was determined from the equation

$$q/A = \frac{R_1/R_2 \cdot EI}{\pi\ d\ L} \tag{11}$$

where R_1 was the electric resistance of the test section and R_2 was the electric resistance of the rotor measured over the brushes. This ratio was 0.978. The voltage over the brushes was measured with a Goerz precision voltmeter with a rated accuracy of 1/4 per cent, and the current was obtained by measuring the voltage across a precision shunt calibrated to yield 60 mV at 3000 amps. For this measurement a millivoltmeter with a rated accuracy of 1/4 per cent was used.

The rotating speed was measured with a calibrated tachometer. For some of the runs the speed was also checked by counting the pulses which a small magnet mounted in the rotor induced in a stationary solenoid. The error of measured angular velocity was estimated at 1 per cent.

4.0 Results and Discussion

157 runs were carried out. During these runs the cylinder rpm was varied from about 100 to 4000, corresponding to a rotating Reynolds number from about 1.000 to 46.000. The water bulk temperature was varied between 15 to 65 °C. It was not feasible to operate at much higher temperatures, since surface boiling should be avoided during this phase of the investigation, and the maximum surface temperature was therefore limited to about 100 °C. Boiling effects have, however, also been studied and will be presented in a separate report (8).

All fluid properties were evaluated at the arithmetic mean of the surface and the bulk temperatures. The experimental results are summarized in table I in terms of Nusselt, Reynolds, Grashof and Prandtl numbers.

In figure 4, $Nu/Pr^{0.4}$ is plotted against the Reynolds number. The results show that in the range covered by the present investigation the effects of free convection are negligible since the Grashof number is not needed in order to correlate the data. The data are correlated by the equation

$$Nu = 0.11\ Re^{0.68} \cdot Pr^{0.4}$$

and the deviation of the measurements from this equation is less than $\pm$ 5 per cent except for a few runs.

The exponent for the Reynolds number of 0.68 compares extremely well with the exponents determined for air by Anderson and Saunders, Etemad, Dropkin and Carmi and Kays and Bjorklund, who found 0.667, 0.7, 0.7 and 0.7 respectively. Concerning the exponent of 0.4 for the Prandtl number, it is remarkable to note that this value is identical with the exponent used in the well-known McAdams equation for forced convection inside tubes

$$Nu = 0.023\ Re^{0.8} \cdot Pr^{0.4} \tag{12}$$

If the present results are extrapolated to a Prandtl number of 0.72 valid for air, equation 12 reduces to

$$Nu = 0.096\ Re^{0.68} \tag{13}$$

In figure 5 this equation is compared with the experimental equations mentioned earlier that were obtained for air. The agreement is considered to be excellent.

The present data may also be used for testing the analogy solution by Kays and Bjorklund at different Prandtl numbers. Figure 6 shows a comparison between the analogy solution and the present results for Prandtl numbers of 2 and 5. The theoretical solution and the experimental results compare rather well.

As mentioned in an earlier section Anderson and Saunders (1) suggested that an analogy exists between the present problem and natural convection from a horizontal plate facing upwards. By employing the analogy they solved the problem for air. If the analogy is also applied to the general case of any fluid the following equation is obtained.

$$Nu = 0.111\ Re^{2/3} \cdot Pr^{1/3} \tag{14}$$

In figure 7 equation 14 is compared with the experimental results. The agreement between the theoretical solution and the measurements is rather good, the experimental results being about 20 per cent higher.

It should also be noted that the scatter of the data is larger in figure 7 than in figure 4, revealing that an exponent of 0.4 for the Prandtl number is preferable as opposed to a value of 1/3.

5.0 Summary and Conclusions

In this paper consideration of heat transfer from a horizontal rotating cylinder has been extended to the case of water. All measurements presented have been obtained in the region where the effects of natural convection are negligible and the heat transfer rates depend on the Reynolds and Prandtl numbers only.

On the basis of the experimental results a correlation in terms of Nusselt, Reynolds and Prandtl numbers has been established. Extrapolating our results to the case of air, they compare very well with the available theoretical and experimental information in published works.

Our results have been compared with the theoretical momentum and heat transfer analogy solution of Kays and Bjorklund, and good agreement has been found to exist.

