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INVESTIGATION OF HEAT TRANSFER
FROM A TUBE TO AN AIR STREAM
FLOWING NORMAL TO ITS AXIS

THESIS FOR THE DEGREE OF M. S.

G. E. Gollwitzer
1934

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Heating

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I wish to express my appreciation to Professor L. G. Miller as director of my problem and to the many members of the Mechanical and Electrical Engineering departments for their assistance in obtaining and building apparatus.

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I N T R O D U C T I O N

With the ever increasing use of warm air under forced circulation as a means of heating homes and other buildings, the problems of forced convection become more important. One of the many problems is how to design the heated surface so that it will give up the most heat with the least resistance to air flow. In fact one might say that forced convection is a method of employing high grade energy to make a heating surface more effective. Thus it is evident that in such a system the over-all efficiency will include the power consumed to further heat transfer in addition to the usual factors considered in computing the over-all efficiency of a heating system. Hence with this thought in mind, some companies have adopted stream-lined tubing for forced convection since that form of tubing does give a higher ratio of heat transferred for the amount of resistance it offers to the flow of air past it. It does not follow, however, that that type of tube is the best type.

In order to ascertain what effect the behavior of the fluid around a tube has on the heat transferred from the tube, it seems logical to start with the round tube in order to arrive at some general theories which might help to guide in the design of the best type. The purpose of this paper is, therefore, to find the relative amounts of heat removed from the various portions of a tube by a stream of air moving past it, and to propose reasons

for this distribution. Thus for this particular problem it is desirable to keep all important factors as nearly constant as possible. To this end an attempt is made to obtain a measure of heat loss from as small a portion of the tube as is practicable and to have the air which removes this heat not be preheated by any of the up-stream portion of the cylinder. In other words, only a minute part of the tube is to be heated. This is at variance with most similar investigations as will be shown.

While few investigators seemed willing to advance any reason for their results, one does find a variety of explanations for the increase of heat transferred by forced convection over that by free convection. Most of these explanations make use of boundary films or layers which surround any object in a fluid. Since the fluid used in this work is air, it will be used in this discussion as the surrounding medium. While in still air this boundary film is a very good heat insulator, so it is natural to suppose that the introduction of a blast of air does in some way remove a part of this insulation. Hence there are the following possibilities: (1) That the increase in heat transferred by forced convection is due to the sweeping away of the layers of films which act as insulators. (2) That this increase is due to the impact of the particles of air which hurl themselves into the boundary film in exchange for heated particles which come out to join the air stream. (3) That the increase is due to the increase in the velocity of the outer layers until these layers are moving

at a velocity which exceeds Reynold's number for laminar flow conditions and hence gives a thinner laminar film and less insulation. (4) That the increase in heat transfer by forced convection is greater than that for still air due to a combination of two or more of the above possibilities.

REVIEW OF RELEVANT DATA

Most of the investigation that pertains to round tubes has shown only the total heat transferred and not that removed from the various portions of the pipe. In addition to this work much has been done toward determining the behavior of a fluid near a solid in a fluid stream. By use of minute pitot tubes, thermocouples, hot wire anemometers, and the camera it has been shown that there is no slip between the fluid film and the surface but rather a velocity gradient beginning at zero at the surface and increasing, finally, to the velocity of the stream as the distance from the surface becomes large. By similar means it has been shown that eddy currents appear behind the cylinder and seem to come first from one side and then the other with a definite frequency depending on the value of Reynold's number (1) (2).

Elias (3) found indications of a thick laminar film at the up-stream edge of an object exposed to fluid flow. Photographic work done by Ray (4) indicates a relatively cool region in the film at the downstream side of a hot wire. All of these investigations indicate what may be expected of a flowing fluid near a solid and give some basis on which to form explanations of the results found in this work.

The first heat transfer distribution was proposed by King (5). His work was theoretical and was based on the assumption that the greatest heat transfer would be at the position on the cylinder

from which the film could be most easily removed by the fluid stream. This assumption proved to be incorrect, or at least inadequate. At about the same time Stanton (6) reported that the most heat was given from an electrically heated platinum foil fastened to an ebonite rod when the foil was faced upstream if the rod were round, or downstream if the rod were square. This work, which was carried on by Jakeman, apparently was not designated to show the peripheral distribution. It does indicate, however, that the thick laminar layer at the upstream edge was much larger in the case of the square rod than when the round rod was used.

Lohrisch (8) took advantage of the similarity of the differential equations governing diffusion and thermal conduction. He used hydrochloric acid and phosphoric acid to remove ammonia from dampened filter paper fastened to glass models placed in the fluid stream. He obtained the total heat transferred by covering the entire model with the filter paper. To get the distribution, however, 12 small strips were used and the ammonia diffused from each was measured first with hydrochloric acid in the fluid and then with phosphoric acid. His results indicate a high heat transfer from the leading edge of the cylinder gradually tapering off for 60 degrees on each side. The drop at the sides was rapid and the lowest values were found at about 30 degrees downstream from the sides, that is 60 degrees upstream from the trailing edge. The heat transferred from the back was large and increased in almost direct proportion to the increase in Reynold's number until for large values of

Reynold's number this down stream heat transfer became the maximum.

Reiher's (7) work on the cooling effect of a water cooled pipe indicated high rates of cooling at the front gradually changing to low rates at the downstream edge. This gradual change does not agree with any other experimental work, however.

The work of Drew and Ryan (9) indicated a heat transfer similar to that of Lohrisch. In this work an especial effort was made to approximate actual conditions as found in forced convection over steam pipes. Therefore although the air striking the leading edge was of known temperature, that striking other points had been preheated by the upstream portions and the resulting temperature differential between solid and gas was variable. The reason these results approach those of Lorisch is, of course, that Lorisch obtained a similar effect due to the law of mass action. While both of these experiments show fine corresponding results for their purpose, they do not give the downstream portions of the tube a fair chance to show their ability as heat dissipaters. Hence they give no clue as to the probable optimum shape of a convector tube.

