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TECHNICAL NOTE

D-130

USE OF A THEORETICAL FLOW MODEL TO CORRELATE DATA FOR FILM COOLING OR HEATING AN ADIABATIC WALL BY TANGENTIAL INJECTION OF GASES OF DIFFERENT FLUID PROPERTIES

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SUMMARY

An equation is derived, by use of a simplified theoretical flow model, that predicts the temperature of a film-cooled wall within 5 percent for a range of cooling effectiveness from approximately 0.2 to 1.0. The equation is easily solved for required coolant flow rate by a simple iteration process involving continuity. Both helium and air were used as the coolant gas. Data from the General Electric Company are also presented; these data are from a larger air-film-cooling facility and cover lower gas-stream temperatures and velocities than those of the NASA tests. The range of variables covered by the NASA and General Electric data is as follows: main-gas-stream temperature, 502° to 1965° R; main-gas-stream velocity, 104 to 1040 feet per second; velocity ratio V_g/V_c, 0.45 to 33.3; injected gas temperature, 530° to 980° R; and coolant slot height, 0.063 to 0.50 inch.

INTRODUCTION

Various empirically derived parameters have been used in the past to correlate data for film-cooling an adiabatic wall using air as a coolant (refs. 1 to 4). These correlations apply only for use of air as the coolant and, with the exception of reference 4, apply only for a relatively low temperature gas stream. Unfortunately, the empirically derived correlating parameters are in such a form that they are not easily or universally applicable to many design problems.

The purpose of this report is to present a correlation of film-cooling data that contains coolant flow rate directly and applies over a wide range of main-gas-stream temperatures and velocities. Since this correlation also applies when two coolants of widely differing gas properties (air and helium) are used, it may be inferred that the correlation would hold for any gas in the absence of chemical reactions. Data obtained from the General Electric film-cooling facility of a larger size are also correlated.

The pertinent correlating parameters were developed by employing a highly simplified heat-flow process for the model in deriving the differential equation. The resulting equation was solved, and experimental data were used to refine this equation for use in design applications.

The present work is limited to tangential cooling slots, but it possibly could be extended to nontangential slots by incorporating an angle factor in the parameters.

DERIVATION OF EQUATION

In the derivation, the flat plate was used as the cooled model (fig. 1). The increment of cooled area was $L\,\Delta x$, and the assumptions were as follows (all symbols are defined in the appendix):

(1) The coolant film exists as a discrete layer (no mixing).

(2) The temperature profile in the coolant film does not change rapidly with x.

(3) The temperature gradient through the coolant is small.

(4) No heat is conducted through or along the wall.

(5) Conditions are uniform in the y-direction.

(6) The main gas stream is fully developed turbulent flow. The property values involved in calculating the heat-transfer coefficient, $h = 0.0265 \frac{k_f}{D_h} (Re)_f^{0.8}(Pr)_f^{0.3}$, are evaluated at the arithmetic mean of the static gas temperature t_g and the coolant slot exit static temperature t_c.

(7) The gas traverses the distance x' before heat diffuses through the gas layer to raise the wall temperature.

From figure 1, the heat balance for the incremental distance Δx along the plate may be written as

$$(\dot{w}c_p)_c(\bar{t}_2 - \bar{t}_1) = Ah(T_{ad} - \bar{t}) = L\,\Delta x\, h(T_{ad} - \bar{t}) \tag{1}$$

By means of assumptions 2 and 3 the following substitutions may be made in equation (1):

$$\bar{t}_2 - \bar{t}_1 \approx t_{w,2} - t_{w,1}$$

and

$$\bar{t} \approx t_w$$

The result is:

$$\left.\frac{t_{w,2} - t_{w,1}}{\Delta x}\right|_{\substack{\lim \\ \Delta x \to 0}} = \frac{dt_w}{dx} = \frac{hL}{(\dot{w}c_p)_c}(T_{ad} - t_w)$$

Since $T_{ad} \neq f(x)$, this may be written as

$$\frac{1}{T_{ad} - t_w} d(T_{ad} - t_w) = - \frac{hL\ dx}{(\dot{w}c_p)_c} \qquad (2)$$

If equation (2) is integrated and the limits substituted between the point x' ($t_w = t_c$ up to this point) and the point x, the result is

$$\ln \frac{T_{ad} - t_w}{T_{ad} - t_c} = - \frac{hL(x - x')}{(\dot{w}c_p)_c} \qquad (3)$$

or

$$\frac{T_{ad} - t_w}{T_{ad} - t_c} = \eta = e^{-\frac{hL(x - x')}{(\dot{w}c_p)_c}} \qquad (4)$$

This development is obviously highly simplified and probably does not contain all pertinent parameters. However, equation (4) meets the requirement that, as x approaches infinity, η approaches zero or the wall temperature approaches the adiabatic wall temperature (or recovery temperature) of the main gas stream. For values of $x \lesseqgtr x'$ the wall temperature has not increased above the coolant temperature, so that $\eta = 1$ for $x \lesseqgtr x'$.

The next step was to use the parameters in equation (3) for plotting the experimental data to determine whether they were correlated and, if not, what modifications were needed.

CORRELATION PROCEDURE

Before proceeding with the correlation of experimental data, it was necessary to determine what factors affect the length of x'. Figure 18 of reference 5 indicates that, for one-dimensional conduction through a material, the value of dimensionless time $\alpha\theta/S^2$ required for heat to reach the back side of a slab S units thick ranges from approximately 0.05 to 0.2 for a range of hS/k from infinity to 0.05, respectively. Obviously, this gas film probably doesn't very closely approximate the one-dimensional conduction case, since there may be considerable conduction along the x direction in the coolant and mixing between the coolant and main gas streams. However, the parameters may be considered as significant and the data used to determine unknown constants. If

$$\frac{\alpha_c \theta}{S^2} = \frac{C}{f(hS/k_c)}$$

the value of x' may be determined as follows for the flowing coolant gas:

$$x' = V_c \theta = C\left(\frac{S^2 V_c}{\alpha_c}\right) f\left(\frac{1}{hS/k_c}\right) \tag{5}$$

The data showed that equation (5) can be expressed as

$$x' = 0.04\left(\frac{S^2 V_c}{\alpha_c}\right)\left(\frac{1}{hS/k_c}\right) \tag{6}$$

If equation (6) is substituted in equation (3) and expanded, a rather amazing result is noted. Since $(S^2 V_c/\alpha_c)[1/(hS/k_c)]$ is the reciprocal of $hL/(\dot{w}c_p)_c$, the resulting equation is

$$\ln \eta = -\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right] \tag{7}$$

Furthermore,

$$x' = 0.04\left(\frac{S^2 V_c}{\alpha_c}\right)\left(\frac{1}{hS/k_c}\right) = 0.04\,\frac{(\dot{w}c_p)_c}{hL} \tag{8}$$

Data obtained from the NASA film-cooling facility described in reference 4 were plotted as η against $h\,\frac{Lx}{(\dot{w}c_p)_c} - 0.04$ on semilog coordinates. The data were taken for a wide range of velocity ratios V_g/V_c for fixed values of coolant slot height S and main-gas-stream Mach number and temperature. Several tests were made with different values of slot height and main-gas-stream Mach number and temperature. Figure 2 presents η plotted against $\frac{hLx}{(\dot{w}c_p)_c} - 0.04$ on semilog coordinates for a velocity ratio of approximately 1.0 for several runs. Considerable scatter is present. However, it appears that the approximation for x' is a good one as evidenced by the fact that all curves reach a value of unity as the abscissa approaches zero. The plots of the preceding parameters indicated that, for a velocity ratio of 1.0, the data from various tests could be made to fit the equation

$$\ln \eta = -\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right]\left(\frac{SV_c}{\alpha_c}\right)_{V_c=V_g}^{0.125} \tag{9}$$

by incorporating the dimensionless parameter SV_c/α_c. The thermal diffusivity coefficient for the coolant was evaluated from reference 6 at the coolant slot exit static temperature. This parameter may be regarded as the ratio of directed molecular velocity to heat-diffusion velocity; and, as its value approaches unity, equation (9) more nearly approximates the theoretical equation (3). Physically it may be regarded as implying that the narrower the coolant gas-stream slot S, the lower the coolant velocity, or the higher the coolant thermal diffusivity, the more effectively the whole mass of the coolant is absorbing heat. Figure 3 gives the same data shown in figure 2, but the abscissa has been multiplied by $(SV_c/\alpha_c)_{V_c=V_g}^{0.125}$. The solid line in this figure represents equation (9). The data now appear to have the same slope as equation (9) over the probable usable range of values of η, and the variation from the line representing equation (9) is small.

Typical data for one run, which covered a wide range of velocity ratios, were then plotted in figure 4 as η against $\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right]\left(\frac{SV_c}{\alpha_c}\right)^{0.125}$ on semilog coordinates. The spread of the data indicates a dependency on velocity ratio. It can be seen that the cooling effectiveness η decreases for velocity ratios that differ from 1.0. This is possibly explainable by the greater mixing between the coolant and main gas streams when their velocities are not equal. The next step was to determine simple empirical functions of velocity ratio $f(V_g/V_c)$ that would correlate the data for velocity ratios in the neighborhood of 1.0. The parameters developed were as follows:

$$f\left(\frac{V_g}{V_c}\right) = 1 + 0.4 \tan^{-1}\left(\frac{V_g}{V_c} - 1\right) \qquad \text{when} \quad V_g/V_c \gtreqqless 1.0 \tag{10}$$

(The angle in equation (10) is expressed in radians.)

$$f\left(\frac{V_g}{V_c}\right) = \left(\frac{V_c}{V_g}\right)^{1.5(V_c/V_g-1)} \qquad \text{when} \quad V_g/V_c \lesseqqgtr 1.0 \tag{11}$$

The data were then computed and plotted as η against $\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right]\left(\frac{SV_c}{\alpha_c}\right)_{V_c=V_g}^{0.125} f\left(\frac{V_g}{V_c}\right)$ on semilog coordinates. Figure 5 presents the data from figure 4 with the $f(V_g/V_c)$ correction applied; it may be seen that the scatter is reduced and the data points fall more nearly on the line representing the equation.

PRESENTATION AND DISCUSSION OF DATA

The data correlation is presented in figures 6 to 14. These figures may be divided into three general groups: The first group, covering tests made using the NASA film-cooling facility with air as the coolant, is presented in figures 6 to 9. The second group presents data obtained on the NASA facility with helium as the coolant and is presented in figures 10 to 12. General Electric Company allowed presentation of their data, which may be found in reference 3, for comparison with a facility of different geometry. This is the third group, and the data are presented in figures 13 and 14. Also shown on all figures is a line whose equation is

$$\ln \eta = -\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right]\left(\frac{SV_g}{\alpha_c}\right)^{0.125} f\left(\frac{V_g}{V_c}\right) \tag{12}$$

which applies for $\frac{hLx}{(\dot{w}c_p)_c} \gtreqqless 0.04$. It should be noted that the gas velocity V_g is used in place of V_c in $\frac{SV_c}{\alpha}$ (velocity ratio of 1.0) and $f\left(\frac{V_g}{V_c}\right)$ corrects for velocity ratios that differ from 1.0.

$$\eta = 1 \quad \text{for} \quad \frac{hLx}{(\dot{w}c_p)_c} \lesseqqgtr 0.04 \tag{12a}$$

NASA Data with Air as Coolant

Figures 6 to 9 indicate that the data closely follow equation (12) down to values of η of approximately 0.3 with slot heights ranging from 1/2 to 1/16 inch. For values of η below 0.3, the answers obtained from equation (12) would be pessimistic. Most of the scatter on these plots occurs for velocity ratios that are much higher or lower than 1.0. The data in figure 9 are for a slot height of 1/16 inch and show considerable scatter. This may be due to the large values of dt_w/dx to be found with a very narrow slot and due to the possibility of large variation of results caused by distortions of the slot when high gas temperatures are used. Figure 6(b) presents data obtained when the air temperature discharged from the slot is higher than the main-gas-stream temperature; these data agree well with equation (12). Reference 7 indicates that a slightly different form of the equation for computing heat-transfer coefficient should be used when heat flow is from the wall to the gas stream. However, in this correlation the distinction was not made, since it was felt that the differences would be small.

NASA Data with Helium as Coolant

The data obtained when helium was used as the coolant gas are presented in figures 10 to 12 for slot heights ranging from 1/2 to 1/8 inch. These data fit equation (12) as well as the data obtained when air was used as the coolant. Again, most of the scatter occurs for velocity ratios that are appreciably different from 1.0. These data have not been previously presented in the literature and are found in table I.

Since the properties of helium (density, specific heat, and thermal conductivity) are markedly different from air, it may be inferred that the correlation would apply for any gas in the absence of chemical reactions.

General Electric Data with Air as Coolant

In order to determine if the correlation would hold for a test facility of different size, permission was obtained from the General Electric Company to present their data of reference 3. Their test section was 18 inches square (D_h = 18 in.), compared with the 8-inch-square test section for the NASA facility. The slot height S in the General Electric test section was 0.106 inch.

Data obtained with air as the coolant are presented in figure 13. Figure 13 illustrates the agreement of the data with equation (12) when the slot air is cooler than the main stream. It should be noted that the agreement is good down to cooling effectiveness of approximately 0.3 to 0.4.

In the film-heating cases, the slot air was hotter than the main gas stream, and data are presented in figure 14. The data of these two figures are somewhat below equation (12), especially for figure 14(b). However, the slope of the data appears to be the same as equation (12), and the various velocity ratios are correlated.

ACCURACY OF CORRELATING EQUATION

From the presentation of figures 6 to 14 it is difficult to tell what the deviation of the data from the line depicting equation (12) represents as a percentage error in wall temperature or coolant flow. Consequently, representative data were selected to present curves to show the magnitude of these errors. The deviation of wall temperature calculated by equation (12) from the measured wall temperature for the data presented in figures 7(a), 7(b), and 10(a) is found in figure 15. The deviation is roughly ±5 percent for all three sets of data. Although the data appear to depart noticeably from equation (12) for low values of effectiveness (high t_w) in figures 7(a), 7(b), and 10(a), the error in wall temperature is approximately only 5 percent.

Figure 16 presents the deviation of measured coolant flow from the coolant flow determined by equation (12). It should be noted that deviations as large as 60 percent are obtained by using the data of figures 7(a), 7(b), and 10(a). This is understandable, since at values of the abscissa near zero small variations of the measured flow from the computed value result in large percentage deviations.

APPLICATION OF EQUATION AND EXAMPLE

The approach to using equation (12) in a design application depends on what factors are known. If it is required to find the coolant flow when the coolant and gas velocities are specified, both coolant flow and slot height are unknowns and an iteration procedure is required. However, $(SV_g/\alpha_c)^{0.125}$ is very insensitive to changes in the variables. Therefore, the logical approach would be to assume a slot height and compute a coolant flow $\dot{w}_c$, and then see what slot height is required to satisfy continuity and use this value in the second iteration. The second calculation should give approximately the right answer. It should be noted from equations (10), (11), and (12) that the minimum coolant flow rate for a particular cooling job is attained when the velocity ratio is 1.0. If the main gas stream is near or above sonic velocity, it may be necessary to use a velocity ratio greater than 1.0 because of choking in the coolant slot.

For example, suppose that it is necessary to find the coolant flow rate required to cool a point 2 feet from the coolant slot to 800° R when air at a static temperature of 1200° R is flowing in a pipe 2 feet in diameter. The gas velocity is 2000 feet per second and the pressure in the pipe is 14.7 pounds per square inch absolute. The velocity and temperature of the cooling air are 1000 feet per second and 540° R, respectively.

Equation (12) may be solved for coolant flow rate, and the result is

$$\dot{w}_c = \frac{hLx}{c_{p,c}}\left[\frac{1}{\dfrac{-\ln\eta}{\left(\dfrac{SV_g}{\alpha_c}\right)^{0.125} f\left(\dfrac{V_g}{V_c}\right)} + 0.04}\right] \qquad (13)$$

Assume that $S = 0.25$ inch $= 0.0208$ feet:

$$\left(\frac{SV_g}{\alpha_c}\right)^{0.125} = \left[\frac{(0.0208)(2000)}{0.24\times10^{-3}}\right]^{0.125} = 4.51 \tag{14}$$

$$f\left(\frac{V_g}{V_c}\right) = 1 + 0.4 \tan^{-1} 1.0 = 1.31$$

$$T_{ad} = t_g + (Pr)^{1/3} \frac{V_g^2}{2gJc_p} = 1200 + (0.66)^{1/3} \frac{(2000)^2}{64.4 \times 778 \times 0.255} = 1472^{\circ} \text{ R}$$

$$\eta = \frac{1472 - 800}{1472 - 540} = 0.722; \ \ln \eta = -0.326$$

$$h = \frac{0.0265}{D_h} k_f \left(\frac{\rho g V D_h}{g\mu}\right)_f^{0.8} \left(\frac{c_p g\mu}{k}\right)_f^{0.3}$$

where properties are evaluated at a film temperature of $\frac{1200 + 540}{2} = 870^{\circ}$ R. The property values for air were obtained from reference 8.

$$h = \frac{0.0265 \times 0.64\times10^{-5}}{2} \left(\frac{0.0456 \times 2000 \times 2}{1.76\times10^{-5}}\right)^{0.8} \left(\frac{0.245 \times 1.76\times10^{-5}}{0.64\times10^{-5}}\right)^{0.3}$$

$$= \frac{0.0306 \text{ Btu}}{(\text{sq ft})(\text{sec})(^{\circ}\text{R})}$$

$$L \approx 2\pi = 6.2832 \text{ ft}$$

$$c_{p,c} = 0.24 \quad \text{at } 540^{\circ} \text{ R}$$

Substituting these values into equation (13) gives

$$\dot{w}_c = \frac{0.0306 \times 6.2832 \times 2}{0.24} \left(\frac{1}{\frac{0.326}{4.51 \times 1.31} + 0.04}\right) = 16.84 \text{ lb/sec}$$

From continuity,

$$S = \frac{\dot{w}_c}{\rho_c g V_c L} = \frac{16.84}{0.0735 \times 1000 \times 6.2832} = 0.0365 \text{ ft}$$

Substituting this value into equation (14) and the corrected values into equation (13) yields

$$\dot{w}_c = 17.56 \text{ lb/sec}$$

for which the new S is 0.0381 feet, which would not greatly change the result from equation (14).

$$\frac{\dot{w}_c}{\dot{w}_g} = \frac{17.56}{0.0331 \times 2000 \times 3.1416} = 0.0844$$

or the coolant flow required is 8.44 percent of the main-gas-stream flow.