The analogy suggested by Anderson and Saunders between natural convection from a horizontal plate and the present problem has been used to analyse the problem. Analytical and experimental results have been found to compare well with each other.

Finally we conclude that experimental information is lacking on the one hand in the intermediate flow regime where also the Grashof number is of importance, and on the other for a larger range of Prandtl numbers.

Acknowledgements

The auther wish to record his appreciation of Mr Henry Looft, who designed and built the apparatus and Mr Folke Wancke who participated in obtaining the experimental data.

Nomenclature

Symbol	Definition	Units
c_p	Specific heat	KJ/kg ^oC
d	Diameter of test section	mm
d_i	Inner diameter of test section	m
E	Voltage	volt
f	Friction coefficient	Dimensionless
g	Acceleration due to gravity	m/s^2
I	Current	amps
t_B	Bulk temperature	^oC
t_{wi}	Inner wall temperature	^oC
t_{wo}	Outer wall temperature	^oC
Gr	Grashof number	Dimensionless
Nu	Nusselt number	Dimensionless
Pr	Prandtl number	Dimensionless
Re	Reynolds number	Dimensionless
α	Heat transfer coefficient	$KJ/m^2\ s\ ^oC$
β	Coefficient of thermal expansion	$^oC^{-1}$
λ	Thermal conductivity	KJ/m s ^oC
ω	Angular velocity	s^{-1}
ρ	Density	kg/m^3
μ	Viscosity	kg/m s
θ	Temperature difference	^oC

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Table 1 Summary of Experimental Results

Run No	Re	Pr	Gr x 10^{-5}	Nu
1	7777	3.10	37.71	80.25
2	8866	3.00	39.14	87.20
3	10685	3.00	36.54	95.31
4	13066	2.95	36.79	107.49
5	15546	2.88	37.11	116.54
6	17738	2.85	37.11	127.64
7	20530	2.81	38.12	135.69
8	23462	2.77	37.85	152.20
9	24895	2.76	37.96	155.23
10	27658	2.76	37.25	164.86
11	30279	2.73	37.37	169.70
12	33140	2.74	35.88	186.00
13	9885	3.41	31.37	91.15
14	12793	3.19	37.93	103.93
15	14564	3.20	36.15	113.12
16	17317	3.13	36.59	128.20
17	7834	2.90	45.89	72.73
18	9483	2.79	45.12	80.27
19	11670	2.73	45.08	92.55
20	12876	2.69	43.90	97.74
21	14490	2.65	42.23	108.68
22	16505	2.62	42.16	116.35
23	19131	2.55	41.79	125.02
24	22297	2.55	39.21	138.58
25	24663	2.53	38.96	148.12
26	27030	2.53	37.49	160.39
27	29491	2.50	36.44	167.30
28	32807	2.49	35.18	175.33
29	35806	2.47	34.77	193.77
30	37724	2.46	35.14	199.65
31	39560	2.47	34.32	205.92
32	37306	2.45	33.76	197.40
33	35036	2.43	32.68	190.16
34	32276	2.43	30.96	177.80
35	30027	2.36	32.33	163.32
36	27583	2.37	30.64	156.53
37	23301	2.32	32.02	142.02
38	20092	2.28	33.71	124.22
39	17137	2.25	34.26	111.14
40	16095	2.24	33.87	101.12
41	13942	2.29	29.80	96.52
42	12189	2.27	30.24	90.17
43	11112	2.26	31.42	82.08
44	4533	4.14	7.86	58.71
45	6258	4.01	8.87	72.73
46	8161	3.94	8.90	86.96
47	10042	3.83	9.60	99.29
48	11819	3.79	9.50	113.97
49	13916	3.75	10.07	121.89
50	4148	6.38	3.09	65.63
51	4776	6.22	3.54	73.44
52	5819	5.61	5.88	71.42
53	6785	5.57	5.92	76.53
54	7274	5.48	6.10	86.00
55	7242	5.42	6.20	90.14
56	8025	5.40	6.16	100.20
57	9232	5.44	6.09	110.82
58	10630	5.43	5.99	111.85
59	11958	5.39	5.87	127.87
60	13446	5.34	5.88	142.85

