CENTRIFUGAL AND AXIAL FLOW FAN APPLICATIONS AND DESIGN CRITERIA

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ABSTRACT

The application of fan equipment covers a wide range in today's market and requires that attention be given to factors which were not a consideration in the past. The high costs of energy, installation, replacement parts, and loss of production, force the engineer to carry out a more detailed examination of the process and the equipment offered to ensure that optimum efficiency, reliability, maintainability, and quality are designed into the product.

With the use of sophisticated computer analysis for aerodynamic low and stressing, and the availability of a wide range of structural materials, the fan designer is able to produce equipment with higher operating efficiencies and structural reliability suitable for the application.

INTRODUCTION

The movement of gaseous fluid covers a wide range of capacity, pressure, temperature and gas composition, which consequently requires a considerable variety of machine types to cover all applications (Figure 1).

In recent years there has been a rapid increase in the size of utility and process plants forcing the engineer to consider the signi icant amount of energy absorbed and also performance and mechanical reliability of the equipment.

This paper will consider the application of axial low and centrifugal fans for the utility and industrial fields and discuss the considerations taken into account in selecting and designing the fan for that particular service.

AERODYNAMIC DESIGN

Considerable literature exists on the theory of fan design which, when applied, can result in a reasonable degree of success in achieving the required pressure/volume performance. The use of the computer makes it possible to quickly examine a large number of designs for the requirement and produce an optimized design in accordance with the known fan theory. This is particularly true of axial flow fans. The centrifugal fan, however, is more difficult, particularly when high efficiency is required. On many occasions, lengthy trial and error methods are involved before an optimized design is obtained.

The performance is established on a test bed utilizing a model, which generally has an impeller between 36 to 40 inches in diameter. Testing is conducted in accordance with recognized procedures which ensures that all fans, irrespective of design or manufacturer, can be compared on a common basis. The code applicable in North America is Air Motion and Control Association Bulletin 210. The fan model is tested with all accessory equipment which would affect the aerodynamic per ormance, but no attempt is made to simulate duct configurations likely to occur on the actual plant. From this data the performance of geometrically similar fans can be predicted in accordance with the "Fan Laws." This applies to both axial and centrifugal designs.

In the axial low impeller the fluid enters the impeller in an axial direction and leaves in an axial direction (Figure 2). The blades generally have an airfoil pro ile with a heavier cross section at the root than at the tip. Constant thickness plate bladed designs are also available. Blades are attached to a cylindrical or spherically shaped hub.

In the centrifugal fan the fluid enters the impeller in an axial direction and discharges in a radial direction (Figure 3). With this type of impeller there are four basic blade shapes: (a) radial, (b) forwardly curved, (c) forwardly curved radial tipped, and (d) backwardly inclined. The backwardly inclined type can be further subdivided into three distinct shapes; viz, airfoil, plate and curved plate (Figure 4).

Typical characteristic curves for the axial flow and four basic centrifugal blade types are shown on Figures 5-9. The typical axial low fan pressure/volume characteristic has a considerably lower pressure to the left of peak. Operating the fan in this area is not recommended since the fan will be in surge. Severe vibrations will be evident with the possibility of blade failure occurring should this condition be allowed for an extended period.

It will be noted that all the basic centrifugal fan blade types, with the exception of the backwardly inclined machines, have continuously rising power characteristics. The backwardly inclined fan has a self-limiting power curve which can be advantageous when sizing the driver.

The backwardly inclined bladed centrifugal fan was first developed in the late 1940's, providing a significant increase in

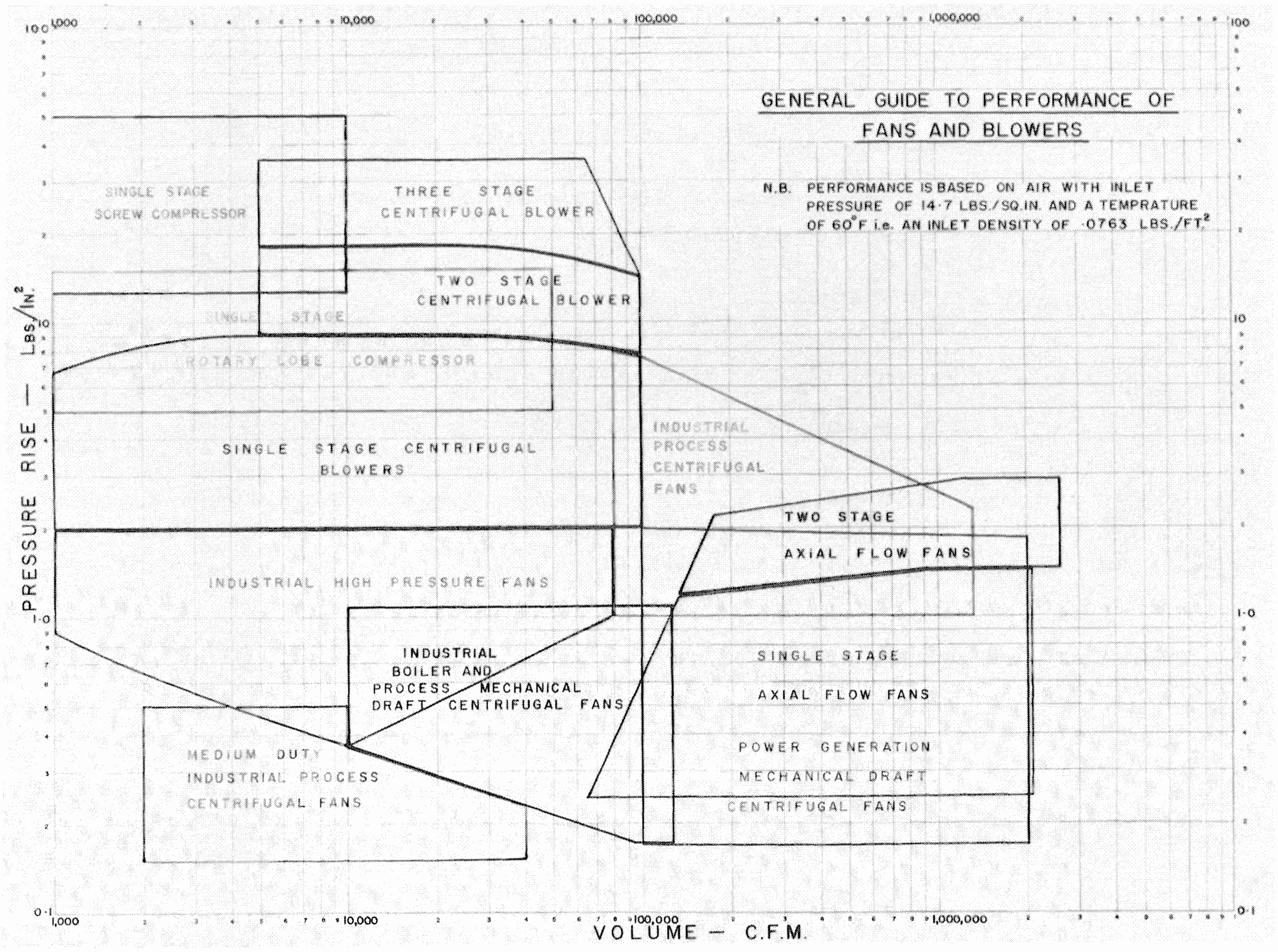

Figure 1. General Guide to Performance of Fans and Blowers.

