SELECTION OF FANS FOR THERMAL POWER PLANTS

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CERTIFICATE

This is to certify that dissertation entitled "**SELECTION OF FANS FOR THERMAL POWER PLANTS**" being submitted by **Mr. Pankaj Sharma** to the department of Mechanical engineering, Delhi college of Engineering, Delhi for the partial fulfillment of the degree of Master of Engineering in Thermal Engineering, is a record of bonafide work carried out by him under my guidance and supervision.

The contents of thesis have not been submitted to any other university or institute for the award of any degree or diploma.

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ABSTRACT

The present study aims to provide a guideline for the selection of fans for modern large capacity steam generators and to carry out the site testing of a forced draft fan with a view to compare its performance with supplied data. The points regarding the fan selection have been identified and critically analyzed to assess the suitability of the fans. Electricity boards on the thermal power plants are installing different types of fans such as centrifugal, axial impulse and axial reaction with a suitable combination of control mechanism for the same duty. An attempt has been made to clearly identify the selection procedure for these fans and recommend the type of fan along with specific control mechanism for the primary air and the draft system of 210 MW steam generators.

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CHAPTER-1

INTRODUCTION

The present study aims to provide a guideline for the selection of fans suitable for modern large capacity steam generators and to carry out the site testing of a Forced Draft fan with a view to compare its performance with supplied data.

The points regarding the fan selection have been identified and critically analysed to assess the suitability of the fans. Problem areas have been drawn upon. Recommendations have been made for the Induced Draft, Forced Draft and Primary Air fans.

The axial flow fan working as forced draft fan on 210 MW, unit at Badarpur thermal power station has been chosen for site testing. This fan was instrumented for the measurement of velocity and pressure at fan inlet, outlet and diffuser exit. Three hole probe traverses have been carried out from casing to hub for three-blade setting angles and three flow rates at such settings. The results have been presented graphically.

It has been found from the experiment that at higher blade setting the fan characteristics matches well with the designer data.

1.1 FUNCTION OF FANS

Fans are used to supply air or gases at moderate pressure. Different types of fans are used in the industry depending upon the discharge pressure and flow rates and the type of medium. Fans are used in various industries viz. Power, Cement, Fertilizer, petrochemicals and air conditioning, etc. With the growth in the capacity of these plants, airflow requirements have increased. This has necessitated the increase in fan size, their number and speed to meet the varying flow demands.

Fans were considered as simple devices with only the initial cost factor as a prime consideration in their selection. The present day emphasis is on higher efficiencies over wide operating regimes; ease of operation and maintenance, precise flow control and to maintain a clean environment.

The basic purpose of a “fan” is to move a mass of gas or vapour at the desired velocity. For achieving this objective there is a slight increase in the gas pressure across the fan rotor or impeller. However the main aim remains to move air or gas without any appreciable increase in its pressure. The total pressure developed by fans is of order of a few millimeters of water gauge.

1.2 FANS FOR THERMAL POWER STATIONS

1.2.1 BASICS OF FANS

The air we need for combustion in the furnace and the flue gas that we must evacuate would not be possible without using fans. A fan is capable of imparting energy to the air/gas in form of a boost in pressure. We overcome the losses through the system by means of this pressure boost. Fan performance is represented as volume vs. pressure boost. The basic information needed to select a fan is:

- Air or gas flow- Kg/hr
- Density (function of temperature and pressure)
- System resistance (losses)

1.2.2 CLASSIFIACATION OF FANS

In boiler practice we meet the following types of fans

A) Axial fans-These are impulse type or reaction type

B) Centrifugal (radial) fans- These are backward curved or forward curved or straight bladed type

1.2.3 AXIAL FANS

In this type of fan, the movement of air or gas is parallel to its exit of rotation. These fans are better suited to low resistance applications. The axial flow fan uses the screw like action of a multibladed rotating shaft, or propeller, to move air or gas in a straight path.

1.2.4 CENTRIFUGAL FAN

This fan moves gas or air perpendicular to the axis of rotation. This fan is best suited when the air is to be moved in a system where the frictional resistance is relatively high. The blade wheel whirls air centrifugally between each pair of blades and forces it out peripherally at high velocity and high static pressure. Air is sucked in at the eye of the impeller. As the air leaves the revolving blade tips part of its velocity is converted into additional static pressure by a scroll shaped housing. There are three types of blades

a) Backward curved blades

b) Forward curved blades
c) Radial blades

1.2.5 DRAFT SYSTEM

Before a detailed study of industrial fans it is in the fitness of things to understand the various draft systems.

The term draft denotes the difference between the atmospheric pressure and the pressure existing in the furnace. Depending upon the draft system we have

a) Natural Draft
b) Induced Draft
c) Forced Draft and
d) Balanced Draft System

1.2.6 NATURAL DRAFT- In natural draft unit the pressure differentials are obtained by constructing tall chimney so that vacuum is created in the furnace. Due to the pressure difference, air is admitted to the furnace.

1.2.7 INDUCED DRAFT-In this system the air is admitted to the furnace directly from the atmosphere and the flue gases are taken out by means of induced draft fans and the furnace is maintained under vacuum.

1.2.8 FORCED DRAFT-A set of forced draft fans are made use of, for supplying air to the furnace and so the furnace is pressurized. The flue gases are taken out due to the pressure difference between the furnace and the atmosphere.

1.2.9 BALANCE DRAFT-Here a set of Induced and Forced draft fans are utilized in maintaining a vacuum in the furnace. Normally all the power stations utilize this draft system.

1.2.10 INDUCED DRAFT FANS

Induced Draft Fan: The induced draft fans are generally of axial type. Impeller nominal diameter is of the order of 2500mm. The fan consists of following sub assemblies.

a) Suction Chamber
b) Inlet Vane Control
c) Impeller
d) Outlet guide vane assembly

e) Diffuser
f) Bearings
g) Flexible Coupling

1.2.11 PRIMARY AIR FAN

Primary air fan is flange-mounted design, single suction, and backward curved bladed radial fan operating on the principle of energy transformation due to centrifugal forces. Some amount of the velocity energy is converted to pressure energy in the spiral casing. The fan is driven at constant speed and varying the angle of the inlet vane control controls the flow. The special feature of the fan is that it is provided with inlet guide vane control with positive and precise link mechanism. It is robust in construction for higher peripheral speed so as to have unit sizes. Fan can develop high pressures at low and medium volumes and can handle hot air laden with dust particles. The fan consists of the following sub assemblies

Suction Chamber: It is of welded sheet steel construction and split horizontally for the assembly and dismantling. Manholes are provided for inspection of the same.

Inlet Vane Control: It consists of a number of aerofoil fixed to individual shaft, which is connected by means of angular joints to a central ring. A control lever is connected to ring, which can be operated by an electric servo meter.

Rotor: The rotor consists of a shaft and impeller. The impeller is mounted on a shaft with a taper fit and locking nut. The critical speed is maintained well above operating speed.

Housing: The fan rotor is placed in cylindered roller antifriction bearings. An inclined ball bearing absorbs the axial thrust on the impeller side and scrapper rings seal off the bearing casing. For controlling the bearing temperature, thermometers are provided in the bearings.

Flexible Coupling: Coupling is of flexible pin type with rubber bush inserts.

Shaft Seal: It is a two part-labyrinth seal, which seals off the box section casing at the shaft passage.

1.2.11 IGNITOR AIR FAN

Ignitor fan provides necessary combustion air to all the ignitors. Fan makes the suction from the atmosphere directly and supplies air to the wind boxes of the individual ignitors at a fixed constant uncontrolled rate

at ambient temperature. Fan impeller is directly mounted on the motor shaft and installed in the casing. Casing is so designed that one side panel (cover) can be easily removed off by loosening the fixing bolts. Dampers are provided and flanged at inlet and outlet of fan to control the airflow.

1.2.12 SCANNER AIR FAN

Scanner fans are installed in the boiler for supplying continuously cooling air to the flame scanner provided for flame supervision. Normally one fan remains in service while the other one remains as standby. Scanner air fan is centrifugal type and impeller is directly on the motor shaft with the help of hub. Sets of dampers are mounted at inlet and outlet of the fan.

1.2.13 CONTROL OF FAN OUTPUT

Very few applications permit fan to operate continuously at the same pressure and volume discharge rate. Therefore to meet requirements of the system, some convenient means of controlling fan output becomes necessary. This will also help in avoiding the overload on drive during starting.

Variable Speed Control: Speed variation of fan also varies the quantity and pressure developed in proportion to the load and square of the load. Hence, by selecting speed variation as a control method the fan operating point can be kept always at the optimum point which results in overall higher efficiency of fan.

Inlet Vane Control: In the case of axial fans the variation in output is achieved by changing of angle of inlet with respect to the moving blades.

Damper regulation: In this method a damper is introduced in the circuit and by varying the damper opening the resistance offered by it is altered.

Variable Pitch Type: This is applicable only to variable pitch type axial fan. The blade itself is rotated such that optimum efficiency can be maintained at part loads also.

1.3 FACTORS AFFECTING FAN PERFORMANCE

Given fan size, system resistance and fluid density when speed changes:

a) Capacity varies directly with speed

$$\frac{Q1}{Q2} = \frac{(RPM1)}{(RPM2)}$$

b) Pressure varies the square of speed

$$\frac{P1}{P2} = \frac{(RPM1)^2}{(RPM2)}$$

c) Power varies the third power of speed

$$\frac{HP1}{HP2} = \frac{(RPM1)^3}{(RPM2)}$$

1.3.1 CONSTANT PRESSURE

a) Speed, capacity and horsepower vary inversely with the square root of density or,

$$\frac{RPM1}{RPM2}, \frac{Q1}{Q2}, \frac{HP1}{HP2} = \frac{(d2)^{1/2}}{(d1)}$$

b) Speed capacity and horsepower vary inversely with the square root of absolute temperature.

$$\frac{RPM1}{RPM2}, \frac{Q1}{Q2}, \frac{HP1}{HP2} = \frac{(b2)^{1/2}}{(b1)} = \frac{(T1)}{(T2)}$$

1.3.2 CONSTANT SPEED AND CAPACITY

Horsepower and pressure vary directly with density and barometric pressure and inversely with absolute temperature.

$$\frac{HP1}{HP2}, \frac{P1}{P2} = \frac{d1}{d2} = \frac{b1}{b2} = \frac{T2}{T1}$$

A typical arrangement of these fans along with pressure at various points and pressure drop across various equipment used in

modern thermal power station is indicated in fig.1.1 and 1.2. Table 1 indicates details of various types of fans used in steam generators for 210MW unit at a particular power station.

Table 2 indicates the type of fan along with the regulation mechanism for induced primary air and forced draft fans used for 100/110, 200/210, and 500 MW sets already in operation on Indian scene.

1.4 NECESSITY FOR STUDY ON FANS

The high rates of development of the national economy of India with simultaneous increase in the role played by electric power predetermine a continuous increase in the annual addition of new capacities of the order of 3000 MW or more. The major addition in installed capacity is of coal based thermal power plants of higher size units. Because of the large gap between supply and demand for electricity, it is becoming difficult to keep our units on par with out any consideration to the overall station efficiency and regular unit maintenance requirements. This sort of attitude seriously deteriorates the health of the running units and in fact further reduces the unit availability.

Recent studies carried out on the performance of 200/210 MW units installed in the country show that each unit of this capacity losses about 225-230 equivalent hours in a year due to complete shut down or partial loading necessitated by problem in I. D., F. D. and P. A. fans. If this loss can be eliminated there can be an improvement of 2.6% in the present plant load factor level of 45% for these units. This has aroused considerable interest among power station engineers in the selection of more reliable and efficient fans for use in modern large capacity thermal power stations.

The lesser availability of the plant equipment may be attributed to factors such as quality of equipment, sizing and selection of equipment, operation and maintenance and the quality of coal. Out of the above sizing and selection of equipment is of prime importance and the same has been discussed of I. D., F. D. and P. A. fans.

1.5 PRESENT PROBLEM

Electricity boards on the thermal power plants are installing different types of fans such as centrifugal, axial impulse and axial reaction with a suitable combination of control mechanism for the same duty.

An attempt has been made to clearly identify the selection procedure for these fans and recommend the type of fan along with specific control mechanism for the primary air and the draft system of 210 MW steam generators.

A site test has also been carried out on a variable pitch axial reaction forced draft fan to study and obtain the actual fan characteristics and flow conditions at various stations of the fan airways for different flows.

1.6 OUTLINE

In the thesis, the test conducted enables to know the installed performance of fan on site in conjunction with the system to which it is connected. This test determines under actual operating conditions, the volume flow rate, the power input and the total pressure rise across the fan. The result of such tests may also provide an actual value for flow resistance of the air ways which can be compared with the value specified and will also provide an experimental fan performance point to compare with the fan characteristics determined by tests with standardized airways.

A site test may also form the basis of fan acceptance test, agreed between the manufacturer and the purchaser. More careful and thoughtfully planned flow tests on site on fans also help in accumulating information regarding the blade performance which may act as a check for the design and thereby improvement in it if need be.

CHAPTER 2

LITERATURE SURVEY

2.1 Fan Materials, Mechanics and Noise

The following data are directed specifically at alerting the fan designer to the many interface problem interface problems that are of importance in achieving the best overall design solution. Fortunately, there is little incompatibility between good aerodynamic design and optimum solutions in the associated disciplines.

General background information in these interactive areas is readily available in standard textbooks. However, the special requirements of fans demand that the interface problems and relevant research findings be discussed here.

2.1.1 MATERIALS

Environmental Factors

The environment in which the fan is to operate should be one of the first considerations affecting material and manufacturing method selection. For instance, in a furnace system the material should have a high melting point, low creep, and resistance to oxidation. The construction method should avoid stress concentrations; particularly as metal expansions and contractions due to temperature cycling will increase the tendency toward fatigue failure. The high Ni-Cr alloy steels are favoured for such installations. When cambered plates are used, extra root strength and stiffness can be obtained by inserting a tapered under surface gusset plate to match the concave curvature (Fig.2.1). The ensuing surface irregularities should have an inconsequential influence on the blade aerodynamic properties. Similar type gussets on both outer surfaces of a double-skinned airfoil-shaped blade will result in drag and pressure rise penalties.

It is not possible to design all duct\ fan systems to be free of changing load conditions. In some cases this may involve “shock” air loading of the blades; extra stiffness and strength must be designed into the unit.

However, the most common causes of “shock” loading are gross turbulent inlet conditions and fan stalling. This latter operational condition is often present but can remain unrecognized. Some fan manufacturers who present catalogue data on the complete volume/pressure characteristics without cautionary notes perpetuate the continuing ignorance about partial or complete stall properties. In most

cases sufficiency ruggedness, based on experience, is built into the equipment to ensure its reliability.

Blades and vanes may have to operate in moisture-and/or dust-laden flows and hence are liable to water and dust erosion problems. In brief, the major variables are dust concentration, hardness, size, shape, velocity, and the strike incidence with respect to the target material. The physical properties of the target will determine the subsequent erosion experienced.

Erosion is of either a ductile or brittle variety where each type is associated with a specific incidence range. The wear rate is a maximum at approximately 25° where the ductile mechanism is active. As the incidence approaches 90°, the brittle fracture process progressively assumes dominance.

The wear rate (mg/g of dust) is proportional to velocity raised to a power that varies within the range 2.0 to 2.3; this relationship is independent of the erosion mechanism for nonbrittle target materials. Exponents of 3 have been recorded for brittle erosion on brittle surfaces.

Hard crystalline dusts such as silica, silicon carbide, and alumina are most destructive. Quartz, which is a particular type of silica abundantly present in nature, provides a ready source of erosion. On impact, the sharp corners of freshly broken crystalline dusts have a much greater destructive power than impacting faces or edges. Rounded noncrystalline dust particles tend to peen metal surfaces at the higher speeds.

Aerodynamic drag forces will modify dust trajectories in regions of rapidly changing airflow direction. This is particularly true for dust particles below 10-μ m in size. The actual incidence of these smaller particles relative to the blade nose surface is consequently less than the apparent one. The percentage of particles striking the surface is progressively reduced as dust size and the related incidence are decreased.

Experiments show that wears rate rises rapidly with particle dimension, reaching a plateau for sizes above approximately 50 µ m. the threshold, earlier is the particle size at which this plateau commences.

The more normal definition of threshold is the condition below which erosion is arrested. Attempts have been made to relate this threshold to particle size, velocity, or kinetic energy. However, an important related variable that has not received much experimental

attention is the apparent dust incidence. Experience has shown a substantial threshold improvement with reducing incidence

There are a few reports of wear occurring for particles down to 3-µ m in size. Hence dust size. Hence dust size in a general sense is an inappropriate threshold criterion although some experiments on axial flow compressors have demonstrated zero wear for particle sizes less than 10 µ m.

Extensive testing with dust at the incidence for maximum erosiveness has established kinetic energy as the most universal measure of erosion threshold; the target materials were steel and aluminum.

Hence for common dusts at roughly equal specific weights and size, it is possible to fix an approximate velocity at which erosion ceases, for the maximum wear incidence. For steel and aluminum this velocity has a mean value of 45 m/s when dry silicon dust having medium and maximum particle sixes of 35 and 100 µ m, respectively, is active. Confirmation of this figure was obtained for an aluminum bladed rotor exhausting dusty air from an ore crushing station in an underground mine. The blade leading edge profile was of a type that facilitated a high wear rate.

As foreshadowed earlier, the trajectory and strike rate changes that accompany small apparent dust incidences will lead to increased threshold velocities. Hence in dusty situations the designer should make a special effort to ensure a smooth inlet flow and the avoidance of blade surfaces permitting dust strikes at incidences promoting peak erosiveness.

Restrictions on airfoil camber, especially in the prime wear and blade speed increases of up to 90 m/s for the previously specified dust before erosion becomes a problem; however, the dust must enter the blading annulus in a smooth and controlled manner.

Recent experiences with a large mine fan having a tip speed of 110 m/s show considerable promise. The blade leading edge has been carried forward by 3% to a very small nose radius that unites the upper and lower surface extensions. These flat extensions meet the original airfoil contours along tangential planes; the centre line of the wedge shaped extension must approximate the direction of the oncoming stream relative to the blade.

Hence erosion is restricted to the thin extreme leading edge, being mainly of the brittle type. The active dust quantities are consequently a very small percentage of the total throughput; this promised longer blade life. But since a ‘shock free’ local incidence is provided, drag will remain low ensuring high efficiency. When this local incidence condition is departed from, increased wear on one surface will sharpen the leading edge and operational efficiency will fall. In other indicates the importance of individual design attention to each blade element shape in erosive airstreams.

The lower blade surface in the vicinity of the trailing edge constitutes an erosion area of secondary importance. As the blade camber increases, the dust to surface incidence and the erosion rate both become greater.

Model tests to obtain quantitative material loss data on actual turbo-machinery are reported the former being concerned with the severe problem faced by the operators of induced draft, axial flow fans in power station boiler installations.

The two most common blading materials for large fans are steel and aluminium. The weight loss per nit dust mass is approximately identical for both metals, which leaves steel with an advantage of nearly 3 in respect to volume loss. Annealing, tempering, and age hardening of these metals has no substantial influence on the wear rate. This feature is in contradistinction to the abrasion process in which wear is directly proportional to surface hardness. (Abrasion is defined as the rubbing action of an abrasive material over a surface, under applied pressure.)

Metal carbide surface coatings reduce the erosive wear rate, but the thinness of the flame-applied layers greatly restricts their practical value. The layer eventually erodes away, but in the process undesirable surface roughness is left; the latter affects aerodynamic performance

High alloy white cast irons possessing densely dispersed metal carbides in a tough matrix material represent a possible solution for leading edge protection under extreme conditions. The high cost of such inserts may, however, restrict their use. On the other hand, the development in Japan of a coating process, designated SIGMA, is apparently showing considerable promise

Composite non-metallic blading materials have relatively poor erosion resistance. When elevated temperature and oil mist are nonexistent, resilient materials such as special rubbers or polyurethane's can be used as coatings to resist erosion. The integrity of the underlying blade material can be maintained by regular coating replacements.

Many exhaust fans expel a corrosive and moisture laden air stream. Provided erosion is not a complementary problem, suitable epoxy paints are available. However metal surfaces capable of withstanding erosion, but not corrosion, may have to be covered with a material resistant to both these mechanisms. Resilient materials have been the most successful in this regard. However, because fan performance is sometimes severely compromised by these surface coverings, aerodynamic allowance should be made in the design or fan selection exercises for this event.

2.1.2 Material Selection

In addition to the foregoing considerations, the weight, strength and cost of the material have to be taken into account. For any particular application, the blade attachment details will have some bearing on the choice of material. For instance, with a small, fixed pitch rotor the cheapest solution consists of a cast or moulded integral boss/blading assembly. Steel or aluminum is used where strength is required. Moulded foam plastics provide an adequate and cheap unit for low powered units in protected environments. However, large fans necessitate the blades being cast or fabricated as separate items. Fixed pitch blades of the latter variety are often welded to the boss and then stress relieved. Cast or moulded blades are normally clamped into the boss, or supported in a trunnion-bearing set-up for variable pitch operation.

Large benefits accrue from blade weight reductions. A lighter hub follows, making for dead weight reductions on the drive shaft and lesser vibration when ort of balance forces develops. In addition, the "flywheel" effects on starting and stopping problems are reduced. The blades must, however, retain sufficient strength and rigidity.

Wind tunnel fans have been constructed with blades possessing a fiberglass skin and a stiffness adding foam plastic filling. Starting torque and braking force are both substantially reduced; increased response to wind speed controls is an additional benefit.

Because of its weight advantage over cast or forged steel, aluminum is a common blade and hub material. Although the weight loss due to erosion is comparable to that of steel, the volume loss is, of course, greater. The corrosion and erosion resistant properties of cast aluminum alloys with high silicon content (e.g., U.S. Federal Specification QQ-A-601E, A356 with T6 heat treatment), in instances could lead to its selection as the best overall solution.

Contrary to popular belief, the normal varieties of stainless steel are subject to corrosion in certain acidic, dust laden air conditions. Blades possessing skins of this material quickly develop a series of pinhole flaws, which by admitting moisture into the interior provide a potential vibration situation. Special corrosion resistant alloy steels are costly.

Blades with a spar, ribs, and sheet metal skins are favoured for large, variable pitch fans. Weight is kept to a minimum and the ribs assist in maintaining blade profile. The leading edge can be of solid construction, forming a secondary spar.

Air tightness is essential in avoiding water ingression, with attendant corrosion and balance problems.

Laminated wood construction, of an integral hub and blade type, is now seldom utilized. However, separate laminated or moulded wooden blades are in use, having the advantage of high profile shape accuracy. The surface is frequently protected against moisture by a plastic sheath covering.

The recent advent of production of carbon and boron fibres in commercial quantities is expected to encourage a greater use of fibre reinforced moulded blades. Their cost is currently the main deterrent.

Blade life and replacement costs are two important factors when dealing with fans in adverse environments; these should be considered when selecting the blade material and construction.

2.1.3 Other Design Considerations

The design of the boss, drive shaft, and other mechanical features will follow normal procedures, but allowance should be made for significant unsteady aerodynamic and out of balance forces; the latter may be present during the fan life and are often due to uneven erosive wear, a broken blade, or mud build-up. The duct wall surrounding the blade tips must be stout enough to resist fracture in the case of blade failure, thus containing the potentially dangerous debris.

2.1.4 NOISE

In achieving a minimum fan noise fan noise level, it is important to design, install, and operate the fan in accordance with good aerodynamic principles. For instance, the fan unit of plates 2 and 3 at the design blade pitch angle and one diameter removed from the fan

casing, registered an overall sound pressure level of 75 dB with a peak of 71 dB in the 150 to 300-octave band. The tip speed was 94m/s and the power 675 kW. Electric motor hum constituted part of this emitted noise. At a distance of 100 m, the high frequencies had been dissipated and the low frequency components were barely discernible in the quiet surrounding environment. There was a similar measure of noise control in the fan unit of plates 4 and 5; it was possible to hold conversations at normal speech levels in the immediate vicinity of the installation, which would have been impossible with the units they replaced.

The preceding illustrations are just two of a number of “quiet” fans designed or specified by the author.

The commonly held belief that fans are inherently noisy, and consequently must be either tolerated or provided with additional noise control equipment, is obviously out of step with the evidence. Excessive noise can usually be interpreted as indicative of some aerodynamic design flaw and should be treated as such; it should not be ignored. A striking example of the consequences that can accompany a failure to put the correct interpretation on a grossly excessive model noise level is available from the Hinkley Point nuclear power station fan and gas ducting failures. Aerodynamic design modifications at the model stage would have resulted in great savings of money and manpower.

In the past the emphasis has been on producing fans of low capital cost for industrial use. The customer has consequently accepted aerodynamic crudity. It is therefore assumed by the latter that the emitted noise is normal; where noise must be controlled, the fitting of attenuators is becoming standard practice. However, the total cost of the installation often exceeds that of a “quiet” fan installation, of good aerodynamic and mechanical design. Attenuator maintenance and additional space requirements must also be considered in the cost comparison.

2.1.5 Design Objectives

By applying the aerodynamic principles with respect to minimizing unsteady aerodynamic forces, the main noise abatement objectives are achieved. The requirements are as follows:

1. Design the upstream duct system to be free of highly disturbed and swirling flows.
2. Provide a smooth flow into the fan annulus by a suitable outer wall “bellmouth” and by fitting an adequate nose fairing or spinner.
3. Design the upstream vane or support struts to possess low drag and to be free of flow separation, ensuring that their thin trailing edges are at least one half vane or strut chord upstream of the rotor leading edge .
4. Ensure that the rotor blades are properly matched, in twist, chord, and camber, to provide efficient and low drag operation at the design condition; also, small tip clearance is most desirable
5. The straightener vanes or streamlined support struts must be of good low drag design, free of flow separation, and located at least half a rotor blade chord downstream of the blade trailing edge.
6. The number of rotor blades and stator vanes must be unequal and preferably should not possess a common factor. In addition, the product of the stator vane number and rotor rotational speed must not equal rotor blade natural frequency. Support struts must be subject to the same requirements; when restricted to a sector, the preceding conditions must be met on the basis of the blade speed and the period between adjacent struts.
7. An unseparated flow condition immediately downstream of the fan blading is desirable.

