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COOLDOWN TIME FOR SIMPLE CRYOGENIC PIPELINES*

by

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ABSTRACT

An uncooled pipeline which is used to transfer a cryogenic fluid from one point to another must ordinarily go through a period of cooling down from ambient temperature to near the liquid boiling temperature. During most of the cooldown period the liquid boils and the pipe delivers only warm gas. This paper offers a quick method by which cooldown time for a simple system can be estimated from a dimensionless parameter read from a graph. To use the method it is necessary to know the fluid and pipe enthalpy, density, and velocity of sound in the warm gas. The idealized model and closed form solution are described, and comparison with experimental results is shown.

KEY WORDS: Chilldown, Cooldown Time, Cryogenic Fluids, Transfer Lines

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INTRODUCTION

When an uncooled pipeline is used to transfer a cryogenic fluid from one point to another, it must ordinarily be cooled from ambient temperature to near the liquid boiling temperature. During most of the cooldown period the liquid boils and the pipe delivers only warm gas; thus, how soon it will start delivering liquid can be an important operational consideration. This paper offers an easy method of estimating cooldown time for simple piping systems of relatively large length-to-diameter ratio. A simple system is defined here as one which has a constant flow area, is without concentrated masses, orifices or other constrictions, and is well insulated. Since the model is greatly simplified the results should be considered as design aids only; however, agreement with experimental results is good.

Earlier works [1, 3, 4]* on this subject have been few in number. The methods of Burke et al. [1] and Chi [4] involve the use of properties and flow rates averaged over the entire cooldown process. Since the variations are large and nonlinear, such averaging seems undesirable. The analysis of reference [1] accounts for heat transfer from the surrounding atmosphere and estimates the effect of concentrated masses of warm material. A liquid-gas interface assumption used by Burk et al. is also used in the present analysis. Macinko's method [3],

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* Figures in brackets indicate literature references.

cryogenic fluid in a time considered short when compared to the total cooldown time. Then the temperature histories at all stations along the pipe would essentially coincide. This could be referred to as "heat transfer-controlled" cooldown, and it might be approached in pipelines of small length-to-diameter ratios or in submerged bodies. The opposite extreme could be termed "flow-controlled" cooldown in which gas flow resistance is the important factor, and the resistance to heat flow is effectively zero. If axial heat conduction were neglected, the pipe temperature at a given point would drop instantly to the liquid temperature as soon as the liquid reached that point and the fluid temperature would rise instantly to that of the warm pipe. Progress of the temperature step along the pipe would be controlled by the rate at which the boil-off gas could be pushed out of the way.

The persistence of warm gas at the discharge of a long transfer line suggests that an analysis based on the "flow controlled' assumption, as in reference 1, is appropriate--at least for engineering design estimates. It is worth noting that the deviation of the assumed temperature step from some of the actual upstream temperatures shown in Fig. 1 may not have a serious effect on cooldown time computation. The reason is that for compressible flow with large length to diameter ratio the pressure drop per unit length increases rapidly toward the discharge end. Thus, upstream temperature does not bear as strongly on overall pressure drop as does the downstream temperature. The assumption

although a rather laborious, incremental computation, has produced good agreement with data of other experimenters [5, 6]. Chi's model [4] is probably best adapted to short transfer lines. A modification of Burke's solution has been used successfully [1, 2] for systems in which the flow is controlled by a restriction at the discharge. The work of Jacobs [9] is useful in estimating the total quantity of cryogenic liquid consumed during cooldown.

PROPOSED ANALYTICAL MODEL

The proposed flow model was developed largely from observation of the experimental cooldown data obtained from the system shown in Fig. 1 and from other reported observations [1, 6]. A complete description of that system is given in [7]. Briefly, the system, as shown in Fig. 1, consists of a supply dewar with a 200-ft long, 0.625-in. I.D., 0.75-in. O.D., vacuum-insulated, copper transfer line discharging to the atmosphere. Tests were also conducted with the line shortened to 150, 82, and 25 ft. The typical temperature histories shown in Fig. 2 indicate that the discharge temperature remains near the warm starting level for most of the cooldown period.

Two important factors which control the cooldown of pipes are the resistance to flow of vaporized liquid and resistance to the transfer of heat. The cooldown behavior of any pipeline might be expected to lie between two extreme cases in which one or the other of these controlling factors predominates. If flow resistance were unimportant compared to heat transfer resistance the entire pipe could be filled with

of adiabatic gas flow permits an integration of the time per unit of distance moved by the liquid vapor interface. This integral (in a generalized form) over the length of the pipe is taken as the cooldown time, and requires averaging only of friction factor. Further assumptions are as follows:

1. Temperature drop in the pipe is a step from ambient temperature down to the liquid saturation temperature. This temperature step, or cold front, advances down the pipe at a velocity, u_i.