Inspection of equation (13) reveals that, for equal values of velocities, pressures, and temperatures, the coolant flow required to cool a specified length of duct to a specified temperature varies approximately as the diameter to the 0.8 power. The gas flow varies as the diameter squared. Therefore, $\dot{w}_c/\dot{w}_g \propto 1/D^{1.2}$. Consequently, with increased diameter of the duct to be cooled, the percentage coolant flow required decreases rapidly.

CONCLUDING REMARKS

An equation, based on a simplified theoretical flow model, has been derived that correlates film-cooling data for a wide range of slot heights, velocity ratios, main-gas-stream temperatures, and two coolant gases of widely differing properties. This equation predicts the wall temperature within approximately 5 percent for ranges of cooling effectiveness between approximately 0.2 and 1.0, which should cover the usable range. The equation applies equally well when helium and air are used as coolants; and, because of their widely differing fluid properties, it may be inferred that the correlation would hold for any gas in the absence of chemical reactions. Data from the General Electric film-cooling facility, which was considerably larger than the NASA facility and which had main-gas-stream velocities and temperatures that were considerably lower, were also satisfactorily correlated; this indicated that all pertinent physical parameters appear to be incorporated in the equation.

Lewis Research Center
National Aeronautics and Space Administration
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APPENDIX - SYMBOLS

No units are given because the parameters used in this report are dimensionless. Consequently, only a consistent set of units is required.

A area of cooled surface

C constant

c_p specific heat

D_h hydraulic diameter

e base of natural logarithm

f function

g acceleration due to gravity

h convective heat-transfer coefficient

J mechanical equivalent of heat

k coefficient of thermal conductivity

L length of coolant slot (fig. 1)

M main-gas-stream Mach number

Pr Prandtl number

Re Reynolds number

S coolant slot height (fig. 1)

T_{ad} gas stream adiabatic wall (recovery) temperature,

$$T_{ad} = t_g + (Pr)_g^{1/3} \frac{V_g^2}{2gJc_p}$$

t static temperature

$\bar{t}$ average coolant temperature

V velocity

$\dot{w}$ flow rate

x	distance along cooled wall in direction of flow (fig. 1)
x'	distance coolant flows along wall before wall temperature increases
y	distance on cooled wall normal to flow direction
α	thermal diffusivity, $\frac{k}{\rho c_p}$
η	cooling effectiveness, $\eta = \frac{T_{ad} - t_w}{T_{ad} - t_c}$
θ	time
μ	coefficient of viscosity
ρ	mass density

Subscripts:

c	coolant slot exit
calc	calculated
f	properties evaluated at $\frac{t_g + t_c}{2}$
g	main body of gas
meas	measured
w	wall
1	position closest to coolant slot
2	position Δx farther from coolant slot

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TABLE I. - HELIUM-COOLANT DATA

S, in.	L, ft	W_c, lb/sec	W_g, lb/sec	V_c, ft/sec	V_g, ft/sec	T_{ad}, ^{o}R	t_c, ^{o}R	Wall temperature, t_w, at station x, ^{o}R											
								x, ft											
								0.132	0.209	0.376	0.543	0.710	0.876	1.042	1.293	1.501	1.751	2.000	2.822
1/8	0.667	0.0102	11.91	294	992	1475	950	1048	1115	1248	1303	1328	1342	1342	1347	1368	1373	1377	1372
		.0146	11.93	325	993	1475	800	928	1010	1177	1250	1283	1303	1312	1317	1338	1348	1353	1355
		.0207	11.94	429	990	1470	690	800	885	1068	1165	1210	1245	1257	1272	1293	1305	1317	1327
		.0293	11.92	582	985	1470	665	698	740	925	1035	1100	1150	1173	1195	1223	1238	1258	1282
		.0422	11.90	846	986	1470	645	643	640	757	870	942	1000	1037	1068	1098	1125	1158	1203
		.0587	11.92	1081	985	1473	620	625	618	658	727	785	845	890	932	955	998	1042	1105
		.0826	11.88	1458	982	1470	597	600	593	622	657	683	722	757	792	810	875	908	980
		.1339	11.91	2224	975	1475	555	555	555	585	622	648	688	725	748	783	808	838	888
		.2013	11.90	3260	967	1470	525	525	525	560	608	648	693	735	762	795	810	847	893
		.2909	11.83	3683	961	1470	505	505	505	533	583	627	675	717	748	785	790	835	885
1/4	0.667	0.0090	12.05	135	1005	1470	980	1107	1188	1295	1337	1358	1363	1365	1367	1385	1385	1390	1385
		.0135	12.04	175	1000	1475	840	983	1073	1227	1295	1323	1337	1338	1343	1365	1368	1372	1370
		.0189	12.03	222	1001	1470	760	872	957	1143	1232	1273	1298	1303	1315	1340	1345	1350	1355
		.0268	12.05	288	999	1465	700	762	828	1035	1145	1202	1235	1250	1267	1295	1305	1313	1325
		.0378	12.10	364	1001	1465	635	660	722	902	1025	1097	1145	1167	1195	1223	1240	1257	1282
		.0691	12.00	626	991	1465	605	605	602	642	747	832	902	945	988	1008	1065	1095	1148
		.1091	12.03	933	986	1465	580	580	582	597	637	678	725	765	812	808	890	915	985
		.1664	12.03	1360	984	1470	570	570	570	595	635	663	698	723	750	770	810	828	878
1/2	0.667	0.0142	11.96	99	1037	1475	890	975	1045	1197	1277	1313	1335	1342	1352	1375	1375	1377	1375
		.0198	11.93	125	1032	1475	840	897	963	1128	1227	1277	1305	1317	1332	1355	1355	1360	1362
		.0280	11.89	166	1028	1475	785	825	873	1030	1153	1223	1262	1282	1300	1325	1332	1338	1343
		.0438	11.87	228	1034	1485	690	700	723	867	992	1108	1172	1203	1235	1268	1283	1297	1315
		.0785	11.76	376	1014	1483	640	640	640	640	755	872	953	1008	1063	1113	1148	1170	1218
		.1231	11.74	553	1002	1483	605	605	605	602	605	650	730	795	862	923	973	1005	1087
		.2098	11.69	885	990	1480	580	580	580	580	577	567	570	587	603	630	673	710	837
		.2978	11.67	1195	982	1480	558	558	558	557	553	543	547	558	558	558	573	588	653
		.4206	11.71	1572	975	1485	530	530	530	532	530	522	525	533	533	535	555	572	632
		.0156	14.87	85	863	1003	720	768	803	882	920	935	942	945	947	957	963	963	958
		.0217	14.85	110	861	1003	680	720	750	845	893	915	928	933	933	945	950	953	952
		.0363	14.88	166	859	1000	605	640	670	755	823	865	892	902	910	922	930	932	937
		.0542	14.91	236	856	1000	580	580	580	658	743	800	835	857	868	887	898	903	915
		.1125	14.86	475	848	1000	570	570	570	562	568	612	667	702	735	765	793	808	838
		.1652	14.34	678	821	1015	570	570	570	560	557	550	562	587	618	655	693	718	783
		.2346	14.31	908	820	1020	540	540	540	540	532	523	530	542	550	555	575	593	657
		.1192	14.38	476	832	1020	545	545	545	545	545	577	628	672	712	743	772	783	823

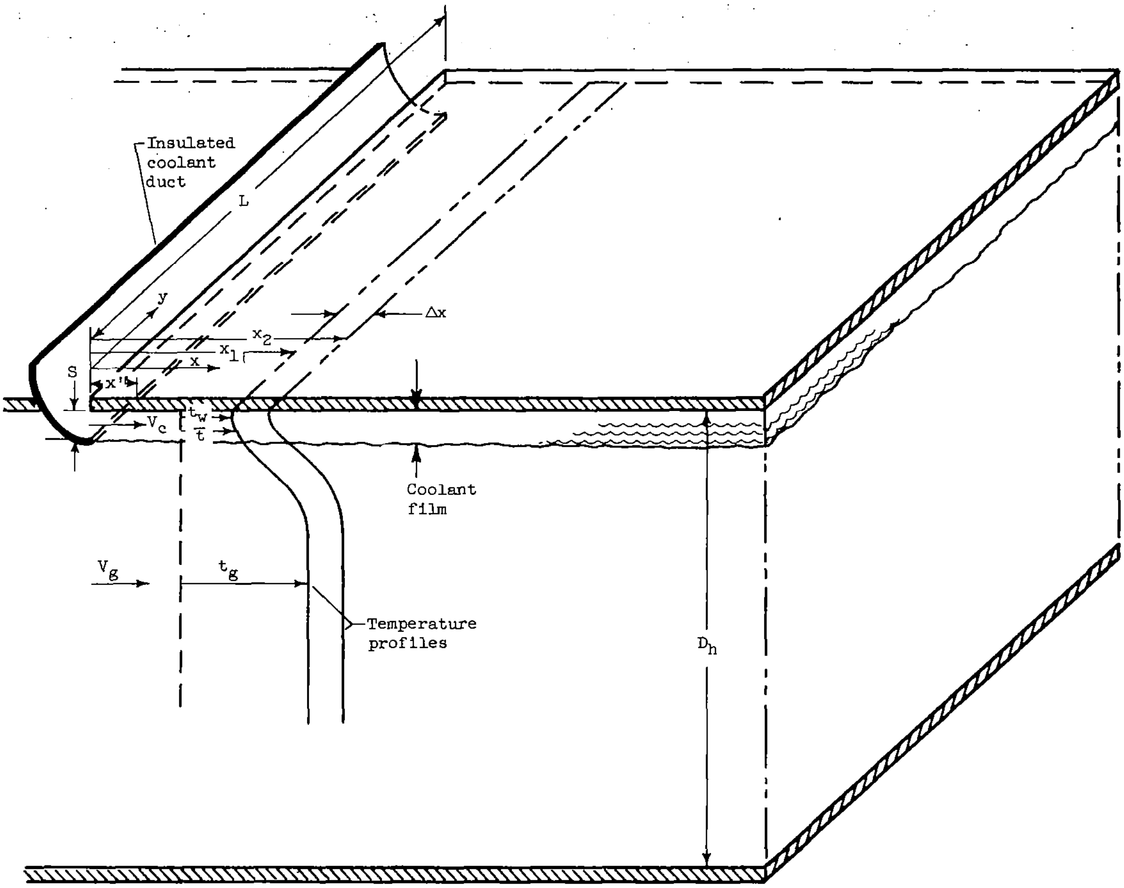


Figure 1. - Flat-plate flow model.

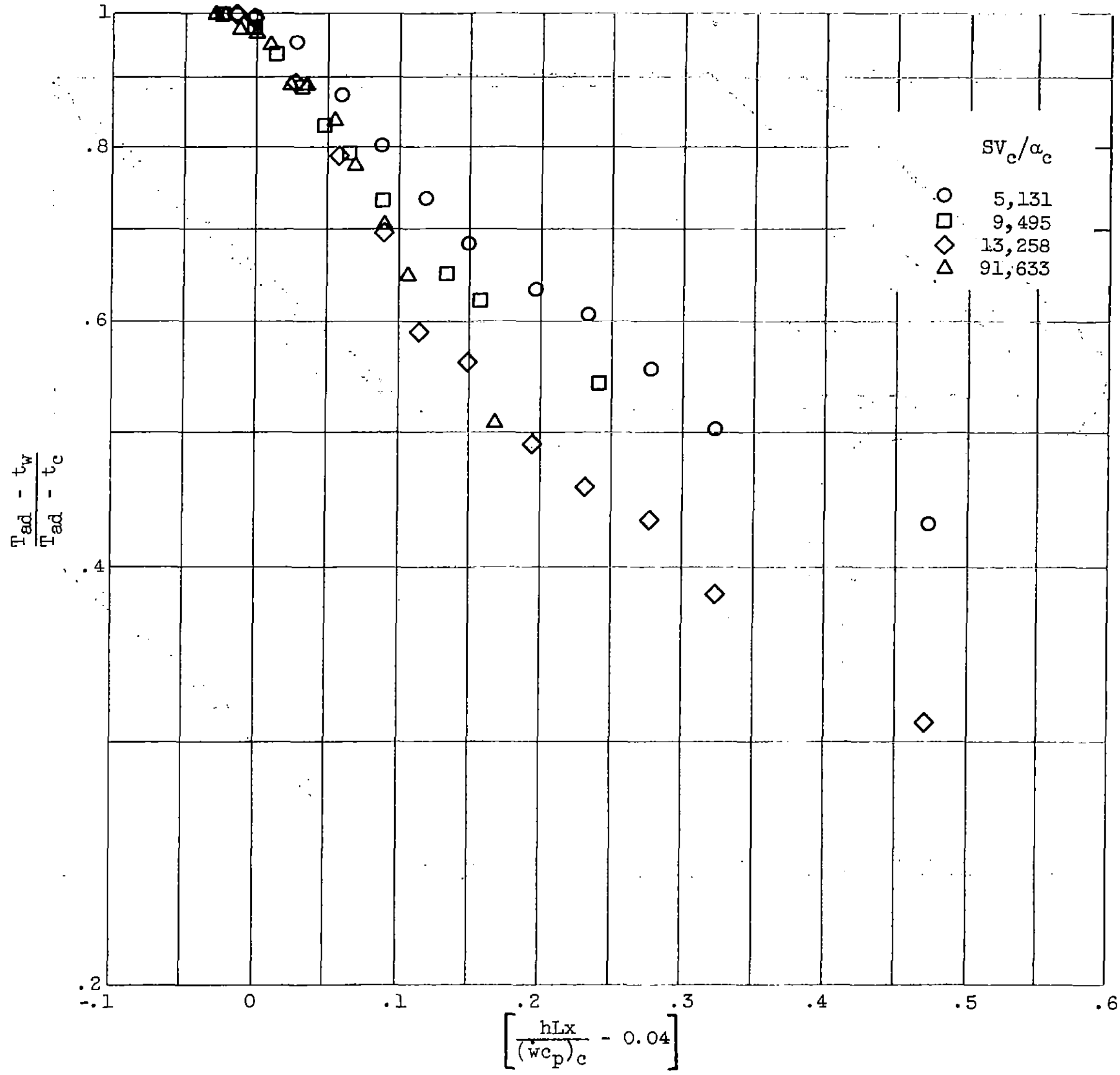


Figure 2. - Variation of cooling effectiveness with $\left[\frac{hLx}{(wc_p)_c} - 0.04\right]$ for velocity ratio of approximately 1.0 and range of values of SV_c/α_c.

