HUDSON
Products Corporation

The Basics of
AXIAL FLOW FANS

CONTENTS

FOREWORD

This manual is designed to familiarize users with applications for axial flow fans, velocity recovery stacks, seal discs, and variable flow fans. Calculations are provided for estimating fan power consumption and noise.

It should be noted that final fan selection should be made by using Hudson's Tuf-Lite®, Fan Selection Program or by contacting Hudson Products Corporation at 713-914-5700 or 1-800-634-9160.

FIGURES

NOMENCLATURE

English Letter Symbols

ACHE	=	air-cooled heat exchanger
ACFM	=	actual cubic feet per minute
ACT	=	actual conditions
ASP	=	actual static pressure
ATP	=	actual total pressure
AVP	=	actual velocity pressure
BHP	=	brake horsepower
BPF	=	beam or blade passing frequency
CS	=	current source
CFM	=	cubic feet per minute
DR	=	density ratio
Eff	=	efficiency
D	=	fan diameter
FPM	=	feet per minute
HP	=	horsepower
LP	=	sound pressure levels (metric units)
LW	=	sound power levels (metric units)
MSL	=	mean sea level
N	=	number of fans
NFA	=	net free fan area
PWL	=	sound power level (I-P units)
r	=	inlet bell radius
PWM	=	pulse wide modulation
R	=	feet from center of noise source
RH	=	relative humidity
RPM	=	revolutions per minute
SCFM	=	standard cubic feet per minute
SF	=	speed factor
SP	=	static pressure
SPL	=	sound pressure level (I-P units)
SR	=	solidity ratio
STD	=	standard conditions
Eff_{Total}	=	total efficiency
Eff_{Static}	=	static efficiency
TP	=	total pressure
V	=	velocity
VFD	=	variable-frequency driver
VP	=	velocity pressure
VR	=	recovered velocity pressure
VVI	=	variable-voltage inverter

Greek Letter Symbols

ρ = density

Subscripts

ACT (act) = actual condition
STD (std) = standard condition
exit (ex) = velocity recovery stack exit
c = fan curve condition at standard density

1 = inlet, condition 1
2 = outlet, condition 2
in = inlet
out = outlet
SL = sea level
des = design
eff = effective

1.0 Definition of Axial Fans

An axial flow fan moves air or gas parallel to the axis of rotation. By comparison, a centrifugal or radial flow fan moves air perpendicular to the axis of rotation. Axial flow fans are better suited for low-resistance, high-flow applications, whereas centrifugal flow fans apply to high-pressure resistance, low-flow conditions. Typically, the types of fans discussed in this manual can handle "resistances" up to approximately 1 in. of water. Axial fans can have widely varied operating characteristics depending on blade width and shape, number of blades and tip speed.

The most common type of fan for air-cooled heat exchanger (ACHE) is less than 14 ft. diameter and has four blades. The most common type for wet cooling towers is 28 ft diameter and has eight blades. A typical ACHE fan is shown in Fig. 1.

Typical Air-Cooled Heat Exchanger Fan
Fig. 1

2.0 Fan Engineering Nomenclature

ACFM - Actual cubic feet per minute of air moved by the fan.

Actual Conditions - Resistances related to actual inlet or outlet temperature and fan elevation above mean sea level compared to standard conditions.

Air Density - Air density at the plane of the fan based on standard or actual conditions.

Beam Passing Frequency - Number of times per revolution that one fan blade passes over a beam or strut — thought of as "how the structure interacts with the fan blade" expressed in cycles/sec (Hz).

Blade Natural Frequency - Frequency at which a blade freely vibrates when it is struck in cycles/sec (Hz).

Blade Passing Frequency - Number of times per revolution that a fan tip passes a point on the fan ring expressed in cycles/sec (Hz) — thought of as "how the fan interacts with the structure".

Brake Horsepower - (BHP) - Net power required by the fan at actual conditions to perform the required design work.

Chord - Straight line distance between the leading and trailing airfoil edges.

Fan Diameter - Width/distance between opposite blade tips.

Fan Laws - Set of laws that predict performance changes if one or more parameters are changed from one fan or operating condition to another. These laws govern airflow, pressure capability and power required among many other parameters.

Fan Ring Diameter - Inside diameter of fan housing at the plane of fan.

First Mode Resonant Frequency - Frequency at which a blade freely vibrates when struck ("natural" frequency) in cycles /sec (Hertz).

Forced Draft ACHE - Fan is located below the heat transfer surface forcing ambient air through the bundle.

Harmonic Frequency - Integer multiples of fan RPM and expressed as 1x, 2x and 3x fan speed in cycles/second (Hertz). Harmonic frequency is checked against resonant frequencies to prevent vibration and fatigue.

Induced Draft ACHE - Fan is located above the heat transfer surface drawing ambient air through the bundle. The fan is exposed to the heated exhaust air.

Leading Edge - Thicker portion of the airfoil that is the first part of the blade to meet the air.

Net Free Area - Net area at the plane of the fan through which all air must pass. Usually based on the nominal fan diameter minus seal disc area or hub diameter. Note that blade area is not considered.

Pitch Angle - Blade tip angle below the horizontal required to do the design work and move air upward. Hudson fans all rotate clockwise looking into the airflow.

Resonant Frequency Safety Margin - Percent difference between the closest resonant frequencies of 1st mode resonant frequency, blade and beam pass frequencies, and 1x Harmonics.

SCFM - Airflow rate moved at standard conditions, in standard cubic feet per minute.

Solidity Ratio - Measure of a fan's pressure capability solidity — sum of the tip widths divided by the fan circumference.

Stall Point - Fan operating condition where the boundary layer of air separates from the airfoil and causes turbulence. This can be compared to boat propeller or pump cavitation.

Standard Conditions - Resistances related to the standard density of air at 0.075 lbs/ft^3, at 70°F dry bulb temperature and sea level (29.92 in. Hg).

Static Efficiency - Fan efficiency based on static pressure and fan brake horsepower at the same density.

Static Pressure - Sum of all the system resistances against which the fan must work, expressed in inches of H_2O. This is the useful work required from the fan. (Velocity pressure excluded).

Tip Clearance - Distance between the tip blade and the fan ring or housing, sometimes expressed as a percent of the fan diameter.

Tip Speed - Peripheral speed of the fan tip expressed in feet per minute.

Total Efficiency - Fan efficiency based on the total pressure and fan brake horsepower at the same density for standard or actual conditions.

Total Pressure - Sum of the static pressure and velocity pressures.

Trailing Edge - Thinner portion of the air foil.

Velocity Pressure - Parasitic loss caused by work done to collect all the air into the fan's inlet, expressed in in. of H_2O. It is based on the fan's net free area at the plane of the fan.

Velocity Recovery Stack - Device frequently used in cooling towers but rarely in ACHEs. It captures the kinetic energy of the exit air velocity and converts it to useful work. It "reduces" air velocity and decreases the total pressure acting on the fan.

WR^2- Flywheel effect of fan that relates mass moment of inertia about the axis of rotation. in lb - ft^2.

2.1 Fan Engineering Units

I-P: inch-pound U.S. system
metric: metric system
SI: international system of units

• LENGTH

  I-P: feet (ft)
  metric: meter (m)
  SI: meter (m)

• VOLUME

I-P: cubic feet per minute (CFM)
metric: cubic meters per second (m^3/s)
           cubic meters per hour (m^3/h)
SI: cubic liters per second (l^3/s)

• PRESSURE

I-P: inches of water (in. - H_2O)
metric: mm of water (mm - H_2O)
SI: Pascals (N/$m^{2)}$)

• DENSITY

I-P: lbs per cubic ft (lb/ft^3)
metric: kilograms per cubic meter (kg/m^3)
SI: (kg/m^3)

      Note <u>standard</u> density

      I-P   =   0.075 lb/ft^3
      metric =   1.201 kg/m^3
      SI   =   1.201 kg/m^3

• ROTATIONAL SPEED

I-P: revolutions per minute (RPM)
metric: turns/min
SI: turns/min

• TEMPERATURE

I-P: degrees Farenheit (^oF)
metric: degrees Celsius (^oC)
SI: degrees Celsius (^oC)

• AREA

I-P: ft^2
metric: m^2
SI: m^2

• POWER

I-P: HP (horsepower)
metric: kW (kilowatt)
SI: kW (kilowatt)

• TIP SPEED

I-P: ft/min
metric: m/s
SI: m/s

• VIBRATION AMPLITUDE

I-P: mils (1/1000 inches)
metric: microns (1/1000 mm)
SI: microns (1/1000 mm)

• VIBRATION FREQUENCY

I-P   cycles per minute (CPM)
        cycles per second (Hz)
metric: cycles per second (Hz)
SI: cycles per second (Hz)

• SOUND POWER LEVEL

I-P: PWL dB (decibels)
metric: Lw dB (decibels)
SI: Lw dB (decibels)

• SOUND PRESSURE LEVEL

I-P: SPL dB (decibels)
metric: Lp dB (decibels)
SI: Lp dB (decibels)