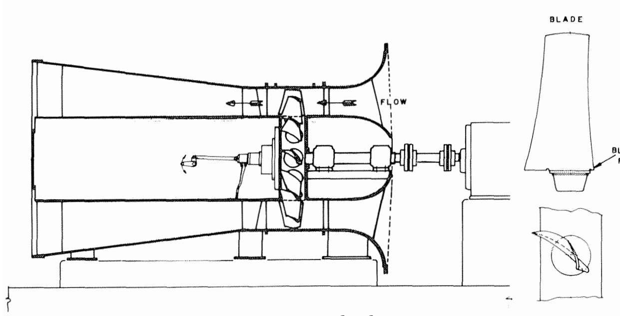

Figure 2. Axial Flow Fan.

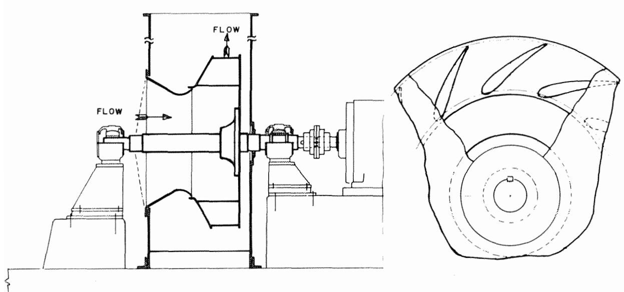

Figure 3. Centrifugal Fan.

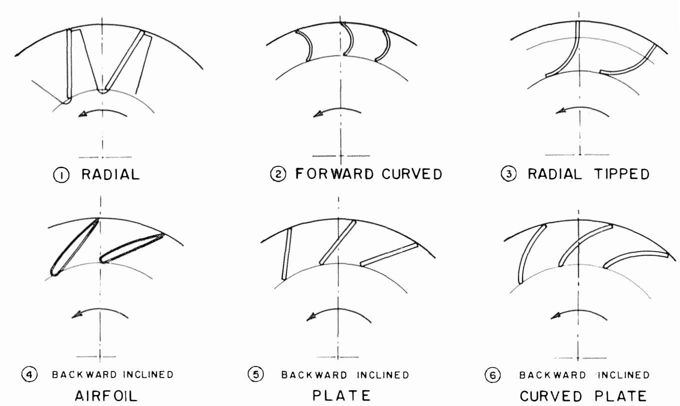

Figure 4. Blade Types.

efficiency over other blade shapes available. It was eventually established that comparable performance could be obtained with the backward curved plate blades. It was at that time assumed that the increase in efficiency with the airfoil fan was due to a reduction in friction through the blade passages; however, by far the greatest contribution to this increase is in fact due to the improved configuration at the inlet to the impeller.

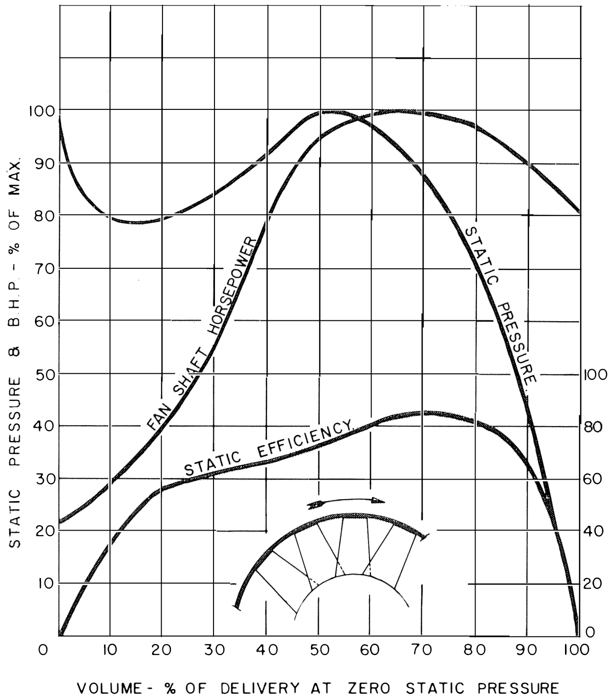

Figure 5. Characteristic Curve of Axial Flow Fan Fixed Blade Pitch.

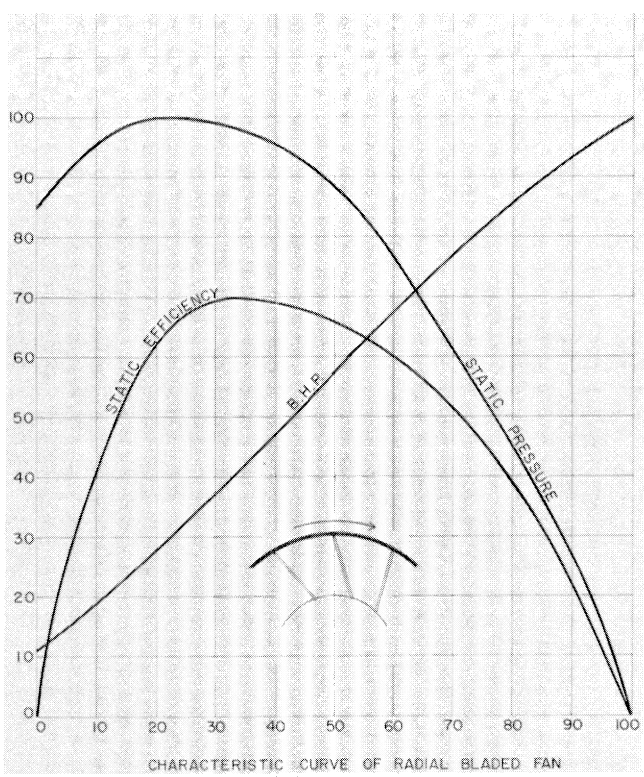

Figure 6. Characteristic Curve of Radial Bladed Fan.

