Improving Cooling Tower Fan System Efficiencies

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After a look at the problem for air cooled heat exchangers and cooling towers using axial fans, ways to improve system efficiencies in three areas are discussed: before the fan system design is finalized, improvements in the physical equipment as installed, and recognition of performance problems caused by adjacent equipment. Results of a full-scale test illustrating fan efficiency contributions of various components are discussed.

Although not commonly thought of as a Turbomachine, an Axial Fan is a Turbomachine in *reverse.* Webster says a turbine is a "rotary engine actuated by the impulse of a current of fluid subject to pressure and usually made with a series of vanes on a central rotating spindle." My discussion concerns devices that impart energy to the current of fluid . . . axial fans. More specifically I will discuss the things that rob the fan system of efficiency, and how they can be improved. Some of these principles can be applied to the turbomachine system as well to obtain the maximum amount of work by expending the least amount of energy. As we all know, conserving energy today is the "name of the game."

Fan Systems

In this discussion I will limit the scope of two types of fan systems: those used in Dry Cooling Towers (air-cooled heat exchangers) and Wet Cooling Towers.

Each of these devices are used to transfer heat and both have several things in common:

1. Both have an axial fan to move the air.

2. Both have a shroud to contain the fan and funnel air into the fan.

3. Both have plenums into which the air is directed so that heat can be transferred by direct or indirect contact. In the case of air coolers or radiators the contact is indirect while in cooling towers the air comes in direct contact with the heated water.

Fan System Efficiency. When we design an air-moving device one of the most important tools we use is the fan performance curve. Using this curve of fan performance we plot a system resistance line to establish an operating point at which the fan performance exactly matches the system requirements.

From the operating point we can define the fan pitch and power requirements. With almost any fan the pitch can be changed from the original estimate, if airflow is too low, to a higher pitch and greater flow. However, if our system efficiency or losses are not as assumed and we need more air, horsepower increases by the cube of the flow. A ten percent increase in flow requires a thirty-three percent increase in horsepower.

Fan performance curves generally are obtained under ideal, reproducible, conditions. The Engineering Test Lab at Texas A&M's Research and Extension Center is the only independent test laboratory in the country with an AMCA certified wind tunnel. The lab tests everything from kitchen ventilators to scale model 60 ft. diameter fans. The test conditions for high performance axial fans usually require blade tip clearance on a five foot model of about .040 of an inch with a large inlet bell, conditions as ideal as possible. As a result of good aerodynamic design and minimized losses, Total Efficiencies are generally in the 75 to 85% range.

However, from experience with many full-scale fan tests it is rare that "real life" performance exceeds 55 to 75% total efficiency. The difference is in "Fan System Efficiency." Although the fan efficiency is exactly the same, the *system* efficiency is greatly different.

To refresh your memory as to terminology, the head or total pressure that an axial fan works against is made up of two components. These are static pressure which is the sum of the system resistances and velocity pressure which is a loss associated with accelerating the surrounding air from zero to the design velocity. The only useful work done is by the static pressure component. This is measured in inches of water and an axial fan normally works in the regime of 0 to 2 inches total pressure. Air Horsepower is calculated by:

$$HP_{Air} = \frac{\text{Tota Pressure} \times \text{CFM}}{6356 \times \text{Efficiency}} \tag{1}$$

where Total Pressure is in IN-$H_2$0, flow is in Cubic Feet per Minute.

Assume we have to design a forced draft air cooled exchanger required to move 200,000 CFM of air, operating against a system static pressure of 0.42 inches of water. Assume the air temperature is 70° and the elevation is sea level. From fan performance curves we choose a 14 foot diameter fan for this job and find 21.0 Brake HP would be required. Using the equation for Air Horsepower we further calculate that the Total Fan Efficiency at this operating point is 87%, a very respectable number. Incidentally, the Evaluated Horsepower on this unit was very low and won the contractor an order. The air cooler was constructed using these numbers and a 25 HP motor was installed. This would seem to have an adequate "pad" even allowing say 5 percent for drive losses. This gives: 25 – (21 x 1.05) = 2.95 Hp excess or a 12% pad on horsepower. Figure 1 shows the Fan Operating Point.

Alas, the air cooler is put on stream and we find its capability is sadly insufficient, requiring expensive field modifications. Let's look at what most likely caused the problems: Figure 2 shows tip losses, reverse flow at the hub and recirculation loss.