The most recent work seems to be that of Small (10). This work most nearly parallels the problem at hand in that it consisted in a relatively small heating element. The apparatus consisted of a wooden tube containing an electric heating element placed in a slot with a piece of fibre covering the element. The fibre contained two thermocouples at different distances from the center along a radius. The wood and fibre were turned down together to give a smooth round

surface. Since the conductivity of the fibre was known, as was the distance between the thermocouples, then the temperatures as measured by the thermocouples gave the data necessary to compute the heat flux through the fibre. As the fibre was of appreciable size the desired effect of heat given off at the measuring point only was not entirely realized since the air was slightly heated by the up-stream portion of the fibre. Also this method assumes radial flow of heat since the thermocouples are located radially. In addition to these objections there is the difficulty of accurately locating the thermocouples. But in spite of these objections this method remains a nice way of handling the problem since the percent error is probably small.

The results obtained in this experiment were comparable with those advanced in the present paper. The heat transferred was the greatest at angles between 45 degrees and 60 degrees away from the leading edge and slightly lower at the leading edge. He records his lowest points directly at the sides with an increase again at the down stream portions. His minimum rate of heat transmission was from .3 to .4 times that of his maximum.

While much of the above mentioned data seems contradictory it will be shown that when the particular conditions are considered in addition to the data obtained reasonable deductions can be made.

MECHANICAL APPARATUS

The first requisite to an experiment in forced convection is, of course, a steady, controllable supply of air so confined that it can be measured readily. The blowers available were, for purposes of speed variation, equipped with direct current motors. Since there were many large direct current motors nearby which were run intermittently, the direct current supply was of variable voltage and would, therefore, produce a variable fan and air speed. This being undesirable it was thought better to use an alternating current motor at a constant speed and to vary the size of the inlet. For lack of a synchronous motor a large induction motor was used at a small fraction of its rated load thus giving a constant fan speed of 1200 revolutions per minute. Special gates were built for the purpose of varying the size of fan inlet. The fan was of the forward curved blade centrifugal type with a 14-inch round connection at the exhaust side.

The wind tunnel was a 14-inch round pipe in 30-inch sections. The first section was filled with 2-inch pipes for the purpose of straightening the turbulence set up by the fan and thus assuring as nearly stream lined flow as possible in the tunnel. Since the general engineering rule is that air will flow normally past a point 10 pipe diameters down stream from an irregularity, it seemed reasonable that the air issuing from 2-inch pipes 20 inches long would be flowing more nearly stream lined than would be the case if 20 inches of open 14-inch pipe were used. In addition to this

precaution, 10 diameters of the 14-inch pipe were passed before the air arrived at the test section of the tunnel. The entire pipe was kept straight by use of tight wires stretched one-half inch from it which were used to check the alignment. This prevented wave formation due to crooked pipe joints. A canvass connection was used near the blower to dampen vibration in the tunnel.

The test section used was merely a short piece of 14-inch pipe equipped with machined castings bolted to top and bottom for the purpose of supporting the test element to be studied. A drawing of this section complete with the test element is shown on plate 2, entitled "Wind Tunnel Assembly". The bottom view is a view as seen from the blower. The two castings bolted to the top and bottom are identical except that the bottom one has holes drilled and tapped for adjustable legs and the top one has a small scratch indicating the upstream direction. Both castings are supplied with reamed holes through which the element is inserted. The element is so designed that it can be easily removed by merely disconnecting one wire and lifting it from the tunnel.

A cast iron dial was carefully machined and marked at five degree intervals over half the circumference by use of a dividing head. The dial is marked "D" in the upper view of plate 2 and the zero position is marked "0". For simplicity graduations are shown for 60 degrees only instead of 5 degrees as was the actual case. The dial measured 7 inches. It was made large so that the element

could be accurately set in position.

ELECTRICAL APPARATUS

The test element referred to is shown in greater detail on plate 3 called "Details of Heating Element". In the full view shown at the left is a textolite tube with an 8-inch strip of metal exposed. The textolite was chosen because it had a low conductivity of known value. The metal is a part of a number 30 hytemco wire. This metal was selected because it has a high temperature coefficient of electrical resistance coupled with a reasonably high electrical resistance and non-corrosive properties at ordinary temperatures. It is a high nickel alloy steel. In the cross section view at the left this wire may be seen to pass to the interior of the tube and to binding posts at the ends of the tube. Potential leads are placed to measure the potential drop over a three-inch section at the middle of the tunnel. The exposed wire is continued two and one-half inches each side of the test section in order to minimize end losses from that section. The potential leads are fastened to bushings on the opposite side of the tube, these bushings in turn being connected to binding posts at the top of the element. These bushings were used merely to facilitate the assembly of the element. The bottom binding post, partially shown in the cross section view on plate 3, is faced downward so it will not interfere with the removal of the element from the tunnel. The exposed metal strip referred to was made by cementing a piece of number 30 hytemco wire in a small V groove on the surface of

the cylinder. The wire was then carefully honed down flush with the surface of the textolite, the amount honed off being computed by the change of electrical resistance between potential leads.

Plate 1, the wiring diagram, will be used to show the set up of the apparatus.

The large rectangle marked "potentiometer" includes all equipment used in a potentiometer circuit and will not be discussed here. The potentiometer itself is a Leeds & Northrop Student potentiometer. A sensitive portable mirror type galvanometer is used. It will be sufficient to say that a potential applied at the two connections shown may be readily measured by the instrument.

The heat supplying circuit starts at the storage batteries, passes through the variable resistances, through the test element, the fixed resistances and back to the battery. The current flow in this circuit is controlled by the variable resistances which are placed in parallel to allow minute adjustments.

By using switch S it is possible to measure the potential drop across either the fixed resistance or the element itself. It will be well at this point to consider the relationship between these two.

The element is made of hitemco wire which has the property of changing its resistance appreciably with small changes of temperature. The relation between temperature and resistance is known in this case so it may be shown that a piece of this wire having a resistance of .4008 ohms at 68 degrees F. will have a resistance

of .4353 ohms at 100.5 degrees F. It will be readily seen that by using a fixed resistance of .4353 ohms in series with the test element, the element will be at 100.5 degrees F. when the potential drop across the fixed resistance is the same as that across the element. This condition may be brought about by adjustment of the current flowing in the circuit and hence the heat supplied to the element. Furthermore, the value of this potential drop indicates both the amount of current flowing and the heat given up by that portion of the element lying between the two potential leads. The fixed resistance is made of advance wire, the resistance of which is very nearly constant over the range of temperatures used. It is immersed in an oil bath.

P R O C E D U R E

For each run the setting of the air inlet gate was made at approximately the desired value by use of a blade type anemometer. This was, then, the air velocity used for the run and its actual value was determined accurately by use of a pitot tube traverse with the Whalen gage. Other factors such as humidity, barometric pressure, and temperatures were noted to be used in computing the actual air velocity and the value of Reynold's number.

