How to Select Turbomachinery

For Your Application

Barber Nichols

6325 West 55th Avenue
Arvada, CO 80002

Kenneth E. Nichols P.E.

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Barber-Nichols Inc.

Abstract:

Turbomachinery finds many applications in commercial, industrial, military, and aerospace applications. These turbomachines include pumps, compressors, fans and blowers, and turbines for prime movers. Other applications include turbines for generating shaft power. These turbines can be associated with closed Rankine or Brayton cycles for long duration power systems or open cycles where the driving fluid is expended to the atmosphere.

The paper describes a method of predicting turbomachinery performance using dimensionless parameters of specific speed and specific diameter. This approach provides the preliminary designer information as to the best type of machine for an application. In addition, it provides parameters to determine the size of the machine. It also aids in the selection of the optimum speed for a turbine or pump, combination turbopump applications, or turbine-compressor matching in a Brayton cycle power system.

Turbines that must operate over a range of power levels or with a changing exhaust pressure can be designed to have a broad performance range. Design techniques to improve the performance range of turbines are discussed.

The paper presents examples of turbomachinery that have been designed for space applications. These include the turbines for a 15 KW closed Rankine Cycle Solar Power System, a hydrogen-oxygen fueled open cycle power system for short duration manned space missions, and a turbine driven hydraulic pump.

Discussion:

Through the technique known as dimensional analysis, the similarity parameters specific speed N_S, specific diameter D_S, Reynolds number R_e*, and suction specific speed S, or Mach number M_a*, are derived which serve as convenient parameters for presenting the performance criteria of turbomachines. These four parameters are sufficient to describe completely the performance of geometrically similar turbomachines. For a given volume flow rate and a given head change through a turbomachine, specific speed is a number indicative of the rotative speed of the machine and specific diameter is a number indicative of the rotor diameter or size of the machine. Reynolds number expresses the ratio of inertia force to viscous force and reflects the properties of the fluid being pumped and the speed of the machine. Suction specific speed for machines such as pumps operating on non-compressible fluids will indicate whether or not cavitation exists. If cavitation does not exist, then pump performance will be as expected. If serious cavitation exists the pump performance can not be predicted from the similarity parameters. For machines operating on compressible fluids such as turbines and compressors, Mach number M_a*is used as the fourth similarity parameter in place of suction specific speed.

It is beyond the scope of this paper to go into detail concerning the derivation of the similarity parameters; however, as shown in reference 1, there are seven variables that completely describe the conditions of a turbomachine and three primary dimensions (length, mass and time). The π–Theorem, proposed by Buckingham in reference 2 serves as the basis for formal dimensional analysis and states that complete dynamic similarity can be specified by a number of π –terms equal to the number of variables minus the number of primary dimensions. The number of π –terms is therefore seven minus three or four.

The Seven Variables Are:

1.	Q	=	Rate of Volume Flow	$\frac{L^3}{T}$
2.	H	=	Head Change Through Machine	$\frac{L^2}{T^2}$
3.	N	=	Rotational Speed	$\frac{1}{T}$
4.	D	=	Rotor Diameter	L
5.	ρ	=	Fluid Density	$\frac{M}{L^3}$
6.	μ	=	Absolute Fluid Viscosity	$\frac{M}{L \cdot T}$
7.	H_S	=	Net Positive Suction Head	$\frac{L^2}{T^2}$
	or			
	a*	=	Speed of Sound in Fluid	$\frac{L}{T}$

Dynamic similarity presupposes that geometric similarity also exists. For example, Reynolds number effects, as in pipes are a function of the relative roughness of the flow passage, expressed in dimensionless form as the ratio of roughness height to passage diameter. Therefore turbomachines that operate at the same specific sped N_S, specific diameter D_S, Reynolds number R_e*, etc., must be geometrically similar to have the same performance. Since it is not practical to consider al possible geometries some restriction is needed. Of all possible geometries, one configuration will yield an optimum value of performance or highest efficiency, and this optimum geometry or design criterion is a function of N_S, D_S, etc. Now, with the restriction that only optimum geometries are considered, a graphical representation of turbomachine performance as a function of N_S, D_S, R_e*, and H_S or M_a*can be generated.

It is difficult to present the performance of any machine as a function of four parameters at one time. Fortunately, two of these variables, namely Reynold's number R_e* and Mach number M_a* have only a secondary effect on turbomachine performance; and more significantly, if the Reynolds number is above 10^6 for turbines and compressors or above 10^7 for pumps the effect of Reynolds number is very nearly constant which eliminates this variable.