2.2 FAN APPLICATIONS

The information presented, should aid designers faced with the problem of selecting in any instance the most suitable fan type.

2.2.1 DUTY-RELATED FAN TYPES

The axial flow fan, in one or other of its various forms, can be designed for efficient operation at any point in the entire fan pressure duty range.

Fan sizes advance according to an arithmetic progression. Under the Imperial unit system each fan is 25% larger than the previous one. This arrangement has limited the number of fan sizes, but in most cases a suitable fan to carry out a specific duty can be chosen. The International Standards Organization has recently put forward a metric series in which the above interval is halved, with a special option for halving this increment once again. The listed sizes range from 100 to 2000 mm. In practical terms, it is suggested that the 25% interval be adopted, giving 250.315, 400, 500, 630,800,1000,1250,1600, and 2000 as the major fan sizes. However where space limitations or other special circumstances arise, the ISO recommendation of 12½% should be given first consideration when selecting an intermediate fan diameter.

Fan applications can be divided roughly into three categories, in terms of flow coefficient and pressure rise capability. These are as follows:

1. A<0.20. These low-pressure-rise fans usually possess a small boss ratio (0.4) and are frequently operated as exhaust fans, without stator vanes.

2. A ≈ 0.20 to 0.40. These units normally have boss ratios between 0.5 and 0.7 and incorporate stator vanes; they are capable of substantial pressure rise gaining from an optimum efficiency design approach.

3. A>0.40. Fans in this category are usually of the multistage in line type, since the single stage total pressure coefficient capability is restricted. Compactness and a possible tip speed reduction are associated features.

A study of fans in the second category indicated that the number of "off-the-shelf" fan types could be limited by selecting two boss ratios and designing these units to achieve the highest nondimensional pressure rise, and efficiency, in their respective classes .The larger boss ratio is 1.25 times the smaller and hence this boss diameter for a given fan equals the smaller boss diameter relative to the next fan size. Some economy of hub and centre fairing components can be thereby achieved.

With the available variables of boss ratio, fan diameter, and rotational speed or adjustable rotor blade pitch, a wide range of duties can be covered efficiently by the preceding two design possibilities.

High efficiency is normally assumed for units possessing rotors fitted with airfoil section blading. However, unless the design and installation of the equipment are fully optimised and the fan is well matched to the actual operational duty, this belief could be ill founded. On the other hand, a well-good efficiency when operating in the designed duty region. Producing a good blade shape can be more expensive when three-dimensional curvature is indicated. The required blade stiffness may result in a metal thickness that requires the construction of a special die for pressing.

A simple computational procedure is presented earlier, that permits the use of two-dimensional curvature but results in curved leading and trailing edges. Hence the centroids of the various span wise stations are radially displaced. Special attention has to be given to root fixing. The preceding procedure is confined to constant chord and blade camber, on cylindrical surfaces.

Any deviation from these restrictions to conical blade surfaces, with varying chord and camber,would require calculations on a trial and error basis. In any case, this technique for avoiding a three-dimensional curvature need has limitations for free vortex flow rotors with zero preswirl. The ensuing rate at which twist is changing with blade span (the major cause of sheet metal blade three dimensionality) accelerates with reducing boss ration. Hence to restrict leading and trailing edge curvatures the fan must have a relatively large boss ration and/or be of an arbitrary vortex flow type.

A prerotator rotor unit design, illustrates an approximate procedure for developing blades and vanes out of sheet material, rolled to cylindrical and conical contours, respectively.

2.2.1.1 ROTOR-ONLY UNIT

Induced draft cooling tower fans are almost exclusively of this type. The discharge diffuser is normally short and have restricted area ratio, thus allowing increased wall angles.

Since the installation is a low pressure rise one, swirl angles will be correspondingly small; this flow rotation can prove beneficial to diffuser performance when careful design and development techniques are employed.

The reversing fan used in many furnaces and drying kiln applications possesses symmetrical blades with flat, wedge, or contoured cross section. Symmetrical, axially aligned stators upstream and downstream of the rotor will only be effective for limited degrees of swirl; this condition is associated with low-pressure rise duties and small flow coefficients. However, the resistance of furnace and kiln equipment is usually appreciable, making the rotor only configuration the enforced one. The loss in efficiency due to swirl is normally unimportant (provided the pressure duty is attained), as it represents an additional low grade-heating source. However, the swirl should not be allowed to persist in closed circuit equipment, thereby affecting fan inlet flow.

Optimised equipment in the preceding two categories usually involves a degree of arbitrary vortex flow in the blade design. This is related to

design problems associated with small boss ratios. Suggestions for dealing with these are contained in Appendix .The fan industry is presently producing a relatively large proportion lf its axial flow units are “tube axial” equipment,
as demanded by a capital cost conscious market. However, increasing power costs must in the near future force the user and fan industries to revise their thinking on stator use.

2.2.1.2 ROTOR-STRAIGHTENER UNIT

Correctly designed units of this type are potentially the most efficient. As a result, the preceding axial flow fan type is employed wherever the power requirements are high, for example, in relation to large mine ventilation and boiler plant aspiration equipment. Because of the inherent flow stability of the unit it is the common choice for wind tunnel applications.

This blading arrangement is particularly suited to multistage units, as the stators remove, for all practical purposes, an appreciable range of swirl angles resulting from changing duty conditions.

Associated with the use of stators is the need for a suitable inner termination. The normal fluted motors used in direct drive assemblies have no stator attachment capability, and hence a motor casing component must be provided. Motor cooling is then achieved in the manner illustrated in Fig.2.6

When dealing with large powers and solids transport, external air must be ducted in and out of a sealed motor compartment. Hollow stator vanes, or special streamlined fairings, are required as airflow passages. An auxiliary fan, the motor cooling fan, and/or static pressure differentials, maintains the cooling airflow. The cooling air can be rejected to the duct airstream when the static pressure of the later is less than the motor compartment pressure. Chamber and/or stator partitioning may be required to separate the ingoing and outgoing airstreams. A large fan of this type is illustrated in plate 6; the unit features an expanding centre body diffuser.

Straighteners normally consist of sheet material of a single. Cambered and twisted plate type or have a double skin airfoil variety. The former are simpler to produce but carry reduced performance and increased noise penalties; these are particularly evident when the fan is operating well away from its design duty.

2.2.1.3 PREROTATOR-ROTOR UNIT

In applications where the axial dimension of the blading unit is limited, the preceding unit can provide an efficient, high-pressure rise solution. For a given duty the solidities of both the stator and rotor blading are less than those for the previous fan type, and the rotor blade setting with respect to the plane of rotation is also less. Provided an adequate gap between the blading rows exists, to minimize noise, an extremely compact unit can be designed. Also, both the non-linear twist change and total twist for a rotor blade is significantly less than those for the previous blading arrangement, using free vortex flow design procedures. This latter feature can be exploited to simplify the construction procedure to allow for sheet metal blading of two-dimensional curvature.

For a given rotational speed the flow acceleration induced by the prerotators increases the relative rotor blade velocity and gives the potential for significant gains in fan unit pressure rise, as opposed to the rotor straight enter unit. However, there is a moderating influence on this increment due to drag increases arising from the blade passage effects .

The use of untwisted, symmetrical inlet vanes of variable incidence as a means of fan duty control on rotor only installations is an efficient method of achieving modest changes in the fan characteristic. However, outside a restricted range of vane incidence angles, departure from free vortex flow condition, and eventual flow separation from the vanes, will lead to a lowering of overall efficiency.

2.2.1.4 PREROTATOR-ROTOR-STRAIGHTENER UNIT

The characteristic curve of a rotor-straightener unit can also be changed by the addition of the foregoing variable incidence inlet vanes, with similar provisos. At the other end of the range fixed, axially oriented, symmetrical vanes are occasionally used in the dual role of residual swirl removers and structural supports for electric motors, bearings, fairings, and so on. From the aerodynamic design viewpoint, these units belong to the previous two categories.

Units in which both prerotators and straighteners contribute to the design duty of the fan are uncommon. Their use is confined mainly to installations with special requirements. For example, the 6.1-m-diameter, 900-kW vertical mine ventilation fans at Mount Isa Mines Ltd. lend them to such a solution .The blading arrangement is identical for both the upcast and downcast units (fig2.76&2.8, Plates 2 and 3,).

Since these fans are sited in housing areas the need to minimize noise led to a limit on tip speed. This in turn results in a relatively large total swirl, ($\epsilon_p+\epsilon_s$) at the blade root. In addition, structural support for the upstream casing and enclosed motor in the upcast fan case necessitates the use of a considerable number of faired struts. Therefore the use of prerotators in the combined structural/aerodynamic role represents an acceptable solution. In the downcast case, the straighteners fulfill a similar motor support purpose. The requirement for identical blading arrangements in relation to both fan types, with its attendant gains of rotor interchangeability and lower capital and maintenance costs, highlighted the advantages of the dual stator arrangement.

At the rotor design condition, 40% of the total swirl is induced by the prerotators. However, since the fans are of a variable rotor blade pitch type, the straighteners are designed for swirl amounts in excess of the 60% remainder, to cover the full range of likely mine duty requirements.

For the downcast fan unit, the prerotators provided a flow smoothing and orientation device under windy atmospheric conditions, thus minimizing rotor blade stresses.

The flow accelerating action of the prerotators can be used to obtain a greater sensitivity for volume flow rate measurement. The pressure differential between upstream wall tapings and those at a selected position on the vane surfaces represents a more adequate quantity than that obtained when the downstream ring is located in the annulus wall, for boss ratios of 0.5 or less. This eliminates the need for expensive contractions, or nozzles.
On a subsequent unit designed for changed duty requirements, the preswirl had to be increased to 50% of total design swirl, owing to speed and fan sizing constraints.

The preceding blading arrangements carry slight efficiency penalties, but overall these are relatively small.

Limits on rotor blade root loading actually fix the amount of total swirl, ($\epsilon_p + \epsilon_s$) b , achievable in design. The increase over the design value for the rotor straightener case is significant but not substantial.The rather detailed description of these mine fans illustrates the type of solution that evolves when all design and operational requirements are permitted to interact.

2.2.1.5 CONTRAROTATING UNIT

This compact multistage unit is currently used in regard to high pressure rise applications. However, excessive noise from existing commercial versions has constituted a major acceptability problem,

particularly when the rotors are adjacent to one another. In the latter case the noise has a peak of approximately twice the frequency of a single rotor; the resultant frequency and intensity are both critical from a hearing loss viewpoint.

There are three different arrangements of fan unit, namely,

1. Motor-rotor/rotor-motor.
2. Rotor-motor/motor-rotor.
3. Rotor-motor/rotor-motor.

Since the motors of the first arrangement can be supported on axially aligned plate systems, aerodynamic interference is minimized, but rotor proximity maximizes the noise. Tie rod mounting of the motors for the latter two arrangements minimizes the aerodynamic and noise problems associated with the motor support systems; unfortunately, some fans in these categories have foot mounted motors supported by a bench and plate system. In certain instances the terminal boxes are directly attached to the motors. Excessive noise and poor aerodynamic performance are the inevitable consequences of such crudity, which is condoned on the grounds of capital cost and ease of manufacture.

To function correctly over a wide range of duty requirements, variable speed is required on the downstream rotor if the axial outflow condition is to be maintained. This refinement is justified for wind tunnel fan equipment where swirl free flow is essential.

2.2.1.6 MULTISTAGE UNIT

Despite the current widespread industrial use of contra rotating fans, the author believes that in most instances a more acceptable result can be achieved by staging rotor straightener units of identical and appropriate design.

These units have been favourably received by Australian mining industry for use in the underground auxiliary ventilation role. As the duct length is increased, rotor units are progressively added to the total fan assembly. To minimize noise and hence eliminate the need for noise attenuation equipment, fan units operating off four pole motors (1450 rpm) were developed. The increased axial velocity that accompanies a boss ratio of 0.7 obtains the pressure capability. However, careful attention must be paid to all aerodynamic design features if excessive losses, potentially associated with high velocities, are to be avoided. Nose and tail fairings for the first and last stages,

respectively, are essential. The duct flow passage should be parallel and free of nonaerodynamic features, with the stators providing the structural supports. When the fan assembly is installed as a blower unit, a curved or conical inlet bell is required with the large open area safety screen attached at the bell inlet (Plates 4 and 5,). Compactness is a feature of this multistage fan equipment.

These fans are in the third category.The restriction on tip speed calls for an increase in the axial velocity component, hence the higher Λ value Since the latter velocity is maintained through multiple stages, the losses are mainly skin friction ones. The subsequent loss over the tail fairing is then only a small proportion of the total gain in pressure for the multistage unit.

When designing a multistage unit for a given high pressure duty, economy and capital cost considerations nay dictate a smaller number of stages, with a tip speed increase and reduced. As a consequence, noise will become an increasing problem for rhich solutions must be found. However, the high blade passing frequency of likely arrangements will simplify the design of satisfactory noise attenuation devices. The ensuing unit could prove more satisfactory from a noise viewpoint than existing less compact centrifugal fans that presently cover this portion of the fan pressure range in the broad industrial scene. However, a crude approach to such a development would, in the author's opinion, have limited or little success and could lengthen the time for eventual acceptance by industrial clients.

With multistage units a "work done" factor is required for use in determining the design total pressure rise coefficient. This constitutes an arbitrary correction for velocity nonuniformity influences.

2.2.1.7 COCLUSION

Broadly speaking, the last two of the three duty categories listed previously favor a "free vortex" design approach, whereas fans in the first category may for practical reasons require an "arbitrary vortex" design approach. The fans in this latter category cover a wide range of cooling and unitized equipment applications. Unfortunately, the restricted upstream and down stream flow spaces, in certain cases, make application of the design equations and methods presented here of limited value. For example, a motor vehicle cooling fan will experience substantial radial velocity components and large scale secondary tip flows, when unshrouded. However, the guidance provided here should lead to improved fan installation layouts and design development procedures

2.3 VARIABLE PITCH FANS

2.3.1 General

A principal factor determining the performance of all fans-axial, mixed flow, or centrifugal is the angular setting of the blades, particularly the angle made by the outlet edge with the tangent to the direction of rotation .As this angle is increased the forward component of air velocity on which the pressure depends remains substantially constant. It follows that volume flow could be controlled if the angular settings of the blades could be altered while the impeller was rotating.

The variable pitch aircraft propeller and the Kaplan runner for hydraulic turbines show that pitch adjustment in the axial configuration can be developed to the highest levels of reliability. Fans with variable pitch axial impellers have been in successful service for many years, though in an unnecessarily limited range of application. With increasing emphasis on flexibility of control and energy economy, both in air conditioning and process work, their use is spreading. Centrifugal impellers have been produced with variable pitch trailing edges to the blades, rather after the pattern of aircraft wing flaps. However, the mechanical design problems of this configuration are severe and it is seldom encountered.

Variable pitch axial fans can be used with constant resistance, constant volume flow, or constant pressure systems in fact with any form of system characteristic. The overall energy savings are among the highest available with any form of duty control. A unique feature is the ability to control volume flow down to zero even at constant pressure, and to produce reverse flow if required. Standard pneumatic or electric continuous control system can be employed, and the speed of response tailored to requirements. Noise level falls with reduction of volume flow, whereas it tends to rise with damper or vane control; the fall is not nearly so rapid as it is with damper or vane control but the use of a multi speed motor with a variable pitch impeleer largely bridges this gap.

2.3.2 Performance of typical variable pitch fan

The aerodynamic design of a variable pitch axial fan can be exactly the same as that of the general purpose adjustable pitch range..Fig. 2.9 is an example: a 1600mm fan with downstream guide vanes and a diffuser expanding to a 2000mm diameter outlet duct in which the fan performance is measured. The fan total pressure and inlet volume flow are plotted for a pitch angle range from 32° (at the blade tips) to -8 ; this range could be extended if negative flow was required. Contours of constant fan total efficiency are also plotted from the peak of about 86% down to 50%.

The broken lines A, B, C and D are examples of the range of system characteristic which can be dealt with by this fan.

A is a constant resistance characteristic for which the required total pressure drop equals:

$$0.40\ Q^2$$

This is the commonest system, and applies when the required volume flow is delivered (or exhausted) through a system of airways, filters, heater batteries, etc. with the specified total pressure drop varying with the square of the volume flow. In the example the volume flow can be continuously and stably adjusted from 50m^3/s to zero-or to reverse flow if required. The fan efficiency is between 75% and 85% from 50 down to 31m^3/s, that is from 66 down to 16 kW impeller power input.

B is a part constant pressure, part constant resistance characteristic with a total pressure drop equal to:

$$300+ 0.33Q^2$$

Characteristics of this type are met with in air conditioning systems requiring a fixed pressure (here 300 Pa) to operate the final room outlet distribution and control units. The rest of the pressure drop arises in the airways and equipment supplying the outlet units. The example shows a volume flow adjustable from 46 to 21m^3/s down to zero with 75% tc 85% efficiency from 46 to 21^3/s-60 to 13 kW input.

C is a constant pressure system maintaining a constant static pressure of 600 Pa in the 2000mm duct. It is assumed that the velocity pressure at the outlet of this duct (which equals $0.06Q^2$) is all lost (though some could well be recovered) which leads to the following total pressure requirement:

$$600 + 0.06\ Q^2$$

Characteristics of this type arise when the fan supplies a plenum chamber feeding a number of outlets, only some of which are in service at any one time. These may be process supply points, each requiring a fixed volume flow and pressure, or the plenum may be on the suction side of the fan, drawing from a varying number of fume cupboards, for example. In air conditioning the supply might be to a number of floors in a building, each with its own conditioning unit, and each operating for different periods according to occupancy. The fan pitch would be automatically adjusted to maintain constant pressure or suction in the plenum. In the example the volume flow may be adjusted from 60m^3/s to zero and 75%to 85% efficiency is maintained over the wide range of 58m^3/s to 18m^3/s -62 to kW input.

D is a constant volume system maintaining a volume flow of $40m^3/s$ over a range of total pressure drop from 1000 Pa down to zero (or negative pressure if required). 75% to 85% efficiency is maintained over a pressure range from 1000 to 400 Pa with characteristic is required when a specified volume or mass flow is to be maintained against a variable flow resistance, as might arise for example in the fuel bed of a combustion process, in a flotation chamber or pneumatic transport system for powered material or across a filter as it becomes fouled. The fan pitch would be automatically adjusted to maintain constant mass flow irrespective of pressure and the set flow rate could be altered from time to time as required.

2.3.3 Applications of variable pitch fans

The versatility of the variable pitch axial fan will be evident from the above example. It will be seen that over 75% efficiency is maintained over a power input range of 100% to 25% in each variable volume case; efficiency is still 50% at less than 10% of full power. It is not necessary for the whole duty to be dealt with by one fan. Large volume or pressures can be dealt with by using one or more variable pitch fans in parallel or series with a number of similar fixed pitch units, which are switched in and out to provide coarse control steps, the fine control between steps being provided by the variable pitch unit.

Among successful applications of variable pitch axial fans may be cited:

1.Air conditioning system in a variety of designs for commercial and public buildings.
2.Vehicle tunnels and garages where control of carbon monoxide and smoke is required.
3.Underground railways for station ventilation and heat removal.
4.Power stations for forced and induced draught and gas circulation.
5.Heavy industry, chemical and metallurgical, for process control, drying and cooling.
6.Textile industry for humidity control.
7.Oil industry for large scale “fin-fan” heat exchange.
8.Laboratory for fume cupboards, wind tunnels, etc.
9.Mines for main ventilation.
10.Factories for ventilation and pollution control.

2.3.4 Mechanical features of Variable pitch fans

Centrifugal forces are predominant in the mechanical design. The centrifugal force on a mass of m kg revolving at n rev/sec at a radius of r metres is:

$$mr(2\pi n)^2$$

For the 1600mm 975rev/min fan of 9.4.2 $2\pi n$ =102radians/sec. Taking is M, the mass of one blade, as 3 kg, and its radius of gyration, rg, as 550mm, the centrifugal force per blade will be:

$$3 \times 0.55 \times 102 \times 102 = 17{,}200 \quad N$$

This is nearly 600 times the weight of the blade, which is 3×9.8 = 29.4 newtons and the situation is sometimes spoken of as equivalent to operating in a gravitational field of 600g.g = $9.8m^2/s$ is the acceleration due to gravity at the earth's surface.

Such a force, and indeed forces several times as great are within the static capacity of ordinary ball thrust or needle roller thrust bearings, and these are commonly used to carry the centrifugal force at the blade root while permitting blade rotation with minimum friction. The taper roller thrust bearings used for aircraft propellers are normally unnecessary for fans.

Levers at the base of each blade translate the common pitch angle adjustment into axial movement of a sliding member within the hub. This may be actuated in four ways:

(a) Automatically, by the expansion of a pneumatic bellows of reinforced rubber within the hub against a spring. The bellows is fed with compressed air through a rotary air seal on the shaft extension.

(b) Automatically, by an external pneumatic or electric thruster through a lever system which applies pressure to the stationary race of a ball thrust bearing the revolving race of which is coupled to the sliding actor within the hub.

(c) Automatically, by an external pneumatic or electric thruster which applies actuating force as in (b) while the reaction force is transmitted from the hub to the body of the thruster through a second thrust bearing. This arrangement relieves the main fan bearing of control thrust load.

(d) Manually, by means of a screw jack when the fan is at rest.

Fig.2.10 shows the cross section of the hub of a variable pitch axial fan with inbuilt pneumatic control of type (a). The air pressure in the bellows is adjustable by the control system to any value there is a corresponding compression of the spring, position of the sliding actuator, and pitch angle of each blade. When the fan is running forces must be applied to each fan blade to keep it at the required pitch angle. Left to itself it would rotate to a position near zero pitch angle where the centrifugal forces on it were in balance. Weights W are attached to the blade toot in such a position (at right angles to the blade) that they apply a counterbalancing turning moment and minimize the actuating force required.

2,4 GENERAL ASPECTS FOR SHAPING OF RADIAL-FLOW FANS

2.4.1 SURVEY

The assembly of an ordinary radial-flow fan is extremely simple and offers few constructional difficulties. Taking the common design (Fig. 2.11), the essential features will be stressed. The example is a low-pressure fan by Sulzer with forward-curved blades.

The impeller is free-running. The two outside bearings are connected to the spiral housing by means of a pedestal block. The bearings used are ring –lubricated bearings and ball-bearings. The rear wall of the impeller very often seals with the spiral wall. Since the spiral housing, which is generally of a rectangular cross-section in low-pressure fans, possesses a larger width than the impeller, it is necessary to fit a nozzle or even a tapered piece to connect to the circular duct. The seal between impeller and housing is formed by keeping the intermediate space as narrow as possible. The spiral housing is mainly made of steel sheet which is riveted or welded. The straight surfaces are stiffened by angle-iron. From the standard shape a rectangular air discharge cross-section has evolved, and special transition pieces are necessary if round ducts are to be connected up. Figure2.12 is the outside view. Figure 2.13 shows the section of a medium pressure fan made by casting. This is a design equipped with shaft seal and which can be used for gases.

Coupling the motor with the fan can also be accomplished without using plate pedestals if an integra! Connection shown in Fig. 2.14 is made by using the tubular construction. This is a fan by Sulzer with acid-protective coating.

In order to improve fans with high suction capacity the firm Sulzer has transferred the appropriate designs from centrifugal machines with double-cured blades to fans. Thus we have blades which are extremely similar to Francis turbine runners (Fig. 2.15). The impellers are stamped and welded together with disc and ring. Figure 2.16 shows a double groove runner of this type.

In double-groove fans the design of the housing creates some difficulty.Figure 2.17 shows in ingenious solution of this problem by Messrs. Sulzer.

The use of external rotor motors with fans in various cases leads to considerable simplification of the overall assembly. The firm Zeol-Abegg is occupied entirely with this field. A very interesting type, example, is the installation of a motor of this type in a double-ended suction drum impeller as shown in Fig.2.18

A new housing design with many applications has been brought out by Messrs. P.Pollrich (Fig. 2.19). The housing for the drum impeller is connected up to the base plate and is adjustable, so that in this way any required discharge opening angle can be obtained. Another feature is that the drum impeller can be replaced by a high-duty impeller which creates a higher pressure and reduces the delivery volume.

Single-stage fans for high pressures incorporate many features of turbo-compressers. The features of a few examples of this type will be discussed. Heavy rings reinforce the cover plate on the internal diameter. They are useful at the same time for receiving special labyrinth seals. One of the features is that with gases through the shaft is sealed by means of carbon stuffing boxes, but these will not be adequate with very dangerous gases. An absolute seal, however, is obtained by using a fluid stuffing box.

Figure 2.20 shows a small spiral housing for pressures of approx. 0.6 atm. The fan shaft, which is made in one piece solid with the pinion, is driven by a gear. The peripheral speed of the impelleu is about 240 m/sec. Figure 2.21 shows the impeller, which looks like a turbocompresser wheel.

The important point in this design is the gear form. The oil used for gears and the bearings has to be cooled artificially. For this purpose Messrs. Demag and other firms use water-cooled tube coils, which are immersed in the oil bath. The efficiencies obtained with these fans are in the region of 80%.

Figure 2.23 shows a gas fan dedinced by Messrs. Demag for high pressures. Recently Messrs. Demag have been constructing impellers with radial-tipped blades which have an axial intake. These impellers, which were originally developed for superchargers, have proved eminently suitable for general application in high pressure installations. Figure 2.24 shows an impeller of this type. This is a remarkable new merhod of construction. The blades are at first welded to the shroud plate (Fig.2.25) and then to the impeller disc.

Still higher pressures can be achieved if the disc design of the impeller is ignored in principle and only the blades retained. Figure 2.26 shows one of these impellers by Messrs. Rateau of Paris, a firm that has made great efforts in the development. Figure2.27 shows a fan of this design with suction at both ends: the fan is mounted in an open housing.

Great demands are often made on the materials used for fans according to the gases which pass through. Processes in the chemical industry involve gases which would rapidly attack steel. In this industry there are fans made of wood (for use with chloride vapours) lead, stoneware, silicon, and other materials. Figure 2.28 shows a fan made by Messrs Sulzer built entirely of wood.

The high duty fan made by Messrs. Schnackenberg (Wuppertal) was constructed wholly in PVC. In this way advantage is taken of the properties of new materials.

