CAVITATION PERFORMANCE OF BERKELEY MODEL 1-1/2 WSR CENTRIFUGAL PUMP WITH WATER AND RELATED SCALE AND THERMODYNAMIC EFFECTS

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Acknowledgements

The major share of data reductions and the calculations for this report were carried out by, and under the supervision of, Mr. Paul T. Chu. The test instrumentation was largely conceived and installed by C. L. Wakamo, assisted by J. Schmidt. The tests were supervised and conducted by V. F. Cramer, assisted by M. J. Robinson. Major contributions to the test program were also made by T. A. Sheehan and A. Travers.

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I. Introduction

Cavitation performance tests have been conducted on the Berkeley Pump Company Model 1-1/2 WSR centrifugal pump, which is used to power the liquid metal cavitation loop for two basic purposes:

1) To obtain a direct comparison of cavitating performance in a turbomachine under conditions of identical geometry between hot and cold water on the one hand and various liquid metals on the other.

2) To obtain basic information on cavitation in a turbomachine for comparison with similar information from a stationary component as a venturi, which is also employed in the present project.

Due to the fact that the facility and pump were designed primarily for operation with high-temperature liquid metals, and hence somewhat compromised to avoid mechanical complexities, the cavitating performance of the pump cannot be obtained as easily or accurately as might be expected under ordinary conditions. However, the difficulties were overcome to the extent that meaningful data could be obtained.

II. Equipment and Instrumentation

A. Limitations

The Berkeley pump is a sump-type centrifugal pump with shaft overhung from a bearing housing located above the sump tank. It is connected into a closed piping loop. The facility and the pump have been previously pictured and described in detail[1,2]. The sump is sealed by a stuffing-box from atmosphere. In liquid metal operation it is intended that the sump be blanketed with inert gas at a pressure slightly above atmospheric allowing a small, controlled out-leakage. Even with the maximum allowable speed of the pump (pump drive is through a variable-speed fluid coupling from an induction motor), cavitation with water can only be attained in the pump if the sump pressure is reduced to a near vacuum or the water vapor pressure is raised to values near atmospheric. It was desired to use both approaches in order to determine the possible "thermodynamic effects," since the parameter denoting the ratio between vapor and liquid volume under equilibrium conditions in the cavitating regions[3,4] changes very considerably between, for example, water at 80 F and 160 F. It was also desired to cover a range of operating speeds to obtain information on "scale effects."

Using either approach to obtain cavitation (both were used to cover the desired range of parameters), it is difficult to obtain accurate data. For cold water or low speed tests, it is necessary to produce quite a considerable vacuum in the sump. This was done by using a vacuum pump, the capacity of which was balanced against the stuffing-box leakage and a controlled by-pass valve (Figure 1). Since the stuffing-box leakage was not constant or negligible (overheating difficulties were experienced if it were unduly tightened), it proved very difficult to obtain steady-state even for brief periods.*

Vacuum requirements for high temperature or high speed runs were of course less, so that the above difficulties were somewhat reduced in these cases. However, data obtained at the higher temperatures includes increased inaccuracy due to temperature measurements, since the

*These difficulties should not be present with mercury since the atmospheric pressure in the sump is only about 2-1/2 feet of suppression head in this case.

dependence of vapor pressure on temperature increases greatly. The value of NPSH becomes quite sensitive to temperature since the vapor pressure is nearly as large as the static pressure. It should be noted that this difficulty, which is not excessively severe in the present case, exists only for tests with low NPSH pumps (which was occasioned in this case by the very low flow, i.e., 40-100 GPM, rather than good cavitation performance).

B. Instrumentation

Two sets of tests were made using quite different instrumentation arrangements. The first was quite conventional, using manometers and/or calibrated gages for pressure measurement. The second used high-response rate pressure transducers feeding a recording potentiometer. It was felt that these were necessary because of the previously mentioned difficulty of attaining steady-state conditions. Actually, the transducers did not give as great an improvement in precision as had been hoped, probably because of drift during the runs.

The remaining instrumentation was typical. Flow was measured by calibrated venturi; temperature by a thermocouple inserted at first into the pump sump, later into the discharge line. No significant differences between these locations were observed. Pump speed was measured through a magnetic pick-up feeding an electronic counter. Accuracy is believed to have been extremely good. Gas content of the water was measured by a Van Slyke meter.

III. Test Results

Twelve distinct categories of test runs were planned. These are classified according to pump speed as a proportion of design speed, flow as proportion of design flow at the applicable speed, and temperature (approximately 90 R or 165 F. A temperature of approximately 90 F, rather than say 60 F, was used to avoid vaporization in the instrument lines which were cooled by the somewhat lower room air temperature.). Actually, three speeds were used (1750, 2420, and 3000 rpm), and two flow ratios (0.93 and 1.20). Afinally, only 10 of the possible 12 categories were run, since it proved impossible to provide sufficient vacuum in the sump to obtain cavitation at the low speed with cold water. The runs and their results are listed in Table I.

Each run consisted actually of several passes through the cavitating performance region (an average of approximately four per run). Thus some information is available upon which to base a calculation of standard deviation. This was done for all runs having more than two passes, and the results are also listed in Table I. The standard deviation is shown on the points in Figures 2, 3, and 4, where it is seen to be sufficiently small that the trend of the curves is significant. This is particularly true in the case of the Thoma cavitation parameter.

A typical plot of the raw data from a run having a somewhat less than average standard deviation is shown in Figure 5 as pump pressure rise versus H_{sv}. The H_{sv} value upon which the cavitation parameters are based is defined as that value for which the pump pressure rise has decreased to 95% of the noncavitating value.

H_{sv} itself was inferred from a static pressure tap located approximately 6 diameters upstream from the elbow leading directly to pump suction (Figure 1), corrected for a small calculated friction loss and elevation change to impeller centerline. Pump discharge also was corrected for the elevation change to the impeller centerline. However, the frictional effects were negligible in this case. The calculation details are given in the Appendix.

H_{sv} values were taken from plots of the data as, for example, Figure 5, and used to compute the Thoma cavitation parameter, σ_T, and the suction specific speed, S. For the presentation of these results, (Figures 2 and 3), pump speed and flow were normalized by dividing by the design speed (1800 rpm) and design flow (40 gpm), respectively.

Examination of these curves shows a considerable "scale effect" in that σ_T decreases between 30 and 50% (depending upon flow) for a speed increase of about 75%. Similar, but inverse, behavior is noted for S. The curves are clearly divided according to the flow ratio. σ_T is appreciably lower for the lower flow and S higher. The standard deviation of the points is such as to make the scale effect and the flow effect both unquestionably significant for σ_T. In the case of S, the scale effect is certainly significant, while the flow effect is also, but less so.

There is no significant differentiation between hot and cold water points, or noticeable effect of gas content (as long as water is settled over a period of several hours to remove large entrained bubbles, as it was for these tests). However, the gas content could only be varied over a range from saturation to about 50% of saturation.