If the Mach number of the machine is less than or near 1.0 the compress-ibility effects are negligible which eliminates this variable, and turbomachine performance can be presented as a function of two parameters, specific speed, N_S and specific diameter, D_S. For pumps, if the suction specific speed, S, is below the critical value then operation will not be affected and again performance can be presented as a function of N_S and D_S. This method of presenting turbomachinery performance as a function of the similarity parameters as described herein is discussed in detail in references 1, 3 and 4. The results of that work and their application form part of the subject of this paper.

First, in order to better understand the basis of presenting turbomachinery performance as a function of the similarity parameters, an example and discussion of a performance plot for turbines will be covered. The correlating similarity parameters selected as discussed previously are the specific speed, N_S, and specific diameter, D_{S*}

Where: $$N_s = \frac{N \cdot Q_3^{1/2}}{H_{ad}^{3/4}}$$

And $$D_s = \frac{D \cdot H_{ad}^{1/4}}{Q_3^{1/2}}$$

N = Rotational Speed (rpm)
Q_3 = Rotor Flow Rate (ft^3/sec)
H_{ad} = Adiabatic Head (ft)
D = Diameter (ft)

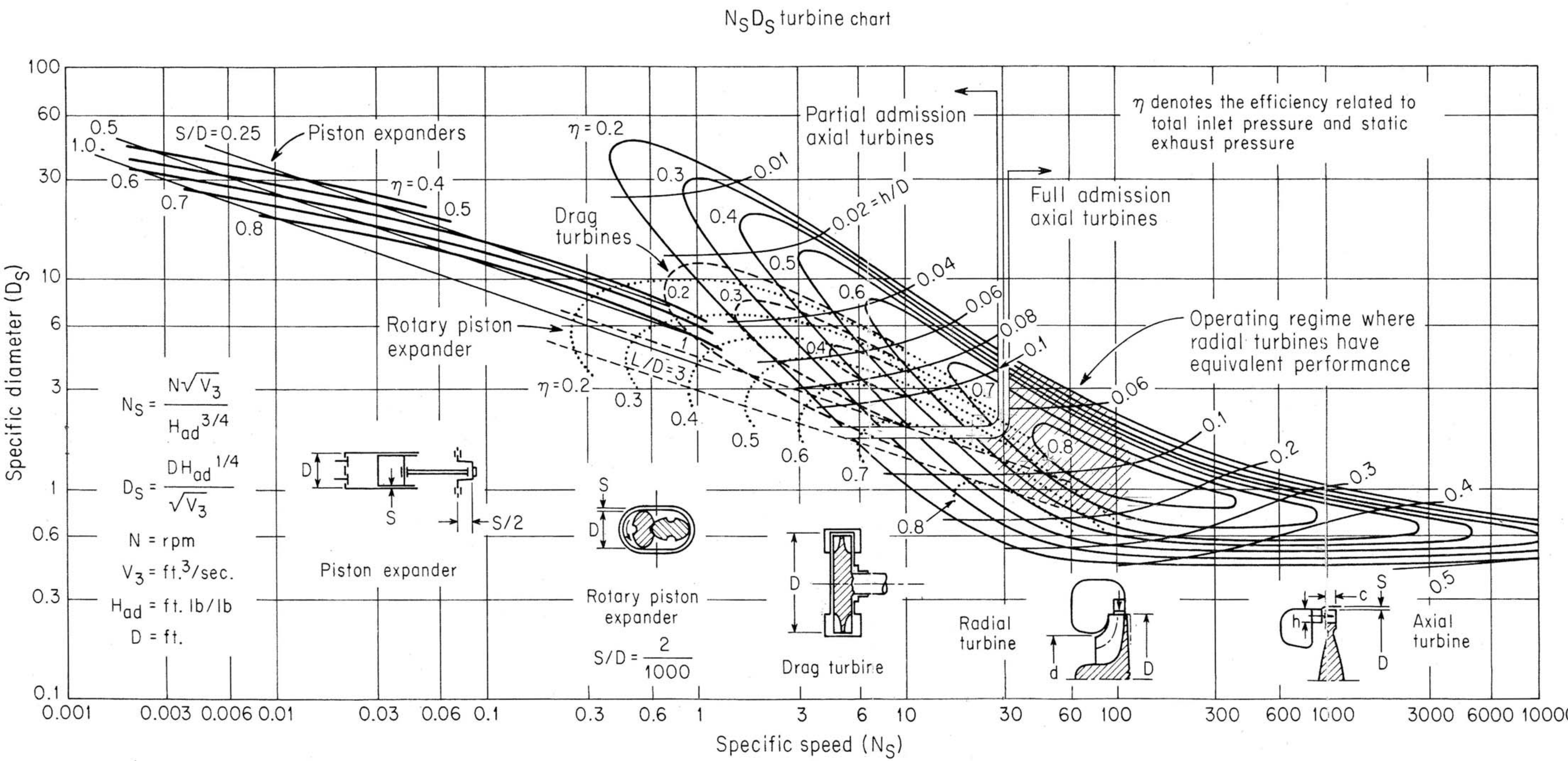
It may be noted that the specific speed and specific diameter are not dimensionless in the form presented above; however, the parameters are truly dimensionless when reduced to a form using angular velocity. An example of a typical specific speed, specific diameter correlation for full admission turbines is presented in Figure 1. This figure is calculated for a Reynolds number greater than 10^6, a rotor inlet Mach number of 1.0, a side and tip clearance to blade height ratio of 0.02, and a nozzle angle of 16 degrees. A good approximation of the turbine efficiency may be obtained by iteration using an applicable curve such as Figure 1. As may be noted in Figure 1, the turbine efficiency may be greater than 80 percent at specific speeds greater than 40.