The sum of these losses, if neither eliminated nor provided for, can easily reduce fan efficiency 20 percent, as we will see later, so the real fan efficiency was nearer 67 percent and the design horsepower was 27.2, not the ideal fan curve horsepower of 21. Since only 25 HP was installed, the user will have to either improve system efficiency or install a larger motor to meet design duty.

This is what Fan System Efficiency is all about and the subject of this paper. Wet cooling towers have still other aspects, which will be discussed.

In the following examples a forced draft air cooler is often cited. Another type, an induced draft air cooler with the fan above the bundle, is also in wide use. It has more advantages but problems of a different nature. Induced draft fans are in the hot exit air which may create problems with maintenance although there are several other offsetting advantages over forced type units. Each type unit would be subject to the fan tip losses and fan hub seal losses. However, the major problems of inlet conditions to the fan ring and hot air recirculation are magnified by the high inlet velocity to the fan and the low exit velocity from the bundle of a forced draft unit.

System Losses-The Holes in the Bucket. Potential losses in system efficiency occur in several areas:

a) Losses caused by the fixed system design rather than by variable physical properties. Once the operating point of the fan is fixed these losses are built-in and cannot be easily detected or corrected. They are losses because they rob the system of potential efficiency. Examples of this type of system "loss" would be: choice of fan design, fan diameter selection, fan design operating point.

b) Losses caused by "variable environmental properties" would be: lack of fan hub seals, excessive fan tip clearance, poor inlet conditions of the fan ring or stack, excessively high approach velocity to the fan, or random air leaks in the fan plenum. Often allowances for losses in louvers, bug screens, recirculating ducts, and steam coils are simply omitted in design.

c) Other performance losses could occur because of hot air recirculation.

Of the above losses, the only easily corrected problems would be in category b) which we call "variable environmental properties."

In the following discussion category a) will be covered in The Fan Itself. Category b) will be discussed in The Fan Housing and c) will be covered under Unwanted Air Movements.

(a) ***The Fan Itself.*** The ways a fan system could be inefficient are sometimes obvious but most of the time they are not. For instance, the *blade design itself is* a major factor. Modern axial fans are usually made by extruding aluminum or molding fiberglass. Extruded aluminum blades are by nature always of uniform chord width while molded fiberglass blades can have an irregular shape. See Figure 3, Fan Blade Shapes. One of the basic design criteria for blade design is to produce uniform air flow over the entire plane of the fan. One of the aerodynamic principles involved is that the work done at any radius along the blade is a function of blade width, angle of attack and tangential velocity *squared.* The "angle of attack" in airfoil design dictates the amount of blade twist required at any particular radius along the blade.

It follows that as a point on the blade decreases from tip toward the hub the tangential velocity sharply decreases and in order to produce uniform airflow, the blade width and twist must increase. If the blade chord cannot increase in width, the twist must be increased to compensate. With an extruded blade the twist is created by mechanically yielding the blade to a prescribed degree. Due to limits in elasticity only limited twist can be created. In a molded blade there is no such limitation to chord width or twist so the "ideal" blade can be more closely approached.

The point is, that the blade design itself can create problems of non-uniform air flow and inefficiency.

Another inherent property of an axial fan is the problem of "swirl." Swirl is the tangential deflection of the exit air direction caused by the effect of torque. The air vectors at the extreme inboard sections of the blade actually reverse direction and subtract from the net airflow. This is a very measurable quantity. An inexpensive hub component, the *Hub Seal Disc* prevents this and should be standard equipment on any axial fan.

A real example that illustrates performance differences due to blade shape and seal disc usefulness is shown in Figure 4, Results of Air Flow Test. This data was obtained by a major cooling tower manufacturer who carefully measured air flow magnitude and direction across a blade in a full scale cooling tower. Curve "A" shows the performance of an extruded type blade with no hub seal disc. Curve "B" shows performance of a tapered fiberglass blade with a seal disc. Both 20 ft. diameter fans were tested under identical loading conditions of horsepower and speed. Note that significant negative air flow occurs at approximately the 40 percent chord point on the straight blade but no negative flow was found with the tapered blade.

Another component of the fan system efficiency would have to be the fan *Operating Point.* By this I mean the point where the system resistance line meets the fan performance line. This would be the particular blade pitch angle that produces the desired air flow against the required system resistance. This pressure capability and flow is a function of the fan tip speed. For a certain fan speed, only *one* pitch angle will satisfy the system design operating condition. This fan operating point will have a discrete efficiency. However, efficiency varies as much as 10-15 percent from pitch angle to pitch angle and even along the usable portion of each pitch. By usable I mean the portions of the curve beyond the "stall" condition. This "stall" condition is easily discernible on the fan curve and is analogous to cavitation in a pump.

