THE UNIVERSITY OF MICHIGAN
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EVALUATION OF DESIGN PARAMETERS, INCLUDING CAVITATION-EROSION EFFECTS, FOR LIQUID METAL PUMPS APPLYING TO NUCLEAR POWERPLANTS

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PREFACE

This report is presented as a preliminary result of an investigation into the problems of large-scale mechanical liquid metal pumps, which are utilized as fuel and coolant circulating pumps in present and projected nuclear powerplants.

The work upon which this report is based was undertaken at the request of Dr. R. G. Folsom, Director, Engineering Research Institute, University of Michigan.

ACKNOWLEDGEMENTS

Grateful acknowledgement is extended to Professor H. A. Ohlgren for his kind assistance and suggestions in the preparation of this report. The author also wishes to thank Raymond Knight and Doris Thompson for their design and drafting assistance, and Mrs. Diane Etzel and Mrs. Jean Sinclair for their secretarial efforts.

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1.0 OBJECTIVES

The investigations reported here were undertaken for the purpose of examining those problems involved in the design and development of large-scale liquid metal circulating pumps for application in nuclear powerplants. The investigation was to be particularly concerned with the fluid dynamic problems, the most serious of which appear to be cavitation and erosion damage to the structural components at the high relative velocities obtained in the pump impeller.

2.0 ABSTRACT

A large number of the presently proposed and constructed nuclear powerplants utilize liquid metals as a coolant and as a solvent carrier for the fissioning and breeder blanket materials. Due to their excellent heat transfer, vapor pressure, and nuclear properties, it appears that liquid metals will become increasingly important in future nuclear developments as operating temperatures increase with improving reactor technology.

At the present time, the state of the art of liquid metal pumping is largely undeveloped with respect to full-scale, high temperature powerplant units. Along with seal and bearing difficulties, there are the unknowns of fluid dynamic performance. Perhaps the most serious of these deficiencies in the present knowledge relates to cavitation performance and effects, and high velocity erosion effects, which may be encountered in a full-scale unit. The pumping conditions demanded for the nuclear powerplant application typically involve high flows (up to and in excess of 10,000 gpm) and relatively low submergence, so that pump design speed is limited by the present requirement for conservative cavitation design. This results in excessive pump cost, size, and weight. Approximate curves allowing the evaluation of these trends are included in the report.

3.0 INTRODUCTION

Many of the presently proposed and projected nuclear power-plants utilize a liquid metal as either a primary or secondary coolant or perhaps as a fuel carrier as in the homogeneous reactor. In all of these cases, one or more liquid metal circulating pumps are required.

The liquid metals may be of several sorts. The most prominent at the present time are, on the one hand, sodium, potassium, and their alloys, and on the other, bismuth and its alloys. Temperatures under consideration are high and some cases of the order of 1500°F. Quantities to be pumped depend on the fluid, the power range, and the allowable temperature differentials. Pump flow rates above 10,000 gpm have been planned, and no doubt even higher flows will be required in the future.

Pump inlet pressures are restricted by the added complexities of the pump design for increased pressure and by the penalties throughout the entire system caused by an increase. This latter is important because of the high temperatures, large pipes, and expensive materials. It is also important where weight is a factor.

Considering the foregoing, it is obvious that in many cases the restricting factor in the pump design will be cavitation* and/or high impeller velocity. The former will necessitate pumps of unduly large size, cost, and weight, and the latter may necessitate a different type of design, such as the use of multi-stage units. Due to

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* Even though vapor pressures are low, the combination of flow and pump inlet pressure are such that fluid pressure less than the vapor pressure (and even negative) would be attained if pump speeds, desirable from the mechanical design viewpoint were used. There are at the present time several incidences of liquid metal cavitation.

the special requirements of the application, even under ideal conditions the pump is an extremely expensive and weighty item.

Over the years, empirical limits have been established for water pumps with respect to cavitation performance loss and damage effects, and also with respect to fluid velocity. However, considering the differences between basic fluid properties, chemical properties, container material strength, temperature, etc. there is no reason to believe that these limits will apply to the liquid metal pumps. Even the direction of the change is not certain in all cases. Consequently, it seems imperative to examine these phenomena in the actual medium under consideration, and to run carefully controlled, definitive experiments. While considerable effort has been expended to date on liquid metal flow tests in the range of pipe-line velocities, very little has been done in the range of the necessarily much higher velocities to be encountered in pump impellers, and nothing has been done with respect to cavitation.

4.0 LIQUID METAL PUMP APPLICATIONS

In any power reactor, it is necessary that the heat produced by nuclear fission in the reactor core be removed to a thermodynamic working fluid so that it may produce mechanical or electrical power in a more or less conventional heat engine. Perhaps the most suitable media for the transfer of this heat from the reactor (and also as a solvent carrier for the fissioning and blanket materials in a homogeneous reactor) are liquid metals. This will become increasingly the case as the temperature levels in the reactor are raised (as reactor technological development proceeds). The great desirability of liquid metals in this application is attributed to the following factors:

a) excellent heat transfer properties both with respect to film resistance and favorable volume heat capacity (allowing very low pumping power).

b) low vapor pressure at very high temperature minimizing reactor core container problems.

c) in most cases, nuclear properties suitable for the use of the liquid metal in the reactor core.

One of the problems at the present time associated with the use of the liquid metals is the relatively undeveloped state of the art of handling and pumping these fluids. With respect to pumps, this raises problems of sealing, bearing arrangement, mechanical construction with high temperature gradients, and other problems related to the physical construction. It also raises fluid dynamic problems with respect to cavitation, erosion, and pump flow-path design in general. The presently available meager experience data seems to indicate that the parameters of head, flow, and efficiency scale fairly well in the conventional way. However, no information is available from long-term tests under the high velocity, high temperature, and perhaps cavitating conditions which would be encountered in a full-scale pump for a large plant. To further complicate the matter there is the question of radioactivity effects being added to the conventional erosion and cavitation effects. The cavitation phenomenon itself might be complicated in this respect by the introduction into the fluid of fission product gases, as perhaps polonium. This situation represents a serious deficiency in the developmental programs presently underway.

At the present time liquid metal systems are projected or in use in

a) a liquid metal homogeneous reactor system where the fissioning and the breeder blanket materials are suspended as solution (or slurry) in bismuth. This system may also include secondary circuits to carry heat, exchanging between the UBi circuit and a power circuit of either water-steam or a gas. A general schematic diagram of such a system is shown in Figure 1. Liquid metal flow rates for a fairly large plant are approximately 30,000 gpm.

b) fast plutonium breeder reactors under development at Arco, Idaho and projected in large scale by the Detroit Edison Company for erection in the Detroit area. These systems utilize two liquid metal circuits: 1) the primary sodium loop heat exchanges directly with the fuel rods in the core, while 2) a secondary NaK circuit transfers heat from the sodium to steam. Flow rates in this plant are of the order of 30,000 gpm and pressure rises of 150 psi are contemplated. A simplified schematic diagram of such a system is shown in Figure 2.

c) a sodium graphite reactor as designed and constructed by North American Aviation. Here sodium is used as a primary coolant removing heat from a matrix of graphite and uranium. If this were to be designed into a complete powerplant there would also be the possibility of a secondary sodium or NaK loop.

While a few typical examples of reactor applications utilizing liquid metals have been cited, there are numerous other potential applications. As reactor temperatures increase, in those applications where high efficiency and light weight are of importance, there is the strong possibility of utilizing a liquid metal as the thermodynamic working fluid, perhaps in a binary cycle with water. Possible liquid metals for such an application would be mercury, sodium, potassium, rubidium, and possibly others. Thermodynamic investigations along these lines have been and are presently being carried through.

5.0 PRESENT LIMITATIONS ON LIQUID METAL PUMP DESIGN

5.1 General Pump Types

Considering the large flow rates which are usually involved, in general only the centrifugal pump, of the various conventional pump types, is of importance. However, since liquid metals are usually good electrical conductors, there is also the possibility of an electromagnetic pump. Compared to the centrifugal pump, the electromagnetic design offers the great advantage of a completely welded system. However, with respect to weight, cost, and efficiency, it is in general very much inferior. Because of the excessive weight of the electromagnetic pump and its associated components, it is indeed prohibitive in designs where weight is at a premium. Considering the variation between the various fluids to be pumped, sodium and potassium are far more suitable for an electromagnetic pump than are bismuth and its alloys because of the difference in conductivity.

5.2 Pump Requirements

As was mentioned in Sections 2.0 and 3.0, liquid metal flow rates in excess of at least 10,000 gpm will be required. Generally, it is not desired to increase pump suction pressure above a rather small value which depends on factors involving the overall system. A typical specification for example would allow 10 feet in excess of atmosphere (or about 50 feet) for a 12,000 gpm sodium pump. (The net positive suction head in this case would be somewhat less because of the vapor pressure of sodium.) It will be noted that a similar arrangement for bismuth would give a total suction pressure of only about 14 feet because of the difference in densities.

Pump head rise requirements cover the full range from a few feet to several hundred feet This quantity is closely connected with the overall design, and is difficult to discuss in general. It is usually a compromise value between the high velocities desired for high heat transfer on one hand, and the pump cost, complexity, and power requirements on the other. Because of the greater power required to pump a given volume flow rate of bismuth through a given head drop, the design head requirements for a bismuth-cooled system would tend to be smaller than for a sodium system. Heat capacity per unit volume is about the same for sodium and bismuth. The required head for the typical example previously cited (12,000 gpm sodium system) is about 400 feet. Presumably if bismuth were the coolant, a lower figure would have been selected since otherwise the power requirements would be excessive.