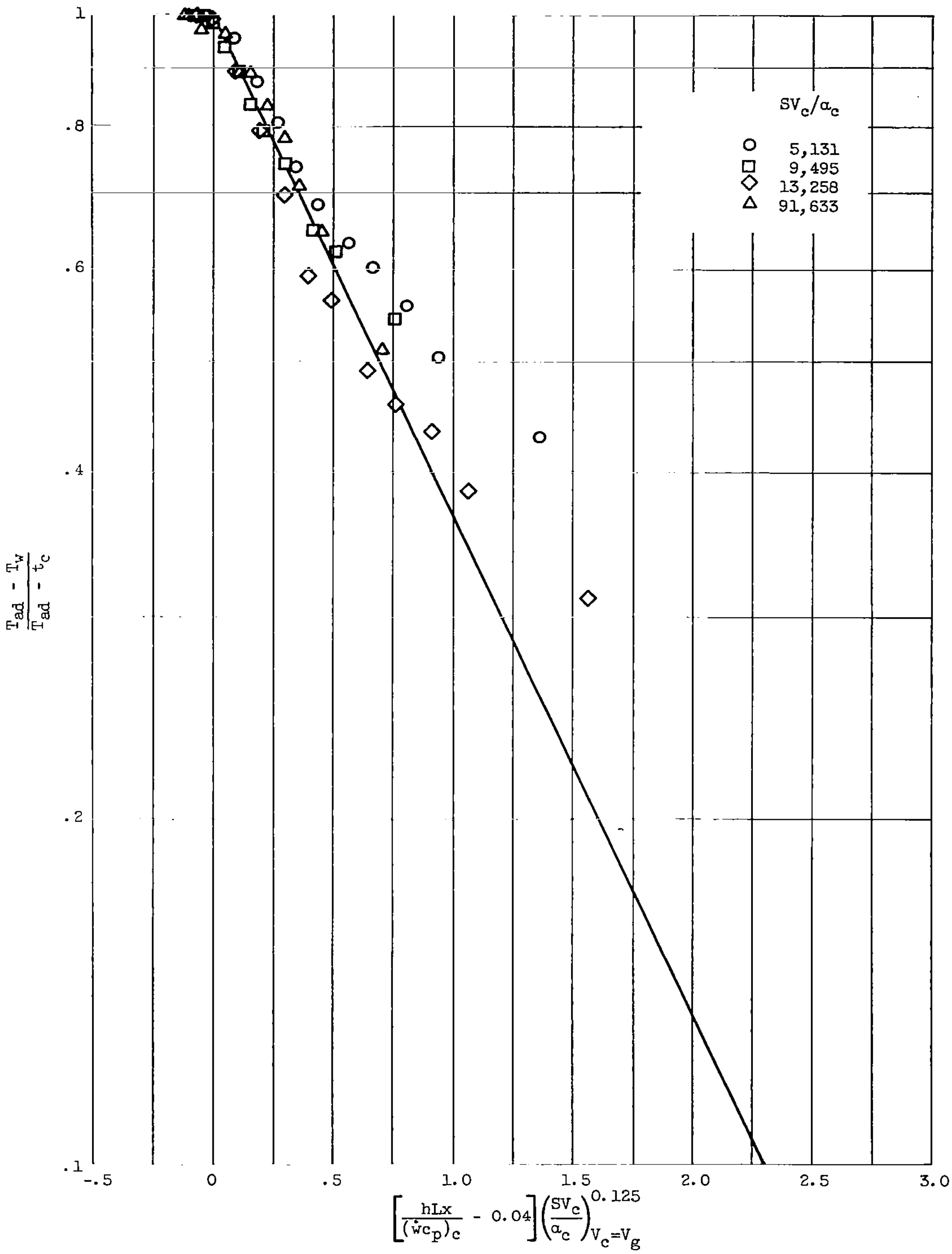


Figure 3. - Cooling effectiveness plotted against $\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right]\left(\frac{SV_c}{\alpha_c}\right)^{0.125}_{V_c=V_g}$ for velocity ratio of approximately 1.0 and range of values of SV_c/α_c.

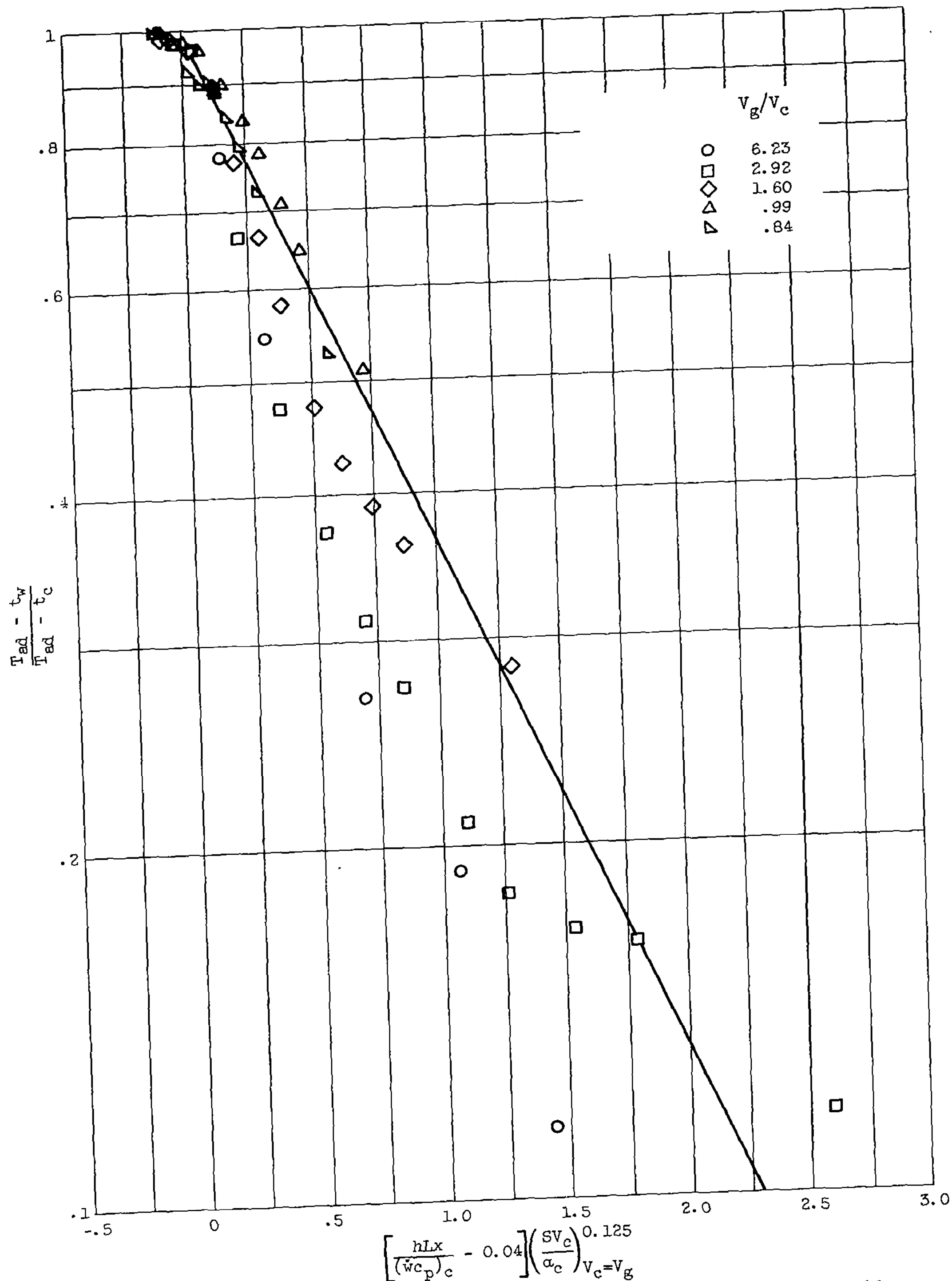


Figure 4. - Variation of cooling effectiveness with $\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right]\left(\frac{SV_c}{\alpha_c}\right)^{0.125}_{V_c=V_g}$ for a range of velocity ratios.

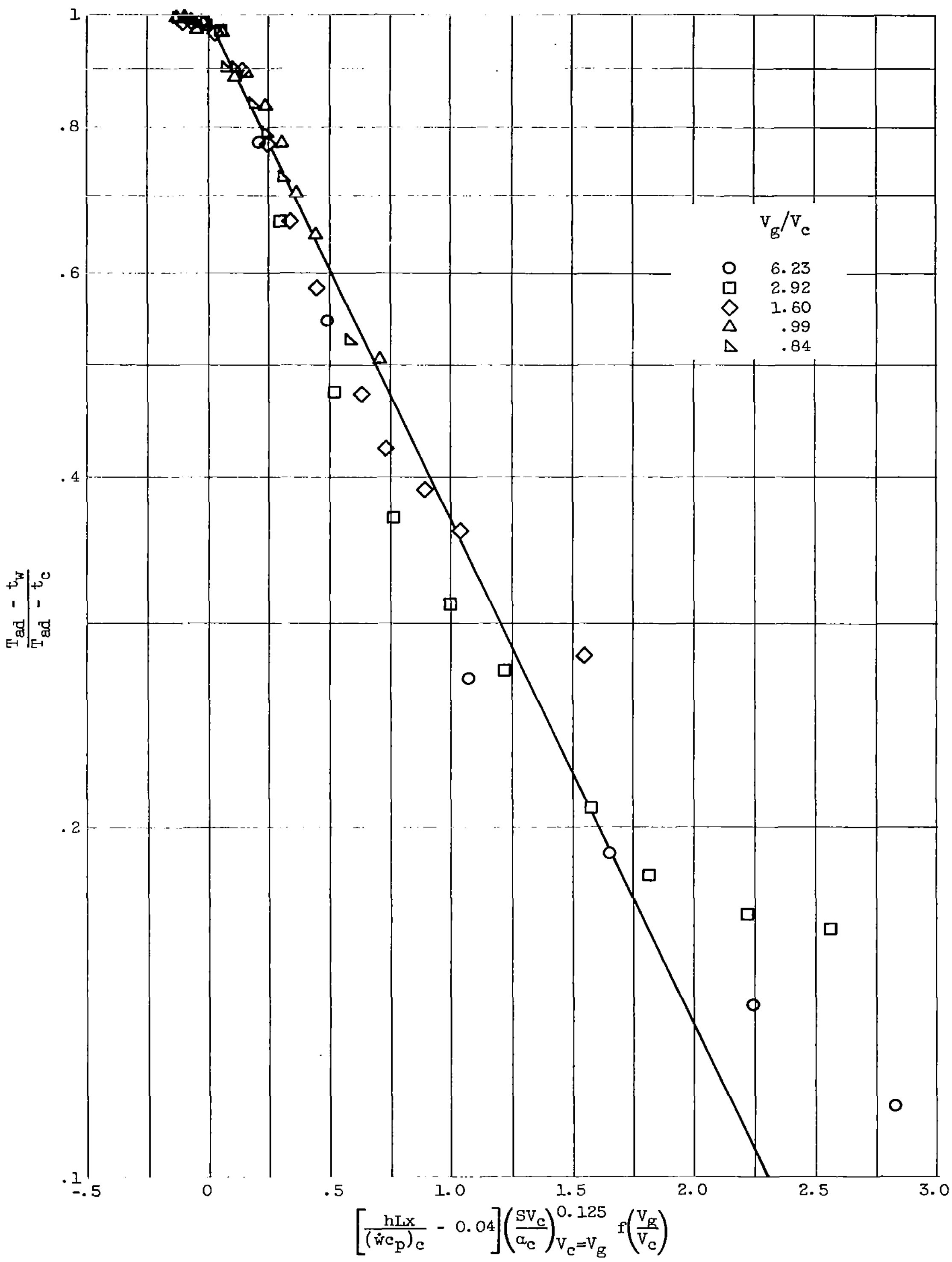
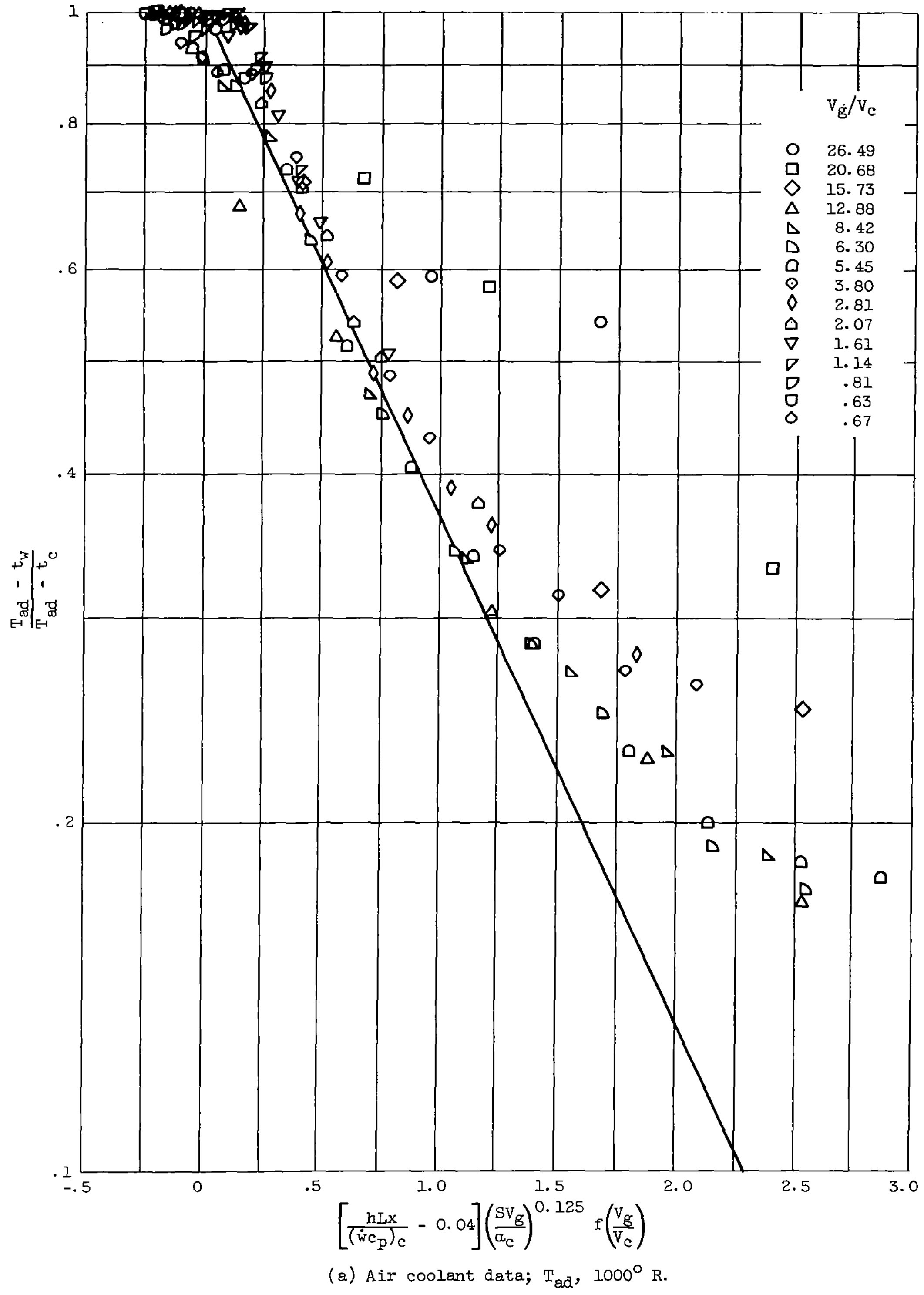
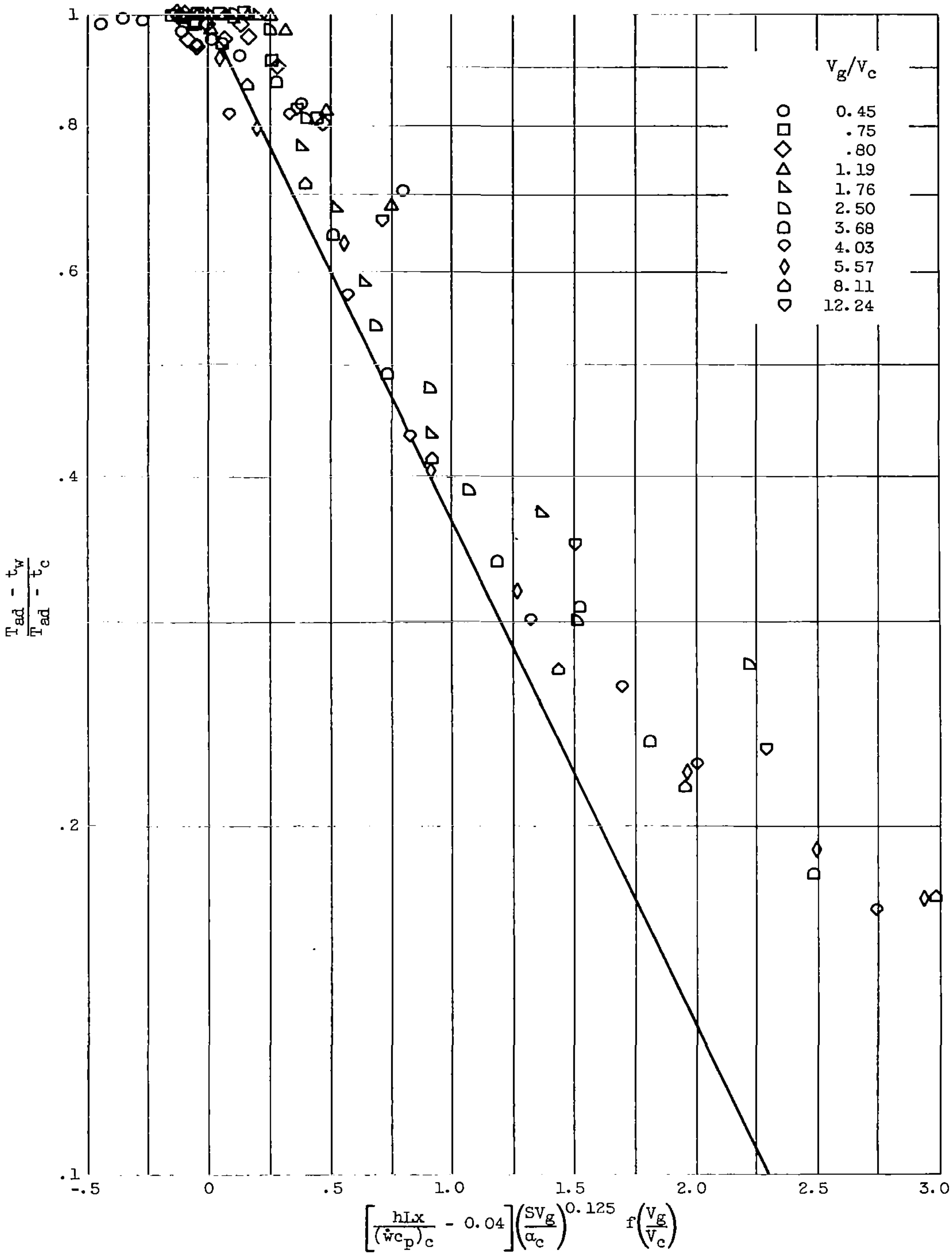


Figure 5. - Variation of cooling effectiveness with $\left[\frac{hLx}{(\dot{w}c_p)_c} - 0.04\right]\left(\frac{SV_c}{\alpha_c}\right)^{0.125}_{V_c=V_g} f\left(\frac{V_g}{V_c}\right)$.