2.2 Useful Conversion Factors

• AIRFLOW

$2118.64 * m^3/s = ft^3/min$
$0.000\ 471\ 947 * ft^3/min = m^3/s$

• PRESSURE

$25.40\ \ * in.\text{-}H_2O = mm\text{-}H_2O$

$249.08\ \ * in.\text{-}H_2O = Pa$

$9.806 * mm\text{-}H_2O = Pa$

$Pascal\ (Pa) = N/m^2$

• VOLUME

$35.314 * m^3 = ft^3$
$0.02832 * ft^3 = m^3$

• DENSITY

$Standard\ density = 0.075\ lb/ft^3$
$16.018\ 463 * lb/ft^3 = kg/m^3$
$Standard\ density = 1.201\ kg/m^3$

• LENGTH

$25.4 * inch = mm$
$3.281 * m = ft$
$1000 * mm = micron$
$0.3048 * foot = m$

• TEMPERATURE

$(^{o}C * 1.8) + 32 = {^{o}F}$

$(^{o}F - 32) / 1.8 = {^{o}C}$

• POWER

$0.746 * HP = kW$

3.0 Fan Selection Criteria

Many parameters are important in selecting a fan to meet the specified operating conditions such as:

Fan diameter - to suit bay or cell size,

Hudson blade type: "H", "HW", "B", "C", "D", "K",

ACFM - Airflow required for design heat transfer duty,

ASP - Actual static pressure from the sum of all resistances to airflow,

Air temperature at plane of fan,

Fan elevation above mean sea level or air density at fan, and

Fan speed in either tip (FPM) or rotational speed (RPM).

Also, a velocity recovery stack installed? Is there a noise limitation? How much allowable horsepower can the fan use? Will the fan avoid resonant frequency problems?

In the most simple case, design airflow, static pressure, and density are calculated for the fan "curve" conditions at 12,000 FPM tip speed and standard density with no noise limitations. The operating point is then plotted on a fan curve of the appropriate diameter.

The fan curve shows the correct pitch angle and horsepower required by the fan at the design point.

Selection is checked for the following:

(a) Confirm that the operating point is not close to a "stall" condition. (Explained in detail later).

Typically, a 2° pitch angle safety margin is desired before reaching a stall point on the curve or overloading the motor. The American Petroleum Institute has specific limitations that must be met for ACHE applications.

(b) Confirm that the fan brake horsepower (BHP) is low enough that when environmental, drive and motor losses are added, the installed motor HP is not exceeded.

(c) Be sure that the "horsepower per blade" mechanical limit (for Hudson fans) is not exceeded or select a fan with additional blades.

(d) Calculate the blade passing frequency, and compare it to the first mode resonance frequency for the selected blade (for Hudson fans).

(e) Repeat step (d) for the beam passing frequency. (Most structures have four main beams.)

(f) Repeat step (d) for 1x harmonic of fan RPM, and compare to first mode resonant frequency.

(g) Check resulting noise level against any noise limit specifications. Noise level is most often specified as sound pressure level (SPL) at a given distance from the tower or ACHE. Convert the SPL to a sound power level (PWL) to evaluate the selected fan. Some adjustments to tip speed and/or the number of blades may be required.

(h) Check air temperature at the fan. Does it exceed maximum fan operating or startup temperatures?

If you have any questions, or require a copy of the Hudson Tuf-Lite® Fan Rating Program or specification curves please contact the Hudson Fan Sales department for assistance at 1-800-634-9160.

4.0 Basic Fan Equations - (I-P Units Only)

For standard conditions (0.075 lbs/ft^3 density)

- $TP = SP + VP$
- TP = Total Pressure
- SP = Static Pressure
- VP = Velocity Pressure

Normally, pressures are given at actual conditions (density) so the equation becomes:

$ATP = ASP + AVP$ also:

$TP = ATP * 1$ / Density Ratio

$$\text{Density Ratio} = \frac{(\rho)_{ACT}}{(\rho)_{STD}}$$

$SP = ASP * 1 / DR$

Note: Hudson fan curves are based on standard conditions and 12,000 FPM tip speed. If operating conditions are given in actual conditions, they must be converted to standard conditions before referring to the fan curve to determine performance.

- ACFM = Actual ft^3/min

- $BHP = \dfrac{TP_{ACT} * ACFM}{6356 * Eff_{TOTAL}}$

Note: 6356 is a constant relating to the power required to move one CFM of air against a static pressure of one in.-H_2O at standard density using one horsepower.

Brake HP is the net power absorbed by the fan and does not include motor and drive losses.

Conversely:

- $Eff_{TOTAL} = \dfrac{TP_{ACT} * ACFM}{6356 * BHP}$

- $Eff_{STATIC} = \dfrac{SP_{ACT} * ACFM}{6356 * BHP}$

Note: Pressures and horsepower must be at the same density, either all standard or actual.

- Velocity (ft/min) $= \dfrac{ACFM(ft^3/min)}{NFA\ (ft^2)}$

where NFA is the "net free area" of the fan

- Velocity Pressure $= \left(\dfrac{V}{4005}\right)^2$ for std air

$$AVP = \left(\dfrac{V}{4005}\right)^2 \left(\dfrac{\rho_2}{\rho_1}\right)$$

where ρ_2 = density at the fan

ρ_1 = std air density (0.075 lb/ft^3)

Note: Velocity pressure (noted on Hudson's fan curves) must be determined based on the fan used and required ACFM.

- Basic fan laws:

  CFM = fn(RPM)[1]
  Airflow varies in direct proportion to RPM

  SP or TP = fn(RPM)[2]
  Pressure capability varies with the square of a change in RPM

  HP = fn(RPM)[3]
  Power required varies with the cube of a change in RPM.

- When velocity recovery stacks are used:

  $VR = (VP_{fan} - VP_{exit}) *$ Recovery efficiency

  Recovery efficiency is usually 0.6 - 0.8

  Use:
  VP_{fan} - velocity pressure at plane of fan
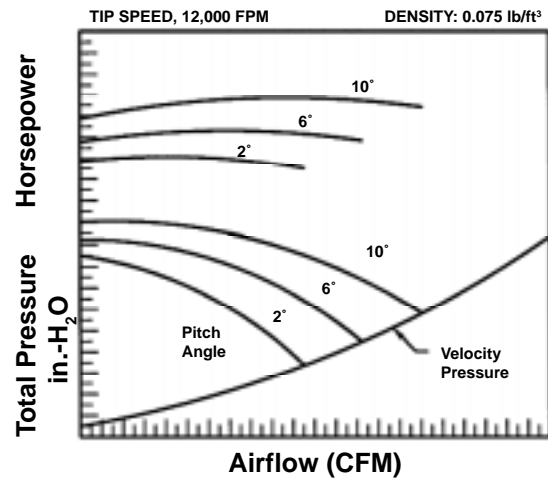
  VP_{exit} - velocity pressure at top of stack

  When velocity recovery stack is used, horsepower savings are calculated using:

  $TP_{eff, act} = TP_{act} - VR$

  Note: All pressures should be based on the same density.

- Other noise related equations are found in Section 9 (Fan Noise).

5.0   Fan Performance Calculation

TIP SPEED, 12,000 FPM   DENSITY: 0.075 lb/ft³

Typical Fan Curve
Fig. 2

Manual fan selection procedure (See Fig. 2, Fan Curve).

1. Define operating conditions

   - Fan diameter

   - Blade type: select Tuflite II®, designated by "H" after fan diameter.

   - ACFM (Actual Cubic Feet per Minute)

   - ASP (Actual Static Pressure)

   - Air temperature at fan

   - Elevation above MSL or

   - Air Density

   - RPM or tip speed

   - Velocity recovery stack used? If so, height of the venturi section above the fan or fan exit diameter is required .

   - Noise requirements will affect selection, and will be discussed later.

2. Find density correction factor 1/DR

3. Find velocity pressure (STD) from appropriate fan curve or:

$$VP_{std} = \left(\frac{V}{4005}\right)^2$$

Velocity ressure line is shown at the bottom of the Hudson curves.

4. Calculate standard total pressure:

$$TP_{std} = ASP * (1/DR) + VP_{std}$$

5. Calculate speed factor (SF) $= \dfrac{\text{Curve RPM}}{\text{Actual RPM}}$

Calculate SF^2 and SF^3

Note: If fan is operating at 12,000 FPM tip speed, SF = 1 when using Hudson curves.

6. Prepare to enter curve: c = curve cond.

$$ACFM_c = ACFM_{design} * SF$$
$$TP_c = TP_{std} * SF^2$$

7. Enter curve:

Read up from $ACFM_c$ to TP_c. Find pitch angle, and interpolate if necessary. Read up to design pitch angle on HP lines. Read left to design BHP (std).

Note: To determine the fan operating point, pitch angle and standard BHP must be found from curve.

8. Calculate actual BHP:

$$BHP_{act} = \frac{BHP_{std}}{(1/DR) * SF^3}$$

9. Calculate total efficiency:

$$Eff_{TOTAL} = \frac{TP_c * ACFM_c}{6356 * BHP_{std}}$$

Note: All values must be at the same density conditions. TP_c and BHP_{std} are at standard density.