5.3 Centrifugal Pump Design Limitations - Cavitation and High Fluid Velocity Erosion

As is well known, the weight and cost of a centrifugal pump and driver installation can be most effectively reduced by increasing the operating speed. In general, this effect is greatest in the low specific speed range where large reductions in pump diameter are attained through speed increase. Thus the importance of increasing pump speed depends on the relation between flow rate and required head rise. This is illustrated in Figures 3, 4, 5, and 6* where the impeller diameter is plotted against pump speed for a range of required heads. Each curve is for a given flow rate--1000, 5000, 10,000 and 100,000 gpm

* The assumptions and sample calculations for these curves are shown in the Appendix.

were considered. Noting the 10,000 gpm case for example (Figure 5) if the required head were 300 feet, increasing speed up to about 5000 rpm would still result in a substantial reduction in diameter. However, if the required head were only 10 feet, most of the benefit of speed increase would be achieved at 1000 rpm. In the 1000 gpm curve (Figure 3), a substantial diameter reduction with speed up to 10,000 rpm is shown. These curves are, of course, approximately applicable to single stage* centrifugal and axial flow pumps in general, and do not depend upon the particular fluid.

To illustrate the approximate variation of pump-driver weight with flow rate, head rise, and pump speed for a particular type of design for a particular fluid, the curves shown in Figures 7, 8, 9, and 10 are included. These illustrate only approximate trends and are not intended as accurate data. Each curve applies to a particular flow rate: 1000, 5000, 10,000 and 100,000 gpm were considered. The weight values are based approximately on a typical bismuth circulating pump design with vertical long-overhang shaft, inert gas blanket, mechanical seal, electric motor drive, with which the writer is familiar.** Since for similar types of machines in a somewhat restricted size range, cost and weight are more or less proportional, the vertical axis is called a cost and weight index. Although these curves are meant to apply only to a specific type of design, they show trends which are more or less applicable for any type of centrifugal or axial flow pump design. It will be noted that up to a certain point, i.e.,

* Single stage pumps only are considered for the present because of the added mechanical complexities (overhung shaft, difficult bearing design, seals, etc.) of multistaging with high temperature liquid metals.

** The detailed assumptions, sample calculations, and a sketch are given in the Appendix.

for low specific speed units, the pump cost and weight for a given flow and head rise are approximately proportional to the square of the speed, so that doubling the speed would achieve a four-fold saving.

In most actual liquid-metal circulating-pump designs for nuclear powerplants, the pump speed is limited by the necessity of avoiding any possibility of cavitation. However, there is also the possibility of destructive erosion due to the high fluid velocity in the impeller. In general, for pumps in the range here considered, the relative velocity in the inlet section is greater than that at discharge. This value, as affected by the cavitation parameter S, is shown in Figure 12, which is explained in detail later in this report. However, there is still the necessity of achieving the discharge velocity to achieve the required head in a single stage unit, and this value is only slightly affected by pump speed. Due to the lack of knowledge regarding the cavitation phenomenon in the various liquid metals with respect to both damage and performance, it is at present necessary to design with extreme conservatism in this respect. Thus the cost, size, and weight of the proposed designs may be unnecessarily great; also the designs may be unduly complicated mechanically by the necessity for added shaft overhang, etc.

A commonly employed measure of the likelihood of cavitation in a given design is the suction specific speed S, which is defined below:

$$S = \frac{N\sqrt{gpm}}{(NPSH)^{3/4}}$$

N = rpm
NPSH = net positive suction head - ft. of fluid

It has been shown,* considering the uncertainties of the actual flow conditions in a centrifugal rotor, and the likely obtained pressure coefficients with present-type blades, that it is most probable that the vapor pressure is reached at some points within the impeller with S values above about 8000. Thus, for conservative design in an unknown medium, an S of about 6000 should be selected. However, there are many examples of pumps operating at S values of the order of 12,000 in water and other fluids with no discernible cavitation effects. Thus, even though water pumps can in many cases attain this latter value, and operate without damage even under fairly high velocity conditions, there is no basis for the assumption that the same value is feasible in an elevated temperature liquid metal system (or, on the other hand, for the assumption that it cannot be exceeded in some cases). Also, there is no assurance that the fluid velocities necessary to attain the desired heads in single-stage unit will not result in prohibitive metal erosion. For fluids other than water it is necessary to consider not only the effect of density (up to 10 for bismuth), high temperature which weakens the structural materials, but also various chemical effects. On the other hand, in some cases, the tension bearing capability of the fluid may be such as to prevent cavitation even if negative pressures are reached.

To show the effect of the S value on pump design, Figure 11 is included. For three typical NPSH values, and for three typical S values under each of these, the allowable pump speed is shown as a function of flow rate. It was assumed that an S of 6000 represents

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* G. F. Wislicenus, "Critical Considerations on Cavitation Limits of Centrifugal and Axial-Flow Pumps," ASME Paper No. 55-A-144.

present day conservative design practice for liquid metal pumps. It is believed that this statement would approximately represent the opinion of the major pump manufacturers. However, it is felt that an S of 12,000 or even more might become feasible in certain liquid metal pump applications if further basic knowledge on cavitation in liquid metal were available. This value then was used for the upper limit on the curves.

It will be noted that if the previously mentioned example of a 12,000 gpm, 400 foot head, 50 foot suppression head is considered, the speed based on the conservative S value is about 1030 rpm, and the cost and weight index about 1600.* For S of 12,000, the speed is 2060 rpm and the cost and weight index is about 400, resulting in a four-fold saving. As another example a 10,000 gpm bismuth pump might be considered with a head rise of 20 feet and a net positive suppression head of 15 feet (about 10 feet over one atmosphere). This is approximately typical for one type of application. Then for S of 6000, the pump speed is about 450 rpm and the cost and weight index is 26; for S of 12,000 the speed is 900 rpm and the cost and weight index is 15. Here the saving is not as great as in the first case although it is still substantial. It will be noted that for lower flow rates, if the heads are as given, the proportional saving becomes greater, approaching the ratio given for the case of the sodium pump. The cost and weight indices for the two cases should not be directly compared since the ten-fold density difference between sodium and bismuth would considerably increase the cost and weight of the bismuth

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* Impeller diameter is about 37 inches. The entire pump casing diameter would be about six feet.

pump with respect to the sodium pump. Also there may be differences in temperatures, applicable materials, sealing arrangements, etc.

It thus appears that from the viewpoint of economics there is considerable justification for a liquid metal cavitation study program, since the cost of a single pump-driver unit may be several hundred thousand dollars. From the viewpoint of those applications where weight saving is important, as for example in aircraft or other forms of transportation devices, such a program appears to the writer to be absolutely vital. Also, it may be necessary to conduct high velocity tests, even when cavitation is absent, to demonstrate the feasibility of single-stage centrifugal pumps producing the desired heads, since relative fluid velocities in the impeller are of the order of five to ten times the conventional pipe-line velocities (Figure 12).