IV. Discussion of Results

A. General

Examination of Figure 3 shows the suction specific speed for the unit is about 2500 at the design point and about 4500 at 1.66 x design speed (and 0.93 x design flow). The fact that these values are less than conventional practice would indicate is believed due to two factors:

1) There is a standard long-radius elbow directly upstream of the pump suction. This is not a part of the pump itself, but rather of the loop.

2) The impeller was not especially designed for good cavitation performance.

The low S values are not believed injurious to the overall test objectives since only a comparison between fluids in the same geometrical configuration was desired. No attempt to develop a high suction specific speed impeller is involved at present. The low values are an advantage in that cavitation with liquid metals should be more easily obtained.

B. Scale and Thermodynamic Effects

It has become increasingly recognized of late that significant scale effects and also thermodynamic effects, causing departures from the idealized theory (on which the concept of the Thoma parameter and suction specific speed are based, for example) do exist in cavitation performance[4,5,6,7,etc.]. An almost identical effect (wherein suction specific speed increased 40% for a speed increase of 50%) to that shown in Figure 3 is listed by Stepanoff[4], quoting a Russian paper[8]. Also a somewhat similar effect (significant decrease of cavitation number for increase of Reynolds' number has been noted in the present investigation for cavitation in a venturi (Figures 18 through 21 of Reference 2).

The venturi tests correlate well in terms of Reynolds' number (Figure 6), but only very poorly in terms of velocity (Figure 7). These figures are replotted from the data of Reference 2; Figure 6 is Figure 19 of Reference 2 with standard deviations added, and Figure 7 is the same data plotted against normalized throat velocity. This partial correlation in terms of two different parameters is possible because of the relatively small temperature range, and the fact that most of the high Reynolds' number points are also high temperature points, and hence to some extent all shift together when the abcissa is changed from Reynolds' number to velocity.

Visible initiation in the venturi was chosen for comparison with the pump data because the proportional head effects are similar and hence, presumably, the degree of cavitation.

The pump curves were shown in Figures 2 and 3 plotted against normalized velocity. It is noted that the correlation is quite good in terms of the standard deviation which is shown. Figure 4 is a replot of the data of Figure 2 in terms of Reynolds' number. A reasonably good correlation is obtained although not quite so good as that in terms of velocity. The shape of the curves is similar, but the slope of the curve in terms of velocity is somewhat steeper.

Fairly clear-cut arguments can be advanced to explain the correlations in terms of Reynolds' number, although it seems unlikely that Reynolds' number should be the only factor of significance. A summarization of these arguments follows.

If it is considered that cavitation bubbles originate in the boundary layer and also that local under-pressures in the fluid are a function of degree of turbulence, it is not surprising that Reynolds' number should to some extent at least correlate cavitation data. Other possible parameters, as listed for instance in Reference 5, include Froude number, Mach number, Weber number, and Peclet number. If tests at a fixed temperature and over only a moderate velocity range are considered, it seems unlikely that effects due to variation of Mach or Peclet numbers could be serious. Weber number does not vary significantly over the test temperature range since surface tension is only a weak function of temperature. Froude number does not vary significantly in the tests. Hence, it might be concluded that, at least for fixed temperature, Reynolds' number alone would be significant. In fact, a variation of cavitation number with Reynolds' number has recently been derived by Oshima[6] by considering only inertial, pressure, and surface tension effects. However, the predicted variation is in the opposite direction to that observed in the present tests.

In the pump tests, the cavitation conditions are defined as those producing a pump head reduction of 5%. As suggested by Stepanoff[4] and Salemann[9], tentatively this can be taken to mean a given vapor volume within the pump. Again, as originally suggested by Stepanoff[4], considerably greater local head depression for hot than for cold water, at least under constant equilibrium conditions, is required to produce the sensible heat from the surrounding liquid for production of the requisite vapor. This was presented in slightly different form by the present author[3] as a parameter equal to the ratio of vapor volume to liquid volume per unit head depression. This parameter is plotted over the range of the present test temperatures in Figure 8, where it is seen to vary by a factor of about 5. In other words, under equilibrium conditions, a head depression, 5-fold increased over the cold water tests, should be required to produce the same proportionate head reduction in the hot water tests. It is apparent that this is not quantitatively meaningful in the present case, since the head depression for the cold water tests is of the order of 4 feet, and about the same for the hot water tests. However, the trend characterised as "thermodynamic effects" in the current literature certainly exists as shown by the test results of Stepanoff[4] and Salemann[5], and also Jacob[10].

It thus seems reasonable to assume that separate Reynolds' number and thermodynamic effects exist. Assume for centrifugal pumps, on the basis of the present tests, as well as those of Reference 8, that the

Reynolds' Number effect alone, for tests at constant temperature, results in a decrease of the Thoma parameter with increase of Reynolds' Number (pump speed), and that the thermodynamic effects result in a further decrease of Thoma parameter (assuming a fixed proportionate head loss) for increased B-factor[4,3], i.e., as occasioned by increased temperature for a given fluid. Then the data shown in Figure 4, which is plotted in terms of Reynolds' Number, should show separate hot and cold water curves. That this is not the case either with the pump (Figure 4) or venturi (Figure 6) data may be the result of the fact that the thermodynamic effect is relatively negligible over the temperature range tested.

No direct comparison can be made with the test results listed by Stepanoff and Salemann since the vapor pressure values for the present tests are far below those for which they tested. However, extrapolation of their data does indicate that the thermodynamic effect would not be appreciable in these tests.

The venturi cavitation data[2] from the present research investigation correlates considerably better in terms of Reynolds' Number than of velocity. (Figures 6 and 7). However, in these tests, a fixed proportionate head loss for the different tests does not exist. Rather, the extent of the cavitating region was fixed visually, and it was shown that the head loss, under these conditions, for substantial cavitation was greater for cold water than for hot, as expected (see Figures 28 and 29 of Reference 2, although the differentiation for visible initiation, for which comparison should be made, is too small to be significant within the precision of these tests - Figure 27, Reference 2). However, the argument upon which prediction of the thermodynamic effects is based, i.e., fixed proportionate head loss, does not apply. If the points of Figure 6 (visible initiation for the venturi), or similar curves for more advanced cavitation (Figures 20 and 21 of Reference 2), are adjusted according to the venturi loss data (Figures 28 and 29 of Reference 2, for example) so that the head loss ratio for points at a given Reynolds' number is constant, then the hot water point for a given Reynolds' number should be considered at a greater visual degree of cavitation (as visually observed) to give the same head loss as a cold point at that same Reynolds' number. This would result in a separate curve for hot water which would be at somewhat lower cavitation numbers than the cold water curve (i.e., the present curves of Figures 18-21 of Reference 2), since cavitation number generally decreases as the cavitating region is increased (Figure 22 of Reference 2). If curves on this basis were plotted against velocity rather than Reynolds' number, the separation between hot and cold water curves would be increased, rather than tending to collapse into a single curve when plotted in this fasion as the pump curves apparently do.