Another formula for total efficiency is:

$$Eff_{TOTAL} = \frac{TP_{act} * ACFM}{6356 * BHP_{act}}$$

10. Calculate static efficiency

$$Eff_{STATIC} = Eff_{TOTAL} * \frac{ASP}{ATP}$$

11. If a VR Stack is used:

1. Calculate stack exit area . If not known, assume 7° angle of sides and calculate exit diameter and area. Use height of venturi section above plane of fan.

   Calculate SF^2 and SF^3

2. Calculate exit velocity pressure:

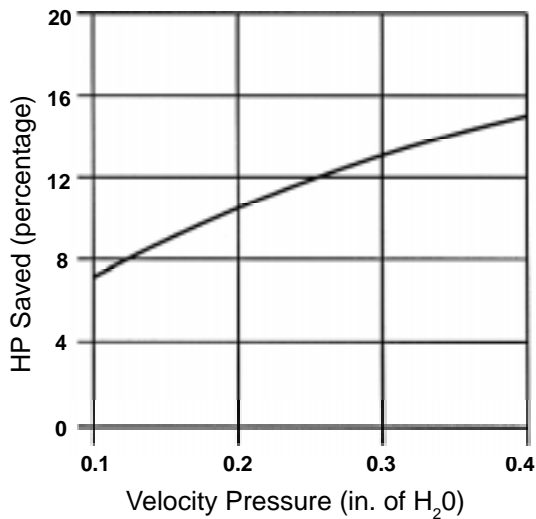$$VP_{ex} = \left(\frac{ACFM}{Area_{exit} * 4005}\right)^2 * \frac{P_{exit}}{P_{std}}$$

3. Calculate VR assuming 70% recovery efficiency
   $$VR = 0.7 * (VP_{fan} - VP_{ex}) \text{ in.-H}_2\text{0}$$

4. Calculate $TP_{eff} = (TP_{fan} - VR)$

5. Recalculate BHP $= \left(\dfrac{TP_{eff} * ACFM}{6356 * Eff_{TOTAL}}\right)$
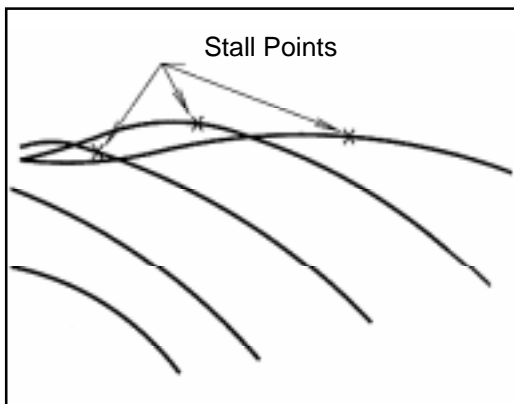
   Fig. 3 shows the HP saved by using VR stacks. See Section 6.3 for example calculations.

Approximate HP Savings with VR

Fig. 3

12. Check operating point on curve with proximity to stall. A stall point is where the curve changes inflection. (See Fig. 4 below).

Note: The Tuf-Lite® program avoids all stall conditions.

Stall Points on Fan Curve

Fig. 4

Allow a 2° pitch angle margin of safety before stall for a conservative fan operating point.

13. Calculate blade passing frequency (BPF):

$$BPF = \frac{No.\ Blades * RPM}{60} \quad in\ Hz$$

$Hz = cycles/second$

Compare with 1st mode resonance for the blade being used. There should be a 5% margin of safety on either side of a resonant condition. If not, change the speed or number of blades.

14. Check horsepower per blade:

$$HP/Blade = \frac{BHP_{act}}{No.\ Blades\ in\ fan}$$

Refer to Hudson's Tuf-Lite® Fan Selection Program for maximum recommended HP/blade chart.

6.0 Some Basic Assumptions

Following are basic assumptions that can be safely used for fan selection if certain data is not available.

1. Typically for cooling towers or ACHEs, tip speed will be 14,000 FPM maximum.

2. Typical blade quantities per fan are:

up to 14 ft dia.	4 blades
16 - 20 ft	6 blades
24 - 30 ft	8 blades
36 - 40 ft	8 blades

3. Horsepower ranges:

6-10 ft dia	7.5 - 15 HP
12 ft	20 - 30 HP
14 ft	30 - 60 HP
16 - 20 ft	50 - 100 HP
24 - 30 ft	100 - 250 HP
36 - 40 ft	150 - 300 HP

4. Temperatures at the fan (Farenheit):

Forced draft air coolers	80°- 100°
Induced draft air coolers	150° - 200°
Cooling towers	100°, 100% RH

5. Fan stacks:

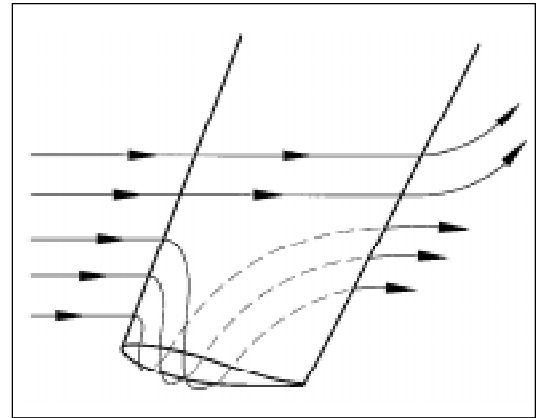   Assume 7° angle per side and 70% (0.7) recovery efficiency.

6. Noise levels:

    @ 12,000 FPM   102 PWL dB(A)

7. Typical Hudson fan:

    Total efficiency $\cong$ 75% minimum

    Static efficiency $\cong$ 55% minimum

6.1   Effect of Tip Clearance

Tip clearance between the blade and inside of the fan cylinder or fan ring is critical to proper axial flow fan performance. Hudson fan curves are derived from tests conducted at Texas A&M University's Engineering Laboratory, based on very close tip clearances with optimum inlet conditions. If close tip clearances are not maintained, a loss of performance will be noted, in pressure capability and airflow. The primary reason for close tip clearance is to minimize air loss or leakage around the tip, known as a tip vortex. (See Fig. 5). Hudson recommends the use of safety factors to correct for the environment in which a fan is placed.

Tip Vortex (Leakage)
Fig. 5

Exit air will be at a higher pressure than incoming air due to the work expended by the fan. Note that the blade performs most of the work in the outer portion of the airfoil. If a leakage path (or clearance) exists, the air will seek the path of least resistance and bypass the tip, causing a vortex and loss of performance. Tip clearances can be reduced with Hudson accessories.

Fig. 6 shows the API recommended tip clearances for smaller (16ft and less) diameter fans. Also included in this figure are the nominal tip clearances for larger diameter fans.

Fan Diameter	Minimum	Maximum
3ft through 9ft	1/4 in.	1/2 in.
>9ft through 11ft	1/4 in.	5/8 in.
>11ft through 16ft	1/4 in.	3/4 in.
18ft through 40ft	1/2 in.	1 in.

Tip Clearance
**Fig. 6**

6.2 Effect of Inlet Conditions

Consider the air around the entrance to the fan. In a re-entrant tube, like a fan ring, air must be collected from all directions and accelerated to the velocity at the plane of the fan. Some of this air is moving at 90° to the fan axis. If a smooth transition is not present, the inability of the airflow vectors to make rapid changes in direction will create a "Vena Contracta" at the blade tip, starving the blade of air in this area. (See Fig. 7)
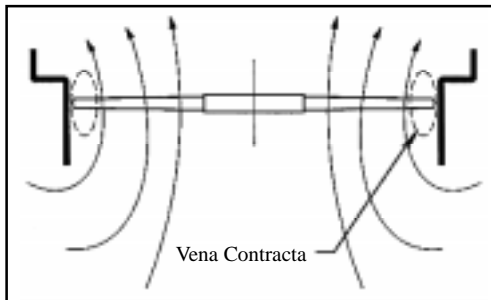
Airflow with No Inlet Bell
Fig. 7

Since most of the work is done by the outer portion of the fan blade, the result can be a loss of efficiency as high as 15-20%. An additional adverse effect can be increased vibration.

To prevent this loss, consider the use of an inlet bell attached to the bottom of the fan ring. A properly designed inlet bell provides a smooth transition as the air is gathered from all directions into the plane of the fan as shown in Fig. 8.
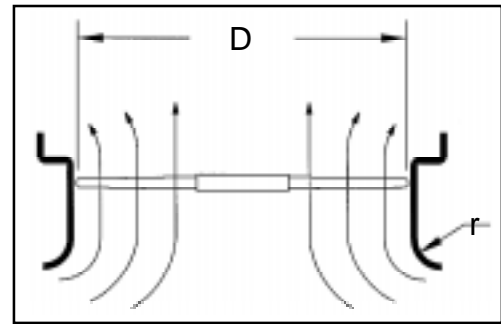
Airflow with Inlet Bell
Fig. 8

Note: Hudson Products' Fin-Fans® Air-Cooled Heat Exchangers all have inlet bells as standard equipment.

Hudson's Tuf-Lite® fan rating program allows several choices of inlet conditions: No inlet bell, elliptical inlet bell, rounded outlets with r = 0.05D, 0.1D or 0.15D and conical inlets with r = 0.05D, 0.1D or 0.15D.