2. Axial heat conduction is negligible.

3. A liquid-warm gas interface coincides with the position of the pipe cold front at all times so that the interface velocity is also u_i.

4. Heat transfer in the gas stream is zero; hence, the upstream gas temperature is the initial wall temperature.

5. The final temperature of the liquid in the pipe is the saturation temperature corresponding to the inlet driving pressure.

6. Heat transfer from the outside environment is negligible. This may not be appropriate in non-vacuum insulated systems.

7. The velocity changes are gradual enough that the process may be considered quasi-steady, that is, local acceleration terms in the momentum equation may be neglected. Thus, the flow and pressure surging that are known to exist are smoothed out.

8. Because of the relatively low velocity of the liquid stream as compred to the low density, high velocity, warm gas stream, all of the pressure drop is assumed to occur across the gas stream. Thus,

the upstream gas pressure is equal to the inlet pressure, $P_i = P_l$. Since the gas is warm, ideal gas relations are used. The ratio of specific heats was taken as 1.4 (for perfect diatomic gas) in the calculations presented. Other values are equally possible.

DETERMINATION OF THE INTERFACE VELOCITY

The velocity of the interface is the liquid inflow velocity minus the velocity at which the liquid front evaporates.

$$u_i = u_\ell - \frac{W}{\rho_\ell A_f} \tag{1}$$

and the upstream gas velocity is

$$u_{gi} = u_i + \frac{W}{\rho_{gi} A_f} \quad . \tag{2}$$

The rate of heat transfer from the wall is

$$q = u_i \rho_w A_w \Delta h_w \quad . \tag{3}$$

A simple heat balance equates the rate of heat transfer from the wall to the rate of enthalpy increase of the fluid. (See assumption 5.) Kinetic energy, viscous dissipation of energy, potential energy, etc. are neglected.

$$q = \rho_\ell A_f u_\ell \Delta h_\ell + W \Delta h_t \tag{4}$$

The following expression for the interface velocity has been obtained by eliminating q, u_ℓ, and W from the system of equations (1) through (4):

$$u_i = \frac{u_{gi}}{\beta} \tag{5}$$

where

$$\beta \equiv 1 + \frac{\rho_w A_w \Delta h_w - \rho_\ell A_f \Delta h_\ell}{\rho_{gi} A_f (\Delta h_\ell + \Delta h_t)} \tag{6}$$

According to the assumptions, for a given set of starting conditions β is constant and dependent only on the beginning and end conditions.

GAS FLOW

The gas flow is assumed adiabatic, constant area, and quasi-steady. If it is further assumed that the gas obeys the ideal gas equation of state, then the conditions which have been set up are those of a special case of one dimensional flow called "Fanno" flow. For Fanno flow many text books such as Hall [8], present formulae and tables which result from the solution of the combined continuity, energy, momentum, and state equations. Two of these formulae may be expressed in the following form:

$$\frac{P_i}{P_2} = \frac{M_2}{M_i} \sqrt{\frac{1 + \frac{k-1}{2} M_2^2}{1 + \frac{k-1}{2} M_i^2}} \tag{7}$$

and

$$\bar{f} \frac{(L - X_i)}{D} = \frac{M_2^2 - M_i^2}{k M_i^2 M_2^2} + \frac{k+1}{2k} \ell n \left[\frac{M_i^2}{M_2^2} \frac{1 + \frac{k-1}{2} M_2^2}{1 + \frac{k-1}{2} M_i^2} \right] \tag{8}$$

In these expressions the downstream Mach number, M_2, is not permitted to exceed 1. k is the ratio of specific heats, taken as 1.4

for perfect diatomic gas in this paper. $\bar{f}$ is treated as a known, constant value; its determination will be discussed in a later section.

COOLDOWN TIME

The cooldown time is defined as the time needed for the interface to travel the length of the pipe:

$$t = \int_0^L (1/u_i)\, dX_i \tag{9}$$

From the above equation the following dimensionless group may be obtained by substitution of u_i from (5) with u_{gi} replaced by $M_i u_s$.

$$t u_s/\beta L = \int_0^1 (1/M_i)\, d(X_i/L) \tag{10}$$

Since the upstream gas temperature is assumed always equal to T_0 the velocity of sound u_s is constant as well as β.