The performance of turbines as a function of N_S and D_S is essentially a solution of the theoretical hydraulic efficiency equation and the calculation of the parasitic losses. Included in the analysis are the effects of Reynolds number and Mach number, and the losses due to disc friction, blade pumping, partial admission, incidence, rotor scavenging, leakage, and rotor blade profile. These effects are generally considered separately in the analysis and are described below.

The Turbine Hydraulic Efficiency as Derived From the Momentum Equation is:

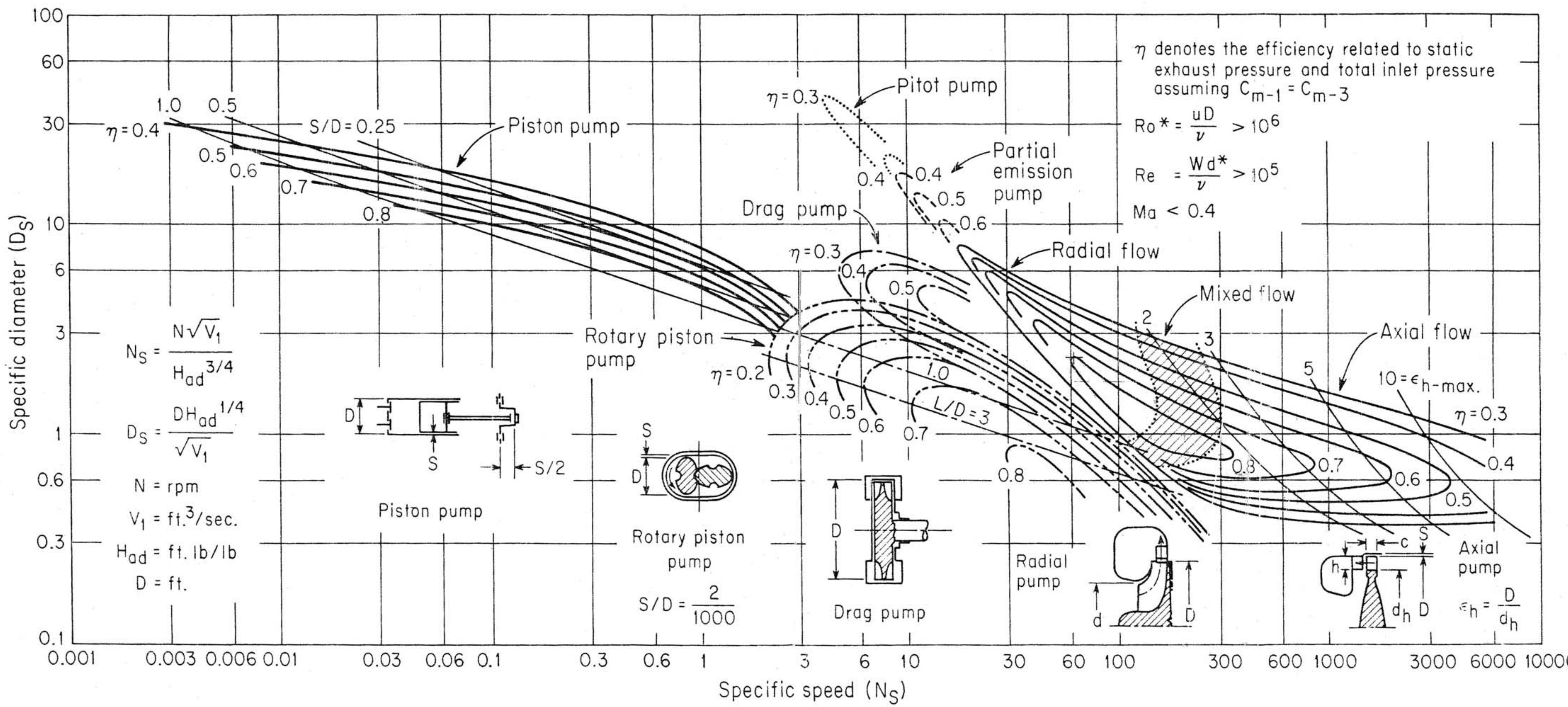
$$\eta_H = \frac{2 \cdot U}{C_0}\left[\psi_N \cdot \sqrt{1-\rho} \cdot \cos\alpha - \frac{U}{C_0} + \psi_R \cdot \cos\beta_3 \cdot \sqrt{\rho + \psi_N^2 \cdot (1-\rho) - \frac{2 \cdot U}{C_0} \cdot \psi_N \cdot \sqrt{1-\rho} \cdot \cos\alpha + \frac{U^2}{C_0^2}}\right]$$