The point here is that, within limits, the fan speed can be varied so that a pitch angle can be selected which will *optimize fan blade efficiency* and will satisfy the required system resistance. Often it would be desirable to slow the fan down to attain a higher, more efficient operating pitch angle as an operating point.

This also has a side benefit of reducing noise and vibration because normally the lower pitch angles which appear obvious choices to handle the duty have lower efficiencies.

An example of this is shown in Figure 5, Optimized Fan Blade Efficiency.

Still another aspect of system efficiency is the proper selection of the Fan Diameter for any given conditions, *operating* and *economic.* There are several things which influence the choice of fan diameter.

1. Air Flow Range
2. "Fan Coverage"
3. Optimum Cell Size
4. Evaluated Horsepower
5. Standard Sizes Available

Of these, the most logical influence is that the fan must provide the amount of Air Flow required for any duty in a sensible operating range. A quick look at any vendor's fan curve will yield several sizes of fans to do any particular job. A poorly sized fan will waste horsepower at the least and fail to do the required duty at the worst.

For proper air distribution we suggest that the area of fan coverage shall not be less than 40 percent of the bundle face area, for induced draft units, 50 percent for forced draft units. This is a simple means to define the minimum fan diameter for dry cooling towers.

For wet cooling towers, and recently for dry towers as well, the optimum cell size and evaluated horsepower comes into play. Both are purely economic considerations. Optimum cell size is obviously matching fan size to minimized construction cost per cell. The Evaluated Horsepower is increasingly becoming the major factor in deciding fan diameters. E. H. is a "dollars per horsepower" penalty added to a bid which is a measure of the operating costs of that design over the capitialized life of that particular tower. Evaluated Horsepowers of $550/HP to as much as $2,500/HP are becoming common. The significance of E. H. is that very frequently the difference in evaluated horsepowers of several fan selections can exceed the cost of the fan by many times.

For instance consider the recently requested fan selection for the following:

Required:	102 fans, 16 feet diameter
Air Flow:	600,000 CFM/fan
System Actual Static Pressure:	0.38 IN-H_20
Density Ratio:	0.823
Horsepower Evaluation:	$827/HP
Find:	Fan horsepower and number of blades

For this application both 16 ft., 20 ft. and 24 ft fans were studied. Table 1 is a summary of the results.

In each case, the total air flow and fan efficiency were the same. By using the larger fans considerable savings can be attained. Although in the above example efficiency was held constant for each selection, additional savings in horsepower could be achieved by manipulating the fan speed and the operating point to optimize the fan efficiency for each size.

It is the variation of the Velocity Pressure Loss at each fan's operating point which greatly effects the required horsepower. The Velocity Pressure Loss is a fixed loss in every fan which reflects the energy used to collect the air into the throat of the fan. It is dependent on the Net Free Area of the fan and not on the exact entrance conditions which will be discussed later.

In reviewing the potential losses in efficiency in the fan itself we have discussed two inherent losses that were built-in to the system by design:

1. Poor fan blade design
2. Poor selection of operating point

We also discussed the factor of optimized diameter which was decided economically before the air moving device was built.

The two factors which could be physically modified to reduce fan system losses would be the addition of the Hub Seal Disc and the revision of the fan operating point to a more efficient condition, although a change in the number of blades or gear ratio might be required for the latter.

(b) ***The Fan Housing.*** The components that make up the fan housing would be considered a Fan Ring for air coolers or a straight or Velocity Recovery Stack for cooling towers.

The most important system loss for both types would be the air leakage around the tips of the fan blades. This loss is a direct function of the *Tip Clearance* with the ring or stack and the Velocity Pressure at the operating point.

This leakage is caused by the tendency of the high pressure exit air to recirculate around the tips into the low pressure air in the inlet. Figure 6 shows the effect of excessive tip clearance on a small fan at one flow.

The loss takes the form of reducing the Total Efficiency and Total Pressure capability of the fan.

Tests were performed on a 5 ft. diameter fan with an initial clearance of 0.047 inches per side. The clearance was progessively increased to 0.188, 0.375 and finally 0.464 inches per side.

A typical 14 ft. diameter fan with 0.75 inches tip clearance would relate to 0.27 inches or a 28 ft. fan with 1.0 inch tip clearance relates to 0.18 clearance on a 5 ft. diameter fan.

Inlet Conditions

There are several areas where inlet conditions can seriously affect the fan system. For instance, the inlet condition to the Fan Ring for a forced draft air cooler, the inlet condition *into* the Velocity Recovery Stack of a wet cooling tower or the *approach area* itself to the whole air cooler or cooling tower.