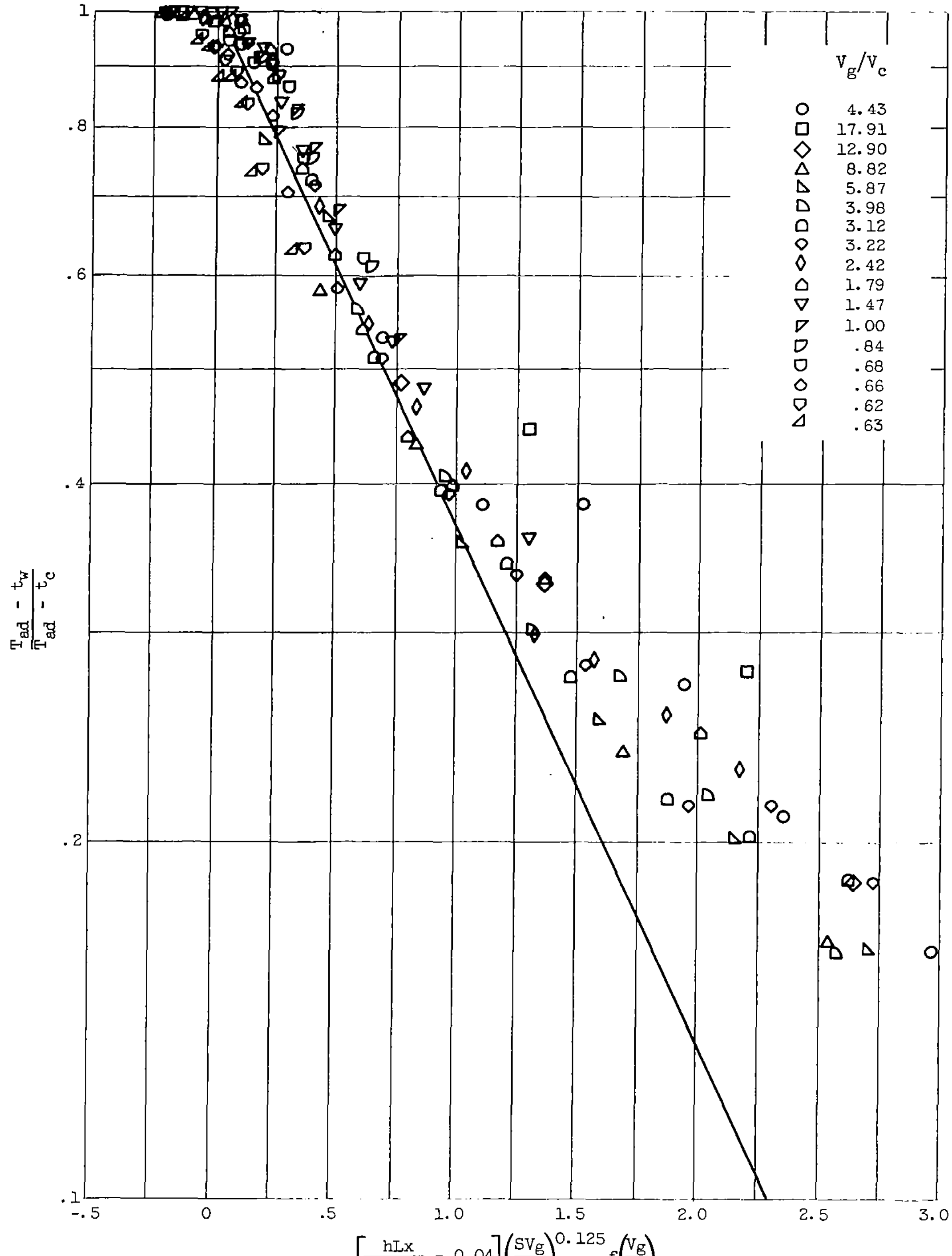
(a) Air coolant data; T_{ad}, 1000° R.

Figure 6. - NASA air coolant and heating data for slot height of 1/2 inch. M, 0.5.



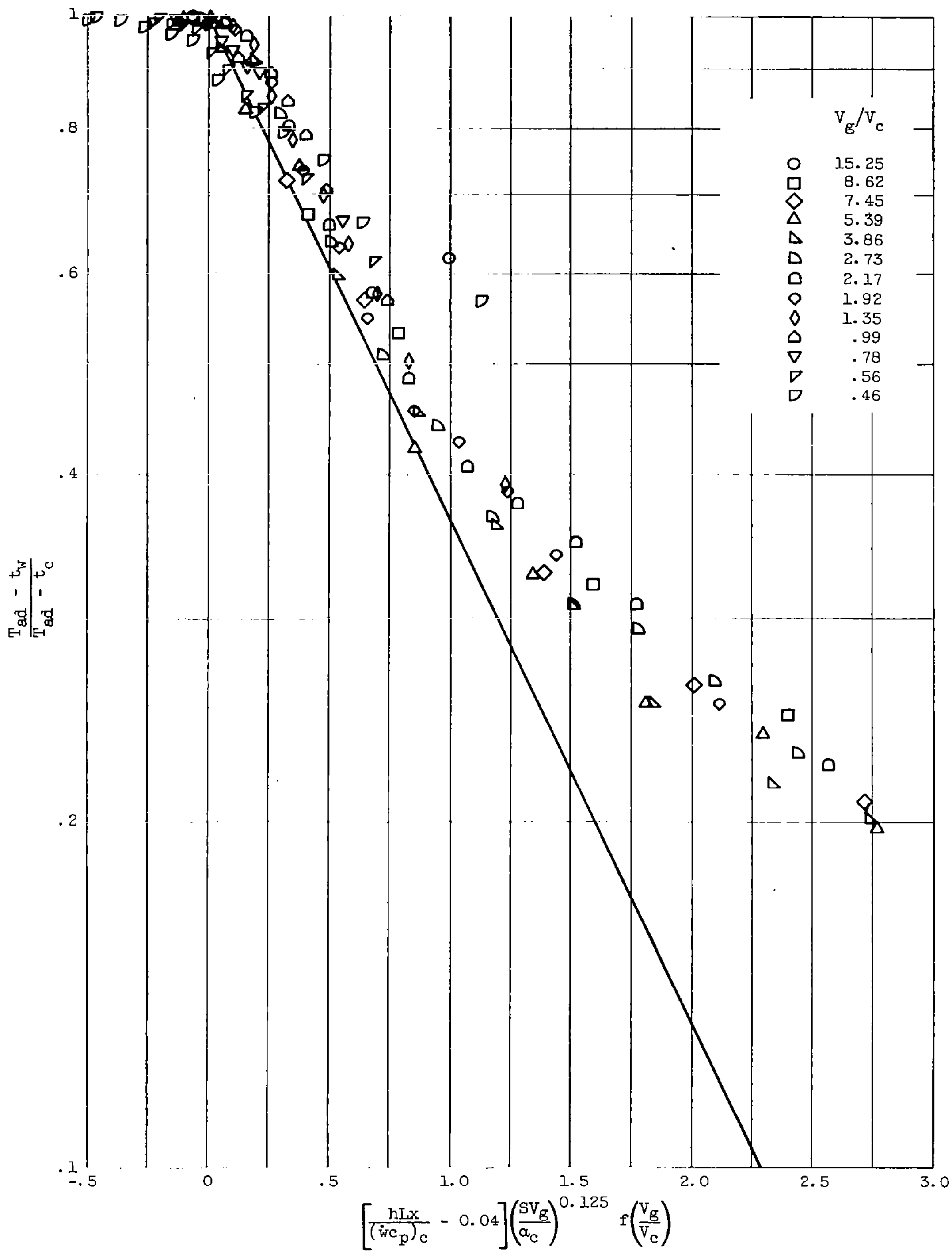
(b) Air heating data; $t_c \approx 800^\circ$ R.

Figure 6. - Concluded. NASA air coolant and heating data for slot height of 1/2 inch. M, 0.5.



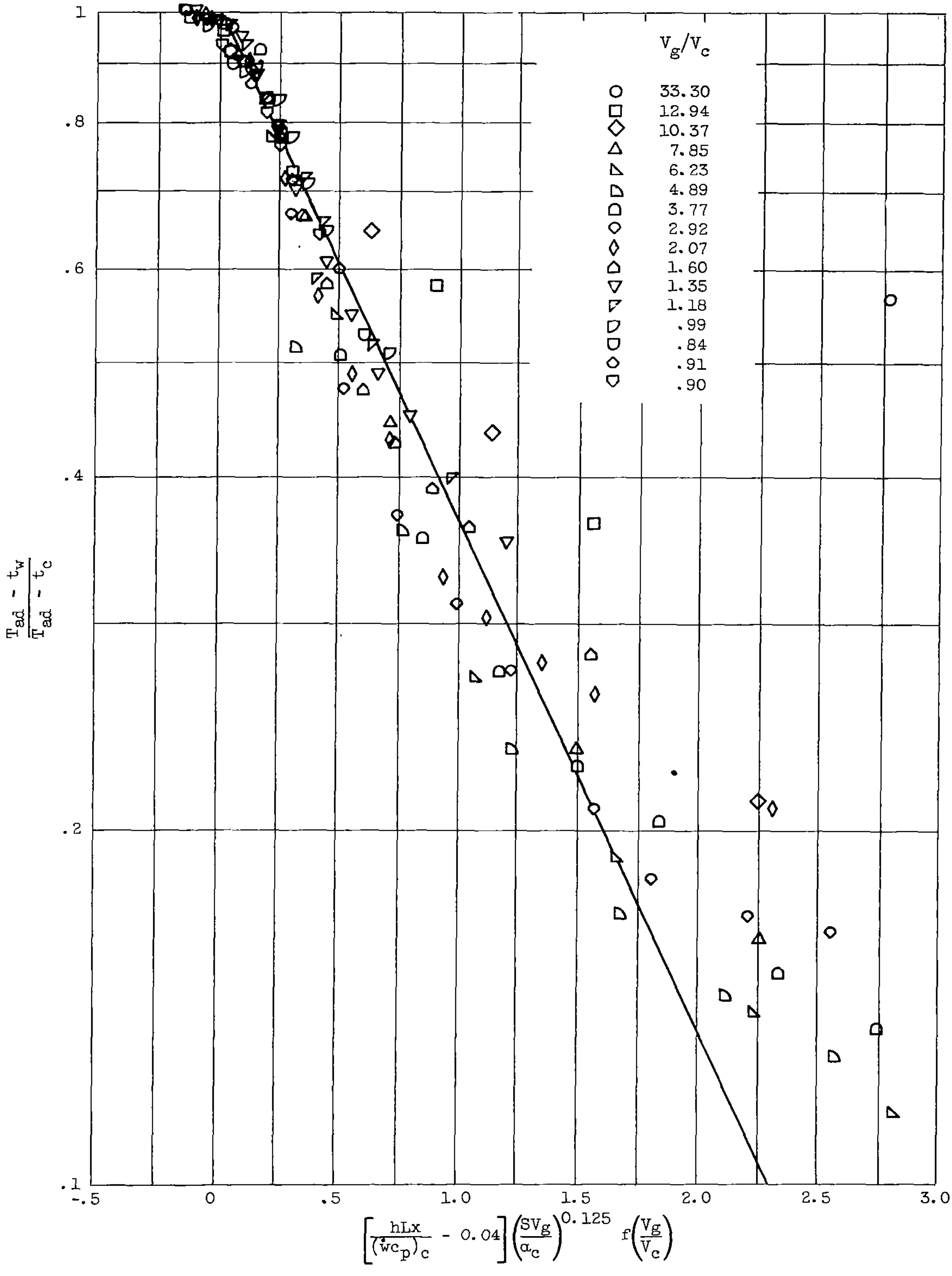
(a) T_{ad}, 1000° R; M, 0.5.

Figure 7. - NASA air coolant data for slot height of 1/4 inch.



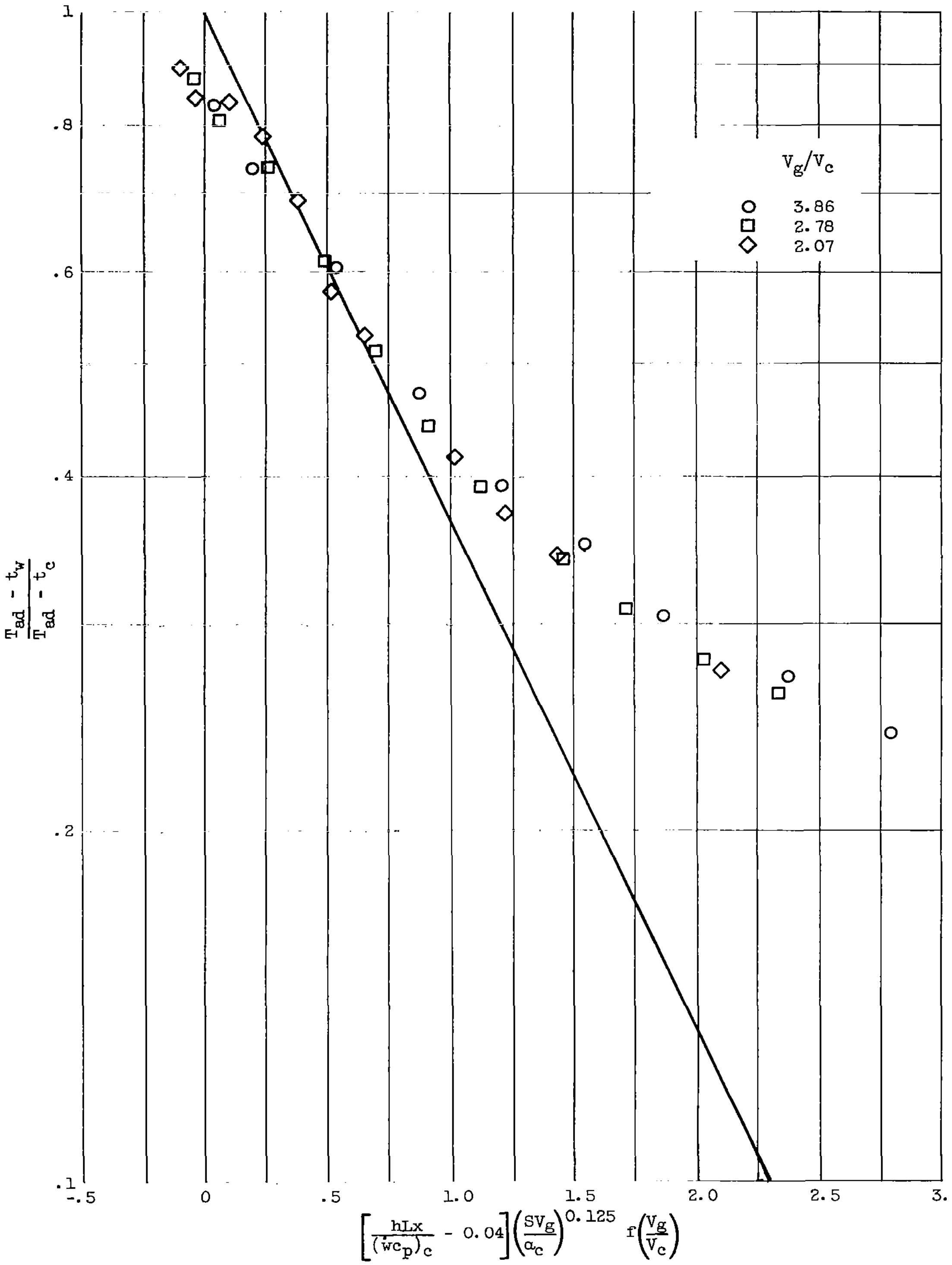
(b) T_{ad}, 1000° R; M, 0.2.

Figure 7. - Continued. NASA air coolant data for slot height of 1/4 inch.