Where: U = Rotor Tip Speed (ft/sec)
C_0 = Isentropic Spouting Velocity (ft/sec)
Ψ_N = Nozzle Velocity Coefficient
Ψ_R = Rotor Velocity Coefficient
ρ = Reaction Fraction
α = Nozzle Angle
β_3 = Rotor Exit Blade Angle



$N_S D_S$ turbine chart
Specific diameter (D_S)
Specific speed (N_S)
Piston expanders
S/D=0.25
Partial admission axial turbines
Full admission axial turbines
η denotes the efficiency related to total inlet pressure and static exhaust pressure
Drag turbines
Rotary piston expander
Operating regime where radial turbines have equivalent performance
0.02=h/D
L/D=3
$N_S = \frac{N\sqrt{V_3}}{H_{ad}^{3/4}}$
$D_S = \frac{D H_{ad}^{1/4}}{\sqrt{V_3}}$
N = rpm
V_3 = ft.^{3}/sec.
H_{ad} = ft. lb/lb
D = ft.
Piston expander
Rotary piston expander
S/D = $\frac{2}{1000}$
Drag turbine
Radial turbine
Axial turbine

$N_S D_S$ pump chart



The nozzle velocity coefficient is defined as the ratio of nozzle exit velocity to isentropic exit velocity and generally has a value of approximately 0.96 at the design point. When convergent-divergent nozzles are operating at pressure ratios other than the design point the nozzle coefficient decreases. The magnitude of this decrease becomes greater with increasing design Mach numbers.

The rotor velocity coefficient is defined as the ratio of the rotor exit velocity to isentropic rotor exit velocity. The rotor coefficient includes empirical correlations for several losses including those for partial admission, Mach number, blade turning and profile loss, and aspect ratio. The rotor coefficient is also modified to include incidence losses.

As with all aerodynamic machines, Reynolds number effects are significant in turbomachinery.

The Reynolds Number for Turbines is Defined as:

$$\mathrm{Re} = \frac{D_h \cdot W_2 \cdot \rho_2 \cdot 3600}{\mu_2}$$

Where: D_h = Blade Passage Hydraulic Diameter (ft)
W_2 = Blade Inlet Relative Velocity (ft/sec)
ρ_2 = Blade Inlet Density (lb/ft^3)
μ_2 = Blade Inlet Viscosity (lb/ft-hr)

The turbine efficiency decreases rapidly at Reynolds numbers less than 5000. Corrections for Reynolds numbers effect is included in the analysis.