Run No	Re	Pr	Gr x 10^{-5}	Nu
61	14754	5.30	6.05	149.25
62	16089	5.33	5.86	152.01
63	17620	5.31	5.82	166.08
64	18877	5.30	5.74	174.36
65	19605	5.27	5.66	180.91
66	18431	5.23	5.79	160.94
67	17017	5.24	5.58	162.66
68	5115	5.22	7.03	71.76
69	5795	5.29	6.53	75.36
70	6466	5.30	6.51	85.33
71	7102	5.33	6.20	89.73
72	7806	5.19	6.83	97.04
73	8517	5.17	6.85	100.53
74	9241	5.24	6.40	108.94
75	19746	3.75	25.86	141.27
76	19515	3.61	26.43	143.46
77	18793	3.57	26.83	139.52
78	18088	3.52	28.30	135.49
79	17084	3.53	27.01	114.76
80	16313	3.49	27.94	112.19
81	15602	3.45	29.08	111.18
82	14767	3.42	29.94	107.49
83	13241	3.54	26.09	110.98
84	12681	3.45	28.41	105.14
85	12027	3.37	31.09	100.26
86	10597	3.47	27.85	95.17
87	9103	3.73	24.67	86.34
88	8421	3.54	29.81	79.90
89	7242	3.72	26.18	74.67
90	6660	3.52	31.84	68.08
91	5680	3.54	31.02	61.71
92	4681	3.58	29.52	52.76
93	3656	3.68	26.82	49.09
94	2948	3.52	31.15	41.65
95	8265	3.66	26.56	83.74
96	9414	3.60	27.68	87.92
97	10100	3.66	26.03	94.96
98	11179	3.60	27.11	96.13
99	11950	3.65	25.26	100.86
100	3263	4.19	15.29	46.64
101	3607	4.28	14.14	48.77
102	3927	4.38	12.88	50.89
103	3686	5.33	6.93	58.07
104	3972	5.41	6.44	60.53
105	4290	5.43	6.32	65.28
106	4779	5.21	7.62	69.39
107	5062	5.28	7.12	71.11
108	5335	5.36	6.63	74.15
109	5680	5.35	6.72	76.92
110	6028	5.33	6.65	80.99
111	6370	5.33	6.69	84.19
112	2720	5.23	7.45	52.72
113	2987	5.39	6.45	58.26
114	3345	5.34	6.76	53.84
115	995	3.36	29.62	19.01
116	1669	4.08	12.95	32.86
117	991	3.37	30.10	19.48
118	2358	4.38	9.92	39.04
119	3028	4.58	7.75	45.84
120	2385	4.32	9.97	39.86

Run No	Re	Pr	Gr x 10^{-5}	Nu
121	1696	4.00	13.75	32.51
122	1031	3.23	32.99	19.20
123	3104	4.45	7.93	49.17
124	1003	4.07	13.97	20.25
125	1505	4.61	7.45	30.95
126	2174	4.84	5.54	39.06
127	2841	4.96	4.62	45.11
128	2208	4.74	5.88	39.26
129	1552	4.45	8.85	29.11
130	1018	4.00	15.45	20.88
131	2869	4.90	4.96	47.96
132	1301	2.52	32.87	19.91
133	2371	2.79	17.26	32.69
134	3420	2.91	12.75	41.30
135	4438	2.99	9.96	50.39
136	3405	2.92	12.37	42.45
137	2359	2.81	16.75	33.44
138	1302	2.51	32.44	19.60
139	4451	2.99	10.15	49.69
140	31729	2.93	25.02	194.41
141	34788	2.96	22.74	211.34
142	38284	2.95	21.37	224.10
143	41237	2.98	19.66	239.37
144	44243	3.00	18.59	250.88
145	44486	2.99	18.71	253.51
146	41774	2.94	19.89	242.62
147	39039	2.89	22.10	226.66
148	36207	2.84	24.32	209.16
149	33481	2.77	27.20	195.06
150	31925	2.91	23.16	196.14
151	34870	2.95	21.24	208.10
152	37873	2.98	19.87	222.93
153	44236	3.01	18.30	251.95
154	41184	2.98	19.50	236.67
155	46530	3.07	16.31	269.42
156	44206	2.01	18.63	250.44
157	41232	2.98	19.92	236.78

