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"OCEAN THERMAL DIFFERENCE POWER PLANT TURBINE DESIGN"

by

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DISCLAIMER

INTRODUCTION

A Rankine cycle operating on the thermal differences of the Gulf Stream is very similar to a conventional fossil fuel power plant. All of the major components are similar in function but different design criteria and operating conditions are required for each system. A schematic diagram of the closed cycle ocean thermal power plant is shown in figure 1. The most significant design condition imposed on this cycle is that of the low temperature differences of the ocean site. In the case of the turbine this temperature difference or available head between states one and two is further reduced by heat exchanger losses. In considering an overall system analysis it is necessary to carefully study the tradeoffs between heat exchanger and turbine efficiencies. This interaction is particularly significant because of the fixed temperature differences. Power output can be written for the power plant as follows:

$$P = \dot{m}\ \eta_p \eta\ H \qquad (1)$$

In this η_p is the power efficiency of the system and is defined as the ratio of Net power output to gross power generated. This term is a function of the parasitic losses in the system and therefore decreases as the mass flow rate of the working fluid increases. As a result a one percent decrease in turbine efficiency at a fixed power output would require significantly more than a one percent increase in mass flow rate.

Satisfactory plant performance is critically related to cycle and therefore turbine efficiency. It is to our interest to choose a turbine design that will give the highest practical efficiency. Relative to other components in the system turbine cost does not appear to be a major factor. It is therefore on a basis of choosing turbine design for maximum efficiency that we have

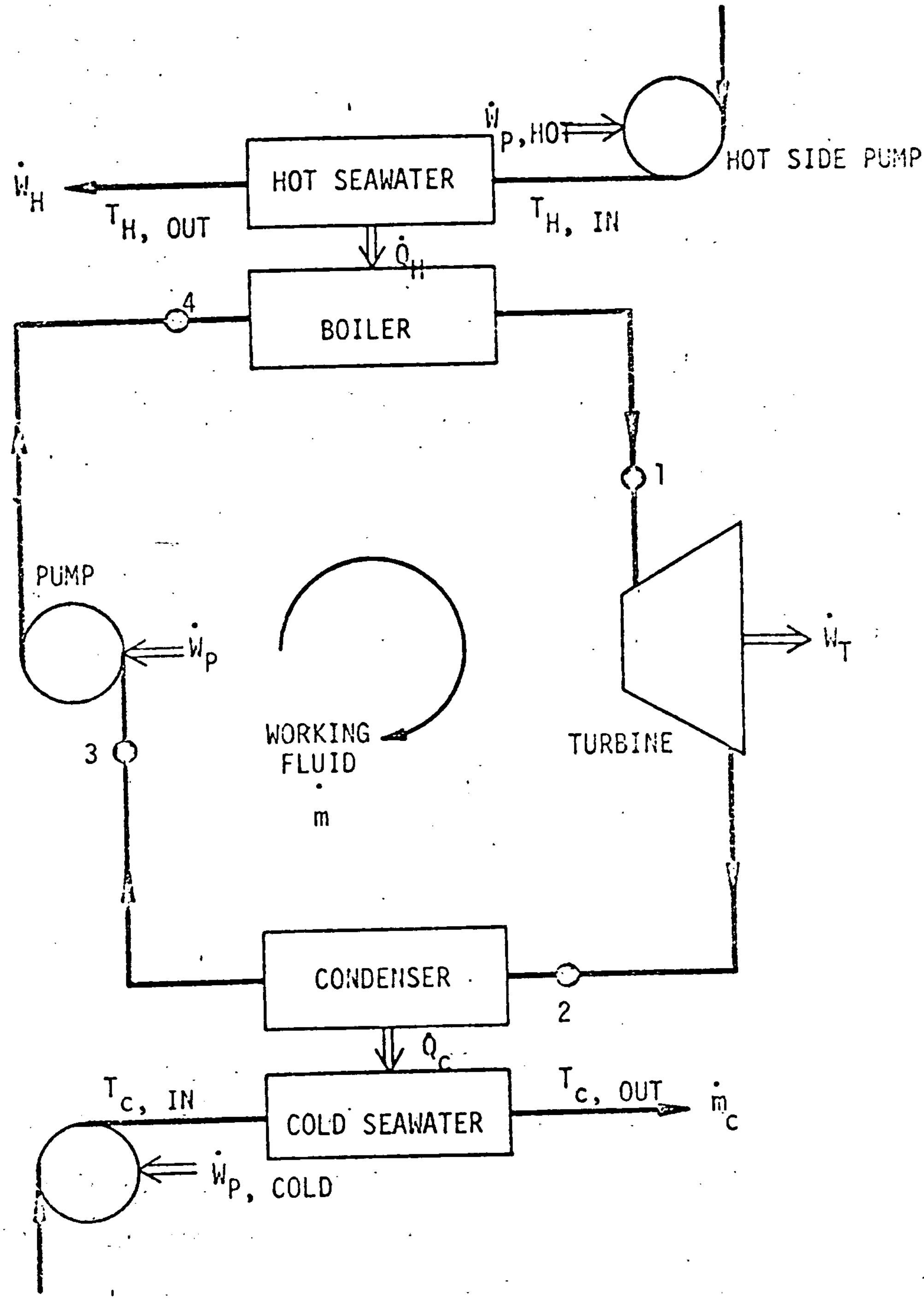

Figure 1 OTGM Schematic Diagram

proceeded. The purpose of this report is to describe considerations in design point selection for the turbine based on thermodynamic and fluid dynamic criteria.

Turbine Types

Both axial and radial flow turbines could be used in this application. Efficiencies above 90% have been obtained with both[1]. When range of operation at high efficiency is compared on the basis of specific speed (turbine speed for unit head and unit flow), the axial machine has a wider range. Blade tip speeds in this application are low enough so that stress considerations are not severe for either configuration. Radial flow machines are usually able to achieve higher stage loadings. For most fluids and the temperature differences in this application, only a single stage is required for radial or axial flow.

Although both radial and axial flow machines appear acceptable at design point,some consideration must be given to off-design performance. For the OTG power plant where variation in operating conditions are found during the year at each site,this is significant. Axial machines of fixed geometry operating at constant head have less variation in efficiency with speed than a radial machine. If partial admission is used for thottling,axial machines again appear to be favored. These considerations have led us to choose the axial flow machine for our application.

Working Fluid Influence on Design

Working fluid selection is as important to the turbine as it is to other components in the cycle. However because of the restrictive cycle conditions of an ocean thermal gradient power plant the working fluid influence is somewhat different from a conventional power cycle. Using a hot side temperature of 70°F and a cold side temperature of 40°F it is possible to determine cycle operating properties for working fluids. A summary for ten selected working fluids is given in Table I. With the exception of water and carbon dioxide all have reasonable vapor pressures at the upper cycle working temperature. And although low pressure differential exists across the turbine for some of the candidate fluids all are practical. The isentropic enthalpy drop across the turbine does vary by almost one order of magnitude for the different fluids. Excluding water this leads to mass flow rates which differ by a factor of eight for any given power output. It would appear that ammonia and normal butane would be attractive fluids if flow losses are to be reduced.

Generally it is said that a high molecular weight is desirable for a working fluid so as to reduce turbine blade speed. This is usually useful to lower the number of turbine stages required. In this application the enthalpy drop across the turbine is so low that any of these fluids would only require one stage and in most cases the spouting velocity (isentropic velocity when expanded to zero pressure) is so low that speed induced blade stresses are not a problem.

Turbine size can be a problem however with OTG power plants. With fluids having low isentropic head large flow rates are required for a given power output. For example a 100 mw simple steam power plant (800 psia, 900°F) would have a mass flow of about 250 lb/sec. All the candidate fluids in Table I have flow rates from 20 to 160 times greater. A grouping of terms which is a better indicator of relative size is also given in Table I. It can

TABLE I WORKING FLUID CYCLE PROPERTIES

WORKING FLUID	VAPOR PRESS. AT 70°F	ΔP 70° to 40°	L BTU/lb	H BTU/lb	v_g 40°F	$\dot{m}$ lb/sec for 100 mw	$MP_cL^{1.5}$	M	Turbine Size MW
Water	.3	.13	1071.	36.	2445.8	2,900	170,343	18	.15
Propane	124.3	82.8	156.7	6.	1.348	17500	7.85×10^6	44	3.1
Carbon Dioxide	853.4	199.8	95	2.9	.144	36400	26.6×10^6	44	4.72
Ammonia	128.8	39.6	536.2	20.2	3.971	5200	18.8×10^6	17	22
Sulfur Dioxide	49.1	16.2	162.2	6.1	3.02	17200	4.36×10^6	64.1	1.44
n-Butane	31.6	13.9	163.5	11.0	4.88	9600	1.99×10^6	58.1	3.9
Freon-12	61.4	23.4	64.1	2.5	.774	42100	2.36×10^6	120.9	.60
Freon C318	40.1	12.8	48.5	3.2	1.13	33000	2.06×10^6	200	.77
Freon 502	151.3	39.7	63.1	3.1	.447	34000	6.24×10^6	111.6	1.8
Genetrar 12/31	85.7	33.8	72.2	4.3	.907	24210	3.29×10^6	103.5	2.0

be shown from thermodynamic principles that the grouping [Molecular Weight * Condenser Pressure * (Latent Heat)$^{1.5}$] is proportional to the turbine output per unit area of the turbine exit.(2) In order to reduce turbine size and cost, this parameter should be numerically large. This parameter is compared along with turbine mass flow rate for the candidate working fluids in figure 2. If low flow rate and small turbine size are considered desirable, ammonia again appears to be the best fluid followed by normal butane and propane. This examination can be carried one step further. Trouton's rule states that the product of molecular weight and latent heat divided by the absolute temperature is approximately constant for all fluids at atmospheric pressure. This would suggest that the grouping given above be modified so that turbine output per unit area would be proportional to the product of condenser pressure and square root of the latent heat, indicating that a desirable fluid should have low molecular weight and high vapor pressure at the temperatures in question. Again ammonia followed by propane appear to be favorable candidates.