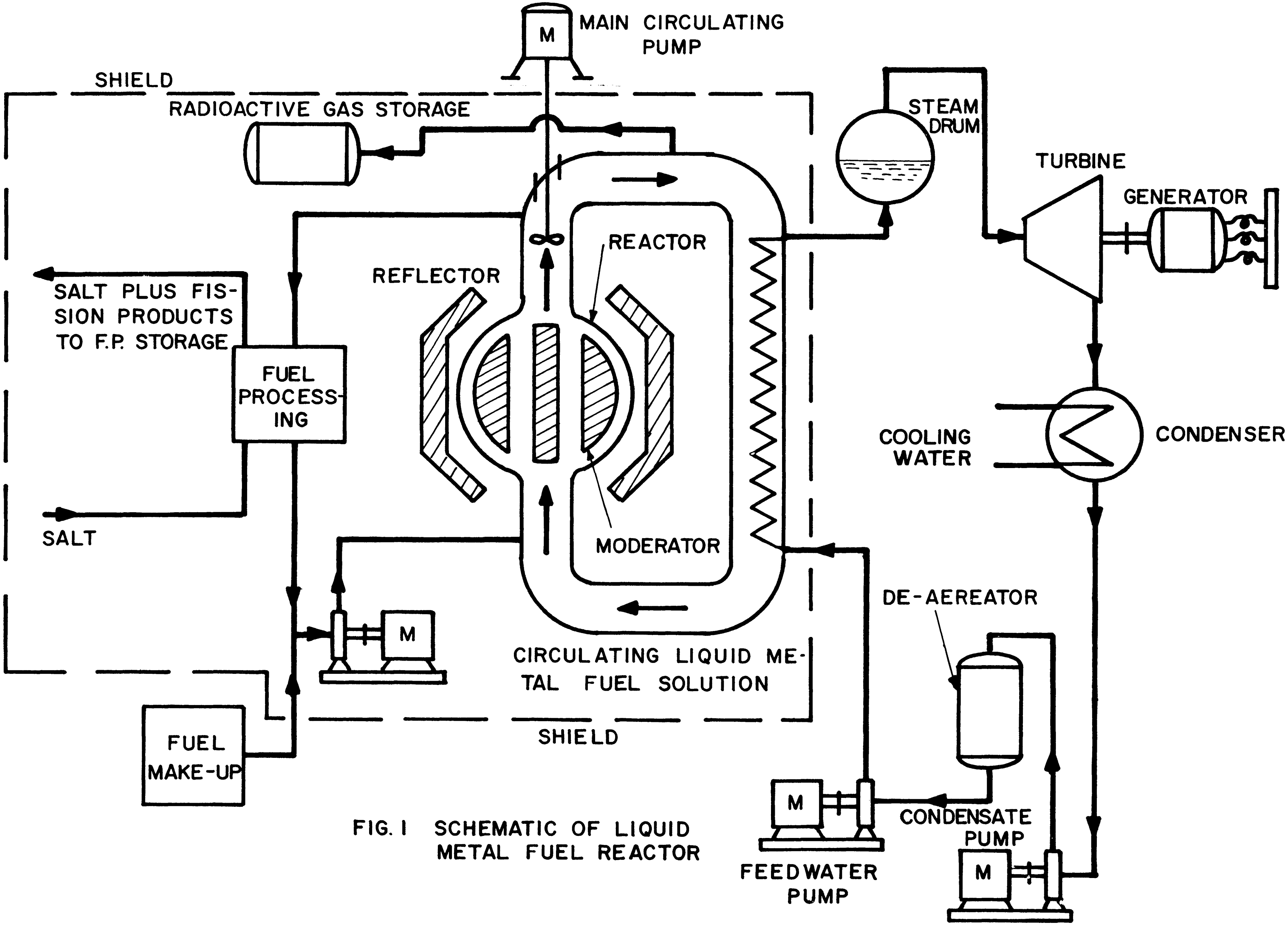
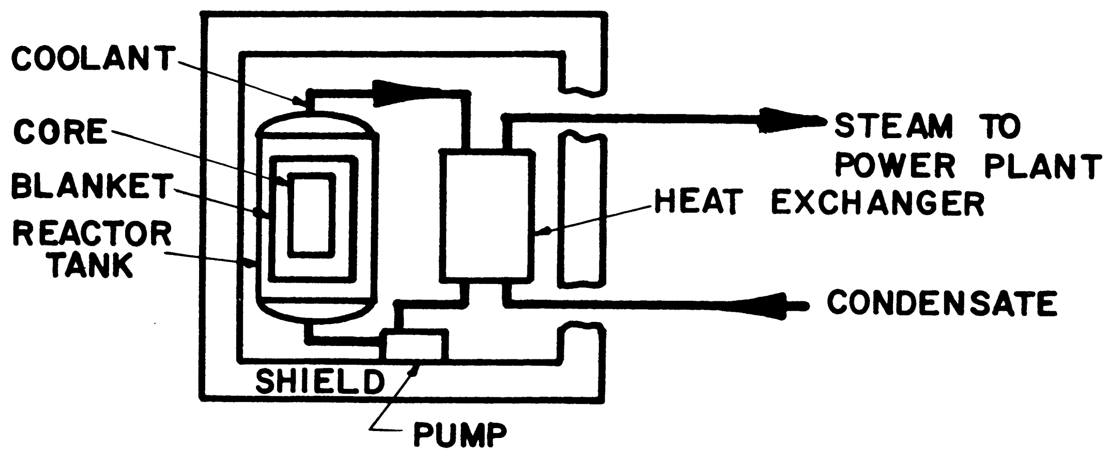