No rationalization is at present possible of the fact that the pump data correlates best in terms of velocity and the venturi data only in terms of Reynolds' number. The above arguments regarding thermodynamic effects indicate adjustments in the wrong direction if it were to be assumed that the venturi and pump plots were equivalent. Of course, there are basic differences between rotating and stationary systems which may be responsible. It is noted that a similar correlation of cavitation number over a restricted temperature range with Reynolds' number for cavitation on a stationary body was presented in Reference 7.

An examination of the existing literature regarding scale effects in pumps and on stationary objects[2,4,5,7,8] shows that cavitation index sometimes increases with increased velocity or Reynolds' number and sometimes decreases. In particular, the tests involving centrifugal pumps (the present tests and Reference 8) show a decrease. The results on high pressure coefficient bodies, as summarized in Reference 5, show the opposite variation. However, results for a low pressure coefficient body (an NACA 16012 Hydrofoil in Reference 5) shows a decrease with Reynolds' number, as does the data from the venturi tests of the present investigation[2]. The hydrofoil and venturi seem somewhat comparable in this respect. Also, some similarity between a hydrofoil and the blades of a centrifugal pump may be justified.

Predicted Mercury Performance

For similar velocities in the same equipment, the Reynolds' numbers for mercury will be considerably greater than those for water (factor of 3 to 8 depending upon temperature). The thermodynamic effects as compared through the equilibrium volume ratio (B-factor) should not be great since the vapor volume per liquid volume per unit head depression is a factor of 10^7 less for mercury than for cold water[3]. Since the inlet head depression required to obtain a given cavitation effect on pump head differential is only a small portion of H_{sv} with cold water, and since it will be presumably greatly decreased for mercury, the change in cavitation parameter from this effect should not be very substantial. While this is in the opposite direction from the Reynolds' number effect, an examination of Figures 2 and 3 indicates that the Reynolds number effect will probably be fairly large. Hence, if these effects alone control, it would be expected that the cavitation parameters for mercury will be substantially less than for water.

On the other hand, the effect of surface tension is proportionately less for mercury , perhaps making cavitation easier and raising the cavitation number.

V. Conclusions

It is concluded from the pump cavitation tests with water that the Thoma cavitation parameter decreases substantially for increasing pump speed for a fixed flow coefficient. Curves for hot and cold water over the range from 90 to 160 F coincide. Corresponding comments apply to suction specific speed. These trends are in approximate quantitative agreement with the only similar tests known to the author in the literature.

It is tentatively concluded that major effects resulting in deviation from classical theory in a pump or a stationary member can be divided into either Reynolds' number or velocity effects (it is not certain at present which is most suitable, although tests with mercury in the present facility may go far to resolve the uncertainty) and thermodynamic effects. A detailed examination of the results indicates, however, that these alone are probably not sufficient, and the correlations are probably partly fortuitous.

The direction of variation of cavitation number with Reynolds' number in the venturi tests was the same as in the pump tests. This does not conflict with present literature which shows Reynolds' number effects in either direction for stationary objects. No similar tests on venturis are known to the author for direct comparison.

It is also tentatively concluded that cavitation numbers for mercury in the same geometry and at the same velocity will be less than for water because of the large Reynolds' number effect and the relatively small thermodynamic effect in the applicable range of variables. However, a counter trend could be indicated by a Weber number effect.

Run No.	Temp. °F.	N/No.	Q/Q_o	σ_T	S	σ_σ	σ_S
9	166	0.97	0.93	0.1732	2351	0.033	353
10	162	"	1.2	0.232	2572	-----	----
2	83	1.343	0.93	0.802	4144	0.0067	247
1,12	85	"	1.2	0.209	2927	0.0251	240
8	167	"	0.93	0.1071	3437	0.0301	569
11	162	"	1.2	0.2065	3033	0.0216	225
3	88	1.665	0.93	0.0865	3930	-----	---
13	97	"	1.2	0.1846	3192	0.0155	216
7	166	"	0.93	0.0687	4935	0.0182	979
6	161	"	1.2	0.1599	3516	0.0046	83
4	93	"	1.2	0.192	3747	0.0163	274
20	120	1.343	0.93	0.1214	3200	0.0438	725
21	110	"	1.2	0.1925	3650	0.0239	345
22	125	1.665	0.93	0.0884	4240	-----	
23	125	"	1.2	0.1618	4040	-----	
Averages						0.023	387

TABLE I

Summarization of Results and

Standard Deviation

VI. Appendix

A. Error Analysis

Each of the experimental points used (with one exception) is the result of several actual runs, usually four. According to standard procedures*, these are used to compute the standard deviations for each point which consists of more than two separate runs. These are summarized in Table I, where it is noted that the average standard deviation for the Thoma cavitation parameter is 0.023 and that for the suction specific speed is 387. The actual values do not vary by orders of magnitude from the average values. The runs with standard pressure gages, rather than high-response transducers (runs 20-23) show somewhat greater deviation but not by an order of magnitude.

The average deviations are shown on the figures with each point to give some idea of the reliability of the points.

The standard deviations computed from the repetitive runs do not quite include all possible errors. The runs were made in quick succession so that a single temperature reading was used for all. For the low temperature runs, the dependence of vapor pressure on temperature is so small as to make error on this account definitely negligible. For the highest temperatures used, an error of 1°F corresponds to a vapor pressure error of about 0.2 feet. On this basis it is believed that the likely vapor pressure error is no more than 0.1 feet, corresponding to an error in N_{sv} of no more than 2% and in most cases less than 1%. This is quite negligible compared to the scatter of the data.

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*

$$\sigma_x = \sum_{i=1}^{n} \frac{(x_i - \bar{x})^2}{n - 1}$$

where x_i = data
$\bar{x}$ = average of x_i
n = number of runs
= standard deviation

B. Data Processing

The basic equations used in reducing the suction specific speed data are

$$S = \frac{N\sqrt{GPM}}{H_{sv}^{3/4}} \quad \text{—— (1)}$$

$$H_{sv} = \frac{P_{in}}{\rho} + \frac{v^2}{2g_c} - \frac{P_v}{\rho} \quad \text{—— (2)}$$

$$P_{in} = (P_{static})_{in} - \Delta Z - \frac{L}{D}\frac{v^2}{2g_c}f \quad \text{—— (3)}$$

$$P_{out} = (P_{static})_{out} - \Delta Z - \frac{L}{D}\frac{v^2}{2g_c}f \quad \text{—— (4)}$$

The meaning for the above symbols is listed in Nomenclature.

Thus, according to the above equations, the measured quantities should be RPM, GPM, temperature and the static pressures at both ends of the pump. Since the transducer registers only pressure differences, the barometer pressure reading becomes necessary.

As mentioned in the text, pressure readings are obtained directly from a recording potentiometer in most of the tests and from Bourdon gages in some. Eqs. (3) and (4) are then computed. H_{sv} in Eq. (2) can be calculated from the result of Eq. (3). ΔP of the pump is readily obtained from $P_{out} - P_{in}$. One thus plots ΔP vs. H_{sv}. The critical H_{sv} is taken at the point where the pump pressure differential is reduced by 5%. A typical curve is shown by Figure 5.