The air cooling of Otto and Diesel engines has opened up a new field of application for fans. As the flow resistance, brought about mainly by cooling fins, can now be accurately determined by tests on models, it is possible to create a precise design which allows high grade axial flow fans to be utilised to advantage. As a guide it nay be stated that the temperature of air flowing through the cooling fins is raised by about 50°C. The loss of pressure is about 150-200 mn WG. With a repeatedly varying load the delivery volume of the fan must be careful adjusted. Owing to their flatter characteristic, centrifugal fans are gecerally the most suitable, since with axial flow fans speed regulation or blade adjustment has to be accepted. The interesting solution shown in Fig. 2.30 was utilised in the housing was divided to feed the two cylinders, and a fitted oil cooler was added before the division of flow. In spite of its flat characteristics, the fan is still interesting from the point of view of regulation. In the suction space of the impeller a tubular component is inserted which ensures only partial loading of the impeller. The movement of this component is controlled automatically by a temperature sensitive device. Eckert has given detailed information on the use of axial flow fans in motor vehicles.

Special impellers are used for circulating air in furnaces, where the problem is to circulate air at high temperatures (500-600°C) at any speed. In large industrial furnaces this gives rise to considerable pressure differences when starting up with cold air: the impeller stresses are considerable and it is imperative that the peripheral speed should be the lowest possible. The stipulation that bearings should be kept out of the hot zone is a further complication. In addition, the presence of cold air when starting the furnace calls for three to four times the normal motor power other conditions being equal. In this case it is important for the blower to have the lowest possible driving power at zero delivery so that overloading of the motor may be avoided by throttling when starting up. No conventional impeller satisfies ail the above stated criteria; neither an axial flow fan nor a centrifugal fan hence been found suitable. Figure 2.31 is a view and Fig. 2.32 is a typical installation in a large bolt-butting oven rated at 1500 kw

2.4.2 FORM OF AXIAL-FLOW FANS

In axial flow fans the form of the impeller is of primary interest. The outside housing is of simple construction and need not be discussed at length. The simple form of an axial impeller is to be found in fans whose main task is the delivery of large volumes of air. Curved blades preponderate here. With correct sizing and observation of the basic principles previously outlined, it is possible to achieve good efficiency.

For greater requirements aerofoil profiles will be called for throughout as the basic form for the propeller blades.

Aluminium is the most suitable material for low peripheral speeds, e.g. u<50 m/sec. At higher speeds, aluminium alloys, like Dural, Lautal, Elektron, etc., are appropriate. High efficiencies can only be achieved, however, if the blades are precisely machined and polished after casting. For the individual sections, templates of the aerofoil profiles have to be used to enable the designed form to be reproduced. The need for this extremely difficult finishing work is the reason for the high prices of the rather simple aerofoils.

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The efficiencies indicated in section 108 can only be reached if great care is taken in manufacture. As machining and polishing are simpler and cheaper when wood is used in the construction material, very good results can also be obtained with wooden aerofoils. Escher Wyss developed a design with a relatively small hub for low-pressure factor and high delivery rate (Fig.2.33).

The fields of application of axial flow fans are often such that the resistances to be over come cannot be stated beforehand. Therefore the customer is often is often left with the problem of adapting the fan later to the conditions which exist. Adjustable impeller blades are very convenient in this case. The firm Escher Wyss, Zurich, have been responsible for initiating the development of adjustable impeller vanes.

Even for multistage fans, impellers with vanes that could be adjusted during operation were developed by Escher Wyss. A notable design is shown in Fig.2.34, here the blades are adjustabie from the outside by means of the hand wheel shown in Fig.2.34 the top part of the housing has been removed and the guide vanes situated in front of the impeller are uncovered. Figure 2.35 shows a wind tunnel fan by Escher Wyss.

Figure 2.36 shows a standard model by Escher Wyss with guide vanes and outside ring. The illustration portrays practically the whole of the fan which has been changed into what mat be regarded as a fitting for a pipeline. At higher pressures the hub diameter must increase considerably. A higher pressure impeller of this type where γ values of 0.6 were achieved is shown in Fig. 2.37 . A very large unit in course of erection is shown in Fig 2.36 . The blades of the divided ring, which is grouted into a concrete duct, serve the purpose of bracing the hub of the impeller.

Figure 2.38 shows the installation of an axial flow fan in a small wind tunnel. Axial flow fans are able to achieve considerable outputs at low peripheral speed and also with very little noise. The small appliance with a driving power of 1.5 kW produces a wind velocity of approximately 40 m/sec.

For handling large volumes of air at moderate pressures, the axial fan appears to be an ideal solution. A propeller fan for a cooling tower delivering 60 m 3/sec at about 9 mm WG, installed by the firm Sulzer is shown in Fig.2.39.The axial flow fan solves this problem with an effective power consumption of only11hp.

Mechanical speed adjustment is advisable for many special tasks, e.g. in the cooling fans of motor vehicles. Figure 2.50 shows how this problem was solved by means of a friction wheel gear developed by FKFS adjustable from the outside.There is probably no fan so widely used as the well known table fan. The low driving power and the satisfaction of being able to obtain results even with simple designs may contribute to the fact that this application has received little scientific attention.

It is therefore to be welcomed that certain firms are attempting to market technically up to date designs in this respect. Figure 2.41 is an interesting design by the Samson United Corporation in Rochester,. The blades, which have a twisted circular section, are made of rubber sheets thickened at the hub end. These blades are inserted into grooves of a hub in the shape of a bell. From the aerodynamic aspect the shape is fairly satisfactory and the use of rubber eliminates danger so that the fan can be used without a guard. Owing to the higher efficiency these impellers can be run at low peripheral speeds.

For an impeller of this type the author measured $\varphi=0.16$, $\psi=0.135$. This shape of blade has already been used previously in the United States) for table fans.
Recently, very small cross flow fans have been produced as table fans.

An interesting special design by Messrs. P.H. Frohlich for generating small pressure differences is shown in Fig 2.42. This is a propeller fan design projecting at a slope into a duct so that there is always a portion of the blade actively effecting the flow. Drive and bearing are located outside the duct. Designs of this kind may be interesting for small gas flues where it is often required to boost the chimney draught which may not be adequate. In such cases the power consumption of the drive is readily acceptable.

In the chemical industry, owing to the use of corrosive gases, many special designs are called for. Plastic fans are very popular and, having fairly good mechanical strength, they present no difficulties. In special instances, however, stoneware is the only material which can be used.

Figure 2.43 shows a propeller fan where the impeller and the housing are made exclusively of stoneware. This type of fan is produced for volumes of 10-20 m3/min and pressures up to 12 mm WG.

2.4.3 IMPORTANT CONSTRUCTIONAL DETAILS

The construction of impellers is governed by the peripheral speed. At very high peripheral speeds, as in superchargers, star blade configurations are used which are ail based on the original design by Rateau (Fig.2.26). Highly stresses impellers with side walls call for forged plates in alloy steel.

The blades are riveted on. U- or Z-shaped blade profiles, as shown in Fig 2.44,a,b and c, are used in these designs (Demag design). However blades with rivet heads which are machined (Figs.2.45) (type BBC) are used alternatively. The opened up impeller (Figs.2.45and 2.46) portrays the difference. Welding is not suitable for the high alloy discs which are required in this design. For the smaller peripheral speeds commonly occurring in fan engineering, welded designs are usual, the distortion which was formerly a frequent source of complaint can be eliminated entirely by using modern welding devices and modern welding machines. Semi automatic butt welding is a rather important feature in this construction. Figure 2.47 shows a very large construction by MAN which is completely welded. It will be observed that the impeller has blades of double curvature.

With wide blades these are inserted at various depths in the plate and the whole is then welded together as shown in Fig.2.48. Hollow air-screw blades made from steel are being used lately for axial flow fans.Adjustable guide vanes of radial flow fans have been developed by the firm BBC. Figures 2,49 and 2.50 show typical components of this design.

The main constructional problem in wide blades such as are used for drum impellers is the strength of the drum impeller union. A thick intake ring as shown in Fig.2.51 for instance. Will meet this requirement provided there is an adequately small diameters ratio. Bracing offers further possibilities. This method can be observed in the designs shown in Figs.2.52,2.53 and 2.54 . Finally, the blades can be relieves by intermediate rings. Figure 2.54 shows a single ring and Fig.2.55 several rings. In order to provide relief to the hub and simultaneously self adjustment of the impeller without vending stresses, BBC suggest that for highly stressed impellers the wheel be fastened by a resilient annular spring, a design which clearly points to the influence of steam turbine engineering (Fig.2.56).

In the construction of the fan housing it is often important that the housing can be easily turned in any direction. For this purpose the scroll housing can be mounted on a pedestal which accommodates the driving motor on the inside (Fig.2.57). In the manufacture low cost impellers, each blade may be connected to the hub by a bracket. Fig.2.68 shows examples.

In gas blowers shaft seals are of great importance. The toxicity of the gases may call for absolute tightness of the shaft entry. Fig 2.59 shows four typical designs which have been developed by Messrs. Demag.

2.4.4 AIR-COOLED BEARINGS FOR HOT-GAS BLOWERS

In hot-gas blowers one of the most important applications for fans, the safe cooling of the bearings during operation, is the major factor. If it is not possible or convenient to provide water cooling, artificial auxiliary air currents can be arranged for cooling the bearings. Fig 2.60 shows an arrangement in which a small radial flow impeller is situated behind the main impeller providing air for cooling. Both bearings are encased in steel sheet and the air exhausted by the auxiliary impeller first has to pass over the outer surfaces of the bearings. It is then sucked through the hollow shaft which is provided with radial openings at intake points adjoining the bearings. The air then passes through further radial openings in the hollow shaft into the blades of the auxiliary impeller which are also hollow.

A more difficult problem is that of cooling a bearing built into the hub of an axial flow fan. Such an arrangement is shown in Fig 2.61 . Incorporated in the hub of the axial-flow fan is a centrifugal impeller which is divided by means of a displacer a into two narrow parts. Air is again exhausted through radial openings in the hollow shaft. The air exhausted by the radial blades then passes into the axial flow fan blades which are hollow and is discharged at their open ends. A second current is conducted to the point the guide wheel.

A remarkable design is shown in Fig.2.62. Here cooling air is forced into the motor from the pressure end of an axial flow fan through the hollow wall bracket which supports it. As shown by the arrows, the cooling air is then directed back to the axial impeller.

As relatively small air volumes are involved, the efficiency of the cooling fans is unimportant. Therefore a drum impeller can be used without any disadvantage thus making for compactness. Fig2.63shows an instance where a miniature drum impeller is arranged adjoining the fan housing wall and right next to the bearing. Guide blades enforce the circulation round the bearing.

The most effective cooling is by water. For this purpose generally a bearing bush, i.e. the top bearing bush, is provided with a water-cooling passage. Fig2.64 shows an example of this arrangement. The water enters through branch a and leaves through branch b.

CHAPTER-3

SELECTION PROCEDURE

3.1 INTRODUCTION

Many a time the difference in performance, efficiency and operation reliability among the commonly used fans is apparently so little that one tends to choose either of them without realizing the best. Over the period of time some experience has been accumulated on the choice of fan, which at times is not very much different than the first stage selection.

It has been realized through literature that the following criteria will decide the type of fans to be selected for a particular application.

a) Efficiency of operation and economic benefits.
b) Performance and control characteristics of the fans
c) Maintenance and reliability
d) Effect of erosion
e) Fan acoustics

These parameters are inter related and overlap in certain aspects and one or more of the above points may become more important in certain cases and will form the basis for the type of fan selected.

3.2 FAN SELECTION CRITERIA

The fan selection is an important exercise as Primary air (PA) , induced draft fan (ID) and forced draft (FD) fan together represents a total power consumption of 2 to 3% of power produced in power plants . These fans are generally chaterized by a large volume flow and a moderate to large pressure rise and can be centrifugal or axial type.

The central electricity generating board, UK (CGEB) uses aerofoil bladed centrifugal fans for the induced draft, forced draft, primary and gas recirculating service on boilers. Flat bladed fans with reinforced wearing surfaces are used to draw coal/ air mixture to the burners. However now a days variable pitch axial flow fans are in increased usage for FD fan applications e.g. Castle peak (660 MW boiler) Rihand (500 MW boiler). Centrifugal ID and FD fans have wide impellers and run at comparatively lesser speeds (490 rpm, 590 rpm, 740 rpm) because of large volumetric capacity and moderate pressure requirement. Some of these fans in usage are centrifugal ID fan with double suction being used at 500 MW units of Singrauli Super Thermal Power Station (STPS) and Vindhyachal STPS at 550 and 530 rpm respectively. Similarly, Centrifugal Boiler ID fan at Castle peak B station and Yuanbao Sham , China are operating with these configuration (diameter 3.3m, 743 rpm) and diameter 4.36 m, 493 rpm) respectively. The PA fans have narrow impellers and run at 1490 rpm to generate high pressure required by the system resistance of the mills and burners.

Various configurations and data for ID, FD and PA fans for 500 MW, 210MW and 100 MW boilers, in use in NTPC, are given in appendix B&C. The draft system at 210 MW units, typically icludes two ID fans and two FD fans. At NTPC Badarpur, two axial flow reaction type FD fans are provided to supply the necessary air for combustion. Two axial flow impulse type ID fans are provided to evacuate the flue gases while two single inlet centrifugal PA fans provided primary air.

Aerodynamically, any fan type can be designed to perform any duty. However mechanical consideration dictates that each fans types is most suited for a particular combination of flow and pressure. The selection of fan (centrifugal or axial) for a particular application is primarily made on the basis of the specific speed. The specific speed can be defined as

$$Ns=NQ/H$$

Where H is the head devloped in meter of gas col.

The centrifugal fan is to be chosen whenever the specific speed is less than 80-90.
For all specific speed requirements more than 80-90, the choce should be axial fan. If N in rad/s and H is replaced by gH, the respective values are 1.2 and more than 1.2 for axial fans. Godichon gives the criterion for choice for power plant fans as:-

	Centrifugal fan	Centrifugal fan	Centrifugal fan	Axial fan
Flow control	IVC	variable speed	2 speed & IVC	variable pitch blade
Equipment cost	Low	High	Moderate	High
Flow regulation	A	E	G	E
Sound power level	A	E	G	B
Reliability	G	E	G	A
Ease Of maintenance	G	G	G	A
Energy saving at part load	A	E	G	G

A=average, G=good, E =excellent, B= bad

3.2.1Fan Specification

The right specification of the parameters of the fan at the initial stage, is vital for chossing the correct fan and to operate in an energy efiicient manner. The following parameters need to be specified to enable choice of right fan

Design operating point of fan volume and pressure.

- Normal operating point volume and pressure.
- Maximum continous rating.
- Low speed operations
- Ambient tempratures
- Density of gas at different temprature at fan outlet
- Composition of the gas
- Dust concentration and nature of dust
- The proposed control mechanism that are going to be used for controlling the fan
- Plant altitude
- Operating frequency

There must be a margin in pressure and volume between boiler continous maximum rating (BCMR) duty and the design duty of the fan. The margin from the (BCMR) to fan design duty is normally 20% for volume and approx. 44% increase in head.

3.2.2 WEAR RESISTANCE

A fan handling dust-laden gases is subjected to wear to produced by solid particles contained in the gases. The wear results in metal being cooled away and leads to rotor unbalance and stress modification. Abrasion resistance of fan impellers depend on following factors

- Nature and quantity of dust handled
- Type of impeller
- Tip speed of impeller

The rate of erosion is directly dependent on the relative velocity between fan blade and gases. Fan types ranked in order of increasing velocity are centrifugal, mixed flow and axial. Centrifugal fans have relatively small relative velocity and offers good wear resistance to dust erosion. For centrifugal fans, choosing smaller diameter ratio can further reduce the maximum relative velocity. It has been found that dust erosion is inversely proportional to the pressure coefficient, so greater the coefficient, and the lesser the wear. Typical pressure co-efficient of normal axial impeller, the wear in the latter is considerably less. For this reason, these types of fan have proved very successful in ID application. Additionally, the reduction of wear is obtained by fitting anti-wear devices such as wear plates and anti-wear blades.

3.2.3 VOLUME FLOW CONTROL

Control of volume flows in fans can be obtained by using three different principles

a) Introduction of an adjustable resistance within total system.
b) Aerodynamica principle (change in fan geometry)
c) Speed variation principle

The various aerodynamic methods of controlling the volume flow of the centrifugal fan require the creation of rotating airflow at the inlet of fan impeller in the same direction as the impeller itself. This rotating flow reduces the power absorbed by the fan at the same time as the volume flow is reduced.

Constant speed centrifugal fan can be fitted with two types of inlet control i.e. radial vanes fitted in the impeller inlet (inlet vane control or IVC) or louver blade fitted in the inlet box (inlet louver dampers or ILD). The IVC is more efficient at part load whist ILD is cheaper. A comparison of various volume flow control mechanisms for centrifugal fan (350 MW plant ID fan) is given in appendix 3.2

3.2.4 FAN NOISE

The nose produced by fan comes from both mechanical and aerodynamic sources. Mechanical sources include the drive motor, the bearings, and vibration caused by residual out of balance forces. Sources of aerodynamic nature are as

- Due to the blade passage through air
- Due to forces exerted by fan blades on the air
- Rotation noise due to the passage of blades past any fixed point
- Vortex shredding noise due to flow separation from solid/air boundaries in decelerating flow.
- Air turbulence noise
- Interference noise due to contact by wakes on obstructions and guide vanes.

Noise produced by slipstreams of blades has a frequency (NZ/60) Hz, where z is no. of blades and N is speed of rotation in rpm. The noise is known as siren noise and is prominent factor in the value of overall sound power level. The siren noise of axial fan is louder than that of the centrifugal fan. The sound power level Lw emitted by a fan varies with the fifth power of speed of rotation. Expressed in decibels, the difference in sound power level at different speeds is given by following equation

$$Lw' - Lw = 10 \log (N'/N)^5$$

As stated, centrifugal fan is less noisy than axial fan. This advantage is further increased for all partial loads by using a variable speed drive with centrifugal fan. For a given operating condition, the sound power level (dBA) as measured on ID fan (MCR 250 m^3/s, pressure 3900 Pa) is given below

Fan type	Test block	BMCR
Axial fan with controlled blade pitch	119	119
Centrifugal fan with IVC	112	114
Centrifugal fan with variable speed	110	104

The fan noise spectrum of centrifugal and axial fan is quite different. The centrifugal fan has most of its sound energy concentrated at low frequencies whilst the sound energy of the axial flow fan is spread over a greater frequency range. For human ear, which is less sensitive to lower frequencies, the centrifugal sounds quieter than the axial flow fan.

3.2.5 FAN VOLUTE

A spiral shaped casing is used to collect the rotating gas flow which discharges from the impeller and to deliver it into a single discharge duct. In fan application the diffuser is often dispensed with and the impeller discharges directly into the collecting volute. The volute is usually designed through the application of one-dimensional analysis assuming a free vortex flow and keeping volute inlet flow angle constant.

3.3 EFFICIENCY OF OPERATION

3.3.1 AXIAL FLOW FAN

The axial flow fans in general have consistent high efficiency over a wide range of flow and thus it results in saving in power consumption when operated at part loads as compared to the corresponding radial centrifugal fans. The physical dimensions or the size of axial flow fans for the same duty is also smaller. Characteristics curves of variable pitch axial flow fans with adjustable impeller blades are shown in fig.-3.1

3.3.2 CENTRIFUGAL FAN

Characteristics of the centrifugal radial fans are shown in fig.-3.2 with various inlet vane positions. It can be seen that the constant efficiency eggs are perpendicular to the boiler resistance line and therefore, the constant high efficiency is achieved only for small boiler operating range. In other areas

the efficiency falls abruptly to lower values. Cost wise also, the axial fans are cheaper as compared to the corresponding radial centrifugal fans.

3.4 PERFORMANCE AND CONTROL CHARACTERISTICS

The phenomenon of stall occurs whenever a fan is operated beyond its performance limits. Under the condition when the fan blades are required to provide more lift than they are designed to produce, flow separation takes place around the blades. In this situation, the fan becomes unstable and no longer operates on its normal duty. Each blade/guide vane angle performance characteristics has its individual stall point and the locus of all, the stall points corresponding to the various blade/guide vane angles are generally referred to as stall line.

3.4.1 Stalling in axial fans

The stall phenomenon in axial fan can be more clearly explained on the basis of fig.-3.3 along with the boiler system resistance curve. If the normal boiler system resistance increases for any reason, like furnace pressure excursion caused by a main fuel trip, the normal operating point 'X' will change to meet a new higher system resistance by traveling along the fan performance curve 'A'. As the operating point arrives at point 'I' the fan will stall. Because of the relationship between the fan performance curve 'D' in the stall area and the upset system resistance curve 'B' new operating point 'Y' is found where the system resistance curve 'B' and the stall curve 'D' intersects. If the system resistance remains high the fan will continue to operate at point 'Y' in unstable region.

However if the system resistance reduces, the fan will recover from the stall condition and return it to its normal operation. Further the blade angle is reduced fan will regain stability if the new stall point is higher than the upset boiler resistance curve.

Under the conditions when the stabilization of the fan cannot be achieved by blade angle adjustment of the fan blade angle, the fan must be shut down. In case where the axial flow fans are used, a visual indication of head and volume [or blade angle] in the control room is the minimum requirement for satisfactory operation. Further, in an advanced control it is a practice to have function generator that compares head and flow with the stall line and automatically reduces the fan blade angle to prevent the stall of the fan.

Further, for parallel operation of the axial fans, it is necessary that the two fans share the load equally to avoid one fan operating in / near the stall region. Further while attaching on the second fan care has to be taken to reduce the load on the operating fan slightly so that the new fan is started away from the stall line.

3.4.2 STALLING IN CENTRIFUGAL FANS

The phenomenon of stalling in centrifugal fans is similar to the one discussed for axial flow fans. However it is not so common in centrifugal radial fans. From past experience one can conclude that centrifugal radial fans do not pose any problem in parallel operation where as extreme care has to be taken in case of axial flow fans for parallel operation to avoid operating either fan in stall area. Very narrow band of stall area that too near the shut off head towards the left of the performance curve provides better operational stability for centrifugal fans in comparison to the stall axial flow fans.

3.4.3 PARALLEL OPERATION

Whenever the gas flow of the system is split and each of the branches has its own fan, the resulting arrangement is called parallel operation.

Steam generator, for 60 MW units and above are almost always equipped with several fans running in parallel. In these plants the splitting of the gas flow is mainly done for

- Higher operating safety
- If one fan fails, partial load operation is still possible.
- Smaller components in the single lines, e.g. air pre heaters dampers, ducts fans.
- Better controllability in the partial load range.

Where fans run in parallel, the performance graphs of all fans will add up to one total (fig.-3.4A) or else they can be plotted one above the other (fig.-3.4B).

There are two categories of characteristics lines for single fan operation, one for fans with common resistances and other for fans with common and separate resistances (fig-3.5 & 3.6).

If only one fan is running, this can be represented as shown in fig-3.4B by plotting the plant resistance line on the fan performance curve.

3.4.4 DIFFICULTIES IN PARALLEL OPERATION

Problems will be kept at a minimum if both (or all) fans run in the same duty point of the performance graph, so that dampers, control systems or speed can be adjusted simultaneously.

3.4.4.1Unequal load distribution during parallel operation.

This means that the volume flow of one fan is larger than that of the other one. This mode of operation is however, restricted due to the fact that the fan with the lower capacity might get into the unstable range as indicated in the fig.-3.7

3.4.4.2 Starting a second fan for parallel operation.

One fan is running and other is to be started. Even here the mode of operation be such that neither of the fans run in the unstable range of the performance graph.

Fig.-3.9 shows the principle of starting the second fan (for common resistance only). Fan 1 shows in point 1a of the system performance graph for single fan operation. Fan2, which had been shut off from the system by means of dampers, is started while the inlet guide vanes (or louver dampers) are closed (in case of speed control it is accelerated to the minimum operating speed).

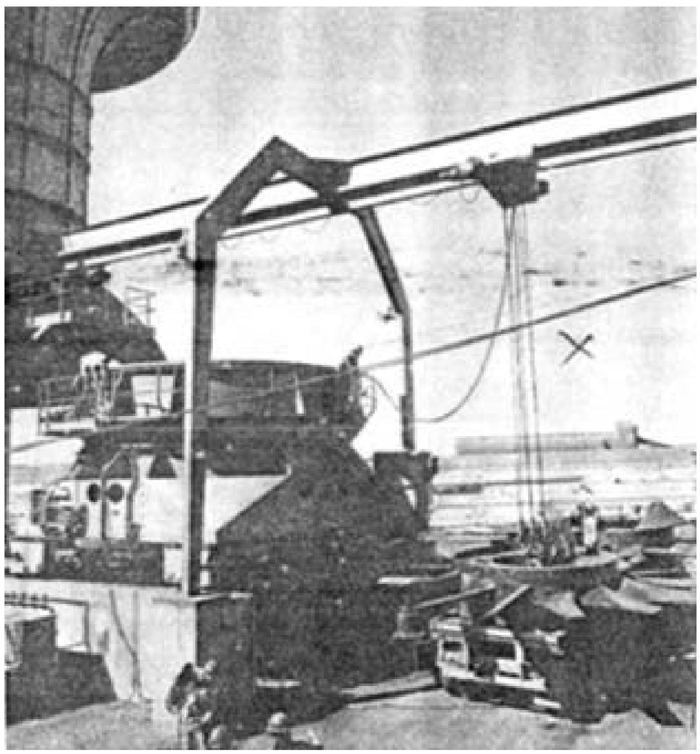


Plate 3. Maintenance provisions for downcast fan, Mount Isa. (By courtesy of Mount Isa Mines Ltd., Australia.)