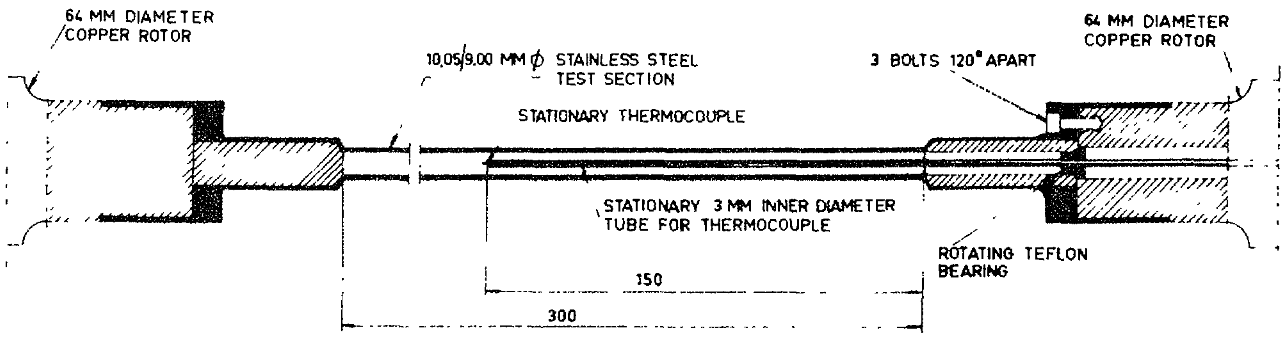

Fig. 1. Details of test section

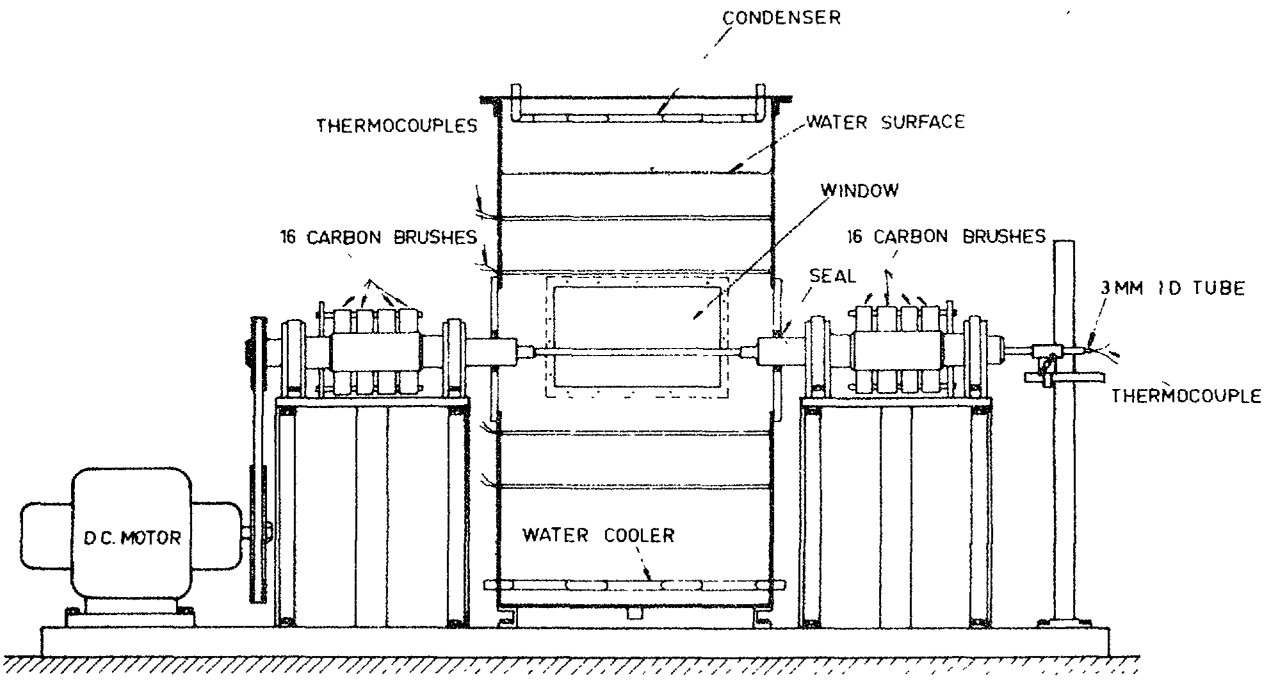

Fig. 2. Apparatus