Rotor frictional losses may be a large percentage of the available power. These friction losses include disc friction and blade pumping losses, (if the turbine is of partial admission design). The friction of a rotating disc has been measured by a number of investigators. The disc side clearance has been shown to have a large effect on the friction loss, decreasing the loss from the value of an unshrouded wheel by a factor of as much as 4 for a closely shrouded wheel.

The loss associated with turbine blade pumping is caused by the “paddle wheel” effect of the blades in a partial admission turbine. This loss only occurs for blades which are in the portion of the wheel which is not opposite a nozzle.

Another loss associated with partial admission turbines is the blade scavenging loss. This loss occurs because of the flow work required to accelerate the “dead” gas trapped between the blades when the blade passage first enters the nozzle flow region.

Another loss to be considered is the effect of leakage. In a turbine flow will leak from the turbine rotor inlet and exit seals. This static leakage will go into the region around the disc and ultimately into the exhaust port. Generally, this leakage flow is not available to do work. Other static leakage occurs over the blade tips which essentially reduces the mass flow available to do work. Another leakage that occurs in single disc multi-stage turbines is dynamic leakage which results from the flow trapped in the volume between blades and carried out of the flow system with wheel rotation.

From the equation for N_S and D_S it can be shown that U/Co is a function of these parameters.

$$\frac{N_s \cdot D_s}{154} = \frac{U}{C_0}$$

This expression can be substituted into the hydraulic efficiency equation in place of U/Co and efficiency can be calculated as a function of N_S and D_S. The parasitic losses previously discussed can also be derived as functions of N_S and D_S and the overall net efficiency is then determined as a function of N_S and D_S as shown in Figure 1.

The performance of pumps and compressors as a function of N_S and D_S is generated in exactly the same manner except the detailed equations of energy transfer and parasitic losses are of course different. They can, however, be expressed in terms of N_S and D_S. A performance plot for pumps and compressors is shown in Figure 2.

N_S – D_S diagrams are used for determining the performance of a turbomachine for a specific application as follows. If a certain efficiency level of operation is desired for a turbine, pump or compressor, the head across the machine is calculated. For a turbine this head is the total isentropic head drop available and for pumps or compressors it is the isentropic head rise associated with the desired pressure ratio. The volume flow is then determined. For pumps or compressors the volume flow is the inlet flow and the state conditions are usually know. For turbines the volume flow is at the exit and a turbine efficiency is assumed in order to determine exhaust specific volume and then total volume flow. Now since it was assumed that a certain minimum efficiency was desired one can determine on the N_S – D_S diagram the lowest value of N_S that provides this efficiency. Substitute this value into the N_S equation and find the required rotative speed. For most applications the rotative speed will already be determined by such considerations as stress levels, maximum bearing or gear speeds or limiting pump speeds as determined by cavitation criteria. When shaft speed is known the N_S can be calculated and the performance potential determined. Since this procedure is much simpler than evaluating the entire detailed equations that govern turbo-machinery performance much time can be saved during a preliminary design phase when selecting or matching turbomachinery components. The D_S value from the N_S – D_S diagram will allow the determination of the machine rotor diameter which of course is indicative of the size of the unit.

Turbines for certain applications may operate over a range of power levels or with a varying backpressure or with both of the variables. The effect of this is that the overall pressure ratio across the machine varies over a wide range. One recent turbine application analyzed and presently under development has to operate over a pressure ratio range from 5 to above 1200. This change in pressure ratio occurs partially across the nozzle and partially across the rotor of a turbine and, of course, neither of these components are operating at their design point. However, by proper design, a nozzle configuration can be selected that offers a broad operating range. An example of this is the use of a plug nozzle in an impulse turbine. Considerable work has been done in the area of evaluating off design performance and is reported in Reference 5. As shown on Reference 5, the performance of the nozzle of an impulse turbine is more influential on overall performance than any other component. The measure of nozzle performance is the velocity coefficient which indicates what percentage of the ideal isentropic velocity is being realized from the nozzle. The off design performance of a nozzle can be presented as shown in Figure 3. The actual nozzle velocity coefficient is normalized to the design point velocity coefficient. The operating pressure ratio is shown in the figure as a ratio of isentropic Mach number to the design Mach number. Also, nozzles may be optimized for off-design performance and do not degrade as much as other nozzles at reduced pressure ratio operation. However, the peak performance at design may not be quite as high; therefore, the application profile must be analyzed in order to select the turbine design with the best operating economy over the entire range of variable conditions.