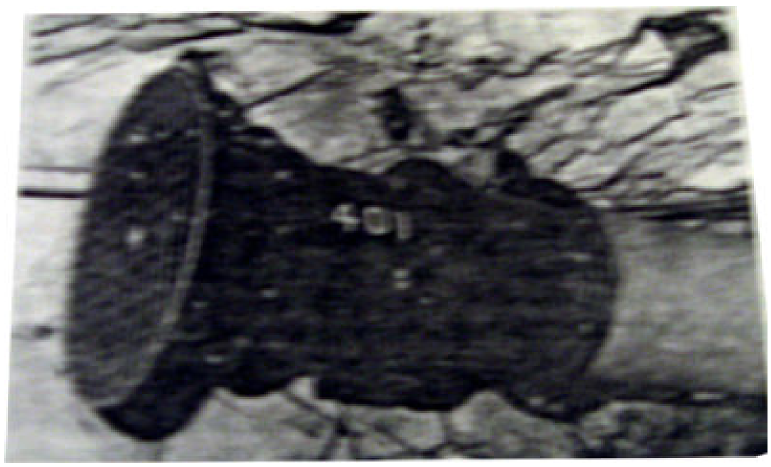


Plate 4. Two stage auxiliary ventilation fan, Mount Isa. (By courtesy of Mount Isa Mines Ltd. Australia.)

Then the control system (louver dampers, inlet guide vanes or the speed) must be adjusted in such a way that the fan runs in point 2a, thus producing the same pressure as fan 1. If the shut off dampers are now opened, the fan will become part of the system. The control device is opened slowly (or the speed is increased).

More the volume, the second fan handles, the higher the pressure rise required for the system, so that the duty point for the fan 1 moves upwards along the performance line, until both fan runs in the same point.

If the second fan started as described above, it is imperative that the pressure rise produced by the fan that had been operating initially is lower than the deepest point of the stall limit of the second fan, as the latter would otherwise have to be run through the unstable range of the performance graph. For separate and common plant resistance, which prevail more than often, the position of the duty point of fan1 must be such that the pressure of the common resistance portion corresponding to the volume handled is lower than the deepest point of the stall limit of the second fan.

In plants having fans with stall limits down to very low pressure values and resistance line which do not run through zero, but require a certain pressure, even at zero capacity, viz., forced draft units, primary air fans units, load is removed from fan1 to ensure that the start up of fan 2 does not cause any problem.

The figures from 3.10 to 3.14 show the characteristics lines of axial and centrifugal fans in common use for utility boilers. The ranges in which parallel fans can be started have been crosshatched.

Presently, the fans are operating on manual mode but unequal sharing of load and occasional tripping of unit is encountered at the time of load variation.

Following are the specifications of axial fans used at these stations

	Unit I	**Unit II**
Total pressure, mm WG (design)	384	358
Flue gas flow m^3 /sec	240	225
I. D. Inlet pressure,mmWG (MCR)	-237	-227
Discharge pressure, mm WG	+39	+33

Method of flow control	Inlet guide vane	0 Inlet guide Vane

The line diagram in fig.-3.15 and the table here under shows the pressure reading at different zones of this boiler.

	Design Draft	***Actual measurement Draft at 200 MW***
Economizer outlet	-47	-54/-55
Air heater outlet	-164	-216/-230
ID fan inlet	-199	-300/-288
ID fan outlet	+33	+115/+125
ID fan outlet	NA	+105/+89

Whenever the unloading phenomenon of the fans occurs, a heavy rumbling noise and vibration in the inlet guide vanes is experienced. The unit load has to be reduced and the corresponding reduction in the inlet guide vanes position of the healthy fan also has to be done. Once the system stabilizes and the fans are equally loaded, loading on the fan is increased in an unpredictable manner with a marked bias corresponding to the unit load. Obviously, this situation leads to disturbance in furnace regime and subsequent tripping of the unit under furnace draft high/low protection.

These observations indicate that there has been a gross under estimation of the system resistance resulting in reduction of operational margin in ID fans. The observations obtained from the fan show that against the predicted pressure of –199 mmWG at ID fan inlet, a pressure of –300 mmWG was been experienced. This has been due to increased drop in pressure all along the system, cumulatively leading to a very high suction pressure at the fan inlet.

Improper discharge duct lay out and high-pressure drop at the chimney also might have contributed to the high-developed head of the ID fan.

3.4.5. Remedy.

The severity of the problem can now only be reduced by reducing the system resistance by removing 12 Nos. of manually/electrically operated dampers in the inlet path of the induced draft fans and by plugging the various sources of air ingress through seals and ducting.

3.5. GUIDELINES FOR GOOD AERODYNAMIC SELECTION OF FANS.

I) The operating point should lie in the peak efficiency zone at the highest possible vane or blade opening.

II) For fans handling clean air, the maximum operating speed should be chosen so that the unit size is small.

The operating parameters selected should not lie in the unstable zone or even close to the "stall line". It should be ensured that the system resistance curve does not cut the "stall line" and enter the unstable zone. Figures 3.16 to 3.18 illustrate the above points.

Figure 3.16 illustrates the selection of an axial reaction fan for two different systems. The poor selection is the result to the test block being very close to the stall line in actual operation, the fan may tend to operate in the stall region due to inadequate manufacturing tolerances or varying operating conditions.

Hence the test block should not be too close to the stall line.

Figure 3.17 shows the characteristics of axial impulse fan. When this fan is used for forced-draft application, the system resistance in the stall zone causes severe vibrations and noise, which are very harmful for the fan.

Figure 3.18 illustrates the selection of a radial fan .The poor selection is the result of the fan being highly over dimensioned for the system requirement, as it is clear from the vane opening position at the test block. Such a selection causes problems in respect of controllability, besides causing the fan to operate at very low efficiency, which results in high power consumption.

Selection of fans for various applications may be made in accordance with the general guideline in Table 3.

MAINTENANCE AND RELIABILITY

(EXPERIENCE WITH FAN FOR 200/210 MW BOILERS)

Before the inception of 500 MW units at Trombay, the 200/210 MW was largest sized units in operation in the country and forms a backbone of the power generation programme.

The forced outage for 200/210 MW was 29.05%, 27.91%, and 31.59% for different years. Out of this, boiler and their auxiliaries have contributed maximum of the order of 44.75%, 37.22% and 33% during different years.

The break up of forced outages in 200-210 MW units on account of different system is indicated in Table 4.

The break up of forced outages due to I.D., F.D. and P.A fans are given in Table 5 and break up of partial loss of generation due to these fans are given in Table 6.

3.7 DRAW BACKS IN VARIOUS TYPES OF FANS

3.7.1 INDUCED DRAFT FANS

Since there are normally two ID fans in a unit and capacity of each ID fan is designed to have 60% MCR load. There should be no outage of complete unit on account of ID fans. There are, however, 63, 38 and 13 days of complete outage of the 200 MW units on account of 10 fans, during different years. Equivalent full load outage due to partial loss of generation due to ID fans are 32 and 39 days during different years.

The main problems in ID fans causing those outages are as below:

i) Overloading of ID fans.

Overloading of ID fans has been observed. It may be due to

a) Excessive air leakage of gas system preheater, duct joints etc.
b) Ash accumulation in the suction duct.
c) More system resistance and losses in multiflue stack.

This has to be analyzed and action taken accordingly. Since the damper available presently is not tight closed, the double dampers need to be provided in the suction of the fans to have 100% tightness. The fans have to be designed considering the system resistance due to double damper.

ii) Failure of expansion joints.

The expansion joints for the ID fans are having asbestos cloth, which is getting disintegrated and requires frequent replacement. The replacement of these asbestos cloth type expansion joints with metallic bellow type expansion joints have to be examined.

iii) Inner Bearing Failure

This is due to improper interference between fan shaft and shaft-sleeve under the bearing. As the bearing is supported on outlet guide vane assembly, the erosion of guide vane failure also leads to bearing failure.
Collaboration has already advised for proper interference fit for shaft sleeve and additional support with struts to outlet guide vane assembly.

IV) Inlet Guide Vane Linkage Failure

The wearing of inlet guide vane linkage is very fast because of turbulence in the suction ducting. Introduction of guide vanes on the suction side has been suggested.

V) Unequal sharing of load between two ID fans.

The unequal sharing of the load between two ID fans has been observed in many units which sometimes even causes the tripping of one fan.

RADIAL VS AXIAL ID FANS

It is felt that the radial fans are having less noise level and gives smooth operation and has more life than the axial impulse fans. The only disadvantage appears are to be large size and cost of the radial fans. Thus option of radial ID fans has to be further looked into.

3.7.2 PRIMARY AIR FANS

It can be seen from Table 5 that these fans have contributed 67, 87 and 46 days of complete outage of the units during different years. The partial loss of generation was 41 and 84 equivalent full days outages during different years.

The problems experienced in P.A. Fans are

i) Cracking of impeller hubs at welding or at surface.
ii) Failure of screws flow hood etc.
iii) Cracking in bearing housing pedestal
iv) High noise and vibration level in fans.
v) Loosening of PA fan motor wedges.
vi) Failure of motor bearings
vii) Passing of Discharge dampers.

BHEL has carried out series of modifications on these radial fans being presently used for this application. The six bladed, wide-width and heavier impeller in Radial fans are being replaced with 18 bladed narrow-width, lighter impellers. The performance of these has confirmed the superiority over earlier design.

Future units should be provided with simply supported type impellers instead of overhung type impellers.

Taking into consideration the large number of outages due to PA fans the feasibility of providing three PA fans need to be discussed.

3.7.3. FORCED DRAFT FANS

It can be seen from Table 5 and 6 that forced outages and partial loss of generation is comparatively very less due to FD fans.

The main problems being experienced in FD fans are:

i) High Noise level
ii) Range able / stability problems especially in axial type of fans having fixed impeller blades.
iii) Loosening of wedges in FD fan motor.
iv) Failure of FD fans motor bearings.

There has been frequent failure of FD fan motor bearings and loosening of wedges in motors.

3.8 EFFECTS OF EROSION

The ID and hot PA fans have to operate in dirty conditions due to the dust present in the flue gases and the dust carried from the flue gases from the air heaters respectively. A great care is therefore needed to ensure cleanliness of lubricating system of their bearings. The lubricating system of these fans should preferably be houses in a closed room, so that relatively dust free atmosphere could be maintained.

Further, the high tip speeds in axial fans made the impeller more susceptible to erosion. Also the pitch changing mechanism in the hub of the impeller may also be effect by the dust erosion and clogging. Due to these reasons, the variable pitch axial flow fans, in general, are not preferred for the ID fan duty, with high ash in flue gases. Operating experience as well as laboratory testing has proven, blades of ferrite alloys and stainless steels are more durable than these blades of aluminum alloy when operated in a fly ash laden atmosphere (figure 3.19). In some cases, chrome surfaced stainless steel strips or nosepieces, have been used to protect aluminum blades from rapid erosion by fly ash. Fan performance will not be affected, as these strips wear because of their minimum thickness of 2 to 3 mm. However, when erosion proceeds completely through these strips, the fan must be removed from service and new strips installed to prevent rapid wear of basic aluminum blades.

However, to drawn the benefit of the higher efficiency and lower cost of axial fans, these should be used for FD and PA fans applications.

As a result of recent development it is possible to achieve the high efficiencies with the radial centrifugal fan with backward curved aerofoil design blades for non-erosive application.

3.9 ACOUSTICS

The FD and PA fans with their inlets open to atmosphere are the main source of noise in the power station. In case of ID fans, only the casing is the source of noise and this too, is reduced by application of proper insulation (absorptive mineral wool insulation).

For FD and PA fans, silencers should be used at the fan suctions to reduce the noise within permissible limits.

The permissible limits of noise being specified for any rotating equipment in the power station, including the fans is 85 dba at a distance of 1.0 meter and at a height of 1.5 meters.

Further, it may be noted that the higher tip speeds in axial flow fans generate more noise as compared to the centrifugal fans.

3.10 COMPARISON OF RADIAL AXIAL IMPULSE AND VARIABLE PITCH, AXIAL FANS FOR SELECTION FOR AN APPLICATION

S/n		Radial	Axial	
			Impulse	Variable pitch
1	Size of impeller	High	Medium	Low
2	Weight of fan	High	Low	Low
3	Weight of rotor	High	Low	Low
4	Moment of inertia	Highest	Low	Low
5	Space required	Highest	Medium	Low
6	Adaptability in system	Medium	High	High
7	Investment cost of fan	Medium	Lowest	Highest
8	Power consumption (considering part load operation also)	Highest	Medium	Lowest
9	Maintenance part (for handling dirty medium)	Low	Low	Medium
10	Maintenance cost (for handling dirty medium)	High	Medium	Low
11	Stability of operation (Without stabilizer)	High	Low	Medium
12 (a)	Stability of operation (With stabilizer)	High	Low	High
12	Wear resistance	High	Medium	Low
12 (a)	Wear resistance (without,chrome surfaced wear plates)	--	--	High
13	Spare parts required	Low	Low	Medium
14	Cost of drive motor	High	Medium	Low

3.11 *ECONOMIC CONSIDERATIONS*

Cost comparison of various types of fans used in stream generators for 210 MW units for different alternative arrangements are indicated in table 7. This also include initial investment on the equipment, saving due to higher operating efficiency and revenue loss due to partial or total unit outages due to fan failures.

3.12 ONCLUSIONS

3.12.1. INDUCED DRAFT FANS

Axial fans have been provided in most of the 200/210 MW units for induced draft purposes. The axial fans are very sensitive to flow path resistance, which has been considered to be one of the minus points of the use of these fans for such services. Also axial fans in many of the installation have an inner bearing which is no t normally accessible. The condition of lubrication of such bearing can be checked only with absolutely reliable thermo-couples and its regressing arrangement is also to be carefully checked. Quite often the supporting structures of this inner bearing also suffer from under cut erosion due to particulate contamination in the gas.

3.12.2. FORCED DRAFT FANS

Outages of generating units due to F.D. fan have been found to be less than those due to primary air fan and induced draft fans. The most efficient from the family of fans i.e. single stage axial reactions fans are normally used for this purpose in 200/210 MW units. Some cases of problems with servomechanism for pitch control have also been met. Manufacturer's quality control and regular inspection by the users seem to be a good remedy. Comparative study of fans for this duty seems to be strongly in favours of axial reaction fans. However, owing to improper response of inlet guide vane actuators and poor resolution of the draft meters provided, there is an increasing tendency among the operators to maintain higher guide vane positions. It has been experienced that operators feel psychologically secure with high guide vane positions, particularly with axial fans, because of their feeling that such fans are prone to instability at low load operation.

It may be possible to program the Data Acquisition System for the fan characteristics and dynamic operating points which could be flashed through graphic messages. This shall enable the operators to know actually at which point of the curve he is at any instant. Such programming will need an accurate input data regarding the actual flow handled by each fan. This arrangement if incorporated could be effective for the correct assessment of operator to operate below the stalling regions of axial fans.

3.11.3 PRIMARY AIR FAN

Radial primary air fans recorded highest number of outages hours in comparison to induced draft and forced draft fans. However, modification of this fan design and replacement of earlier heavy, wide six bladed impeller by narrow, 18 blade, light impeller has shown lot of improvement. Since, fan for this duty has to develop pressures to the tune of about 1350mm W.G. therefore for such low flows and high head requirement improved version of a simply supported 18 blade impeller fan is recommended for use. However, for very inferior coals such as lignite, double stage axial reaction fans may be considered.

CHAPTER – 4

EXPERIMENTAL FACILITIES AND EXPERIMENTATION

4.1Fan Testing Commercial

The current policy of presenting all aspects of fan technology in a sound aerodynamic manner is continued in the present instance. The many complex issue that currently beset the fan industry are not insoluble, provided the problems are approached correctly. Although test codes cannot dictate design procedures or grade their merits, test specifications can be drafted that may provide the customer with additional useful information and incentive from the selection viewpoint.

National technology levels are normally reflected in their test codes. In relation to fans , most would agree that the present situation is unsatisfactory. The difficulty appears to be in reaching agreement on improved methods, particularly on the international level. However, when emphasis is placed on airflow rather than equipment features, satisfactory progress in the upgrading of current test codes can be achieved.

4.1.1SYSTEM EFFECT FACTORS

Commercial laboratory type testing provides the user with “benchmark” data with which to assess the capability and suitability of equipment in regard to the intended duty. When the upstream ducting is designed on good aerodynamic lines and the fan inlet flow is free of swirling, nonuniform flow conditions, then the fan performance can be directly equated to the total pressure loss in the duct system. The performance of adjacent downstream duct components will be modified by poor fan outlet flow quality and this may increase duct resistance above the normal value for such a component.

The code authority for the United States and Canada, namely, the Air Moving and Conditioning Association (AMCA), has accepted the fact that many installed fans suffer substantial performance penalties because of poor inlet flow quality. When applying laboratory test data to a proposed installation, adjustment by the introduction of System Effect Factors (SEF) is recommended . It is a acknowledged that the procedure lacks accuracy, but it is claimed that the correction data are the best available.

However, no distinction between fan types is made. Since the fluid processes by which total pressure is added are different for axial and radial flow fans, the influence of poor inlet flows cannot be identical. The axial unit, which relies entirely on velocity vector changes, will be more affected by a swirl vector than the radial variety, which possesses a large pressure component resulting from centrifugal action. The majority of SEF data has been established from radial flow fans.

In its Fan Application Manual AMCA highlights some of the worst installation features and suggests alternatives. However, this is often a question of relative crudity, as a SEF correction to fan performance can still be required.

The SEF concept is carried through into the field test document of (fig 4.1). Because of the large number of variables in an air moving system layout, adoption of a standard method of field testing has not been considered feasible and hence the document takes the form of a guide. In contrast, field tests in some European countries are covered by code specifications.

The use of System Effect Factors can be avoided by adherence to sound aerodynamic design practice. The duct design and development data presented here, when used in the correct context and within the stated limits, will produce factors of unity.

Greater use should be made of model testing in cases of special design difficulty, as advocated in (fig-4.1). For example, fans in compact heat exchanger units may be subject to substantial radial flow curvatures at fan inlet and outlet, even when the inlet flow is free of separation. These curvatures will affect fan performance; hence experimental studies on a full or smaller scale model are essential in the development of a fan/duct arrangement of satisfactory and known aerodynamic performance .

4.1.2 FAN PRESSURE

Fan pressure must always be considered in terms of total pressure. Static pressure measurements are valid only when used in establishing or estimating mean total pressure at a chosen duct or fan station.

The first fan codes were formulated in the 1929s in both the United States and Europe. It is believed that term *fan static pressure* (FSP) arose out of a test method for centrifugal fans where, for blower configurations, a static pressure measurement in the outlet duct was taken. This quantity relative to atmospheric pressure, which is defined as “fan static pressure,” plus the mean velocity pressure at the measurement station, is equal to the total pressure rise across the fan. However, the flow is accelerated from rest into the rotor inlet, resulting in a negative static pressure; hence the static pressure rise across the fan rotor is not given by fan static pressure. The continuing use of this latter quantity seems to be enshrined in this early tradition, and despite an early attempt to put the emphasis on fan total pressure, the industry continues to give precedence to FSP.

The problem has been compounded by the usefulness of FSP in relation to exhaust fan equipment, where this quantity is equal to the total pressure loss in the duct system. When the system resistance expressed as a K in terms of the dynamic pressure in the ducting upstream of the fan is large, the static and total pressure losses, which differ by only 1.0 are close to equal and hence FSP has a degree of practical if not correct significance to the user. However, under different circumstances the errors can be large and crucial, particularly for axial flow fans

.

The first step toward a rational system is to subdivide fans into two major classes, namely, in line and exhaust equipment. The correct pressures for each of these units are *fan total pressure* (FTP) and *fan inlet total pressure* (FITP), respectively. This eliminates FSP in relation to in line fans and replaces it in the exhaust case by FITP.

In line and exhaust fans are defined as units possessing useful and zero downstream work components, respectively. When the exhaust unit incorporates a downstream diffuser, the combined assembly constitutes the exhaust fan.

Since a rotor unit may be used in either of the two preceding configurations, some conversion between FTP and FITP is required. The total pressure rise through an exhaust fan is equal to outlet total pressure (OTP) minus inlet total pressure (ITP) where OTP is given by the mean dynamic pressure associated with a good quality outlet flow.

For separation and swirl free outlet flows the only matter needing special consideration is in relation to different downstream treatments, when these exist. Adjustments to the outlet total pressure (OTP) for differences in duct resistance and velocity pressure introduced by diffusers or other downstream components are necessary on the basis of calculations or measurements. However, when these flow conditions are not met within reasonable bounds, separate test programs are the recommended alternative. It is an advantage to have test duct components common to both methods.

In seeking a general test solution for all fans irrespective of outlet flow quality, the existing codes have defined the *fan velocity pressure* (FVP) as that based on the mean hypothetical velocity at the outlet flange; this is calculated on gross area, neglecting the presence of internal bodies and surfaces. Unlike the previous proposal, this conversion method carries no provisos in respect to outlet flow quality. Hence in correct aerodynamic terms this procedure in the general sense is erroneous and unsound. Some fans possess outlet flows of gross nonuniformity, high turbulence, and large fluctuating amounts of swirl velocity. These flow features present no measurement or interpretive problems for exhaust fan testing, but for in line fans the situation is different as discussed later.

Therefore the proposal is for testing in relation to two distinct fan types; data conversion from one to the other is subject to an outlet flow quality check.

4.1.3 LABORATORY TESTING CONSIDERATIONS

There are three categories of required testing, namely,

1. **Specific.** A test carried out to determine the precise characteristics of one particular fan. The test conditions of fan speed and air density should approximate those pertaining to the installed fan.
2. **Type**. A test made on a unit that is intended to represent a production run of a particular size and model of fan.
3. **Geometrically similar.** A test made on one particular size of fan that is intended to generate performance data for geometrically similar units of varying size, speed and operational air density.

Testing methods are not normally a function of the preceding requirements, although the detailed data presentations may differ.

Since in line and exhaust fans both have upstream ducting, suitable inlet conditions are not difficult to arrange. A blower fan qualifies as an in line fan when a flared inlet and required for efficiency in addition to test reasons. Unducted diaphragm mounted propeller type fans are excluded from these test categories, although chamber test methods can be adjusted to suit.

The measurement planes have to be located in regions where the accuracy limits of normal instrumentation are not exceeded. On the inlet side, the station must have axial separation from the nose fairing in order to avoid local stream curvature effects, which induce radial static pressure gradients.

The exhaust fan requires no outlet measurement station, since the kinetic energy of the discharge flow is not a useful work component; its influence on fan performance is fully reflected in the power measurement. Since all turbulence and velocity components are eventually dissipated, the ambient atmospheric pressure is the final outlet total pressure and hence becomes a reference pressure.

In the in line fan case, the outlet flow quality will have a pronounced effect on the measurement plane position. Flow separation from motor supports or similar obstructions will always give rise to a turbulent swirling flow. Hence outlet flow quality can be qualitatively discussed in terms of swirl only; this nay be of a steady or unsteady character.

Since a fan is tested over a wide duty range, an amount of negative or positive swirl will be present in all cases, even for fans designed for zero swirl at the duty point; rotor straightener units will minimize such quantities. Provided the swirl does not exceed approximately 10º within the transverse plane, the radial static pressure gradient will be virtually zero, and hence a piezometer ring will record mean static pressure.

Downstream of the motor, for the unit illustrated, three eddies will initially be present, ruling out the prospect of a satisfactory measuring station close to the outlet flange. However these eddies will coalesce further downstream to form a single vortex of a steadier and more stable variety. This condition is usually accompanied by a return to velocity profiles of a more regular type. The measurement plane specified in existing test codes is governed by the latter feature.

Swirl remains the outstanding question with regard to fan performance determination. It leads to pitot static tube errors when the instrument is axially aligned and is responsible for radial static pressure gradients. In addition, swirl is normally considered a nonuseful part of fan output. The codes resolve the instrument problem by adding a flow straightener upstream of the test duct assembly, from fan outlet to the measuring station. This calculation is for swirl-free, fully developed pipe flow, at the appropriate Reynolds number. The additional leading-edge separation loss incurred by the straightener when dealing with swirling flows at large magnitude is tacitly assumed to measure this loss increment.

The user receives the ensuing data in good faith, not realizing that he may have acquired, in terms of flow, a different fan from the assembly-tested one. The problem of excessive swirl, which has been countered by the manufacturer, is now in the court of the user, who may be ill equipped to deal with it in potentially critical circumstances. Incorporating the air straightener into the merchandise an making its losses integral with others in the fan assembly would remove basic aerodynamic opposition to the use of this "fix" device.

Recognition of the swirl properties as part of the fan performance characteristics is recommended. Since swirl and velocity readjustment require approximately the same test duct length, a common test station results. The combined yawmeter and total pressure tube will permit these flow characteristics to be established for fans with a hooked tube oriented along the yawmeter established flow paths. Conservation of angular momentum is the basic principle underlying the preceding assumption, namely, that the swill characteristics measured some distance downstream are representative of the fan properties. Negligible and significant errors will accompany single and multisource vortices, respectively. This procedure should be confined to circular ducts as the rectangular duct walls will modify and place some restraint on the vortex flow pattern.

It is not practicable and barely possible to map the outlet (flange) flow of fans similar to that shown. A reduction in equipment standards will inevitably lead to increased uncertainty in respect to test data and operational properties.

Hence in the overall context the errors arising from the suggested test procedures should be an acceptable risk. The advantage is that the user now has test data that are relevant to the unit purchased.

The preceding suggested development is particularly important when related to axial flow fans. Their limited Kth capacity makes them more vulnerable than the centrifugal type. However, the swirl effects on downstream duct components will remain independent of fan type.

Logically the length of downstream test ducting should be determined by outlet flow quality. However, the codes have to establish a “standard” length of duct covering all fan equipment. A length of 8.5 duct diameters of straight parallel tubing upstream of the measuring station is specified in the AMCA document ; in BS 848 the requirement is for 5 duct diameters. Each code specifies a flow straightener of a completely different type (Fig4.1 and 4.2).

The flow straightener used in the British code originated in Belgium, being developed in relation to the International Standards Organization’s fan code project. The device by permitting radial flow movements enables a more rapid return to uniform flow conditions than does the AMCA version. This is reflected in the shorter duct specified, The outlet duct length should logically be made conditional on flow quality for large scale equipment where available finance and test space are both strictly limited. This could remove the need for model testing in certain instances.

Circular test duct provide greater rigidity and simplify test procedures.

4.1.4 LABORATORY TEST METHODS

In principle, laboratory test methods should be centered on computer techniques, be economic in regard to equipment costs and laboratory space, and be of a limited number to enhance the comparative value of performance data for alternative makes and types of fan.

The AMCA Standard lists 10 test methods from each of which both FTP and FSP data can be obtained. In eight of these an auxiliary fan is required as a load control device, and a means of overcoming the loss in the venturi and nozzle flow measuring systems at the higher volume flow rates. When fan pressures are low. Seven of these methods require a settling chamber of relatively large dimensions particularly when the fan under test is located upstream.