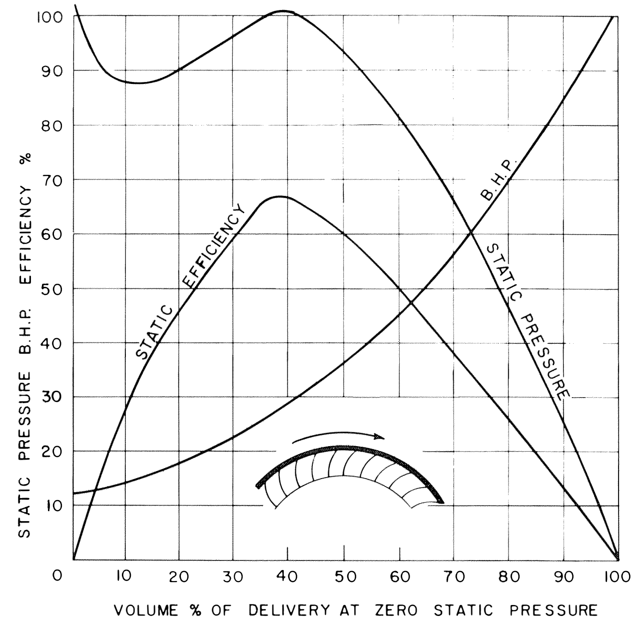

Figure 7. Characteristic Curve of Forwardly Curved Bladed Fan.

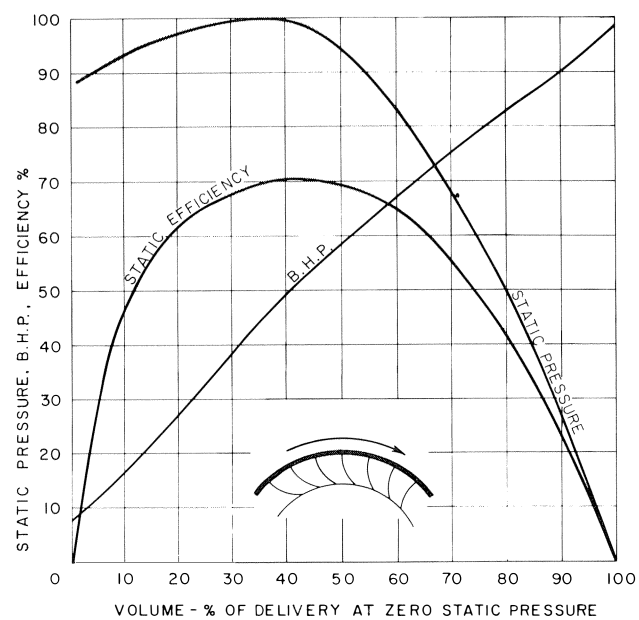

Figure 8. Characteristic Curve of Forwardly Curved Radial Tipped Bladed Fan.

Total efficiencies of 88 to 90 percent are not uncommon and careful control at the impeller inlet is essential in achieving this. Figure 10 illustrates the pressure/volume and efficiency of a range of backward bladed fan design each having the same diameter and operating at the same speed. The efficiency of fan "A," which has a low volume, high pressure characteristic, is considerably lower than the other designs illustrated due to the

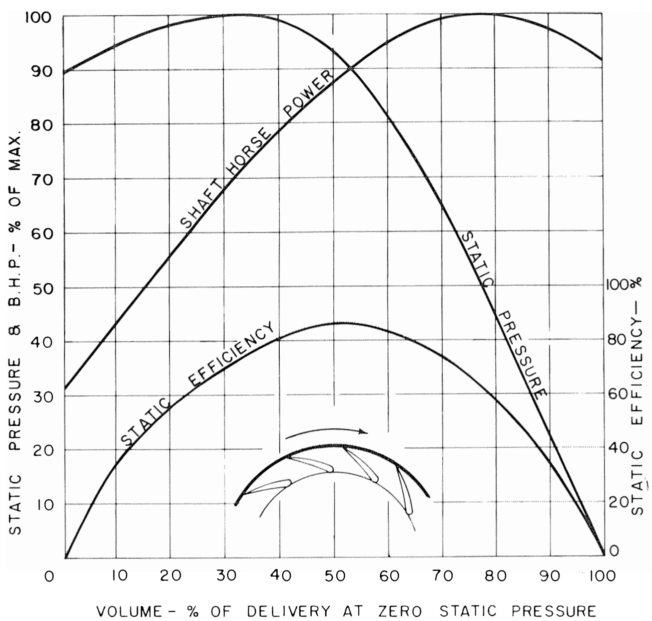

Figure 9. Characteristic Curve of Backwardly Inclined Airfoil Bladed Fan.

proportionately high friction loss between the impeller and the casing sideplate and to some extent in the blade passages. It will be noted, however, that fans "C" and "D," which have curved plate blades, have comparable efficiencies to airfoil designs "E" and "F." It is worthy of note, that replacing the airfoil bladed with the curve plate type on fans "E" and "F" produces almost identical performance, but this results in a considerable reduction in the structural strength of the blade.

The axial flow fan also achieves high efficiency at an optimum blade setting and is only slightly lower than that obtained with the backwardly inclined airfoil centrifugal fan. For all other centrifugal fan blade types, peak efficiencies fall generally in the range of 70 to 80 percent.

CONTROL OF FAN OUTPUT

There are numerous methods of controlling fan output. On centrifugal fans the most commonly used are the preswirl inlet dampers located at the inlet box terminal flange and the variable inlet vane located in the inlet cone. The fan is generally operated at a constant speed. Both of these devices alter the direction of the flow into the impeller. The degree of vane opening, produces a steepening in the pressure volume characteristic, together with a reduction in the work done in the impeller (Figure 11). The total effect is to alter the fan characteristic to more closely follow the system requirement. The backwardly inclined bladed fan illustrated reacts very favour-