Using geometrically similar axial flow turbines operating under similar conditions, the fluids can be compared as to power output size for an individual unit.(2) The maximum power per unit for similar design is given by the following:

$$P_{max} = .0475 \frac{(H)^{2.5}}{v_{g_{40}}} \quad mw \qquad (2)$$

This calculation leads to the results shown in the last column of Table I. Ammonia unit size is larger than all other fluids by at least a factor of four. When building an overall OTG plant with 400 megawatts net capacity ammonia would appear to be the best working fluid from a turbine point of

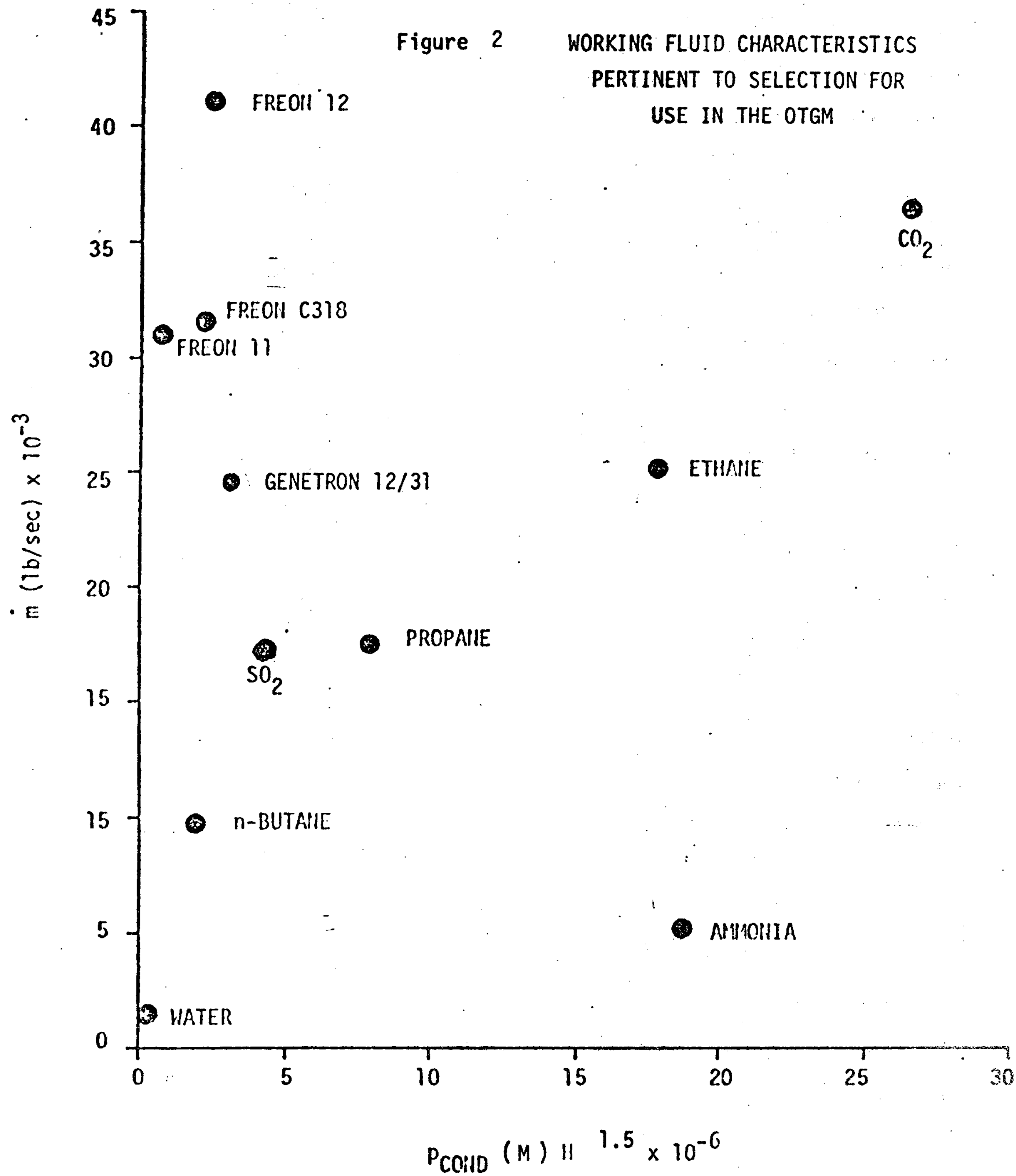

Figure 2 WORKING FLUID CHARACTERISTICS PERTINENT TO SELECTION FOR USE IN THE OTGM

view as it will require the fewest number of units. However for the design studies to be described below, three working fluids were considered: Ammonia, propane and refrigerant 12/31.

Turbomachine Design Point

Turbomachines are usually classified by dimensionless terms. These dimensionless terms can be used to describe general turbine geometry and can be used to correlate turbine performance. Dimensionless analysis can be used to delineate numerous dimensionless groups to describe flow in turbomachines. The usual fluid mechanical groups arise as well as a series especially pertinent to turbomachines. Although a number of different groups have been used to describe turbomachines, for this study it has been convenient to use two particular groups: Specific speed and specific diameter. Specific speed N_s and specific diameter D_s are defined as follows:

$$N_s = \frac{NQ^{1/2}}{H^{3/4}} \tag{3}$$

$$D_s = \frac{DH^{1/4}}{Q^{1/2}} \tag{4}$$

The units used for each term are those conventionally used and it can be seen that their use produces an N_s and D_s which are not truly dimensionless. For most types of operation these two parameters can be used to adequately characterize turbine use. In particular N_s and D_s can be related to turbine efficiency. In order to achieve a high efficiency it also becomes necessary to control other dimensionless variables as well and this will be described below.

Numerous investigations have produced correlations of the effect of design variables on flow efficiency. Those which achieve the most success rely heavily on experimental loss data for turbomachine blades. The work of Balje is the most extensive[3,4,5]. His most recent study involves a correlation of performance as a function of design variables[4,5]. One result of this study, shown in figure 3, is a N_s-D_s diagram which indicates regions of efficient operation. Although there is a region of high efficiency the maximum efficiency is found where $N_s = 120$ and $D_s = 1.2$.

In addition to indicating the numerical values for N_s and D_s, other design parameters are given as well. Figures 4,5,6 can be used to determine axial flow full admission turbine design for a given operating point when specific speed is known. At $N_s = 120$ the design variables giving high efficiency are shown in Table II

$N_s = 120$	$D_s = 1.2$
$S/h = .02$	$h/D = .22$
$\alpha_2 = 15°$	$\alpha_3 = 85°$
$\beta_2 = 42°$	$\beta_3 = 12°$

Degree of reaction = .6

Turbine total efficiency = 91.5%

TABLE II TURBINE DESIGN POINT PARAMETERS

Specifying N_s only determines the shape of the turbine. Other design variables must be established in order to determine turbine size. Of the variables remaining in equation 3 only two are independent. Specifying a working fluid and thermal site conditions reduces the number of independent variables to one. For a turbine, unit power output is related to mass flow rate as follows:

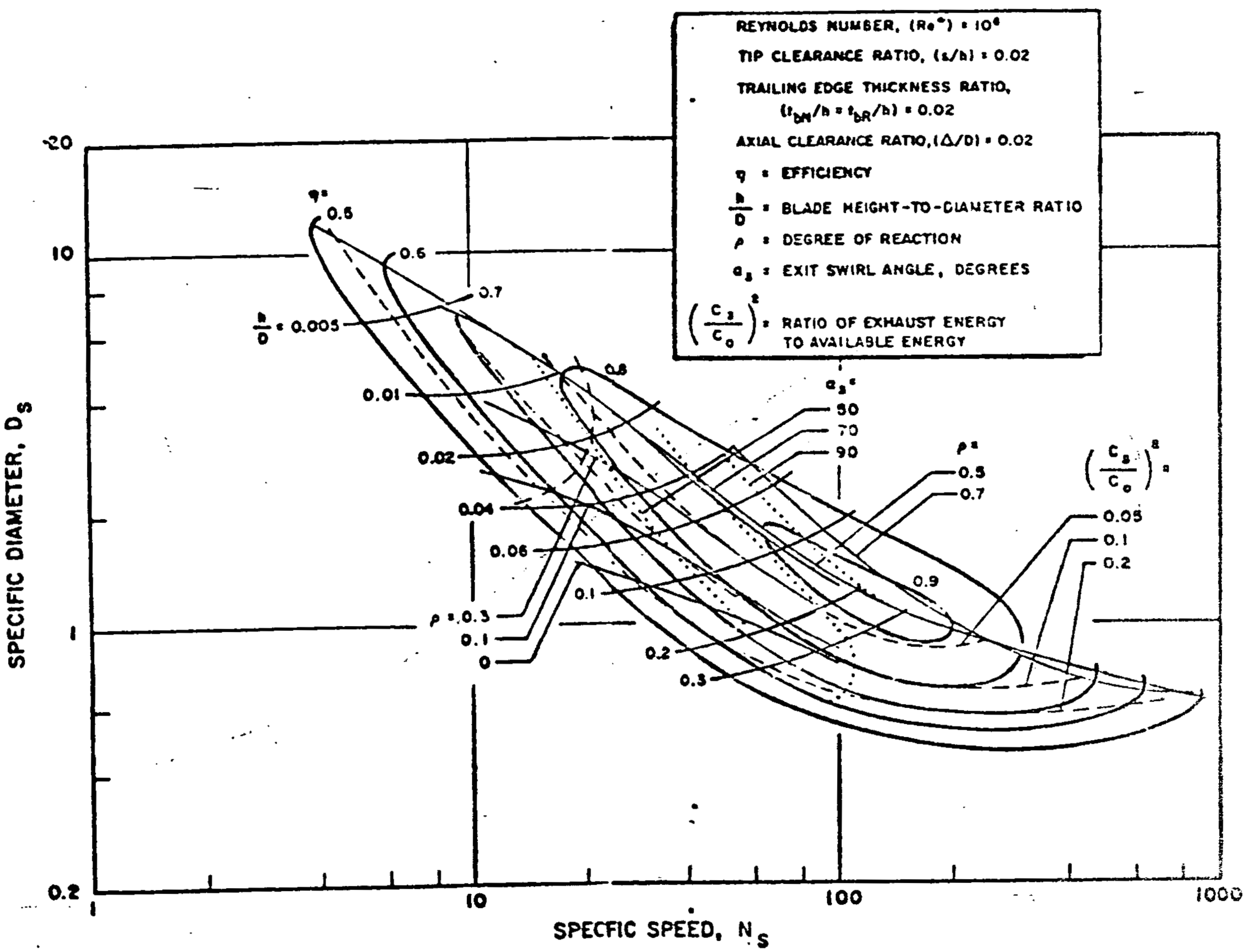

Figure 3 N_s-D_s Diagram For Full Admission Axial Flow Turbine

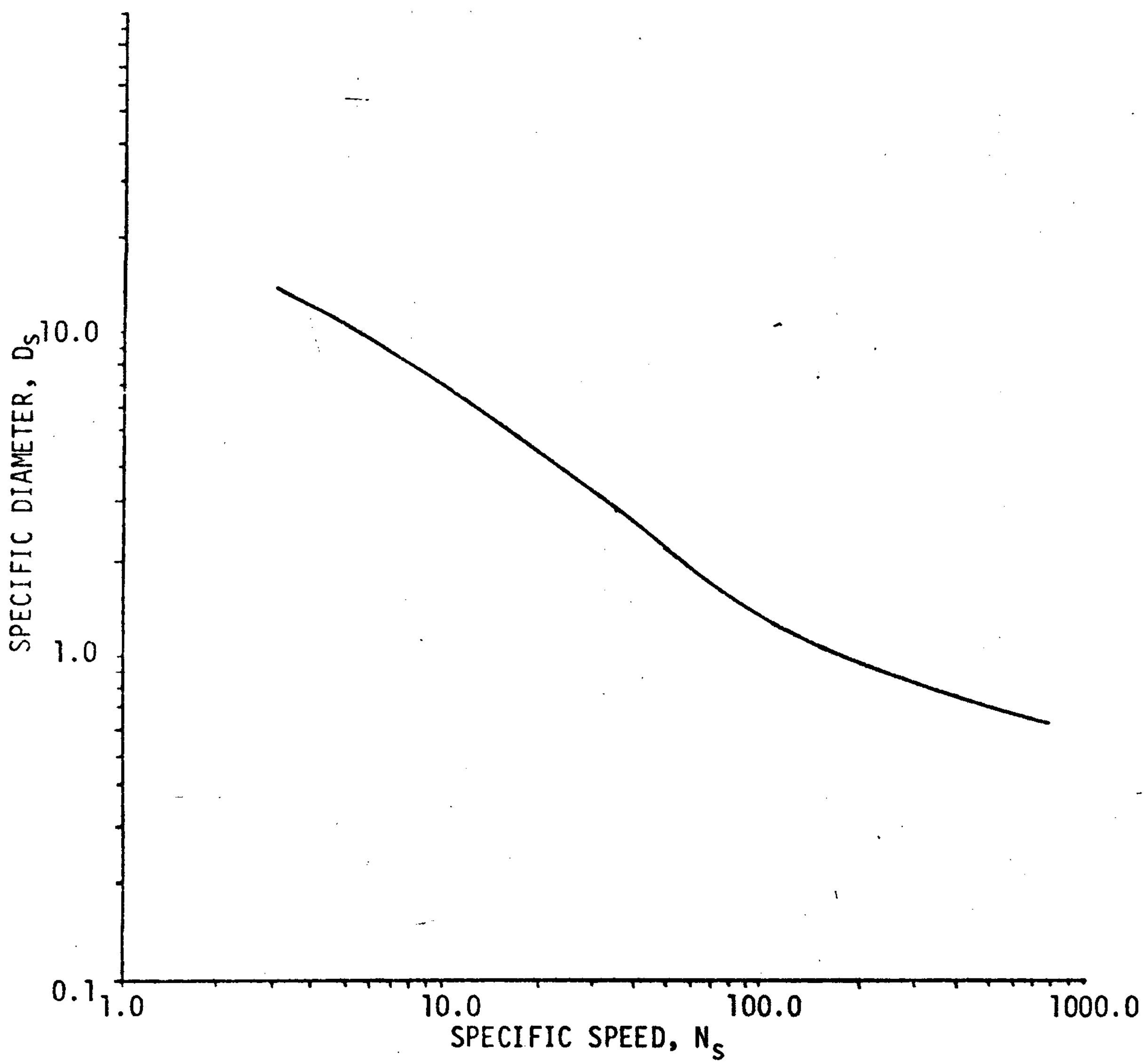

Figure 4 Optimum Specific Diameter for Axial Flow Turbine

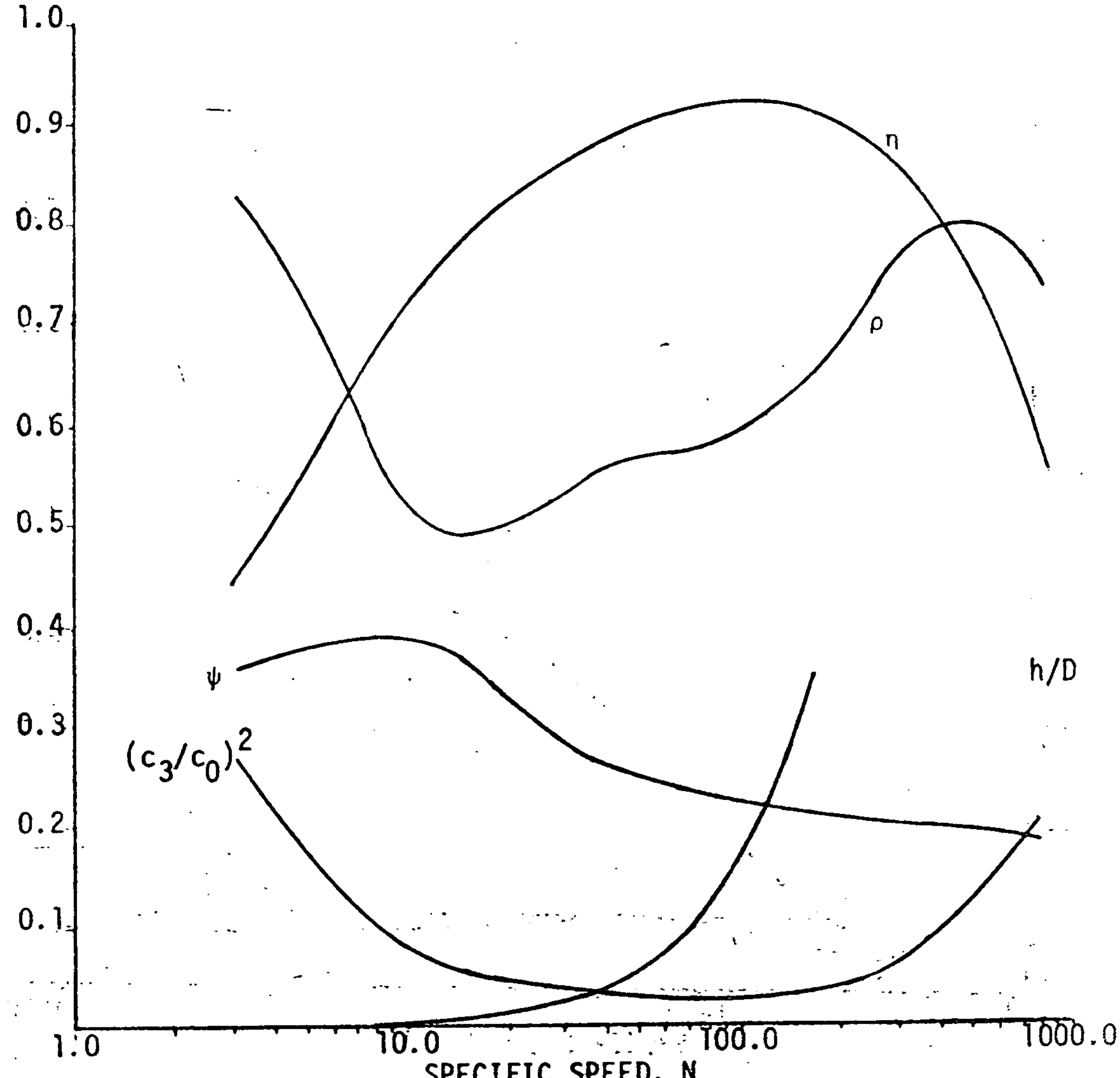

Figure 5 Axial Flow Turbine Design Parameters

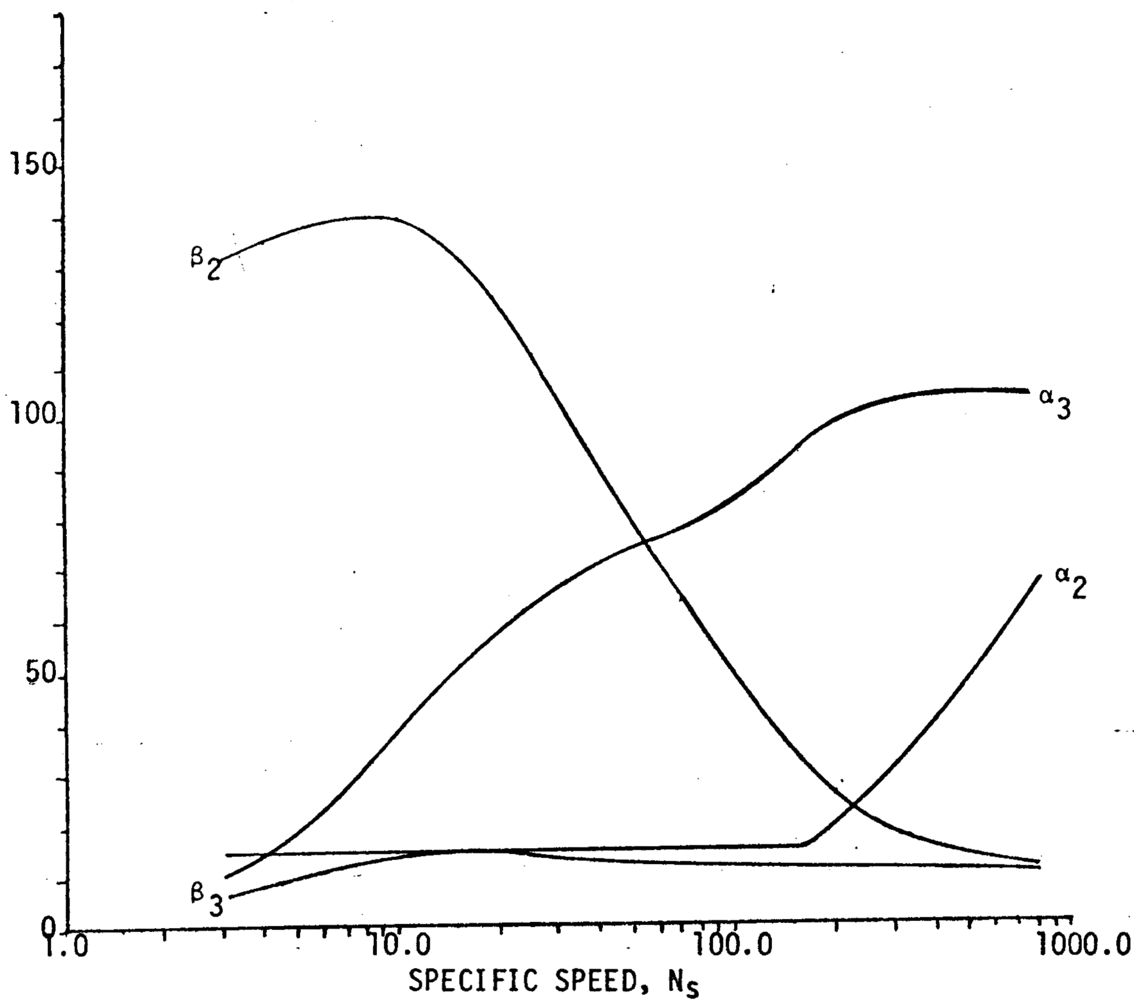

Figure 6 Axial Flow Turbine Flow Angles

$$P = \dot{m}\,\eta\,H \qquad (5)$$

Introduction of this equation into the specific speed equation yields:

$$N_s = \left(\frac{1}{\eta}\right)^{1/2} \left(\frac{v\ X}{H^{5/2}}\right)^{1/2} N\sqrt{P} \qquad (6)$$

For a given working fluid this equation relates power and turbine speed at the design point. As turbine power output increases, optimum turbine speed decreases as the square root of the power increase.

