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THEORY OF RATING

CONTENTS

Theory of Rating

DENSITY AND DENSITY RATIO

In order to select a fan for a given performance, it is first necessary to know the density ratio of the air or other gas being handled—that is, the ratio of the actual density to the density of "standard air".

Standard air is air at a barometric pressure of 29.92" Hg. This is the pressure of dry, 70° F air at sea level. Standard air has a density of .075 lb/ft^3. Therefore,

Density Ratio = Density of air being handled/.075

where density is in pounds per cubic foot. Density as used in some formulas on the following sheets must be expressed in slugs per cubic foot and is obtained by dividing the pounds per cubic foot by the acceleration of gravity (32.2 feet per second per second). Density in slugs per cubic foot for standard air is approximately 0.00232. Do not confuse Density Ratio with Density.

Air density is determined by several factors, which are discussed below.

Elevation

Increasing elevation will decrease air density to the extent that at an altitude of 18,000 feet, the weight per cubic foot of air is just one half that of the weight of a cubic foot of air at sea level under the same conditions. By the same token, a given number of pounds of air would be doubled in volume.

Barometric Pressure

The density of air at a given elevation is, of course, subject to variations in barometric pressure which vary with density. This effect is usually ignored in fan selection and the normal, or standard, barometric pressure is assumed. Accurate barometric pressure readings should, of course, be used in fan and blower testing.

Temperature

The density of air will vary with changes in temperature. Standard air assumes a 70° F temperature under sea level conditions. Increasing or decreasing the temperature will result in a corresponding decrease or increase in density. Air at a temperature of 600° F has a density of one half that of standard air. Temperature used in fan selection must, in all cases, be the temperature of the air being handled by the fan.

Relative Humidity

The relative humidity or moisture content of the air being handled has an effect upon the density. This effect, however, is so small at temperatures below 100° F that its effect upon density ratio may be entirely disregarded.

Definitions and Formulas

Density Ratio $\boxed{\rho}$

ρ = The ratio of the density of the air at the fan to the density of dry air at sea level (29.92"Hg)

$$\rho = \frac{e^{-3.7x10^{-5}A} \; x \; 530}{T + 460}$$

Where:
A = Elevation in ft
T = Temperature in °F

Supply or Exhaust Units

A common inquiry is that of the difference in performance between a fan blowing into a system in which static pressure is present (e.g. a forced draft application) and a fan exhausting from such a system (e.g. an induced draft application). It is true that air is thinned to a lower density when being drawn into the fan and compressed to a higher density when being pushed by the fan. This difference is negligible in fan performance up to several inches of water. Atmospheric pressure itself is in the neighborhood of 400 inches of water pressure, so that a four-inch static pressure, plus or minus, would result in a change of only 1% in the density, which is negligible.

VOLUME IN CUBIC FEET PER MINUTE

Volume $\boxed{Q}$

$$(ACFM) = \frac{(SCFM)}{\rho}$$

(SCFM) = mass flow rate of air through fan in standardcubic feet per minute.

(ACFM) = volume flow rate of air through fan in actual cubic feet per minute.

In determining the volume of air the blower is to handle, the density of the air must again be taken into consideration. Most charts and tables for fan selection are based on standard air of 70° F at sea level.

If the fan is to handle a given number of pounds of air or a given CFM of standard air (SCFM) under conditions other than 70° and sea level, it is necessary first to determine the actual CFM (ACFM) to be handled by the fan *at* the fan. For example, a fan may be drawing air through a radiator or finned tube section. So many pounds or so many cubic feet of standard air are required to pass through the section. In passing through the section, however, the air will take on heat, increasing the CFM which must be handled by the fan. The reverse would be true if the fan were drawing air through a refrigerator coil. In any event, the volume the fan itself is handling must be determined. If CFMis given in standard air, it must be divided by the density ratio at the fan to determine the actual CFM being handled by the fan.

