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HEAT TRANSFER FROM THE REAR OF BLUFF OBJECTS TO A LOW SPEED AIR STREAM

H. H. SOGIN

TULANE UNIVERSITY
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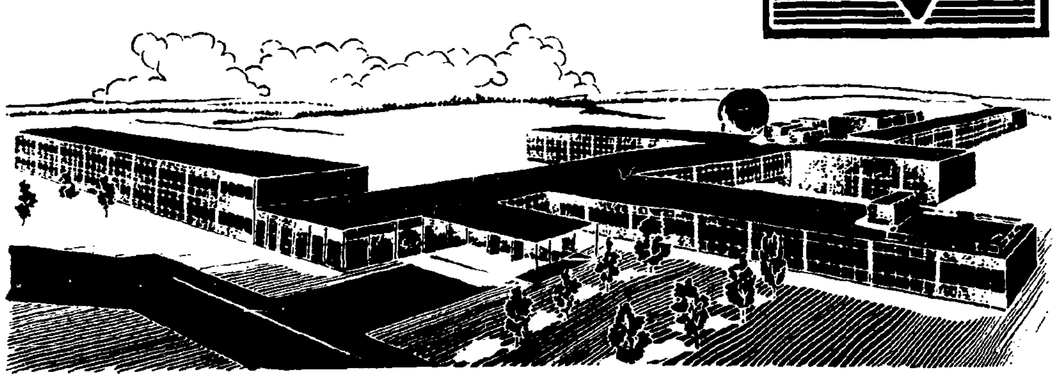

NOTICES

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FOREWORD

This final technical report was prepared by Tulane University, New Orleans, Louisiana, on Contract AF 33(616)-8481 for the Aeronautical Research Laboratories, Office of Aerospace Research, United States Air Force. The work reported herein was accomplished on Task 7063-01, "Research In Heat Transfer Phenomena" of Project 7063, "Mechanics of Flight" under the technical cognizance of Dr. Max Scherberg of the Thermo Mechanics Research Laboratory of ARL.

This report includes results of the forced convection experiments described in reference 1 of the Bibliography, and it includes new results from configurations formed by modifying the shape or orientation of the basic wind-tunnel apparatus. The latter experimental work was performed with the cooperation of Mr. Paul Buettiker during the summer of 1960 at Brown University, Providence, Rhode Island.

ABSTRACT

The local heat transfer by forced convection from the base surface of a blunt obstacle in a variety of configurations has been measured. The data are satisfactorily correlated by an equation of the type

$$\frac{hL}{k_f} = C\left(\frac{U_\infty \rho_f L}{\mu_f}\right)^{2/3}$$

where the properties are evaluated at the mean film temperature. The coefficient C depends upon the configuration and the location on the rear surface. Devices that close the "dead-air" space, or reduce its size, reduce the value of C. Thus, C has practically the uniform value 0.20 for a flat-plate strip at 90-degree angle of attack; but at 25-degree angle of attack it changes from 0.09_3 where the apparent free streamline changes direction through an obtuse angle, to 0.18 where the apparent free streamline changes direction through an acute angle.

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NOMENCLATURE

Symbol	Description
h	coefficient of heat transfer
k_f	thermal conductivity of air at film temperature
L	chord length of the main test plate
$\overline{Nu}_f$	Nusselt number: hL/k_f
q''	local convective heat flux at the wall
$\overline{Re}_f$	Reynolds number: $U_\infty \cdot L \cdot \rho_f/\mu_f$
t_h	heater temperature
t_o	air stagnation temperature
t_r	recovery temperature
t_w	wall temperature
U_∞	wind-tunnel speed of approach
μ_f	dynamic viscosity at film temperature
ρ_f	air density at stagnation pressure and film temperature

The bar over a symbol signifies a mean value.

I - INTRODUCTION

Experiments were performed to determine the heat transfer by forced convection from immersed surfaces to totally separated regions of flow. The purpose has been to provide data that might be useful for engineering design and to learn how the major parameters of the flow affect the heat transfer.

Earlier results of this investigation (Ref. 1) were obtained on a bluff flat plate strip of rectangular cross section in two-dimensional flow. The chord length was 6.732 inches and the thickness 1.00 inch. The nominal blockage ratio was 0.211 when the angle of incidence was 90 degrees. The advantage of this obviously poor aerodynamic shape was that the separation was geometrically fixed and the flow continued without reattachment. The plate was electrically heated and was instrumented to measure distributions of wall flux and temperature in the horizontal plane of symmetry of the wind tunnel. Preliminary naphthalene experiments had shown that the rearward mass transfer at any point was practically independent of the vapor pressure anywhere else on the surface. This implied a strong mixing behind the plate in the range of Reynolds number covered by the tests (100,000 to 440,000). Accordingly, the heat transfer tests were predicated on the proposition that the rearward heat

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transfer would be practically independent of the temperature distribution on both the front and the rear surfaces of the main test plate. In referring to this proposition later, it will be said that the local coefficients are autonomous.

This report reviews the pertinent results of the earlier work on two configurations (Series B and D)[1] and presents new data and results on three more configurations (Series H, J, and K). (See Figure 1.) Series B, K, and J show rearward distributions for angles of incidence of 90, 65, and 25 degrees, respectively. Series D shows the effect of a splitter plate behind the main test plate; Series H shows the influence of a half circular cylinder in front of the main test plate. All the results now include the corrections for chordwise conduction, which were omitted from the earlier presentation.

In summary, a correlation of the type $\underline{Nu}_f \sim \underline{Re}_f^{2/3}$ fairly represents the results in each series. The maximum and most uniformly distributed heat transfer occurs when the plate is at 90-degree angle of incidence, in the absence of both the splitter plate and the half cylinder. This indicates that for a given Reynolds number the heat transfer from the rear of the plate would increase with decreasing blockage ratio, reaching a maximum in an infinitely wide stream, and that it could be exceeded

[1] The earlier report also covers the influence of some turbulence grids and the heat transfer from the forward surface at 90-degree angle of incidence; also, see Ref. 2.