6.3 Effect of Velocity Recovery Stacks

The VR stack recovers wasted velocity pressure energy at the top of the stack and converts it into useful work at the plane of the fan. Amount of recovery is based on the venturi section length and exit diameter. Hudson's Tuf-Lite® program assumes a 7° included venturi angle and a 70% recovery efficiency. Fig. 9 shows a typical VR stack on a large cooling tower fan.

**Typical Large Cooling Tower Fan
with VR Stack
Fig. 9**

Consider the following fan:

28 ft diameter, 8 blades, Type H
14 ft stack, 9 ft venturi height
1,180,000 ACFM
0.38 in. - H_2O ASP
136 RPM
105° F, at sea level
Use $r = 0.10D$ inlet condition

With no VR stack, fan performance is:

137.7 BHP
0.835 Total efficiency
0.512 Static efficiency
0.619 Total pressure actual

With the VR stack:

BHP = 123.6
0.840 Total efficiency
0.571 Static efficiency
0.559 Total pressure (actual)

In this example the VR stack reduced required power by (137.7 - 123.6) = 14.1 HP (10.5 kW). Operating 8400 hrs/yr and assuming a power cost of $0.05/kW-hr, yearly savings are $4410 per fan with the VR stack.

Most large fans now incorporate velocity recovery stacks because of these significant power savings. Fig. 10 illustrates the approximate power costs for various motor sizes assuming 96% of the year in constant operation.
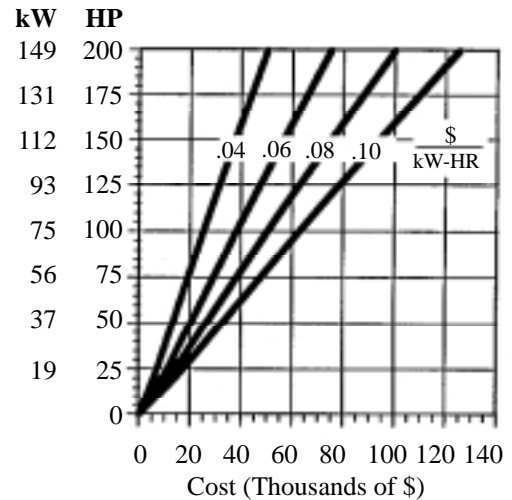
**Approximate Cost of Power
to Operate 8400 hrs/yr
Fig. 10**

6.4 Effect of System Resistance

Fan "system resistance" is defined as the sum of the static pressure resistances of each element in the system, as a function of airflow. It is important to be aware how system resistance changes when a new fan operating point is considered. The most common example is an airflow increase to an ACHE to increase the equipment's heat transfer capability.

Restating the basic fan laws:

1. Airflow is directly proportional to speed.

2. Static pressure requirements vary in proportion to (airflow)^2.

3. Horsepower requirements vary in proportion to (airflow)^3.

Therefore, a 10% increase in CFM requires a 33% increase in HP (1.33 x HP). Likewise, a 10% increase in CFM results in a 21% increase in SP (1.21 x ASP). (Note: Actual operating experiences have determined the relationship between SP and ACFM is more accurately predicted by an exponent of 1.8 rather than 2.0 and is dependent upon the unit design).

For ACHEs: $SP_2 = SP_1 \left(\dfrac{CFM_2}{CFM_1} \right)^{1.8}$

Consider a 25% increase in airflow required for a 14H - 4 fan currently moving 200,000 ACFM at 0.33 ASP with a 20 HP motor. (Assume 0.075 lbs/ft^3 air density and 273 RPM). Hudson's Tuf-Lite program shows:

17.1 BHP, 6.1° pitch

A 25% increase in airflow yields 250,000 ACFM. The new ASP increases to 1.49x0.33 or 0.49 in. - H_2O.

Results from the Tuf-Lite program show a 14H - 4 fan is still a choice, but 33.9 HP is now required to produce the 250,000 ACFM. A 40 HP motor would be required instead of the original 20 HP.

Note: American Petroleum Institute (API) guidelines allow up to 50 HP for belt-driven ACHEs. It is not uncommon to increase blade count to eight blades and increase motor HP significantly to achieve increased airflow. Larger motors, new drive systems or structural modifications may be required to accommodate the required airflow.

Factors limiting use of the existing fan could be:

(a) Fan stall
(b) Allowable HP/blade

If further duty increases were required, consider more blades per fan, a different (wider) blade type or higher operating speed. In some cases where higher noise may not be objectionable, speeds up to 13,500 FPM have been used for small fans . Increased noise and vibration will always accompany fan speed increase.

Fig. 11 shows the typical system resistance line for this example at increased airflows.
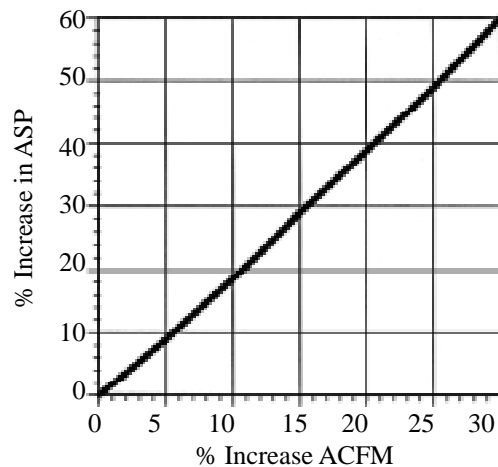
ASP vs. CFM for ACHE

Fig. 11

6.5 Effect of Horsepower

Once the required fan horsepower (BHP) is determined, certain variables including efficiencies and environmental factors, will determine the rated or installed driver horsepower .

1. Hudson's Tuf-Lite fan selection program shows fan brake horse-power required for the selected operating point. Drive system inefficiencies will result in an increased driver horsepower requirement.

2. Environmental losses are more difficult to determine but are a function of actual fan tip clearance, fan inlet conditions, number and size of beams under the fan and structural geometry. These losses must be accounted for when determining driver power requirement.

   Example:
   For a given fan selection:
   Actual Fan Brake Horsepower = 62 BHP (at the fan shaft)

   Example:
   Electric Motor Efficiency $\cong$ 95%

   Example:
   Drive System Efficiencies:
   Gear Boxes $\cong$ 98%
   V-Belts $\cong$ 95%
   Cog Belts $\cong$ 98%

   Example:
   Conservative Estimate of
   Environmental Losses $\cong$ 97%

   Therefore if:
   HP_{in} = Total Horsepower Required
   $$= \frac{\text{Fan Horsepower Required}}{(\text{Eff Motor})\,(\text{Eff Drive})\,(\text{Eff Environment})}$$

   Using a gear box drive:
   $$HP_{in} = \frac{62}{(0.95)\,(0.98)\,(0.97)}$$

   $$= 69.6\ HP$$

   Accounting for environmental losses.

The required electrical power for this example:

$$= 69.6 * 0.746 = 51.9\ kW$$

6.6 Effect of Density

Density, or weight of air per unit volume, is affected by air temperature and altitude at the plane of the fan. These are the inlet air conditions on a forced draft fan or the exit air conditions for an induced draft fan.

A constant RPM, constant pitch fan is considered to be a "constant volume machine". That is, it will move a constant volumetric flow (ACFM) of air regardless of the density.

Consider a change in ambient temperature for a forced draft fan. If the temperature decreases, air density (lb/ft^3) increases. Generally, resistance to airflow by the cooling tower fill or exchanger tubes (static pressure) increases, since the fan is moving the same volume of heavier air thereby increasing the required horsepower.

The opposite occurs when ambient temperature increases: Air density decreases, along with the static pressure and required horsepower static pressure decreases, and the required horsepower decreases. As stated, Hudson's fan curves show operation at standard density of lb/ft^3 at 70°F and sea level.

If a density table is not available, a density ratio can be calculated as:

$$\text{Density Ratio} = \frac{\rho_{ACT}}{\rho_{STD}} = \frac{460 + 70°F}{460 + T_{air}}$$

Density ratio is the ratio of the local air density to standard density.

If the fan plane is other than at sea level, then

$$DR = \left[\frac{(460 + 70)}{(460 + T_{air})} \right] \left[\frac{(AtmPress)_{act}}{(AtmPress)_{SL}} \right]$$

For example, at 95°F air temperature and Pact = 14.6 psia, then

$$DR = \left(\frac{530}{555} \right) \left(\frac{14.6}{14.7} \right) = 0.949$$

6.7 Effect of Vibration

All objects have a "natural" vibration frequency when struck sharply. This includes fan blades, fan rings and the structure. A component will generally vibrate in a sinusoidal wave motion. Vibration frequency is the number of cycles per second, also called Hertz. The distance a component moves per vibration cycle is called the amplitude and is usually measured in "mils" (1/1000 of an in.) "peak-to-peak". Peak-to-peak means the total amplitude on both sides of the components still position (plus and minus directions). This movement can also be characterized by the component's velocity given in inches per second.

A fan's environment continually induces vibration. Each time a blade passes near an obstruction, such as a drive shaft or structural support, blade load fluctuates. Vibration level is a function of the fan's speed and number of blades. If the vibration is at the fan's natural frequency, amplitude and internal stresses on the blade are greatly increased and can be destructive.