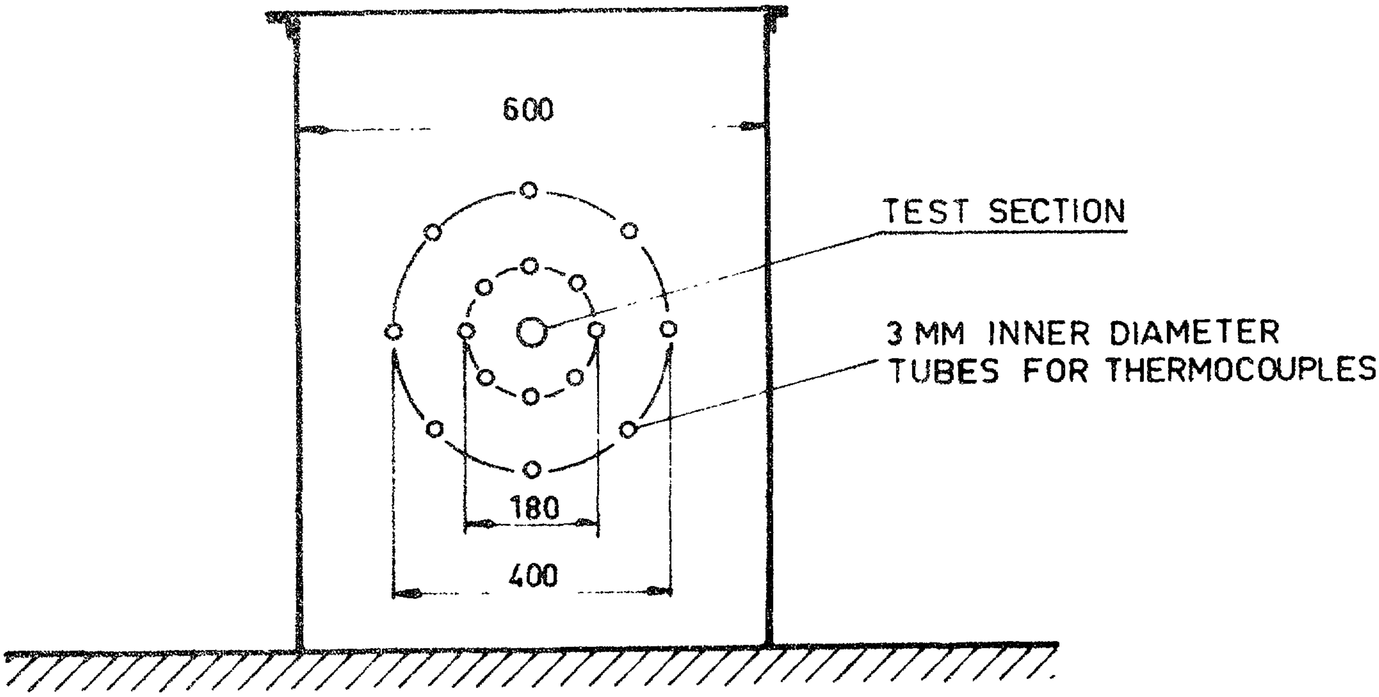

Fig. 3. Location of thermocouples for measurements of water bulk temperature

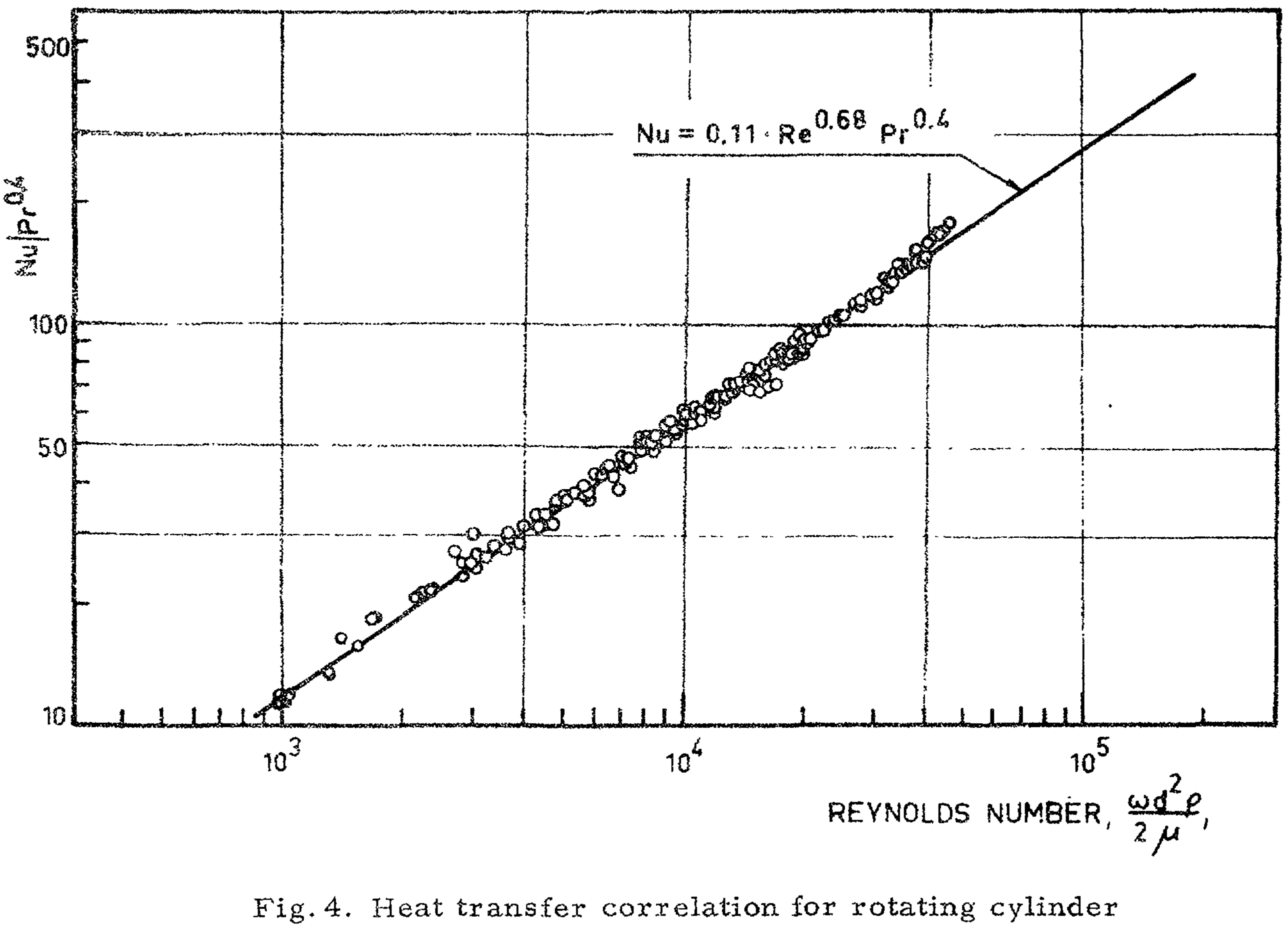