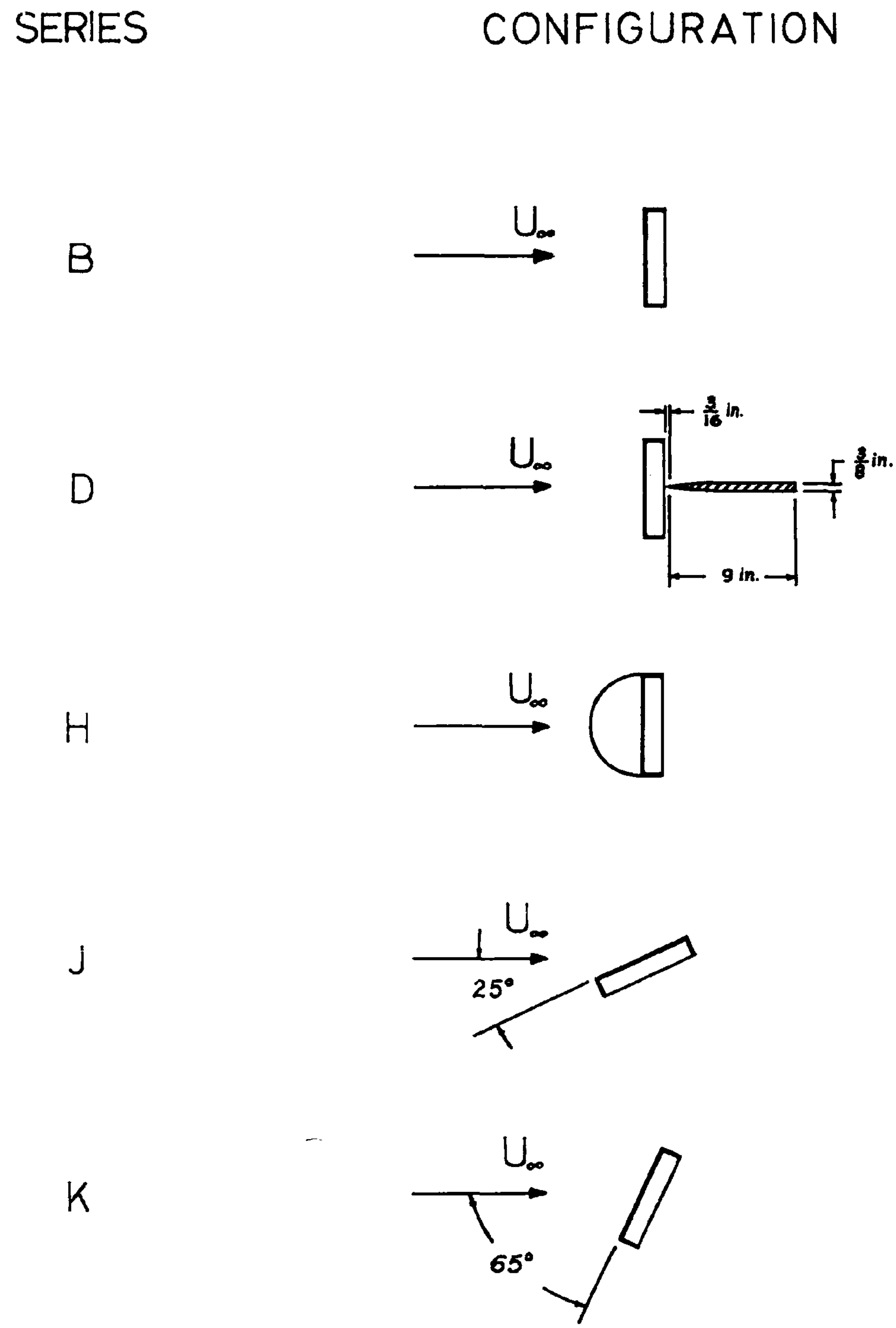

Figure 1. Schematic diagrams of the experimental configurations.

only by the rearward heat transfer from a cylindrical segment with its concave surface facing the flow. Thus an upper limit of over-all turbulent heat transfer to a completely separated flow is indicated. The distributions of wall flux and temperature in all the other configurations are either markedly non-uniform or askew.

The transfer mechanisms behind the bluff objects still defy analysis. Consequently, the trend with Reynolds number and the distributions of the flux in the several configurations are still not explained. It is found, however, that P. D. Richardson's (Ref. 3) analogical argument may be used as a qualitative guide. More directly it may be said that devices which increase the size of the "dead-air" region increase the heat transfer.

To achieve better understanding, additional experiments should be performed. They should include measurements of pressure and temperature fluctuations, hot-wire anemometry, if possible, and flow visualization studies. References 4 and 5 appear to be important contributions in this direction, but they provide no heat transfer determinations.

II - METHOD OF THE EXPERIMENTATION

Details of the main test plate and of the experimental procedure are described in Ref. 1. The essential features are now summarized for convenience.

The chordlength of the basic test plate was 6.732 inches and the thickness was 1.00 inch. An iron channel formed one face and provided structural rigidity. The face towards the wake was instrumented for measuring distributions of temperature and flux. Calibrated thermocouples and Schmidt-type heat-flow meters were used. There were nine chordwise measuring stations, 2/3 inch center to center. They were located on the horizontal plane of symmetry in the tunnel. The wind-tunnel test section was rectangular, 22 inches high and 32 inches wide. The plate spanned the full height of the tunnel. The total span of the plate was electrically heated by a resistance winding. The vertical axis of the plate lay in the vertical plane of symmetry of the tunnel, and the angle of incidence was changed by rotating the plate on that axis.

Each test included the following measurements: distributions of wall temperature and flux; the dynamic and stagnation pressures; and the air stagnation temperature. From these the main heat transfer and flow parameters were calculated.

III - REDUCTION OF THE DATA TO DIMENSIONLESS MODULI

A local coefficient of heat transfer by convection, h, is defined:

$$q'' = h(t_w - t_r) \tag{1}$$

The flux q'' is the metered value corrected for radiation and chordwise conduction [Ref. 1, Eqs (3.7), (4.1), and (4.2)][2]. The maximum total correction was about 4 per cent. The recovery temperature

$$t_r = t_o - 0.9\, U^2/(2c_p) \tag{2}$$

where t_o is the measured stagnation temperature and U is the measured speed in the tunnel. The recovery factor 0.1, which was measured only for the bare plate at 90-degree angle of incidence (Series B) is used throughout (Ref. 1, Section 7).

A Nusselt number is defined:

$$\underline{Nu}_f = \frac{hL}{k_f} \tag{3}$$

where L is the chord length (0.561 foot) and the conductivity is evaluated at the film temperature

$$t_f = (t_w + t_o)/2 \tag{4}$$

A Reynolds number is defined:

$$\underline{Re}_f = \frac{U_\infty\, \rho_f\, L}{\mu_f} \tag{5}$$

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[2] In Eq (4.2) there is a typographical omission: the factor ℓ should appear in the denominator. That correction was not applied in the earlier report. The newly tabulated results of Series B and D now include all the corrections, Also, Table 6 of ARL-4 has been revised to include all the corrections, and the revised table now appears as Table VI of the present report.

The velocity U_{∞} is the wind-tunnel speed of approach, the measured values of speed being reduced 3 per cent where appropriate [Ref. 1, Eq (6.2)]. The density is evaluated at the stagnation pressure and the film temperature, and the viscosity at the film temperature.

The distributions of h in each series are correlated according to the relationship

$$\underline{Nu}_f = C \cdot \underline{Re}_f^{\,n} \tag{6}$$

where C should generally be regarded as a function of location. On the whole, it is found that a satisfactory value of n is 2/3, similarly as in the earlier work.

IV - RESULTS OF THE EXPERIMENTS

Representative results of Series B and of the two tests in Series D are presented in Table I.

In Series B the distribution of C is virtually uniform. The mean value for all metering stations in 17 tests considered most reliable is 0.200. The mean values at each heat-flow meter deviates about 2.5 per cent. They are high at the end stations, where edge corrections are uncertain and have not been applied. The dispersion of the results (ratio of standard deviation to mean value) of the individual tests and at the indivisual stations, also, is about 2.5 per cent. Therefore, for the rear of a flat plate normal to the flow, in the range of the parameters covered by the tests, the results are satisfactorily correlated by

$$\underline{Nu}_f = 0.20\ \underline{Re}_f^{2/3} \quad (7)$$

The distributions of Series D are graphed in Figure 2, where the parameter $\underline{Nu}_f/\underline{Re}_f^{2/3}$ is plotted against the heat-flow meter number. The distance or pitch between adjacent heat flow meters is 2/3 inch, center to center. The splitter plate reduces the over-all heat transfer and induces a maximum rate at the center.

The distributions in Series H, in which the half cylinder was attached to the front of the main test plate, are presented in Table II, and the heat transfer paramenter is presented in Figure 3 for all six tests. This series displays the greatest differences between the distributions of the individual tests, and there is no other constant value of n in Eq (6) that will help correlate the data in a significantly more unified way. In general the local heat transfer reaches a slight maximum near the center, and the over-all heat transfer from the rear of the plate is about 80 per cent of the value found in Series B.