With the air supply functioning properly the element was turned to measure up-stream heat transfer. The potential drop across the three-inch section was made to balance that across the fixed resistance by adjusting the current in the circuit to the proper value. This, of course, indicated that the wire in the element was at 100.5 degrees F. and the element was allowed a few minutes to obtain equilibrium. When the apparatus had attained a constant condition, readings were taken. At each position (i.e. 0, 5, 10, 15, 20 degrees etc. from the position directly upstream) the potential drop across the three-inch section was made to balance that across the fixed resistance and the value in volts of this drop was recorded.

Periodically throughout the run the temperature, humidity, and atmospheric pressure were noted.

Thirty-seven readings were taken for each air velocity and seven velocities were used ranging from 373 feet per minute to 1300

feet per minute. The 378 feet per minute run was in the nature of a preliminary run and was therefore not plotted.

COMPUTATIONS

The first computations were made in order to determine the length of advance wire necessary to give a fixed resistance equal to the resistance of the element when at about 100 degrees F.

The resistance of the test section was found to be .4008 ohms at 68 degrees by comparing it with the resistance of a standard ohm in a low current series circuit.

Data on hytemco shows that a 35 percent increase in electrical resistance corresponds to a temperature rise of 132 degrees F. in the vicinity of 100 degrees F. Hence an increase of 32 degrees F. will raise the resistance of the element to $.4008 \times \frac{32}{132} \times .35$ or about .435 ohms.

Since the resistance of number 24 advance wire is given as .7277 ohms per foot then the number of inches of this wire necessary to give .435 ohms is $\frac{.435}{.7277} \times 12$ or 7.18 inches.

When a resistance reasonably near this was made complete with soldered potential leads, its resistance was found by comparison with the standard ohm to be .4353. This was found to be an 8.6 percent rise above .4008 so that the hytemco wire will be raised $\frac{8.6}{35} \times 132$ or 32.43 degrees to give it a resistance of .4353 ohms. As this is beyond the range of accuracy of the thermometers used, however, 100.5 degrees F. is the value used in subsequent calculations.

In order to determine the temperature of the air corresponding to each reading taken, the temperatures as recorded were plotted

as ordinates against positions of the element as abscissa. An average curve drawn through these points gave a graph from which a value for the air stream temperature at any of the thirty-seven points could be obtained.

Computations for the exact air velocity were made in the customary way using the formula $V \text{ equals } \sqrt{2 g h}$ in which h is the head measured in feet of the fluid being used (air in this case), and $2 g$ is 64.4. For a more detailed discussion of such computations, see (11) or any book on air flow and its measurements. The velocity obtained as above divided by the weight of one cubic foot of air gives the number of pounds of air passing one square foot of area per second. This is the value often designated as G in the formula $Re = \frac{dG}{\mu}$ for Reynold's number. Values of Re were also found in each case by substituting $1/12$ for d and the absolute viscosity of air for μ . The results for these standard computations are shown on the data sheets.

The actual heat transfer computations while simple were not as usual as the above and will therefore be examined in slightly more detail. From Ohm's law $I = E/R$ and watts $= I^2 R$, it is seen that watts also equal E^2/R . Since all measurements have been made in volts and since the resistance is held constant at .4353, the heat loss in each case will be

$$\frac{(\text{volts drop})^2}{.4353} \times \frac{(\text{factor to change to B.T.U./ hour})}{\text{Temperature difference}} .$$

This gives the total heat loss in B.T.U. per hour per degree

difference in temperature. These computations were made by logarithms. The equation then became $\text{Log B.T.U.} = 2 \text{ Log voltage drop} + \text{Log constant} - \text{Log temperature difference}$. The results of these computations are shown on the following sheets and are plotted on plates four and five.

The data as computed was merely relative and while valuable for the purposes of this paper it was thought advisable to so arrange it that it might be compared with work of other investigators as a check. To do this curves were drawn showing the heat loss for each setting of the element over the range of air velocities used. Extra polating these to the zero air speed line gave a point indicating the constant heat transferred independent of position. Values resulting when this correction was applied checked reasonably close to the recent work of Small (10).

TABULATION OF DATA AND RESULTS

Air Velocity = 600 ft. per min. or 45.05 lbs. per min. per sq. ft.

Reynold's number = 5,100

Column A = Position of cylinder in degrees from front or back.

Column B = Potential drop in volts.

Column C = Temperature difference in degrees F.

Column D = Heat loss per hour per degree difference in B. T. U.

A	Front B	C	D	B	Back C	D
0	.5066	27.9	.07837	.4805	25.1	.07836
5	.5167	27.8	.08152	.4808	25.1	.07846
10	.5250	27.7	.08477	.4791	25.2	.07760
15	.5335	27.7	.08753	.4776	25.3	.07681
20	.5421	27.6	.09081	.4750	25.4	.07571
25	.5465	27.5	.09252	.4730	25.5	.07457
30	.5525	27.4	.09491	.4700	25.5	.07380
35	.5575	27.3	.09699	.4660	25.6	.07227
40	.5604	27.3	.09800	.4645	25.7	.07152
45	.5626	27.2	.09914	.4625	25.8	.07063
50	.5641	27.1	.10003	.4590	25.9	.06914
55	.5648	27.0	.10066	.4591	25.9	.06933
60	.5657	27.0	.09991	.4595	26.0	.06918
65	.5583	26.9	.09872	.4608	26.1	.06931
70	.5509	26.8	.09647	.4633	26.2	.06963
75	.5399	26.7	.09301	.4681	26.2	.07125
80	.5263	26.6	.08871	.4735	26.3	.07262
85	.5082	26.6	.08271	.4776	26.4	.07361
90	.4920	26.5	.07782			

Air Velocity = 800 ft. per min. or 56.5 lbs. per min. per sq. ft.