Fig. 4. Heat transfer correlation for rotating cylinder

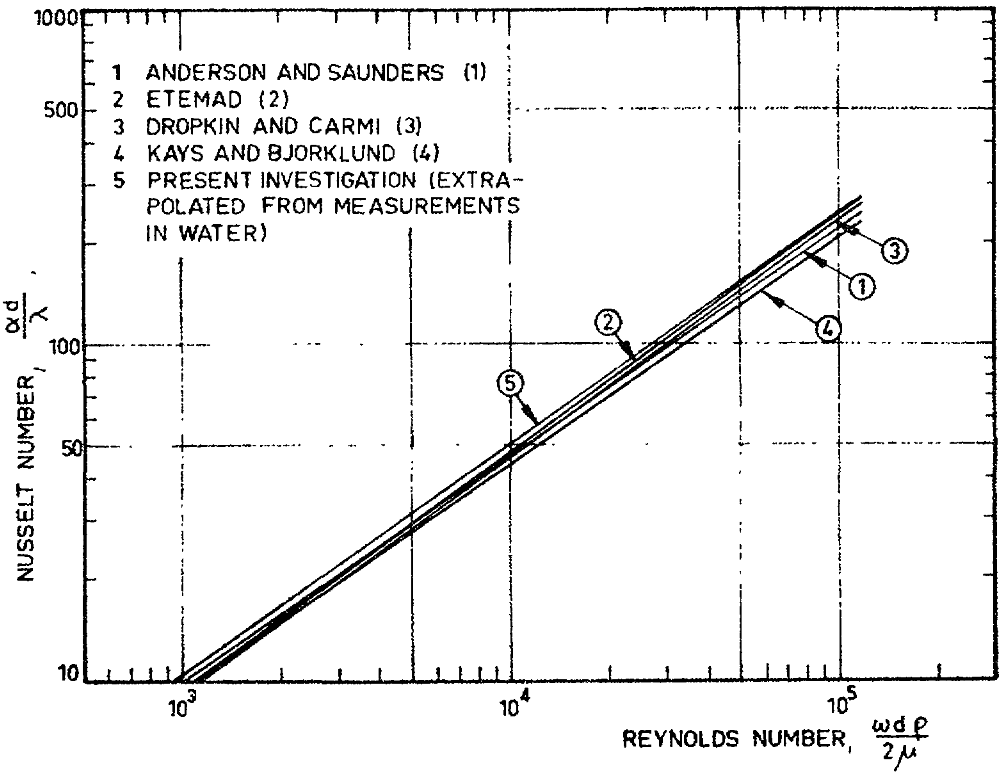

Fig. 5. Summary of experimental results for a horizontal cylinder rotating in air