The results of Series J and K show the effects of reducing the angle of incidence. They are presented in Tables III and IV and Figures 4 and 5. The deviation at the second station in Run K-1 is ascribed to mistaken data rather than to any sort of transition. The effect of reducing the angle of incidence is to reduce the over-all heat transfer and to set the distribution askew. In both series the trailing end of the plate, the end associated with the large fraction of the divided flow, exhibits the greater rate of heat transfer. The spread of the data at the trailing

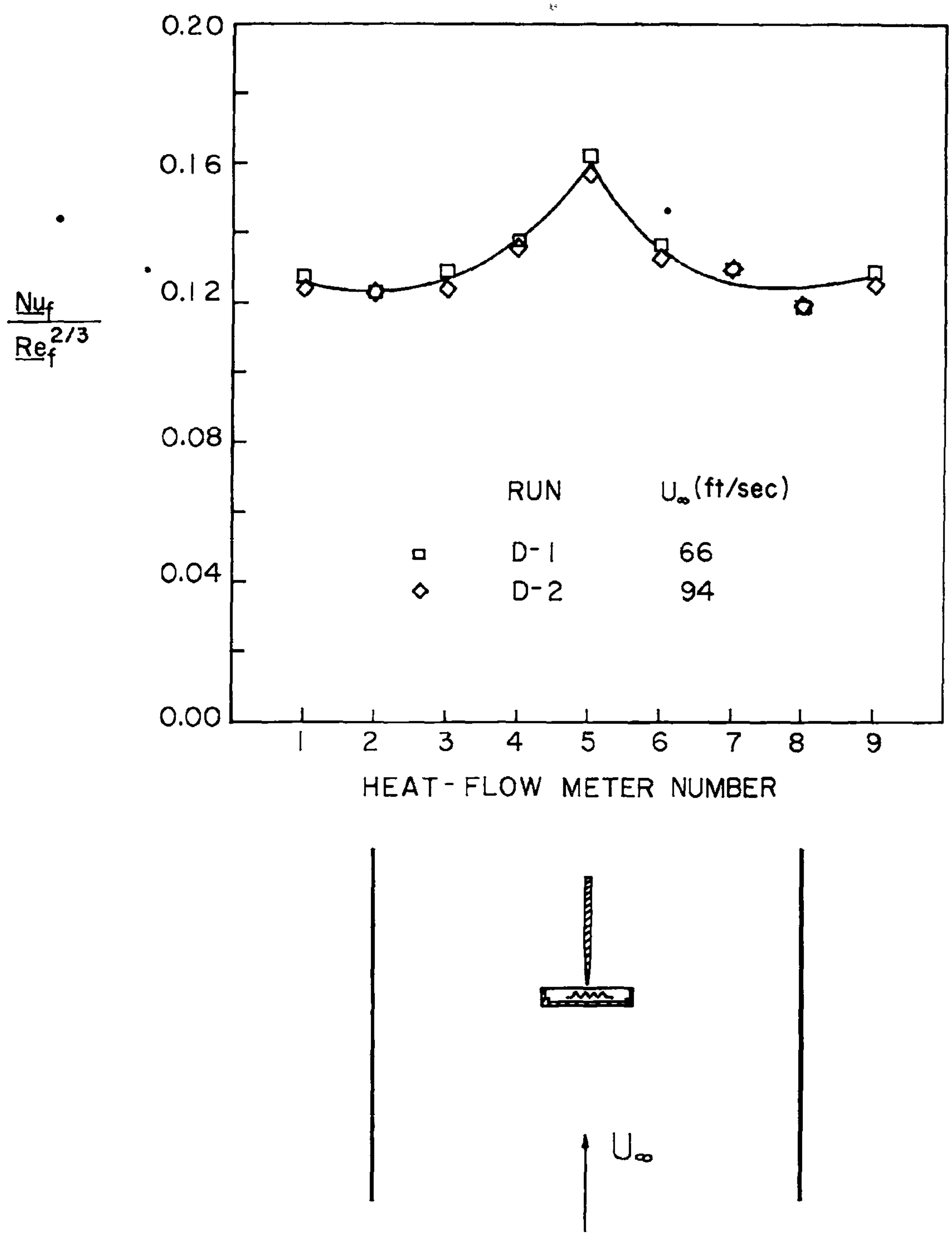
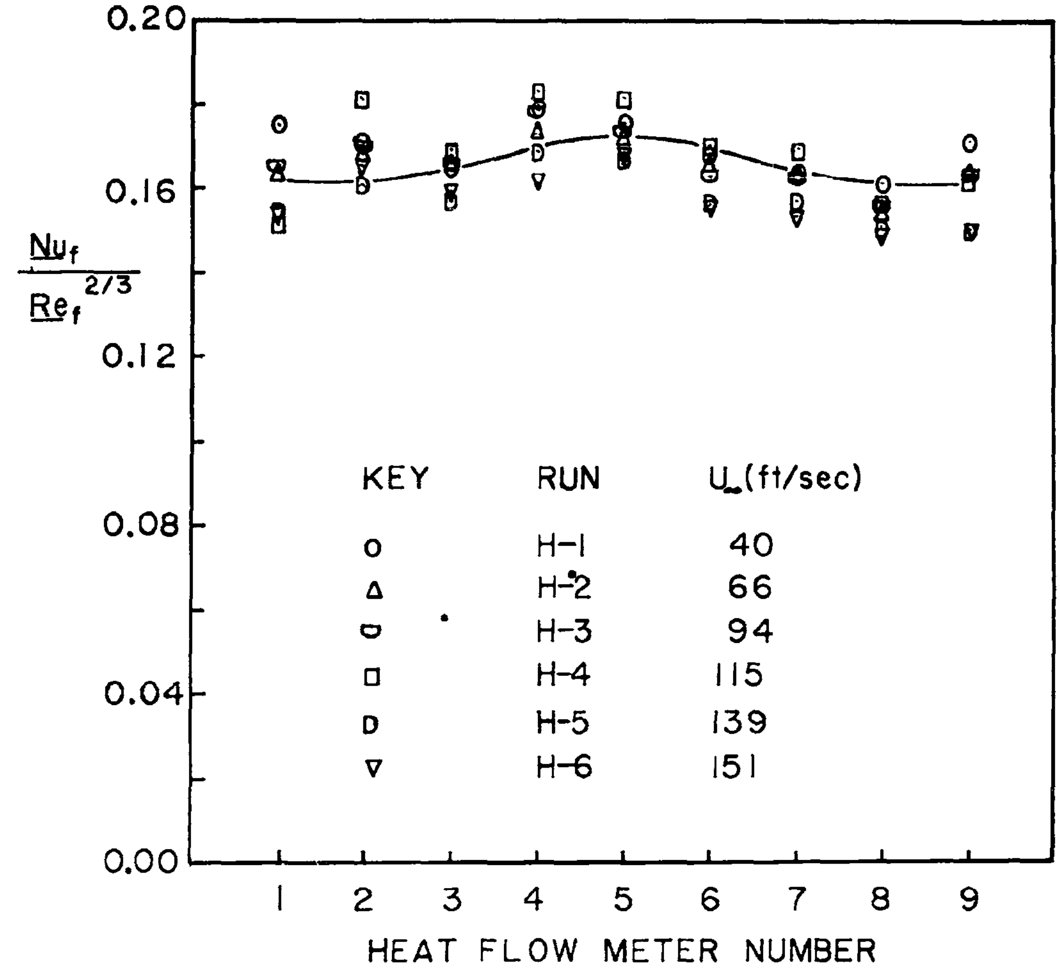


Figure 2. Distribution of the heat transfer parameter $Nu_f/Re_f^{2/3}$ in Series D