The most obvious case is the inlet condition to the fan ring. If we relate to fluid flow, we have all seen the classic effect of the "Vena-Contracta." The effect takes place at the inlet to a fan ring as was shown in Figure 2.

In testing of a forced draft air cooler at 0.2 inches-$H_2 0$ Velocity Pressure, it was found that a loss of 26 percent Total Pressure and 18 percent Efficiency occurred without the use of an Inlet Bell. These losses are not constant but vary with Velocity Pressure.

Velocity Recovery Stacks. In the case of Wet Cooling Towers, a relatively common means of improving inlet conditions and conserving horsepower is known as a Velocity Recovery Stack. These stacks incorporate a slightly tapered exit cone and a well-rounded inlet bell. In theory, there is a significantly reduced Velocity Pressure at the exit compared to the plane of the fan. Since the quantity of air is the same as both planes, the recovery of Velocity Pressure is converted into "static regain" which lowers the Total Pressure requirements of the fan, thus saving horsepower.

Certainly, any axial fan with a Velocity Pressure of 0.3 inches $H_2 0$ or greater can benefit from a V.R. Stack. On some air-cooled units, where recirculation could be a problem, high exit velocities may be desirable so V.R. Stacks would not be recommended. However, they definitely should be considered for wet cooling towers. Figure 7 shows the effect of a V.R. Stack and its potential savings of horsepower from a system efficiency standpoint. Omission of a V.R. Stack would be a loss. Note that the V.R. effect is more pronounced at the higher Velocity Pressures and the horsepower saved at higher flows is very significant.

The entrance *into* the V.R. Stack through the fan deck should not be ignored as often, it in itself creates turbulence and losses. Although the Stack design usually incorporates a generous inlet radius, a sharp corner through the fan deck or heavy structural members beneath can sometimes negate the smooth air flow condition in the stack itself.

Since the user has no control over this condition, once the tower is constructed, these comments serve only to point out the many opportunities for deterioration of system efficiency.

Approach Velocity Consideration. Sometimes the economics of structural costs may unintentionally create very serious effects upon the system performance of heat exchangers. Consider the case of a small air cooler with short columns to allow easy maintenance or a cooler conveniently placed next to a building because of space problems.

As with inlet losses to the fan, the magnitude of the loss is a function of the Velocity Pressure which itself is a function of air velocity. It is considered good practice to insure that the air velocity at the entrance to the fan is no more than approximately one-half of the velocity through the fan throat. To illustrate: consider a 14 ft. diameter fan mounted so that the bottom of the fan ring is 5 ft. from the ground. The entrance area is considered the area of an imaginary plane from the bottom of the ring to the ground. If the air flow through the fan was, say 250,000 CFM, the velocity would be roughly

$$\frac{250{,}000\ \text{ft}^3\ /\ \text{Min}}{.785(14)^2\ \text{ft}^2}\ \text{or } 1{,}625\ \text{ft/min}$$

The entrance area would be π x 14 x 5 or 200 ft^2 giving an approach velocity of 1,136 ft/min. This would cause additional system losses not incorporated in the basic Velocity Pressure computation. Figure 8 shows a rule of thumb, which can be used to compute the approach velocity. The approach velocity should not exceed one-half the velocity through the fan.

(c) Unwanted Air Movements. There are often cases where in order to increase performance, you need to reduce air flow. These are cases where the warm exit air flow recirculates to the inlet side of the fan and decreases the mean temperature difference between the cold entering air and the hot fluid temperature inside the tubes or in the fill thus lowering efficiency of the air cooler or cooling tower.

The main factors which influence the tendency to recirculate are primarily inlet or approach velocity, exit velocity and velocity of prevailing winds. Gunter and Shipes[1] have formulated a simple analytical method to predict recirculation in an air cooler utilizing the above parameters.

The primary causes of recirculation could be summarized as follows:

1. Excessively high approach velocities.

2. Units placed in line with the prevailing wind direction.

3. Units placed at elevations so that the exit of one is upstream of the inlet of the adjacent unit.

4. Low exit velocities, such as those encountered in forced draft air coolers.

Severe performance problems can result if recirculation is encountered.

Recirculation can be confirmed by smoke testing and by temperature surveys of the exit and inlet air streams to a unit. To eliminate recirculation it is usually necessary to increase the exit air flow or change the elevation of the exit flow by adding straight sided fan stacks. In some cases baffles may have to be constructed.