Blades should be tested by the manufacturer to determine vibration characteristics. Hudson checks all "modes" of vibration from 0 - 50 Hertz (cycles/sec) for every fan blade type. A "mode" is the basic shape the blade assumes during vibration. The 1st mode is the most significant with the blade assuming a "flapping" shape. The 2nd mode causes the blade to vibration in a "lazy S" shape with a neutral point in the middle. This neutral point is called a "node". There can be more complex modes that the blade assumes as the frequencies increase from 3rd through "nth" mode.

It is commonly thought that the most critical frequency is the "blade pass frequency" which is:

$$BPF = \frac{No.Blades * RPM}{60}$$

We think of this as "how the fan interacts with the structure". The "critical frequency" is the frequency at which the blade pass frequency coincides with the blade's 1st mode natural frequency.

We believe that a more important frequency is the beam pass frequency. This is the number of obstructions or beams one blade passes over per revolution. For example, if there are four beams in a fan turning 300 RPM, the beam pass frequency is:

$$BPF = \frac{4 \text{ Beams} * 300 \text{ RPM}}{60} = 20 Hz$$

Think of beam pass as "how the structure interacts with the fan blade".

It is also very important to consider "harmonics" of the fan RPM. Harmonics are integer multiples of the fan speed. The 1st harmonic = 1 x RPM.

In Hudson's Tuf-Lite® fan rating program, both blade pass and beam pass frequencies are calculated and compared to 1st mode fan resonance, as well as the 1st harmonic assuming the number of beams are input. A "Frequency Safety Margin" is calculated as the closest margin between any two of these frequencies. We recommend 5% as a minimum safety margin for trouble-free operation.

Consider not only vibration amplitude but also frequency. Frequency tells you where the problem is most likely located. For instance:

- Fan unbalance occurs at fan speed (once per revolution).

- Vibration at blade pass frequency is usually due to aerodynamic problems such as air moving over obstructions, beams or blocked inlets. It could mean a resonance in the structure or fan stack triggered by the blade pass frequency.

- Vibration at motor speed could mean a misaligned coupling, sheave, or drive shaft.

Fan unbalance can be corrected in the field. Field dynamic balancing takes into account the mass or rigidity of the mount and the sheave balance.

Blade pass frequency problems can be expensive, especially when due to resonance in a fan stack or structure. In some cases, it is necessary to add another blade or change motor speed to move away from the problem frequency.

Some final comments about vibration and resonance in fan blades:

Operating at the resonant frequency is much more dangerous for metal blades than composite fiberglass blades, like Tuf-Lite® or Tuf-Lite II®, because of the lower fatigue strength of aluminum. Fiberglass blades also have a high dampening factor that helps to reduce blade vibration.

Further, Tuf-Lite blade resonant frequency peaks are typically very narrow, so a 5% margin on either side of resonance is adequate.

API 661 Standard (Air-Cooled Heat Exchangers for General Refinery Services) states: 6 mils peak-to-peak is the maximum allowable vibration on the motor and primary supporting structure. The Cooling Tower Institute also publishes recommended vibration limits.

6.8   Effect of Number of Blades

The greatest amount of work a fan performs is done by the outer portion of the blade. Consequently, the fan solidity is critical. Solidity ratio is defined as the ratio of the sum of the blade widths to the fan's circumference. For example, consider a 14 ft 4-bladed fan with 13in. wide tips. The solidity ratio is:

$$SR = \frac{4 * 13}{12 * \pi * 14} = 0.10$$

If we increased the number of blades to 6, the solidity ratio would be 6/4 x 0.1 or 0.15. Likewise, for 8 blades, SR would be 0.2. So theoretically at least, 6-blades would do 50% more work than 4 blades, and 8 blades would do 100% more work. In reality a 6 blade fan only does about 40% more work than a 4-bladed fan; however, the principal is that: the work a fan can provide is proportional to the number or width of blades. Likewise, 4 wide blades can do more work than 4 narrow blades by the ratio of the blade widths even if the blade shapes are not similar.

6.9   Effect of Blade Shape

Airflow across the plane of the fan is not uniform varying from positive at the tip to negative at the center of the fan. Blade shape and twist of the airfoil along the blade affects the shape of the velocity profile.

Velocity profile of a well-designed tapered blade with a generous twist compared to a constant chord blade with minimal twist is shown in Fig. 12.
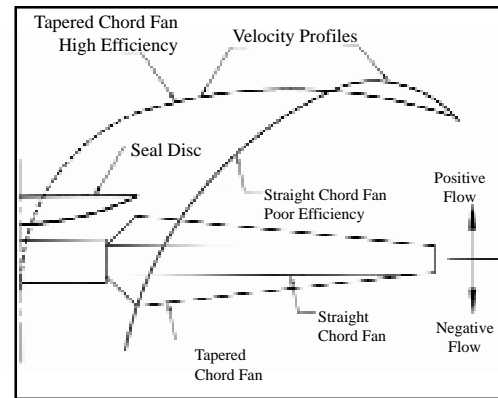
Velocity Profiles-Tapered vs. Constant Chord Blade
Fig. 12

Work preformed by a fan blade is basically a function of three factors at any point or radius:

• Chord width

• Airfoil twist

• Tangential velocity squared

At mid radius (0.25D), tangential velocity is only 25% of the velocity at the tip. To compensate for this decrease in velocity, the chord width and twist must be increased. This is the reason for the increased efficiency and more uniform airflow from a tapered blade. Note that for a constant chord width blade, exit velocity decreases rapidly inboard of the tip and typically becomes negative outboard of the seal disc. A typical aluminum blade's chord width and twist do not vary along the blade.

The advantages of a tapered, well-designed blade with even airflow across the fan result in higher efficiencies and lower horsepower to attain design performance.

6.10 Effect of Seal Disc

Note that the airflow direction becomes negative near the center of the fan. This is the result of the torque applied to the fan which creates a "swirl" effect on the air vectors. Air at the tips is axial or parallel to the fan axis. Toward the fan's center, air velocity decreases by the square of the radius and the air vectors lean further toward the horizontal. At or before the center of the fan, the air is actually moving opposite to the airflow at the blade tips. A well-designed fan will have a center seal disc of about 25% of the fan's diameter. Referring to Fig. 12, note that the air vectors at the center of the fan actually turn to a negative direction. Tests have shown that the seal disc prevents this negative airflow and improves fan efficiency by about 4 - 5%. The seal disc can be as large as 14 ft in diameter for 40 ft fans. Fig. 13 shows a typical seal disc.


Typical Seal Disc
Prior to Installation
Fig. 13

7.0 Variable Airflow Fans (Auto-Variable®)

Variable-pitch or Auto-Variable® fans automatically adjust the pitch angle to provide the precise amount of airflow for controlling process temperature while saving substantial amounts of energy. This section discusses basic operating characteristics, process temperature control methods, energy savings and economic comparisons with other axial fan airflow control systems.

Variable-pitch applications are typically 10 -14 ft (3.04 - 4.3m) in diameter consuming up to 40 horsepower (30 kW). Fig. 14 shows the typical type actuator system found in ACHEs. Fig. 15 shows the inverted actuator system popular in Europe for the ease of access to the actuator.
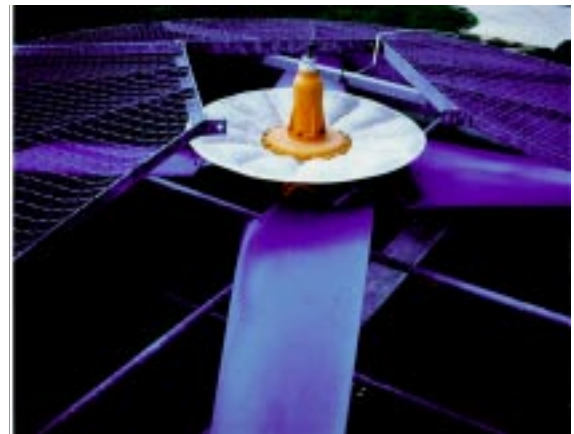

Typical Auto-Variable® Fan
Fig. 14

Inverted Actuator-Variable-Pitch Fan
Fig. 15

Operating and Control Characteristics:

Variable-pitch fan actuators used in ACHEs can be described in general terms as very similar to diaphragm-operated, spring-return valves. Instead of moving a valve stem up and down, the actuator controls fan blade pitch. A force diagram is shown in Fig. 16.
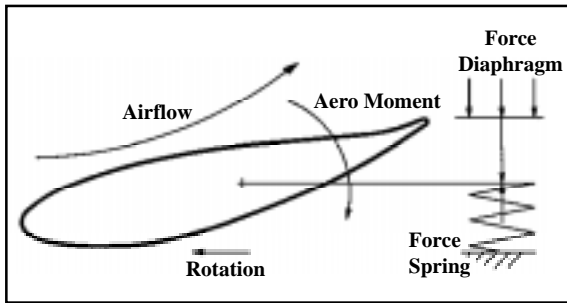

AV Force Diagram

Fig. 16

As the blade moves the air, the aerodynamic moment acts to turn the blade to a lower pitch. The hub spring creates an opposing moment to hold the blade in position. This is a fail-safe mechanism. If air pressure on the diaphragm fails, the fan operates as a fixed-pitch fan providing the maximum design airflow. Variable-pitch hubs can be configured to fail to the maximum or minimum pitch setting.