For each selected pair of the parameters P_i/P_a and $\bar{f}L/D$ the integration of (10) was approximated by summing 100 increments of $(1/M_i)\,\Delta(X_i/L)$. Equations (7) and (8) provided the necessary functional relationship between M_i and X_i/L. For small values of X_i/L (large $\bar{f}(L-X_i)/D$) and small P_i/P_a the discharge Mach number $M_2 < 1$. Under these conditions $P_i/P_2 = P_i/P_a$ and simultaneous solution of (7) and (8) yielded M_i. As X_i/L increased, M_2 increased eventually to unity. Beyond that point $P_2 > P_a$; however, it was only necessary to substitute $M_2 = 1$ into equation (8) to solve for M_i. Newton's method of iteration was used

for both modes of solution, M_i for each increment was the average over the increment. Fig. 5 shows the results of these computations carried out by digital computer over a range of $\bar{f}L/D$ and P_i/P_a. References [10] through [14] give the necessary hydrogen and nitrogen properties.

SELECTION OF $\bar{f}$

In Fig. 3, taken from [15], f for pipes is shown as a function of Re and ϵ/D, and M_i, Fig. 4, is a by-product of the computation of $tu_s/\beta L$. Friction factor may be obtained by two or three iterations starting with an assumed f, using Figs. 3 and 4.

Re and f are practically constant along the gas stream at any given time; however, as the interface progresses the gas length L_g shortens and the flow rate, Mach number, and Reynolds number increase markedly. For most practical cases Re is well into the turbulent range; therefore, the proportional change in f is much smaller than the change in flow rate.

As an example for the 200-foot pipeline with P_i = 75 psig and the roughness of drawn tubing one obtains from Figs. 3 and 4, f = 0.0158 at the beginning of the process (L_g = L). Then fL/D = 60.4. With nitrogen properties β = 73.7 and the cooldown time from Fig. 5 is 84.2 sec.

By the same procedure as above, M_i and f can be obtained for any position of the interface, that is for any value of L_g. In the following table the average f for each segment of pipe was used to get a cooldown time for that segment. (The time to cooldown the first 50 ft with $\bar{f}$ = 0.0158 is the time to cool 200 ft minus the time to cool 150 ft, etc.)

Interface located at	M_i	Segment	Re Avg. over segment	$\bar{f}$	Cooldown time per segment
0 ft	0.104				
		> 0 to 50 ft	2.45×10^5	0.0158	29.2 sec
50	0.120				
		> 50 to 100	2.87×10^5	0.0155	24.3
100	0.145				
		> 100 to 150	3.75×10^5	0.0150	19.2
150	0.202				
		> 150 to 200	1.30×10^6	0.0131	10.5
200	1.0				

t = summation of times per segment = 83.2 sec.

Comparison of the two examples shows that the effect of the decreasing f is slight even though the Reynolds number increased by a factor of five. Since longer times are spent in cooling the upstream segments, the f at the beginning of the process has the strongest influence in controlling the cooldown time. Therefore, it is recommended that $\bar{f}$ be based on $L_g = L$, as in the first example.

CALCULATED AND EXPERIMENTAL DATA COMPARED

Figs. 6, 7, and 8 show data points obtained from the experimental tests plotted along with the curves calculated by the method of Example 1. In all graphs, the driving pressure is the inlet tank pressure which was held constant during a run. Cooldown time was essentially the time from the first admittance of liquid into the pipe until the discharge measuring station registered liquid saturation temperature. Actually the stabilization of the inlet flow rate was used as the indication because it was a more sharply defined point. The designation "saturated liquid

point" means that the liquid in the supply reservoir warmed to the saturation temperature corresponding to its supply pressure before the cooldown began. For "subcooled liquid points" the supply liquid was close to its atmospheric boiling temperature.

The gas temperature rise is not truly a step function as supposed in the idealized model; therefore, toward the end of the process some cold gas is discharged. The result is that fluid enthalpy change, Δh_t is smaller than that of the model. As can be seen from equation (5) a reduction of Δh_t diminishes u_i. This lengthening effect on cooldown time is more pronounced in liquid hydrogen than in liquid nitrogen because of the greater importance of the gas sensible heat as compared to the heat of vaporization. For hydrogen the ratio of gas sensible heat to heat of vaporization is about 8, whereas, for nitrogen it is about 1. Comparison of figures 6 and 7 for the subcooled liquids in the 200 ft pipeline bears out this reasoning, since the calculated time for hydrogen is 10 to 30% below the test data.

The same effect of reduced gas temperature can be expected even for liquid nitrogen when the pipe length is shortened. In Fig. 8 the calculated cooldown times begin to appear low at L = 82 ft and are seriously low for L = 25 ft.