Numerous calculations were performed so as to establish design point conditions for all possible operating conditions with the number of variables involved. Only illustrative results are shown here. Figures 7,8 and 9 give results for ammonia, propane and R 12/31 at a fixed boiler temperature of 68°F and a series of condenser temperatures. At a gross power output of 30 mw per unit ammonia diameters range from 6 to 9.5 feet for condenser temperatures from 42 to 56°F. Turbine speeds run from 3100 RPM to 1300 RPM for the same conditions. For a propane turbine size ranges from 8.5 to 14 feet and speed from 1200 to 475 RPM. With R 12/31 as the working fluid the turbine size would range from 36 to 70 feet and the speed from 240 to 70 RPM. This range of size and speed must be narrowed at the design point for satisfactory operation. At this power level the R 12/31 turbine size is prohibitive. If the same size R 12/31 turbine is used as was found for propane, the gross power output per unit would be about 1 mw. R 12/31 seems to become an unreasonable turbine working fluid for these thermal differences.

Returning to figures 7 and 8 for ammonia and propane it can be seen that the optimum design point dimensions are strongly dependent on site conditions. Sixteen 30 to 35 mw gross turbine units should produce the

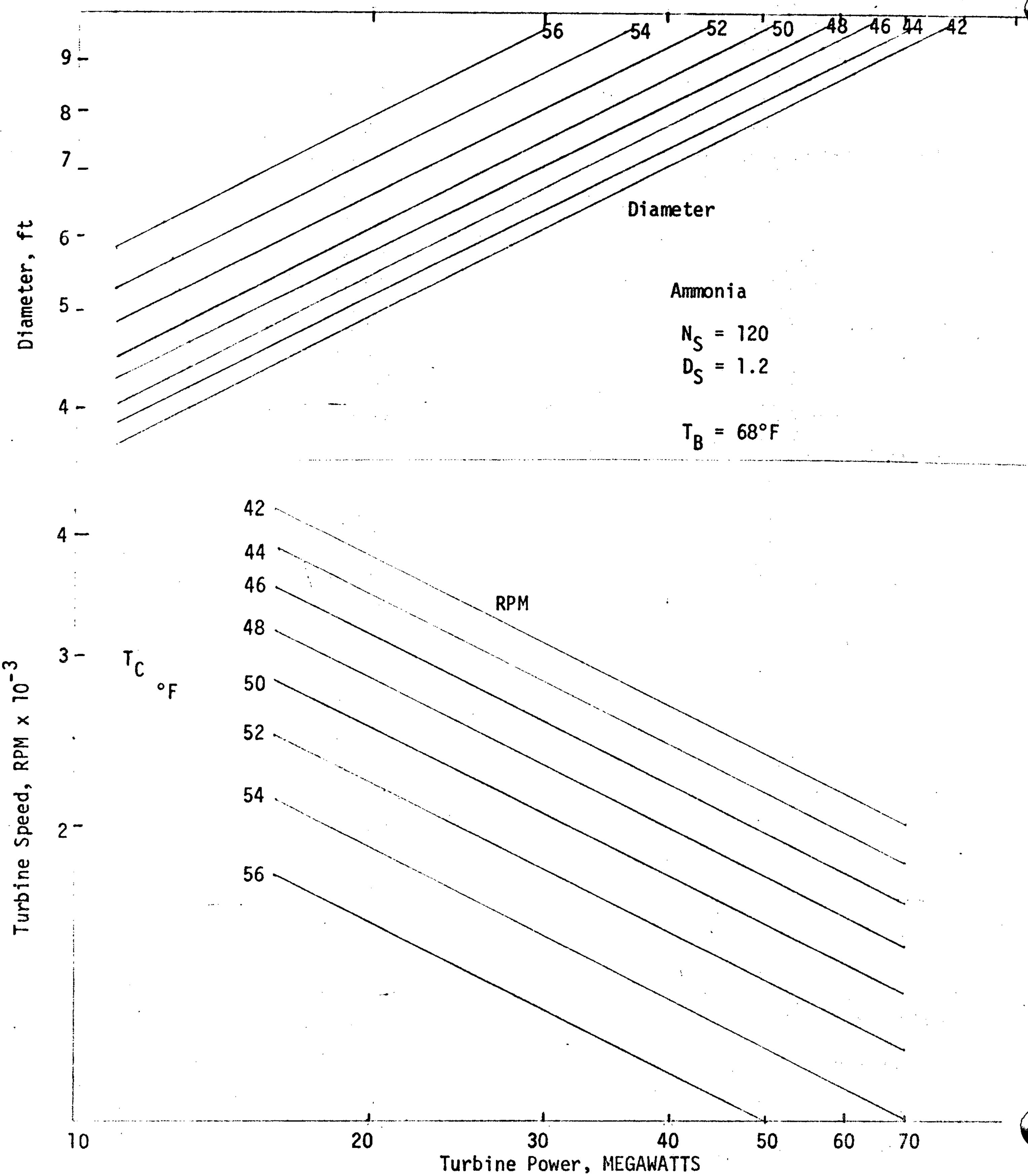

Figure 7 Ammonia Turbine Design For T_B=68 F

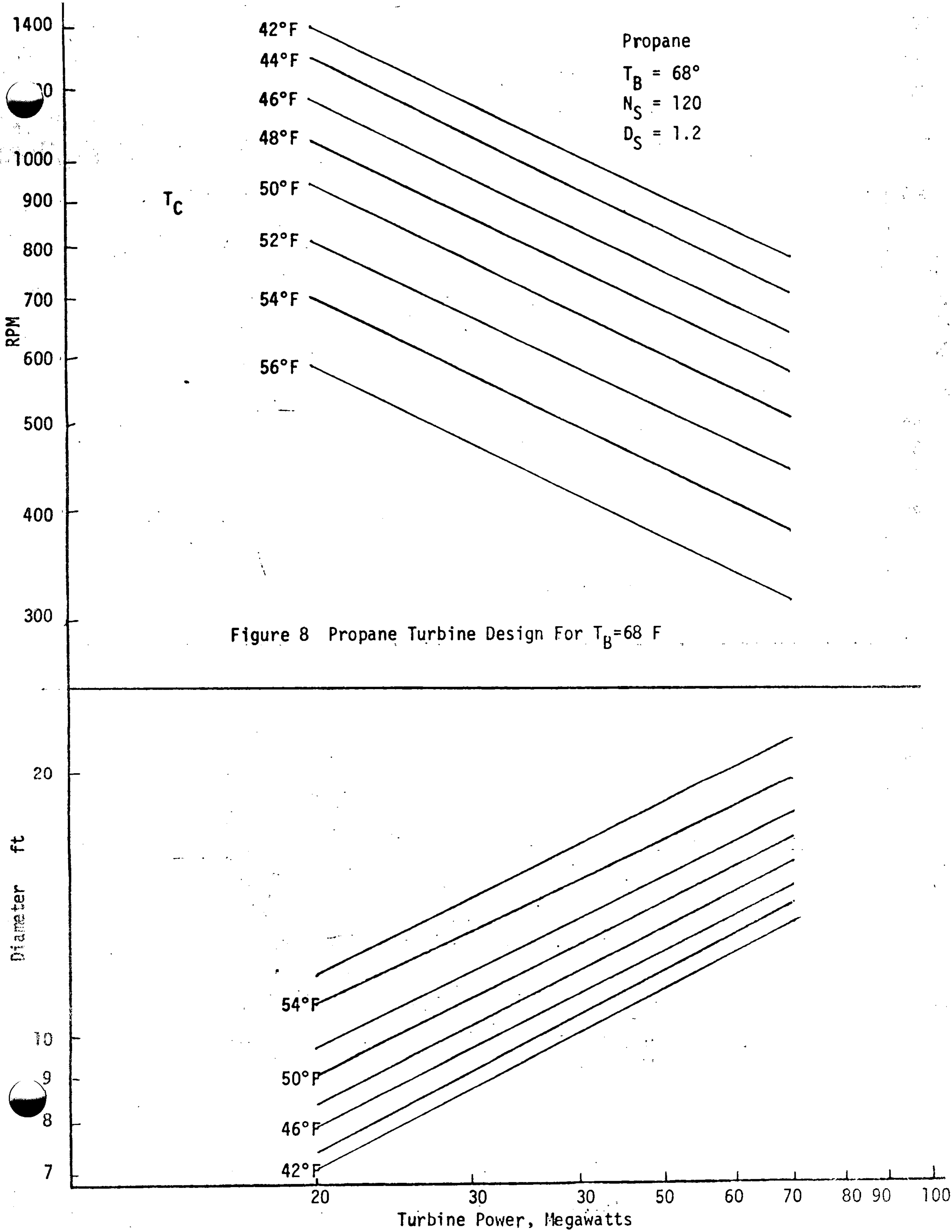

Figure 8 Propane Turbine Design For T_B=68 F

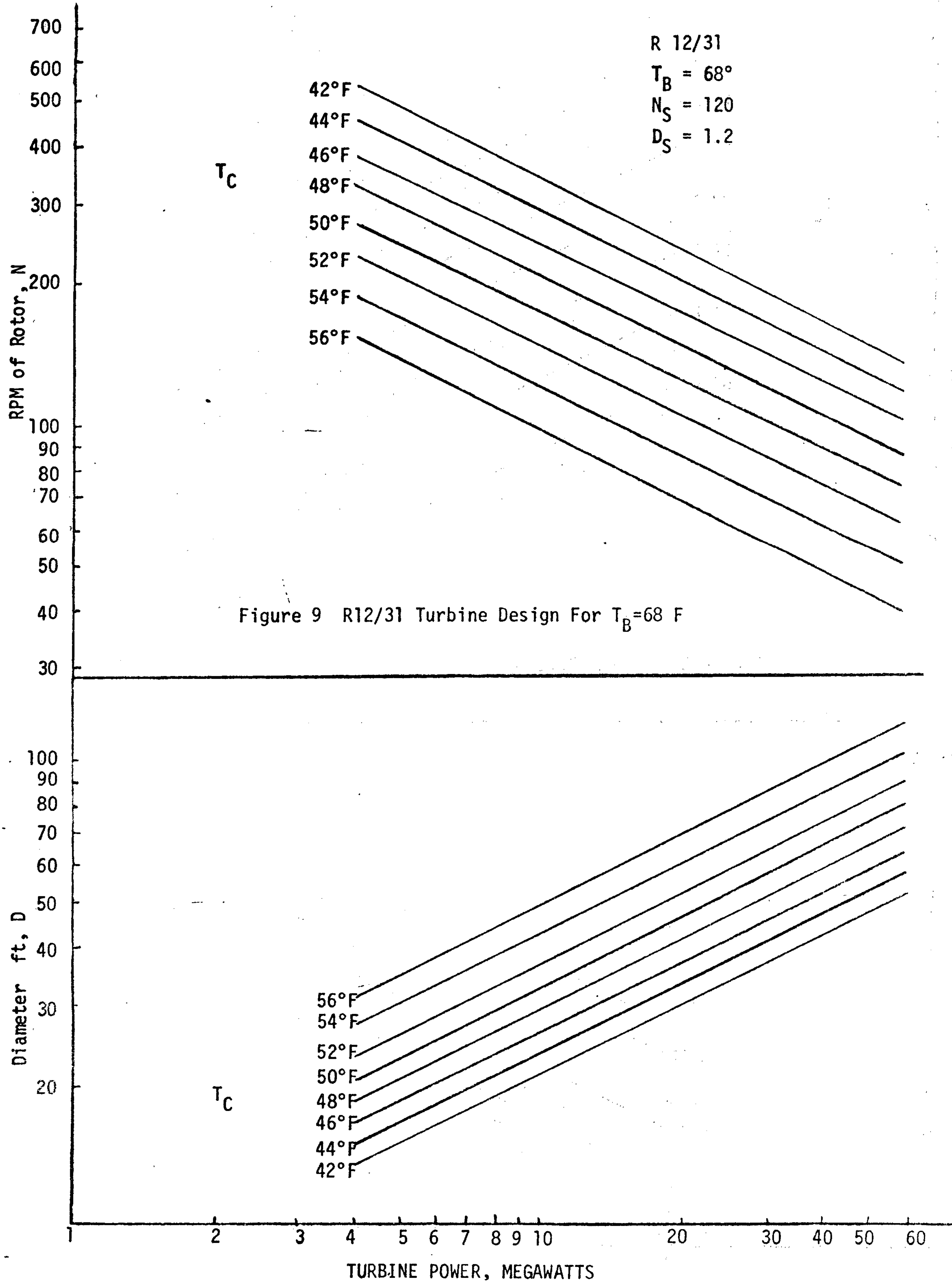

Figure 9 R12/31 Turbine Design For T_B=68 F

400 mw net capacity needed for our proposed plant design. Optimum design diameter and speed will depend on site thermal conditions as well as on heat exchanger design in the boiler and condenser. Further work is required in order to determine the behavior of the design chosen in off-design conditions. This requires a different type of analysis and should include as well off-design conditions for the other components in the overall power plant.

In order to get an approximation of off-design effects on turbine operation the Balje and Binsley design charts can also be used. By fixing turbine power output and/or turbine speed, the range of operation can be determined as a function of specific speed and the other operating parameters. This analysis is not off-design analysis because these charts are based on the optimum specific diameter being used for each specific speed. However this approach does allow a region of operation to be studied and also gives an indication of the range of turbine efficiencies to be expected. Results for ammonia, propane and R 12/31 are shown in figures 10 to 18. In figures 10 to 13, turbine power was fixed and mass flow rate, turbine speed and turbine diameter were calculated as a function of condenser and boiler temperature at a fixed specific speed of 120. For a 30 mw unit, the ammonia turbine can be run at 3600 RPM with high efficiency if the temperature difference the turbine sees is 30°F or greater. For a 15°F temperature difference the turbine speed could be 1800 RPM. For a 20 mw propane turbine the temperature difference for high efficiency must exceed 30°F for 1800 RPM speed. Increasing turbine speed is desirable because this reduces the required turbine size and cost.