Three of the test methods call for pitot static traverses, the remainder relying on static pressure measurements at various stations. The latter data are suitable for computer collection and processing, resulting in a graphical display of performance.

The two test methods that are independent of auxiliary fans feature the test fan at either the inlet or outlet of the test duct setup. In the latter case, fan loading is applied with a throttle at the test duct inlet, and hence a duct length of 8.5 diameters is required from the inlet to the measuring station.

A multiplicity of test methods, all differing from the AMCA ones, is a single test procedure, Hence it is reasonable to assume that in the preceding codes one test method could represent in BS 848 .Once again both FTP and FSP are obtained with a single test procedure. Hence it is reasonable to assume that in the preceding codes one test method could represent the optimum, with a minimum number of alternative procedures to deal with special circumstances, such as small volume flow rates and double entry fans.

The inlet test duct of Fig 4.3 illustrates a desirable combination of a controlled entry flow with swirl remover, a volume flow rate measuring arrangement of low loss (Fig4.4) and a loading device (Fig4.5) that ensures a uniform cross sectional loss in total pressure. The sliding plate system for resistance control must, however, possess excellent hole uniformity and be fitted with a fine threaded screw drive. Alternatively, wire screen may be inserted in the box or upstream of the air straightener.

A minimum length of test ducting is a consequence of the flow control designed into the test duct assembly. Calibration of the flow and pressure measuring systems is desirable but not mandatory, since information from Fig 4.4 should normally be acceptable.

The stated objective of computerized collection and reduction of test data is also achieved since traversing techniques are eliminated with this inlet assembly.

Single entry exhaust fan test can be carried out with the preceding test assembly. When the inlet flow to a double entry centrifugal fan is bificated, the test assembly should be attached per medium of a transformation element ahead of the bification. For comparison of performance with axial Units, and single entry centrifugal fans the bification ducts and inlet boxes must be included in the exhaust fan assembly.

The preceding compact inlet flow measuring and fan loading system can be retained for inline fans. Provided the fan unit possesses good quality outlet flows a minimum length of outlet duct equipped with a piezometer ring produces the required data without recourse to pitot static or yawmeter traverses (Fig 4.6).

When the fan outlet flow quality is below standard, the downstream length of test ducting must be increased and traversing techniques adopted. Measurements of yaw angle, total pressure and static pressure are required at specified radial locations. From these the resultant velocity vectors are established for resolution into axial and tangential components. The former are the basis for volume flow rate determination.

The mean velocity pressure of the tangential components represents a nonuseful contribution to total pressure in the process of establishing fan performance.

When laboratory space does not permit the retention of the inlet duct assembly a short length of parallel duct with a flared inlet may be substituted. A suitably designed screen box can be transferred from the inlet duct to the point of discharge.

In presenting the test data, the angle of mean swirl and the mean swirl velocity pressure should be included as functions of volume flow rate. These quantities assume great importance for low pressure units . The customer will increasingly become aware of the qualitative and quantitative value of this additional information.

Additional test methods should be reserved for special circumstances such as when the velocity pressures are too small for accurate flow rate determination. Employing a test method in which the flow is speeded up through a venture, nozzle, or orifice plate will normally require the aid of an auxiliary fan.

The preferred selection is an inlet chamber method. (The chamber diameter is about 60% of that of an outlet chamber rig for a given fan size and type.) Selected alternatives from the AMCA and BS test codes are illustrated in Figs.4.7 & 4.8. The former incorporates the flow measuring system within the chamber and requires greater chamber length.

Exhaust fan tests require an entry duct of specified form (Fig.4.9). The in line fan must have a similar inlet and an outlet duct of suitable length, as discussed previously. However in cases where low velocities make swirl determination difficult, the OTP for FTP calculation might be assumed to equal the sum of the static and velocity pressure at the downstream station. In the case of low flow high pressure blowers the errors ate negligible, but for fans at the other end of the pressure scale the approximate nature of the test results should be highlighted.

Large manufacturing companies with a wide range of fan sizes to test would probably consider the chamber method as the optimum one for general use. A chamber sized for the largest fan can then be used for all fans of lesser diameter. The large laboratory space requirement and high capital cost can usually be justified in terms of workload and operational scale. However, since no special attention has to be given to the open return flow a more fruitful use of floor space can be achieved.

Provided the specified minimum chamber to fan inlet area ratio condition is met, double entry fans of the blower type can be tested by this method. However, exhaust or in line equipment with inlet boxes could require an outlet chamber method or a test arrangement similar to Fig.4.3 , with the latter providing the most practical and cost effective solution.

As implied earlier, laboratory tests are not normally performed on large scale equipment. When model tests are required, the fan size and experimental conditions should be selected with a view to obtaining data at Reynolds number clearly above the critical range (for axial fans).

In the case of geometrically similar units, the nondimensional test results obtained under working conditions that ensure an above Reerit situation should not be extrapolated downward to equipment operating in or below the critical range. Since Reerit is dependent on many design and operational factors, accurate predictions cannot be made.

The test codes allow little or no scope for extrapolation to Reynolds numbers less than the test value. However, many models with speed control can be tested progressively to be predicted for a wider range of geometrically similar fan units.

Measurable air compressibility effects will be present for fan pressures above 2.5 to 3 kPa. The complex problem is resolved in the fan codes by the use of a compressibility factor based on polytropic compression: this is considered more accurate than the previously used isentropic compression factor.

4.1.5 TEST CERTIFICATION

The foregoing rationalization of test methods has resulted in slight modifications to selected AMCA and BS test methods. However, all codes call for strict adherence to their specified procedures which are meticulously presented in the relevant documents. Hence the suggested modified procedures must remain at relevant at present as voluntary options.

In the United State and Canada the AMCA issues licenses to Approved Laboratories that operate in accordance with AMCA standards. The AMCA also tests fan equipment and carries out research in its own laboratories. Central test authorities exist in a small number of European countries, whereas in Australia laboratory registration for industry in general is given by the National Association of Testing Authorities.

The requirement for an AMCA license are enunciated in specific detail earlier. NATA registration is granted on the basis of favorable reports from competent assessors. In Britain each fan company assumes a test responsibility with the National Engineering Laboratory carrying out the supportive research into related code matters. However, a National Testing Laboratory Accreditation Scheme for general industry under the control of the National Physical Laboratory was inaugurated at the end of 1981.

Information on practices in other countries has not been specifically sought, but naturally large and well developed test laboratories and authorities exist throughout Europe and in other parts of the world.

4.1.6 ON-SITE TESTING

With the approaching energy shortage, greater attention is being given it the installed effectiveness of fan equipment. However, the break with the old tradition where the emphasis was on compactness and capital cost, is slow. An upgrading of available design data and reeducation are urgently required.

A detailed analysis of fan acquisition for the mining industry is given earlier. The procedures recommended can be extended to industry in general, provided acceptance of the basic principles by all concerned can be obtained. The building construction industry with its many diverse viewpoints and areas of responsibility is one in which consensus is extremely difficult to achieve.

Briefly the ideal approach to fan acquisition is for a wide ranging and expert study of various equipment alternatives before completing the job specification and seeking tenders. In addition to indicating any special features desired, the document should detail the on site test requirements. When the preceding study highlights a knowledge gap or area of uncertainty, provision should be made for model testing either by the contractor or other qualified authority.

The implementation of these principles within the Australian mining industry is reported in (section 4.1.8); the same principles standards have been successfully applied in other sectors of Australian industry. The minimal cost increase is often recovered within 6 months in reduced running expenses. On occasion the initial study may result in considerable simplification of operational and maintenance requirements, with very large savings. Personal experience with a variety of industrial fan arrangements has shown that it is possible to gain millions of dollars by using better equipment and eradicating lost production time.

Both the AMCA and BS fully acknowledge the vital importance of collaboration between supplier and user in achieving improved installation design and satisfactory test conditions. In South Africa there is now a realization that improved cooperation between the interested parties is very desirable .

The design information available herein on contractions, corners, and short diffusers must result in equipment assemblies of comparable compactness to current installations. Hence the codes should offer some incentive to designers making full use of these data. For instance, a class A test procedure that specified flow quality requirements at the measurement stations, and the fan inlet would be suitable for inclusion in any national code. Provided these conditions were met, the actual placement of measuring planes and points could and would vary greatly with installation features. A strong inducement to most design and test teams to meet Class A standing would subsequently be created.

In addition to flow quality, prime importance should be placed on establishing duct resistance and fan duty in terms of total pressure, by measurement or summation.

The use of vane anemometers in Class A on site test programs is not recommended. Experience has revealed substantial discrepancies between measurements taken with different makes and sizes of instrument, despite valid wind tunnel calibration certificates. Overestimation of flow quantity is the usual fault.

4.1.7 PACKAGED EQUIPMENT

The development of low noise, high efficiency fans for car cooling systems, air conditioners, air heaters, and similar unitized equipment involves testing the fan in a similar environment to the final assembly.

In normal circumstances only the volume flow rate and fan power are required to be determined. (An approximate estimate of system and fan losses is required in establishing the order of the Kth for design.) The chamber methods of Figs.4.7and 4.8 can be used for measuring the former by adjusting the auxiliary fan to give an identical inlet total pressure condition to that experienced by the actual installation. Discharge may be either to the atmosphere or to a duct system the resistance of which can be simulated by an adjustable throttle. The development of a suitable fan can be greatly speeded by this method, which ensures accuracy in capacity determination. A great variety of duct/fan assemblies can be accommodated, the only requirements being to preserve inlet and outlet geometries and to maintain inlet and outlet total pressure conditions.

For large equipment a scale model can vary greatly with type, make, and care in manufacture and assembly. Hence electrical input measurements will not permit accuracy in fan power determination. A simple rope or prony brake system can readily be produced for motor calibration. In very small motors, the power output remains relatively insensitive to input current. However, motor speed reduces quickly with load; hence this variable may provide a more convenient measure of fan power after motor calibration.

4.1.8 PRESENTATION OF FAN PERFORMANCE DATA

The fan codes specify the procedures that must be followed in obtaining graphical data, of which Fig.4.10 is typical. For a variable geometry fan with adjustable stator or rotor blades, the normal presentation method is illustrated. The efficiency contours represent mean values of the measured quantities at different pitch settings.

In displaying type test data the actual experimental points may be omitted. It is important to qualify such data with regard to manufacturing tolerances, surface finish, and Reynolds number trends.

The appropriate test data presented for geometrically similar fans is illustrated in Fig.4.11, utilizing nondimensional coefficients. From these simple relationship the desired performance curves can be constructed for any combination of fan size, speed and air density. For example, blade shapes-particularly in the vicinity of the leading edge, where burrs, casting dressing, closely controlled; blade settings should be held within ±¼° of the specified value; and a nondimensional similarity should exist in respect to surface roughness and to blade clearances at root and tip. Some adjustment to the performance curves, on the grounds of Reynolds number considerations, will be required.

Making accurate adjustments for the latter raises problems as Reynolds number effects are closely related to leading edge shape and surface roughness; these geometric properties the operating Reynolds number is above the critical range, as determined during the model test program, the degree of uncertainty should be within acceptable limits. Complete dynamic similarity is usually unattainable.

A number of the foregoing recommended features are under consideration with respect to a draft Australian test specification.

4.2 CASE STUDIES RELATED TO PARALLEL OPERATIONS OF FANS

Just as in case of electrical machines, fans should also satisfy certain conditions when they are subjected to parallel operation. The basic consideration for the parallel running would normally be the pressure ranges. This aspect is considered even while designing the discharge ducting to have a uniformity of pressure drop. Other consideration would be leakage at the various points resulting in pressure drop and also idling of the machines and finally loss of power generation. Various resistances to flow of air would be another consideration in parallel operation of the fans. Few cases of difficult situations in parallel operation of fans are discussed hereunder.

4.2.1 Case I

At ukai power station, axial impulse ID fans normally run at higher load and they do not face any problem because of no interconnecting dampers on outlet of electrostatic precipitator. Under normal operation these fans run at their rated capacities far away from unstable region so no problem being faced in their operation.

However when the boiler is lighted up only on one forced draft fan is started thereby running it on higher load till the machine is synchronized. Second fan is started while cutting in the first coal mill and difficulties are faced in control at this stage and it runs always in the unstable condition. The reason being that due to PA fan sharing the secondary air load due to air heater leakages and the loading of F. D. fans comes down.

However condition improves just after overhaul, especially with the better performance of the air heater seals. The condition deteriorates over a period of time and the controllability problem exists almost at all times. It is felt that capacity of the fan is higher than required. Axial impulse fan of 107 M^3 /sec. Capacity is being used as forced draft fan. In actual it is running at around 20-30% of its full capacity. This is almost in unstable region of the fan. But running with one fan only will restrict the reliability in operation, as the disturbances in one fan will result in tripping of unit.

4.2.2 Remedy

By improving APH performance as regard to leakage it is felt that the condition of F. D. fan operation may improve. However it is suggested if there is a bisector APH,this problem will not arise.

4.2.3 Case II

Problems in operation of axial fans for induced draft service

Axial impulse type fan was chosen for induced draft system of the steam generators for NTPC units in Madhya Pradesh to develop a design head of 384 mm WG and a flow rate of 240 m^3 /sec. According to BHEL the supplier 25% margin in the pressure and 29.56% margin in the gas flow was maintained during the course of the fan selection. However, from the inception of the commissioning of these units, NTPC has faced operational problems and their subsequent failures. During the initial stage of commissioning, tripping of steam generators on account of high furnace pressure was encountered. When the problem was examined, it was noticed that with one ID fan operation, the unit could generate up to 140 to 160 MW with out difficulty. As soon as the second fan was put into operation and loaded, the rejection of load on the fan was experienced when the other fan get overloaded. The frequency of tripping was three to four times a day when attempt was made to put the fans on auto control.

4.3 EXPERIMENTATION FACILITY

4.3.1 TEST FAN

Variable pitch forced draft fan 'A' of 210 MW unit no.V at Badarpur Thermal Power Station was selected for site testing. The pressure discharge characteristic curves of the variable pitch fan may be changed over a large range without modifying its efficiency. These fans can thus be easily adapted to changing operating conditions and can be operated in the partial load range without affecting considerably economic efficiency. Angle of incidence of the blades may be adjusted during operation by the control oil system. The schematic diagram/overall view of the fan for test set up and technical data appear in fig.4.12 and Table 8.

4.4 INSTRUMENTATION

4.4.1 VELOCITY AND ANGLE MEASUREMENTS

Velocities and angles of the flows were measured with the help of a 3-hole probe mounted on the test section. This probe consisted of a central pitot tube. It has the facility to rotate the probe combination at its own axis to align it along the flow direction. The probe was calibrated in a low speed wind tunnel before use at the test site. The calibration graph for the three probes being used at three measuring stations i.e. fan suction; fan discharge and diffuser exit are shown in fig.4.12. By rotating the probe a null point, the point at which both the pressures are equal, was found out at each location. The angle was then recorded which gives the absolute angle of flow and the readings of pitot (Pt) and tube (Py) were also noted. Using the calibration graph and the difference between Pt and Py the velocity of the flow was calculated.

These measurements were carried out at fan entry (station-I), fan exit (station-II) and diffuser exit. A large number of traverse points were taken from casing to hub at each station. This has enabled the measurement of pressure and velocity variation in the radial direction.

4.4.2 MASS FLOW MEASUREMENT

The flow rate was monitored using the existing fan flow measurement device available in unit control room. However, the more accurate flow rate through the fan was estimated from the measurement of velocity and the area of cross section at particular location.

4.4.3 PRESSURE MEASUREMENT

The total pressures during the experiment were measured from the probe directly while the static pressure was estimate from the Pt-Py difference and the calibration graph.

4.4.4 SPEED MEASUREMENT

A stroboscope was used for fan speed measurement.

4.4.5 POWER MEASUREMENT

One wattmeter method was used for measurement of power consumption. For this appropriate readings were recorded.

4.4.6 MANOMETER

A vertical multitude water manometer was used during the experiment. It could be read up to one millimeter of water gauge.

4.5 EXPERIMENT METHODOLOGY

4.5.1 METHOD OF VARYING OPERATING POINTS

The experiment was conducted on running 210 MW unit at different loads of 110, 150 & 200 MW. To plot reasonably accurate characteristic a minimum six or seven operating points should be obtained by varying the system resistance with the help of fan discharge dampers or isolating dampers before and after the air-preheaters. Since the air-preheater dampers were jammed and the fan discharge damper interlocked not to close during unit operation none of these dampers could be modulated to vary system resistance for obtaining multiple operating points for particular blades angle. Under such adverse situation the only course left was to fix the blade angle of the fan (say fan A at 15 degree) at a particular position and vary the flow through the other fan (Fan B) by changing its blade angle which will affect the flow and pressure in the system thus automatically shifting the operating point of Fan ‘A’ on the fixed blade angle curve. This method had the disadvantage of limited number of operating points for each curve. Using this method three set of reading were obtained for blade angles of 15degree, 30 degree and 45degree. These curves were superimposed over the curves supplied by the manufacturer in figure 4.23, for comparison of the actual curve for fan operation (in conjunction with the system) with that of the curves obtained or standardized test airways.

CHAPTER – 5

EXPERIMENTAL RESULTS

As reported in the previous chapter, experiments have been done on 200/210 MW variable pitch axial flow fan at Badarpur. Flow transverses in the radial direction for different flow rates have been done. The results on the study being presented in figures 4.14 to 4.23.

5.1 VELOCITY PROFILES

Figures 4.14 to 4.16 show the velocity variation from casing to hub for three different blade settings (15degree, 30degree & 45degree) of the impeller. The effect of flow rate is also being shown on these figures. At station 1 which is the entry to the fan impeller the flow is hub oriented for 15 to the flow. There is a sharp velocity gradient between casing and hub and as the flow rate increases the peak point of the velocity shifts towards the middle of the annular space between hub and casing. For 30-impeller blade setting, figure no.4.14, the flow through is hub oriented but has gentler gradient between hub and casing for all the three flow rates. It is also observed from this figure that in the vicinity of the casing the flow is nearly uniform. The velocity profile for 45 degree setting is parabolic in nature with its peak lying at 60 % of the annular space from the casing. This indicates a continual guidance available to the flow. It should be noted that at this angle the flow passages of the fan are widely open.

At station II, i.e. at the exit of the fan for 15 degree blade setting the velocity initially increases in the vicinity of casing and after attaining a sharp peak it reduces towards the hub. However at the hub the velocity again increases. The effect of flow rate change is not very much defined. The profile for 30degree and 45 degree setting possess nearly the similar nature with velocity peak shifting to the mid height of the blade.

Station –III is at the exit of the diffuser which is long and receiving the flow directly from the fan and therefore more susceptible to flow unsteadiness and flow separation. It is observed at this station for 15degree -blade setting that flow is nearly uniform between hub and casing having very low velocities. For 30degree blade setting the velocities are much higher at the hub having practically constant value for a larger part from the casing. As the flow rate increases for higher angle blade setting the flow distribution tend to become parabolic.

5.2 TOTAL AND STATIC PRESSURE DISTRIBUTION

Total and static pressure distribution along the blade height, have been presented in figures 4.17 to 4.22. At station one, the static pressure is low and remains almost constant along the blade height for all the blade height. The total pressure profile is a combination of velocity and static pressure at this station. All the exit of the fan the total and static pressure vary considerably from casing to hub at all settings of the impeller blades. However, at station-III the static pressure is observed to be constant near the casing. There are large changes in total pressure variation at this station from casing to hub.

5.3 FAN CHARATERISTICS

Fig.4.23 shows a comparison of the experimentally obtained values to that of the design data chart for corresponding blade angle settings. It is observed from this figure that the actual values match very well with the design data for larger impeller openings (at higher blade settings) when the flow rate is large. . It is also seen from this figure that at small blade angle settings there is no matching between the two values being imposed on the fan on site at these blade settings.

CHAPTER – 6

CONCLUSIONS & SUGGESTIONS FOR FUTURE WORK

6.1 CONCLUSIONS

Based on the present investigations, which encompassed the critical analysis of fan selection criterion and site testing of the variable pitch axial fan the following conclusions have been arrived.

1. Considering different points requirements and duties to be met by the fan in a Thermal Plant a guideline for the fan selection has been laid down and it has been recommended to use axial impulse fan for induced draft, axial reaction for forced draft and radial fan for primary air.

2. The tests carried out at site shows that for higher blade setting angles the experimental values match well with the designed data.

3. At small values of blade setting angles the off-design performance is poor than the design value.

4. The flow upstream to the fan tend to remain hub oriented at low blade setting angles while it becomes parabolic in nature at high blade setting allowing large flow rates.

5. The static pressure t exit of the diffuser remains nearly constant in large portion of the flow passages.

6.2 SUGGESTIONS

1. Based on the experience gained during the course of the work it is suggested that to arrive a precise selection of a fan for specific duty on a plant, a suitable mathematical model to be developed to overcome the personal ambiguities.

2. Since at low blade setting angles the actual performance shows large departure from the designer's data it is suggested that detailed performance test be carried out at highly off design conditions and reasons for departure be established clearly.

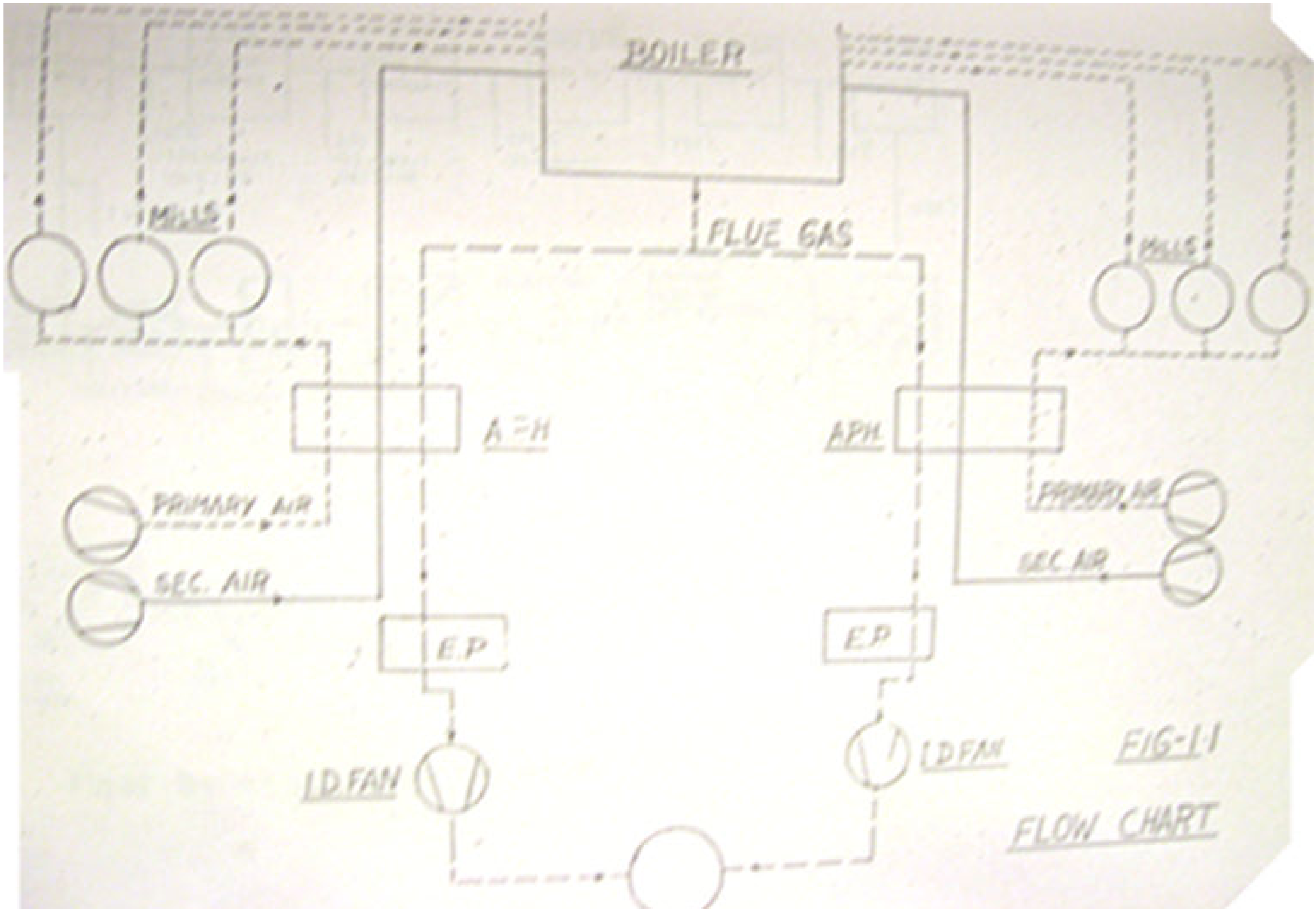


FIG-1.1

FLOW CHART

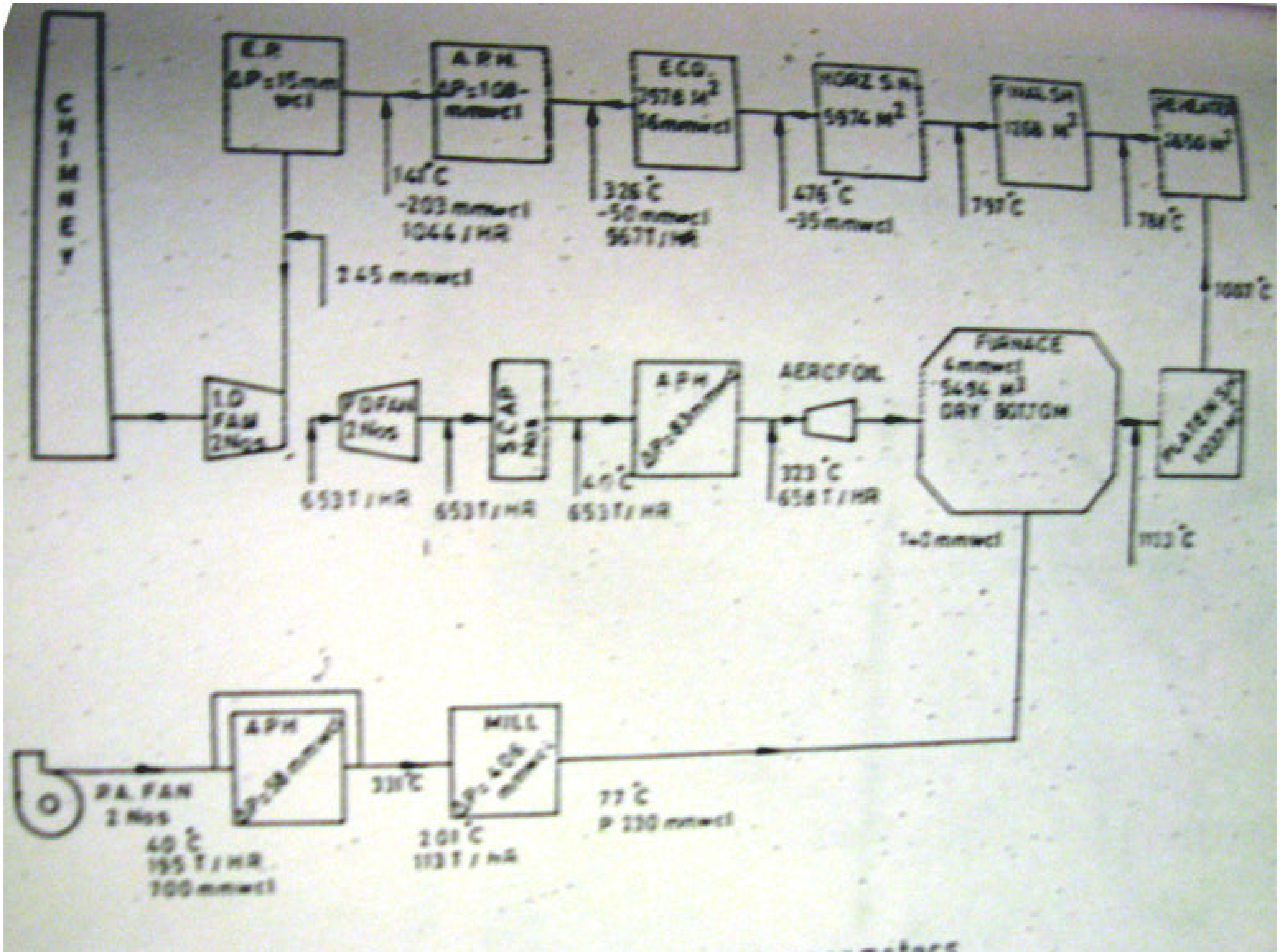


Fig.1-2 Boiler air & fuel gas parameters

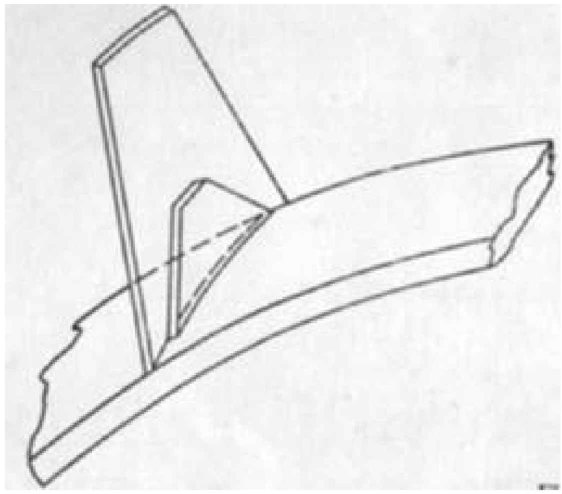


Fig.2.1 Strengthening and stiffening gusset.