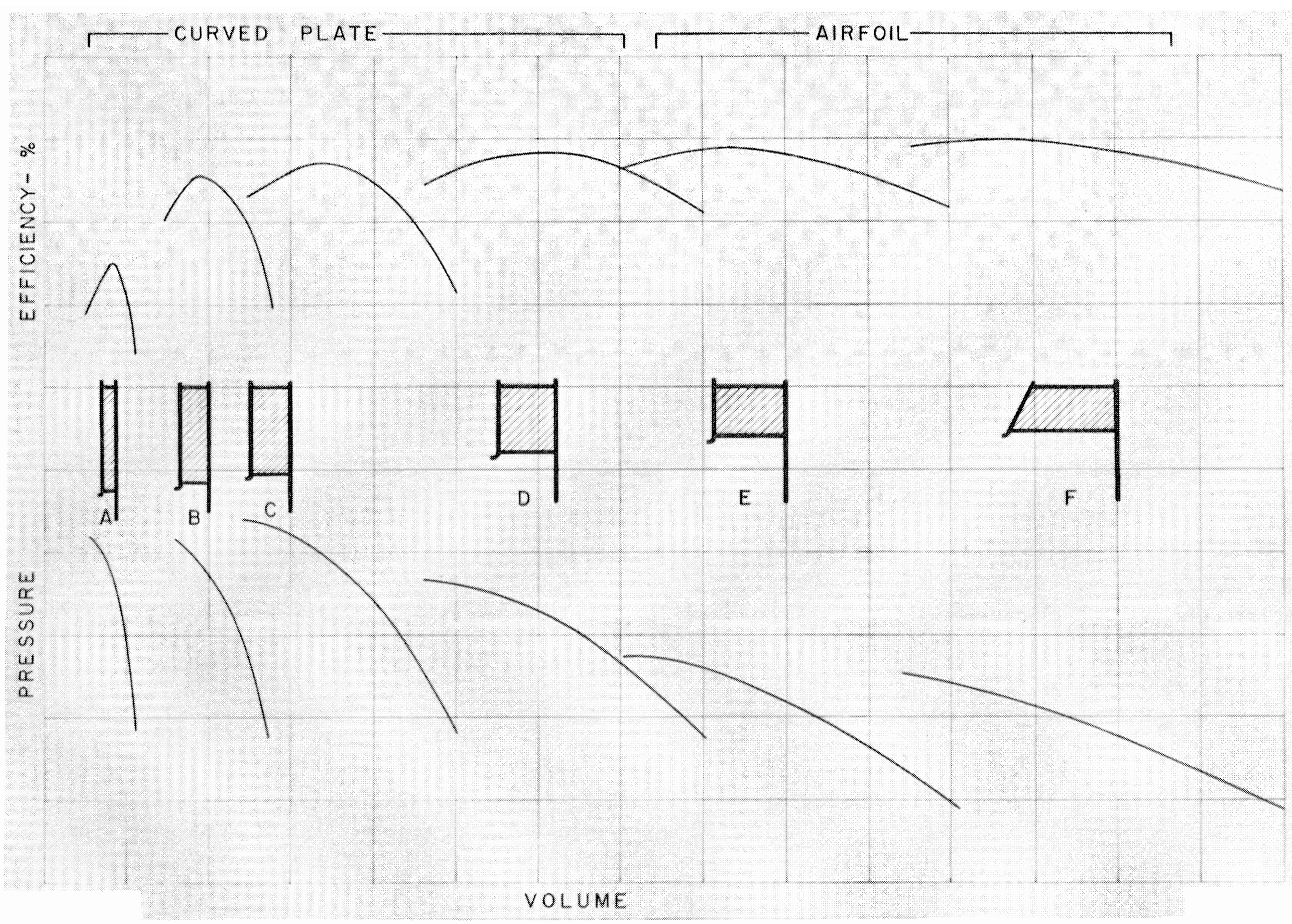

Figure 10. Family of Backwardly Inclined Bladed Fans.

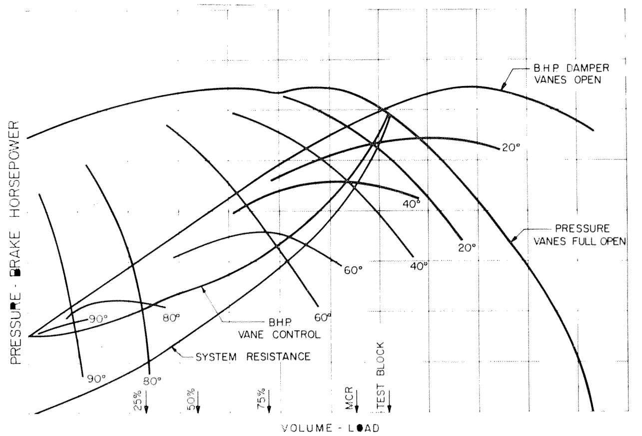

Figure 11. Fan Performance. Constant Speed Control/Inlet Damper · Variable Inlet Vane.

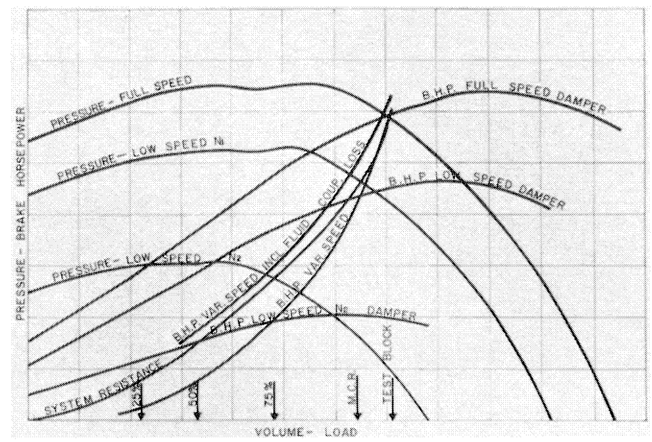

Figure 13. Fan Performance, Variable Speed Motor, Variable Speed Fluid Drive.

ably to these forms of control. The variable inlet vane, due to its location relative to the impeller, is more efficient than the inlet damper and consequently will absorb less power for a given operation condition.

A variation of the foregoing is to introduce a two- or multi-speed drive which allows the fan to operate at peak efficiency at high and low speeds and has the effect of improving power consumption at lower loads (Figure 12). This can complicate the control system for switching speeds and generally results in a more costly prime mover.

Variable speed control of the fan is most efficient over a wide range of loads since the fan will always operate at the most efficient point in the characteristic (Figure 13). However, speed variation is normally achieved with a constant speed drive through some form of slip coupling which has an inherent slip loss and must be considered in the overall power absorbed. To achieve the maximum benefit over the full load range, a direct connected variable speed prime mover is required. This, however, tends to have a high capital cost.

The axial flow fan responds to all of the methods of control previously mentioned for the centrifugal fan but in addition is designed so that the blade pitch angle can be adjusted during operation to follow the load requirement of the system exactly when operating at constant speed (Figure 14). This design also has the advantage of high efficiency over a wide range of control resulting in an absorbed power approaching that of the pure variable speed machine.

Considering a single process with a "square law" system resistance, Figure 15 illustrates the comparison between the control methods described. As previously noted, the most efficient method of control is that of direct connected variable speed drive, closely followed by the variable pitch axial flow fan.