Inquiries and orders should always clearly state whether the volume required is given in terms of standard air or in air at the density being handled by the fan. To be specific, the proper term for standard air volume (SCFM) or actual volume (ACFM) should be used. In the following text, the term CFM, or cubic feet per minute, should, in all cases, be construed as the actual cubic feet per minute being handled by the fan.

VELOCITY

Velocity $\boxed{V}$

V = Axial air velocity through fan in feet per minute.

$$V = \frac{1.27(ACFM)}{D_R^2 - [S/12]^2}$$

Where:

D_R = Diamter of fan ring, in feet.

S = Series of fan = nominal air seal (the aerodynamic "hub") diameter in inches

Velocity, as discussed in t:he following pages, is defined as the average velocity of the air moving in an axial direction through the net area of the blower. The fan rings or throats in which the wheels operate must, of course, be slightly larger than the wheels in order to provide tip clearance. This clearance may vary from 1/4" on the diameter on small units to 1-1/4" on large units. It is this ring or throat diameter that is considered in computing the net area of the blower.

Velocity = A C F M / (Throat Area - Hub Area)

Velocities are given in feet per minute (FPM).

As stated before, the velocity considered is that of air moving in an axial direction. If the air does not leave the fan axially, as in the case of all fan wheels without guide vanes, the term velocity will mean the axial component of the true velocity. (The difference between the squares of the axial component and the total velocity represent wasted power. See "Deflection" on Page 6.)

It should be noted that in some computations, velocity must be expressed in feet per second.

If the unit to be rated is equipped with one or more diffusers on the discharge side of the wheel, the velocity used in obtaining pitch should be that of the air through the blower. However, the velocity pressure to be used in determining horsepower requirements should be the velocity at the discharge of the last diffuser, corrected for diffuser losses.

VELOCITY PRESSURE

Velocity pressure, as it pertains to fans, is the impact pressure of a moving air stream relative to a fixed object. It is necessary, of course, that work be done by the fan to accelerate the air to the velocity required in order to pass a given volume of air in a given time. Velocity pressure, unless otherwise stipulated, refers to the discharge velocity pressure from the fan or, when diffuser sections are used, from the last diffuser section.

Except in the case of circulating fans, velocity pressure is wasted, since the power exerted in bringing the air to the velocity of discharge serves no useful purpose, the velocity being dissipated into the free air surrounding the discharge zone.

Velocity pressure may be measured by a Pitot static tube or, if the velocity is known, may be calculated by formula. From the formula, it will be noted that velocity pressure increases as the square of the velocity and rises quite rapidly at velocities above 2000 feet per minute. It will be noted also that velocity pressure increases in direct proportion to the density ratio.

In the event that diffuser sections are to be installed on the fan, the velocity pressure used in rating should be that of the discharge of the last diffuser section, with a 10% addition for diffuser losses. This velocity may be calculated by dividing the CFM by the area of the diffuser and the velocity pressure for that velocity calculated.

STATIC PRESSURE

Static pressure, as the name implies, is the fixed resistance against which a given volume of air must be delivered. It may be considered the same as the pressure head against which all pumping equipment is rated. For instance, a pump may be selected to raise water to a certain elevation. In addition to the head imposed by the elevation, there is always a certain amount of friction head which also must be considered in arriving at the static pressure which the pump must exert. The velocity of the liquid flowing from the discharge of the pipe is neglected.

In the case of a fan, static pressure may be that offered by passing a given volume of air through a filter, a radiator section, a run of duct work, a mine shaft, or other restrictions in the air stream. It may be either negative (below atmospheric pressure) as in the case of a fan drawing air through a resistance, or positive as in the case of a fan pushing air through a resistance.