Following is a representative calculation:

Temp. = 83°F

Thermocouple correction = 0.65 from calibration curve

Actual fluid temp. = 83.65°F

Saturated vapor pressure = 0.5643 psia (Reference 11)

Barometric pressure = 14.35 psia

Flow rate = 52 GMP from venturi calibration curve

RPM of pump = 2420

(1) ID of the pipe = 1.61 inches

Velocity of the fluid = 8.81 fps

$Re = 1.141 \times 10^5$

$f = 0.017$ for pipe of type used and above Reynolds' number

(2) Suction side pressure correction

$\Delta Z = 5.5'' = 0.458$ ft.

Equivalent length of piping = 4.0 ft.

$\Delta h_f = 0.525$ ft. $\Delta Z + \Delta h_f = 0.983$ ft. = 0.425 psi

$\therefore P_{in} = (P_{static})_{in} - 0.425$ ---------- psia

(3) Density of water = 62.1 lb_m/ft^3

$\frac{P_v}{\gamma} = 1.31$

$\frac{V^2}{2g_c} = 1.04$

$H_{sv} = 2.31\ P_{in} - 0.27$ ----- ft.

(4) Discharge side pressure corrections

$\Delta Z = 1.1$ ft.

Equivalent length of piping = 0.91 ft.

$\Delta h_f = 0.066$ ft.

$P_{out} = P_{static}$ ~~psig~~ 50 ---- psia

(5) Barometric pressure = 14.35 psia

Therefore the working equations are

$P_{in} = (P_{static})_{in} + 13.925$ --------psia

$P_{out} = (P_{static})_{out} + 14.85$ -------psia

$H_{sv} = 2.31\ P_{in} - 0.27$ ------------ft.

$\Delta P = P_{out} - P_{in}$ -------------psi

VII Nomenclature:

P	Pressure
S	Suction specific speed of pump
N	RPM of pump
H_{sv}	Net positive suction head
P_v	Saturated vapor pressure
	Height
V	Velocity of the fluid
σ	Depending on the topic, σ may symbolize the cavitation number which, by definition, $\sigma = \left(\frac{\Delta P}{\rho}\right) / \frac{v^2}{2g_c}$ On the other head, σ may symbolize the "Standard Deviation" in the common statistical sense. Thus σ_σ means the standard deviation of cavitation number
	σ_T means Thoma cavitation parameter
B	Thermodynamic coefficient, equilibrium vapor volume per liquid volume per unit head depression $= \frac{\rho_L}{\rho_V} \frac{C_p(\Delta T/\Delta H)}{h_{fg}}$
ρ_L	Density of liquid
ρ_V	Density of vapor
c_p	Specific heat of liquid
ΔH	Head depression below saturation pressure
ΔT	Temperature change of liquid corresponding to ΔH
h_{fg}	Latent heat of evaporation

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SCHEMATIC OF PUMP TEST SET - UP