Subcooling of the inlet liquid had a slight effect for liquid hydrogen both experimentally and by calculation. However, subcooling

noticeably decreased cooldown time for nitrogen (Figs. 6 and 7), because the subcooled liquid produced less gas for a given heat absorbed from the wall. The ratio $\Delta h_\ell / \Delta h_t$ should be pertinent in gauging the effect of subcooling on cooldown time. For liquid hydrogen at 75 psig, for example, $\Delta h_\ell / \Delta h_t = 0.026$, whereas in nitrogen it is 0.112. Saturation produces two-phase flow throughout the process since even a slight pressure drop or heat addition produces vapor. This two-phase flow in turn produces further pressure drop, not accounted for in the model; thus, as evident in Fig. 7, the prediction for subcooled liquid is more accurate than for saturated liquid.

CONCLUSIONS

1. Experiments indicate that, for pipelines of large length-to-diameter ratio, the cooldown process approaches that of the proposed "flow-controlled" model.

2. The use of the generalized cooldown time parameters graphed in Fig. 5 provides an easy, effective way of estimating cooldown time of long, well insulated transfer lines. For complicated systems, shorter lengths, systems with appreciable heat leak, constrictions, or concentrated masses of metal, a more comprehensive model would be needed.

3. Users of this method should be aware of the limitations brought about by the simplicity of the model in the form presented. A minimum applicable length-to-diameter ratio is probably the most serious limitation. Liquid nitrogen L/D apparently is not severly restricted since

the calculated times were not objectionably low until L/D = 475 (L = 25 ft.). Indications are that liquid hydrogen L/D may be more restricted since the predictions are 10 to 30% low for L/D of 3800 corresponding to the 200 ft pipeline. Further experiments are required to determine the lower limit of L/D for hydrogen.

NOTATION

A_f = flow cross sectional area

A_w = solid cross sectional area of pipe wall

D = pipe inside diameter

f = Moody friction factor ($\bar{f}$ = average value)

G = mass flow rate per unit area

h_ℓ = enthalpy of liquid at inlet temperature

$h_{\ell\ sat}$ = enthalpy of liquid saturated at the inlet pressure

h_g = enthalpy of the warm gas

Δh_ℓ = liquid subcooling enthalpy = $h_{\ell\ sat} - h_\ell$

Δh_t = $h_g - h_{\ell\ sat}$

Δh_w = enthalpy drop of the pipe material in cooling from T_0 to T_{sat}

k = ratio of specific heats for perfect diatomic gas, 1.4

L = total length of the pipeline

L_g = length of the gas stream = $L - X_i$

M_2 = downstream gas Mach number (discharge end)

M_i = upstream gas Mach number (at the interface) = u_{gi}/a_s

P_a = ambient pressure

P_2 = downstream gas pressure ($P_2 \geq P_a$ when $M_2 = 1$)

$P_i = P_1$ = upstream, or interface, gas pressure--taken as equal to the inlet pressure P_1

q = rate of heat transfer into the fluid or out of the wall

Re = GD/μ_g, Reynolds number of the gas stream

t = cooldown time

T_ℓ = inlet liquid temperature

T_0 = initial warm pipe temperature

T_{sat} = saturation temperature corresponding to the inlet pressure

u_i = liquid-gas interface velocity

u_ℓ = liquid inflow velocity

u_{gi} = upstream, or interface, gas velocity

u_s = velocity of sound in the warm gas stream

W = mass rate of evaporation

X_i = location of the upstream end of the gas stream, or liquid-vapor interface. $X_i = 0$ at the inlet end.

GREEK

β = enthalpy change parameter defined by equation (6)