In cooling towers the effect of the Velocity Recovery Stack is to reduce the exit air velocity which could promote recirculation. It may be necessary to utilize straight stacks to jet the hot exit air further away from the approach or inlet areas.

Air Leakage. This is another category of unwanted air flow.

Air leakage could occur in an air cooler at several places which lower system efficiency:

- Ineffective or missing tube bundle air seals along side frames, between tube bundles or at the ends of tube bundles.

- In older units plenums could be rusted out causing holes and loss of effective air flow over the bundles.

In a cooling tower you could have:

- Missing panels in the casing

- Holes in the fan stacks

- Missing boards or holes in the fan deck

The net result of these problems is that the air movement intended to go through the tube bundle or fill takes the path of least resistance and consumes power but does no work.

Fan Tests

To illustrate the negative effects on fan system efficiency we have discussed, a series of full scale fan tests were performed.

The basic scheme was to test a forced draft air cooler at three different air flow rates in four conditions each:

(a) Standard (with Inlet Bell, Seal Disc, and Close Tip Clearance)

(b) Remove Inlet Bells only. Test unit and replace Inlet Bells.

(c) Remove Seal Disc only. Test unit and replace Seal Disc.

(d) Increase blade tip clearance.

A total of twelve tests were performed.

Test Equipment. A 20 ft. x 32 ft., four row forced draft air cooler with two 14 ft. diameter fans was tested. Modifications were made to the same single fan only. The fan operated at 10,000 FPM tip speed and was equipped with a 30 HP Reliance 1,160 RPM motor. The finned section was a typical 1" O.D. — 10 fin per inch extruded finned tube bundle. The unit was equipped with both steam coils and louvers which were locked in an open position during the test period.

The testing equipment used included the following:

Taylor Model 3132 Anemometer
Draft Gauge
Tachometer
Westinghouse Model PG-101 Power Analyzer

Procedure. For each test, air flow (CFM), static pressure, temperature, and electrical power consumed was measured. Electrical measurements included volts, amperes, watts, and power factor. Electrical power input was calculated by the relation:

$$HP_{in} = \frac{V \times A \times P.F. \times 1.732}{746} \tag{2}$$

Velocity Pressure was calculated by:

$$VP = \left(\frac{\text{CFM}}{\text{Net Free Area} \times 4005}\right)^2 \text{IN-H}_2O \tag{3}$$

System Efficiency was calculated by:

$$E = \frac{\text{Total Pressure (act.)} \times \text{CFM}}{6356 \times \text{HP}_{\text{in}}} \tag{4}$$

Thus, the effect of only one variable was investigated for each of three flows which were at 0.061, 0. 10 and 0.13 inches velocity pressure.

Discussion of Results. Table 2 shows a comparison between curve fan efficiency and the tested *system* efficiency.

Tests 1 and 2 showed a 10-15 percent decrease from curve efficiency as might be expected. Test 3 showed a 30 percent decrease from curve efficiency which was surprising. Full scale testing at best cannot achieve accuracy or repeatability better than about plus or minus 5 percent. The effects of ambient winds during the test period are by far the biggest cause of error. Variations in velocity and direction during the test period cause most problems while objects around or on the test unit create eddy currents of wind with corresponding high and low pressure areas. The Total System Efficiency (?) was considered "base" performance for the tests that followed.

Considering the base performance in each case was 100 percent, let us examine the effect of each variable in turn. Figure 9 shows the negative effect of only one variable for each test point with the resulting decrease in base system efficiency. As you will note, each plays a significant role in performance.

In reviewing the results shown, it can easily be seen that the negative effects that rob system efficiency are a function of the Velocity Pressure. While not demonstrated on this test, previous tests have shown also that the effects of the three parameters studied are indeed cumulative. That is, the total decrease in performance will be the sum of each individual effect. Thus we can see that the negative effects within the scope of this study would decrease the base performance of this test fan by magnitudes of 15 to an astonishing 58 percent. Keeping in mind the previous decrease in "base" system performance from the idealized "curve" system performance, this should point out the importance of considering the real system efficiency.

Conclusions

From the previous discussion the most important points worth emphasizing could be summed up as follows:

1. In any real life fan system there are inevitable losses that degrade system performance below that of the idealized curve performance. These should be taken into consideration.
2. Some losses are built-in by poorly designed fans or system designs that are not optimized.
3. Some losses are correctable by inexpensive standard components.
4. It is very important that an analysis is made of the complete fan system so that fan *system* efficiency can be computed. To do this complete information must be furnished from the supplier of the equipment for static and velocity pressure losses for each component in the system.