SELECTION FOR THE APPLICATION

For a particular performance requirement, numerous fan selections are possible covering the various types of impeller previously described. Dependent upon other factors, all of these selections may be suitable but most probably one or two at the most will satisfy the application. To arrive at this optimum selection, good communication must exist between the consulting engineer and the fan manufacturer. The latter has

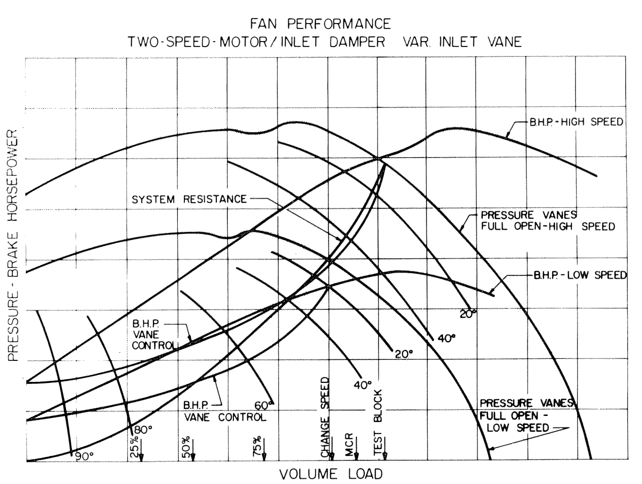

Figure 12. Fan Performance, Two Speed Motor/Inlet Damper · Variable Inlet Vane.

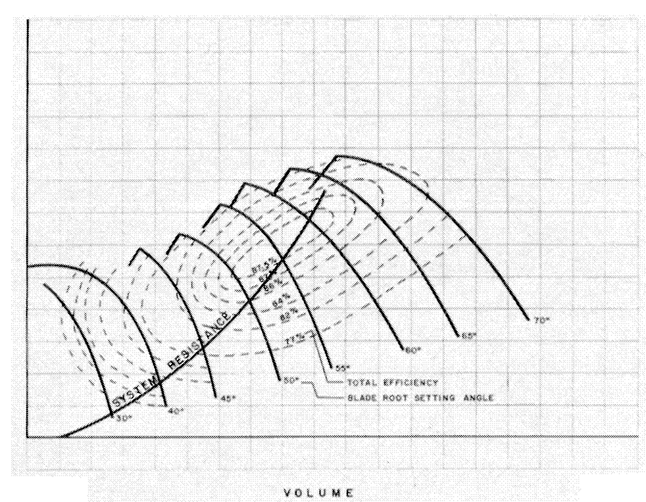

Figure 14. Characteristic Curve of Variable Pitch Axial Flow Fan.

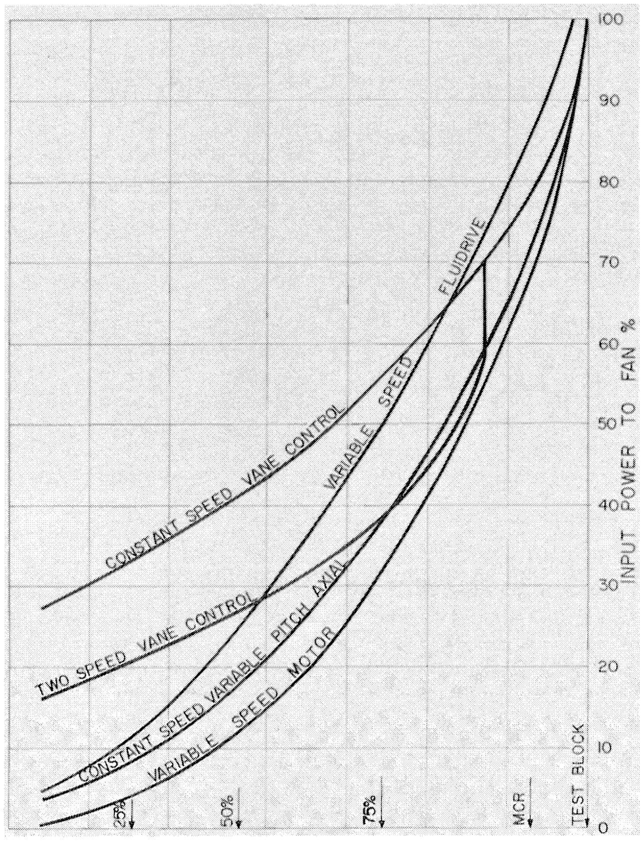

Figure 15. Comparison of Fan Input Power for Various Methods of Capacity Regulation.

the expertise in the moving of gas but may not be familiar with the process, operational modes or production cycles involved in the plant. One of the main considerations affecting the selection of the fan and in many instances dictating the type of impeller and blading is the gas composition. The analysis will determine whether corrosion is an important factor requiring exotic alloy steels, aluminum alloys, titanium or one of the many commercial bonded coatings. Equally important is whether the gas contains particulate matter. The chemical analysis, micron sizing, abrasive and adhesive qualities, and loading presented to the fan could prove extremely critical from a maintainability and structural integrity view point. For example, experience has shown that the high efficiency backwardly inclined airfoil centrifugal fan will operate satisfactorily on induced draft service on a coal fired utility boiler when preceded by an electro static precipitator having a collecting efficiency in the 95 to 98 percent range. Axial flow fans will also operate satisfactorily in this type of environment. However, it would be disastrous to apply these types of impeller to an iron ore pellet machine handling gases containing highly abrasive ore particles entrained from the pellet bed. Velocity of the dust particle through the fan impeller and casing has a considerable bearing on the wear rate experienced. Laboratory work has been carried out, and is still continuing, to determine the abrasive qualities of numerous dusts. Tests have been conducted with controlled weight of dust which has been graded. In an erosion test rig the dust is impinged on a carbon steel test piece at various velocities and angles of impingement. The weight of metal removed is determined and a velocity index established. This provides a yardstick for erosion which is proportional to velocity to the power of the velocity index. Comparison can then be made with other dusts as shown on Figure 16.

The resistance of materials to erosion can also be evaluated in this test rig. In this case the dust sample remains constant while various materials are examined. Figure 17 illustrates the relationship between the air velocity conveying the dust and the amount of material removed from the test specimen. It will be noted that as the velocity increases so does the rate of erosion and also the harder the material the lower the rate of erosion. In practice, the variables are so great that this data is at present only used for guidance. For the highly erosive application several possibilities exist in the selection of the proper fan type, provided suitable wear protection is applied to the impeller and the casing, thus the high efficiency fan need not be eliminated as a possible choice.

Some dusts have adhesive rather than abrasive qualities as in the case of ash produced in a Black Liquor Recovery Boiler in the pulp and paper industry. High efficiency airfoil fans have been in operation in this service for almost twenty years. Dust buildup does occur on the impeller but this is effectively removed with regular steam or air soot-blowing through fixed multi-nozzle blowers while the fan is in operation. This cleaning controls imbalance in the impeller and avoids the possibility of damage to the shaft and bearings.