Front				Back		
A	B	C	D	B	C	D
0	.5336	30.6	.07942	.5069	27.8	.07884
5	.5427	30.5	.08227	.5065	27.9	.07834
10	.5555	30.4	.08493	.5057	28.0	.07781
15	.5643	30.3	.08953	.5033	28.1	.07680
20	.5723	30.3	.09209	.5018	28.1	.07634
25	.5792	30.2	.09464	.4990	28.2	.07522
30	.5873	30.1	.09540	.4955	28.3	.07391
35	.5888	30.0	.09845	.4912	28.4	.07238
40	.5916	30.0	.09939	.4883	28.4	.07153
45	.5945	29.9	.10070	.4844	28.5	.07014
50	.5945	29.8	.10104	.4824	28.6	.06932
55	.5934	29.7	.10101	.4811	28.7	.06871
60	.5920	29.7	.10100	.4811	28.7	.06871
65	.5875	29.6	.09934	.4806	28.8	.06832
70	.5784	29.5	.09661	.4829	28.9	.06874
75	.5650	29.4	.09250	.4866	29.0	.06959
80	.5477	29.3	.08722	.4929	29.0	.07137
85	.5250	29.3	.08014	.4964	29.1	.07376
90	.5067	29.2	.07491			

Air Velocity = 1000 ft. per min. or 70.2 lbs. per min. per sq. ft.

Front				Back		
A	B	C	D	B	C	D
0	.5396	27.9	.08891	.5345	27.5	.08851
5	.5478	27.8	.09196	.5340	27.5	.08834
10	.5607	27.7	.09669	.5340	27.5	.08834
15	.5714	27.65	.1005	.5288	27.5	.08663
20	.5811	27.6	.1042	.5250	27.5	.08539
25	.5907	27.6	.1077	.5205	27.5	.08412
30	.5973	27.5	.1105	.5164	27.5	.08261
35	.6013	27.5	.1120	.5107	27.5	.08080
40	.6050	27.5	.1134	.5060	27.5	.07932
45	.6100	27.5	.1153	.5032	27.5	.07844
50	.6113	27.5	.1158	.5000	27.5	.07745
55	.6098	27.5	.1152	.4968	27.5	.07646
60	.6086	27.5	.1147	.4959	27.5	.07618
65	.6035	27.5	.1128	.4954	27.5	.07618
70	.5942	27.5	.1092	.4934	27.5	.07634
75	.5800	27.5	.1042	.4934	27.5	.07695
80	.5597	27.5	.09705	.5022	27.5	.07813
85	.5365	27.5	.08714	.5089	27.5	.07966
90	.5186	27.5	.08332			

Air Velocity = 1200 ft. per min. or 80.8 lbs. per min. per sq. ft.

Front				Back		
A	B	C	D	B	C	D
0	.5073	27.9	.07859	.5439	25.2	.10000
5	.5588	27.8	.09566	.5437	25.2	.03394
10	.5703	27.6	.10000	.5417	25.2	.09920
15	.5813	27.4	.1051	.5396	25.3	.09804
20	.5915	27.2	.1096	.5349	25.3	.09632
25	.5998	27.0	.1135	.5295	25.3	.09441
30	.6065	26.9	.1165	.5246	25.3	.09267
35	.6118	26.8	.1190	.5191	25.4	.09038
40	.6161	26.7	.1211	.5141	25.4	.08865
45	.6195	26.6	.1239	.5090	25.4	.08690
50	.6215	26.5	.1242	.5061	25.5	.08557
55	.6210	26.4	.1244	.5038	25.5	.08480
60	.6186	26.3	.1240	.5030	25.5	.08453
65	.6138	26.2	.1225	.5031	25.6	.08423
70	.6047	26.1	.1194	.5019	25.6	.08383
75	.5891	26.1	.1133	.5037	25.7	.08410
80	.5661	26.0	.1050	.5059	25.7	.08484
85	.5402	25.9	.09599	.5116	25.8	.08643
90	.5211	25.9	.08932			

Air Velocity = 1600 ft. per min. or 107.4 lbs. per min. per sq. ft.

Front				Back		
A	B	C	D	B	C	D
0	.5484	24.8	.1026	.5492	21.9	.1184
5	.5570	24.7	.1070	.5476	21.8	.1172
10	.5700	24.6	.1125	.5451	21.8	.1161
15	.5840	24.5	.1186	.5427	21.8	.1151
20	.5955	24.4	.1238	.5377	21.8	.1130
25	.6037	24.3	.1277	.5304	21.9	.1120
30	.6103	24.2	.1311	.5265	21.9	.1078
35	.6186	24.1	.1353	.5216	22.0	.1054
40	.6227	24.0	.1376	.5149	22.0	.1027
45	.6270	23.9	.1401	.5086	22.1	.09972
50	.6280	23.8	.1412	.5045	22.2	.09779
55	.6259	23.7	.1408	.5034	22.2	.09725
60	.6254	23.6	.1412	.5024	22.3	.09643
65	.6194	23.5	.1392	.5010	22.4	.09546
70	.6111	23.3	.1365	.4988	22.5	.09421
75	.5905	23.2	.1280	.4985	22.6	.09368
80	.5657	23.1	.1080	.5005	22.7	.09401
85	.5359	23.0	.1064	.5045	22.8	.09510
90	.5181	22.9	.09992			

Air Velocity = 2000 ft. per min. or 130.3 lbs. per min. per sq. ft.

Front				Back		
A	B	C	D	B	C	D
0	.5649	20.2	.1346	.5579	18.4	.1441
5	.5752	20.1	.1402	.5556	18.4	.1419
10	.5906	20.0	.1486	.5532	18.4	.1417
15	.6052	19.9	.1568	.5472	18.4	.1386
20	.6165	19.8	.1658	.5430	18.4	.1365
25	.6251	19.7	.1690	.5382	18.4	.1341
30	.6320	19.7	.1737	.5319	18.5	.1304
35	.6357	19.6	.1756	.5266	18.5	.1277
40	.6400	19.5	.1789	.5170	18.5	.1233
45	.6425	19.4	.1813	.5157	18.5	.1225
50	.6416	19.3	.1817	.5122	18.5	.1208
55	.6436	19.2	.1753	.5066	18.5	.1191
60	.6381	19.2	.1602	.5056	18.6	.1166
65	.6320	19.1	.1782	.5020	18.6	.1154
70	.6200	19.0	.1734	.4983	18.7	.1134
75	.5965	19.0	.1606	.5000	18.7	.1137
80	.5373	18.9	.1452	.5012	18.7	.1147
85	.5336	18.8	.1290	.5061	18.8	.1131
90	.5152	18.8	.1161			

Air Velocity = 400 ft. per min. or 26.05 lbs. per min. per sq. ft.