Velocity Pressure $\boxed{P_V}$

P_v = The impact pressure of the moving air stream relative to a fixed object, in inches of H_2O

$$P_v = 6.2 \times 10^{-8} \rho V^2$$

Static Pressure $\boxed{P_S}$

P_s = The fixed resistance against which a given volume of air must be delivered, measured in inches of H_2O

Total Pressure $\boxed{P_T}$

P_t = The entire pressure against which the fan must operate in inches of H_2O

$$P_t = e P_v + P_s$$

Where: e=inlet factor (usually 1). See "Inlet conditions" in Section 4 System Design

TOTAL PRESSURE

Total pressure is the entire pressure against which the fan must operate and is calculated by adding the velocity pressure to the static pressure. Total pressure determines the amount of blade surface or number of blades in the fan. In conjunction with the CFM and efficiency, it determines the amount of horsepower required. Total pressure, multiplied by the area of the fan, including the hub, determines the thrust of the air upon the wheel and, when the weight of the fan is added, determines the thrust load on the fan shaft bearing.

PRESSURE PER BLADE AND REVOLUTIONS PER MINUTE

Pressure per blade is the term selected to denote the total pressure which one blade of a given series of fan will exert on the air being handled (or will contribute toward total pressure) when rotated a given number of revolutions per minute (RPM). This figure is relatively independent of the fan pitch and is not only theoretically accurate but also has been substantiated by actual test. For example, if one blade at a given RPM will exert a pressure of 0.1 inch, three blades under similar conditions will exert a pressure of 0.3 inches, and so on, until certain limits are reached. Pressure per blade varies in proportion to the square of the speed and directly as the density ratio.

The term "pressure per blade" indicates the pressure one blade will exert over the entire blower area and is not indicative of the pressure on the blade itself. Pressures per blade by the formula given here apply only to Moore Class 5000 fans and are not applicable to units of other design.

NUMBER OF BLADES

The maximum number of blades that can be installed on the various series of Moore fans is shown in the table at right. This maximum is determined by both aerodynamic and physical considerations. From the standpoint of aerodynamic design, blades must be spaced by a sufficient distance so that each blade will have enough air to absorb the blade's potential pressure capability. If blades are installed too close to one another, the air between them is insufficient to develop this potential. This condition, of course, occurs first at the hub where the blades are in the closest proximity.

In fans of fairly large diameter with respect to the hub diameter (D/d), additional blades may be installed to develop a higher total pressure potential, although at some sacrifice near the hub, due to the fact that the distance between blades increases as we move farther out from the hub. Physical conditions may, however, limit the number of blades possible.

In units with stubby blades-that is, for fan wheels having a ratio of fan diameter to hub diameter (D/d) of less than 2.25—additional blades may be required. The correction factor to the number of blades is shown.

The theoretical number of blades (Nt), usually a fractional number, is used in determining deflection. The next larger integral number of blades will actually be used on the unit.

Pressure Per Blade P_B

The total pressure that one blade of a given fan will produce at a given RPM

$$P_b = 3.80 \times 10^{-8} \rho S (RPM)^2$$

Velocity Pressure N

N_t *= The theoretical (fractional) number of blades required*

D/d = The ratio of the fan diameter to the aerodynamic hub (air seal) diameter

$$N_t = \frac{MP_t}{P_b}$$

M= 1 where D/d > 2.25

M=2.05 -(1/2 D/d) where D/d <=2.25

M=1 where 2.05 - (1/2 D/d) <1

N_{MAX} *= The maximum number of blades that can be installed on a given series of fan*

Manually Adjustable Fan		Automatic Fans	
19	4	19	n.a.
27	6	27	n.a.
33	7	33	6
40	8	40	8
49	10	49	10
60	12	60	12
73	15	73	12

PITCH

The pitch of a fan is defined as the advance of the air stream, in an axial direction, per revolution of the fan wheel. In a properly designed fan, air at the tips of the blades as well as air at the hub will theoretically advance a given distance for each revolution of the wheel. In order to simplify calculations, pitch is hereafter referred to in hub diameters per revolution, rather than feet. As all standard series of Moore fans are aerodynamically scaled, the pitch, when given in hub diameters, is a dimensionless figure applicable to all series of Moore units with similar blade shape.