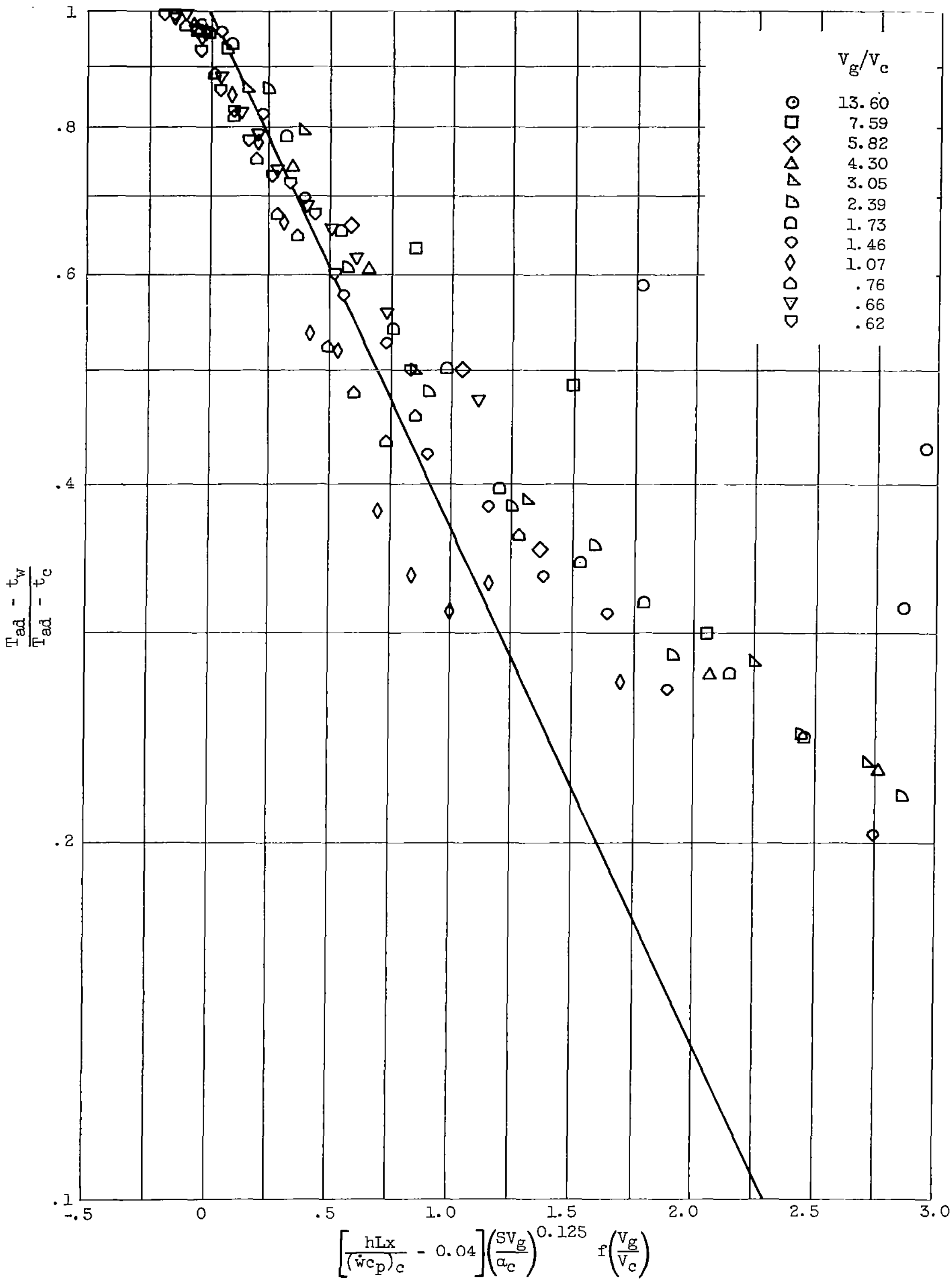
(c) T_{ad}, 1000° R; M, 0.7.

Figure 7. - Continued. NASA air coolant data for slot height of 1/4 inch.



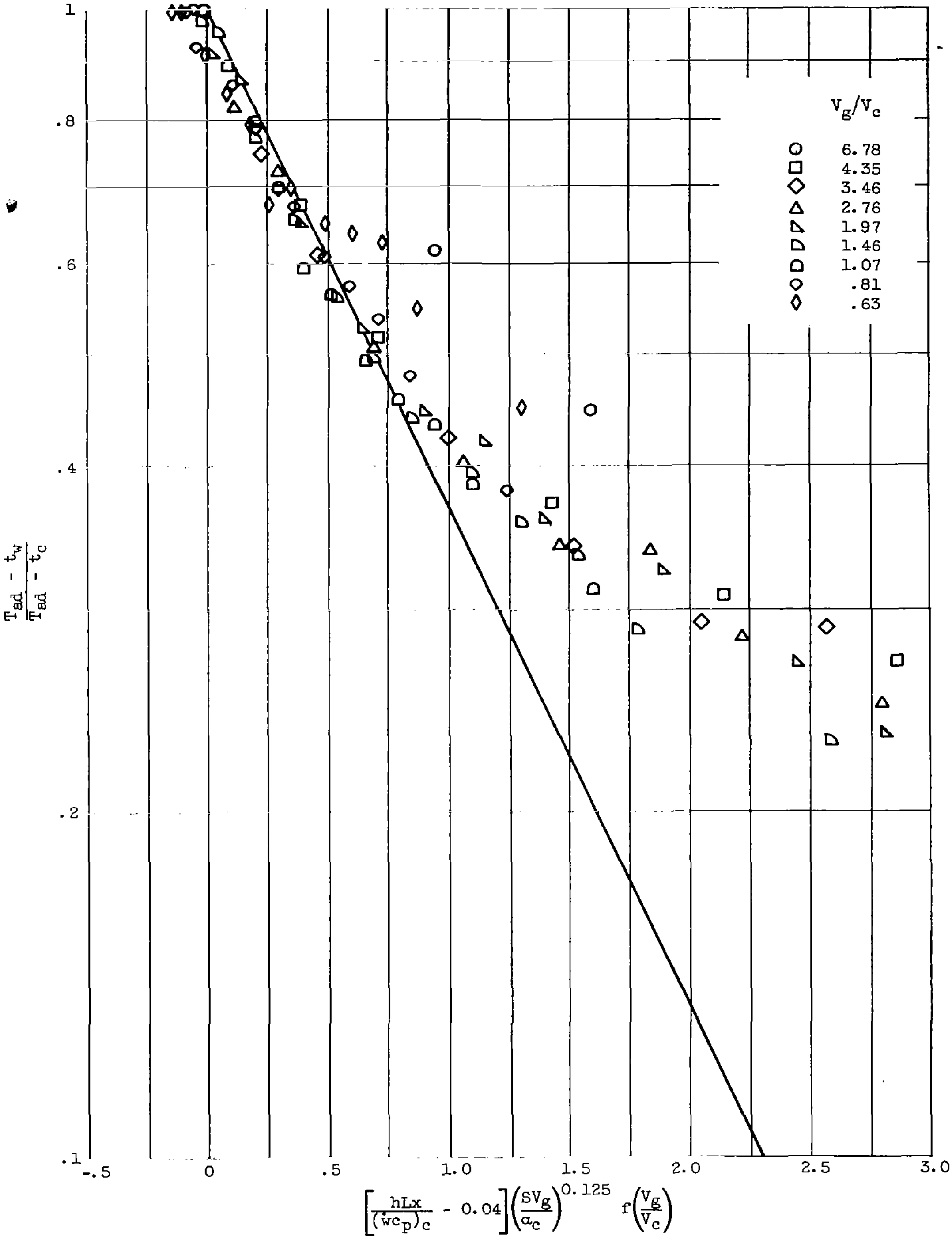
(d) T_{ad}, 1965° R; M, 0.5.

Figure 7. - Concluded. NASA air coolant data for slot height of 1/4 inch.



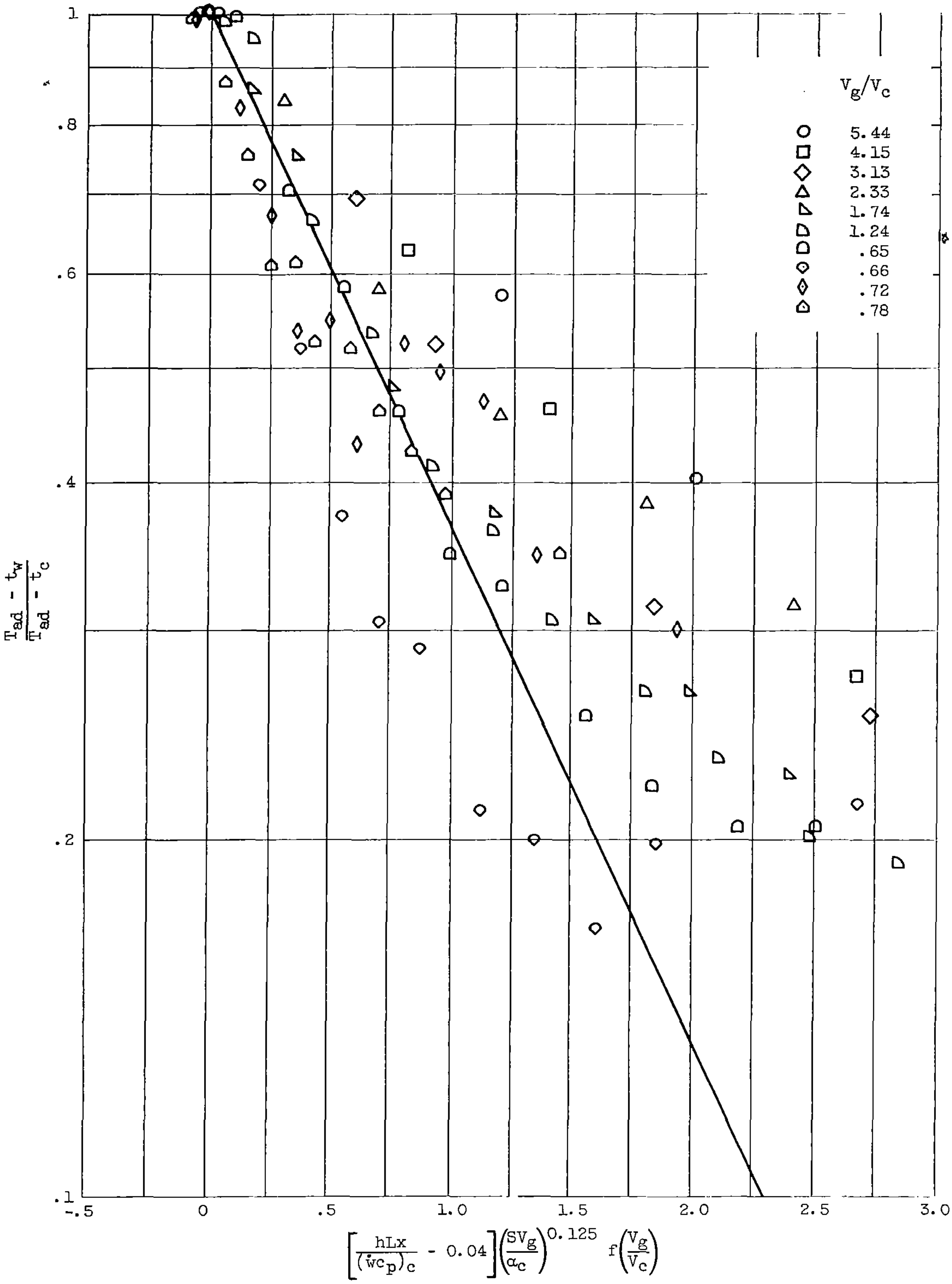
(a) T_{ad}, 1000° R; M, 0.5.

Figure 8. - NASA air coolant data for slot height of 1/8 inch.



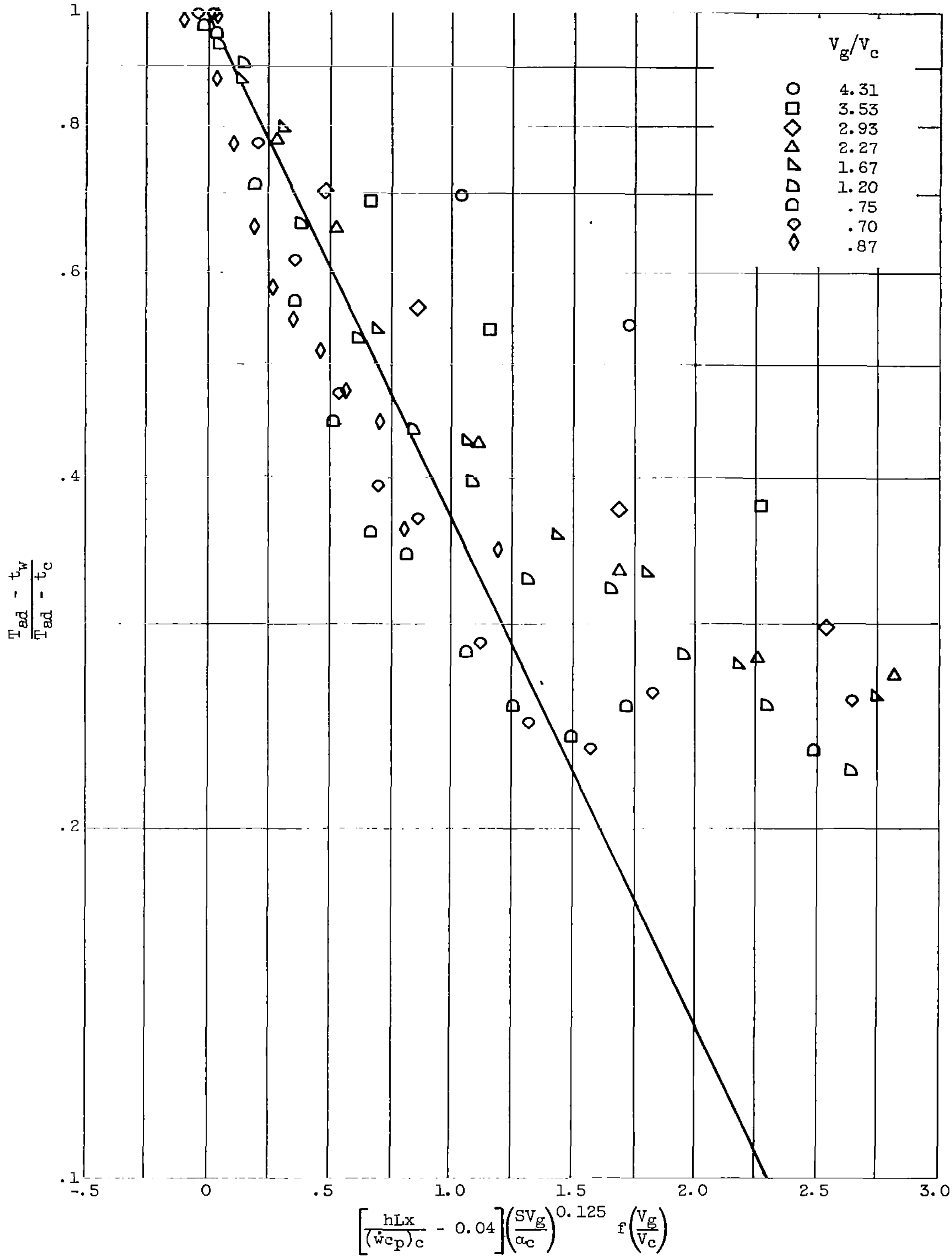
(b) T_{ad}, 1500° R; M, 0.2.

Figure 8. - Concluded. NASA air coolant data for slot height of 1/8 inch.



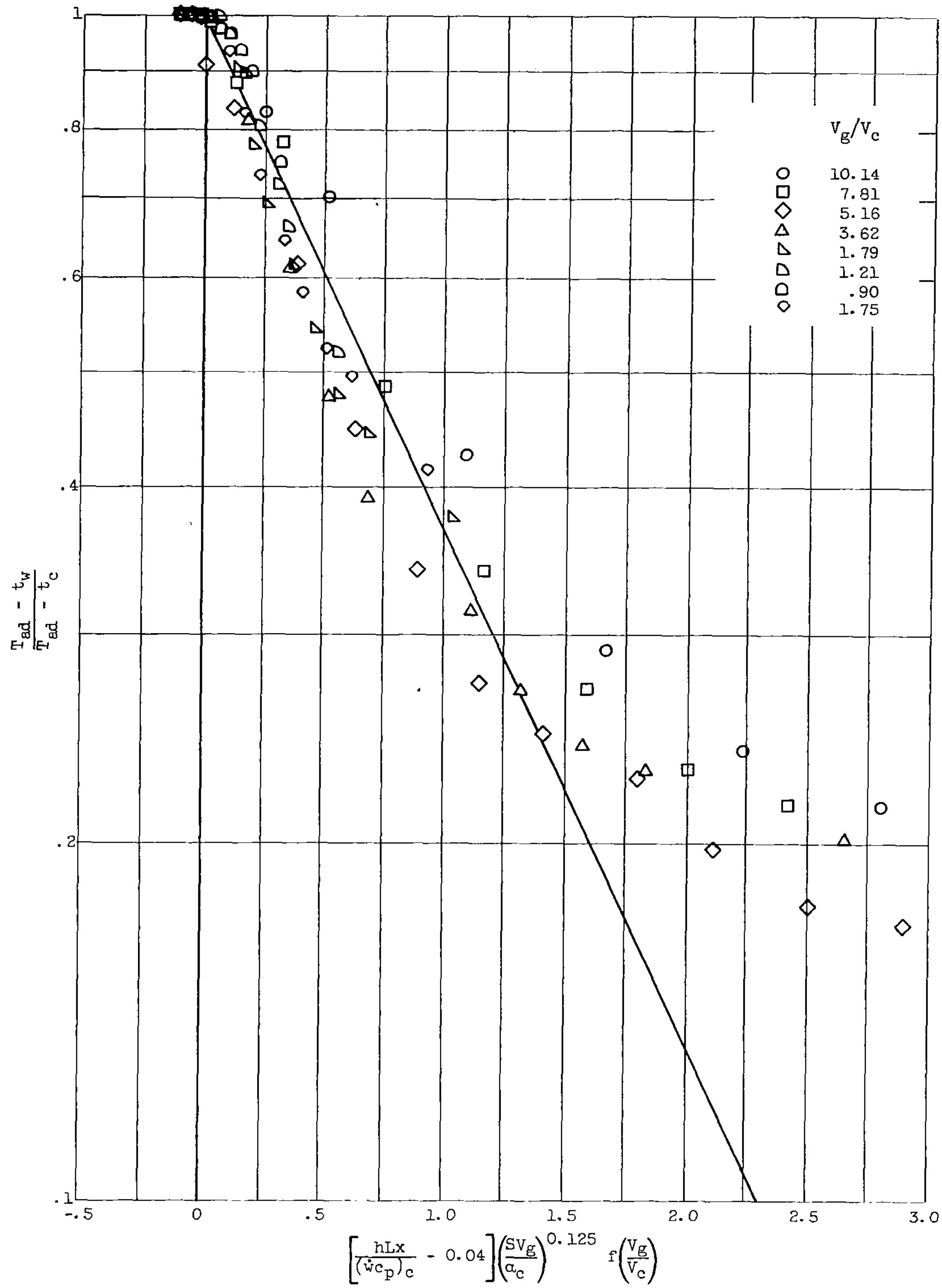
(a) T_{ad}, 1000° R.