Generally specifications are written covering the design operating conditions for the plant but all too often transient or partial load conditions, which in many instances are more arduous than design, are ignored. This could prove particularly critical where temperature excursions exceed design, consequently reducing the margin of safety between the operating stress level and the ultimate strength of the material. In some instances this has resulted in catastrophic failure. A discussion of impeller stressing with follow later in this article.

The final considerations in determining the fan selection are load factor at which the plant will operate and the evaluated cost of energy related to this load factor. With this information it may prove that the installation of a more costly fan is offset by the lower operating cost over the life of the unit. It is therefore important to establish the priority of the various aspects which

EROSION TESTS ON DUSTS

WEAR $\propto$ (VELOCITY)^{N}

	DUST TYPE	VELOCITY INDEX (N)	COMPARATIVE WEAR
1.	P.F. FLY-ASH.	2.40	1.000
2.	P.F. FLY-ASH.	2.52	1.782
3.	P.F. FLY-ASH.	2.92	0.819
4.	P.F. FLY-ASH.	3.00	0.738
5.	STOKER FLY-ASH.	2.31	2.546
6.	STOKER FLY-ASH.	2.77	0.794
7.	SINTER DUST	2.38	1.109
8.	SINTER DUST	3.04	0.875
9.	ALUM. OXIDE	2.18	2.611
10.	SILICON CARBIDE	2.23	2.529
11.	CHROME ORE	2.58	1.621

Figure 16. Erosion Tests on Dusts.

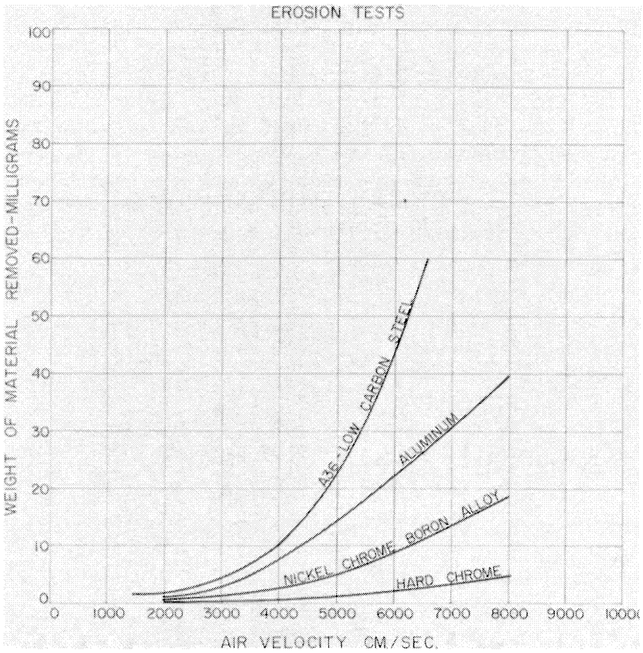

Figure 17. Erosion Tests.

will influence the selection of the fan for the particular application.

IMPELLER STRESSING

An important area in fan development has been the understanding of the stress patterns occurring in centrifugal fan impellers. Prior to 1950, stresses were calculated in a simplified manner. The impeller was considered as independent components comprising blading and rotating discs. The disc stresses were calculated using classical turbine theory with the blade loading being applied as a rim load. The blades were considered as simple beams in bending. Since the technique was rather elementary, large margins of safety were applied. Materials used in construction generally had a considerable margin between the yield strength and ultimate strength which ensured an additional safety factor against possible failure.

With the increasing size of processing plants, the demand for larger diameter and higher speed fan equipment became necessary. This required a more detailed knowledge of the operating stress patterns. Investigations using strain gauge techniques on the rotating impellers proved rather interesting. Previous calculation methods assumed that the maximum stresses occurred at the inlet diameter of the shroud, and at the bore of the backplate or centerplate. It was also considered that the highest blade stress was near the center of the blade span. Strain gauge investigations showed that, on fans with comparatively wide blades, the bending stresses in the blades transmitted twisting movements on the shrouds and the backplate or centerplate as the area of attachment. These stress levels proved to be almost as high as the theoretically calculated stresses. On these wide bladed designs the shroud rigidity tended to make the blades closer to a fixed end beam rather than a simply supported one. High stresses in the area of the attachment could therefore be expected.

More recently, examination of impellers has been carried out using finite element computer analysis. Many proprietary computer programs are available, such as NASTRAN, ANSYS and SUPERB, which can adequately handle this type of analysis. It is only necessary to model a segment of the impeller due to the design symmetry as shown in Figure 18. Plate type elements are normally used in the analysis; however, solid elements and a combination of both have been utilized. The output from the analysis provides extremely detailed information on the stress patterns (Figure 19) and also the deformation of the impeller. It would not be practical to strain gauge an impeller to such a degree to obtain an equivalent amount of data; however, a good correlation has been established between both types of approach. Finite element methods can therefore be used with confidence to establish stress levels in satisfactory operation.

This is of particular importance since such equipment would require construction to be of higher strength materials which generally have a small margin between the yield and ultimate strengths. Accurate prediction of the impeller stresses can, therefore, ensure that an adequate and realistic factor of safety be employed. The designer is aware of the location of the maximum stress and can ensure that the structural strength is provided where necessary. With the introduction of high strength materials, a more rigid control is required on the material itself and on the fabricating processes employed. The material soundness may be checked by ultrasonic examination. Welded joints must also be of high quality. Care must be taken in the conditioning of the weld electrode in controlling preheat, post heat, and welding technique. The welding is subject to non-destructive examination such as magnetic particle ultrasonic and/or X-Ray to ensure soundness.

ROTOR CRITICAL SPEED

Calculation of the first critical speed of the rotating element is carried out generally considering rigid supports (Figure 20). To avoid possible operational problems when the unit is installed, it is recommended that the first critical speed be a

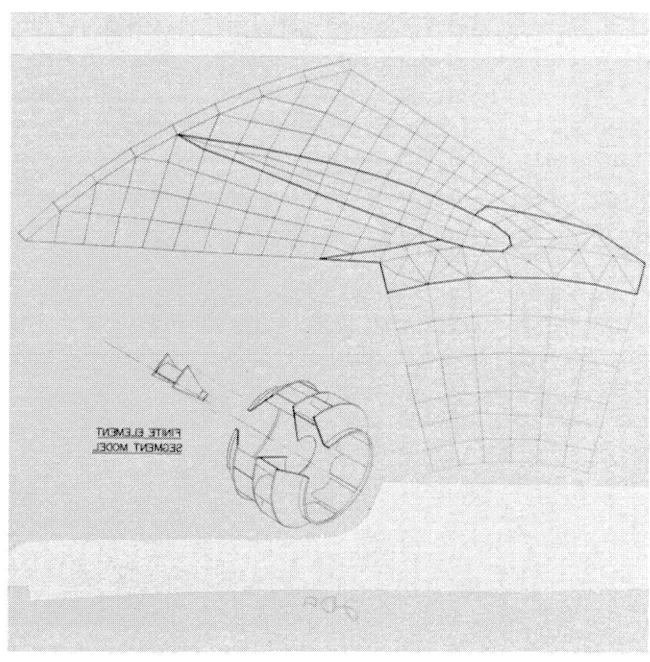

Figure 18. Impeller Segment and Geometry of Center Plate Showing Element Boundary Lines.