FIG. I SCHEMATIC OF LIQUID METAL FUEL REACTOR

FIG. 2

HETEROGENEOUS FAST BREEDER OR
CONVERTER REACTOR

COOLANT – LIQUID Na
PRESSURE – ATMOSPHERIC
PRODUCTS – POWER, RADIOCHEMICALS, FISSIONABLE MATERIAL



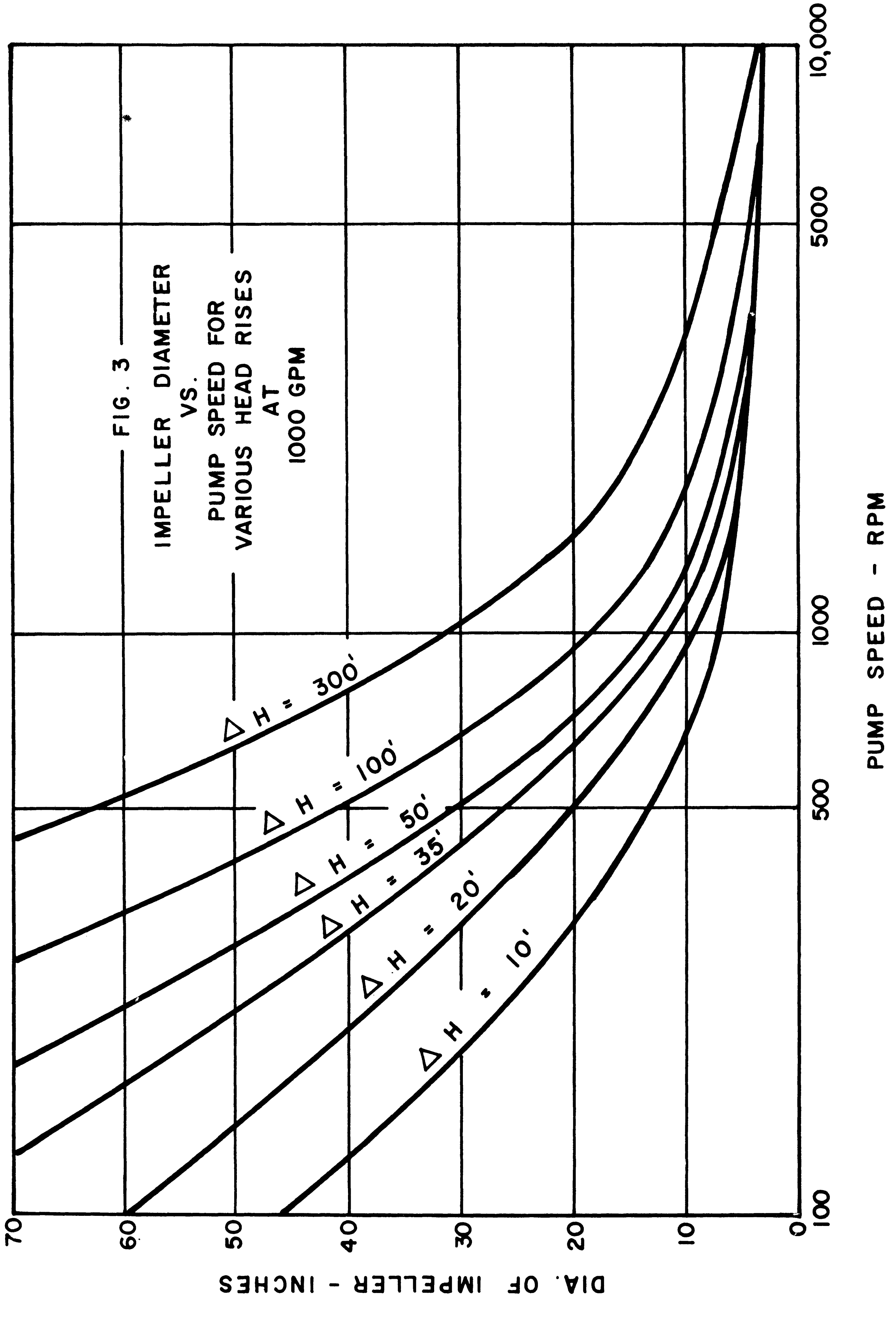


FIG. 3
IMPELLER DIAMETER VS. PUMP SPEED FOR VARIOUS HEAD RISES AT 1000 GPM

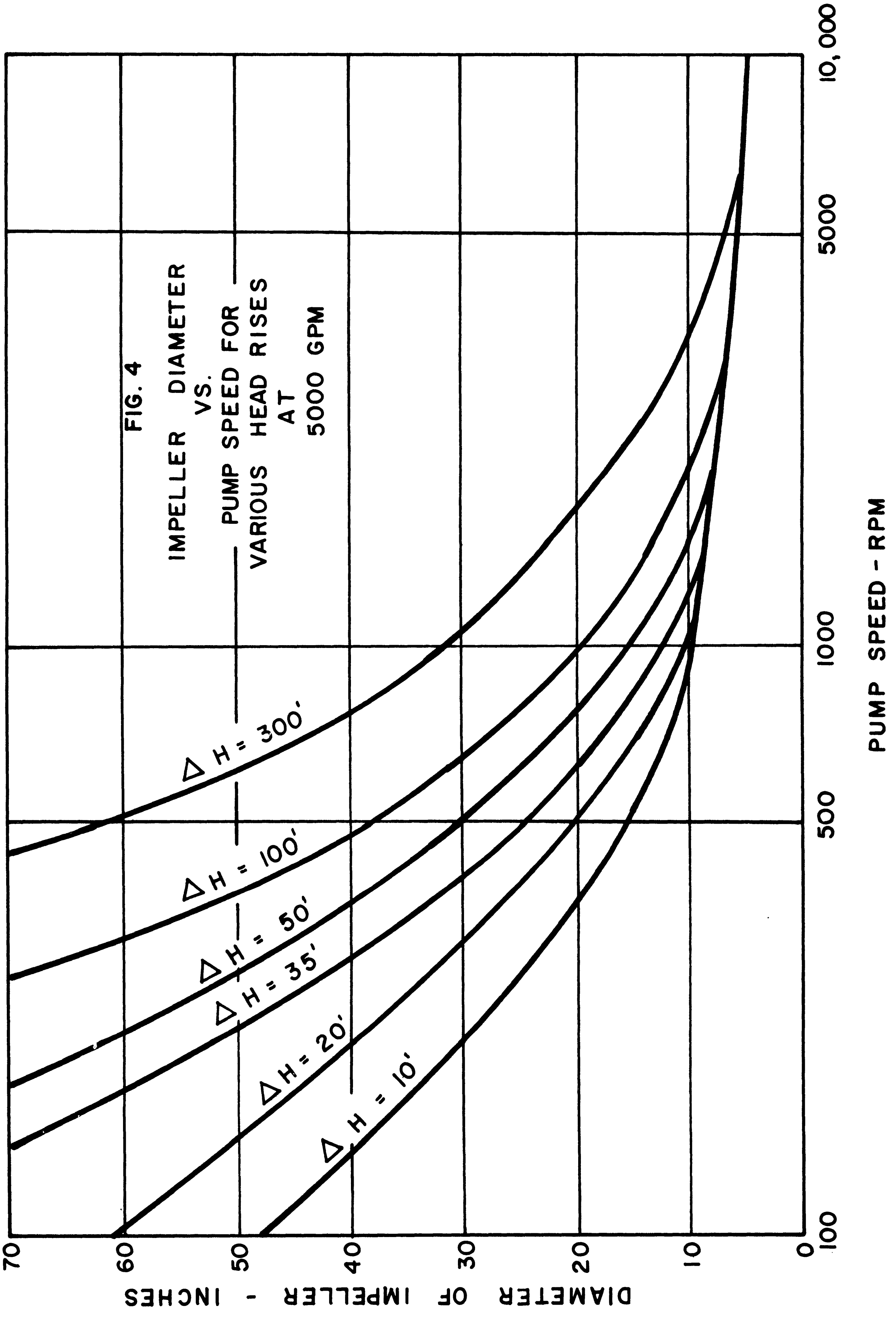


FIG. 4
IMPELLER DIAMETER
VS.
PUMP SPEED FOR
VARIOUS HEAD RISES
AT
5000 GPM
ΔH = 300'
ΔH = 100'
ΔH = 50'
ΔH = 35'
ΔH = 20'
ΔH = 10'
PUMP SPEED - RPM
100
500
1000
5000
10,000
DIAMETER OF IMPELLER - INCHES
0
10
20
30
40
50
60
70

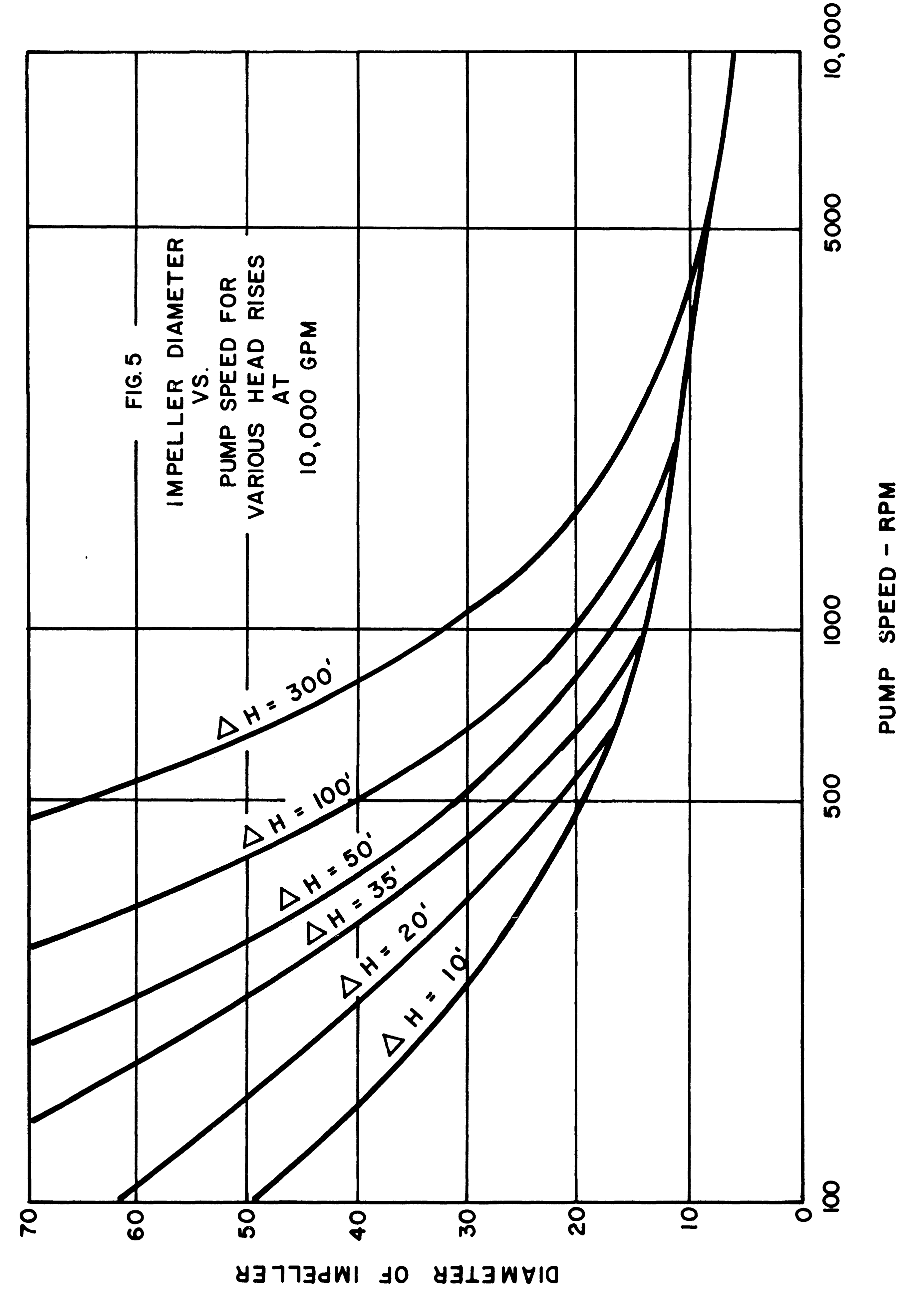


FIG. 5
IMPELLER DIAMETER
VS.
PUMP SPEED FOR
VARIOUS HEAD RISES
AT
10,000 GPM
ΔH = 300'
ΔH = 100'
ΔH = 50'
ΔH = 35'
ΔH = 20'
ΔH = 10'
PUMP SPEED - RPM
100
500
1000
5000
10,000
DIAMETER OF IMPELLER
0
10
20
30
40
50
60
70