Pitch

$\boxed{P}$

P_{HD} = The axial advance of the air stream per revolution of the wheel, expressed in hub (air seal) diameter

$$P_{HD} = \frac{12V}{S(RPM)}$$

Where:
V=velocity in feet per minute
S=Series of fan (air seal) diameter in inches

PITCH LIMITATIONS

Fans will operate at maximum efficiency at their design pitch. For Moore fans, it is highly desirable to keep the operating pitch between 1-1/2 and 2-1/2 hub diameters and essential that it be kept between 1 and 3 hub diameters.

DEFLECTION

Deflection of the air stream-that is, the rotation of the discharge due to the air of necessity rotating in the direction of the wheel — is present in all fans. This deflection, or rotation, may be counteracted by guide vanes located ahead of or following the wheel. Deflection at the hub becomes a limiting factor in fan performance because flow will break down completely when the deflection at the hub exceeds a certain angle. Deflection at the hub may be calculated as shown when pitch and the number of blades required are known.

Deflection At Hub

$\boxed{\phi}$

ϕ= The angle of air deflection from axial near the hub, in degrees

$$\phi = \tan^{-1}(0.14P)(P_f/P_v)$$

HORSEPOWER AND EFFICIENCY

In the following discussion, horsepower is divided into three categories. Certain relationships, however, may be considered to be true for all types of horsepower considered:

1. Horsepower of a given fan in a given installation, with increase or decrease in RPM, will vary approximately as the cube of the RPM. This is easily explained by the following:

   (a) The static pressure will increase as the square of the CFM.
   (b) The velocity pressure will increase as the square of the CFM.
   (c) Since the velocity pressure and static pressure are additive, the total pressure increases as the square of the RPM.
   (d) The volume will increase directly as the speed.
   (e) As a result, the horsepower will increase as the cube of the speed, the horsepower being the product of the CFM and the total pressure.

2. With changes in air density, horsepower for a given volume will increase or decrease in direct proportion to the air density, provided that the static pressure also varies directly with the air density, as is the case in most practical applications.

Horsepower HP

Horsepower Output

Total horsepower output, or total air horsepower, is the actual output of the fan in terms of cubic feet per minute against the total pressure (static pressure plus velocity pressure), and is calculated using the total pressure.

Static air horsepower output contemplates only static pressure and does not give credit for velocity pressure.

$$\text{Total } HP_{output} = \frac{P_t \text{ x } CFM \text{ x } 5.2}{33{,}000}$$

$$\text{Static } HP_{output} = \frac{P_s \text{ x } CFM \text{ x } 5.2}{33{,}000}$$

$$HP_{input} = \frac{P_t \text{ x } CFM \text{ x } 5.2}{33{,}000 \text{ x } E}$$

Where:
P_t and P_s are in inches H_2O
E= Efficiency expressed as a decimal

Horsepower Input

Horsepower input is defined as the brake horsepower (BHP) required to rotate the fan wheel in order to achieve a given performance and equals the total horsepower output of the fan divided by the fan efficiency expressed as a decimal. The horsepower input to the fan is always somewhat less than the power input to the motor due to the inevitable losses in the motor and drive mechanism.

If the efficiencies of the motor and drive and the current draw of the motor are known, the net horsepower input to the fan can be calculated. The formula below assumes a 3-phase AC system:

Where: E = Voltage
I = Amps

(ME) = Motor Efficiency
(DE) = Drive Efficiency
(PF) = Power Factor

$$\text{Nt BHP} = \frac{E \text{ x } I \text{ x } (PF) \text{ x } 1.732 \text{ x } (ME) \text{ x } (DE)}{746}$$

Efficiency E

Mechanical Efficiency

Mechanical or overall efficiency is the best method of comparing the performances of different fans. It gives the fan credit for the work done in accelerating air to the discharge velocity, as well as the work done against static pressure. Mechanical or overall efficiency is the ratio of the total air horsepower output to the brake horsepower input to the fan.