References

[1] A. Y. Gunter and K. V. Shipes, Hot Air Recirculation by Air Coolers, Twelfth National Heat Transfer Conference, Tulsa, Oklahoma, 1971.

Table 1 Fan Selections Based on Evaluated Horsepower

FAN TYPE	NO. FANS REQ'D.	TOTAL DES. H.P.	EVALUATED H.P. (THOUSANDS OF $)	EHP RATIO
16 FT. -8	102	12,842.	10,620.	+ 2.04
20 FT. -6	102	7,446.	6,158.	+ 1.18
24 FT. -8	68	6,304	5,213.	BASE

Table 2 Comparison of Cue Fan Efficiency and Tested System Efficiency

		CURVE FAN ?	TOTAL SYSTEM ?
TEST 1	14° Pitch	0.803	0.707
TEST 2	8° Pitch	0.854	0.712
TEST 3	3° Pitch	0.860	0.586

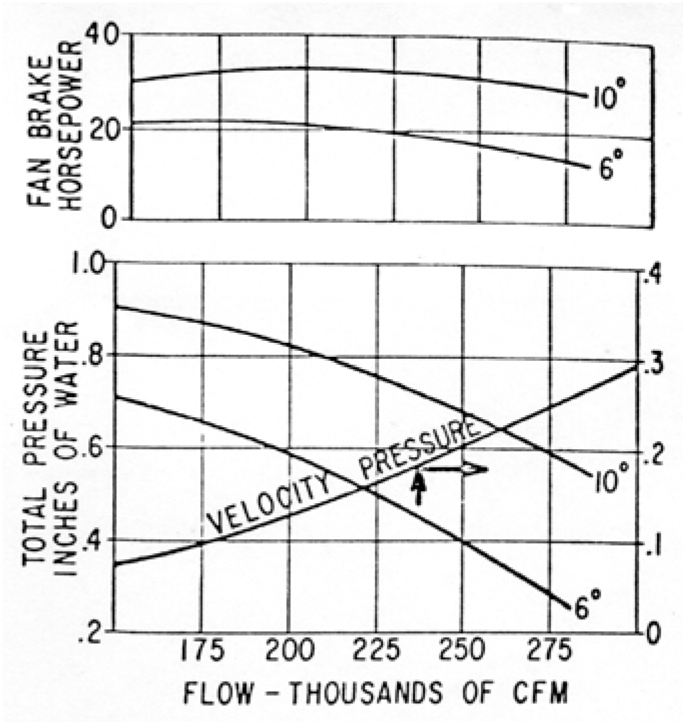

Figure 1 Fan Operating Point.

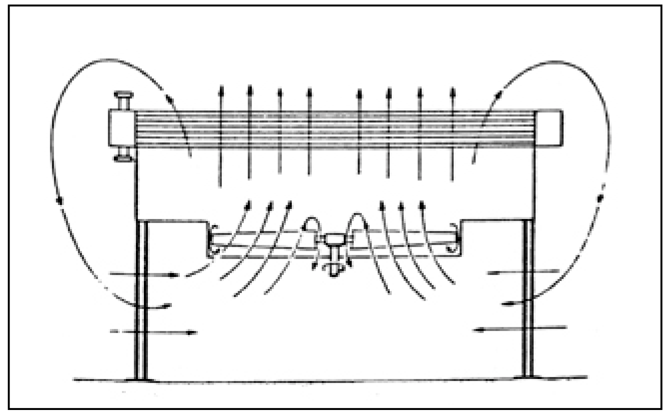

Figure 2 The Problem.

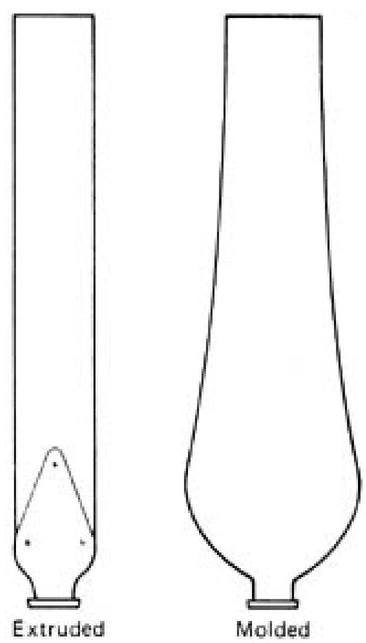

Figure 3 Fan Blade Shapes.