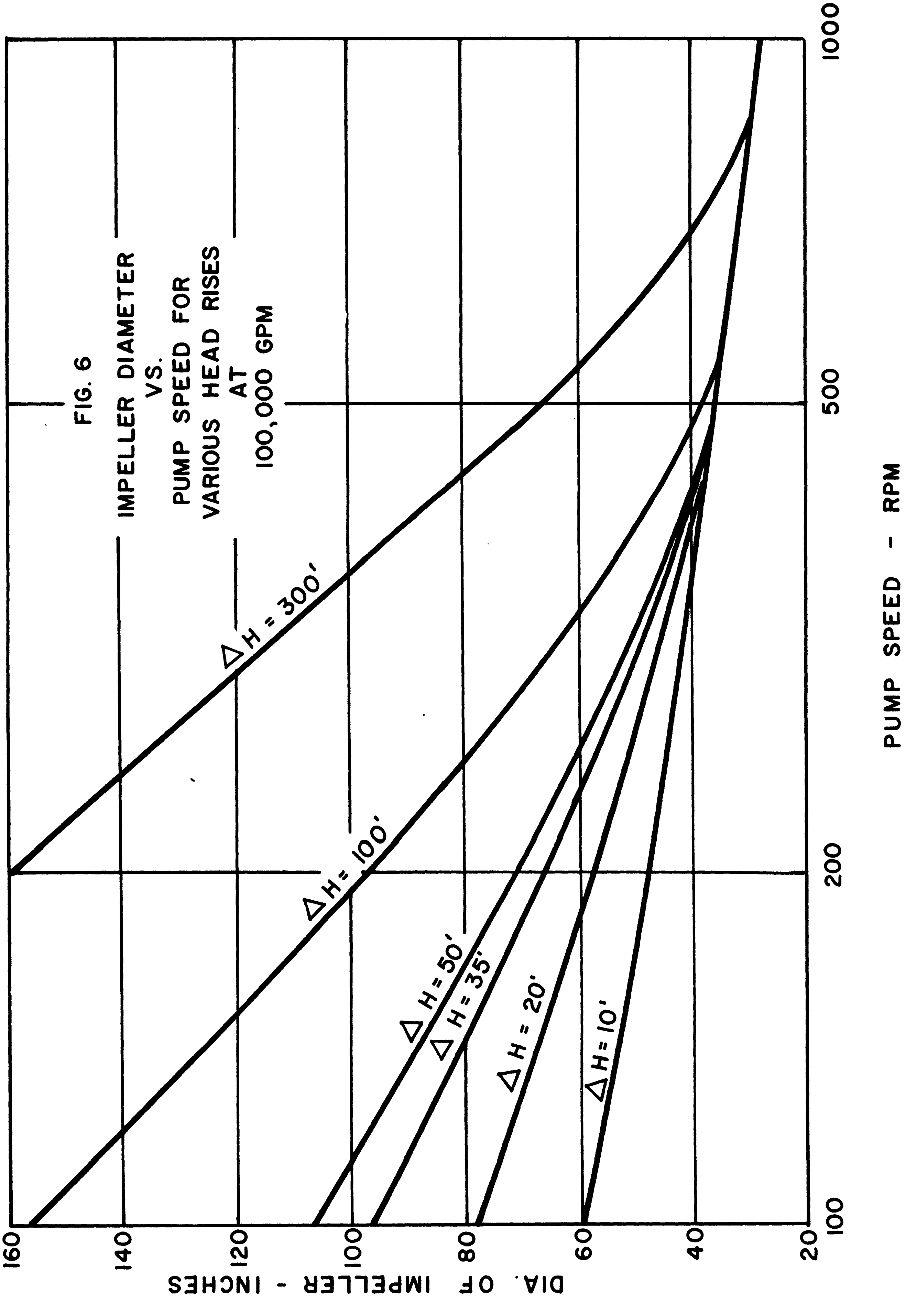


FIG. 6
IMPELLER DIAMETER VS. PUMP SPEED FOR VARIOUS HEAD RISES AT 100,000 GPM

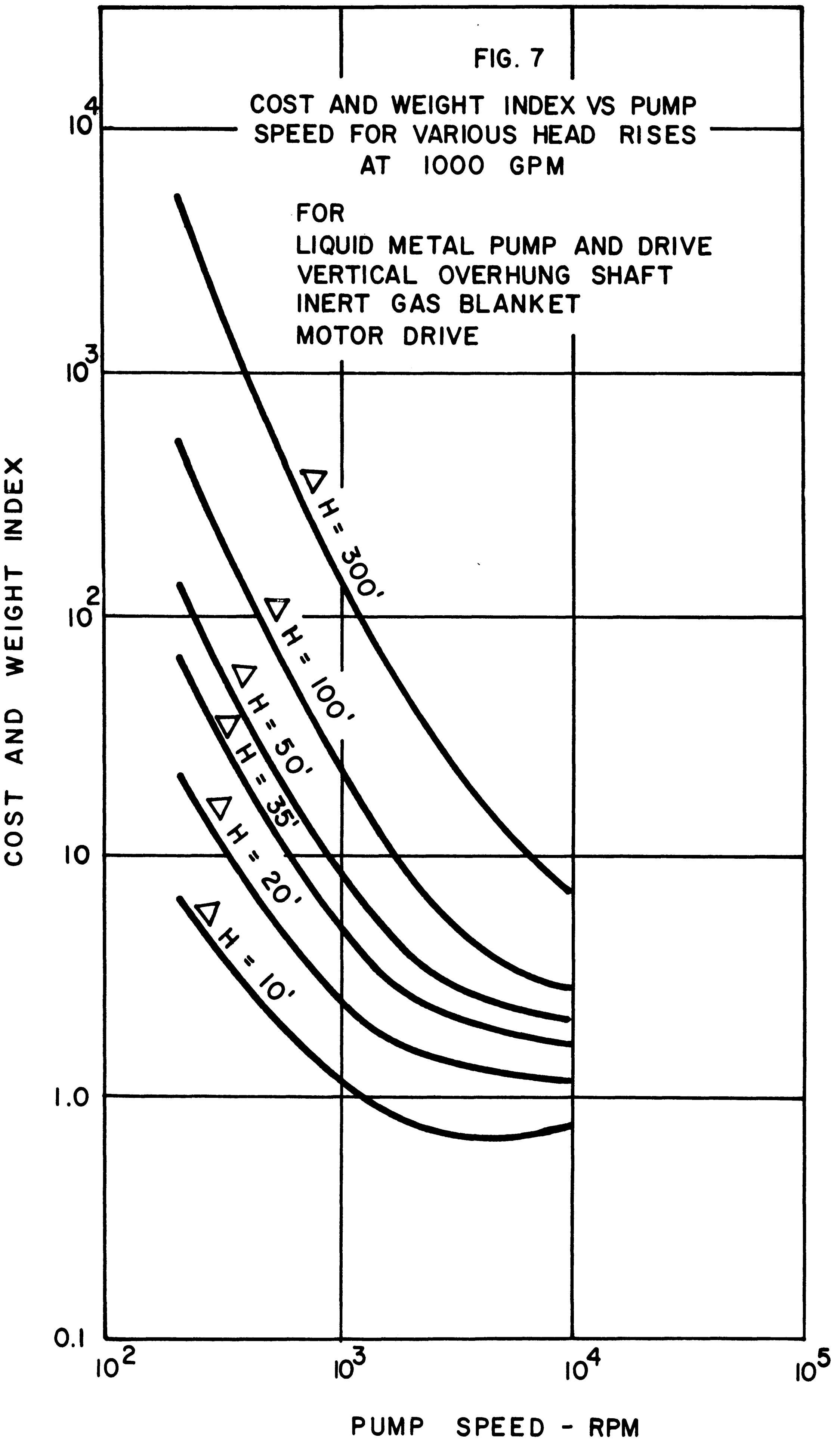


FIG. 7
COST AND WEIGHT INDEX VS PUMP SPEED FOR VARIOUS HEAD RISES AT 1000 GPM
FOR
LIQUID METAL PUMP AND DRIVE
VERTICAL OVERHUNG SHAFT
INERT GAS BLANKET
MOTOR DRIVE
ΔH = 300'
ΔH = 100'
ΔH = 50'
ΔH = 35'
ΔH = 20'
ΔH = 10'
COST AND WEIGHT INDEX
10⁴
10³
10²
10
1.0
0.1
10²
10³
10⁴
10⁵
PUMP SPEED - RPM

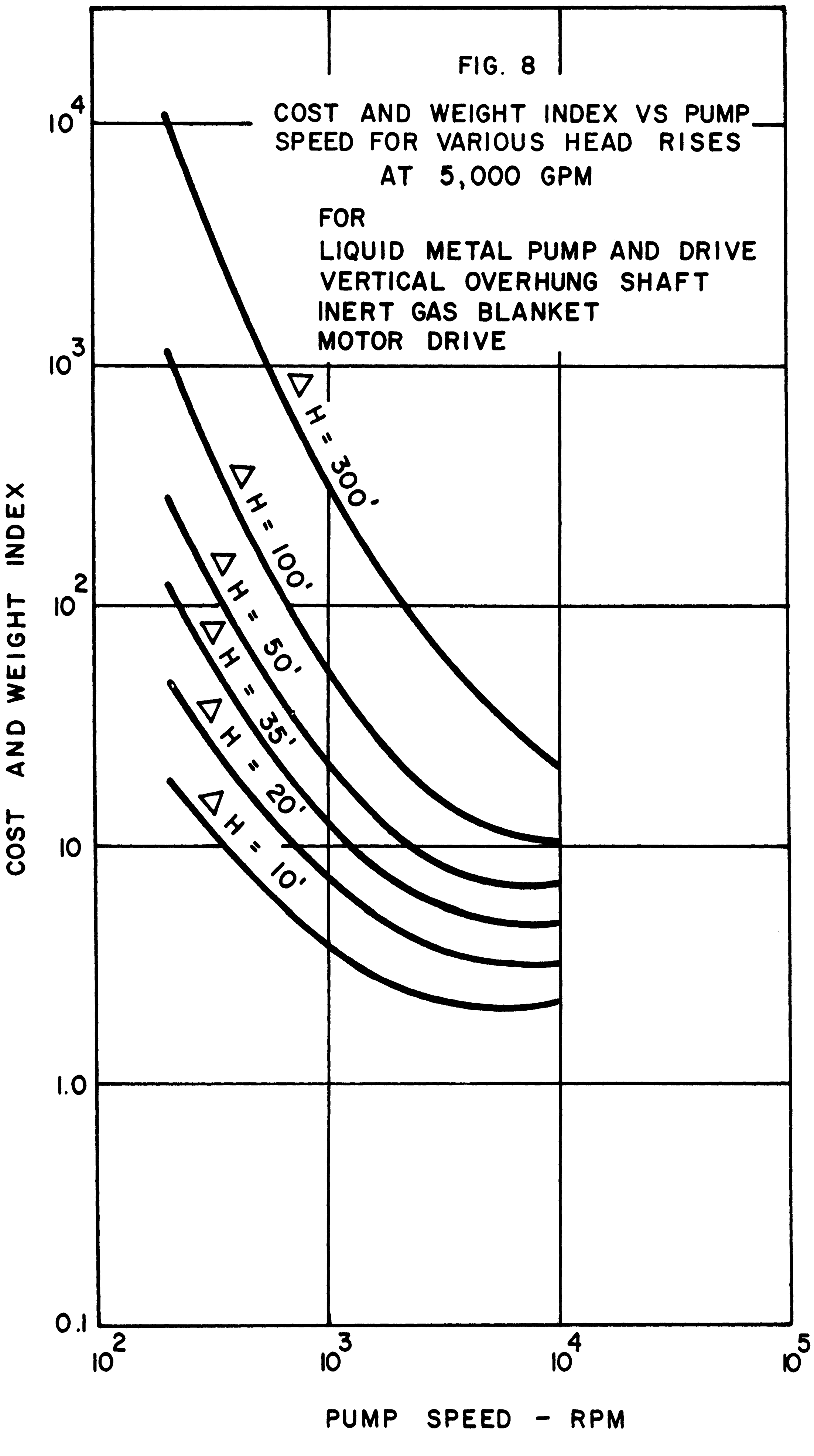


FIG. 8
COST AND WEIGHT INDEX VS PUMP SPEED FOR VARIOUS HEAD RISES AT 5,000 GPM
FOR
LIQUID METAL PUMP AND DRIVE
VERTICAL OVERHUNG SHAFT
INERT GAS BLANKET
MOTOR DRIVE
ΔH = 300'
ΔH = 100'
ΔH = 50'
ΔH = 35'
ΔH = 20'
ΔH = 10'
10^4
10^3
10^2
10
1.0
0.1
10^2
10^3
10^4
10^5
COST AND WEIGHT INDEX
PUMP SPEED - RPM