A well-designed fan will achieve 75% overall efficiency so long as certain minimum conditions are observed and will achieve between 80% and 85% efficiency when selected and installed under favorable conditions.

$$E = \frac{\text{Total } HP_{OUTPUT}}{HP_{INPUT}}$$

Static Efficiency

Static efficiency is the ratio of the static air horsepower output to the brake horsepower input. It is of little value in comparing performances of different fans, but is quite valuable in comparing fan selections for a given installation. Static efficiency neglects the velocity pressure imparted to the air and considers only the CFM delivered against the static pressure. Efficiency in these terms is of the utmost value in fan selection since credit is not given for velocity pressure in this calculation. For example, the larger the diameter of fan selected, the lower will be the velocity pressure, with consequently lower wasted power and higher static efficiency.

$$\text{Static Efficiency} = \frac{\text{Static } HP_{OUTPUT}}{HP_{INPUT}}$$

FAN PERFORMANCE FORMULAS ADAPTED FOR CALCULATORS AND COMPUTERS

1. $\rho = e^{-3.7\times10^{-5}A} \times 530/(T+460)$ -OR
   (Density of fluid in lbs/ft3) /.075

2. $(ACFM) = (SCFM) / \rho$

3. $D_{MIN} = [S / 6\ 4]_{N.I.}$

4. $D_{MAX} = [0\ 36\ S]_{N.I.}$

5. $(RPM)_{MAX} = 4150/D$

6. $V = (ACFM)\ P/4x\ (D+2t/12)^2-((S-3.5)/12)^2$

7. $P = 12V/S(RPM)$

8. $P_v = 6.2235 \times 10^{-8}\ \rho\ V^2$

9. $P_t = eP_V + P_S$

10. $P_{t\text{-}MAX} = Pt(N/N_t)$

11. $F = \tan^{-1}(0.14P)(P_t / P_v)$

12. $F_{MAX} = 55°$

13. $M = 2.05 - (6D / S)$
    $= 1.00$ if $2.05 - (6D / S) < 1.00$

14. $N_t = M\ Pt/\ 3.8 \times 10^{-8}\ \rho\ S\ (RPM)^2$
    If S = 19
    $N_t = M\ P_t/\ 3.8x10\ 8\ \rho\ (40)\ (RPM)^2$

15. $N_{MAX} = [0.205\ S\ ]_{N.I.}$ (manually adjustable blades)
    = 5 (automatic Series 33)
    = 8 (automatic Series 40)
    = 10 (automatic Series 49)
    = 12 (automatic Series 60, 73, 90)

16. $(BL) = N_t / N$

17. $(BL)_{MIN} = 0.70$

18. t = 0.25% x Dia. (ft)
    If an alternate tip clearance is required, use that value in calculating efficiency.

19. $E = (.84 + .029D/(S/12))\ (1-1.4\ t / D)_2 e^x$
    where: $x = -.16\ (P - 1.7 - .23D/(S/12))^2$

    If S = 19 $E = E\ (N_t/N)^{0.2}$

> *Note on Nearest Integer*
> In several cases where integer values are computed, it has been necessary to provide an equation which must be interpreted to the nearest integer. Values to be interpreted are enclosed in brackets with N.I. subscript. $[\ ]_{N.I}$.

Definitions

ρ = *Density ratio of working fluid to dry air at sea level (29.92" Hg3.and 70 °F.*

T = *Temperature in °F of air pumped by the fan.*

A = *Altitude (elevation) of fan in feet above sea level*

(SCFM) = *Mass flow rate of air through fan in standard cubic feet/ min.*

(ACFM) = *Volume flow rate of air through fan in actual cubic feet/ min.*

S = *Series of fan = nominal air seal (the aerodynamic hub) in inches.*

D = *Diameter of fan in feet*

D_{MIN} = *Minimum diameter fan available for particular class and series*

D_{MAX} = *Maximum diameter fan available for particular class and series*

N_t = *Number of blades theoretically required on fan (not necessarily an integer)*

N = *Actual number of blades used on fan-usually the first integer greater than or equal to* N_t *but may be larger*

N_{MAX} = *Maximum number of blades available for particular class and series*

(BL) = *Blade load-Ratio of theoretical number of blades to actual number of blades.*

(RPM) = *Rotational speed of fan in revolutions / minute*

$(RPM)_{MAX}$ = *Maximum allowable (RPM) for particular class, series and diameter*