Turbomachines have been designed and developed for commercial, industrial, military, and aerospace applications. Examples of turbomachines for aerospace applications include turbine-driven electrical and hydraulic power systems, turbopumps for propellant transfer and pressure boosting, and turbo-machinery for long duration space power systems where the raw energy is derived from the sun or man-made nuclear devices.

The Model 876 Accessory Power Unit is a cryogenic-fueled turbine-driven power system. The unit was developed by Sundstrand for the Aeronautical Systems division, USAF Systems Command for use of the X-20 (Dyna-Soar) space vehicle.

The Model 876 is powered by cryogenic hydrogen and oxygen propellants and delivers approximately 37 HP of combined electrical and hydraulic power. The Model 876 APU, shown in Figures 4 and 5, is designed to start and operate on hydrogen from the -423ºF liquid state to the -250ºF gaseous state and with oxygen ranging from -185ºF to -32ºF. Oxygen pressures are maintained from 441 to 2300 psia and can be in either a one- or two-phase condition.

The design incorporates an electronic controller for speed and temperature control and auxiliary logic functions. Flow modulation of the propellants is achieved by individual shutoff and flow control valves which incorporate a jet-pipe torque motor stage to drive a floating-poppet power metering stage. Temperature conditioning of the propellants is accomplished by sensing hydrogen temperature upstream from the hydrogen servo valve and bypassing the proper amount through a regenerator in the turbine exhaust, then through the hydrogen side of an oxygen preheater.

The combustion reaction between the propellants is initiated and maintained catalytically using a combustor and catalyst.

The prime mover section, designed for 250-hour life and a 1,500ºF inlet temperature, incorporates a three-stage single-disc re-entry turbine using an integral Rene' 41 turbine wheel. The ultimate in a fail-safe requirement is satisfied by a unique shaft brake which prevents overspeed destruction of the turbine wheel should a multiple control failure occur.

Figure 6 shows the rotating assembly (combined power unit) of a 14 KW Rankine cycle liquid metal solar power conversion system. The rotating assembly in Figure 6 is in a special fixture used during dynamic balancing of the unit. The combined power unit is an energy conversion machine consisting of a fluid pump, turbine, and an electromagnetic alternator. These components are mounted on a single shaft with two bearings, utilizing rubidium as the working fluid. Liquid rubidium is also used to cool the alternator. The purpose of this machine is to convert the thermodynamic energy of superheated rubidium vapor into 3,200 cps three phase AC power at 120/208 volts. It is designed to operate at 24,000 rpm with a power output of 15 KW.

The turbine of the combined power unit consists of two high pressure turbine stages and two low pressure stages. All four are axial, full admission stages. The first stage of the high pressure turbine is an impulse stage and the second stage is 60% reaction. Inlet conditions are 10 psia and 1,750ºF. The first stage o the lower pressure turbine is 70% reaction and the second stage is 50% reaction. Inlet conditions are 1.2 psia and 1,600ºF. There is reheat between the high and low pressure turbines and the overall cycle efficiency including turbine efficiency is 24%. Figure 7 shows the fourth stage turbine of the rotating assembly. The blades and rotor disks are made integrally.

Figure 8 shows a high speed turbopump used to develop 2500 psi hydraulic pressure. Turbine driven units such as this can be used to provide hydraulic power for short duration demands.

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References:

1. O.E. Balje, K. E. Nichols, and D. G. McPherson, "Study of Turbine and Turbopump Design Parameters," final report. Volume 4, "Low Specific Speed Turbopump Study". S/TD No. 1735, Department of the Navy, Office of Naval Research, Contract No. NONR 2292(C) Task No. NR 094-343.

2. E. Buckingham, "On Model Experiments and the Form of Empirical Equation". Trans. ASME, Volume 37, P 263 1915.

3. O. E. Balje, "A Study on Design Criteria and Matching of Turbomachines:" Part A "Similarity Relatives and Design Criteria of Turbines", Trans. ASME Series A, January 1962.

4. O. E. Balje, "A Study on Design Criteria and Matching of Turbomachines" Part B "Compressor and Pump Performance and Matching of Turbocomponents". Trans. ASME, Series A, January 1962.

5. R. E. Barber, M. J. Schultheiss, "Effect of Nozzle Geometry on Off-Design performance of Partial Admission Impulse Turbines". Sundstrand. Report AER No. 486 April 1967 NONR Contract No. N00014-66-C0204.