Figure 9. - NASA air coolant data for slot height of 1/16 inch. M, 0.7.



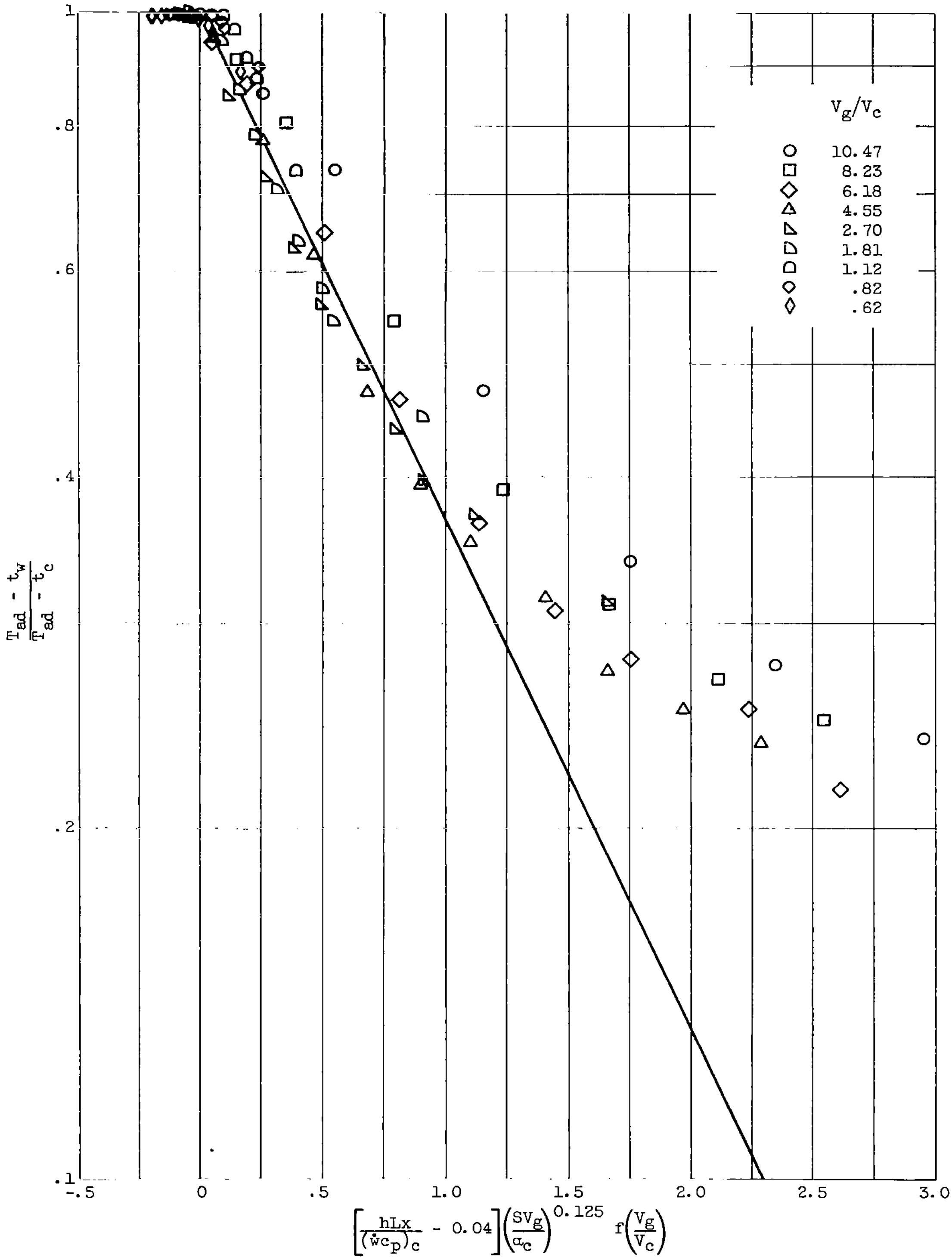
(b) T_{ad}, 1500° R.

Figure 9. - Concluded. NASA air coolant data for slot height of 1/16 inch. M, 0.7.



(a) T_{ad}, 1000° R.

Figure 10. - NASA helium coolant data for slot height of 1/2 inch. M, 0.5.



(b) T_{ad}, 1500° R.

Figure 10. - Concluded. NASA helium coolant data for slot height of 1/2 inch. M, 0.5.

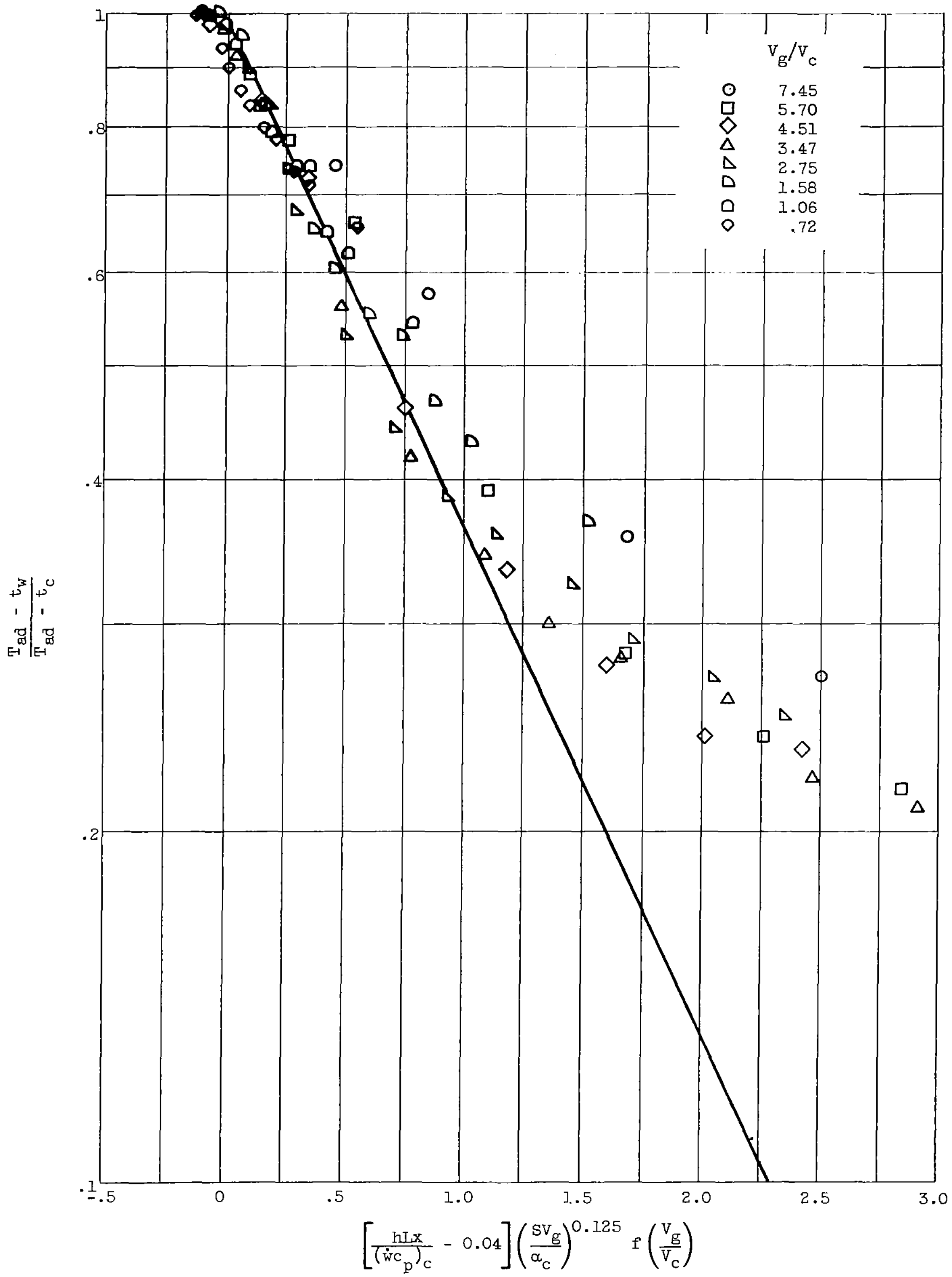


Figure 11. - NASA helium coolant data for slot height of 1/4 inch. T_{ad}, 1500° R; M, 0.5.

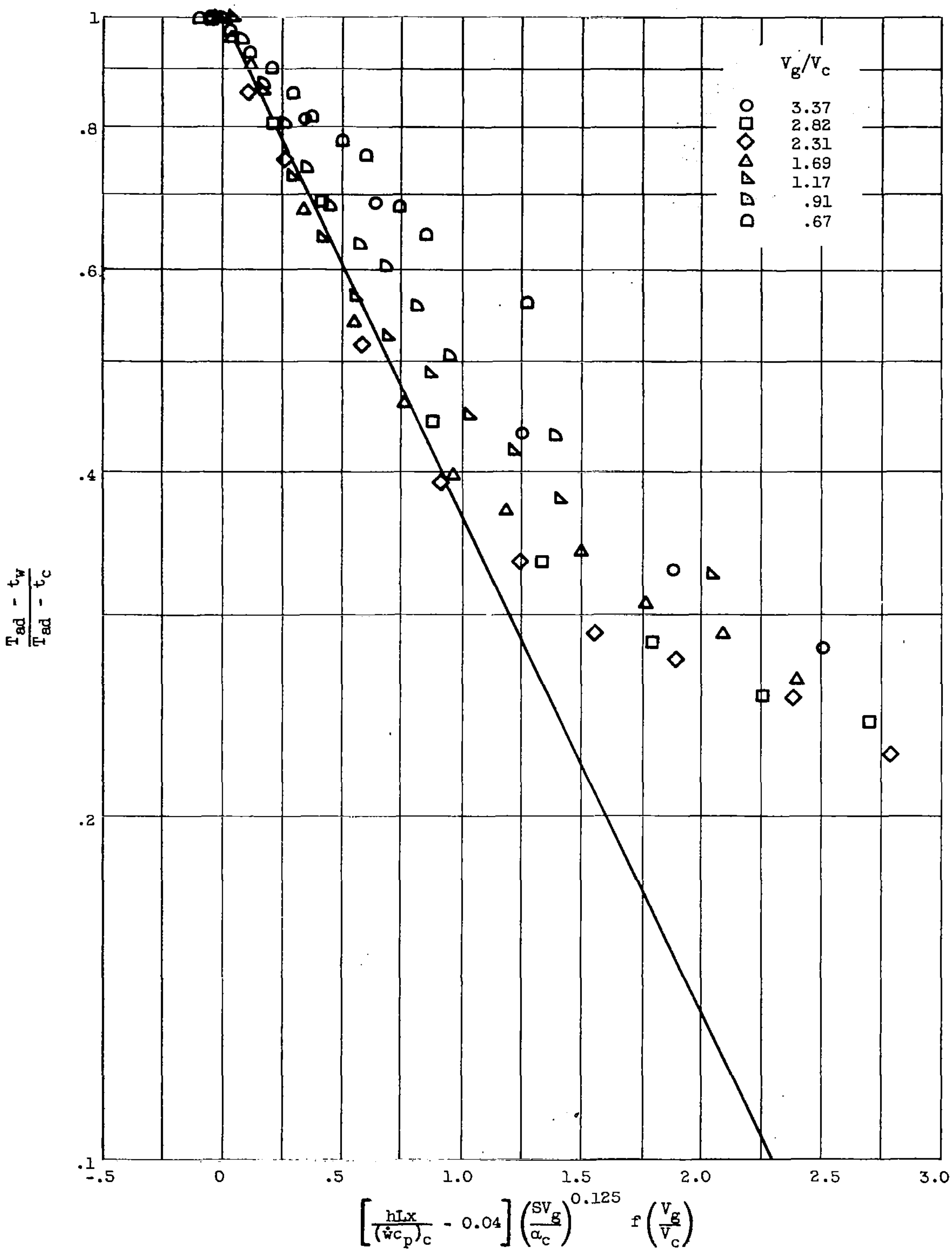
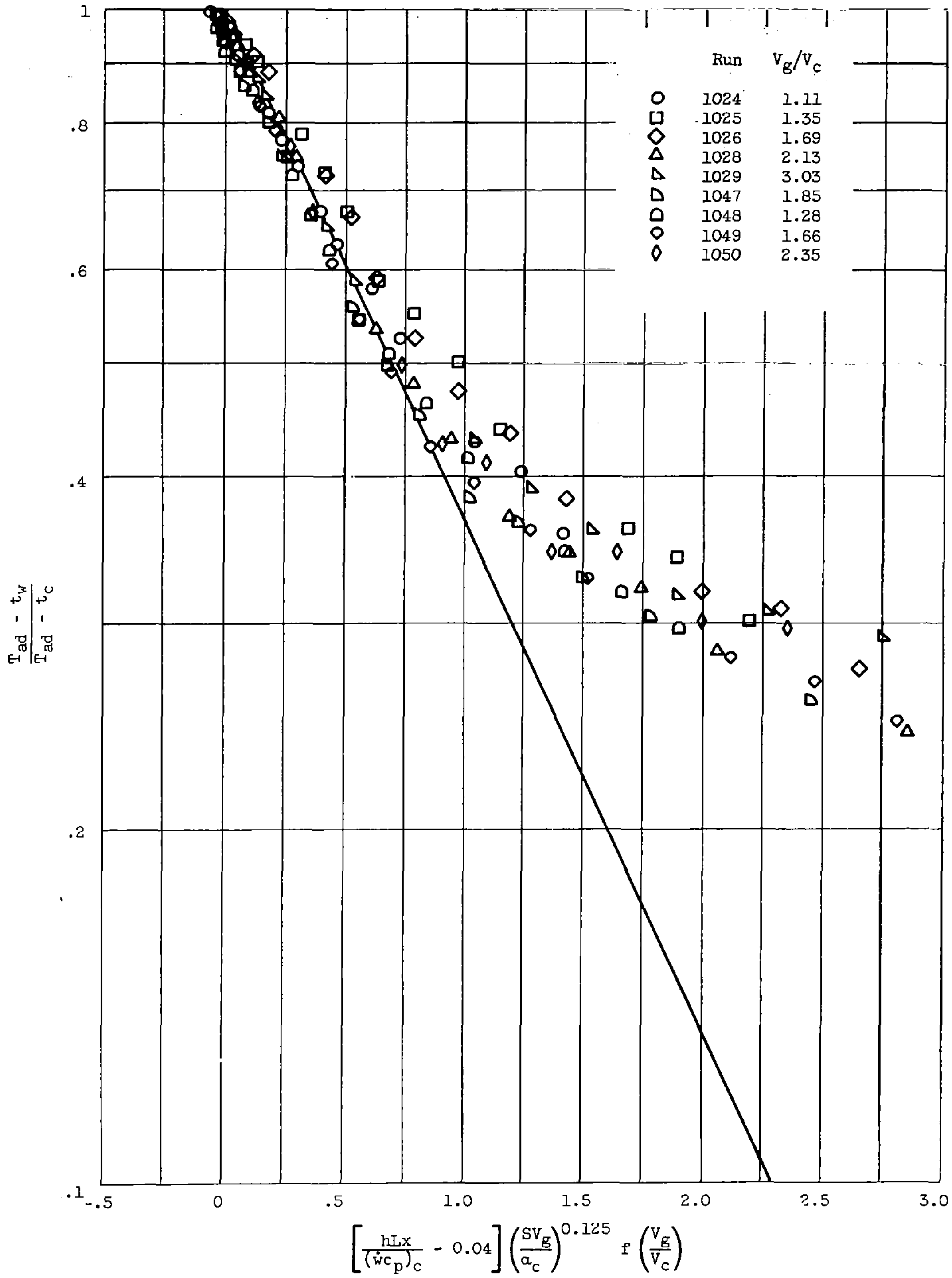
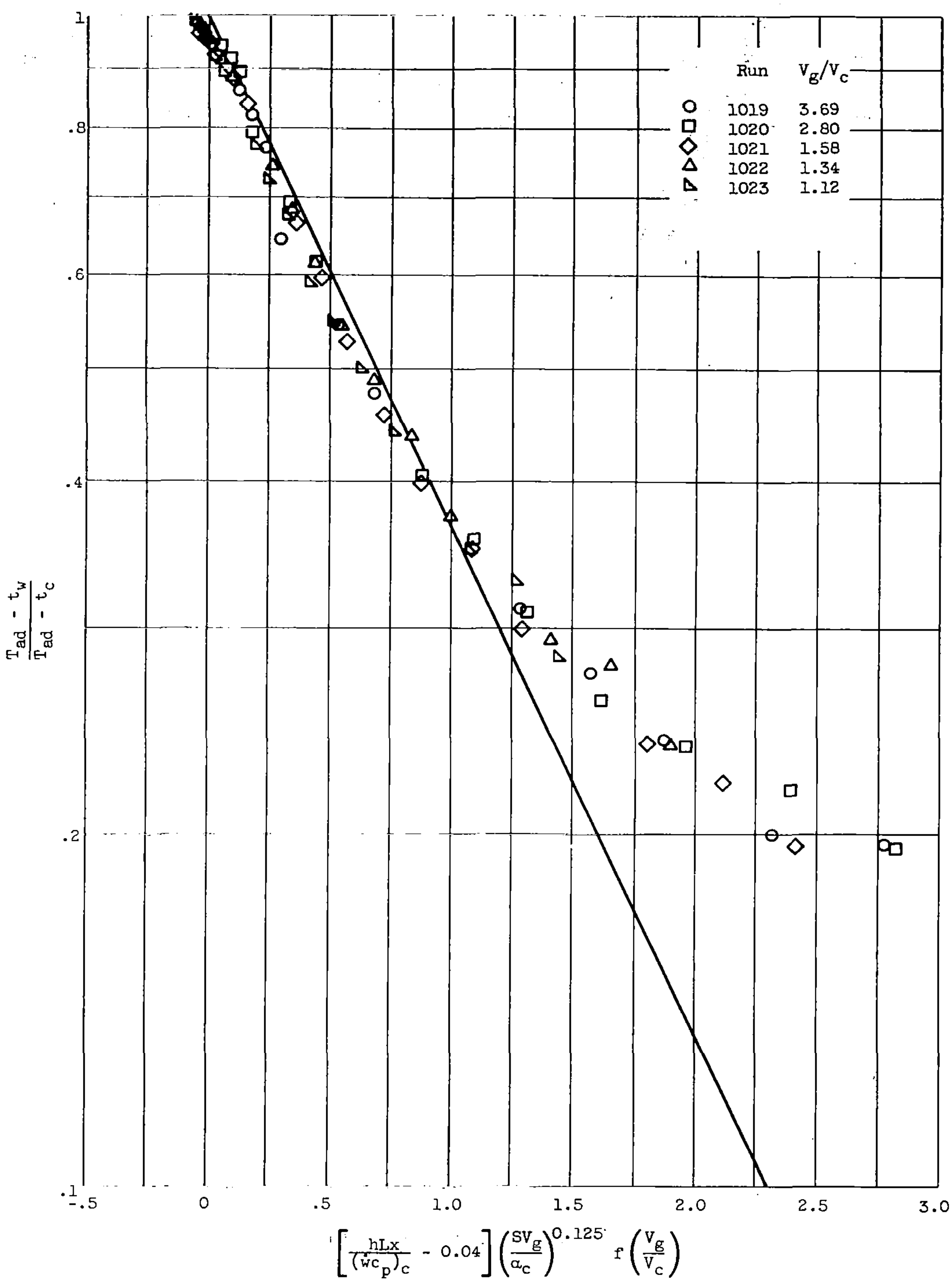