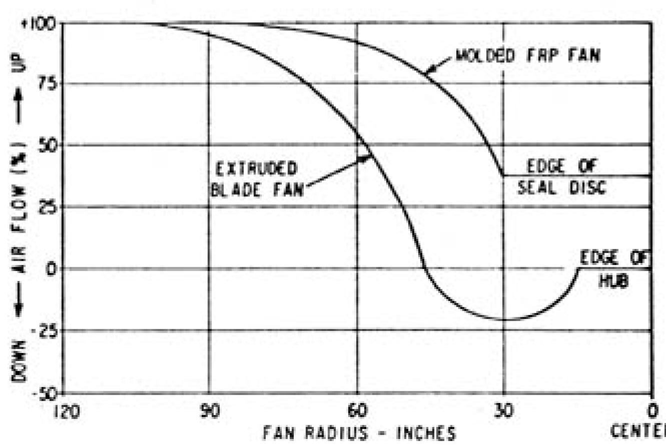

Figure 4 Results of Air Flow Test.

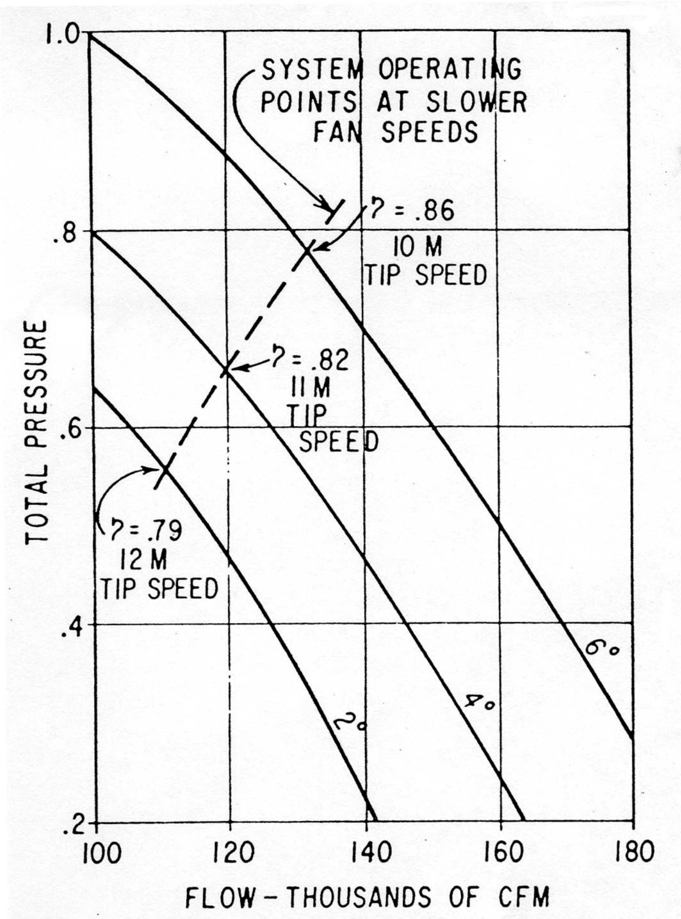

Figure 5 Optimized Fan Blade Efficiency.

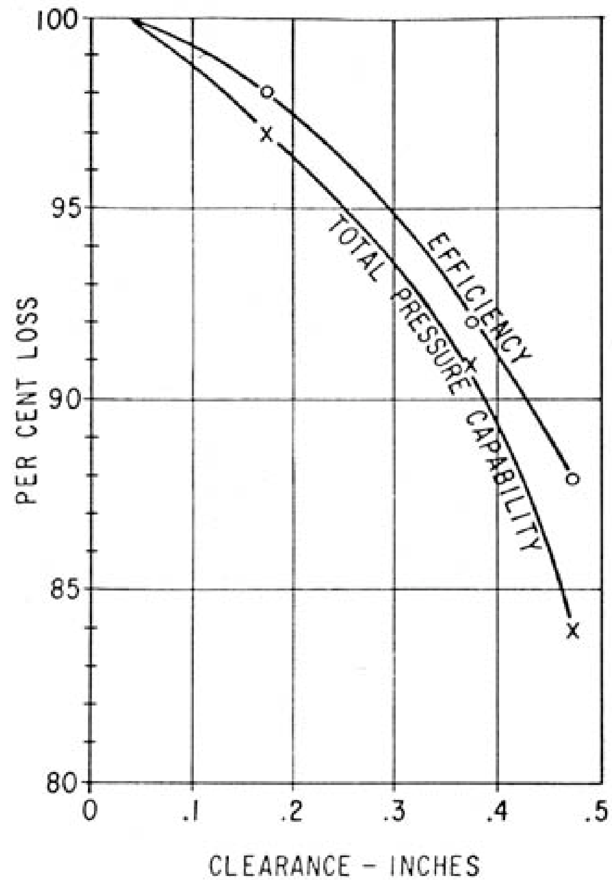

Figure 6 Effect of Tip Clearance on Small Fan.