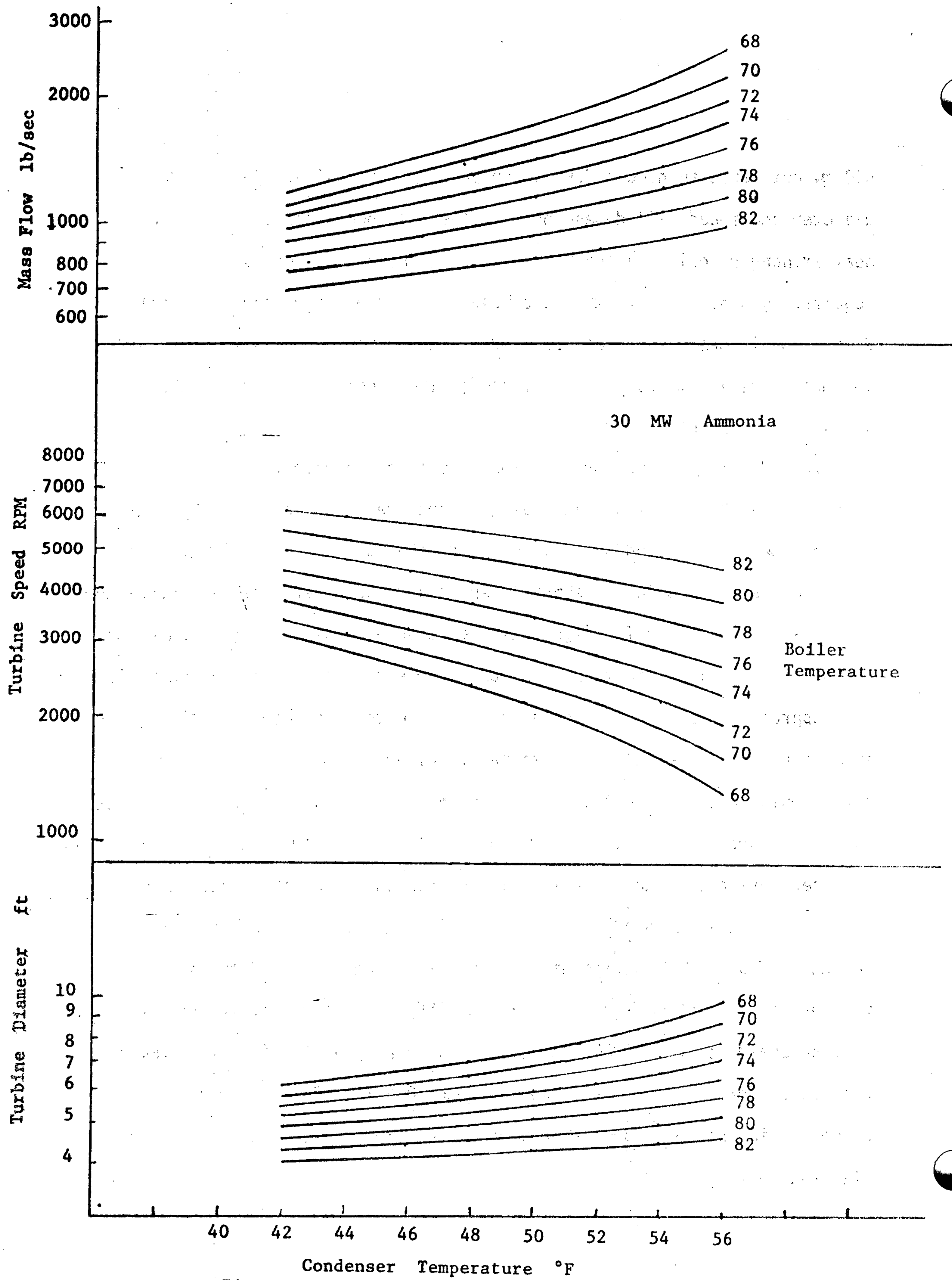

Figure 10 30 mw Ammonia Turbine Design

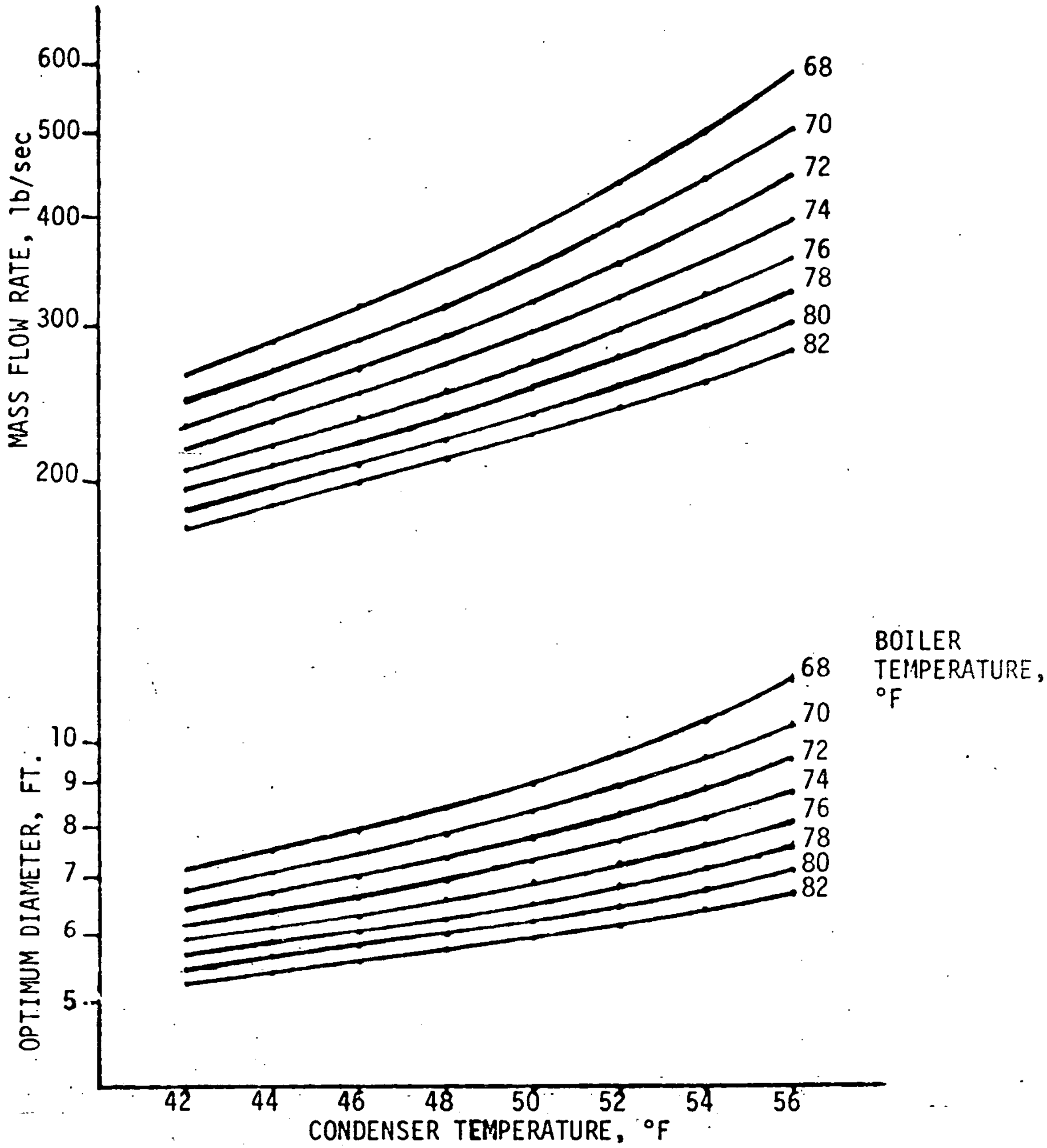

Figure 11 20 mw Propane Turbine Design Size

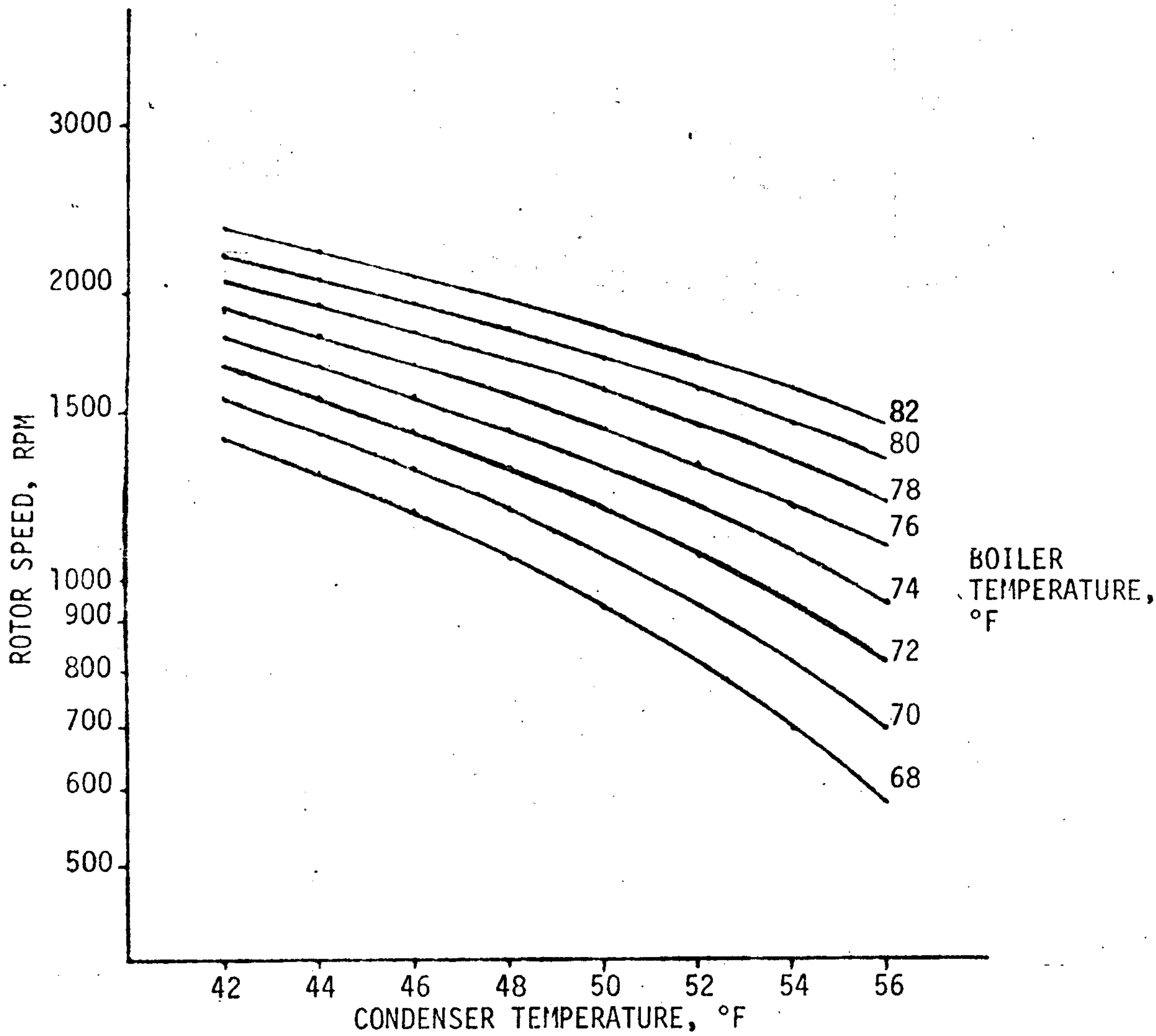

Figure 12 20 mw Propane Turbine Design Speed

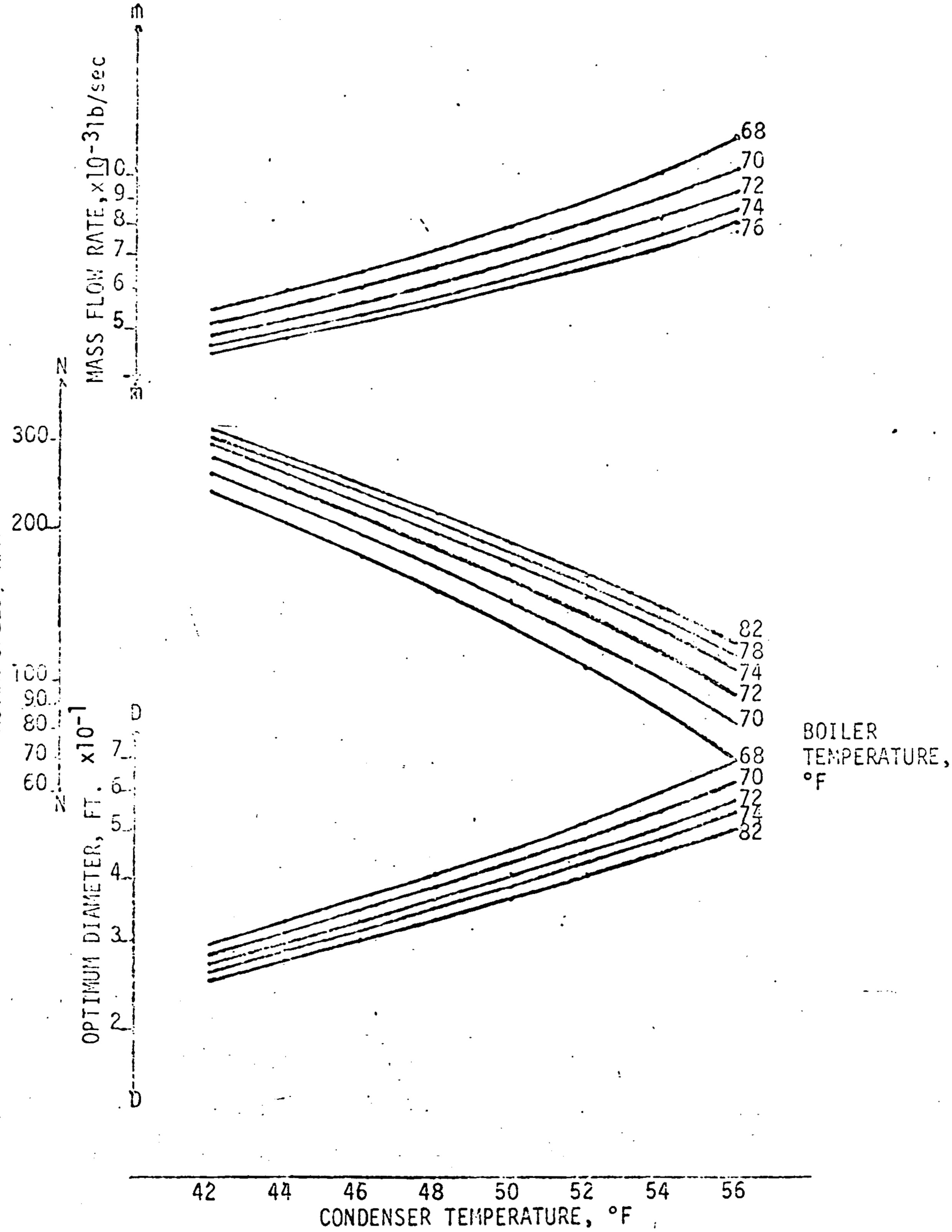

Figure 13 20 mw R12/31 Turbine Design

For purposes of comparison three turbine designs are compared in Table III, one for each working fluid. All are for operation with the same internal temperature difference of 16°F.