Plate 2. Downcast mine ventilation fan, Mount Isa. (By courtesy of Mount Isa Mines Ltd. Australia.)

Plate 5. Assembly of two-stage auxiliary fan components. (By courtesy of D. Richardson & Sons, Melbourne, Australia).

Plate 6 Upcast mine ventilation fan featuring 4:1 diverging center body diffuser, Mount Isa. (By courtesy of Mount Isa Mines Ltd).

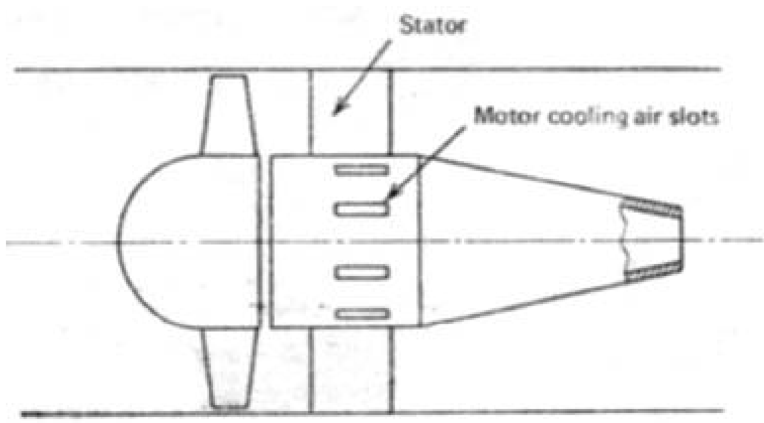


Fig. 2.6 Air exchange system for motor cooling.

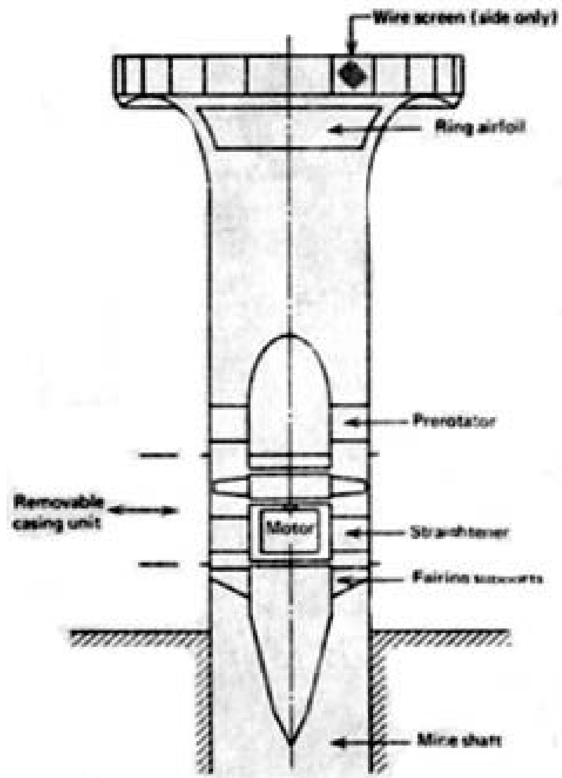


Fig. 2.7 Vertical 6.1-m-diameter downcast mine ventilation fan, Mount Isa Mines Ltd.

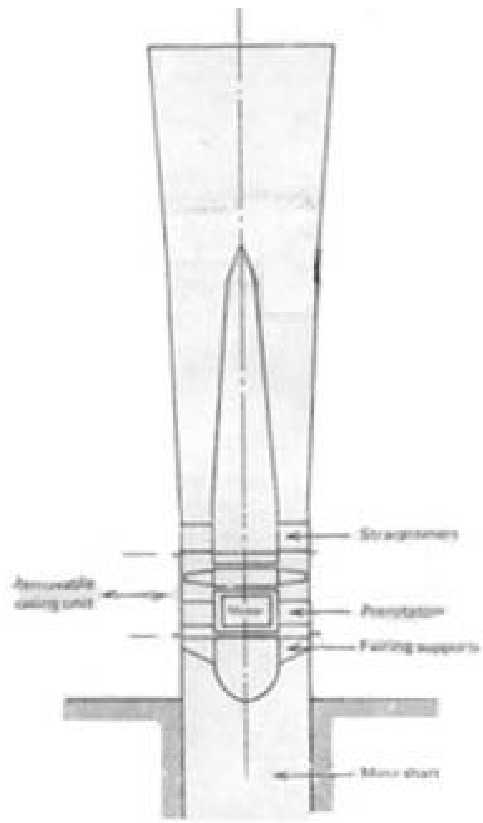


Fig 2.8 Vertical 6.1-m-diameter upcast mine ventilation fan, Mount Isa Mines Ltd.

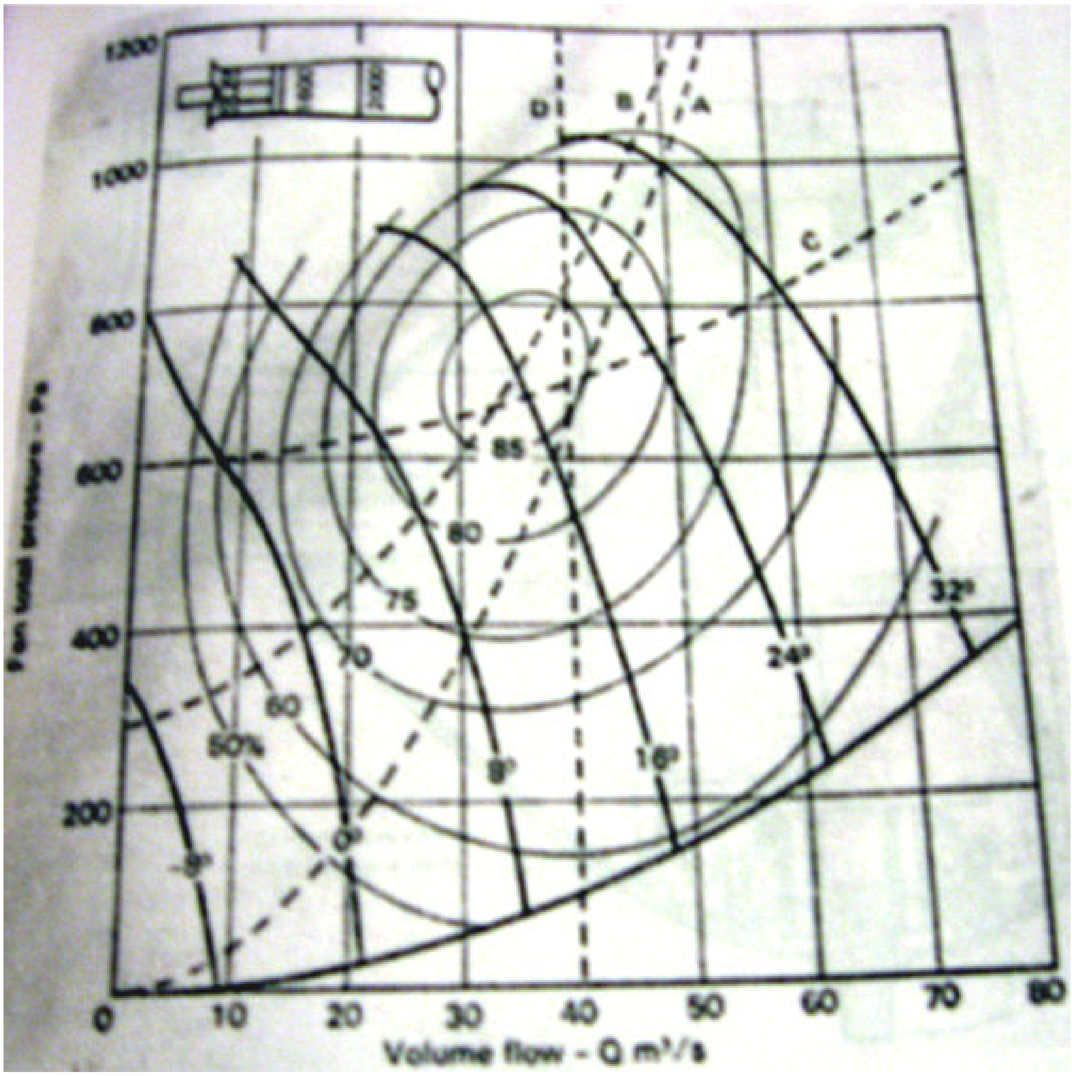


Fig 2.9 Variable pitch axial fan performance: 1600mm, 975 rev/min

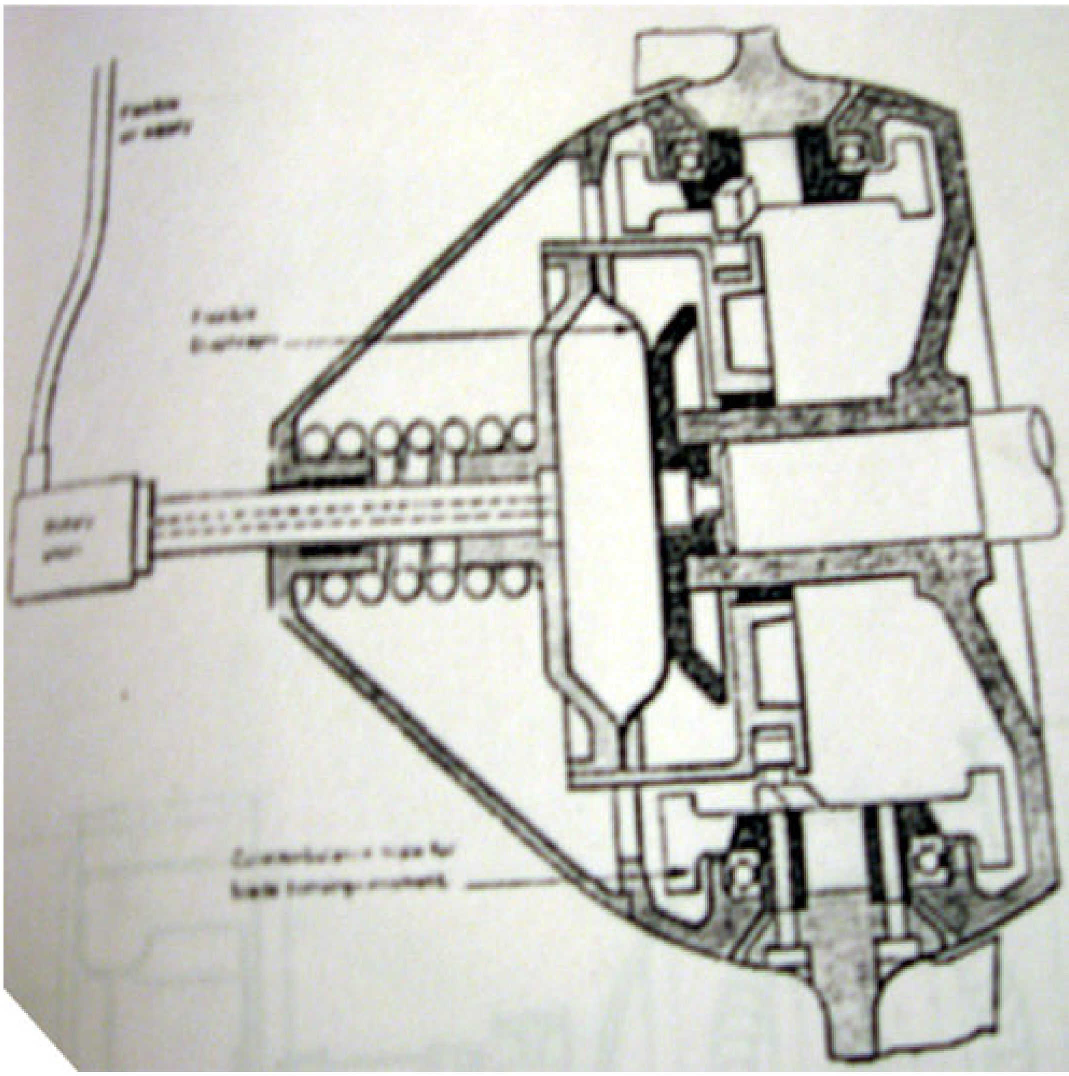


Fig :- 2.10

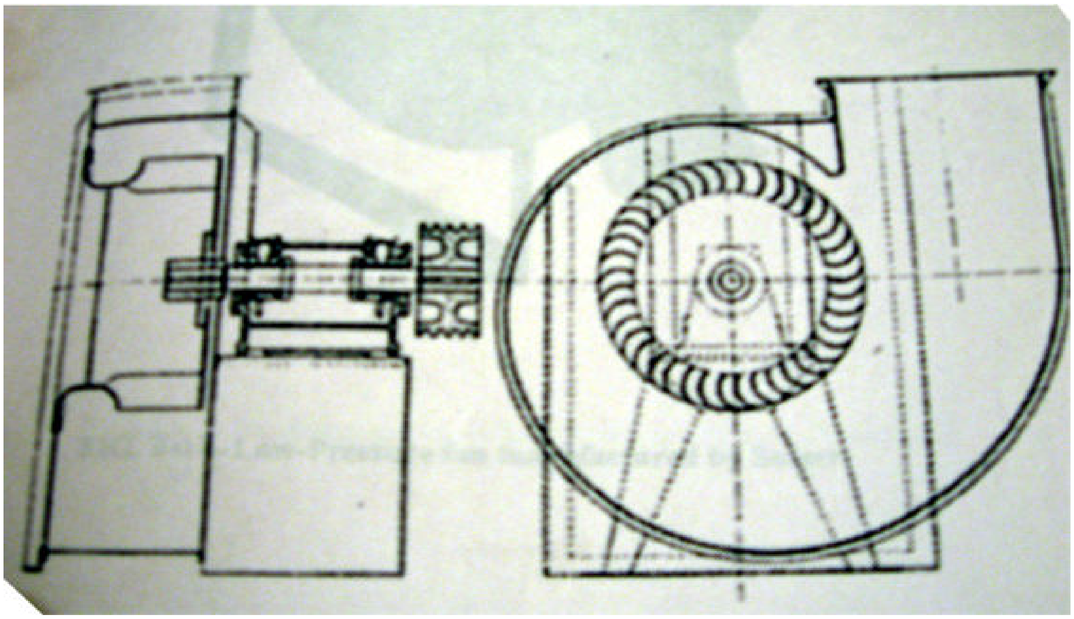


Fig :- 2.11 low-pressure fan manufacturing by Sulzer

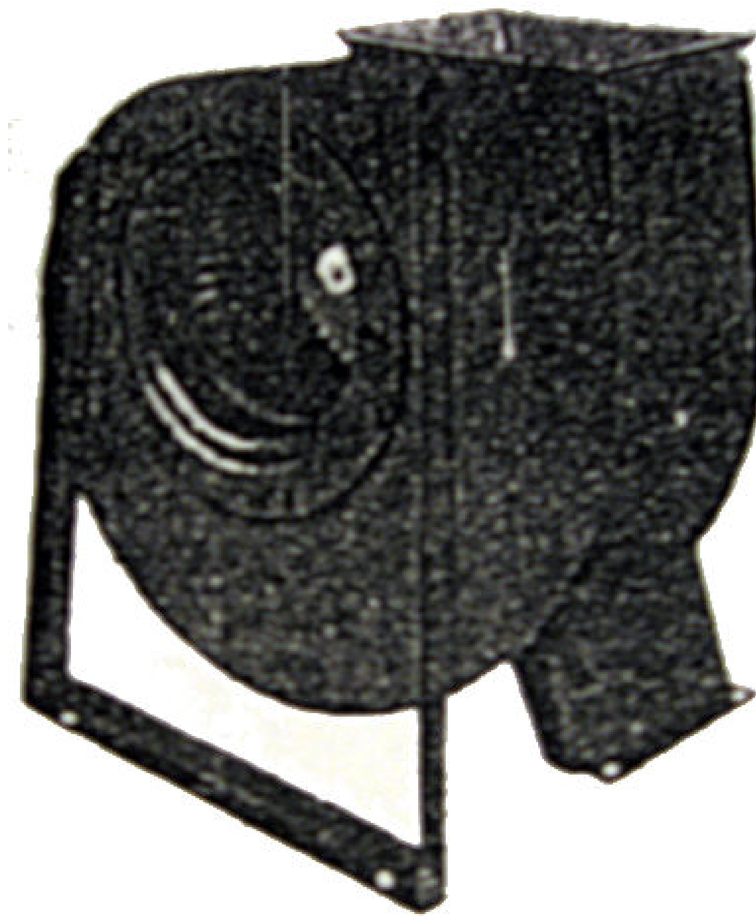


Fig 2.12 Low-pressure fan manufactured by Sulzer

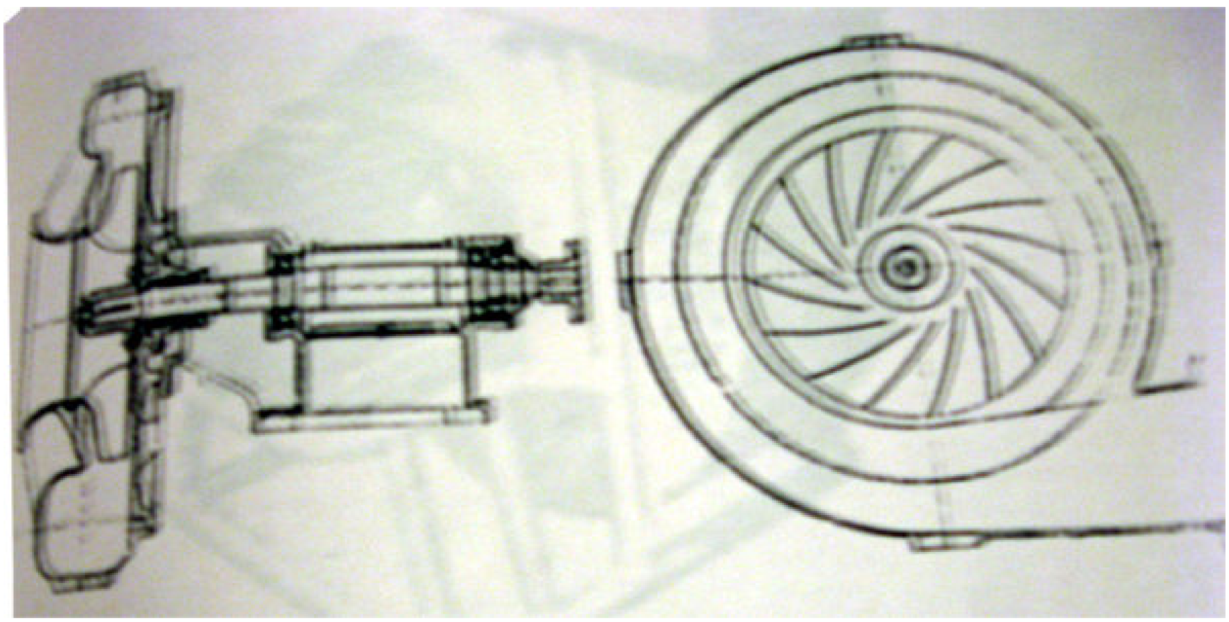


Fig. 2.13 Medium-pressure fan manufactured by Sulzer

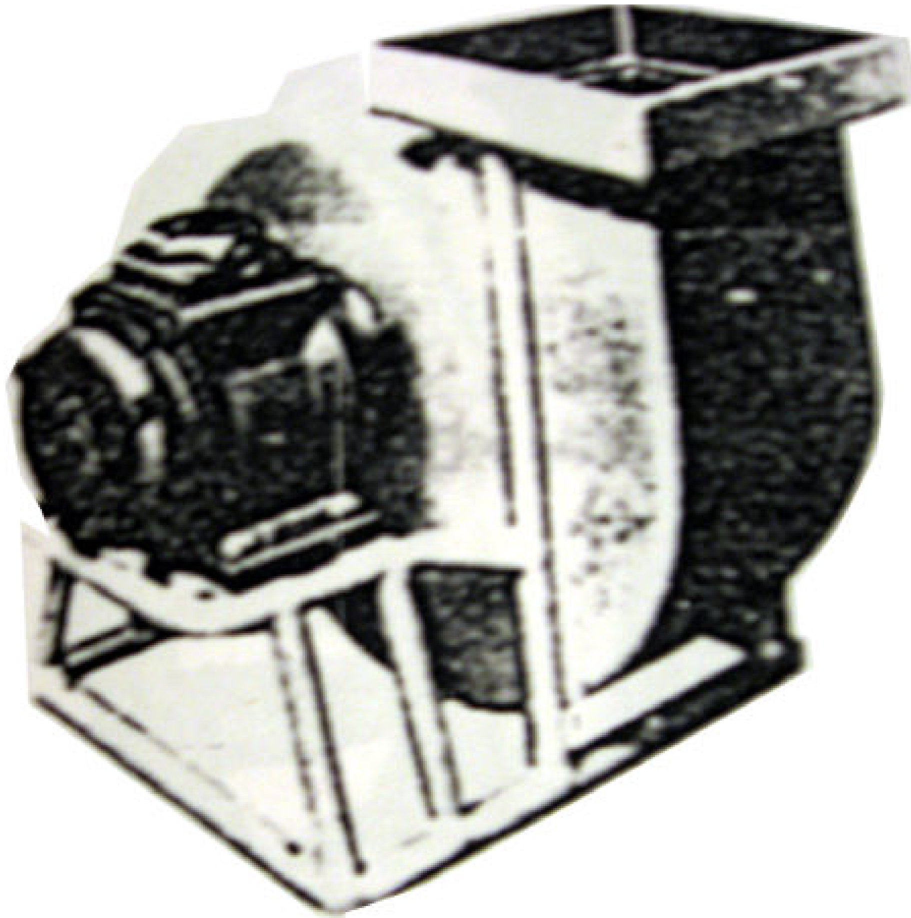


FIG. 2.14 Fan with tubular steel pedestal.

FIG. 2.15 Impeller with double-curved blades manufactured by Sulzer

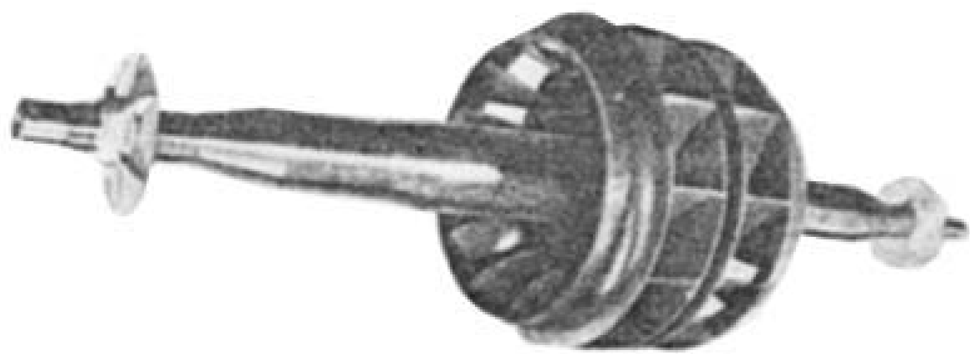


FIG.2.16 Double-entry suction impeller with double-curved blades made by Sulzer.