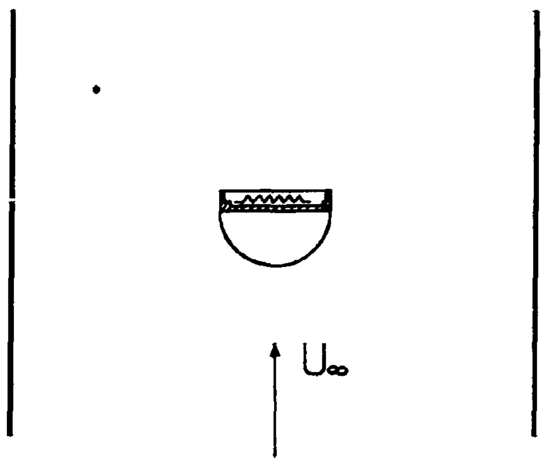


Figure 3. Distribution of the heat transfer parameter $Nu_f / Re_f^{2/3}$ in Series H

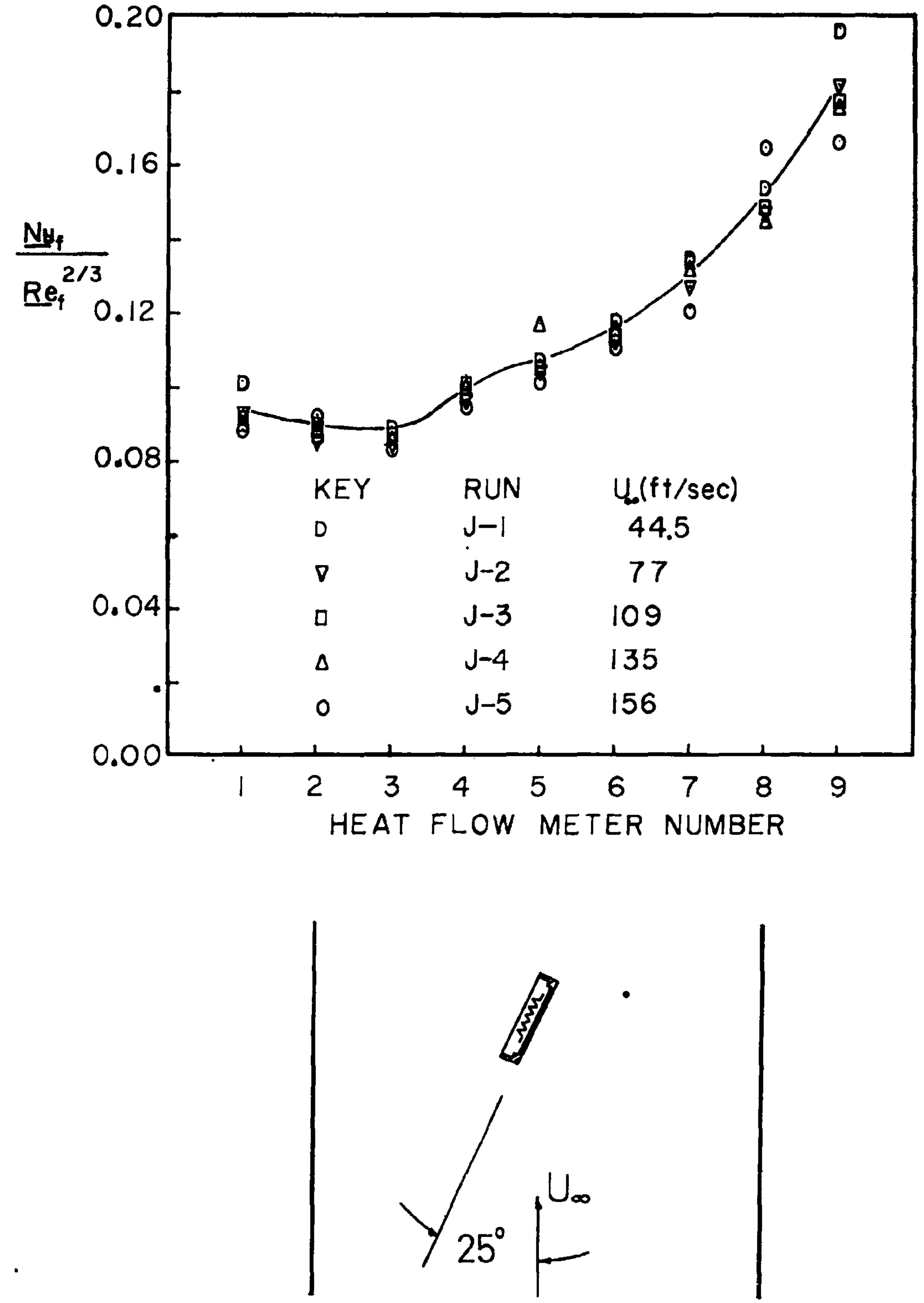


Figure 4. Distribution of the heat transfer parameter $\underline{Nu_f} / \underline{Re_f}^{2/3}$ in Series J