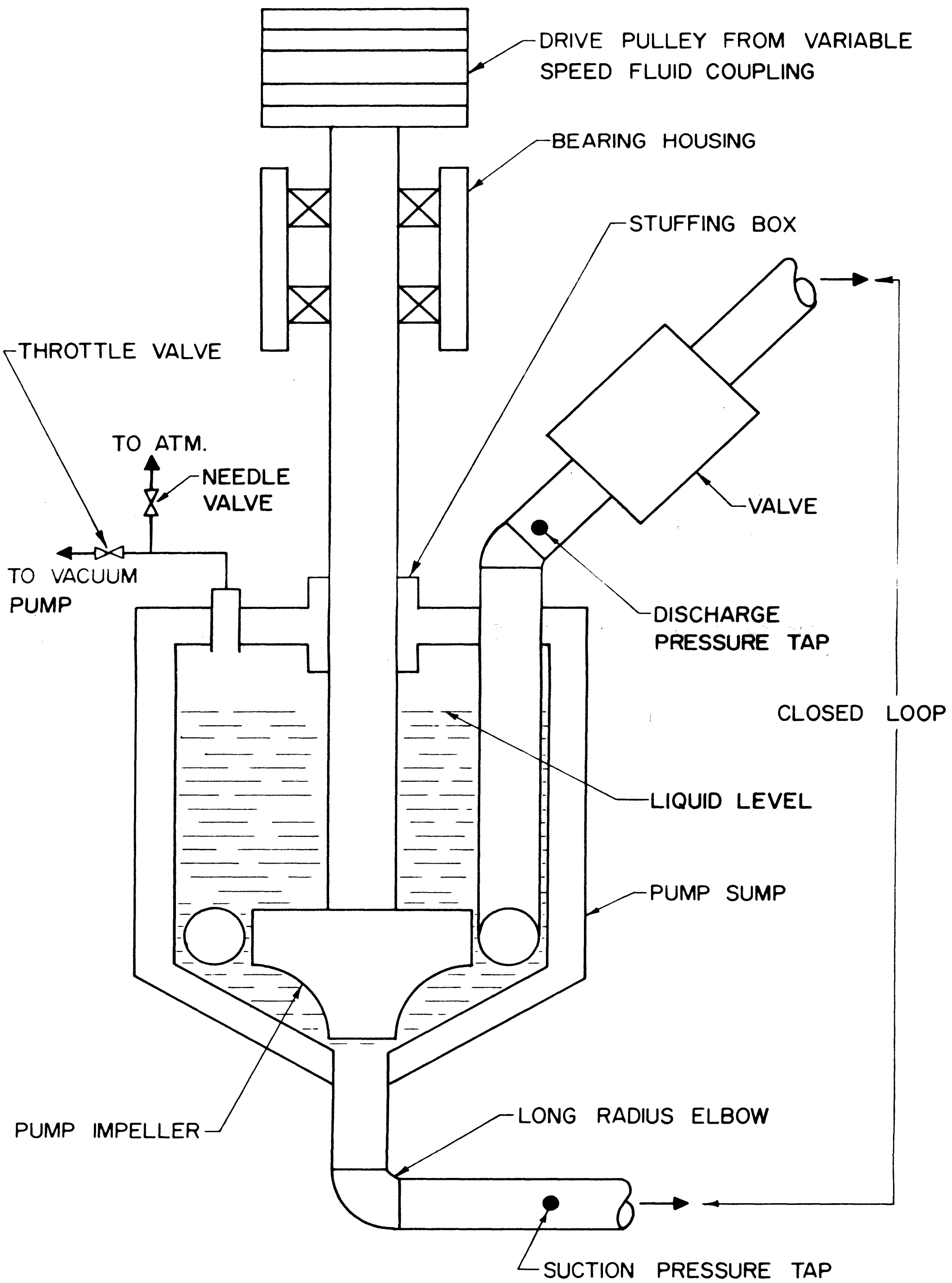


FIGURE 1

THOMA CAVITATION PARAMETER
VS.
NORMALIZED PUMP SPEED

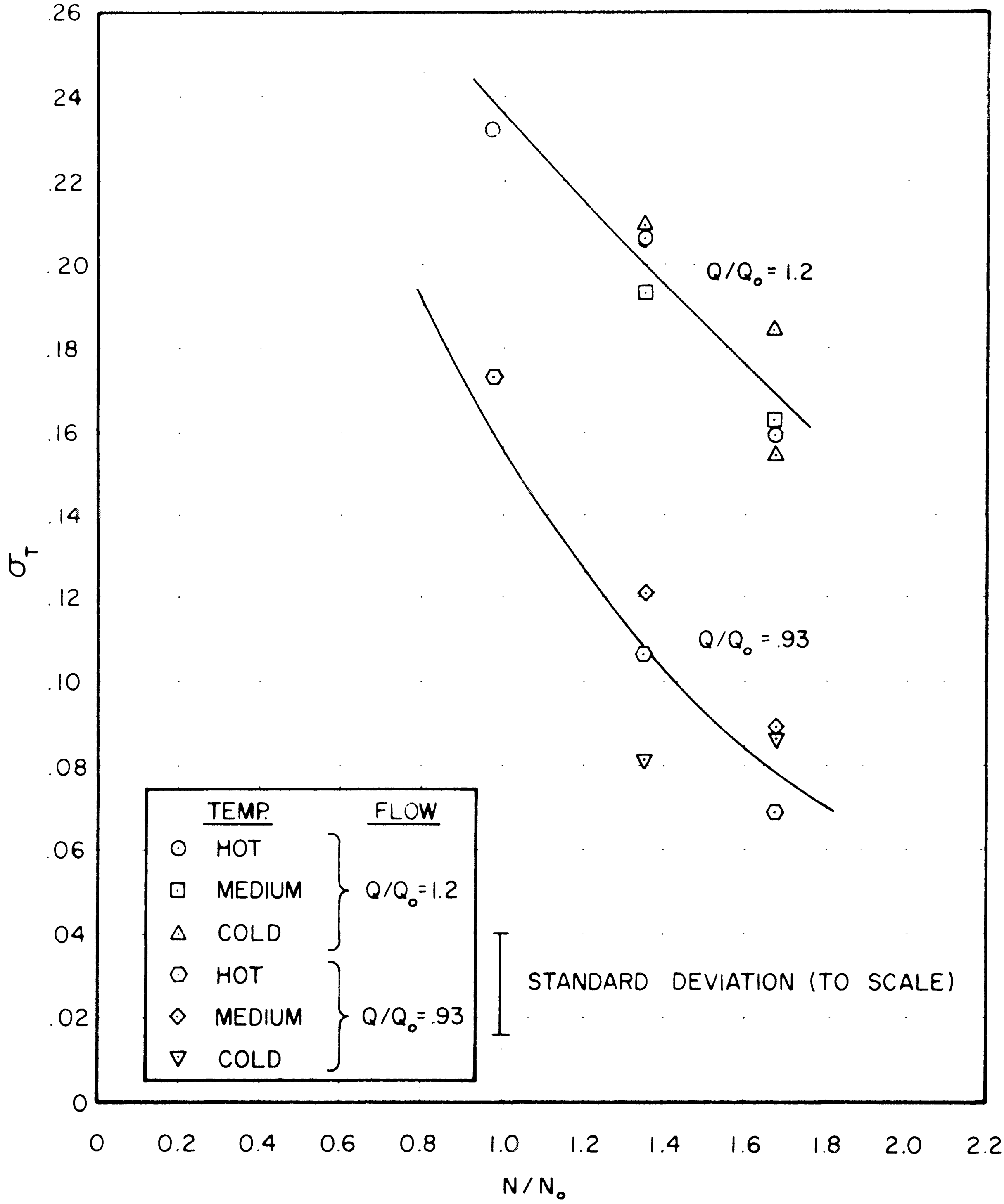


FIGURE 2

SUCTION SPECIFIC SPEED

VS.

NORMALIZED PUMP SPEED

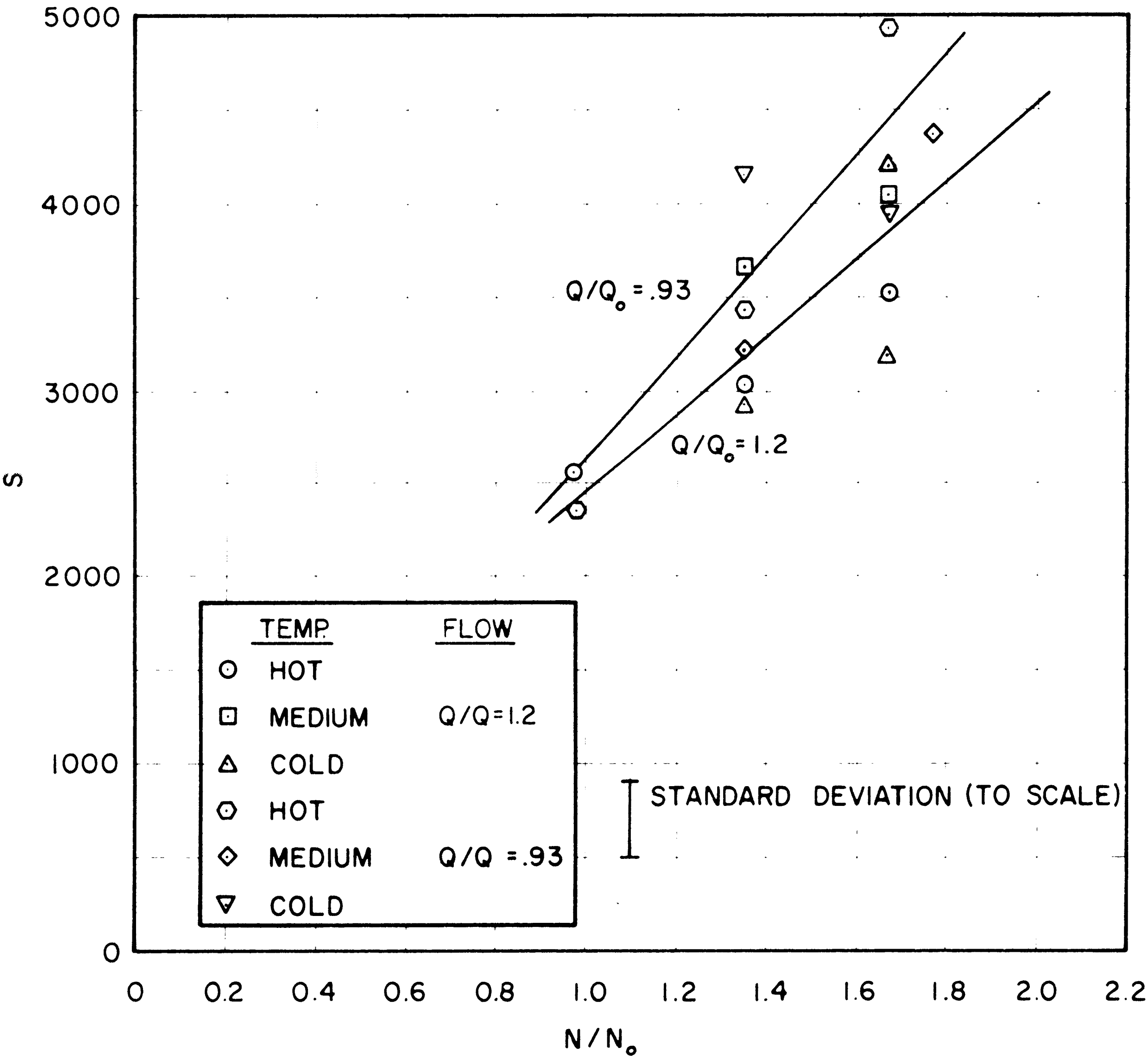


FIGURE 3

THOMA CAVITATION PARAMETER

VS.