To reduce airflow, or even reverse airflow direction, air pressure is exerted on the diaphragm to oppose the hub spring and decrease the blade pitch.

When the blade pitch is about minus 10°, no work is done and essentially "zero" flow is attained. The minimum air velocity obtainable is approximately 50-100 fpm (0.37 - 0.51 m/s). If the hub has the pitch stops adjusted for reverse flow, air is directed downward and can be as much as 60% of the positive pitch airflow at the same horsepower.

A typical variable-pitch hub requires a 3 - 15 psi (21-103 kPa) control signal and operates the blades from the maximum design pitch back down to "zero" airflow. Most hubs are capable of 45° total pitch travel and perform as shown in Fig. 17.
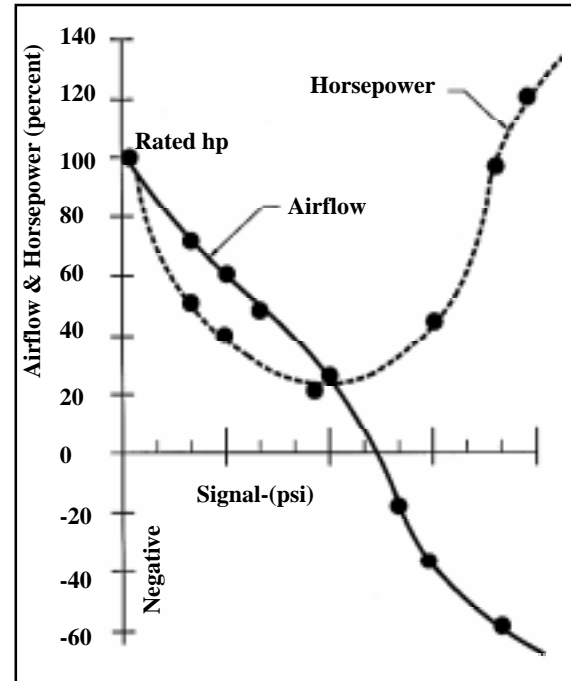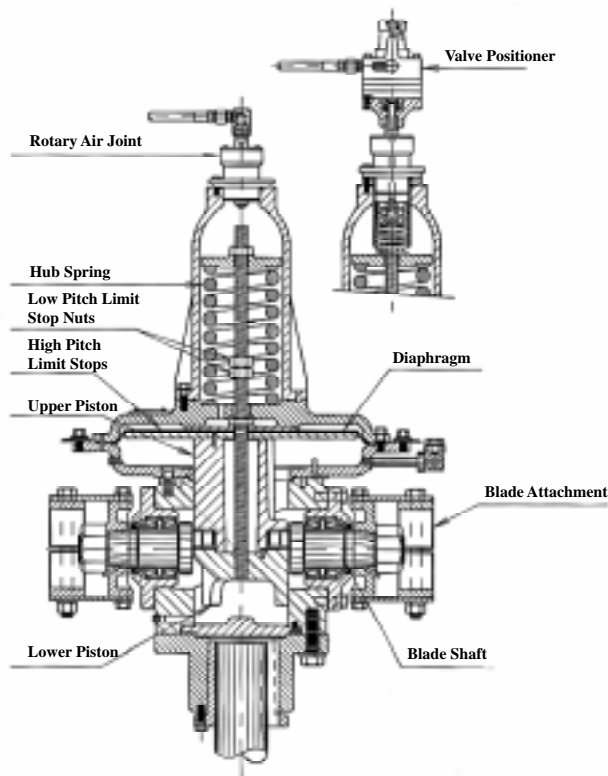

Airflow vs. Signal for Variable-Pitch Hub

Fig. 17

Typical Components:

A typical variable-pitch fan hub mechanism is shown in Fig. 18. The basic components are:

• Hub spring

• Diaphragm

• Piston

• Blade shafts with eccentric actuator

• Rotary air joint

• Valve positioner

Typical Auto-Variable® Hub
Fig. 18

The blade shafts or axles hold the fan blade and have an eccentric bearing on the inboard end. These eccentric bearings engage a groove in the piston. As the piston moves up or down, a twisting motion is imparted to the blades, changing the pitch. The rotary air joint is the static/dynamic interface between the rotating fan and the control air system.

Control Methods:

The valve positioner is a "closed-loop" feedback device that receives the control signal, (usually 3 - 15 psi) and supply pressure up to 100 psi (689 kPa). Valve positioners are used when airflow is critical, such as in ACHE condensing application.

The advantage of the valve positioner is it provides precise airflow control due to blade pitch position feedback and the ability to output high pressure on the diaphragm to quickly change pitch angle.

If airflow control is not critical, an "open-loop" system can be used operating on the 3-15 psi (21-103 kPa) signal alone. This system excludes blade position feedback to the controller and is useful in non-critical installations where the fan system will operate on 15 psi (103 kPa) diaphragm pressure. Operation with only 3-15 psi is generally limited to small fans.

Process Control:

There are several methods used to control process temperature in an air-cooled heat exchanger:

- Fluid bypass
- On-off fan operation
- Two-speed fans
- Louvers
- Variable speed fans
- Variable pitch fans

The oldest control method bypasses a portion of the process stream around the air cooler.

"On-off" fan control is simple and often used if there are a large number of fans in an identical service. However, this non-modulating control method can cause problems in air-cooled condensers such as "water hammer", freezing tube-to-header leakage, or tube buckling due to differential thermal expansion of bundles in parallel. Differential tube expansion can cause tube buckling. Cooling tower fans are a good example of incremental "on-off" fan control.

Two-speed fans are a further refinement giving 0, 67, or 100% of design airflow rate with 1800/1200 rpm motors. Naturally, additional motors increase the degrees of airflow control.

Louvers are the first step to modulated airflow; however, fan horsepower is wasted as airflow is throttled by the louver. At complete shut off, the fan stalls and horsepower actually increases.

Variable-speed fans for fully modulated airflow are available in two types: hydraulic and electric drive. Either type conserves energy and offers good airflow control. The latest development in electrical variable-speed control for fans is the variable-frequency drive (VFD). There are three basic types: VVI (Variable Voltage Inverter), PWM (Pulse Width Modulation) and CS (Current Source). These drives utilize a standard induction motor and automatic control is obtained by a process control device to interface the 4 - 20 milliampere temperature controller output with the VFD.

The older less common hydraulic drive system consists of a motor/variable volume pump/reservoir unit connected to a slow-speed, high-torque, direct-drive motor. Advantages are variable fan speed and eliminating a reduction belt or gear drive. The main disadvantage is inferior system drive efficiency. Efficiencies are approximately: motor 0.97, pump 0.92 and hydraulic motor 0.92. Therefore, the optimum drive system efficiency is (0.97)*(0.92)*(0.92) = 0.82, not counting hydraulic line loses. This must be compared to a typical motor/belt drive efficiency of 0.95 or 0.97 with a gear or timing belts.

Advantages of VFDs are reduced noise and vibration during slow-speed operation. Disadvantages are high cost per horsepower for control of a small number of fans in the

same service and total loss of service if a multiple fan VFD controller fails. Both hydraulic and VFD systems can be designed to operate in reverse airflow to negate the effects of natural draft.

The variable pitch fan can also provide from 0 - 100% positive or from 0 - 60% negative airflow at the same horsepower. Negative airflow is useful, along with louvers, in winterized ACHEs to seal off freezing outside air and recirculate warm air inside the plenum chambers. Internal recirculation systems utilize a positive and negative airflow pair of fans to recirculate warm air.

Power Consumption and Cost Comparison:

When evaluating power consumed by any axial fan operating at a fixed pitch, consider not only the fan performance curve but the system resistance characteristics as well.
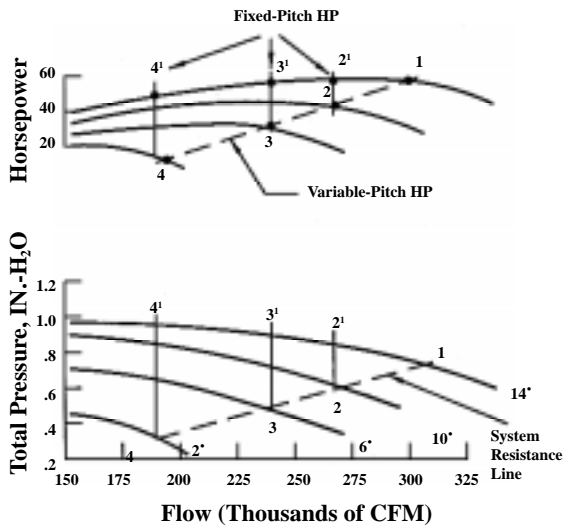
Fan performance curves are obtained in a wind tunnel with a means of varying the resistance or static pressure head against which the fan must work, and measuring resulting airflow and horsepower.