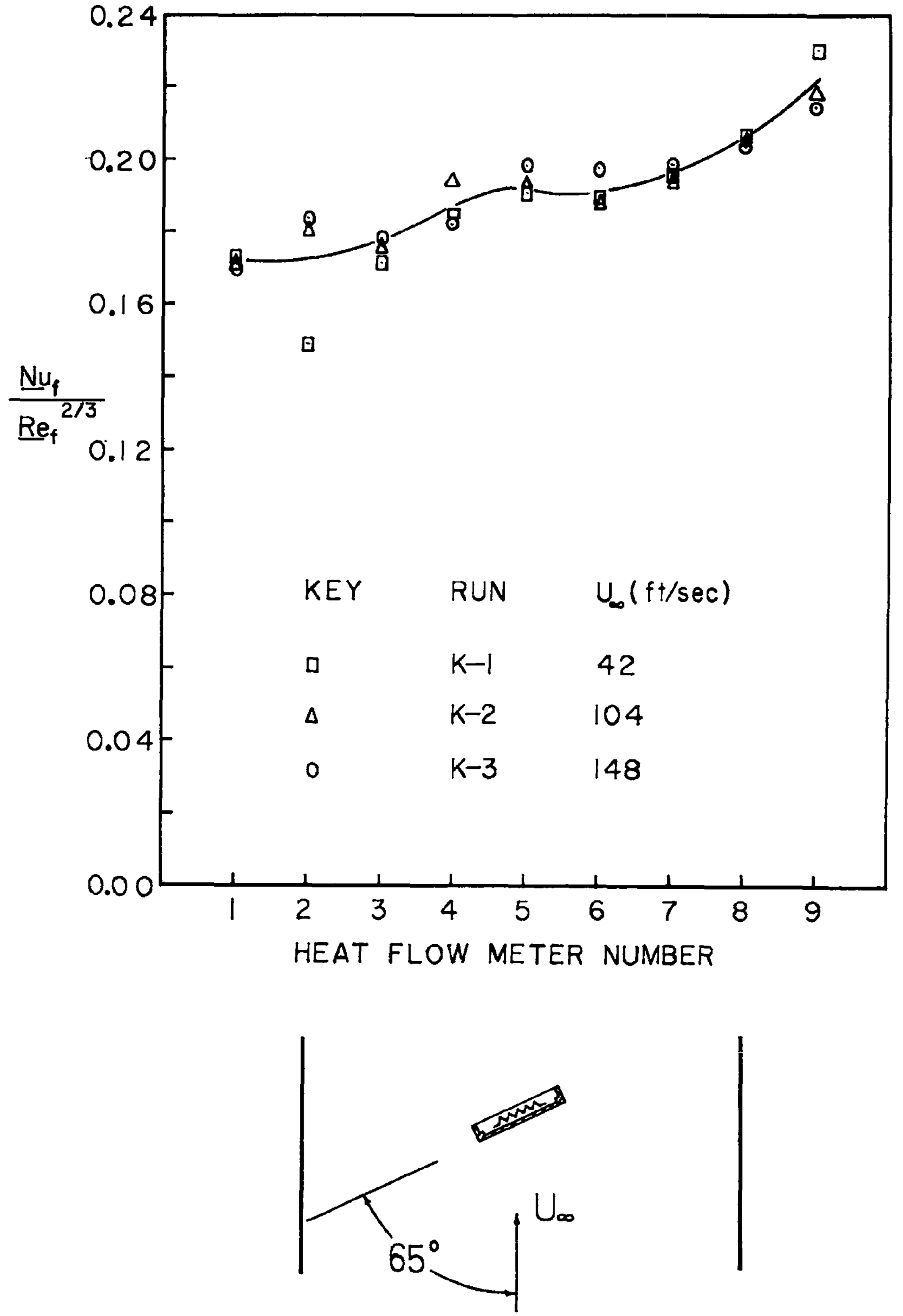


Figure 5. Distribution of the heat transfer parameter $Nu_f/Re_f^{2/3}$ in Series K

end is beyond the experimental error and represents a real departure from the 2/3-power variation. It may possibly be ascribed to alternating separations and reattachments.

The average distributions of the individual series are presented in Table V and depicted in Figure 6. There was an inexplicable asymmetry in Series H; before constructing the representative mean curve the experimental points were reflected about the center line.

V - DISCUSSION OF THE RESULTS

There is no mechanistic analysis of the heat transfer on the solid-wake interface of a blunt obstacle. The governing equations can be written in an extremely general form, but two major difficulties bar the way to construct a mathematical model or to formulate a closed problem in any specific instance. (a) The Reynolds stresses cannot yet be estimated, even on a semi-empirical basis. The motion is not only turbulent, but grossly unsteady, and the unsteadiness is random. Variations with respect to the third dimension probably will have to be admitted even though the configuration is apparently two-dimensional. (b) The boundaries of the region in question, with exception of the interface itself, are nebulous. A practical scheme for specifying reasonable boundaries is needed and probably will be developed empirically. Moreover, the solution has to be joined to the solutions in contiguous regions (boundary layer, mixing layer, or wake) in a fair way.

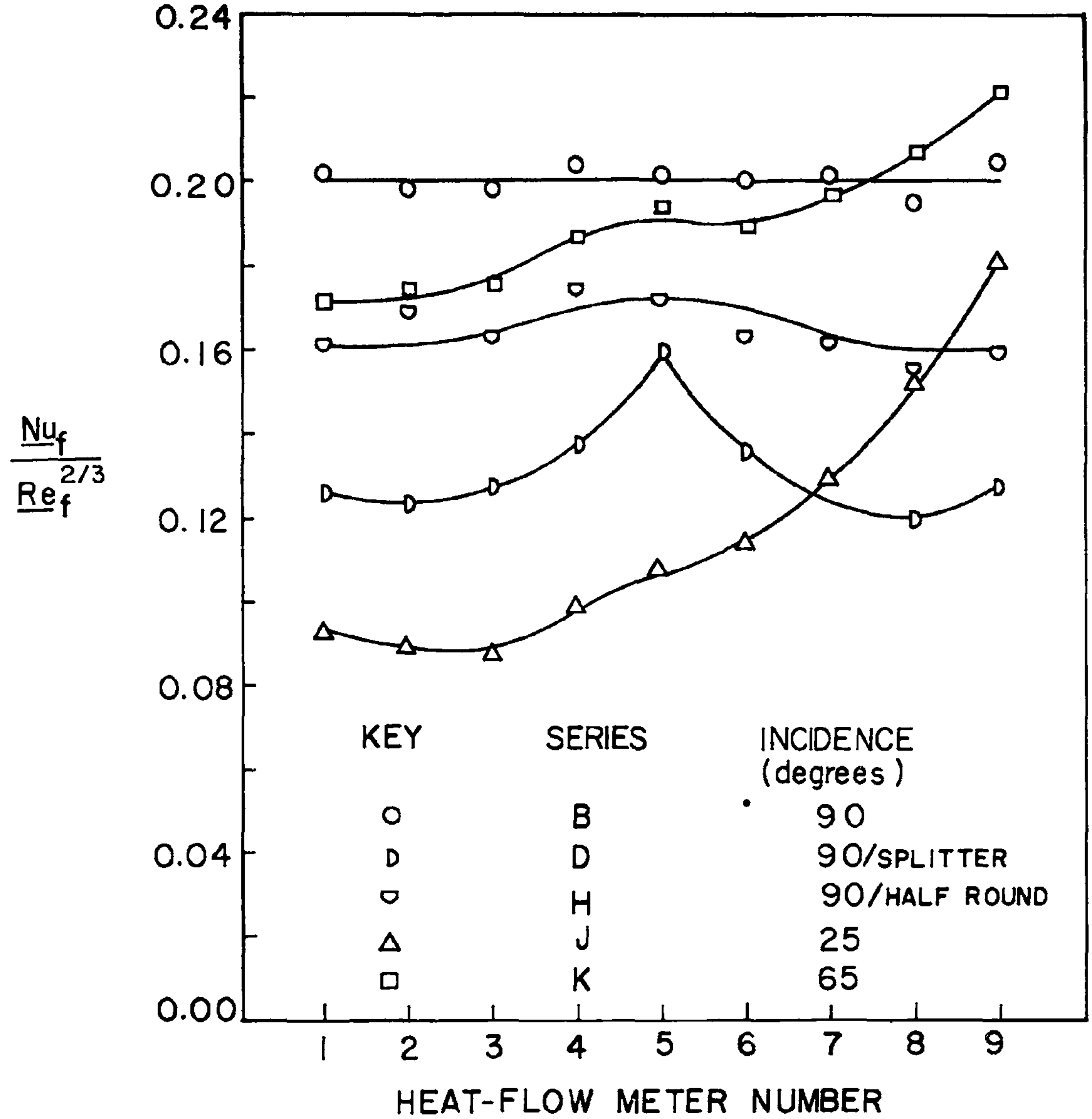


Figure 6. Average distributions in Series B, D, H, J, and K

These difficulties are left to the future.

Meanwhile, Richardson's analogical argument provides a remarkably consistent guide through the trend of the results, if we add a bit of intuition. Briefly, the idea runs as follows. The heat transfer in forced convection appears rather faithfully to follow the empirical rule that $h \sim U^{2/3}$. In certain free-convection configurations, without boundary layers and with apparently autonomous local coefficients, $h \sim (\Delta t)^{1/3}$, or $\underline{Nu} \sim \underline{Ra}^{1/3}$. Now the Rayleigh number, expressed in terms of the terminal velocity for instance, is like the square of a Reynolds number. Hence in both cases $\underline{Nu} \sim \underline{Re}^{2/3}$. This is a remarkable coincidence since in one case the motion arises on account of body forces and in the other is sustained by pressure forces. We note, however, that the two forces are directed toward the heated surface, Roshko (Ref. 6) having shown that the minimum pressure behind a bluff obstacle occurs about two chord lengths downstream from the obstacle (not on the rear surface, as is usually supposed).