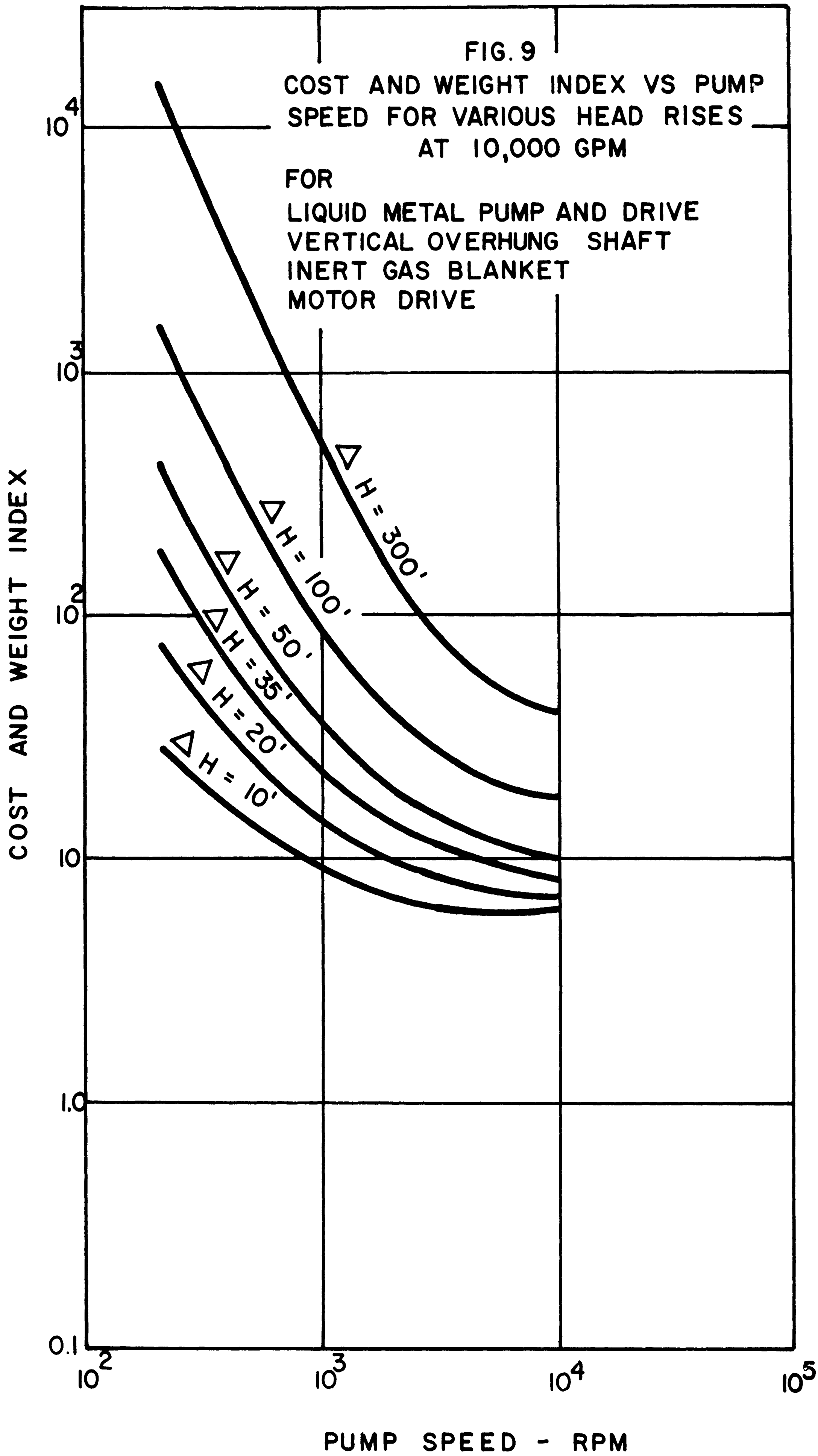


FIG. 9
COST AND WEIGHT INDEX VS PUMP SPEED FOR VARIOUS HEAD RISES AT 10,000 GPM
FOR
LIQUID METAL PUMP AND DRIVE
VERTICAL OVERHUNG SHAFT
INERT GAS BLANKET
MOTOR DRIVE
ΔH = 300'
ΔH = 100'
ΔH = 50'
ΔH = 35'
ΔH = 20'
ΔH = 10'
COST AND WEIGHT INDEX
10^4
10^3
10^2
10
1.0
0.1
10^2
10^3
10^4
10^5
PUMP SPEED - RPM

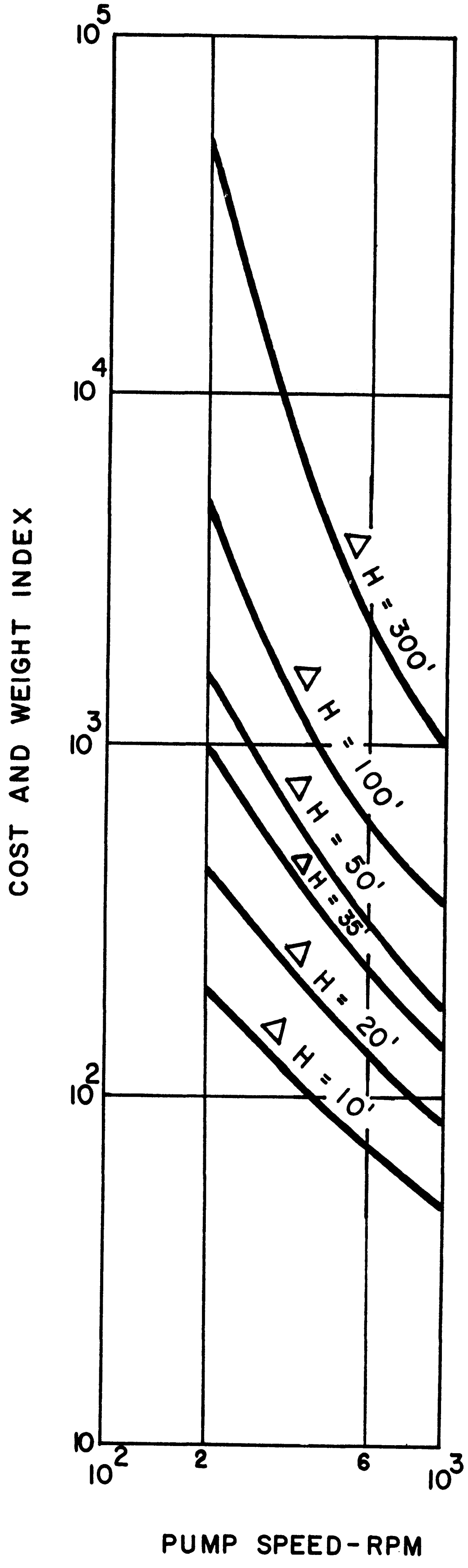
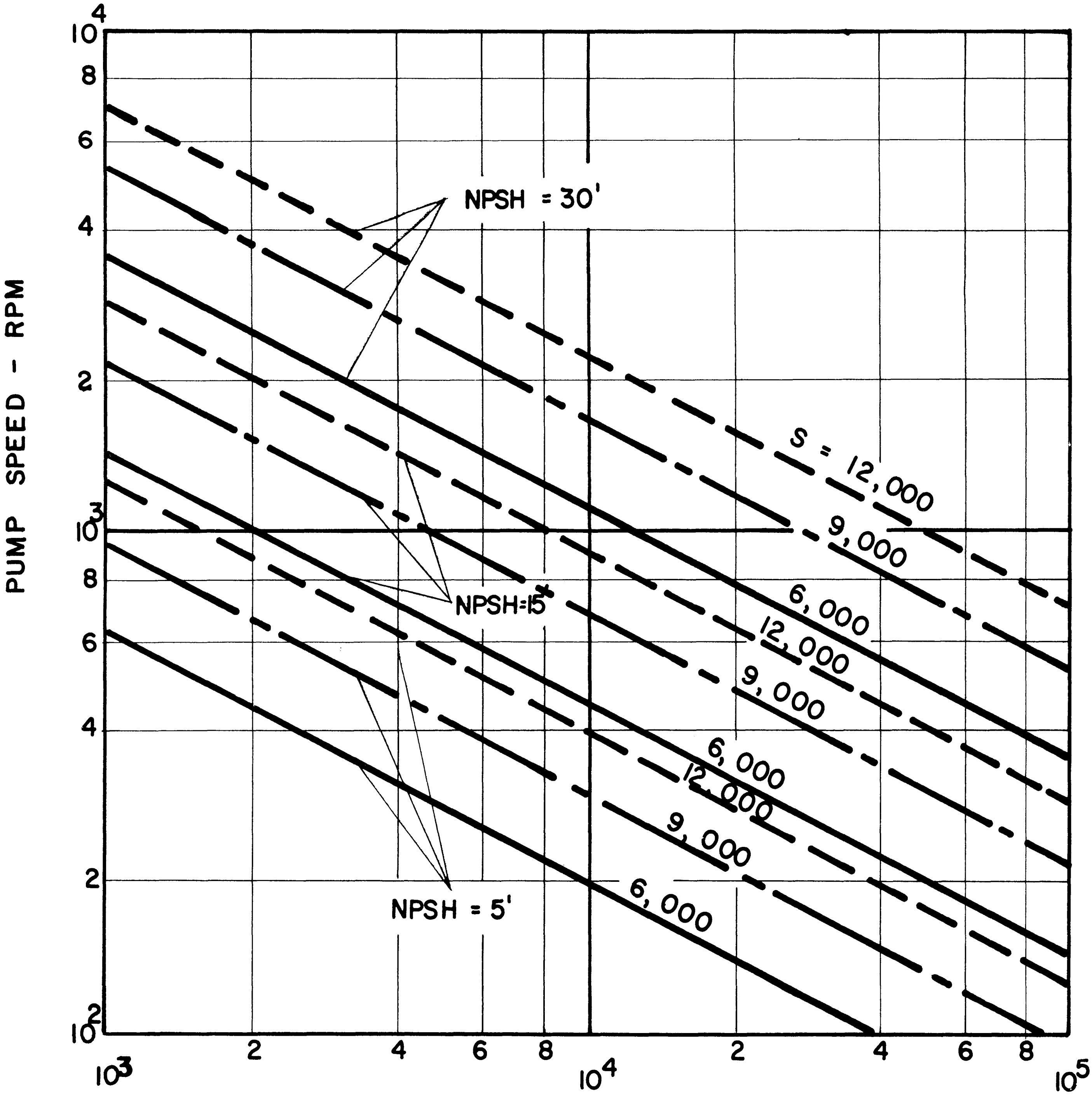