NORMALIZED REYNOLDS' NUMBER

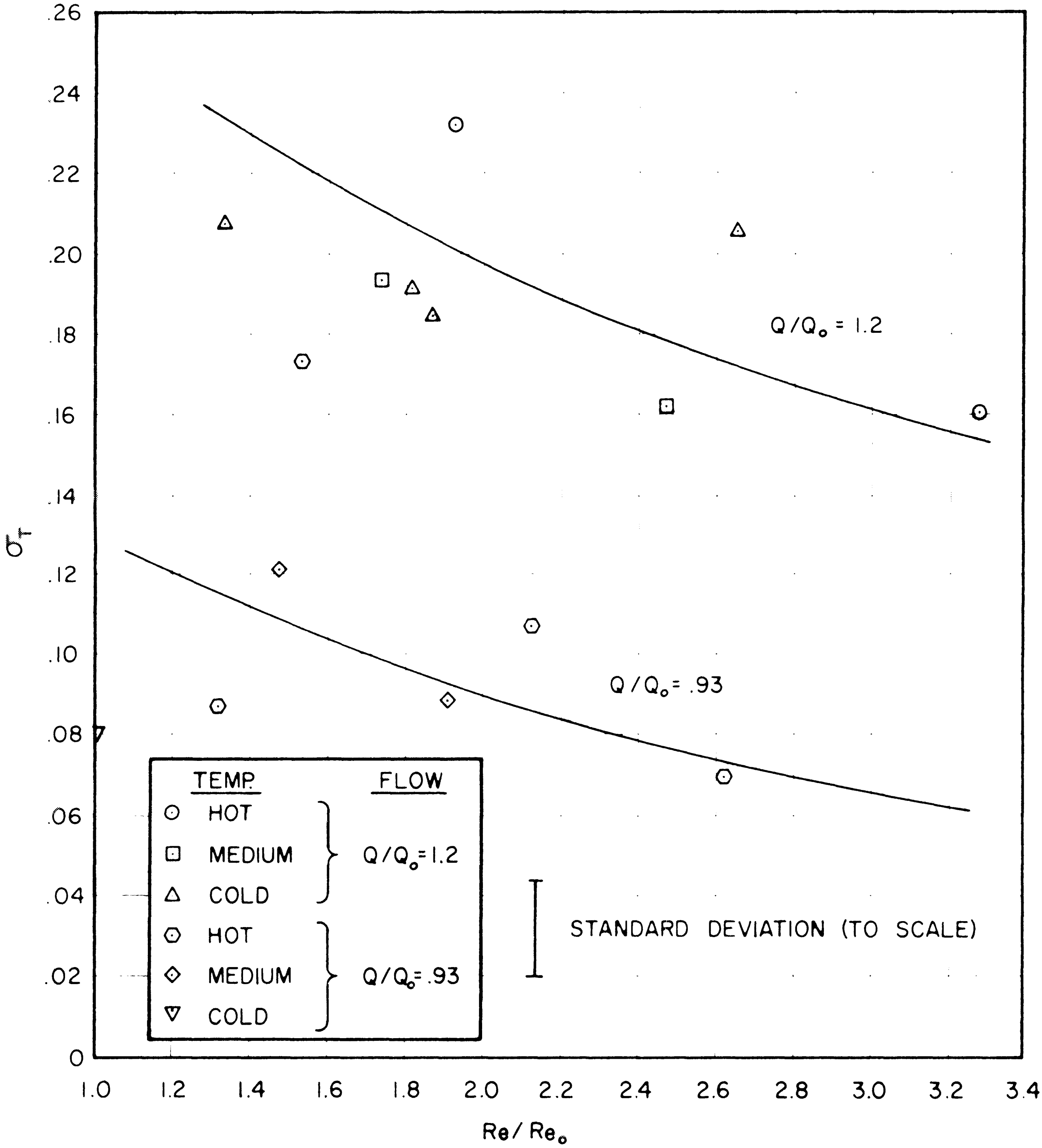
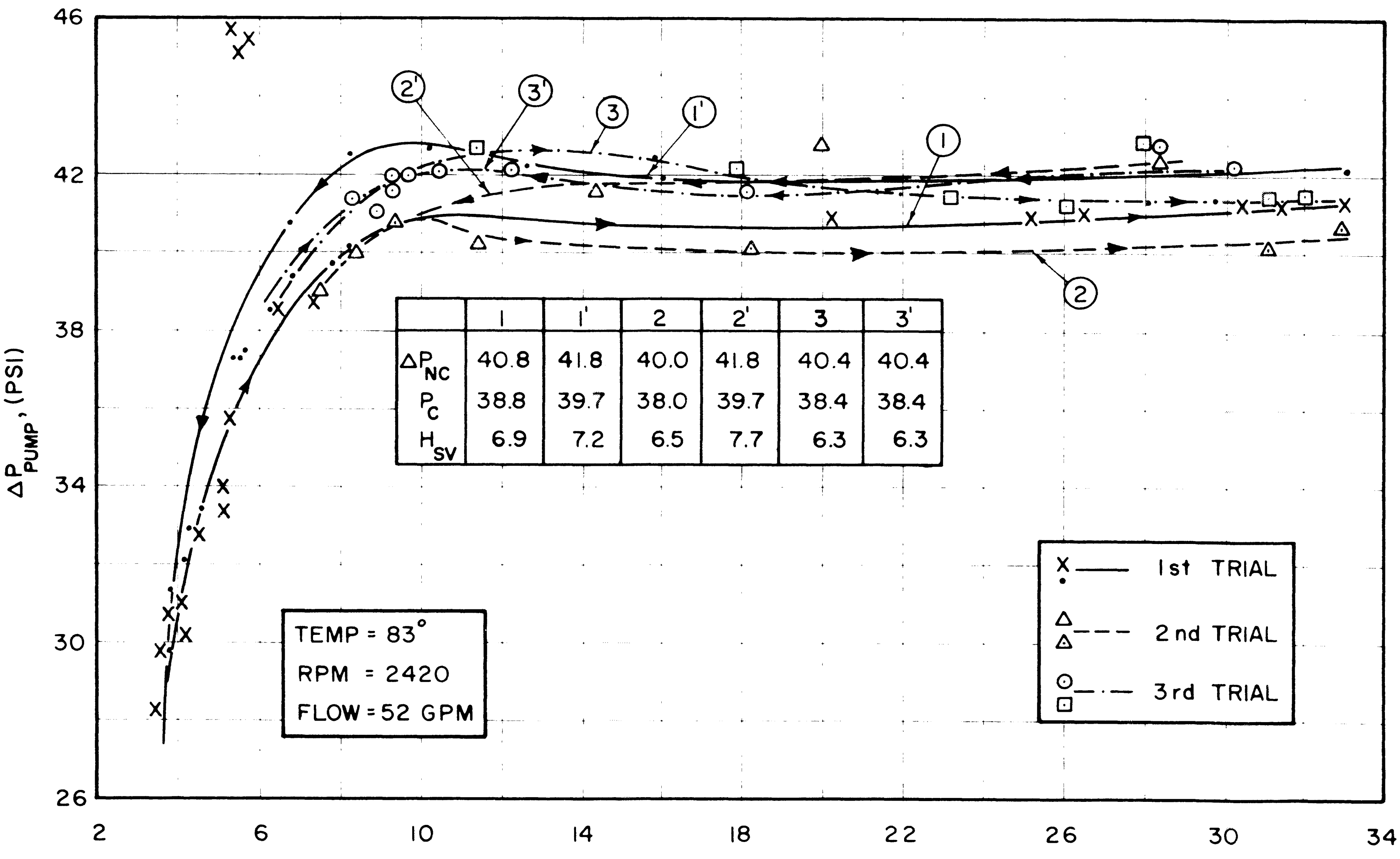