System losses are more difficult to determine, and judgement is necessary in many cases to evaluate the losses due to poor inlet conditions, excessive tip clearance and unusual structural conditions. If a cooling service is critical, it may be wise to model the unit and establish the system losses in a wind tunnel. What must be determined is the sum of the static pressure resistances versus system airflow. Since the fan's air delivery characteristics have been accurately established, the operating point for any airflow requirement can be predicted. This operating point is where the fan output in terms of pressure and flow, exactly meet system requirements and is referred to as the equilibrium point of operation.

Let's examine a typical case of an ACHE with a fixed-pitch fan and a louver for throttling the airflow at the design point and several other reduced airflow operating points.

For example, consider a 14 ft diameter fan operating at a 14° design pitch at point 1 (Fig. 19). The system resistance line, shown as the dashed line, is the locus of points obtained by summing the static and velocity pressure losses vs. flow through the bundle. The pitch angle line for 14° represents the fan's delivery characteristics. Assume density remains constant and the decreased airflow is a result of a reduction in required airflow.

If the louver throttles airflow to control the outlet temperature, the points shown as 1, 2^1, 3^1 and 4^1 in Fig. 19 are total pressure output and horsepower consumed by the fixed-pitch fan. If we were controlling airflow with a variable-pitch fan instead of a throttling device, total pressure output would exactly match the system resistance line as shown at points 2, 3 and 4, and the horsepower requirements are significantly reduced. In this case, the variable-pitch fan used 26, 51, and 73% of the horsepower required by the fixed-pitch fan at points, 2^1, 3^1 and 4^1 respectively.

Energy Comparison:

A typical case study to evaluate cost differentials between fixed-pitch (single-or dual-speed) or variable-pitch fans for air-cooled heat exchangers is conducted as follows:

Step 1. Thermal studies are needed to determine total airflow required as a function of ambient temperature for the particular application. Plot flow versus temperature.

Step 2. Obtain climatological data for the area where the air cooler will be located, and derive a table of "degree-hours" for incremental temperature ranges. This tabulates the number of hours per year that each temperature range occurs.

Step 3. For a fixed-pitch fan, flow is a direct function of speed. Create a plot of fan output versus ambient temperature for each case to be studied. This yields HP-hours per temperature range.

Once the airflow is defined for each range, fan horsepower for any point can be approximated by the following relation:

$$\text{Where: } HP_1 = \left(\frac{cfm_1}{cfm_{des}}\right)^{2.8} \left(\frac{\rho_1}{\rho_{des}}\right) HP_{des}$$

HP_1 = Horsepower at cfm_1

ρ_1 = Density at point 1 lb/ft^3

cfm_1 = ft^3/min flow at point 1

HP_{des} = Design horsepower

ρ_{des} = Design density

Throttled Flow Through Fan
Fig. 19

Step 4. Using HP-hours for each tempera-
         ture range, tabulate the total for the
         year. The power required is kW =
         HP (0.746).   Arrive at an energy
         cost per year for each scheme.

This is a very simple approach but a useful tool.
As an example, consider the following:

A 20 ft x 36 ft (6 m x 11 m), four-row, forced
draft, air-cooled heat exchanger with two 14
ft (4.3 m) fans each having 40 HP (30 kW)
motors.

The unit is a propane condenser, Item 126C
to be located in West Virginia, close to
Charleston. Power cost is $0.035/kW-hr.

Study power and initial investment costs for
one year for three schemes:

1. Two fixed-pitch fans.

2. Same, but with one 1800/900 RPM motor.

3. One fixed-, one variable-pitch fan (single-
    speed).

Design Conditions (per fan):

Flow   224,613 ACFM (106 m³/s)

Power   29.0 HP (21.6 kW)

Temperature   45°F (7°C)

Elevation   350 ft (107 m)

See Fig. 20 for a plot of air delivery vs. am-
bient temperature.
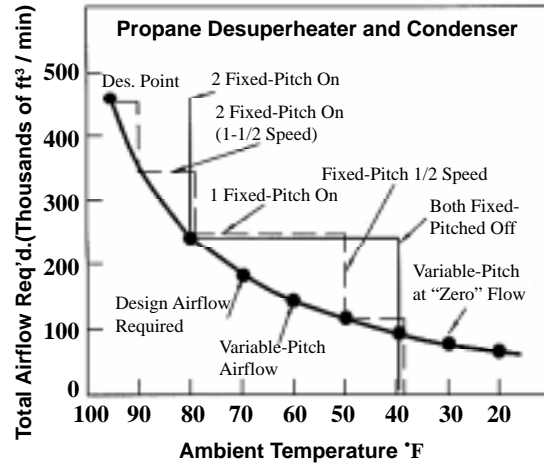
Air Delivery vs. Temperature
Fig. 20

Fig. 21 gives the power study results which
show that the variable-pitch fan would pay
for itself within two years. In this particular
application, the two-speed motor in Case 2
made only a small difference in energy cost.

Estimated Total kW-hr/yr	Energy Cost/yr $.035/kW-hr	First Costs
Case 1 162,670 2 Fixed-Pitch	$5,693	Base
Case 2 142,080 2 Fixed-Pitch 1-1/2 SP (Manual)	$4,973	+$1,000
Case 3 70,887 1 Fixed-Pitch 1 Variable-Pitch	$2,481	+$3,715(a)
(a) Includes temperature controller, variable-pitch cost adder plus estimated $1,500 installation.		

Power Study Results
Fig. 21

8.0 Variable-pitch vs. Variable-Speed

The choice between a variable-pitch and variable-speed fan depends on factors such as: (1) Initial costs and payback period, (2) availability of suitable fans, and (3) the number of fans performing the same duty.

Payback period is calculated from power savings that depend on the "degree days" at the site, power cost, required airflow vs. ambient temperature and installation cost. The $/HP cost of a variable-speed drive drops significantly for larger units.

For heat transfer purposes, an even airflow across a bundle is desirable, so the maximum number of fans should be operated by a single large controller. Variable-speed drives can exceed 1,000 HP.

A study by the U.S. Department of Energy of a typical cooling tower in Houston, Texas, operating with 100 million Btu/hr heat load and a 110-95°F cooling water range, revealed the following relative power consumptions:

No control, full speed   100% of base

Single-speed, automatic on/off 65% of base

Two-speed, automatic on/off   59% of base

Automatic variable-pitch fan   50% of base

The tower was equipped with four fans, each run by a 25 HP motor. The results typify the power savings possible in Houston, where annual dry-bulb temperatures vary from 30°F to more 100°F.

If airflow were controlled with a fixed-pitch fan and a variable-speed drive, rather than by a variable-pitch fan, results would be similar. Speed changing saves power because horsepower consumption is a cubic function of airflow, i.e., $HP_2 = HP_1 (CFM_2 \div CFM_1)^3$.

There is a slight advantage in controlling flow by varying speed rather than adjusting blade pitch angle, since efficiency of a variable-pitch fan declines with decreasing pitch angle.

In both cases, power consumption drops dramatically. Power savings of 50%/yr or more can be realized by replacing a fixed-pitch, continuously operating fan with one where blade pitch or speed is varied automatically.

9.0 Noise Considerations

Fan noise can be elusive requiring sophisticated equipment to measure accurately. Fan noise control must begin at the air-cooled heat exchanger or cooling tower design stage. Noise control as an after thought, can result in a very costly fan and drive component retrofit , and possible addition of heat transfer surface or acoustic barriers.

Noise is a function of fan tip speed. In a classic paper (ASME 72-WA/FE-42) Hopper and Seebold suggest that "the broad band noise varies approximately with the 5.6 power of tip speed, and directly with blade loading. The minimum operating speed that provides effective airflow is a function of the fan's pressure capability.

Since noise (Sound Pressure Level) is a function of the 5.6 power of tip speed, slowing fan rotation can reduce noise.

Unfortunately, a fan's pressure capability decreases with the square of the speed. Therefore, the fan's speed pressure capability must increase to maintain the required airflow.

To increase the pressure capability of a fan, the fan's solidity ratio must be increased, by adding more blades, or using blades with a wider chord, such as Hudson's low-noise blade design. Unfortunately, increasing the number of blades can reduce fan efficiency.

Noise requirements are often more restrictive at night. If this is a consideration, slowing the fan as ambient temperature drops using a variable-speed drive can be one solution to reducing noise. As the nighttime ambient temperature drops, required airflow is reduced, therefore fan speed may be slowed to lower noise levels.

9.1 Noise-Related Nomenclature

Decibel: A number representing relative sound intensity expressed with respect to a reference pressure or power level. The usual reference for sound pressure level is of 20 micro newtons per square meter (20 μN/m^2). A decibel is a logarithm (base 10) of a ratio of the power values abbreviated "dB".

Frequency: Sound vibration rate per second in Hertz (cycles per second).

Low-Noise Fans: A fan able to operate at low speed due to its high-pressure capability. Fan pressure capability is a function of its solidity ratio. Therefore, a low-noise fan will generally have more or wider blades than would be required if the fan operated at normal tip speeds.