FIG. 10

COST AND WEIGHT INDEX VS PUMP SPEED FOR VARIOUS HEAD RISES AT 100,000 GPM

FOR
LIQUID METAL PUMP AND DRIVE
VERTICAL OVERHUNG SHAFT
INERT GAS BLANKET
MOTOR DRIVE

FIG. 11
PUMP SPEED VS. FLOW RATE
FOR VARIOUS SUCTION SPECIFIC SPEEDS



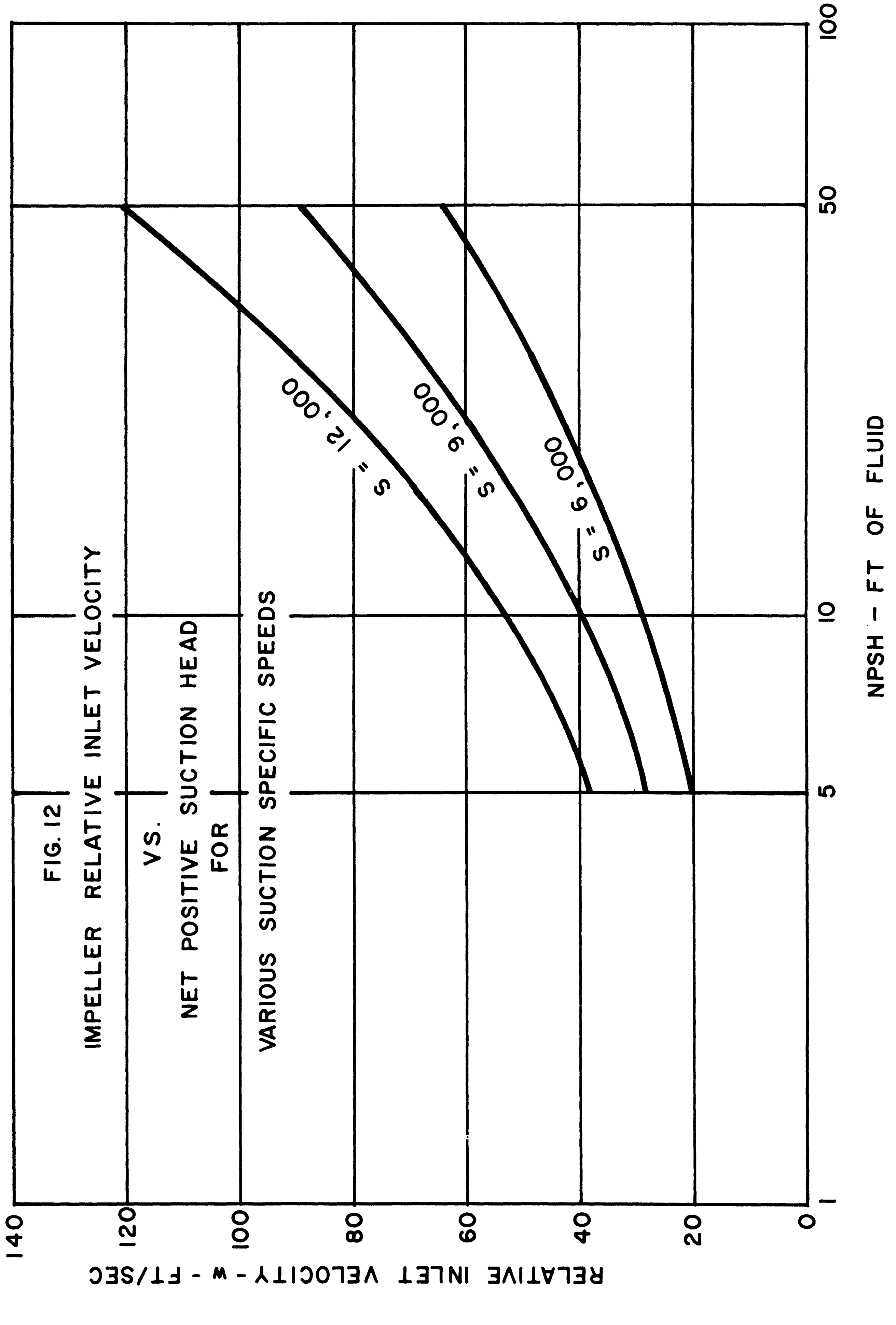


FIG. 12
IMPELLER RELATIVE INLET VELOCITY
VS.
NET POSITIVE SUCTION HEAD
FOR
VARIOUS SUCTION SPECIFIC SPEEDS

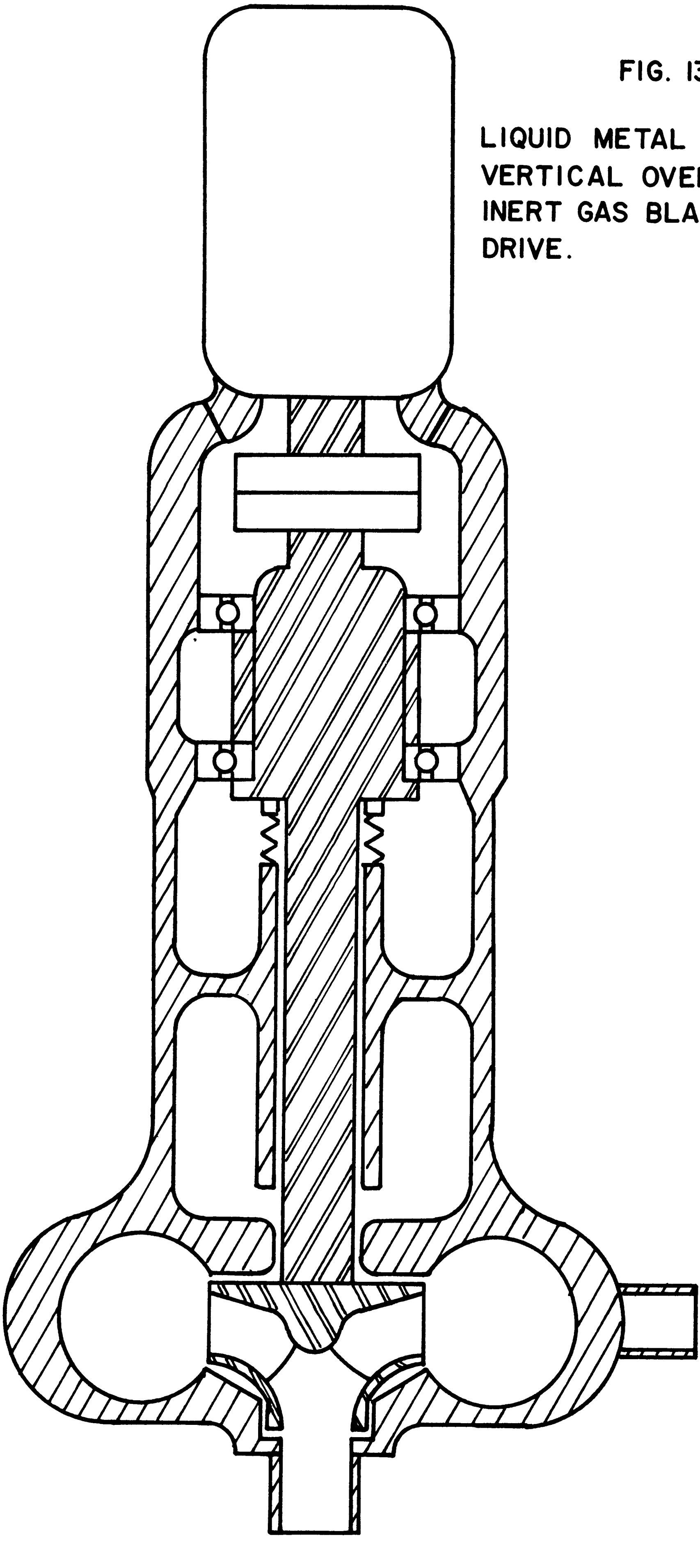


FIG. 13

LIQUID METAL PUMP & DRIVE
VERTICAL OVERHUNG SHAFT
INERT GAS BLANKET-MOTOR
DRIVE.