ϵ = absolute pipe roughness

μ_g = viscosity of the warm gas stream

ρ_{gi} = density of the warm gas stream at the upstream pressure

ρ_ℓ = density of the liquid

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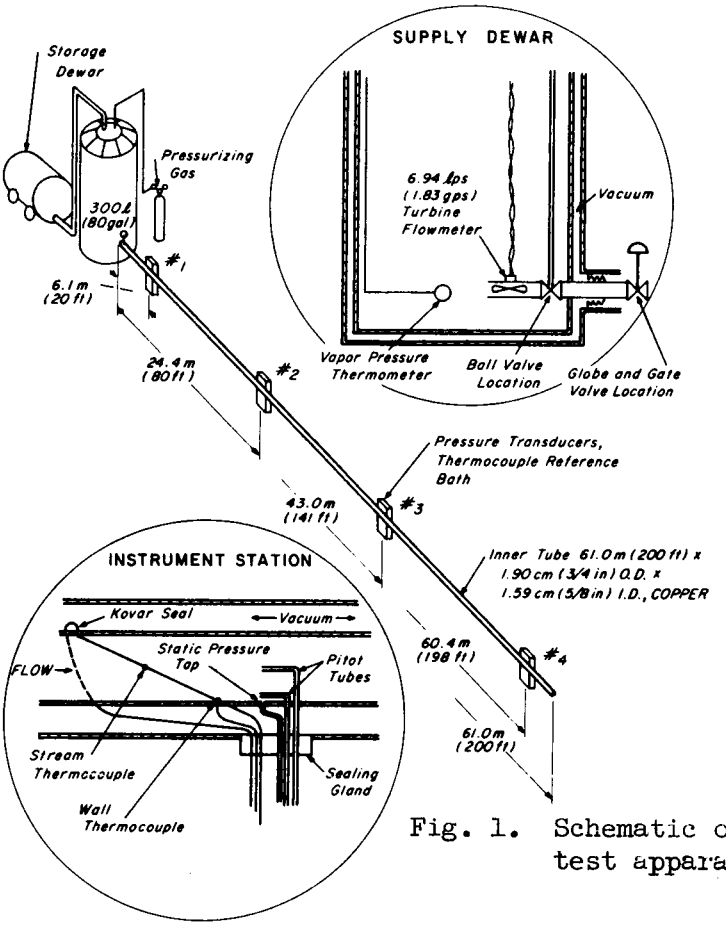

Fig. 1. Schematic of test apparatus.

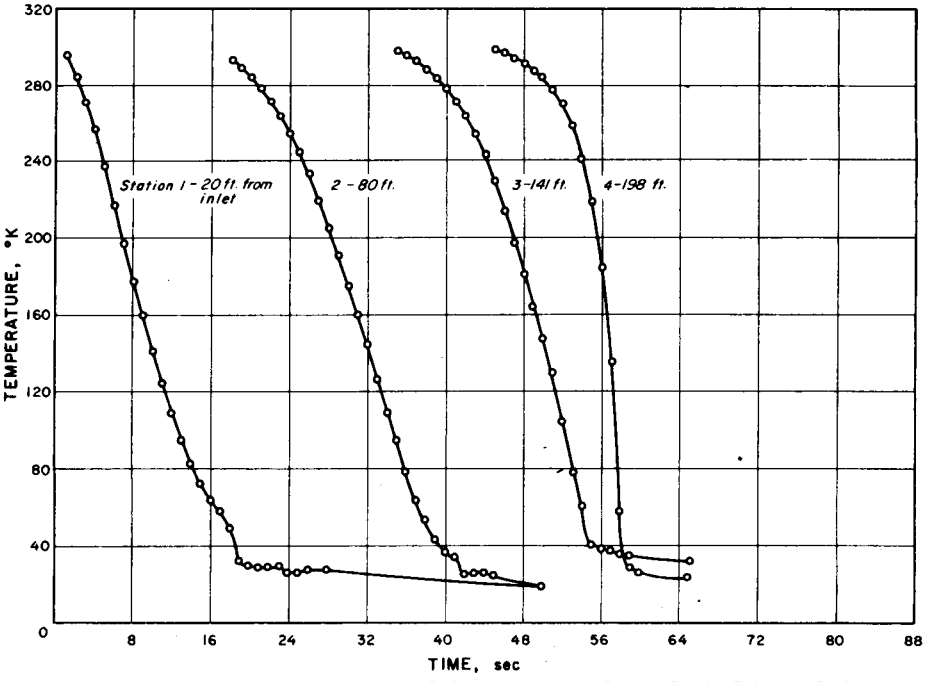

Fig. 2. Typical line temperature history--subcooled liquid hydrogen, driving pressure of 75 psig.