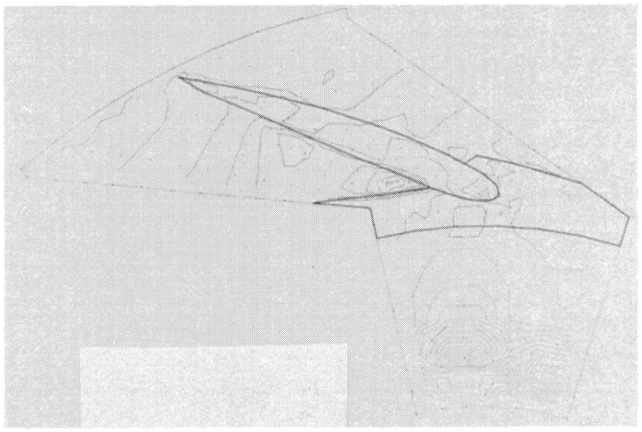

Figure 19. Major Principal Stress Contours on Inside Face of Center Plate.

minimum of twenth-five to thirty percent above the maximum operating speed. Where severe operating conditions exist, such as heavy erosion or build-up, it is suggested that the first critical speed be at least thirty-five percent above operating speed.

The fan manufacturer can also calculate the natural frequency of the combined spring mass system of the rotor, bearing oil, film bearing housing and bearing pedestals but excluding the foundation stiffness. Care should be taken in designing the foundation, since any flexibility will result in lowering the installed resonant frequency of the complete system with consequent operational problems and impaired reliability. The fan manufacturer does not usually have design responsibility for the foundation.

FIELD INSTALLATI●N

Considerable effort has gone into producing a fan which is structurally sound and will handle the requirements for the process. However, by incorrectly installing the equipment, operational problems and performance deficiencies will occur. The fan manufacturer does not generally install the fan in the field but does provide detailed instructions on the installation procedures. All too often the fan will be subject to mechanical vibration problems during comissioning due to misalignment with the driver, inlet cone rubbing and shaft seal rubbing. Performance deficiencies can also occur due to incorrect cone penetration into the impeller and incorrect cone clearances (Figure 22).

Care must, therefore, be taken in following precisely the fan manufacturer's installation specifications to ensure that the predicted efficiency can be obtained and the fan will operate reliably.

FAN S●UND

Considerable interest is being displayed today in reducing environmental pollution, not the least is noise. The fan manufacturer, along with other equipment suppliers, is giving attention to more accurately measuring and predicting the sound generated by existing and new designs.

The fan supplier can reasonably predict the sound emitted from his equipment. This background is obtained from laboratory testing and also conducting sound surveys in the field. However, during the tendering process, he is often asked to make guarantees without having the benefit of knowing where his equipment will be located or whether other equipment will be operating in the area. As this noise factor becomes more critical, the total sound picture in the plant must be the responsibility of the acoustical engineer.

C●NCLUSI●NS

The availability of sophisticated computer programs and a wide range of high strength structural materials enables the fan manufacturer to produce higher efficiency and structurally reliable equipment to meet today the increasing plant size and capacity.

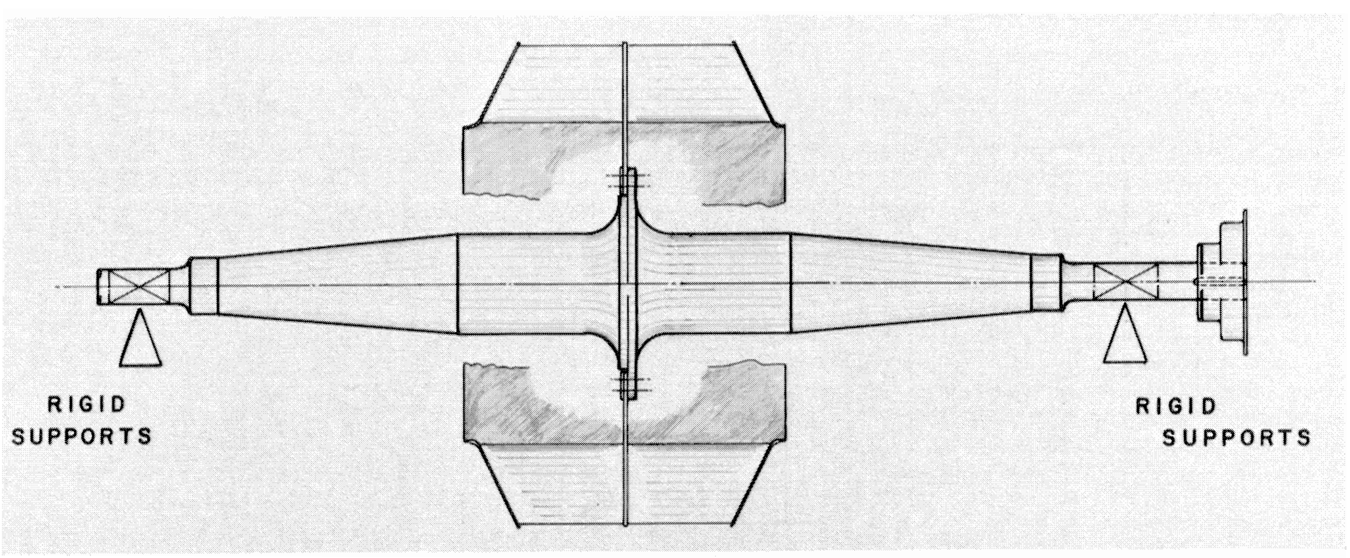

Figure 20. First Critical Speed of Rotating Element.