Front				Back		
A	B	C	D	B	C	D
0	.7515	62.7	.07674	.6965	62.7	.06576
5	.7604	62.7	.07856	.6965	62.7	.06576
10	.7719	62.7	.08096	.6962	62.7	.06586
15	.7834	62.7	.08339	.6943	62.7	.06550
20	.7930	62.7	.08544	.6932	62.7	.06529
25	.7997	62.7	.08689	.6911	62.7	.06489
30	.8052	62.7	.08809	.6883	62.7	.06437
35	.8134	62.7	.08989	.6866	62.7	.06405
40	.8167	62.7	.09063	.6840	62.7	.06357
45	.8158	62.7	.09043	.6813	62.7	.06316
50	.8160	62.7	.09047	.6802	62.7	.06287
55	.8160	62.7	.09047	.6820	62.7	.06319
60	.8086	62.7	.08884	.6808	62.7	.06298
65	.8077	62.7	.08864	.6871	62.7	.06415
70	.8031	62.7	.08763	.6926	62.7	.06518
75	.7921	62.7	.08525	.7014	62.7	.06685
80	.7724	62.7	.08106	.7087	62.7	.06824
85	.7545	62.7	.07735	.7185	62.7	.07014
90	.7354	62.7	.07348			

DISCUSSION OF RESULTS

For convenience consider first what might naturally be expected to occur when an air stream flows past a round tube. The air must, of course, be parted by the tube yet it is common knowledge that a sharp point at the leading edge is not necessary or even desirable for stream lined conditions. The reason for this is, as has been shown by many investigators (3) (6) (10), that a relatively stationary film of air exists in front of the tube to form an effective point and to part the approaching air. This film is held in place by the force of the moving air, or more correctly, by the radial component of that force. In the straight up-stream position this force is entirely radial but as one considers points around toward the side it is seen that the tangential component becomes increasingly large and removes the film rather than holding it in place. This effect will be most pronounced at the side. Thus did King (5) predict high rates of heat transfer at the sides.

Continuing back from the side of the tube there is a low pressure area in which the velocities of the outer film layers are still too great to allow excessive turbulence (12). This is the condition which is maintained as nearly as possible by the gradual curve at the back of a strut or other stream lined object. The purpose in those cases is to maintain a low pressure area with resulting reduction in skin friction while at the same time pre-

venting eddy currents by not allowing the pressure to become too low. In the case of a round tube, however, this condition does not exist for a great distance back of the side before turbulence is set up due to the low velocity of the films compared to the high velocity of the stream and the resulting low pressure behind the tube. These eddy currents provide a condition in which the air actually strikes the back of the tube, or more properly, the film which surrounds it. Also, it has been shown that (2) the eddys come alternately from either side of the tube and it is reasonable to believe that they have a scouring effect capable of removing outer layers of the air film. Finally the picture is a tube blanketed in front by a thick relatively stationary film of air which becomes thinner at the sides. This film probably becomes thinnest at the sides where the total force is tangential to the surface but gets large again downstream slightly just before eddy currents begin to effect it. The downstream portion is probably scoured by eddy currents especially at high velocities and the film there no doubt becomes quite thin at those velocities.

With this in mind, an inspection of the heat transfer distribution curves on plates 4 and 5 will show that the maximum heat transfer rates do not coincide with the points of smallest film thickness. This surprised early investigators who expected a heat distribution similar to that predicted by King.

It follows, therefore, that if present ideas about film distribution are correct, and they have been established by a

number of methods and for a variety of purposes, there is some factor entering into the heat transfer other than just that of the film thickness itself. Also, it may be said that since the largest heat transfer does not occur at the direct up-stream portion of the tube, the transfer is not totally dependent upon the impact and exchange of particles striking the tube. Thus the first two possibilities mentioned in the introduction must be abandoned.

The third possibility, namely, that the increase in velocity changes laminar films to turbulent ones allowing greater heat transfer, cannot possibly explain the effect noted since this would imply higher heat transfer at the sides than is the case. Whether the films are transferred from laminar to turbulent or are entirely swept away is beside the point. Since there are distinct evidences of a turbulent layer, it is probably true that an increase in the velocity of air over a pipe results in decreasing the thickness of the laminar film by moving the turbulent layer nearer to the pipe. Keck (13), by use of sensitive apparatus shows that these transitions, whatever they are, appear to take place periodically with increases of air velocity as though the ultimate strength of the adjacent film bonds was being exceeded at these periods. Obviously, from this data, the process is not a gradual one.

From the above discussion it seems logical to conclude that the rate of heat transfer in this case is a function of each of the components of the air stream force against the tube.

The tangential component tends to tear away (or render less effective by turbulent movement) the insulating film of air surrounding the tube. The radial component tends to bring the moving air in more intimate contact with the air of the film, thus improving heat transfer by conduction or by an actual exchange of cold particles for warm ones.

That this explanation satisfies the actual conditions as shown by the curves on plates 4 and 5 may be seen. The up-stream portion is not affected by a large enough tangential component and hence the transfer is poor due to the thick air layer. The interval between 45 and 60 degrees from this point is high in heat transfer due to the favorable combination of the two forces and the amount transferred varies only slightly over that range where one force is replacing the other. From the 60 degree point to the side position the rate of heat transfer drops rapidly due to the rapid decrease in the value of the cosine of angles near 90 degrees, hence the radial component becomes small. The portion just back of the side position has the lowest rate of heat transfer since the radial force actually becomes opposite in sense. This is the condition present at the back of the streamlined tube and accounts for the smaller heat of streamlined tubing compared with round tubes of the same surface area. Back of this point turbulence is set up and while this portion cannot be readily analyzed it is probable that the radial and tangential forces again come into play as the eddy currents strike the back.

Without further reviewing the data of other investigators it will be seen that this explanation is in keeping with all data except, possibly, that of Reiher (7) on the cooling of air which he himself finds puzzling and explains by supposing eddy currents in the cooling water inside the pipe used. The works of Drew and Ryan or of Lohrisch show maximum heat transfer at the up-stream center, but since isothermal conditions or its equivalent were approached in each case, the temperature differential was variable as previously explained and one would not expect the maximum values as shown in this paper.

Unfortunately these results show that any attempt to streamline a tube used as a convector will result in reducing the heat transfer rate on the back portion. On the other hand, streamlining is desirable since it reduces the wind resistance to heat transfer ratio. A more specialized study would show to what extent streamlining is practicable. A more promising possibility is that of replacing the laminar film at the front of the tube with the tube itself by proper shaping. This should result in higher heat transfer rates without appreciable increases in wind resistance.