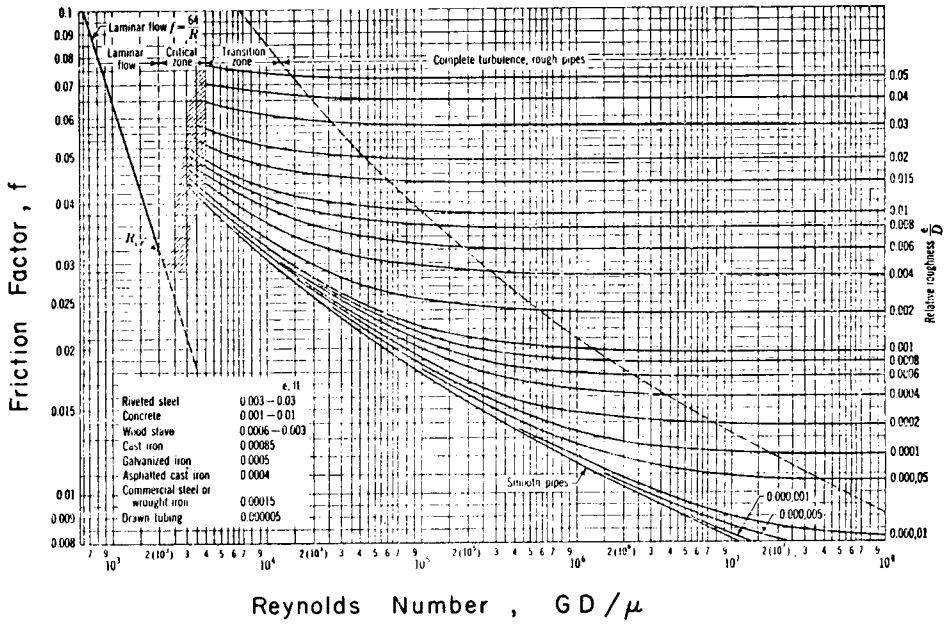

Fig. 3. Friction factor and absolute roughness for circular pipes From Moody, Ref. 15.

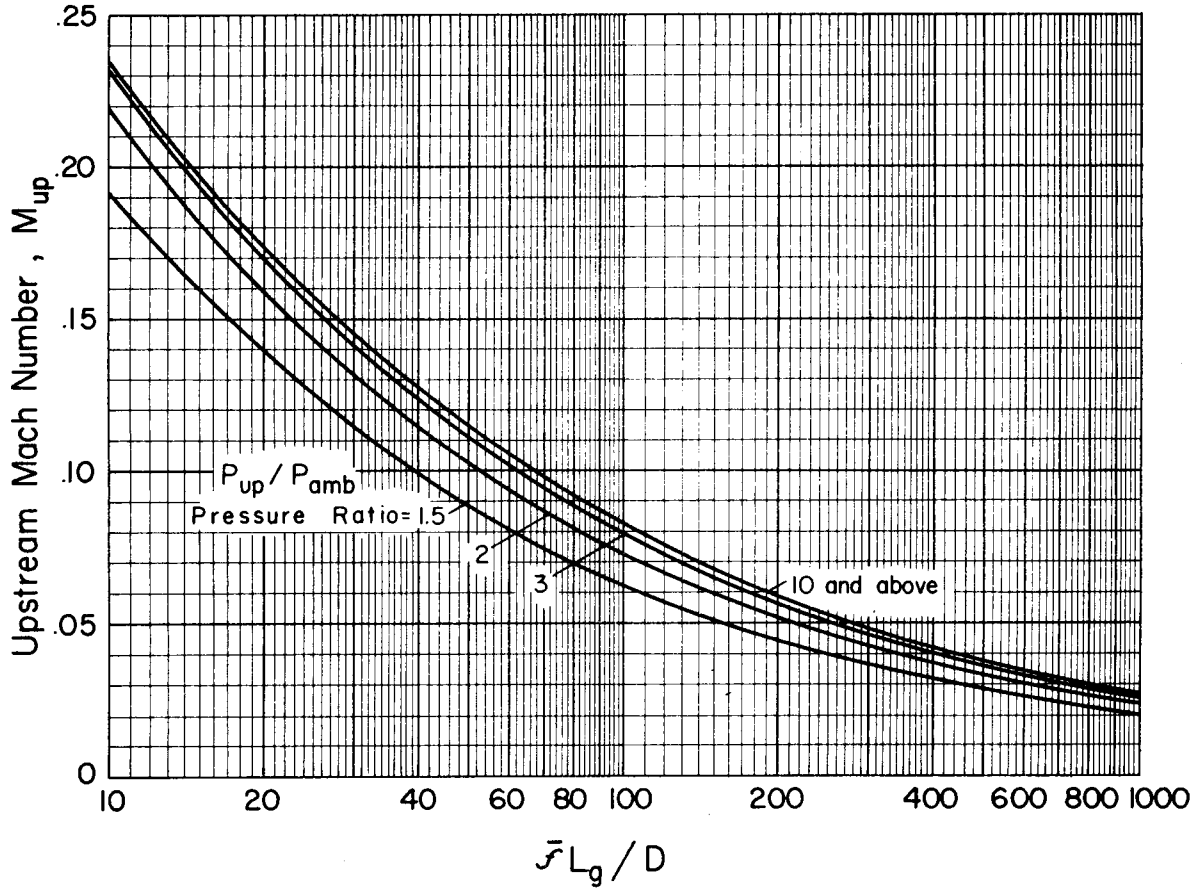

Fig. 4. Upstream gas Mach number for Fanno flow.