Figure 12. - NASA helium coolant data for slot height of 1/8 inch. T_{ad}, 1500° R; M, 0.5.



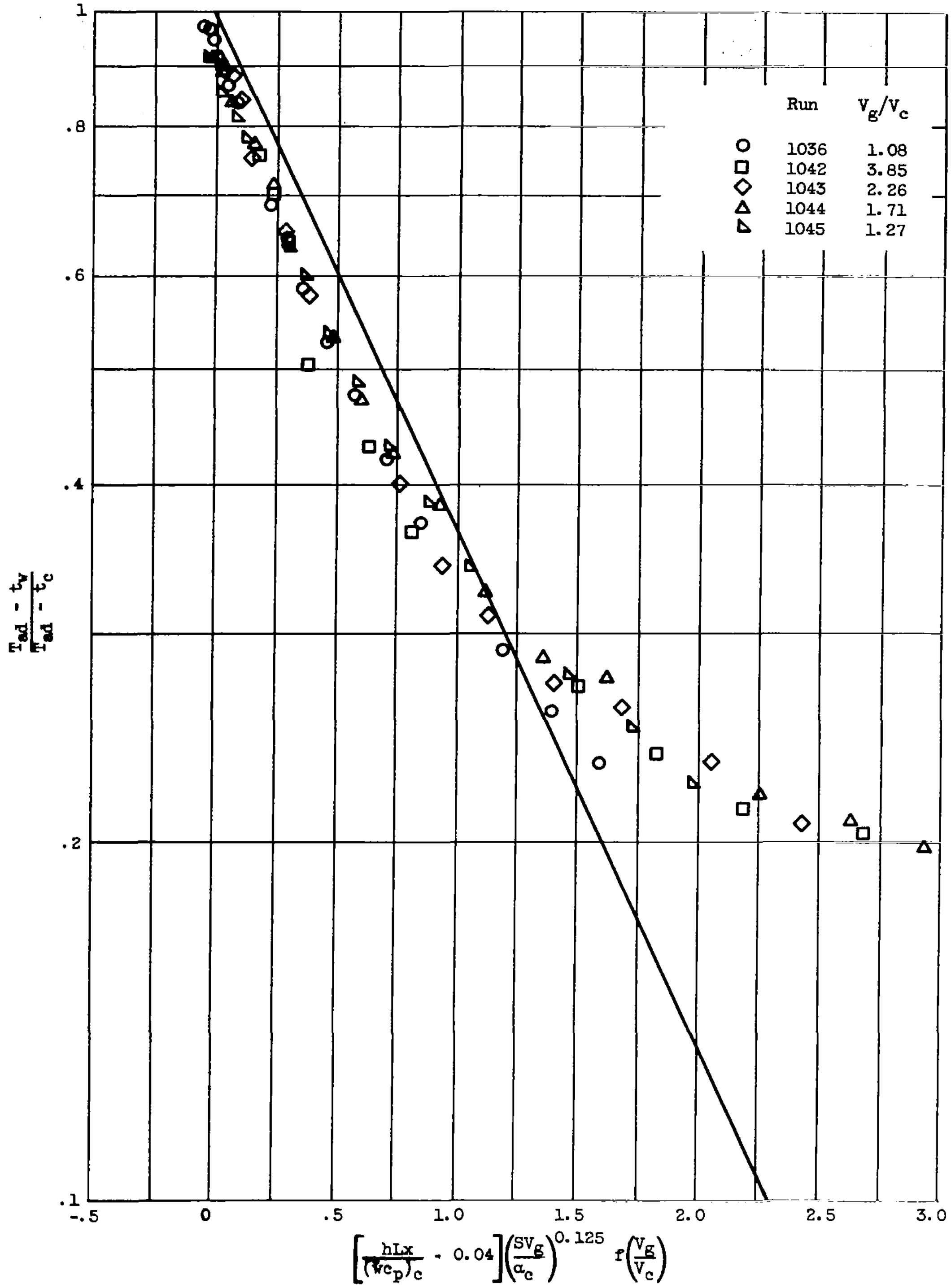
(a) $T_{ad} \approx 615^\circ$ R; hydrodynamic starting length, 41.5 inches.

Figure 13. - General Electric air coolant data for slot height of 0.106 inch. $V_g/V_c > 1.0$.



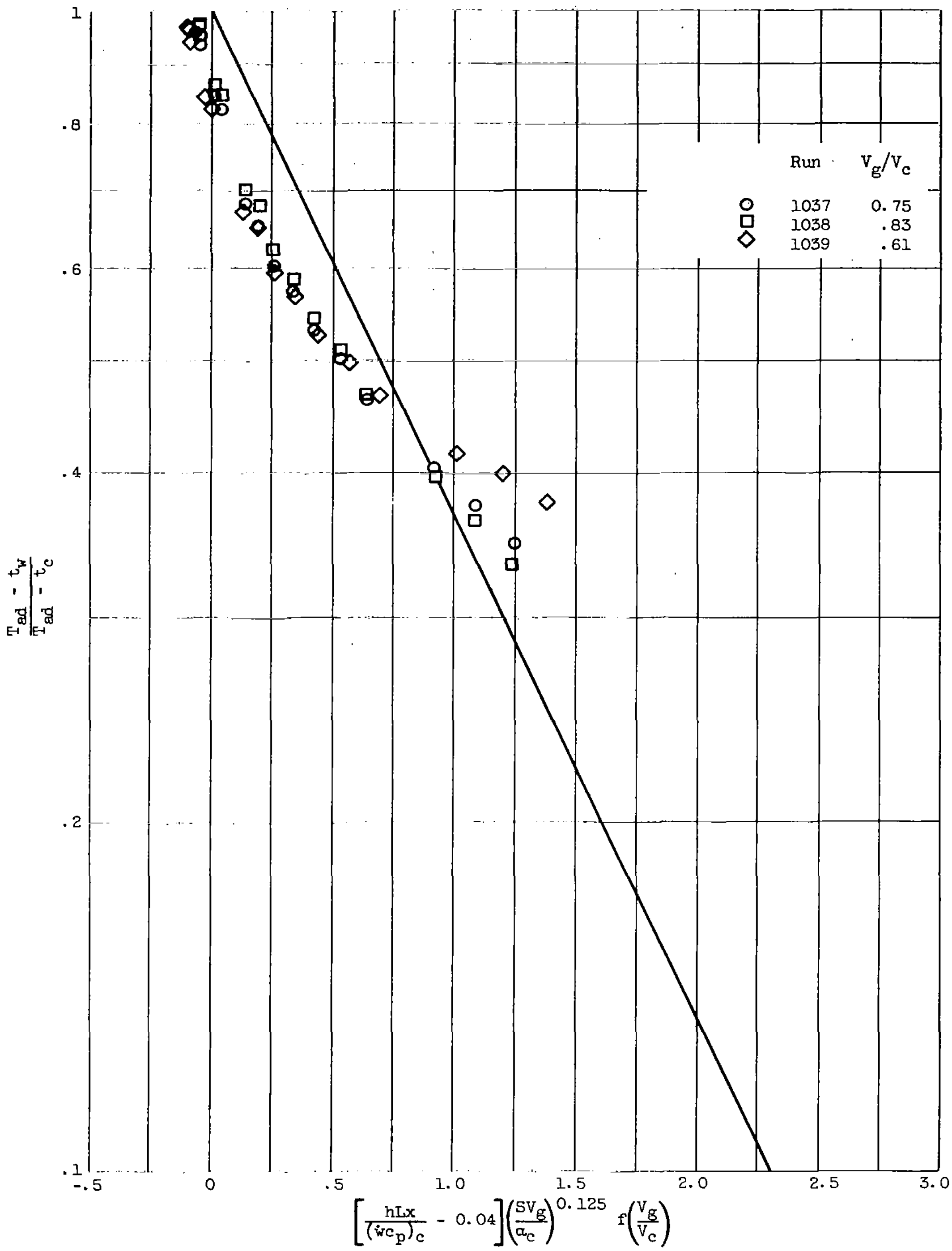
(b) $T_{ad} \approx 620^{\circ}$ R; hydrodynamic starting length, 239.3 inches.

Figure 13. - Concluded. General Electric air coolant data for slot height of 0.106 inch. $V_g/V_c > 1.0$.



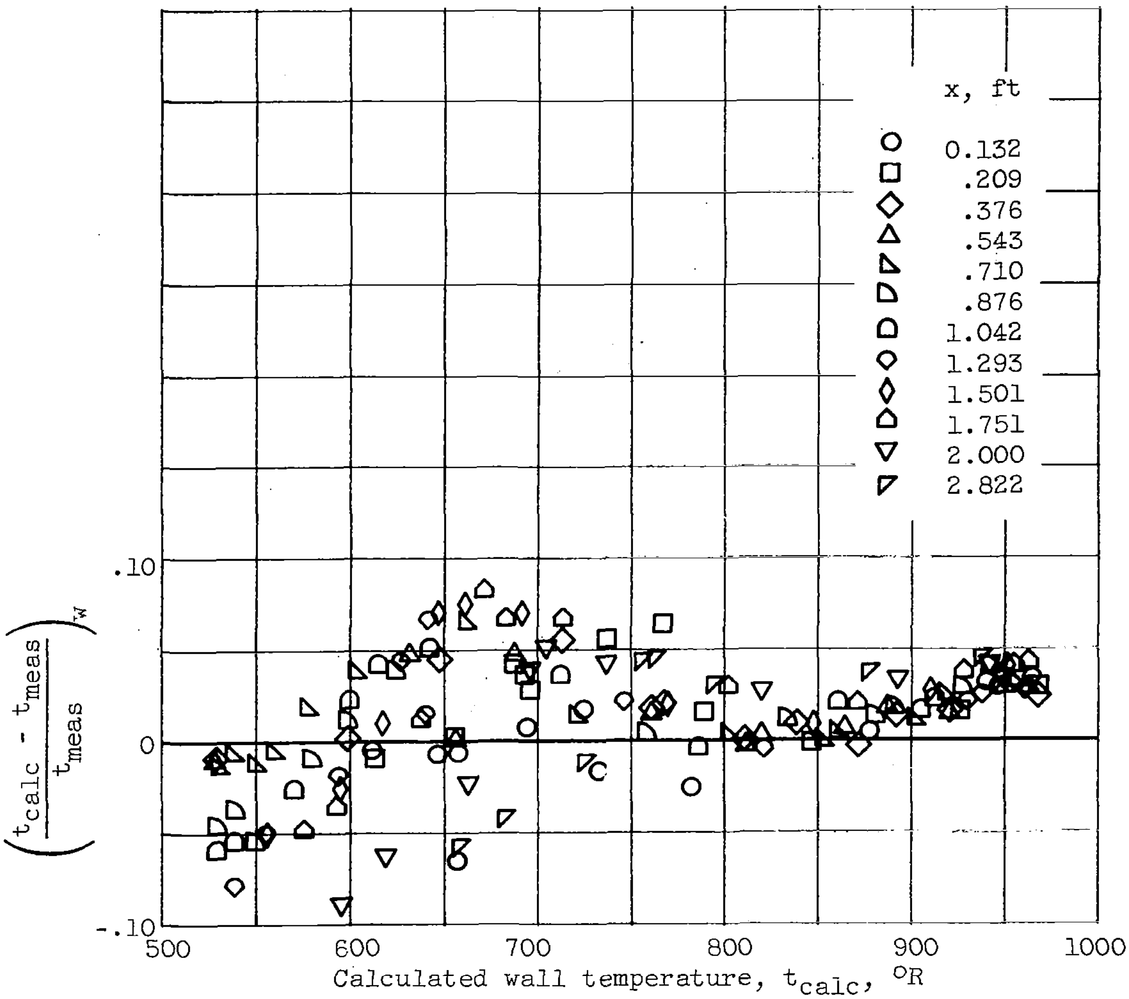
(a) $t_c \approx 534^\circ$ R; $V_g/V_c > 1.0$.

Figure 14. - General Electric air heating data for slot height of 0.106 inch; hydrodynamic starting length, 41.5 inches.