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PLATE I
WIRING DIAGRAM

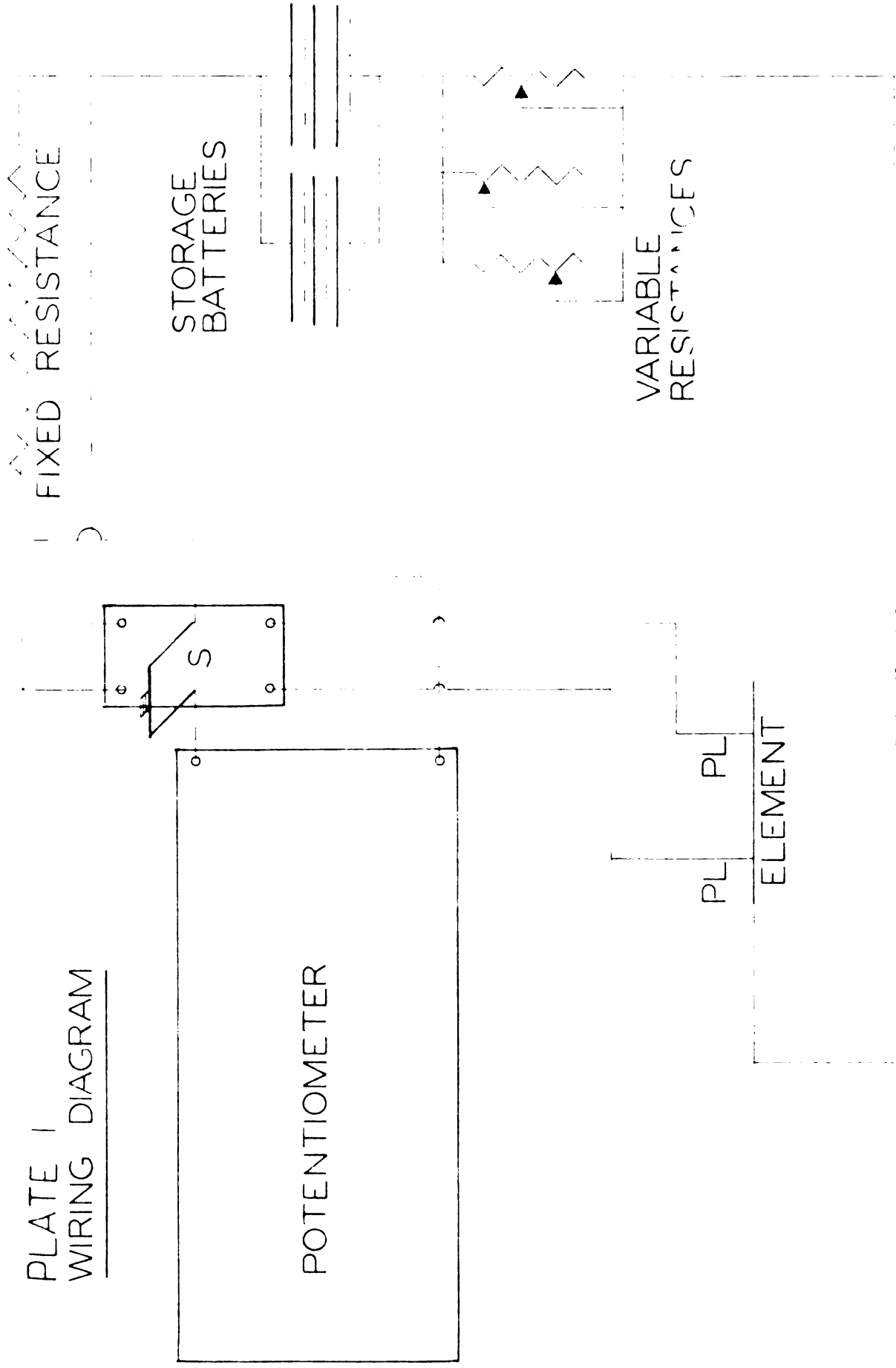