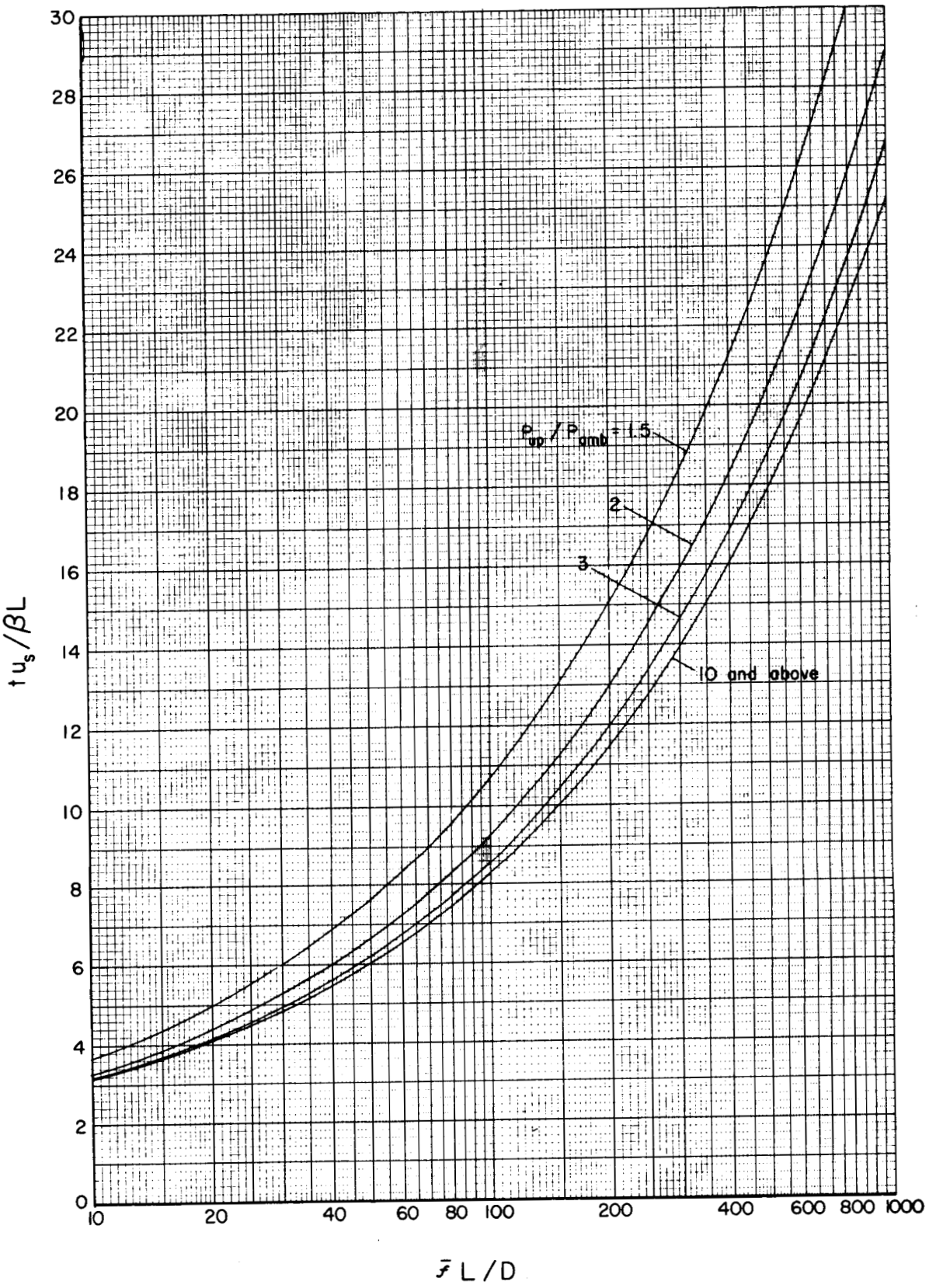

Fig. 5. Cooldown time parameters.

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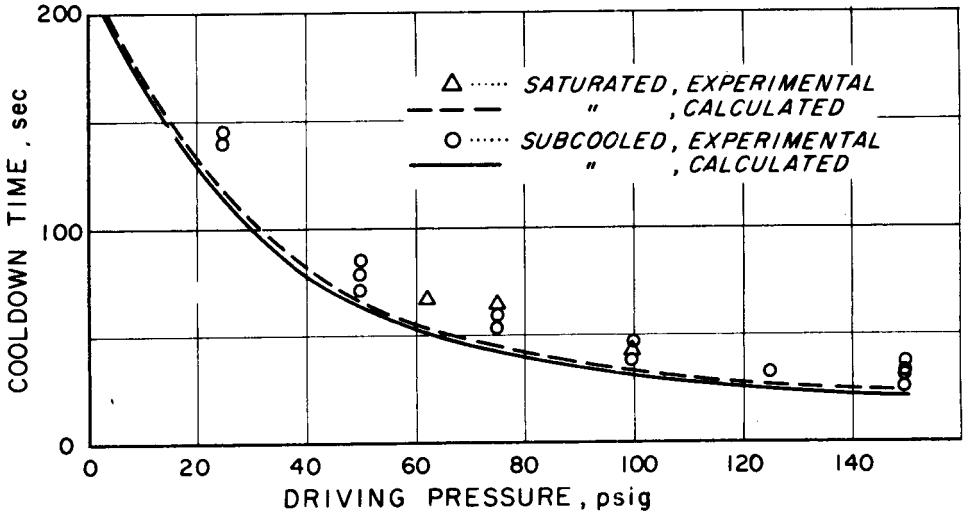