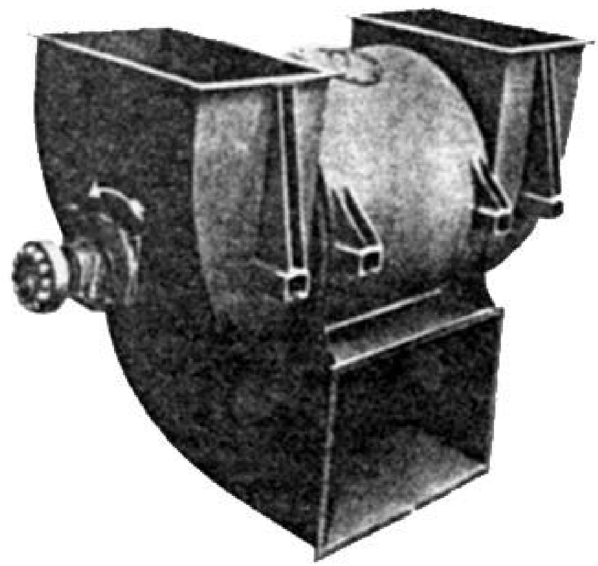


FIG. 2.17 Fan with suction at both ends and discharge branch below.

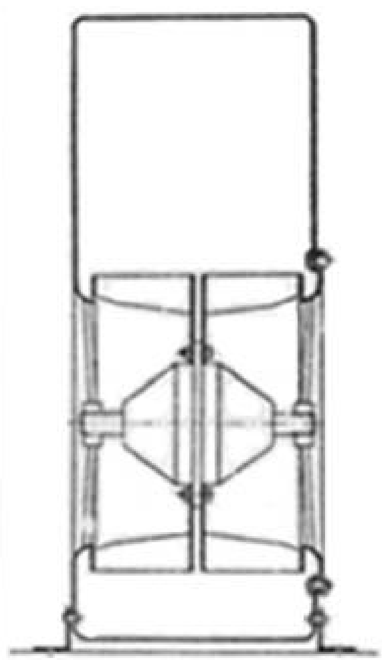


FIG.2.18 Multivane impeller with external electric rotor fitted inside the hub

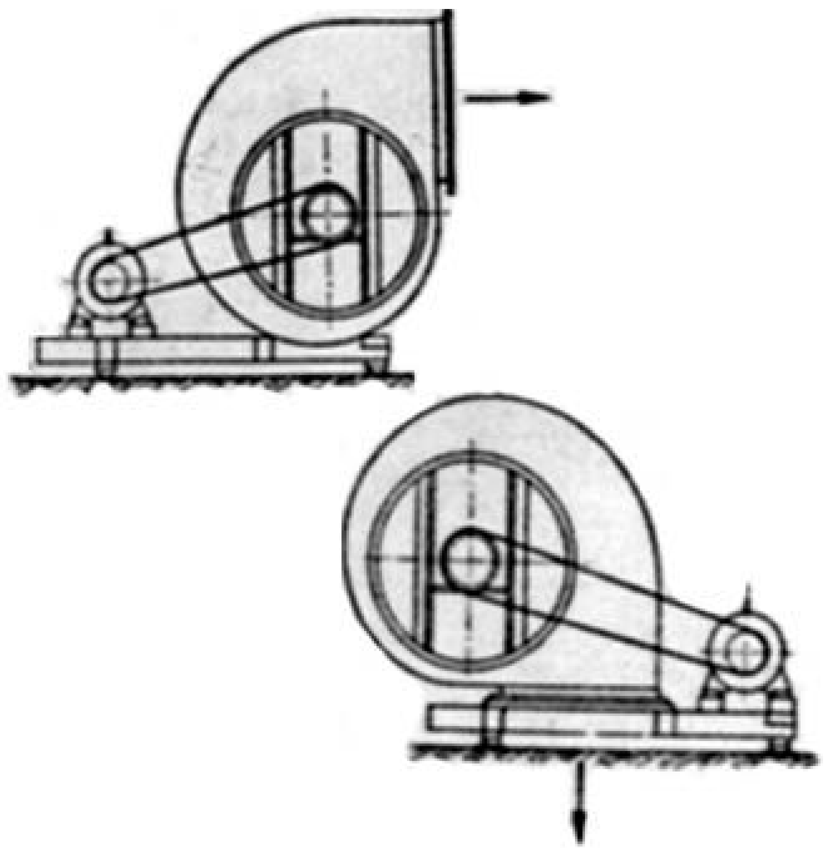


FIG.2.19 Multivane impeller made by Messrs. P.Pollrich with possibilities of orientation in any direction and exchanging impellers.

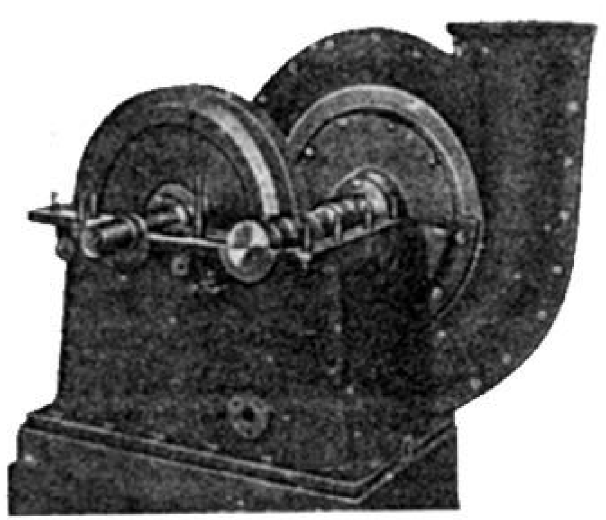


FIG.2.20 Small spiral housing by Demag for 0.6 atm pressure.

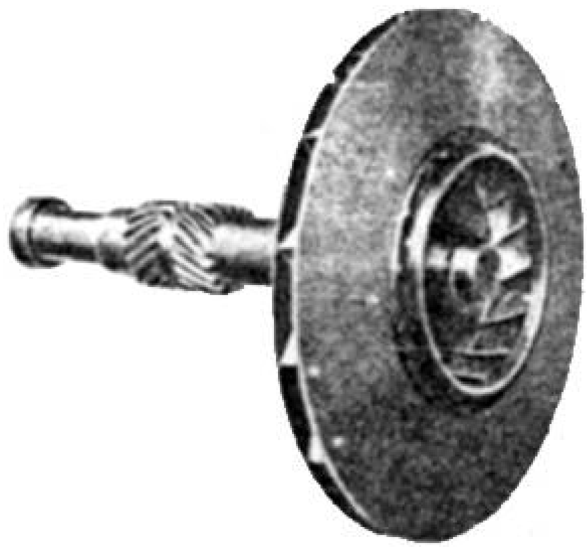


FIG.2.21 Impeller with pinion shaft for FIG.2.30

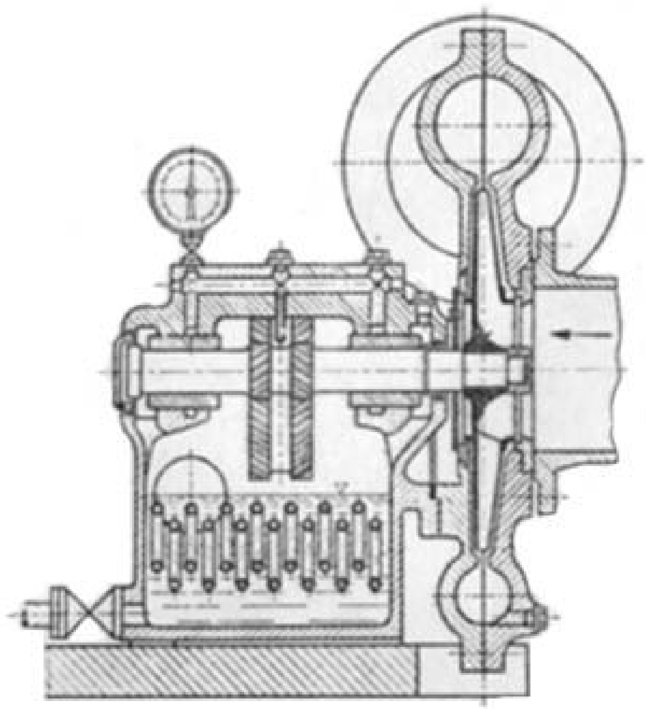


FIG.2.22 Cross section through high duty fan made by Demag. Cooling of the oil bath by water cooling pipes.

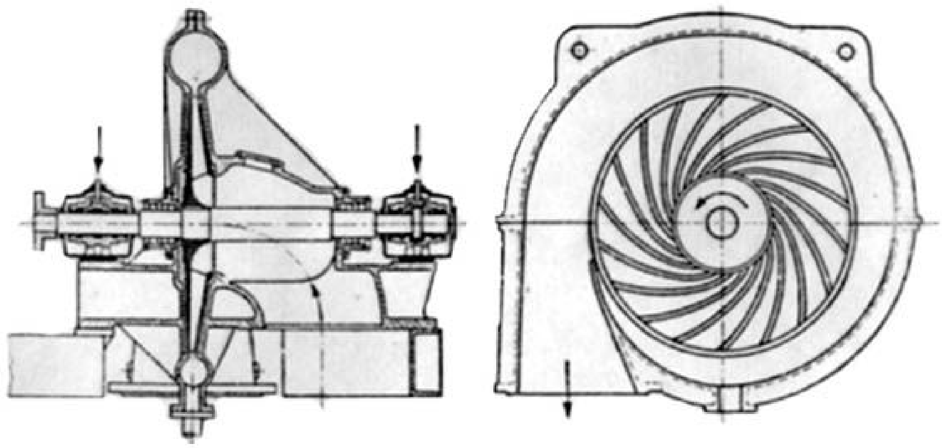


FIG.2.23Gas blower for high pressure (Demag).

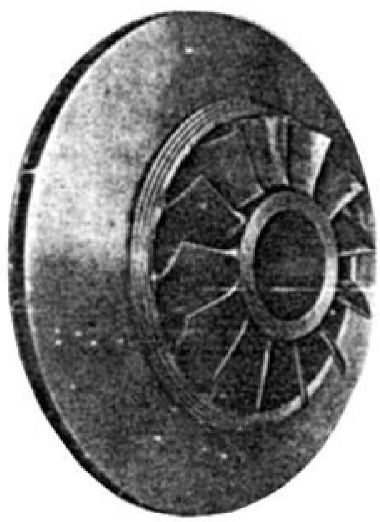


FIG. 2.24 Impeller with fixed blades on the shroud (Demag).

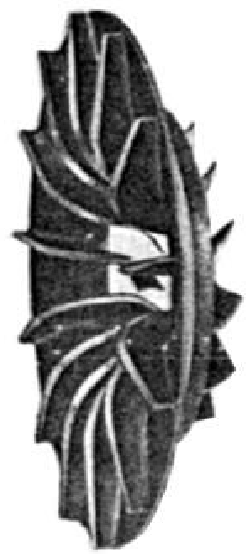


FIG.2.25 Shroud with blades.

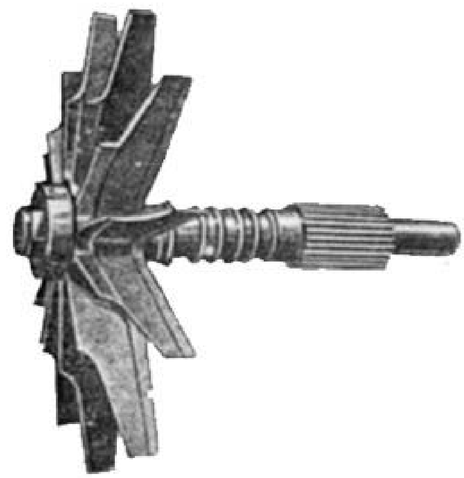


FIG.2.26 Fan impeller for maximum peripheral speed according to Rateau.

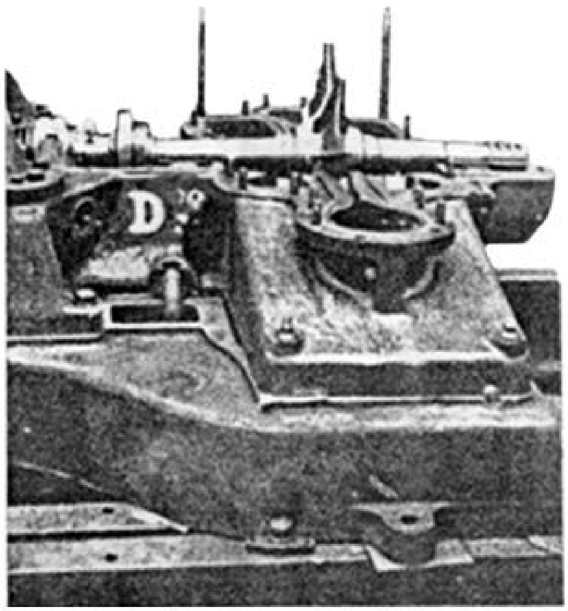


FIG.2.27 Fan designed by Rateau with double entry of air.

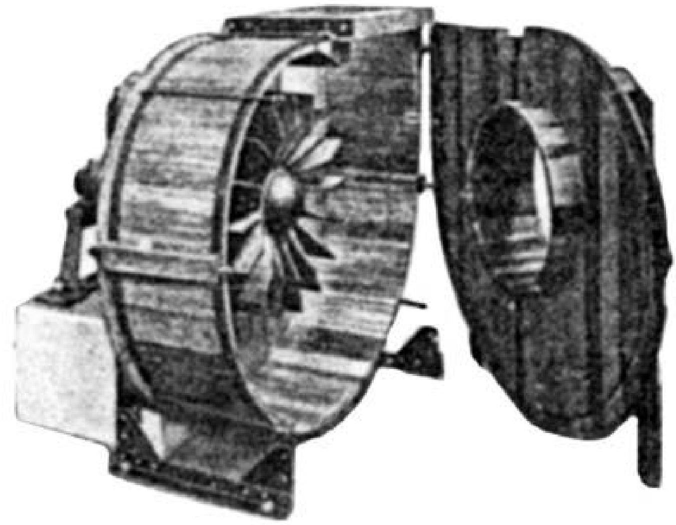


FIG.2.28 Fan constructed wholly of wood made by Sulzer.

FIG.2.29 High duty fan constructed of plastic material made by Messrs. Schnackenberg.

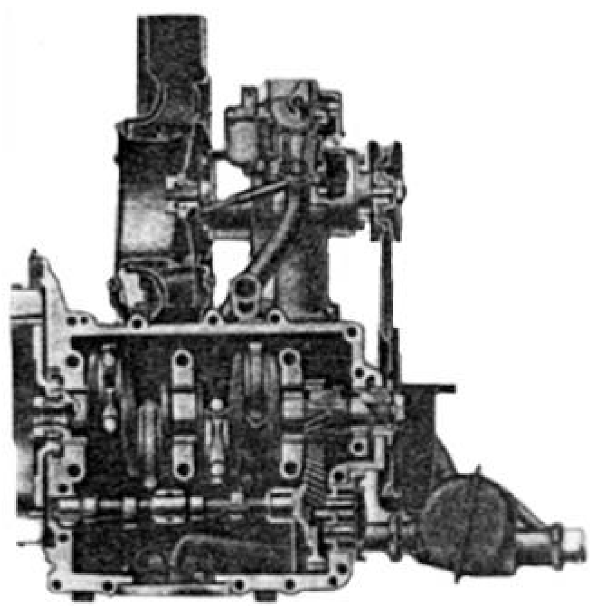


FIG.2.30 Centre line section through cooling fan of Volkswagen engine.

FIG.2.31 High duty fan designed by Eck.

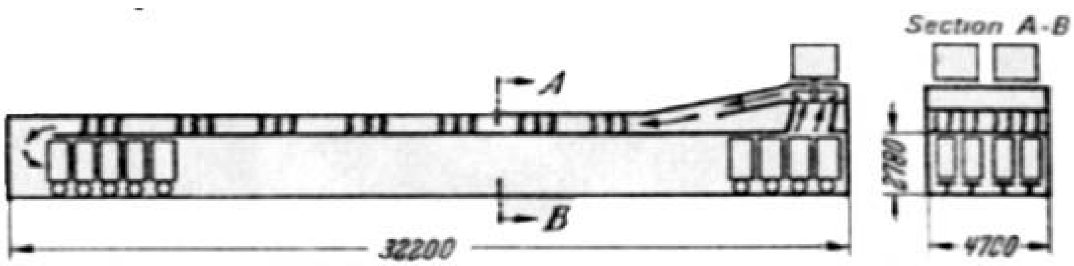


FIG.2.32 Section through a large furnace with air supplied by fans. (Otto Junker in Lammersdorf.)

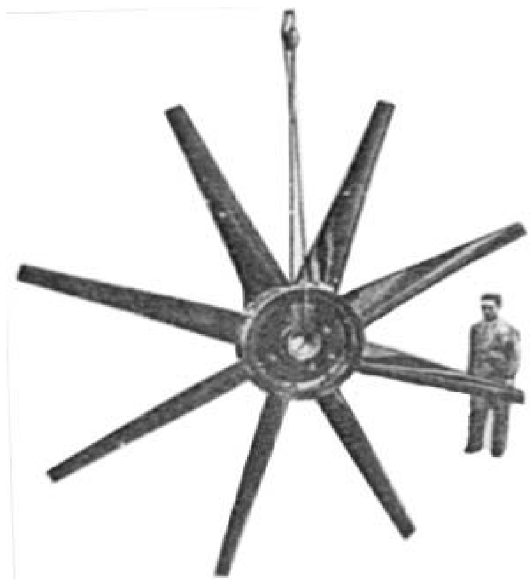


FIG. 2.33 Axial flow fan made by Messrs. Escher Wyss for φ=0.193, γ=0.045;D=300 mm dia.

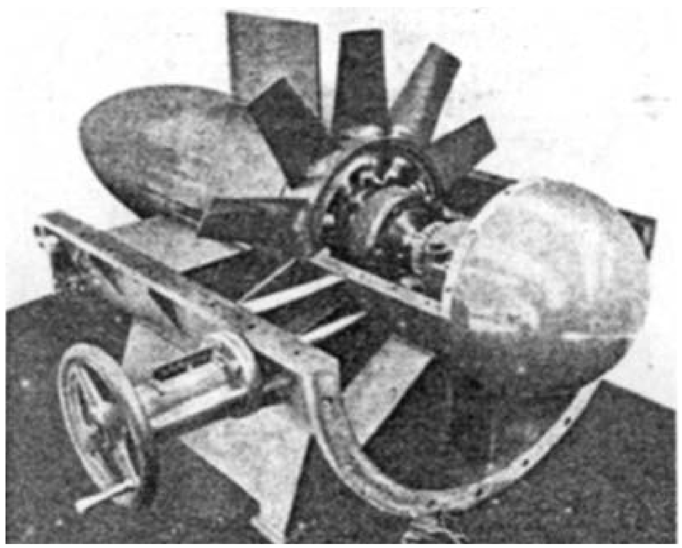


FIG.2.34 Axial flow fan with blades adjustable from outside.

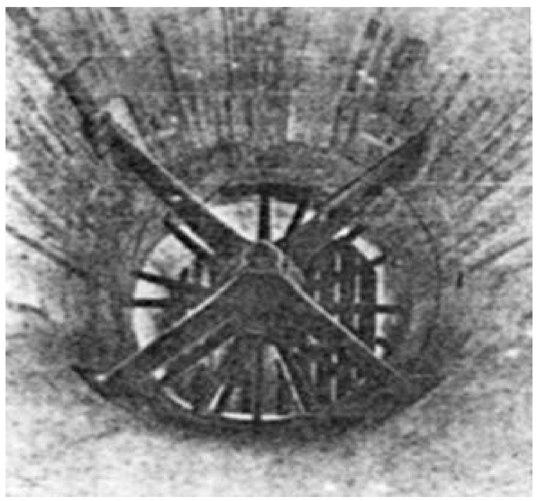


FIG.2.35 Wind tunnel fan made by Messrs. Escher Wyss.

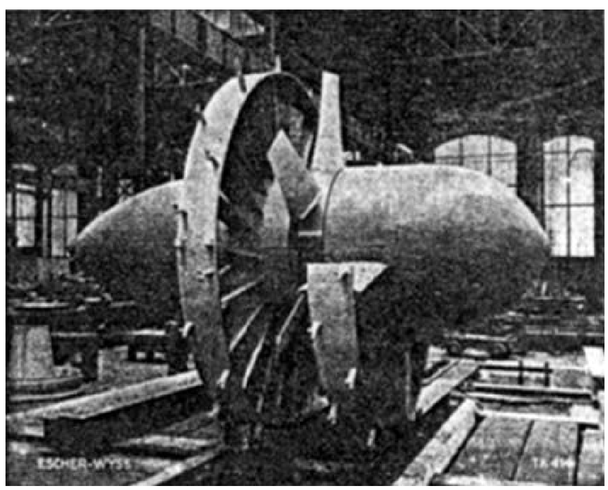


FIG.2.36 Axial flow fan for large delivery volumes made by Messrs. Escher Wyss.

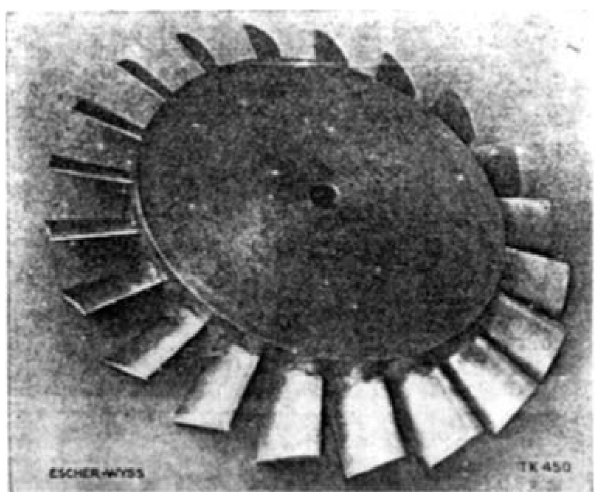


FIG.2.37 High pressure impeller made by Messrs. Escher Wyss.

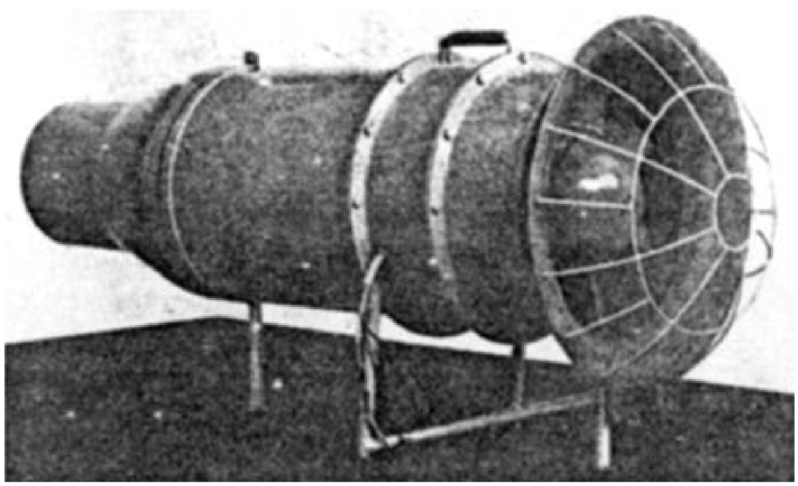


FIG. 2.38 Wind tunnel with axial flow fan.

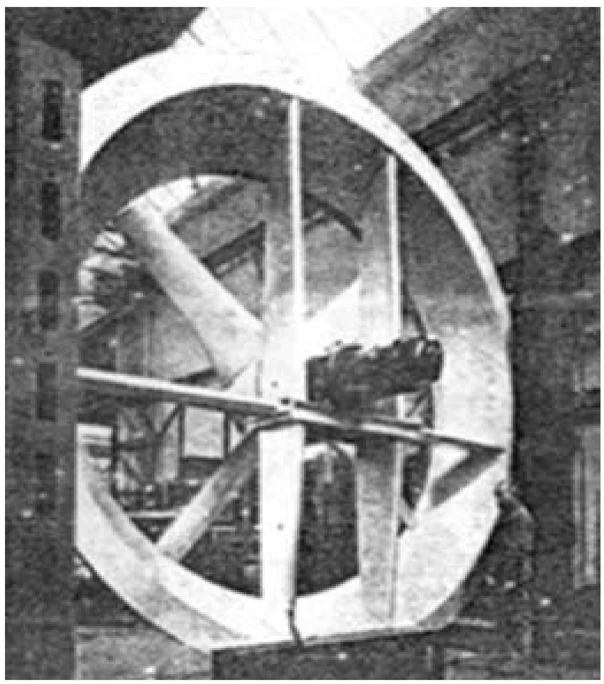


FIG.2.39 Axial flow fan made by Messrs. Sulzer for 60 3/sec and 9 mm WG.

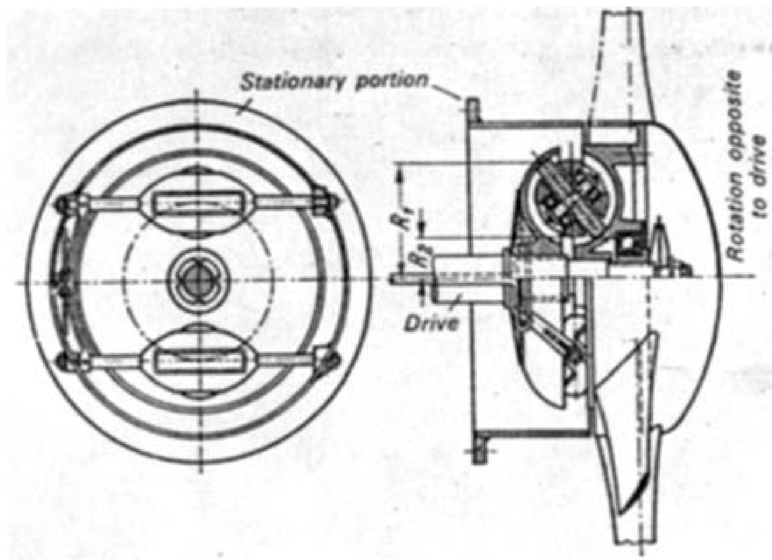


Fig 2.40 Friction wheel drive for speed control in axial flow fans designed by FKFS.