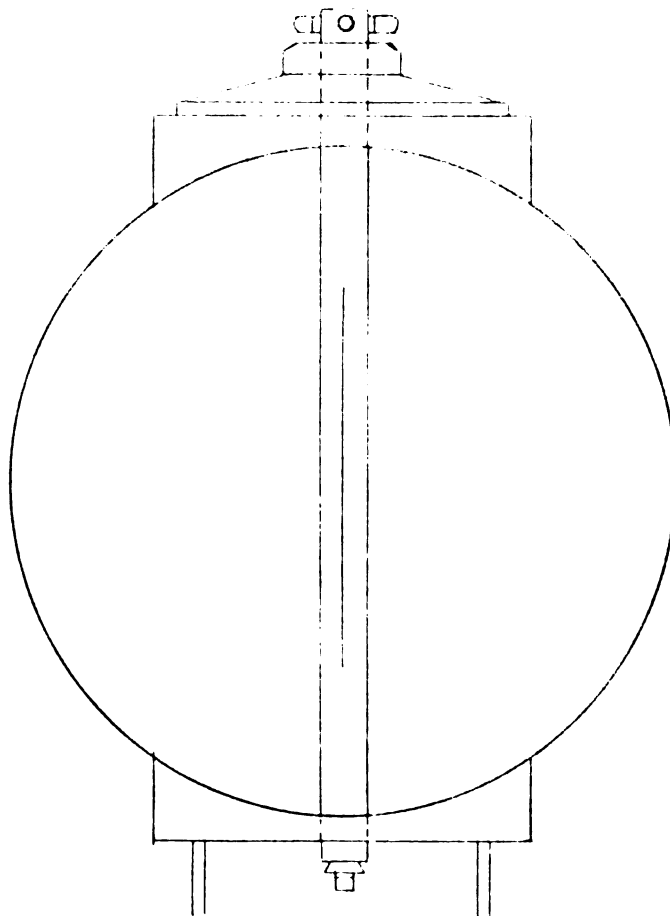
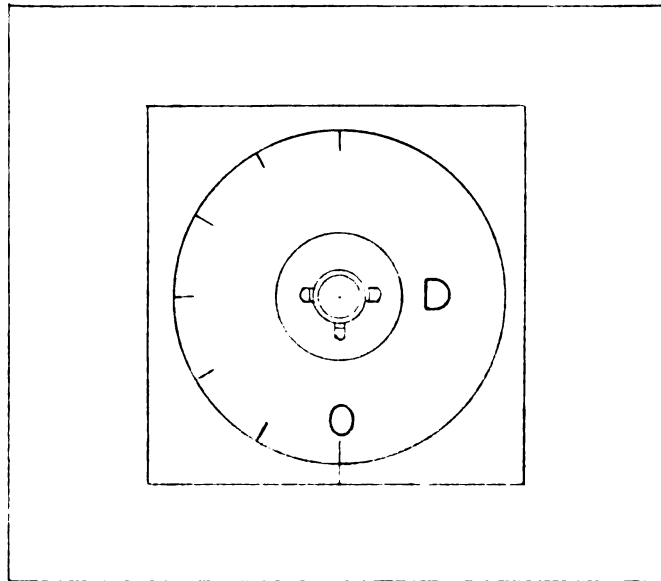
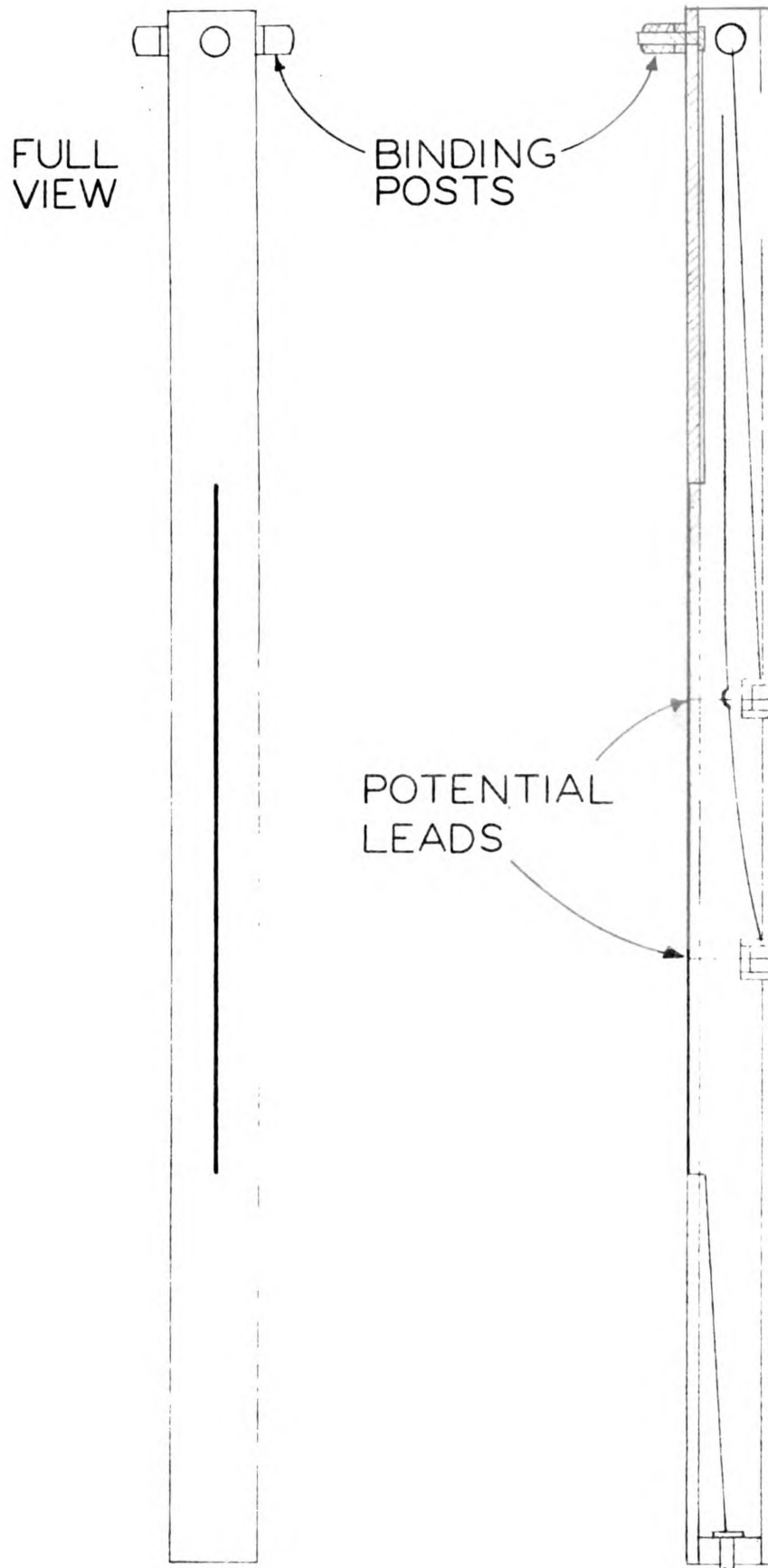