APPENDIX A

IMPELLER DIAMETER VERSUS PUMP SPEED CALCULATION FOR VARIOUS HEADS

(Figures 3, 4, 5, 6)

For radial and mixed flow units:

$$V = \frac{\pi \, DN}{60 \times 12} = \Phi \sqrt{2 \, gH} \qquad \text{(A-1)}$$

where:

D = mean impeller discharge diameter (inches)

N = rpm

g = acceleration of gravity, ft/sec^2

H = pump head rise (feet)

Φ = empirical constant based on specific speed

In general Φ varies from about 1.0 for low specific speed units to perhaps 1.2 for mixed flow units.

For cases where inlet eye diameter exceeded required impeller discharge diameter, the eye diameter was considered to be the controlling diameter.

The eye diameters were calculated utilizing empirical ratios of $NPSH/(Va^2/2g)$ for the various suction specific speeds desired.

APPENDIX B

PUMP WEIGHT AND COST ESTIMATE

(Figures 7, 8, 9, 10)

A motor driven, gas-sealed, vertical, high-temperature bismuth pump design was considered as a basis. This is shown schematically in Figure 13.

The total weight of this design was considered to be influenced by the following, for a radial or mixed flow unit:

1) Pump Proper (impeller and casing up to bearing housing)

2) Shaft and Bearing Assembly and Bearing Housing for Radial and Mixed Flow Units

   a) Radial Load (shaft deflection)

   b) Thrust Load (bearings)

2') Shaft and Bearing Assembly and Bearing Housing for Axial Flow Units

3) Motor

For an axial unit, it was considered that shaft design was influenced primarily by critical speed, rather than radial impeller load. Otherwise the assumptions were the same as for the radial and mixed flow designs.

A typical unit of 35 feet head rise, 5000 gpm, 700 rpm, 18 inch impeller diameter was considered. It was assumed that portions 1), 2), and 3) would have the following relative weights. The other units were prorated from this basis. This assumption is based upon previous design studies of the writer's experience.

1) 7

2) 5

3) 8

1. Pump Proper (Impeller and casing up to bearing housing)

It is assumed that the pump casing thickness depends on pressure stress. Then

$$W_1 \ \alpha \ D_c^2 \ t \ b^{1/2} \tag{B-1}$$

D_c^2 represents the radial area of the sides of the casing, t is wall thickness, b is casing axial length. Since weight is not directly proportional to width, a factor of $b^{1/2}$ was used.

Then t α HD_c where H is head rise and D_c is casing diameter. It is assumed that the pumped fluid is bismuth and thus H α pressure.

$$b \ \alpha \ \frac{gpm}{VD} \tag{B-2}$$

V is peripheral impeller discharge velocity = DN. Thus

$$b \ \alpha \ \frac{gpm}{D_c^2 N} \tag{B-3}$$

Then

$$W_1 \ \alpha \ \frac{D_c^2 \ HD \ gpm^{1/2}}{D_c \ N^{1/2}} = K_1 \ \frac{HD_c^2 \ gpm^{1/2}}{N^{1/2}} \tag{B-4}$$

2. Shaft and Bearing Assembly and Bearing Housing for Radial and Mixed Flow Units

a. Shaft

It is assumed that shaft length is fixed by sealing and installation requirements, and that shaft diameter is determined by deflection under radial load.

Radial force:

$$F_R \ \alpha \ HbD_i \tag{B-5}$$

$$\text{Shaft deflection } \alpha \ \frac{F_R L}{I} \ \alpha \ \frac{HbD_i L}{D_{sh}^4} = \text{constant} \tag{B-6}$$

where L is shaft length and is a constant for this purpose, and allowable deflection is a constant of the design. D_{sh} is shaft diameter. Then

$$D_{sh} \; \alpha \; (HbD_i)^{1/4} \tag{B-7}$$

and

$$W_{2_a} \; \alpha \; (HbD_i)^{1/2} = K_{2_a} \left(\frac{Hgpm}{D_i N}\right)^{1/2} \tag{B-8}$$

since $b\,D_c \; \alpha \; \frac{gpm}{D_c N}$ by (B-3) and $D_i \; \alpha \; D_c$. D_i = impeller diameter.

b. Bearing Assembly and Housing

Assume the total weight for these components is proportional to bearing weight which is assumed primarily affected by required thrust capacity.

Bearing thrust capacity (assuming anti-friction bearings) is approximately proportional to $D_b^2/N^{1/4}$. (Values were tabulated from New Departure Catalog.) Bearing weight must be approximately proportional to $D_b^2\, b_b$, when D_b is bearing width and b_b bearing axial dimension. Also D_b and b_b are proportional to each other. Then thrust capacity,

$$F_T \; \alpha \; \frac{\text{Weight of Bearing}}{D_b \; N^{1/4}} \tag{B-9}$$

On the other hand, pump thrust is proportional to HD_i^2. Equate thrust and thrust capacity. Assuming that $D_b \; \alpha \; D_{sh}$; substitute $D_{sh} \; \alpha \; (HbD_i)^{1/4}$ (B-7), and $b \; \alpha \; \frac{gpm}{D_c^2 N}$ by (B-3) $= \frac{gpm}{D_i^2 N}$

$$W_{2_b} = K_{2_b} H^{5/4} \; D_i^{7/4} \; gpm^{1/4} \tag{B-10}$$

2'. Shaft and Bearing Assembly and Bearing Housing for Axial Flow Units

a. Shaft

It is now assumed that shaft diameter is controlled by critical speed considerations since the radial load is not high for an axial flow pump.

f is approximately proportional to $(\delta)^{-1/2}$ where δ = deflection under impeller weight and f = natural frequency.

$f \alpha N$ (since a safe proportion over the operating speed must be maintained).

$$\delta \;\alpha\; \frac{Wgt_i}{D_{sh}^4} \;\alpha\; \frac{1}{N^{1/2}} \quad \text{or } D_{sh} \;\alpha\; (Wgt_i\ N^2)^{1/4} \tag{B-11}$$

Assume impeller weight for an axial impeller,

$$W_i \;\alpha\; b\, D_i^2 \tag{B-12}$$

and that $b \;\alpha\; D_i$. Then

$$Wgt_i \;\alpha\; D_i^3 \tag{B-13}$$

$$W_{2'_a} \;\alpha\; D_{sh}^2 = K_{2'_a}\, D^{1.5}\, N \tag{B-14}$$

by substituting (B-13) into (B-11).

3. Motor

Data was gathered for existing motors from Kent's Mechanical Engineer's Handbook and from the General Electric Company.

$BHP \;\alpha\;$ gpm x H for the pump.

It was noted that approximately

$$W_3 \;\alpha\; \frac{BHP^{3/4}}{N^{1/2}} = K_3 \; \frac{gpm^{3/4} \times H^{3/4}}{N^{1/2}} \qquad \text{(B-15)}$$

The curves shown in Figures 7, 8, 9, and 10 are approximate plots of the results from these calculations. It will be noted that for a given head and flow, pump weight is reduced more drastically for low specific speed units than for high. This result is reasonable since for the high specific speed and axial units, diameter reduction is not accomplished by increasing speed. In fact extreme speed increases show a weight increase because the shaft weight increases (because of fixed length and critical speed requirement) faster than the motor weight decreases.

These curves are included only to show the approximate trends and are not intended as accurate data.

APPENDIX C

PUMP SPEED VERSUS FLOW RATE FOR VARIOUS SUCTION SPECIFIC SPEEDS

(Figure 11)

This is simply a plot of the relation

$$S = \frac{N \sqrt{gpm}}{(NPSH)^{3/4}} \qquad (C-1)$$

APPENDIX D

RELATIVE INLET VELOCITY VERSUS NPSH

(Figure 12)

For a given suction specific speed there are empirical constants relating the maximum desirable fluid velocities to the NPSH. From one version of these relations the relative fluid velocity at impeller inlet at the outside periphery of the impeller was calculated for various NPSH with suction specific speed as parameter. These are plotted in Figure 12. This is not intended as exact data and does not imply for a given S and NPSH only one relative velocity is possible. However, it shows the order of magnitude and trend of the variation to be expected in present day design practice.