V = *Axial air velocity through fan in feet per minute*

P = *Relative pitch of fan expressed in air seal diameters / revolution*

FOR COMPUTERS AND CALCULATORS

20. (BHP) = $1.575 \times 10^{-4} P_t$ (ACFM)/E

21. ε = 1.0 (wheel in bell inlet or duct)
1.1 (wheel in conical inlet)
1.2 (wheel set well back in long tube)
1.3 (wheel set well back in short tube3
1.4 (wheel set back in re-entrant tube)
1.5 (wheel in inlet end of short tube)
2.0 (wheel in inlet end of re-entrant tube)

22. B for electric motor drive fans = (6.6P) + (15Nt / N) - 12.05
If S=27, B = (6.6P) + ($15N_t$ / N) - 14.05
B for engine drive fans = (6.6P) + ($15N_t$ / N) - 15.55
If S=27, B = (6.6P) + ($15N_t$ / N) - 17.55

23. B_{MAX} = 18° for electric motor drive fans
B_{MAX} = 15° for engine drive fans

24. (TS) = π D (RPM)

25. τ_f = 5252 (BHP) / (RPM)

26. τ_b = τ_f /N

27 $\tau_{b\text{-MAX}}$ = 286 (for small resilient mounts)
= 380 (for large resilient mounts)

28. S = 19,27,33,40,49,60,73 or 90

29. V_{MAX} = 3000 FPM

30. F_t = 5.192 (P_t) ($\pi D^2/4$)

Note on Torque Values

$$\tau_{f\text{-MAX}} = \tau_f \times \text{Torque Factor}$$

Equation No. 25 gives the running torque of the fan. To determine the maximum torque applied to the fan during the starting cycle, multiply this number by the starting torque factor of the motor. For a standard NEMA motor, the torque factor is a. use this maximum torque value to size the QD type taper lock bushing. Use the torque per blade factor (Eq. 26) to determine the required mount size (Eq. 27) and to check that the maximum is not exceeded.

Note on Calculated Efficiency
Moore Fans have sufficient overload capacity that ACFM and static pressure as derived in the formulas are guaranteed. The value calculated for efficiency is an average value for normal blade loading and moderate air deflections at the hub but contingent on selection ant installation, individual fans will vary as much as ± 3% about this calculated efficiency.

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Definitions

P_s = Static pressure increase through fan in inches of H2O

P_v = Velocity pressure of air exciting fan in inches H2O

E = Inlet factor

P_t = Total Pressure

Φ = Angle of air deflection near air seal in degrees

$P_{t\text{-MAX}}$ = Maximum allowable Pt to avoid airflow breakdown at air seal

M = Multiplier necessary to determine number of blades for fans where D is less than 2.26 x (S /12)

t = Recommended mean distance between blade tip and fan ring in inches

E = Efficiency of fan (1 =l OO%)

(BHP) = Power required to drive fan using efficiency E above in horsepower

τ_f = Torque required to drive entire fan, in Ft-Lbs (See Note)

τ_b = Torque per blade required to drive fan (See Note)

$\tau_{f\text{-MAX}}$ = Maximum allowable torque on fan

$\tau_{b\text{-MAX}}$ = Maximum allowable torque per blade installed on fan

B = Blade angle set at clevis in degrees

B_{MAX} = Maximum allowable blade angle

(TS) = Fan tip speed in ft/min

F_t = Axial force in lbs imposed on the shaft due to the air load on the fan

V_{MAX} = Maximum allowable velocity

Fan Law Summary:

The table below is intended to illustrate the relationships between the basic parameters of importance in fan selection. These formulas will not give exact answers applicable to a given situation, however, due to the modifying factors which are not taken into consideration. For example, the change in CFM assumes that the static pressure remains constant. It is also assumed that the change made has no effect on the efficiency, which is rarely the case. It is possible that a change will so improve efficiency that the horsepower will actually be reduced although the basic relationships shown here indicate an increase.