TABLE III

TURBINE DESIGN COMPARISON

T_B = 68°F T_C = 52°F

Ammonia	Propane	R 12/31
P = 30 mw	P = 20 mw	P = 20 mw
D = 8 ft.	D = 9.6 ft.	D = 52 ft.
N = 1800 RPM	N = 820 RPM	N = 105 RPM
$\dot{m}$ = 1950 lb/sec	$\dot{m}$ = 4400 lb/sec	$\dot{m}$ = 9000 lb/sec

In figures 14 to 18 turbine power and speed were fixed and turbine size and efficiency were calculated as a function of condenser and boiler temperature. As can be seen from equation 6 if power and speed are fixed then only condenser and boiler conditions need to be specified in order to determine specific speed and in turn efficiency. First it should be noted that the efficiency stays above 85% for all conditions considered. However when size is fixed the variation in efficiency will be greater. An efficiency above 90% is needed however and conditions which yield efficiencies above 90% are those having low condenser temperatures and high boiler temperatures. In fact it can be noted that large temperature differences reduce the variation in all design parameters. Figure 18 for R 12/31 shows that off design performance for this working fluid would be the poorest of the three candidate working fluids. Over the range considered turbine size varies by a factor of

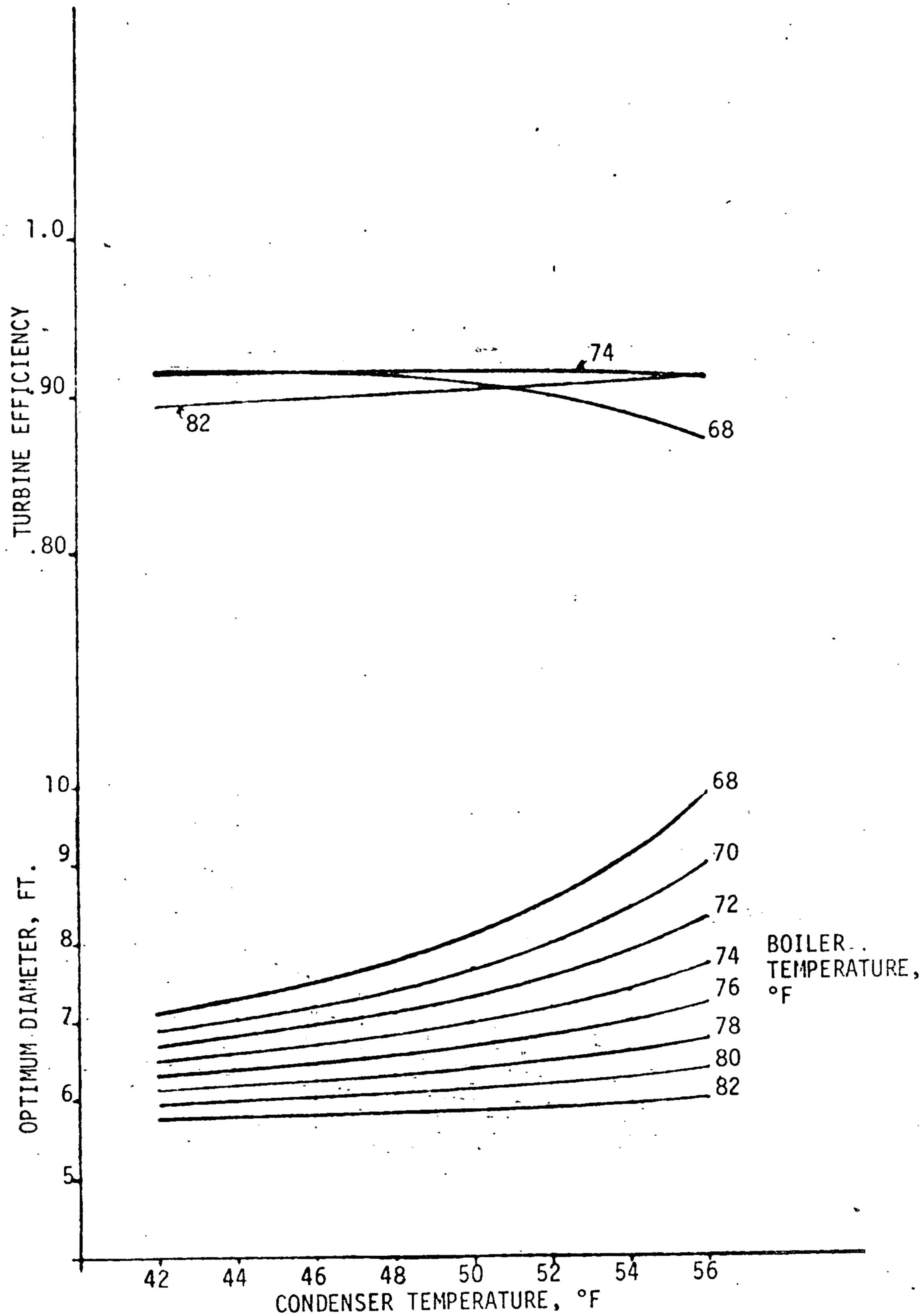

Figure 14 40 mw 2400RPM Ammonia Turbine Operating Range

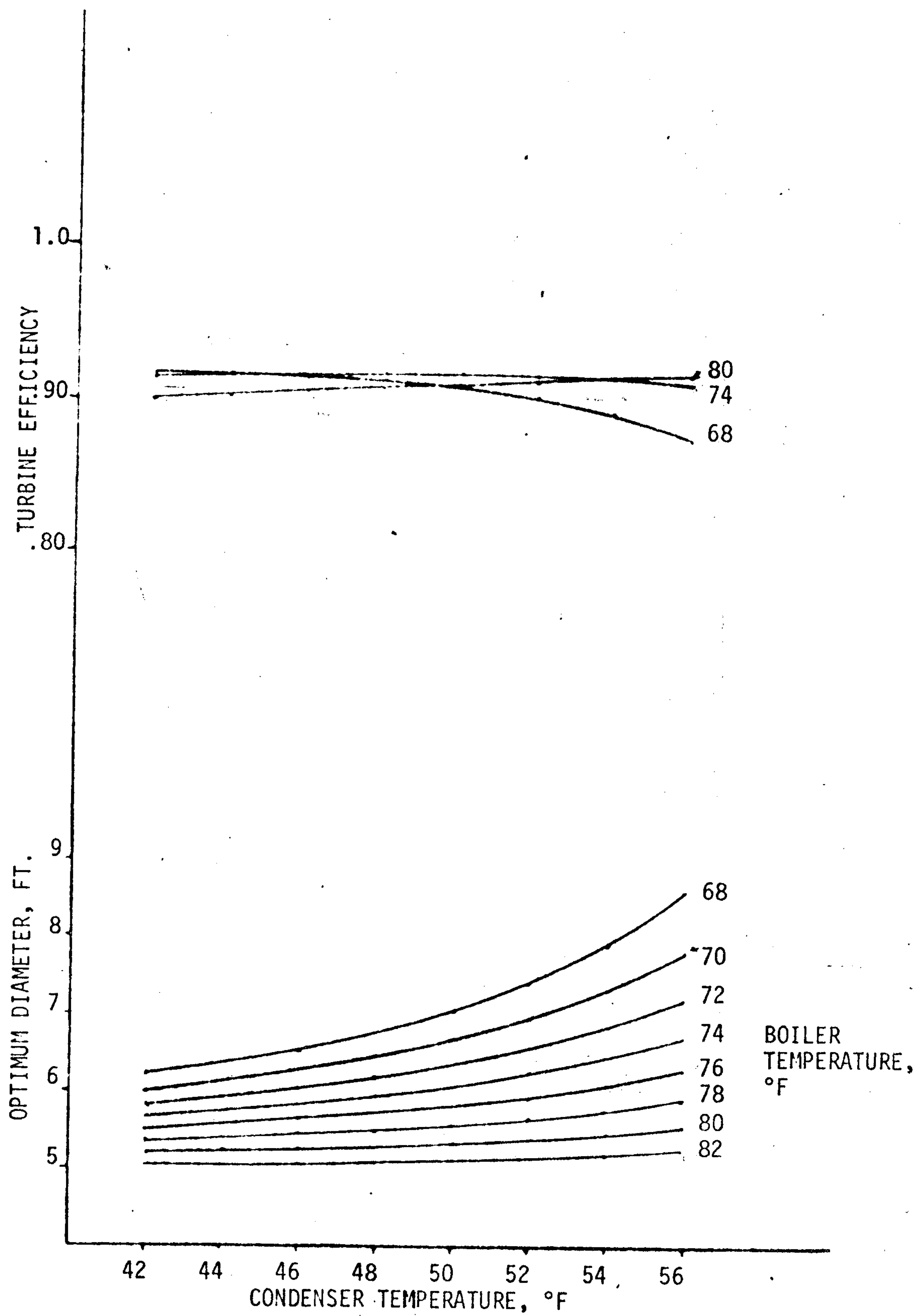

Figure 15 30 mw 2700 RPM Ammonia Turbine Operating Range

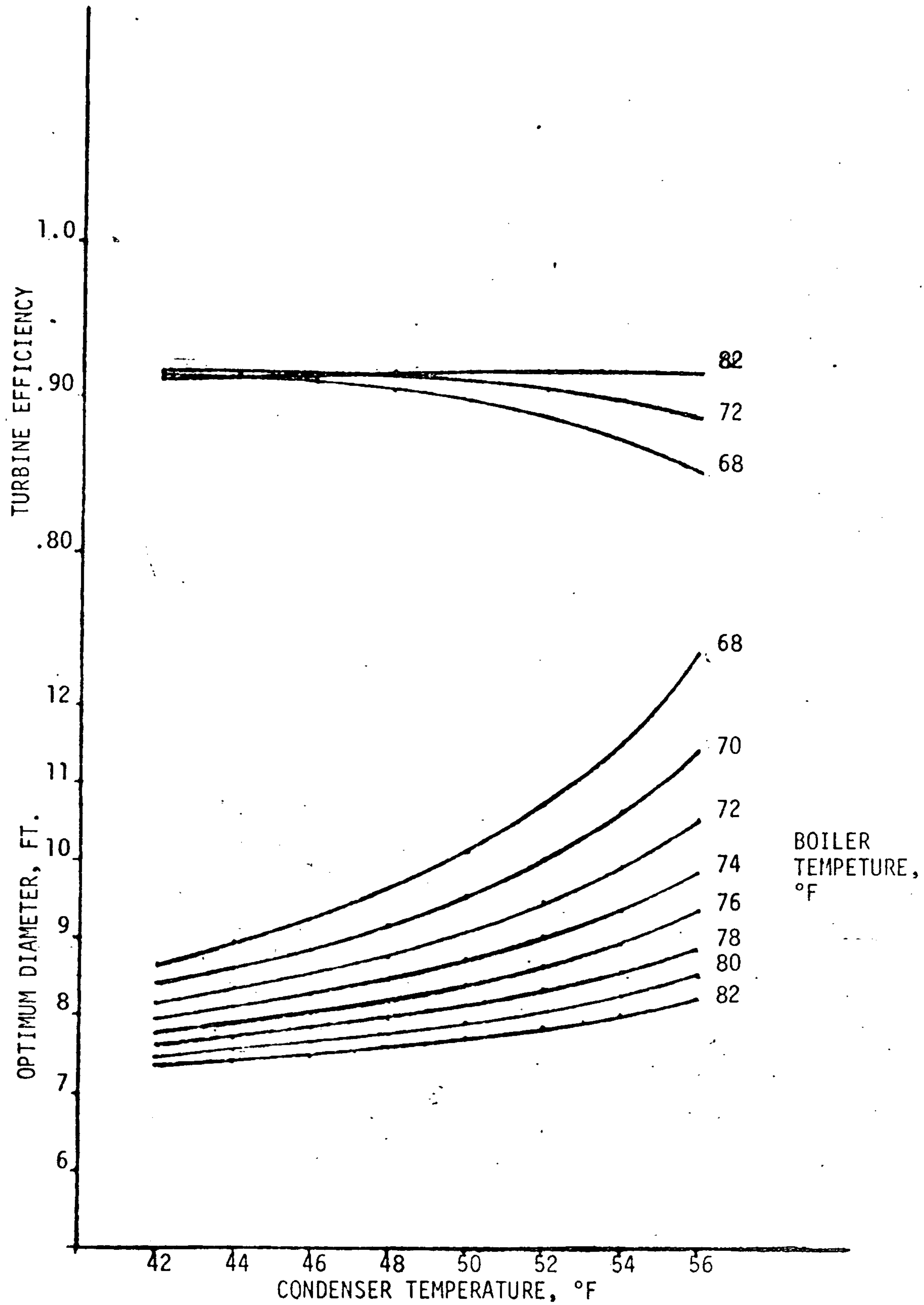

Figure 16 30 mw 1200 RPM Propane Turbine Operating Range

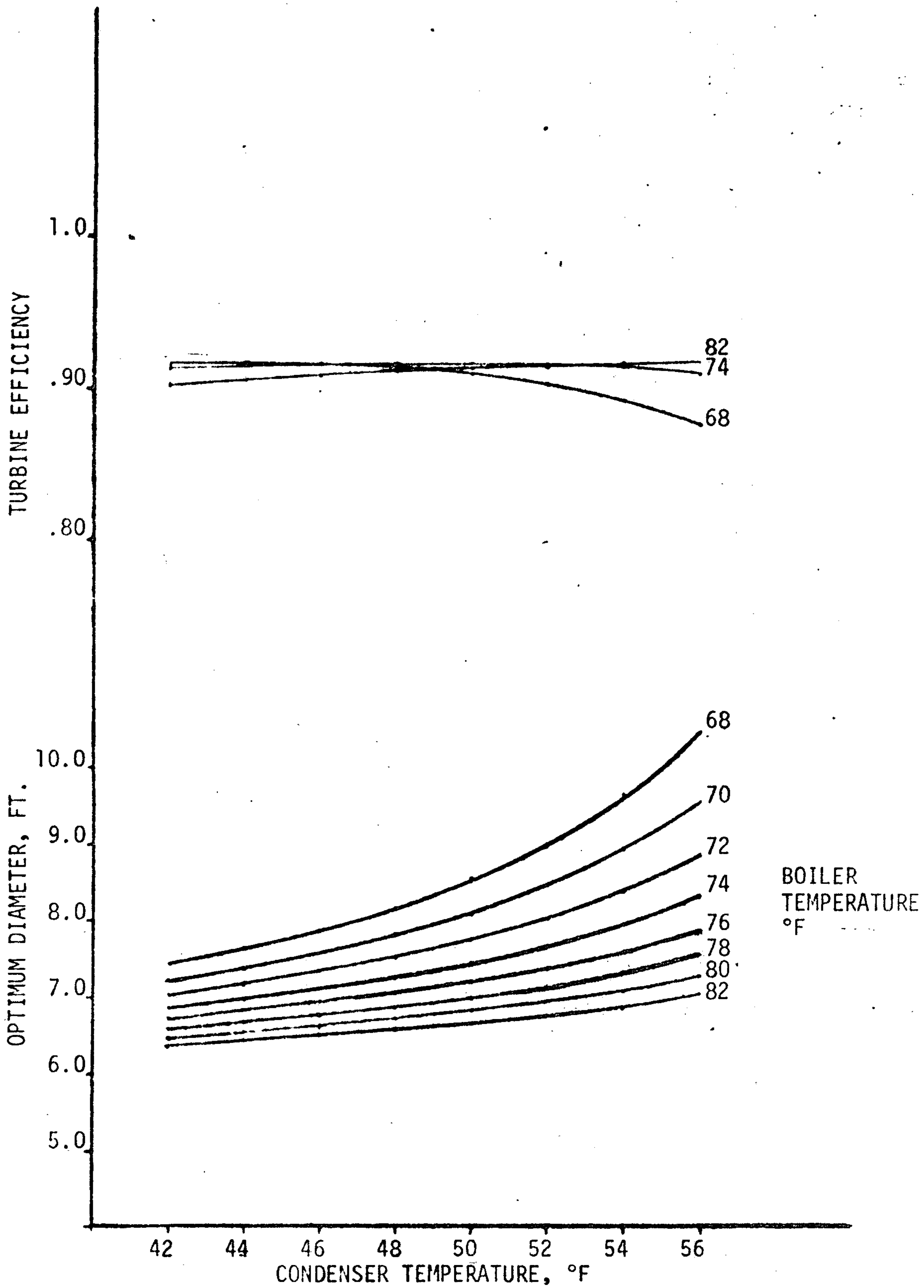

Figure 17 20 mw 1200 RPM Propane Turbine Operating Range

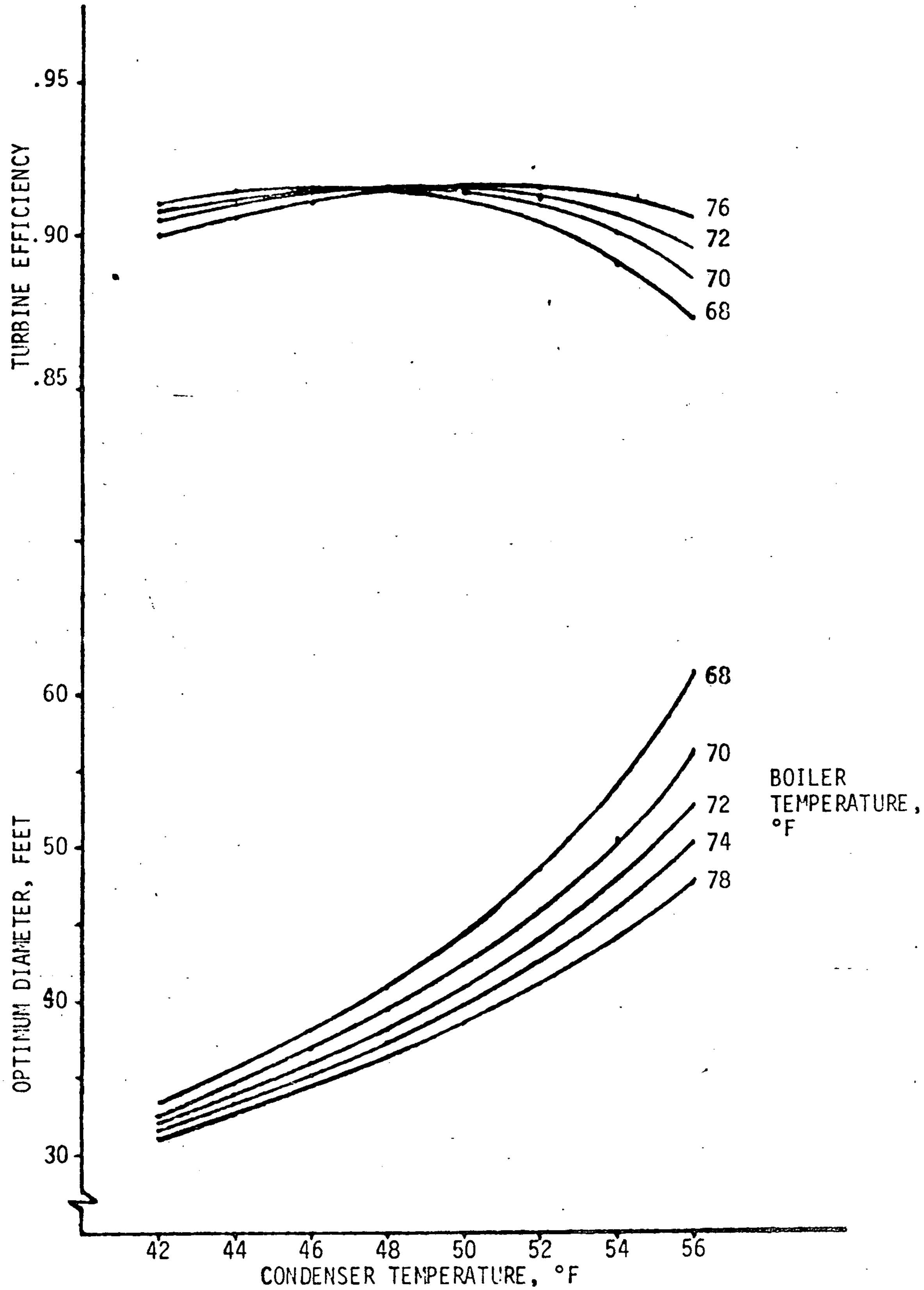