FIGURE 4

PUMP CAVITATION PERFORMANCE CURVE, RUN NO. 2



	1	1'	2	2'	3	3'
ΔP_{NC}	40.8	41.8	40.0	41.8	40.4	40.4
P_C	38.8	39.7	38.0	39.7	38.4	38.4
H_{SV}	6.9	7.2	6.5	7.7	6.3	6.3

FIGURE 5

CAVITATION NUMBER vs. THROAT REYNOLD'S NUMBER
IN A CAVITATING VENTURI FOR VISIBLE INITIATION

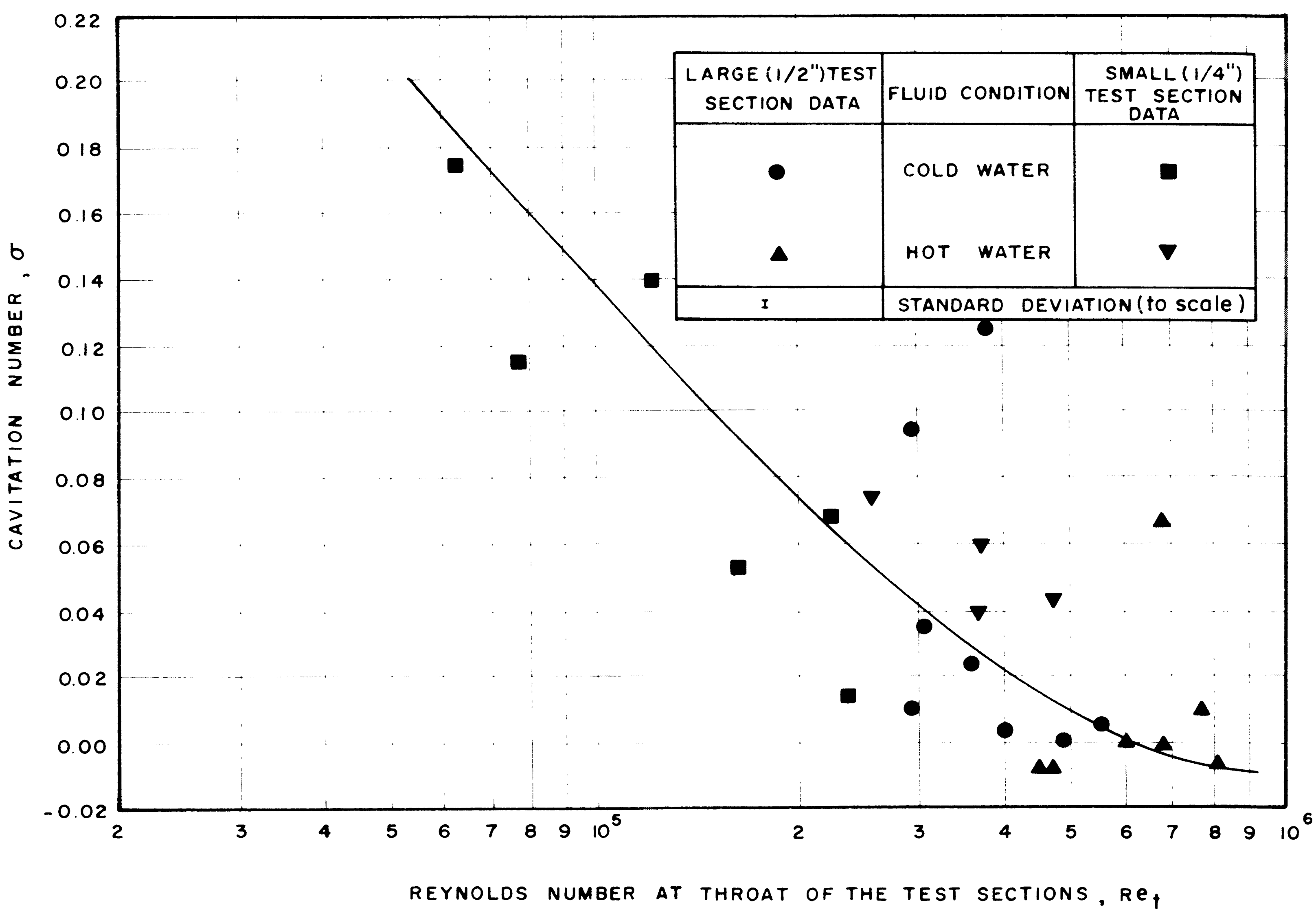


FIG. 6

CAVITATION NUMBER vs. THROAT VELOCITY
IN A CAVITATING VENTURI FOR VISIBLE INITIATION

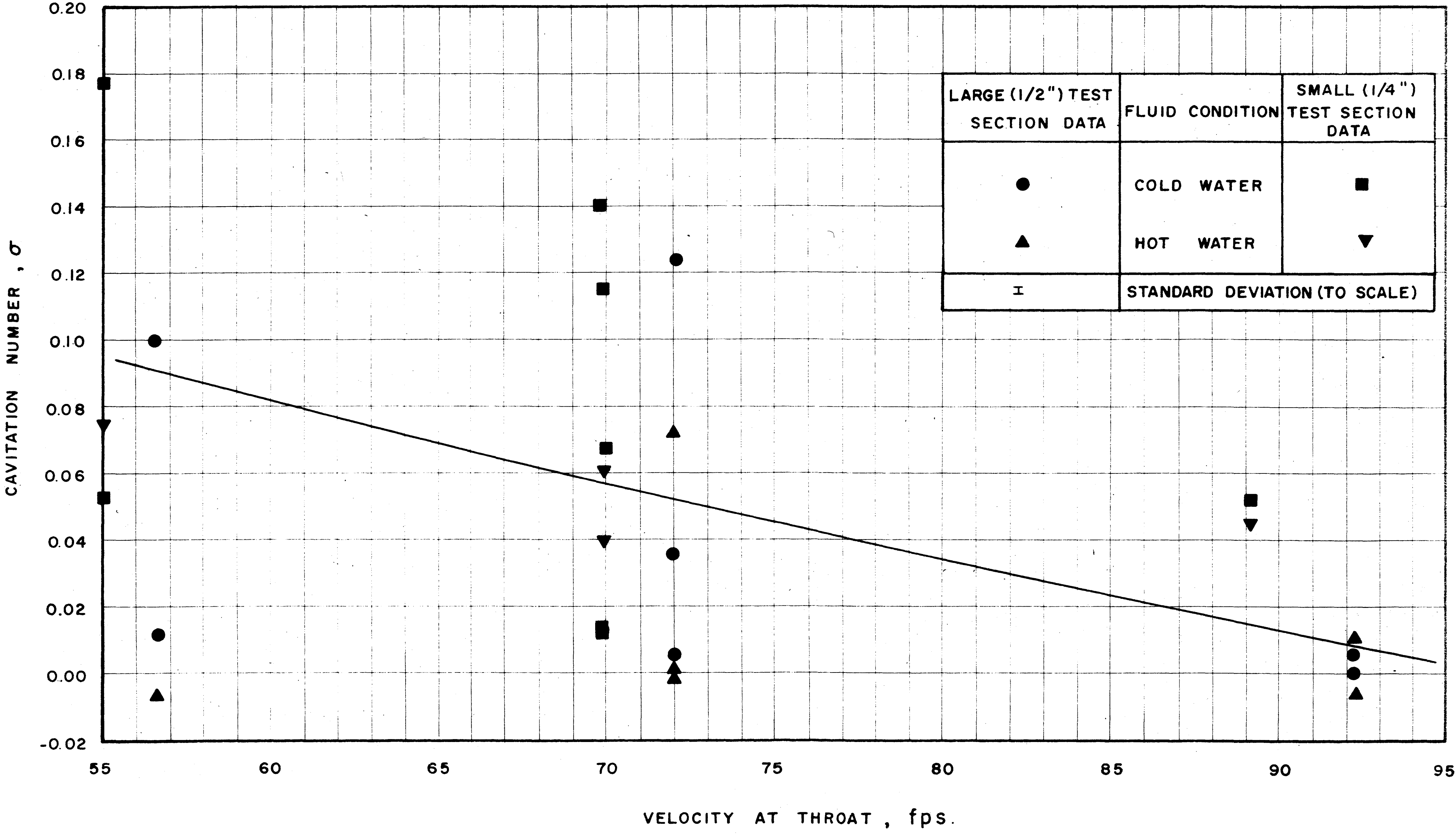


FIG. 7

THERMODYNAMIC PARAMETER, B

VS.

TEMPERATURE

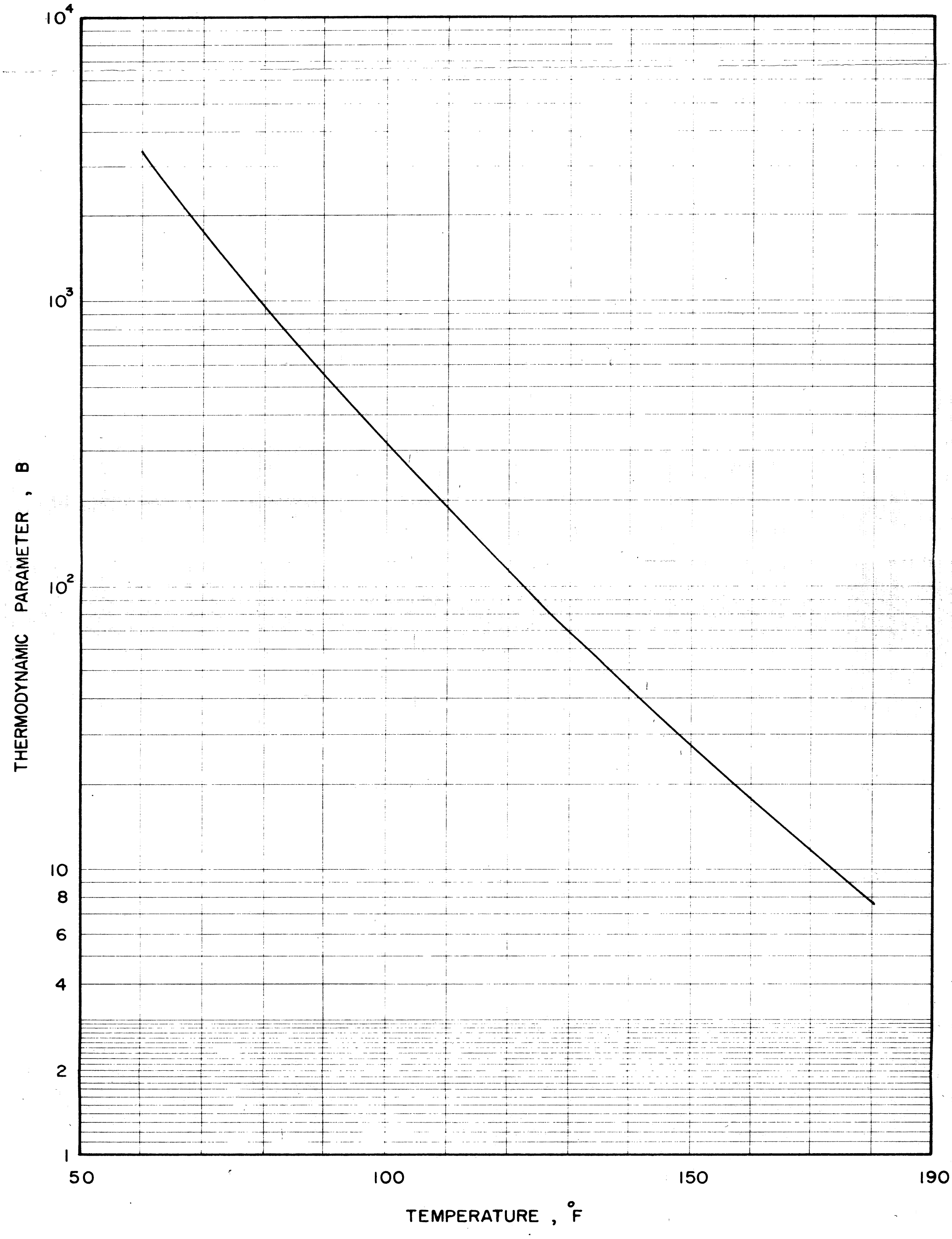


FIGURE 8