Will result in the changes shown in these columns assuming the quantities shown are held constant

Changing the values in this. column ⇓	ACTUAL CFM $(ACFM)_B =$	TOTAL PRESSURE $Pt_{B=}$	REVOLUTIONS PER MINUTE $(RPM)_B =$	HORSEPOWER INPUT $(HP)_B =$	NOISE LEVEL IN dBA $(PWL)_B = (PWL)_A$
DENSITY RATIO p_A to p_B	Constant	$(Pt)_A \times \frac{\rho_B}{\rho_A}$	Constant	$\mathbf{(HP)_A \times} \frac{\rho_B}{\rho_A}$	$+10 \log \frac{\rho_B}{\rho_A}$
	$(ACFM)_A \times \left[\frac{\rho_A}{\rho_B}\right]^{1/2}$	Constant	$(RPM)_A \times \left[\frac{\rho_A}{\rho_B}\right]^{1/2}$	$\mathbf{(HP)_A \times} \left[\frac{\rho_A}{\rho_B}\right]^{1/2}$	$+20 \log \frac{\rho_A}{\rho_B}$
	Constant (SCFM) $(ACFM)_A \times \frac{\rho_A}{\rho_B}$	$(Pt)_A \times \left[\frac{\rho_A}{\rho_B}\right]$	$(RPM)_A \times \frac{\rho_A}{\rho_B}$	$\mathbf{(HP)_A \times} \left[\frac{\rho_A}{\rho_B}\right]^2$	$+ 50 \log \frac{\rho_A}{\rho_B}$
DIAMETER D_A to D_B Assuming fan proportions(D/S) remain constant	$(ACFM)_A \times \left[\frac{D_B}{D_A}\right]^2$	Constant	Constant (TS) $(RPM)_A \times \frac{D_A}{D_B}$	$\mathbf{(HP)_A \times} \left[\frac{D_B}{D_A}\right]^2$	$+20 \log \frac{D_B}{D_A}$
	$(ACFM)_A \times \left[\frac{D_B}{D_A}\right]^3$	$(Pt)_A \times \left[\frac{D_B}{D_A}\right]^2$	Constant	$\mathbf{(HP)_A \times} \left[\frac{D_B}{D_A}\right]^5$	$+80 \log \frac{D_B}{D_A}$
$(RPM)_A$ to $(RPM)_B$	$(ACFM)_A \times \frac{(RPM)_B}{(RPM)_A}$	$(Pt)_A \times \left[\frac{(RPM)_B}{(RPM)_A}\right]^2$	$(RPM)_A \times \frac{(RPM)_B}{(RPM)_A}$	$\mathbf{(HP)_A \times} \left[\frac{(RPM)_B}{(RPM)_A}\right]^3$	$+60 \log \frac{(RPM)_B}{(RPM)_A}$
NUMBER OF BLADES N_A TO N_B	$(ACFM)_A \times \left[\frac{N_B}{N_A}\right]^{1/2}$	$(Pt)_A \times \frac{N_B}{N_A}$	Constant	$\mathbf{(HP)_A \times} \left[\frac{N_B}{N_A}\right]^{3/2}$	$+ 15 \log \frac{N_B}{N_A}$
	Constant	Constant	$(RPM)_A \times \left[\frac{N_A}{N_B}\right]^{1/2}$	Constant	$+ 15 \log \frac{N_A}{N_B}$

If the values calculated lie outside the desired or permissible limits, try the following corrections:

P_t requires RPM > $(RPM)_{MAX}$	Increase N and/or use higher S to reduce required RPM
P_t>P_t-MAX	Use higher S and/or higher RPM and/or increase N
f too large	Use high RPM and/or higher S and/or reduce D
N_t too small	Use lower RPM and/or lower S
N_t too large	Use higher RPM and/or higher S
E too small, P<2.4	Increase P by using lower RPM and/or smaller D
E too large, P>2.4	Decrease P by using higher RPM and/or larger D
T_f too large	Increase RPM and/or decrease motor HP
T_b too large	Inrease N and/or increase RPM