Figure 18 20 mw 150 RPM R12/31 Turbine Operating Range two and turbine efficiency would be expected to change more dramatically than is shown if size were fixed and site conditions varied as is shown. Figure 14 for ammonia and figure 16 for propane can be used to choose design size for these working fluid turbines at the 30 mw unit size.

As can be seen from the results presented here, the actual size of the turbine used for any of the fluids depends upon the site conditions and the heat exchanger design. Shape and other dimensionless ratios for the turbine are specified now. The design criteria used here have been incorporated into the heat exchanger model and are being used to obtain size information for the heat exchanger designs being considered.

Based on the fluid mechanical and thermodynamic performance of the turbine design, ammonia is the preferred working fluid. Although requiring a somewhat larger and slower unit of less power, propane would also be acceptable.

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Nomenclature

D = Diameter, ft.

D_s = Specific diameter

h = Blade height, ft.

H = Isentropic head for turbine, BTU/lb.

L = Latent heat of vaporization, BTU/lb.

M = Molecular weight

$\dot{m}$ = Mass flow rate, lb/sec.

N = Rotative speed, RPM

N_s = Specific speed

Q = Volumetric flow, cuft/sec.

P = Power level, mw

P_c = Condenser vapor pressure, psia

S = Blade tip clearance, ft.

v_g = Specific volume of vapor, cuft/lb.

η = Turbine efficiency

η_p = Power plant Carnot efficiency

α = Absolute flow angle, degrees

β = Relative flow angle, degrees

ρ = Degree of reaction

ψ = Flow factor