Fig. 6. Subcooled and saturated liquid hydrogen cooldown time--200 ft. pipeline.

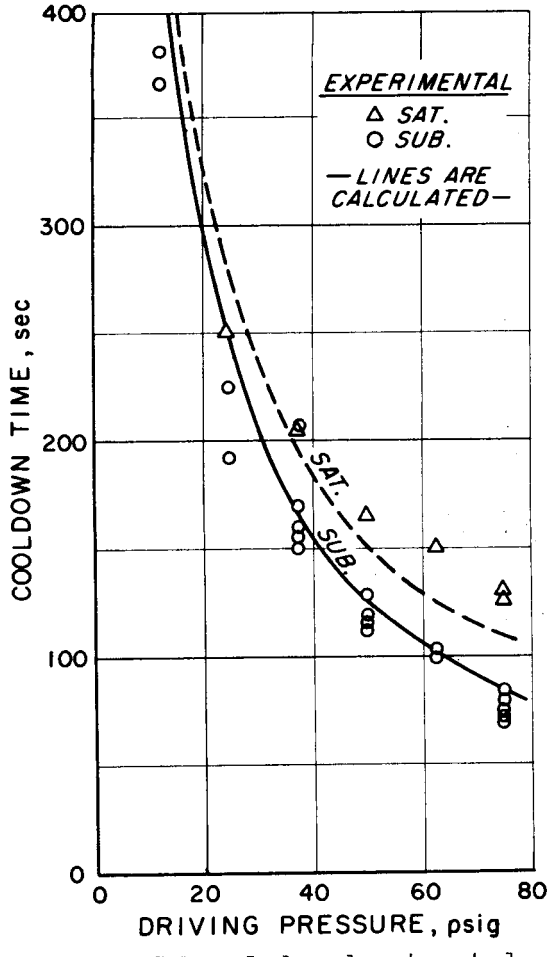

Fig. 7. Subcooled and saturated liquid nitrogen cooldown time--200 ft. pipeline.

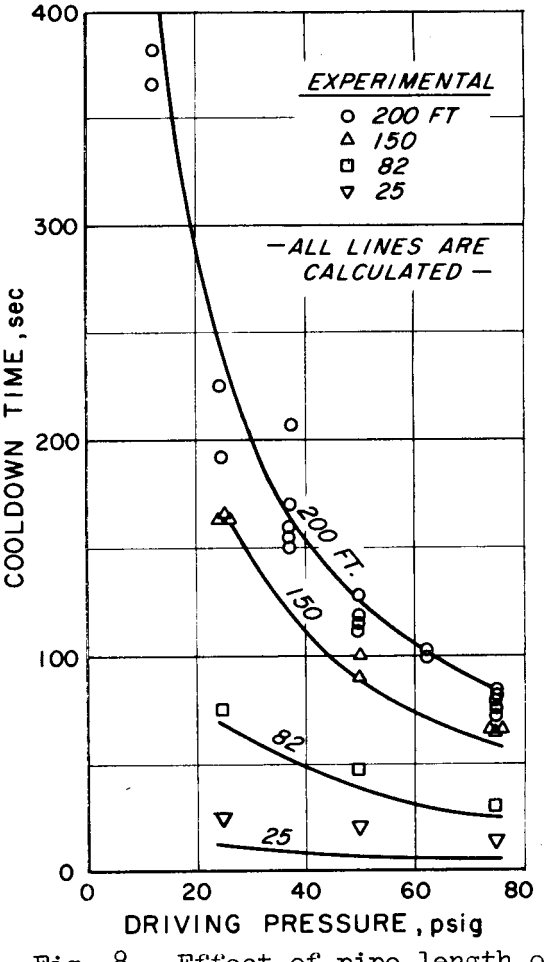

Fig. 8. Effect of pipe length on subcooled liquid nitrogen cooldown time.