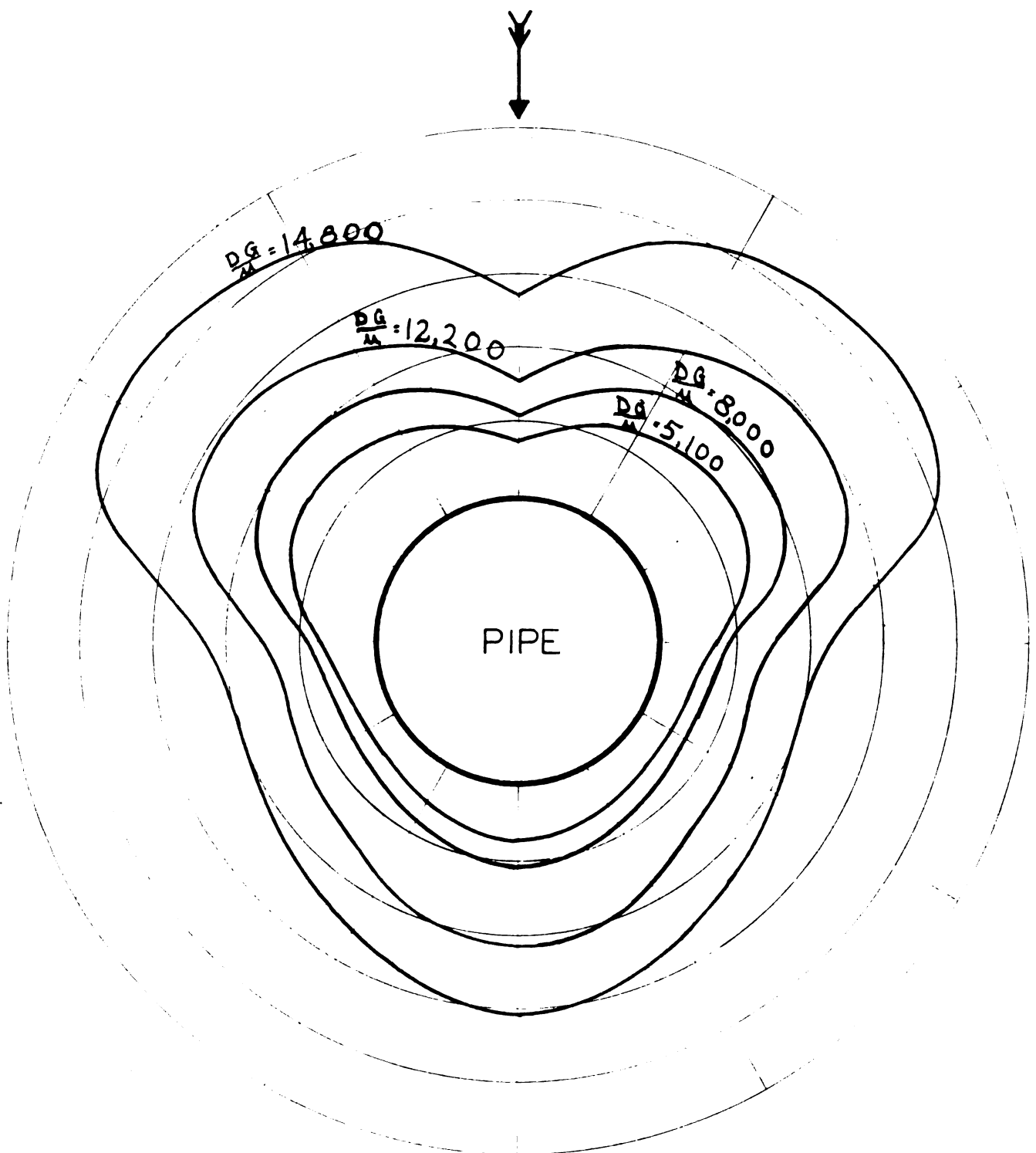
PLATE 2
WIND TUNNEL ASSEMBLY



DETAILS OF HEATING ELEMENT



DISTRIBUTION OF HEAT TRANSFERRED FROM A CYLINDER TO AN AIR STREAM



DISTRIBUTION OF HEAT TRANSFERRED
FROM A CYLINDER TO AN AIR STREAM

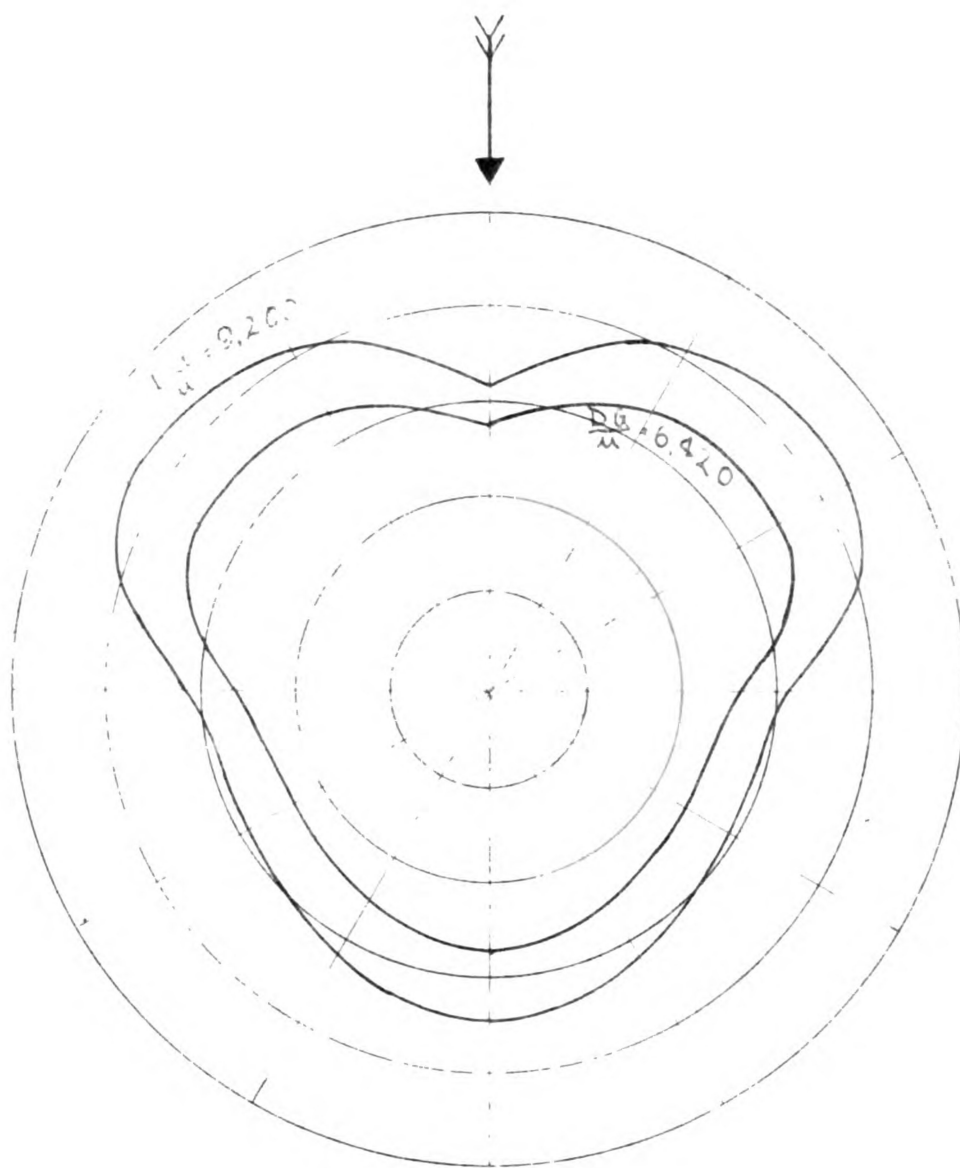


PLATE 5

ROOM USE ONLY

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