Octave Bands: Noise is categorized by dividing it into separate frequency bands of octaves or 1/3 octaves. Generally, we use 63, 125, 250, 500, 1K, 2K, 4K and 8K center frequencies to define these bands in Hertz (cycles/sec).

Sound Power Level: Acoustical power (energy) can be expressed logarithmically in decibels with respect to a reference power, usually 10^{-12} watts. The relationship is given as: Sound Power Level PWL $= 10 \log \left(\dfrac{W}{10^{-12} watts} \right)$.

Sound power level cannot be measured directly but must be calculated from sound pressure levels (SPL) dB. In metric terms, this is known as Lw.

Sound Pressure Level: Known as SPL, or Lp in metric terminology, is the audible noise given in decibels that relates to intensity at a point some distance from the noise source. It can be related to octave bands or as an overall weighted level dB(A).

Weighted Sound Levels: This relates the decibel (loudness) to a frequency. Ears can easily pick up high-frequency noises (both intensity and direction) but are relatively insensitive to low-frequency noise. For a stereo system high-frequency speakers must be very carefully located in a room for best results but low-frequency bass speakers can be placed anywhere, even out of sight.

There are three basic weighting systems: A, B and C. The "A" system, dB(A), most closely relates to our ear, the "B" system, dB(B), has some specific uses and the "C" system, dB(C), is considered unweighted.
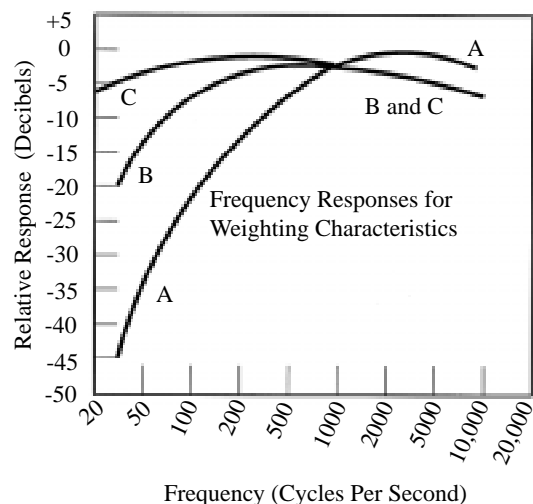
Weighted Sound Levels
Fig. 22

The dB(A) is the most common weighting system. It expresses sound levels with a single number instead of having to specify the noise of each octave band.

Note that the sound range of ACHEs (at close range) is typically between 80 and 105 dB(A).

9.2 Allowable Fan Noise

To establish noise limitations, decide on the maximum allowable noise from the ACHE or cooling tower.

9.2.1 ACHE Noise

If the concern is for overall plant noise or the noise exposure of plant workers in the vicinity of the fans, a different type of specification must be used.

Overall, noise limitations from an ACHE are typically a sound power level (PWL) specification for each bay. This limits the contribution of each bay (typically two fans) to the plant noise as a whole. This is usually needed where noise at the plant boundary is considered. Contributions of each part of the plant must be carefully controlled if overall plant noise is limited. PWLs can be expressed as weighted level dB(A) or sometimes even by limitations on each octave band.

If worker protection is the main concern, a limitation of sound pressure level at 3ft or 1m below the bundle will probably be imposed as "SPL dB(A) at 3 ft". The OSHA limitation is 90 dB(A) for eight-hour exposure, but 85 dB(A) down to 80 dB(A) is becoming more common.

9.2.2 Cooling Tower Noise

The primary difference between cooling tower and ACHE noise is falling water noise in a cooling tower at grade elevation, which also affects noise levels at the plant boundary. Motor and gear noise must also be considered.

Fan designers normally do not include these factors when establishing fan noise. Cooling tower manufacturers will combine the contribution from all noise sources when determining the overall rating of a tower design.

9.3 Predicting Fan Noise

Each manufacturer has proprietary equations for predicting fan noise. API Guidelines use the general formula:

$$PWL = 56 + 30 \log \left(\frac{TipSpeed}{1000} \right) + \log HP$$

This calculates PWL as dB(A)

Hudson's proprietary noise equations (used in the Tuf-Lite program) are based on actual tests at various speeds and operating conditions considering the following effects:

- Fan diameter
- Fan tip speed
- Blade Type
- Blade pitch angle
- Inlet conditions
- Horsepower

Note: Logs are common logs (base 10).

For example:

14 ft fan
237 RPM (10, 424 FPM tip speed)
25.1 HP

Find sound power level

PWL = 56 + 30 log 10.4 + 10 log 25.1
      = 100.5 dB(A)

Fan Noise Profile

It is valuable to consider the noise pro-file of typical Hudson fans. Consider the fan from the previous example with a PWL of 100.5 dB(A). The profile in decibels would be constructed as:

From testing the noise of numerous fan configurations, Hudson knows that fan noise is predominant in the 63-250 Hz bands and mechanical noise dominates from 500 to approximately 2K Hz. Mechanical noise is generated by the drive systems, motors, bearings and induced vibrations.

Some Important Concepts

- When considering multiple noise sources (fans) use the relation:

  $$PWL_N = PWL + 10 \log N$$

  The sound power level for 2 adjacent fans is the PWL of one fan plus 10 log 2 or $PWL_2 = PWL + 3$.

A doubling of the noise source adds 3 dB.

- Noise attenuates with distance by the equation:

  $$SPL \text{ (at distance R)} = PWL - 20 \log R$$

  Where R is in feet from the center of the source. Measure R as a "line of sight" distance.

Consider the noise an observer at grade hears at 50 ft from an operating ACHE with the fan on where the line-of-sight distance from grade to the center of the fan is actually 62 ft. Remember what the ear hears is SPL, the noise energy is PWL. Assume PWL = 100.5 dB(A).

$$SPL = 100.5 - 20 \log 62 = 64.7 \text{ dB(A)}$$

This also assumes background noise is at least 10 dB quieter.

Note: If both fans were running, the SPL would have been 67.7 dB(A).

Noise at 3 ft Beneath the Unit

When considering the noise at 3 ft beneath the unit, the drive system and motor noise become dominant at lower tip speeds.

Factors that influence this noise are:

- Motor noise
- Belt or gear noise
- Bearing noise
- Reflected noise from supports
- Background noise

Gear noise is especially significant in a forced draft unit.

9.4 Noise Testing

Frequently, the ACHE must be tested for confirmation that its noise does not exceed specifications imposed by the purchaser. There are two basic types of tests normally performed before shipment:

a) Measure SPL dB(A) (Sound Pressure Level) at "3 ft. below the bundle or fan guard" - depending on whether the unit is induced or forced draft.

b) Measure PWL (Sound Power Level) using "hemispherical power level test." PWL is specified as either a dB(A) weighted value or by octave bands.

The "SPL at 3 ft" test is by far the most common and least expensive. Usually, only one or two measurements are required. The answer is immediate and read directly from the noise meter.

The hemispherical test is far more complicated and expensive. Several hours, many technicians and a large crane are required to perform this test. Full details of the test are given in API Recommended Practice 631 M, issued in June, 1981. The test consists of measuring SPL at 9 points on the surface of an imaginary 10 m radius hemisphere and 4 points on an imaginary 10 m radius cylinder. The cylinder height is from ground level to the geometrical center of the ACHE. Since the unit is typically mounted on beams 8-10 ft. off the ground, the geometrical center falls 12-15 ft off grade. The crane is required to transport the technicians and noise meter to reach the top points of the hemisphere. The test consists of measuring noise at each point (described in API 631 M) with the fan on and off so that background noise can be determined. After all data is taken, a standard procedure calculates basic fan noise energy, which is the PWL (Sound Power Level). dB(A) and each octave band noise are measured.

This type of test is performed when the noise contribution from the entire bank of ACHEs is required for plant noise calculation.

For cooling towers, noise specifications are generally concerned with the SPL in the vicinity of the fan stack or at some point distant from the tower itself. These tests can be done if background noise is less than 3 dB of the specified level. For SPLs near the stack, water noise is generally not a concern. For distant tests, falling water noise is a major concern beyond the scope of the fan manufacturer. For cooling tower noise tests, gear and water noise is not included in the fan manufacturers fan noise estimate.

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Monroe, Robert, CTI Paper TP74-03, "Cooling Tower Fans - Today and Tomorrow", Jan. 1974

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API Standard 661, Fourth Edition, "Air-Cooled Heat Exchangers for General Refinery Services", Nov. 1997

API Standard 631M, First Edition, "Measurement of Noise from Air-Cooled Heat Exchangers", June 1981

DEDICATION

The Basics of Axial Flow Fans is a compilation of work performed by many Hudson Products' engineers and employees. However, one individual deserves special recognition.

Mr. Robert C. Monroe, who worked tirelessly for Hudson from January 28, 1967 to his retirement on August 1, 1997, was not only a major contributor to the contents of this book but also to the success of Hudson Products Corporation in manufacturing quality fiberglass-reinforced, plastic fan blades. Bob's expertise, vision and hard work were instrumental in Hudson's development of the Tuf-Lite® and Tuf-Lite II® families of superior fan blades.

The Basics of Axial Flow Fans is dedicated to the memory of Mr. Monroe who passed away on May 31, 1998.

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