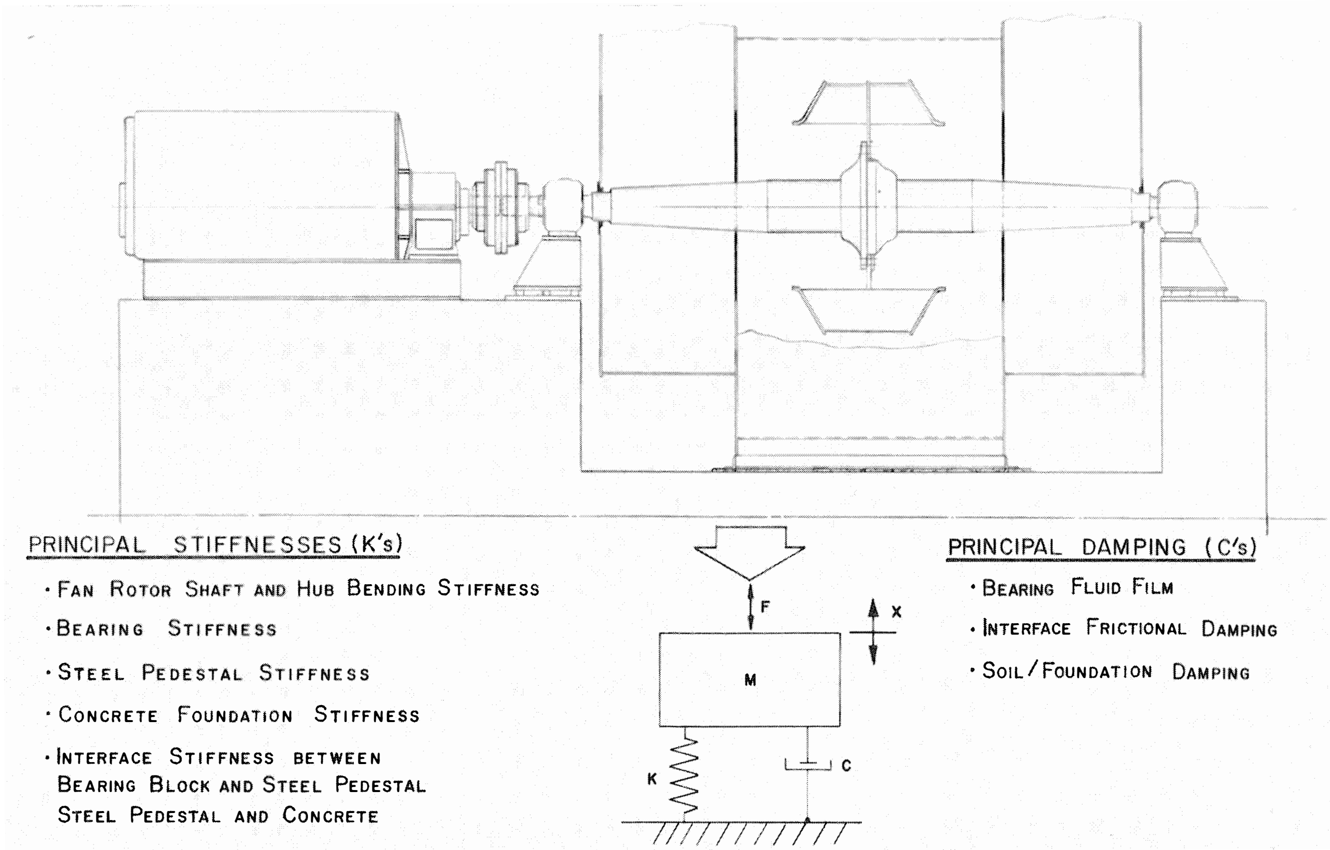

Figure 21. Natural Frequency of Combination Fan Foundation System.

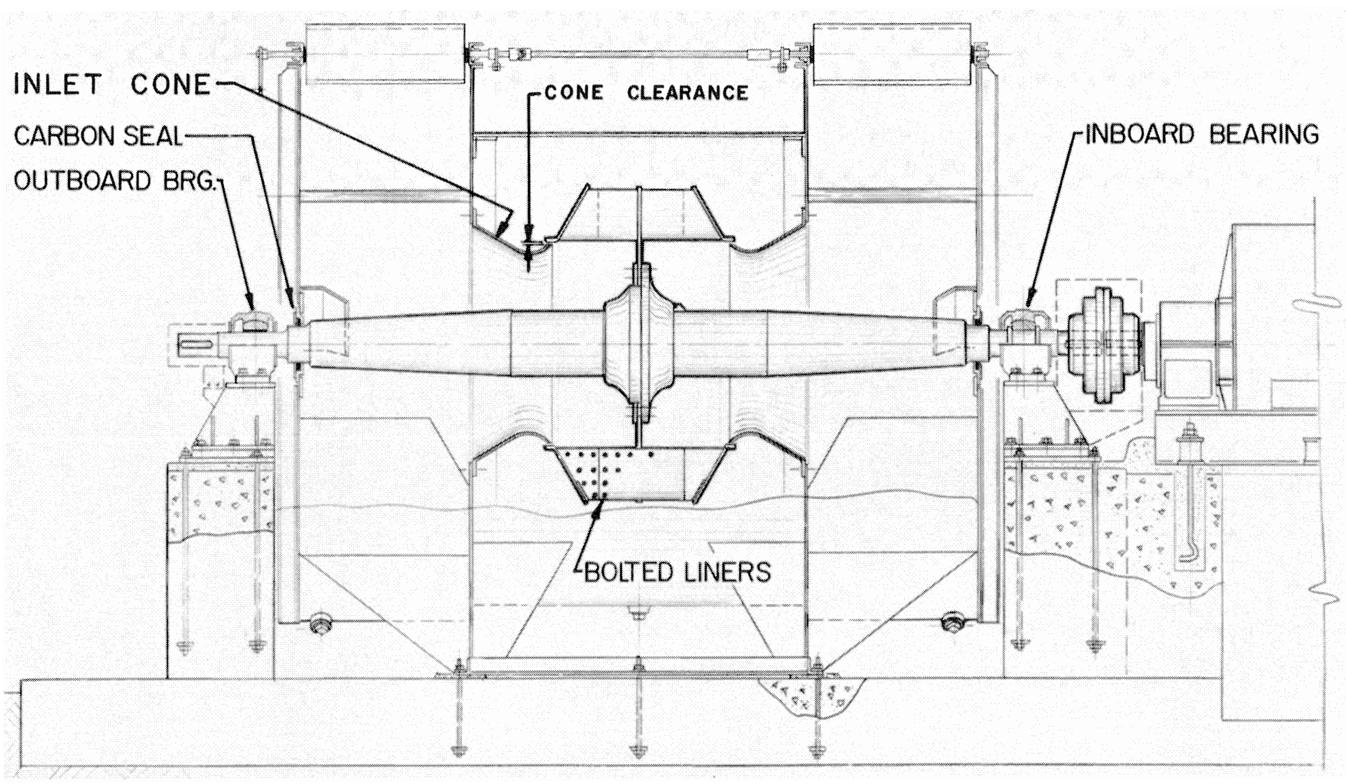

Figure 22. Cross Section of Exhaust Fan.