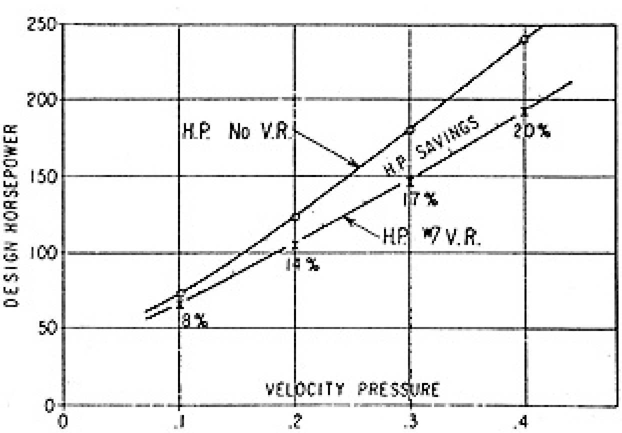

Figure 7 Effect of Velocity Recovery Stack.

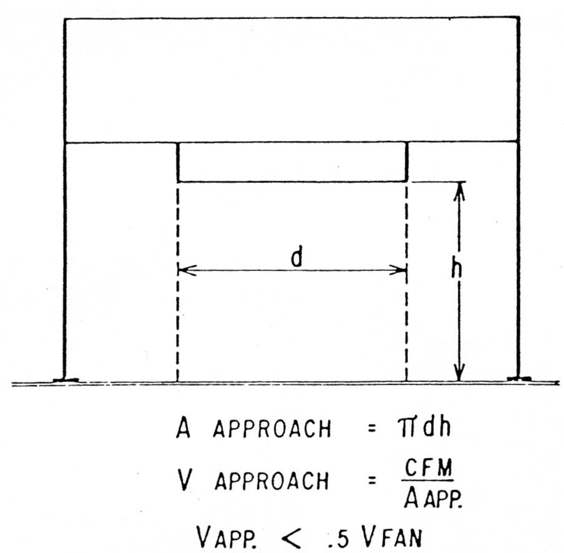

Figure 8 Approach Velocity.

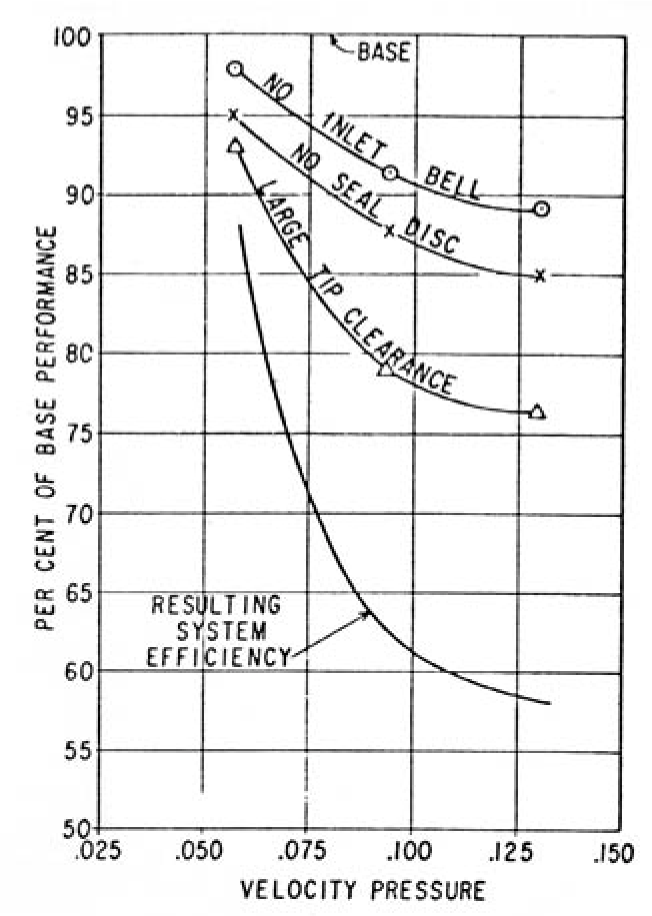

Figure 9 Results of Full Scale Fan Test.