Further, Richardson points out that for sufficiently high Rayleigh number in a gas

$$\underline{Nu} \approx \left(\frac{1}{\underline{Ra}_{cr}}\right)^{1/3} \cdot \underline{Ra} \qquad (8)$$

where, for example, $\underline{Ra}_{cr}$ would be the Rayleigh-Jeffreys critical value for the onset of motion in a horizontal layer of fluid heated from below. Extending the analogical argument to the present study, he writes

$$\underline{Nu} \approx \left(\frac{1}{\underline{Re}_{cr}}\right)^{2/3} \cdot \underline{Re}^{2/3} \qquad (9)$$

where $\underline{Re}_{cr}$ is the first critical Reynolds number, the extremely low speed value at which the fluid presumably separates from the rear. The definition of a low speed separation is vague, there is no specific measurement of $\underline{Re}_{cr}$, and it is very difficult to find either experimentally or analytically. In fact the term "separation" may be inappropriate and might better be referred to in terms of an instability. However, in applying Eq (9) as a qualitative guide, it is convenient to think in terms of a very low speed separation.

First consider the flat plate at 90-degree angle of incidence, Series B. By comparison with Eq (7), $\underline{Re}_{cr} \approx 10$. We suppose that once the fluid starts to separate from the rear of this very blunt obstacle it separates everywhere practically at once. Therefore, according to the aforementioned autonomous nature of the transport processes, the heat transfer at high Reynolds number is uniform along the surface. This uniformity is represented by the single line labeled B in Figure 7, in agreement with the experiemntal result from Series B, shown in Figure 6.

As the angle of inclination is reduced, the bluntness diminishes, the hypothetical separation is less abrupt. Consider the flat plate at 65-degree angle of incidence, Series K. We suppose that the fluid separation starts near the trailing edge as sketched in Figure 7(a). The corresponding critical Reynolds number would be at point a in the lower part of the figure. The hypothetical separation progresses along the

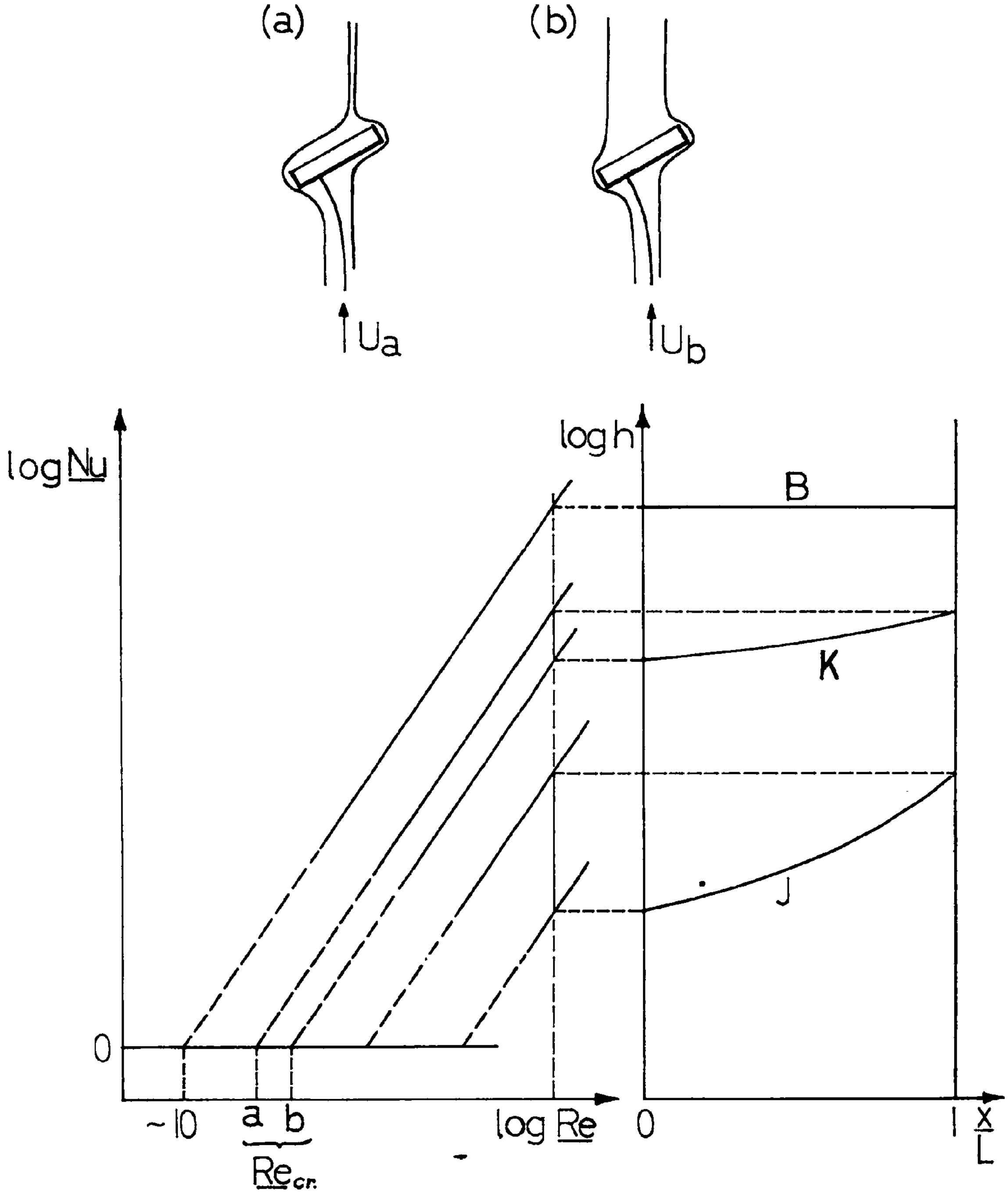


Figure 7. Schematic representation of Richardson's argument

surface as indicated by Figure 7(b), and the final separation occurs at point b in the lower part of the figure. Consequently, the heat transfer behind the leading edge would be reduced. The results are represented by the curve marked K in Figure 7. It is in qualitative agreement with the experimental results. Actually, according to the heat transfer results, $\underline{Re}_{cr,a}$ is also about 10, but the construction in the lower part of Figure 7 has been drawn schematically so that it can be more easily followed; for this reason $\underline{Re}_{cr,a}$ has been displaced to a somewhat higher value. Similar consideration of the flat plate at 25-degree angle of incidence, reinforces this heuristic argument: the total heat transfer diminishes while the variation along the rearward surface increases, as shown by the curve labeled J.

In Series H, one must suppose that rounding the forward surface increases the first critical Reynolds number and that the separation starts at the central region and then spreads abruptly. This would explain the reduction in the over-all heat transfer and the maximum at the center. The significant dispersion of the test results is ascribed to the unsteady and variable position of separation, for the range of Reynolds number includes the critical value of boundary-layer transition and embraces the zone of changing drag. Of course, the fluid at very low Reynolds number is unaware of its misfortunes at high Reynolds number.

Applied to Series D, the argument half succeeds and half fails. On the one hand, the splitter plate delays the separation so that it starts at higher critical Reynolds number and the heat transfer is generally lower than the values found in Series B, in agreement with the results.

On the other hand, the implication is that the very low speed separation starts at the center and gradually moves to the edges, a process that probably would transcend one's intuition in the absence of the experimental results. A better "explanation" is that the splitter plate reduces the amplitude of the gross transverse fluctuations and that there is a reattachment on the splitter plate resulting in a strong reversal of the flow. This might be checked by heating the splitter plate.

With the exception of the above shortcoming, the experimental results are remarkably consistent with the heuristic argument. It is emphasized that the reasoning, such as it is, is intended to be only an interim guide for organizing the results with regard to the trends in the various configurations and that it must not be allowed to obscure a physical explanation of the transport mechanisms.