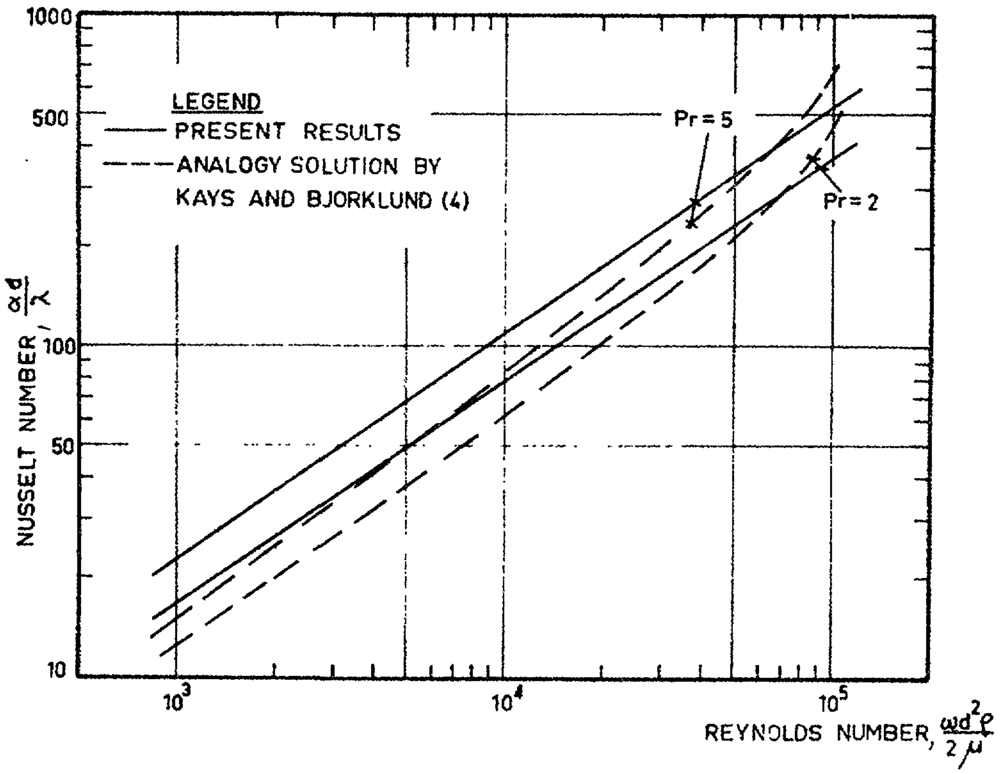

Fig. 6. Comparison between the present results and the analogy solution by Kays and Bjorklund