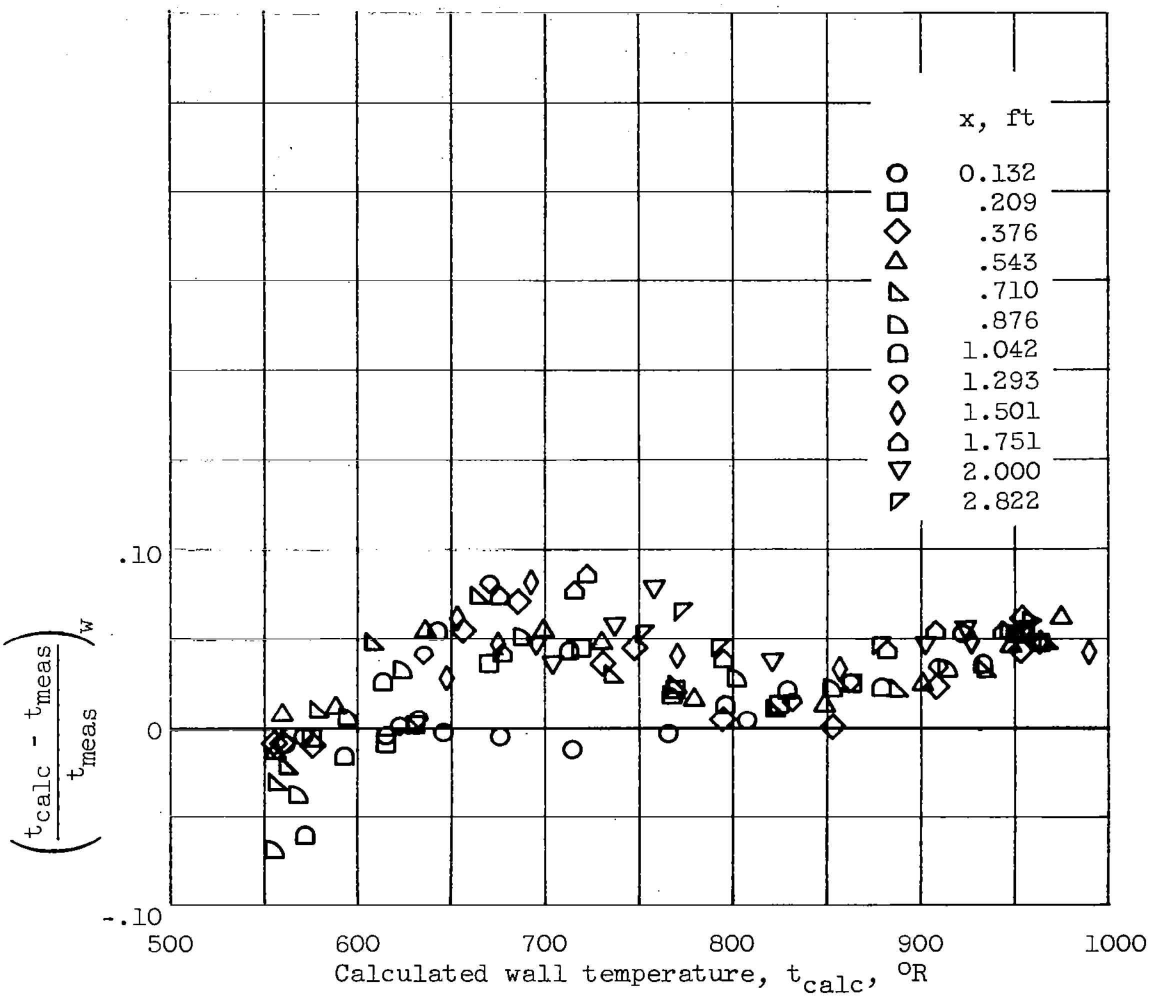
(b) $t_c \approx 605^\circ$ R; $V_g/V_c < 1.0$.

Figure 14. - Concluded. General Electric air heating data for slot height of 0.106 inch; hydrodynamic starting length, 41.5 inches.



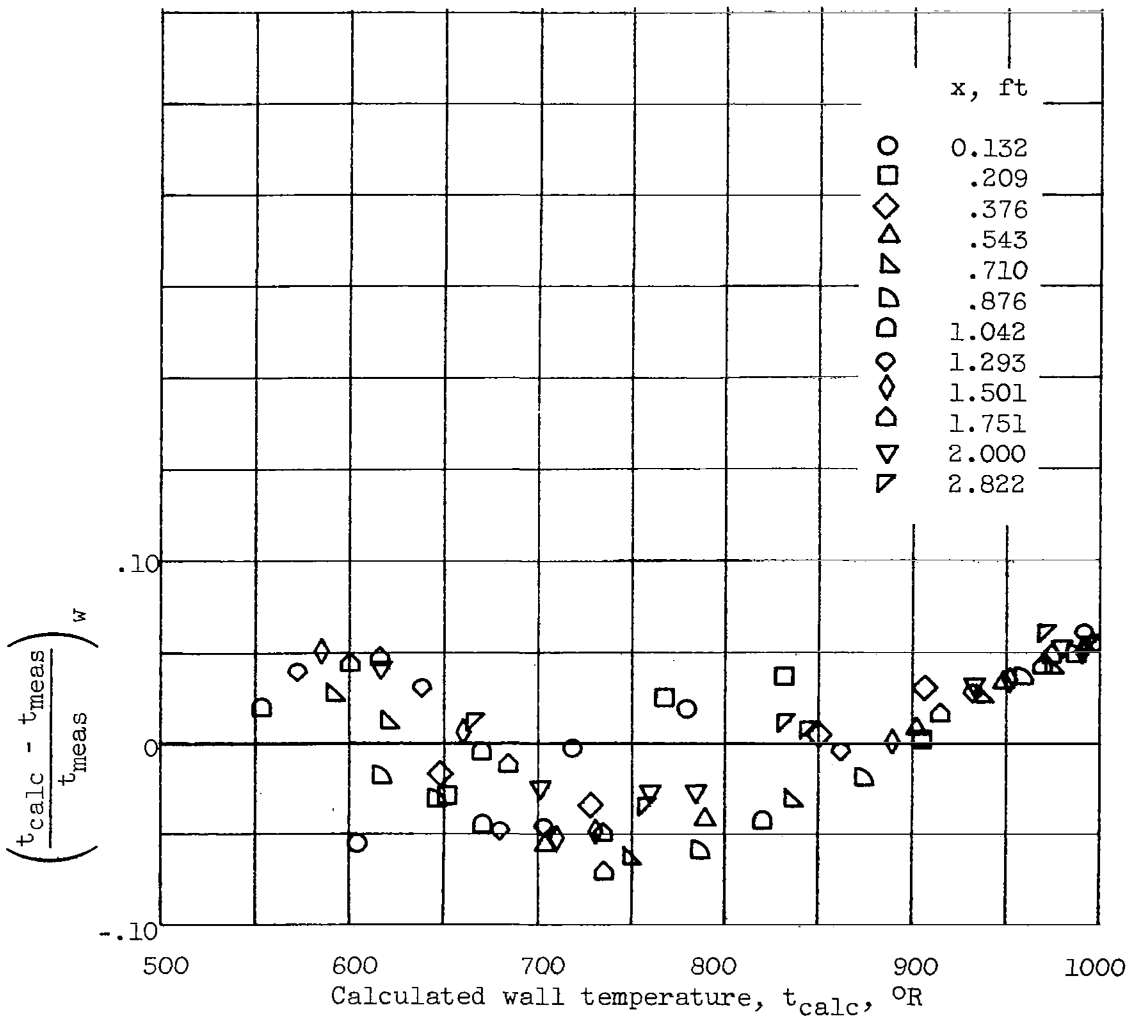
(a) For data presented in figure 7(a).

Figure 15. - Deviation of wall temperature calculated by equation (12) from measured wall temperature.



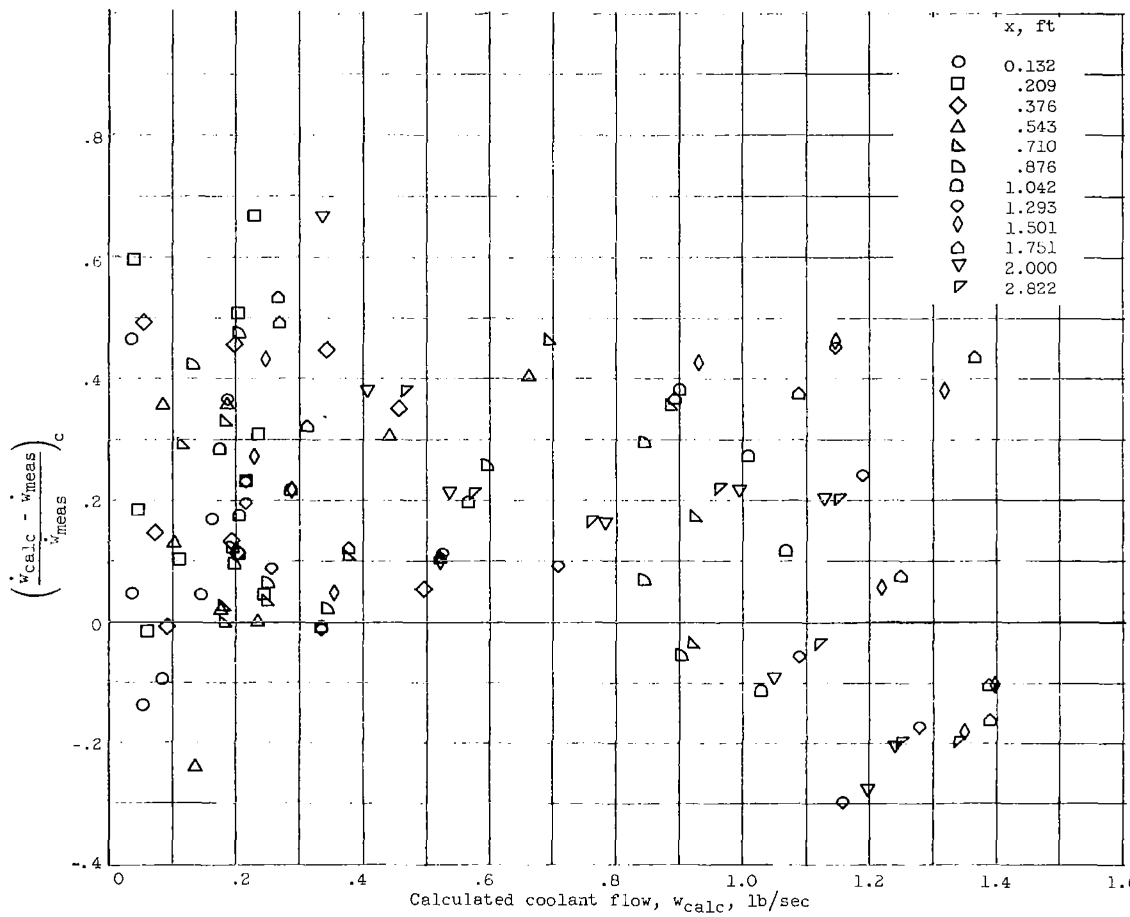
(b) For data presented in figure 7(b).

Figure 15. - Continued. Deviation of wall temperature calculated by equation (12) from measured wall temperature.



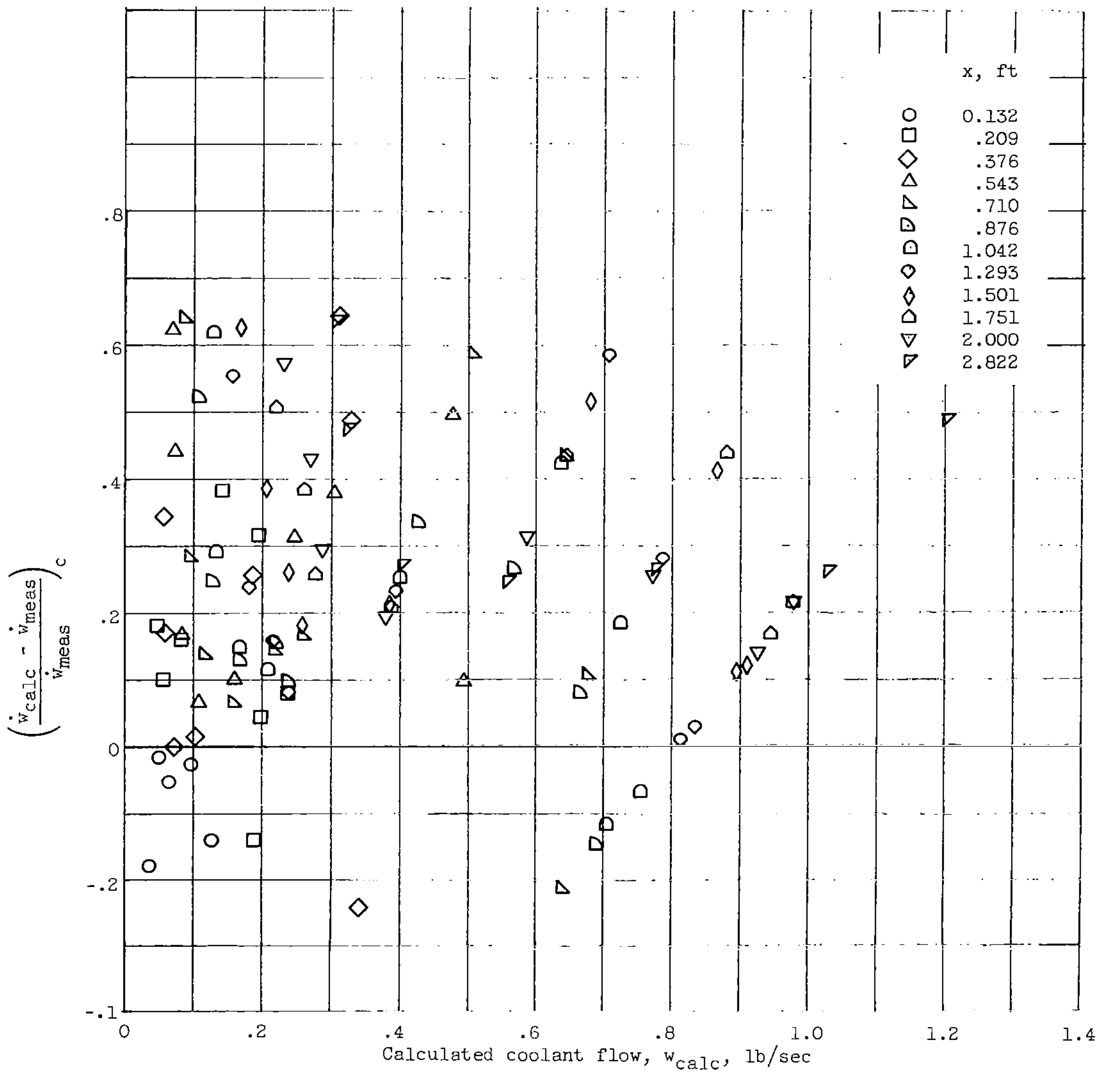
(c) For data presented in figure 10(a).

Figure 15. - Concluded. Deviation of wall temperature calculated by equation (12) from measured wall temperature.



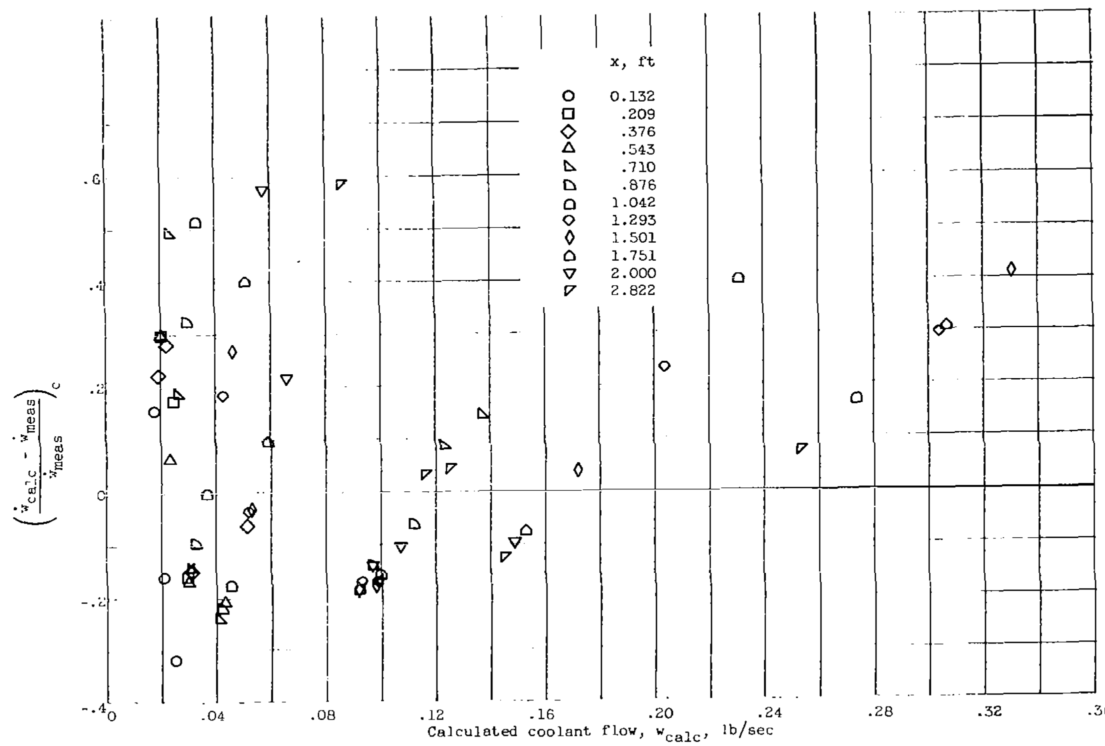
(a) For data presented in figure 7(a).

Figure 16. - Deviation of coolant flow calculated by equation (12) from measured coolant flow.



(b) For data presented in figure 7(b).

Figure 16. - Continued. Deviation of coolant flow calculated by equation (12) from measured coolant flow.



(c) For data presented in figure 10(a).

Figure 16. - Concluded. Deviation of coolant flow calculated by equation (12) from measured coolant flow.