VI - CONCLUSIONS

Heat transfer experiments were performed on a bluff flat-plate strip in the range of Reynolds number from 100,000 to 440,000. The flow was two-dimensional, and the nominal blockage ratio was 0.211.

When the edges of the plate induce a sharp stable and total (sans reattachment) separation the heat transfer at the rear is proportional to the 2/3 power of the velocity.

The distribution at the rear of the bare plate at 90-degree angle of incidence is virtually uniform, and the over-all heat transfer is greater than in any other of the configurations tested. These results are a convenient basis for comparing the results of the other series discussed in the next paragraphs.

The effect of a symmetrically oriented splitter plate at the rear is to reduce the heat transfer about 35 per cent and to induce a prominent maximum at the center.

The effect of rounding the forward surface is to introduce an unstable separation. As a result the data spread into a band which cannot be reduced by any simple similarity rule, but the 2/3 power serves as a fair approximation. The flux distributions exhibit a slight maximum at the center, and the over-all heat transfer is reduced about 20 per cent from the basic value.

The influence of reducing the angle of incidence is to reduce the over-all heat transfer and to set the distribution askew. The maximum value occurs at the trailing edge.

The numberical findings are of course limited to the range of the investigation and only a mechanistic analysis can generalize them. The formidable difficulties of such an analysis have been pointed out in brief, and in its stead a heuristic, intuitive argument has been suggested for following the trends.

Summarizing the results from the several configurations, we may say that devices which increase the size of, or open, the "dead-air" region increase the heat transfer and those which reduce or confine it reduce the heat transfer. This rule applies to both over-all and local changes of the heat transfer. Accordingly, one should expect that the heat transfer would be increased if the tunnel width were increased or if the plate were arched, concave surface facing the flow.

Numerous questions remain open. Will axisymmetric obstacles follow a 2/3 power rule? What is the effect of changing the size, the location or the orientation of the splitter plate? How would the heat transfer be effected if the splitter plate were attached to one edge, as in a rearward facing step? How strongly would the forward face of the flat plate have to be heated before influencing the heat transfer on the rearward face? Over what angle at the rear of a cylinder should we expect the heat transfer to be practically independent of the forward temperature distribution? What is the influence of the Prandtl number?

These questions and others could be answered by the type of experimentation described in this and the previous reports. However, to achieve a rational knowledge of the transport processes, deeper probing is needed. A workable semi-empirical description of the mechanics of the "dead-air" region is needed. Therefore, future studies should be augmented by a more probing type of dynamic mechanical and thermal instrumentation. The Reynolds number range should be extended: although there is much current interest in high speeds and high Reynolds number, much may be gained by reducing the model size and working at lower Reynolds number.

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Table I. Data of Series B and D: Flat plate strip at 90° angle of incidence without and with splitter plate, respectively.

	Run Number					
	B-1	B-3	B-5	B-22	D-1	D-2
Heater temp., t_h (°F)	148.6	140.7	138.5	190.4	168.6	164.1
Air temp., t_o (°F)	74.5	75.4	80.9	82.1	77.6	83.5
Mean Recovery temp., t_r (°F)	74.2	74.7	79.6	81.4	77.2	82.8
Velocity, U_∞(ft/sec)	63.9	89.9	127	90.5	66.2	94.3
10^{-3} x $\underline{Re}_f$	203	290	410	275	203	288

Distributions of convective heat flux (B/hr ft²), wall temperature (°F), and $\underline{Nu}_f$

Heat-flow meter		B-1	B-3	B-5	B-22	D-1	D-2
1	q"	872	902	883	1448	801	827
	t_w	118.8	111.4	109.1	142.3	141.9	136.9
	$\underline{Nu}_f$	704	890	1081	835	437	539
2	q"	872	892	894	1420	790	831
	t_w	119.6	111.7	109.4	143.3	142.8	137.5
	$\underline{Nu}_f$	691	873	1084	806	425	535
3	q"	868	884	896	1444	822	840
	t_w	120.0	112.0	109.6	143.6	142.3	137.7
	$\underline{Nu}_f$	682	858	1079	815	445	539
4	q"	870	904	909	1468	862	888
	t_w	119.4	111.3	109.0	142.6	140.9	135.8
	$\underline{Nu}_f$	693	894	1117	843	477	591
5	q"	873	892	896	1449	980	993
	t_w	119.6	111.6	109.3	142.7	138.8	133.8
	$\underline{Nu}_f$	691	875	1090	830	560	686
6	q"	876	896	900	1446	859	876
	t_w	119.6	111.6	109.4	142.7	141.2	136.1
	$\underline{Nu}_f$	695	879	1091	829	474	581
7	q"	878	897	900	1448	830	864
	t_w	119.6	111.4	109.2	142.5	142.3	136.7
	$\underline{Nu}_f$	696	885	1098	832	449	565
8	q"	852	872	883	1402	774	819
	t_w	120.1	112.0	109.6	143.5	143.5	137.9
	$\underline{Nu}_f$	668	846	1063	794	412	523
9	q"	889	907	894	1482	822	842
	t_w	119.0	111.1	109.2	141.6	142.5	137.3
	$\underline{Nu}_f$	714	902	1091	865	444	544

Table II. Data of Series H: Half cylinder attached to front of basic flat plate strip.

	Run Number					
	H-1	H-2	H-3	H-4	H-5	H-6
Heater temp., t_h (°F)	186.0	168.5	160.5	154.2	156.2	158.6
Air temp., t_o (°F)	83.5	83.1	85.9	84.3	89.8	93.9
Recovery temp., t_r (°F)	83.4	82.8	85.2	83.3	88.4	92.2
Velocity of approach, U_∞(ft/sec)	40.0	65.7	94.1	115	139	151
$10^{-3} \times \underline{Re}_f$	117	198	286	356	424	457

Distributions of convective heat flux, (B/hr ft^2) wall temperature (°F), and $\underline{Nu}_f$

Heat-flow meter		H-1	H-2	H-3	H-4	H-5	H-6
1	q''	879	899	933	934	934	926
	t_w	156.2	139.4	130.8	127.3	126.1	128.2
	$\underline{Nu}_f$	419	559	723	755	877	905
2	q''	870	926	952	1011	965	976
	t_w	157.0	139.5	130.7	122.9	125.8	127.3
	$\underline{Nu}_f$	410	575	739	909	912	978
3	q''	848	921	940	959	953	964
	t_w	158.1	140.2	131.3	123.6	126.3	128.2
	$\underline{Nu}_f$	394	564	721	847	890	943
4	q''	910	948	987	1014	993	980
	t_w	157.0	139.1	130.2	122.4	125.1	128.3
	$\underline{Nu}_f$	429	592	777	923	958	956
5	q''	900	944	966	1011	981	1009
	t_w	157.9	139.5	131.1	122.9	125.5	127.9
	$\underline{Nu}_f$	419	586	744	908	936	995
6	q''	870	919	931	973	950	954
	t_w	158.5	140.2	131.8	123.9	126.4	128.5
	$\underline{Nu}_f$	402	563	706	854	883	925
7	q''	850	909	937	969	949	945
	t_w	158.6	140.3	131.7	123.8	126.3	128.7
	$\underline{Nu}_f$	392	556	712	852	886	912
8	q''	837	878	910	922	925	934
	t_w	158.7	140.8	132.1	124.7	126.6	129.0
	$\underline{Nu}_f$	386	532	686	792	856	894
9	q''	865	895	915	928	906	917
	t_w	156.8	139.4	131.1	123.9	126.1	128.5
	$\underline{Nu}_f$	409	556	705	813	850	889

Table III. Data of Series J: Flat plate strip at 25° angle of incidence.