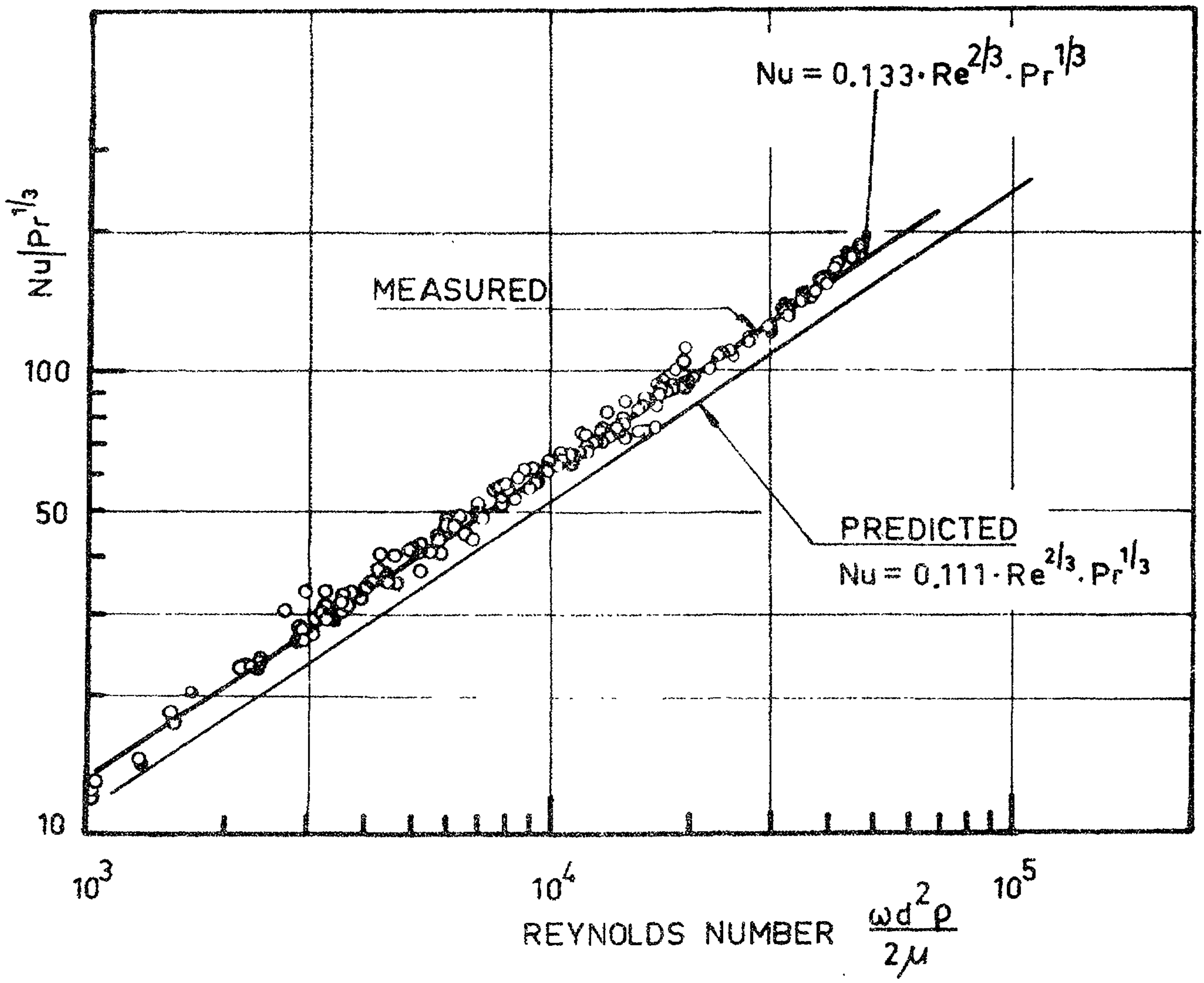

Fig. 7. Comparison between predicted and measured nusselt numbers

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