		J-1	J-2	J-3	J-4	J-5
		Run Number				
Heater temp., t_h (°F)		192.7	178.8	167.2	173.8	167.2
Air temp., t_o (°F)		76.5	81.8	83.1	95.8	92.1
Recovery temp., t_r (°F)		76.5	81.3	82.2	94.4	90.3
Velocity, U_∞ (ft/sec)		44.5	77.3	109	135	156
10^{-3} x $\underline{Re_f}$		130	228	328	395	462
Distributions of convective heat flux (B/hr ft^2), wall temperature (°F), and $\underline{Nu_f}$						
Heat-flow meter						
1	q''	710	732	754	772	774
	t_w	170.0	154.9	142.8	148.7	141.6
	$\underline{Nu_f}$	262	346	437	492	526
2	q''	647	698	742	762	792
	t_w	172.3	156.1	143.7	150.0	140.9
	$\underline{Nu_f}$	233	324	423	474	546
3	q''	643	704	732	759	756
	t_w	173.2	156.2	144.2	150.6	142.8
	$\underline{Nu_f}$	230	327	414	467	502
4	q''	709	772	808	835	844
	t_w	171.2	154.5	142.1	148.3	140.5
	$\underline{Nu_f}$	258	367	474	537	587
5	q''	747	813	851	878	878
	t_w	170.0	153.4	140.8	147.0	139.9
	$\underline{Nu_f}$	276	393	510	578	618
6	q''	804	846	865	904	912
	t_w	168.0	151.9	139.2	145.4	138.3
	$\underline{Nu_f}$	304	418	534	616	665
7	q''	870	916	969	978	940
	t_w	164.5	148.9	135.9	142.6	135.8
	$\underline{Nu_f}$	345	474	637	706	724
8	q''	958	1008	1024	1023	1103
	t_w	160.7	145.4	133.3	140.6	129.6
	$\underline{Nu_f}$	396	551	709	771	988
9	q''	1132	1141	1126	1129	1122
	t_w	154.3	140.2	129.2	135.8	129.8
	$\underline{Nu_f}$	509	682	850	953	1000

Table IV. Data of Series K: Flat plate strip at 65° angle of incidence.

	Run Number		
	K-1	K-2	K-3
Heater temp., t_h (°F)	174.9	150.5	149.4
Air temp., t_o (°F)	83.5	83.5	89.4
Recovery temp., t_r (°F)	83.4	82.8	87.7
Velocity of approach, U (ft/sec)	42.3	104	148
10^{-3} x $\underline{Re}_f$	132	332	467

Distributions of convective heat flux (B/hr ft^2), wall temperature (°F), and $\underline{Nu}_f$

Heat-flow meter		K-1	K-2	K-3
1	q"	837	888	901
	t_w	148.8	121.5	118.8
	$\underline{Nu}_f$	447	817	1025
2	q"	778	934	951
	t_w	153.3	121.0	118.3
	$\underline{Nu}_f$	387	871	1107
3	q"	848	923	940
	t_w	150.0	121.5	119.0
	$\underline{Nu}_f$	444	848	1068
4	q"	888	977	968
	t_w	148.0	120.2	119.1
	$\underline{Nu}_f$	480	930	1097
5	q"	905	972	1010
	t_w	147.5	120.1	117.7
	$\underline{Nu}_f$	494	928	1199
6	q"	897	946	962
	t_w	147.3	120.1	118.0
	$\underline{Nu}_f$	491	903	1131
7	q"	921	966	983
	t_w	146.4	119.6	117.3
	$\underline{Nu}_f$	511	935	1183
8	q"	935	992	998
	t_w	144.8	118.8	116.7
	$\underline{Nu}_f$	534	981	1227
9	q"	1002	1002	1006
	t_w	142.0	117.0	115.5
	$\underline{Nu}_f$	600	1046	1291

Table V. Distributions of mean values of $\underline{Nu}_f/\underline{Re}_f^{2/3}$.

Heat-flow meter no.	Series B	D	H	J	K
1	0.202	0.126	0.161	0.093	0.171
2	0.198	0.123	0.170	0.089	0.172
3	0.198	0.126	0.164	0.087	0.175
4	0.204	0.137	0.175	0.099	0.187
5	0.201	0.160	0.172	0.108	0.194
6	0.200	0.135	0.163	0.114	0.189
7	0.201	0.130	0.162	0.129	0.196
8	0.195	0.120	0.156	0.152	0.205
9	0.205	0.127	0.160	0.181	0.221

Table VI. Data and results of Series B, S, D, and T -- Corrected averages. (Table 6 of ARL-4, revised).

Run No.	t_o (°F)	$\overline{t_w}$ (°F)	$\overline{t_h}$ (°F)	$\overline{q''}$ (B/hr ft^2)	U_∞ (ft/sec)	$\overline{Nu_f}$	$10^{-3} \times Re_f$	$\overline{Nu_f} / Re_f^{2/3}$
B-1	74.5	119.5	148.6	873	63.9	694	203	0.201
B-2	74.6	136.4	164.9	830	36.6	476	113	0.204
B-3	75.4	111.6	140.7	895	89.9	878	290	0.200
B-4	77.8	109.1	138.6	892	110	1000	356	0.199
B-5	80.9	109.3	138.5	895	127	1088	410	0.197
B-6	76.6	118.1	137.7	572	37.5	495	119	0.205
B-7	72.1	101.6	121.2	590	63.4	722	209	0.205
B-8	72.2	95.5	115.4	599	89.7	922	298	0.207
B-9	82.5	90.4	101.0	300	127	1194	424	0.212
B-10	82.9	91.9	102.6	302	110	1105	365	0.216
B-11	80.9	91.6	102.3	303	90.3	974	299	0.218
B-12	80.3	94.3	105.0	298	63.8	763	210	0.216
B-13	79.4	99.2	110.2	289	37.6	529	123	0.214
B-14	82.3	102.1	122.3	604	110	1053	359	0.208
B-15	83.9	102.2	122.2	607	127	1122	415	0.202
B-16	79.4	117.5	156.2	1187	127	1082	405	0.198
B-17	80.4	122.2	160.5	1174	110	981	347	0.199
B-18	78.6	127.0	165.4	1171	90.4	852	282	0.198
B-19	77.6	138.2	176.0	1136	63.8	660	196	0.196
B-20	82.6	131.0	179.4	1485	127	1085	398	0.201
B-21	82.8	135.8	183.7	1471	110	962	341	0.197
B-22	82.1	142.7	190.4	1448	90.5	830	275	0.196
B-23	79.2	138.7	196.7	1785	127	1037	392	0.194
S-1	74.3	142.1	170.0	813	30.5	423	90.3	0.210
S-2	75.4	125.4	154.4	870	54.4	621	166	0.206
S-3	77.4	117.5	146.5	891	76.9	790	237	0.206
S-4	79.7	114.6	144.2	901	94.2	911	291	0.207
S-5	82.3	113.7	143.3	902	108	1003	334	0.208
S-6	78.1	121.3	149.9	861	64.8	709	201	0.207
S-7	80.3	115.2	144.5	891	91.2	890	285	0.205
S-8	77.5	123.2	152.0	867	61.0	675	187	0.206
S-9	74.0	119.3	147.8	862	62.2	680	196	0.202
S-10	78.5	123.0	151.6	856	63.7	684	198	0.201
S-11	81.0	126.5	155.4	863	63.8	671	194	0.200
D-1	77.6	141.9	168.6	810	66.2	441	203	0.127
D-2	83.5	136.6	164.1	840	94.3	551	288	0.126
T-1	84.1	121.0	149.7	890	91.6	845	286	0.195

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