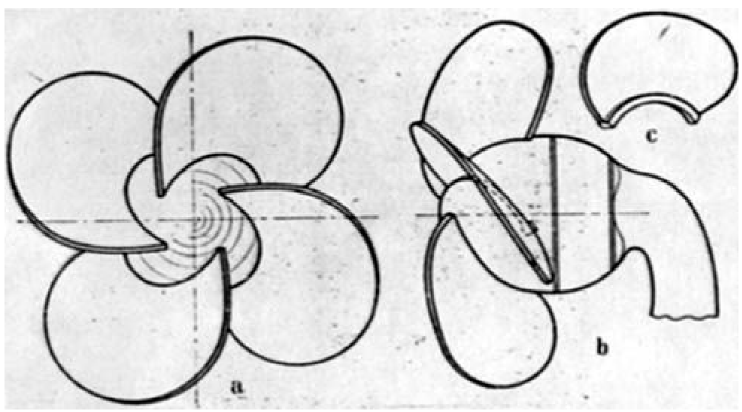


Fig. 2.41(a-c) Axial flow fans with rubber aerofoils and hub discharge.

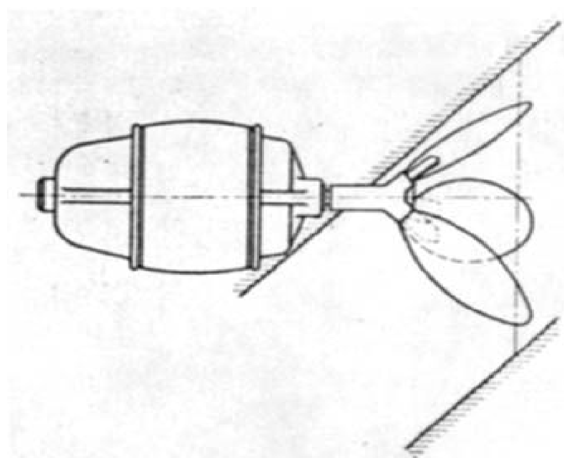


Fig.2.42 Axial flow fan designed by Frohlich.

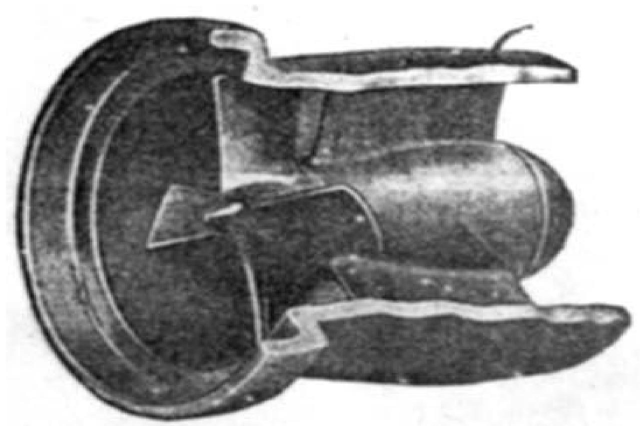


Fig. 2.43 Propeller fan made of stoneware.

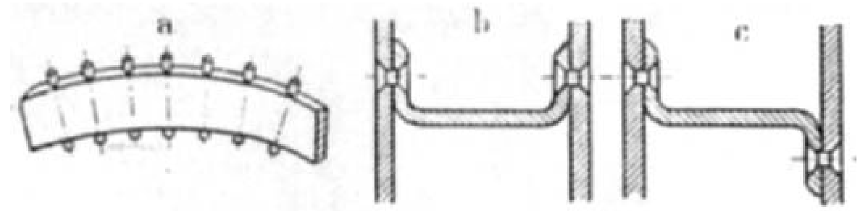


Fig 2.44 (a) Blade with fixed rivets. (b) and (c) typical riveting layout.

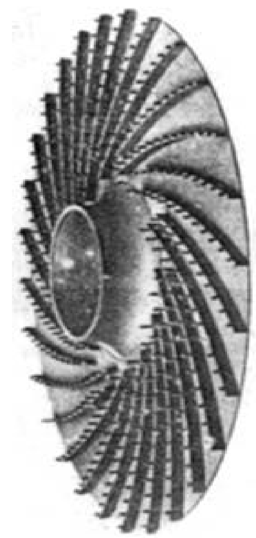


Fig.2.45 Impeller opened up to show milled rivets (BBC).

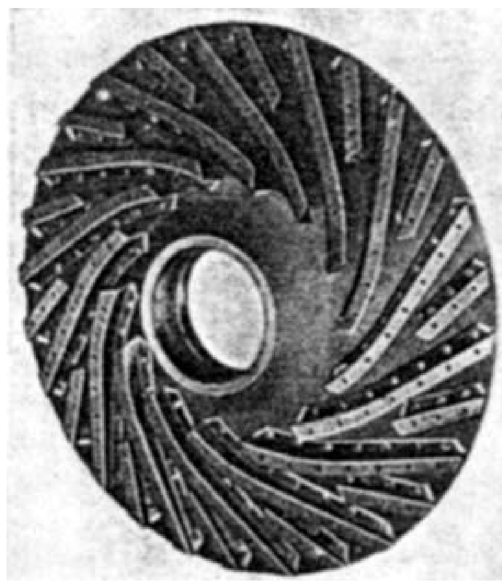


Fig.2.46 Impeller opened up to show riveted blades (Demag).

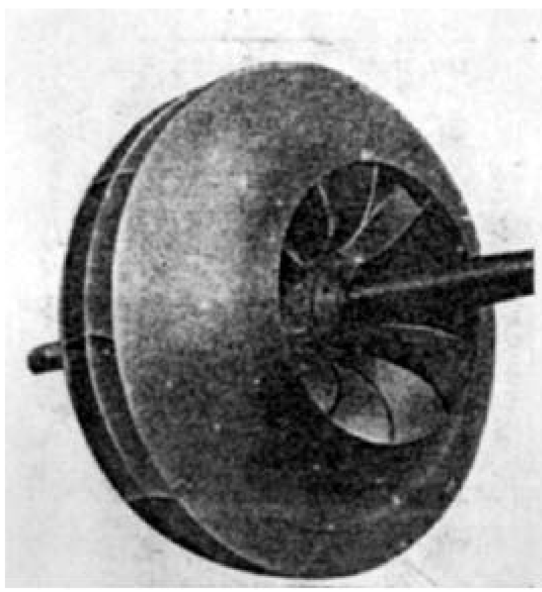


Fig. 2.47 Fully welded impeller designed by MAN.

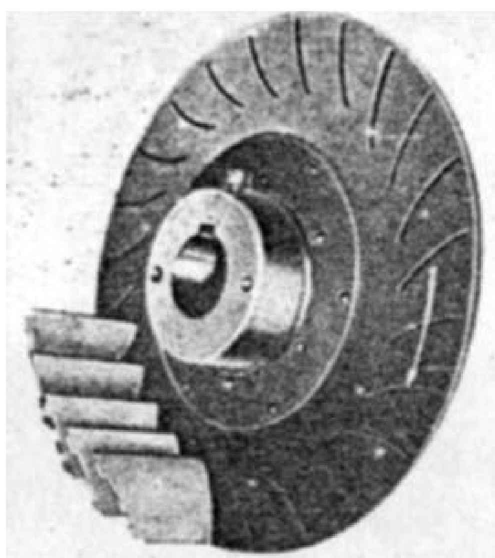


Fig 2.48 Blades inserted in shroud recesses.

Fig.2.49 Guide blades ring with adjustable guide blades according to BBC.

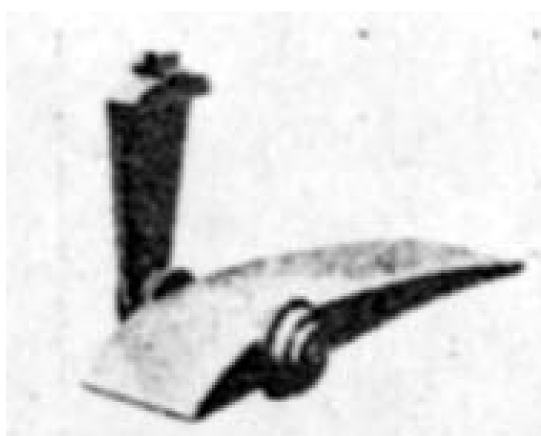


Fig 2.50 Adjustable guide blades according to BBC.

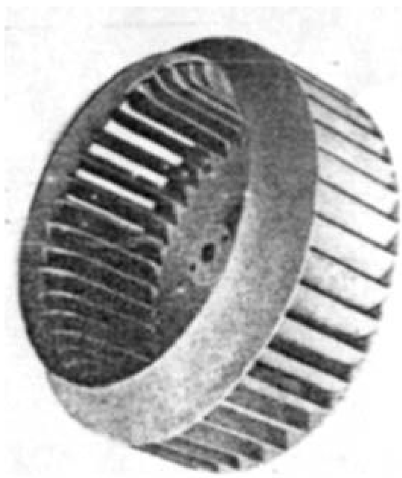


Fig. 2.51 Fig.2.52

Fig.2.51 Drum impeller with open ring.

Fig.2.52 Bracing by struts in drum impeller.

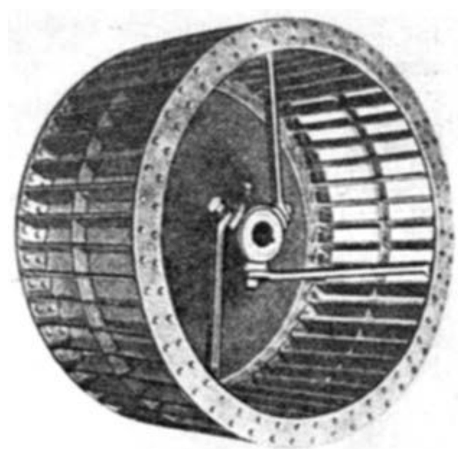


Fig. 2.53

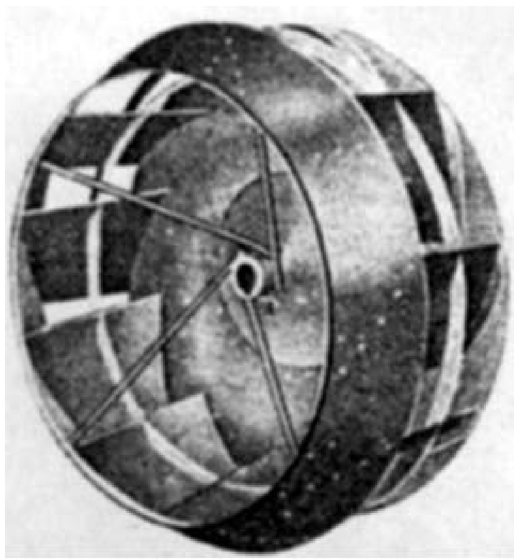


Fig. 2.54Bracing by means of a ring.

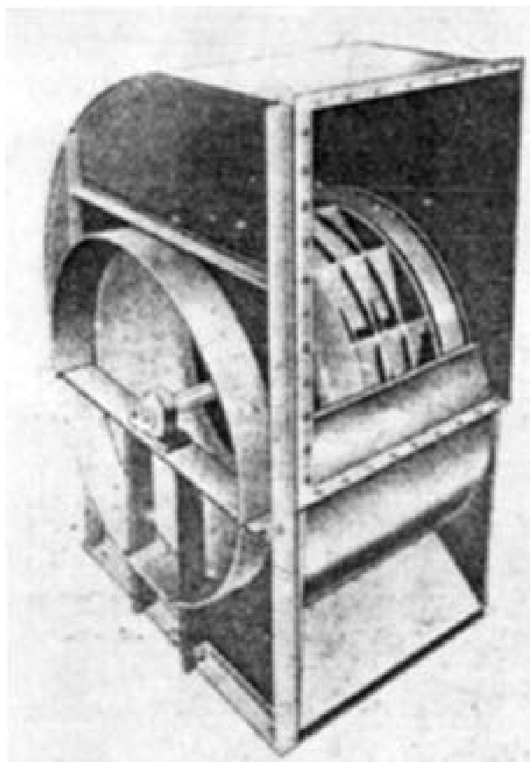


Fig. 2.55 Bracing by means of several rings.

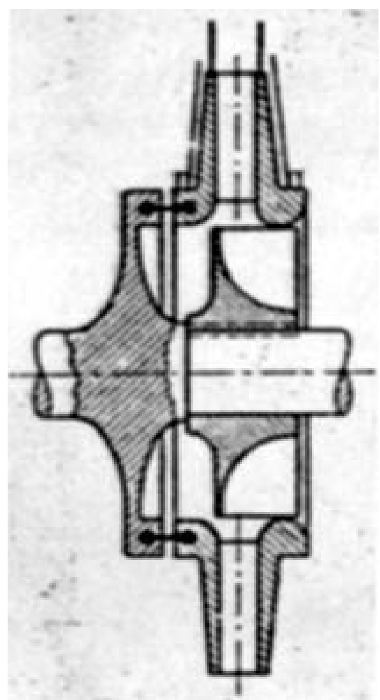


Fig 2.56 Spring mounted impeller according to BBC.

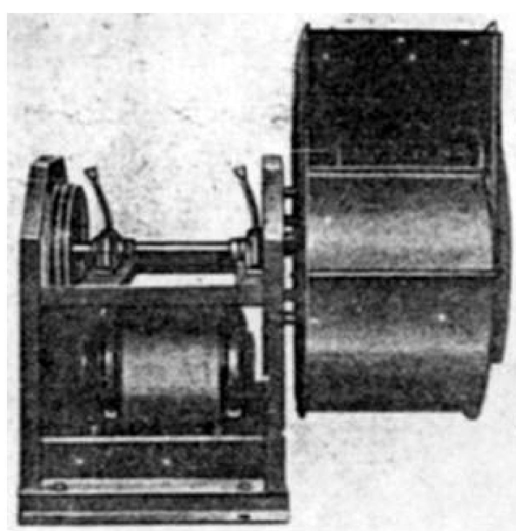


Fig 2.57Spiral housing mounted on a pedestal.

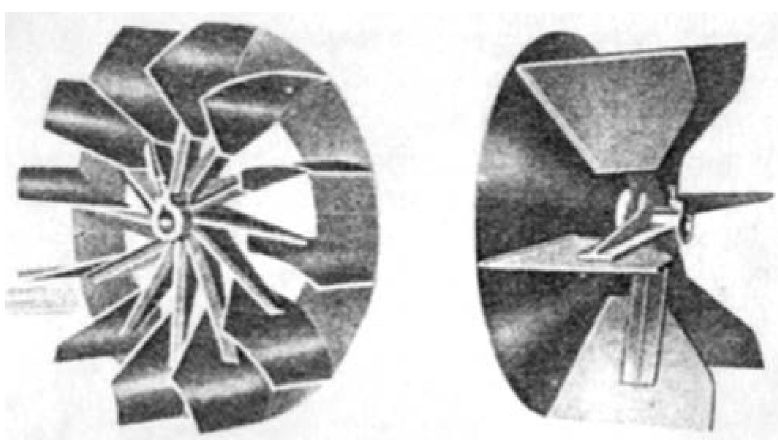


Fig. 2.58Impeller with each blade supported separately.

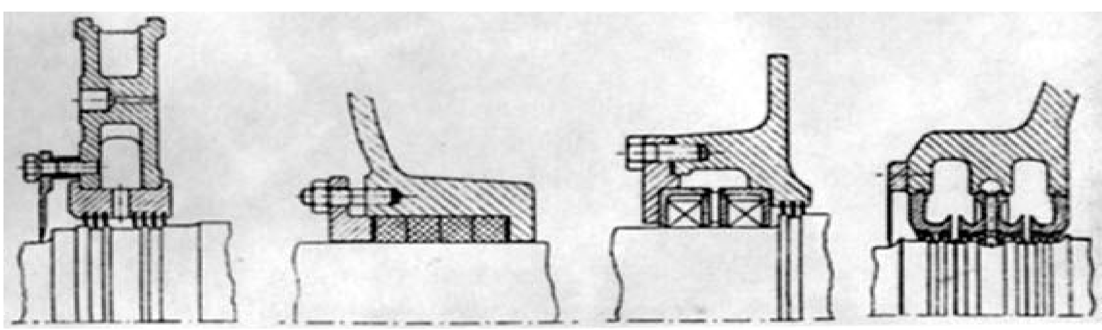


Fig 2.59Various shaft seals: labyrinth seal, soft packing, carbon packing.

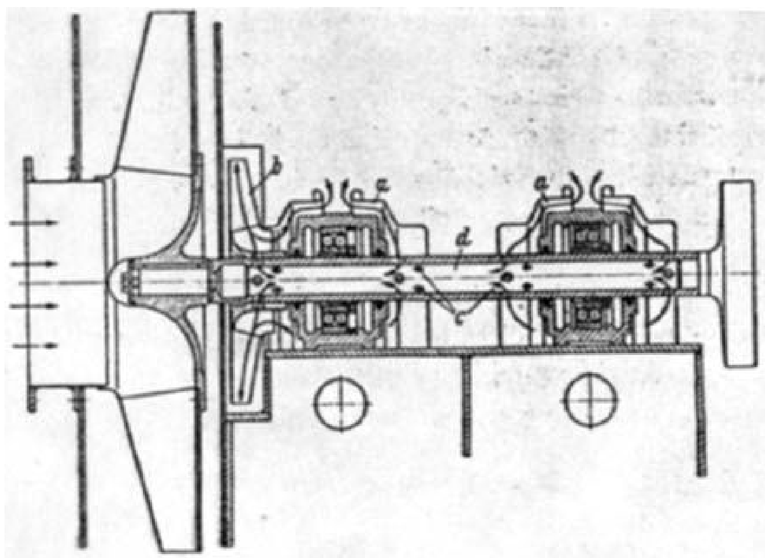


Fig.2.60Air-cooled bearings for fan handling hot gases (Mabag, Sulzbach) a, bearing covers; b, cooling impeller; c, openings in a hollow shaft for sucking air for cooling purposes; d, hollow shaft.

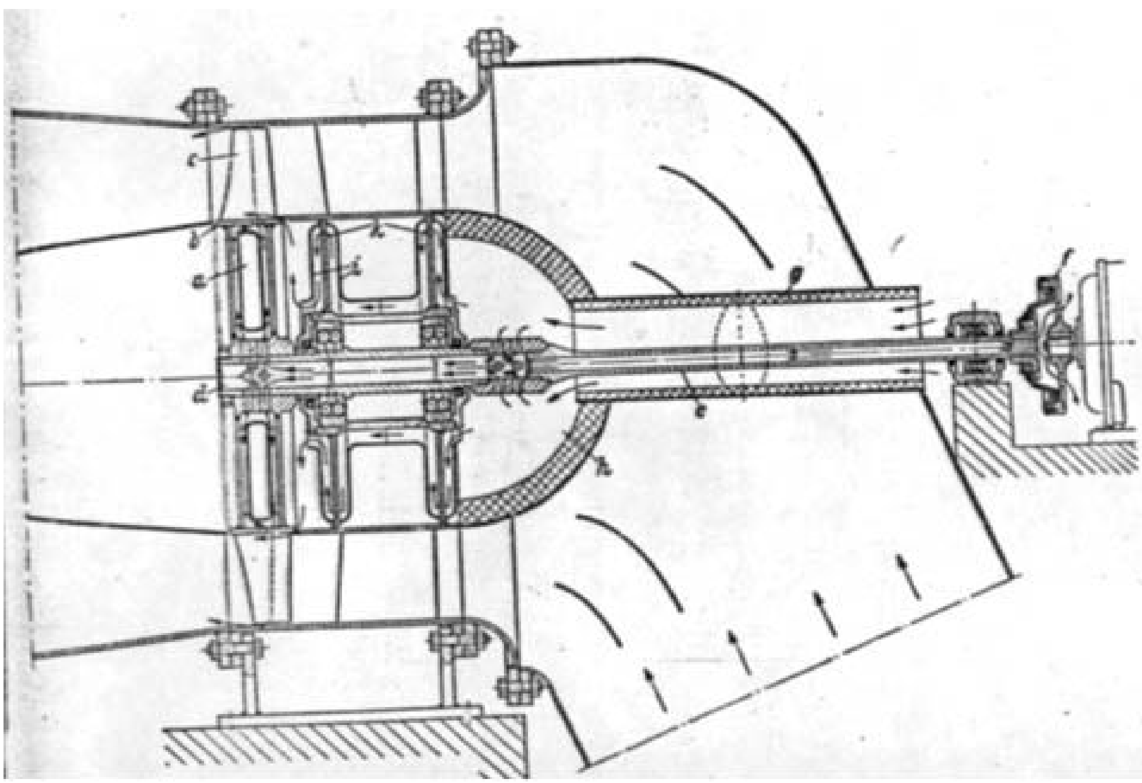


Fig. 2.61 Cooling bearings in the hub of an axial flow fan (Mabag, Sulzbach) a, displacing member, dividing cooling impeller in two parts; b, hub vylinder of the axial flow impeller; v, axial flow fan blades; d, hollow space in shaft through which the cooling air is sucked; e, hollow supply shaft; f, coupling; g, shaft cover; h, hub cover; I and K, passage walls.

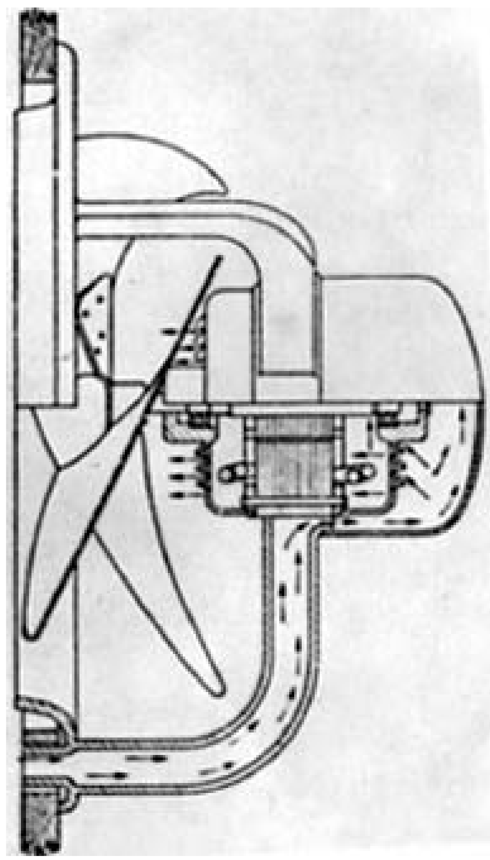


Fig 2.62 Cooling of motor from the pressure side of an axial flow fan.

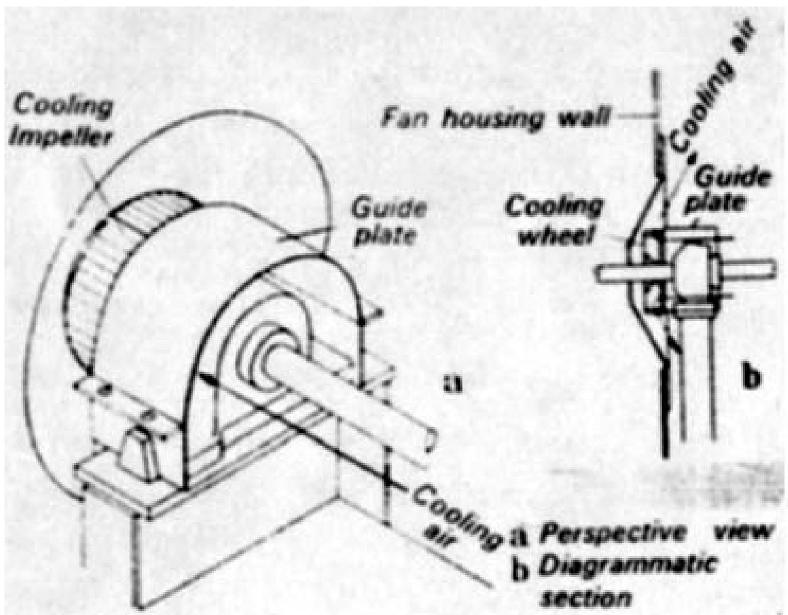


Fig 2.63 (a and b) Bearings cooled by means a special drum impeller.

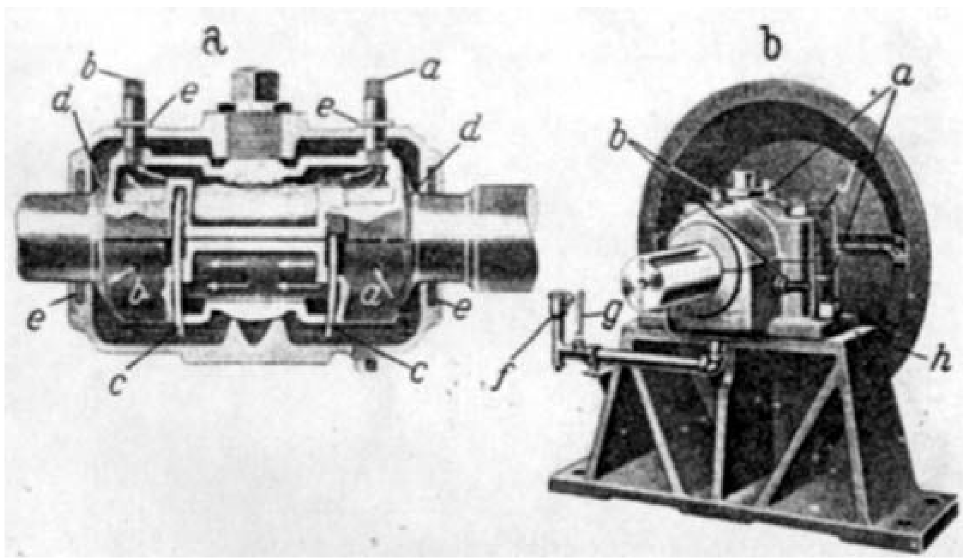
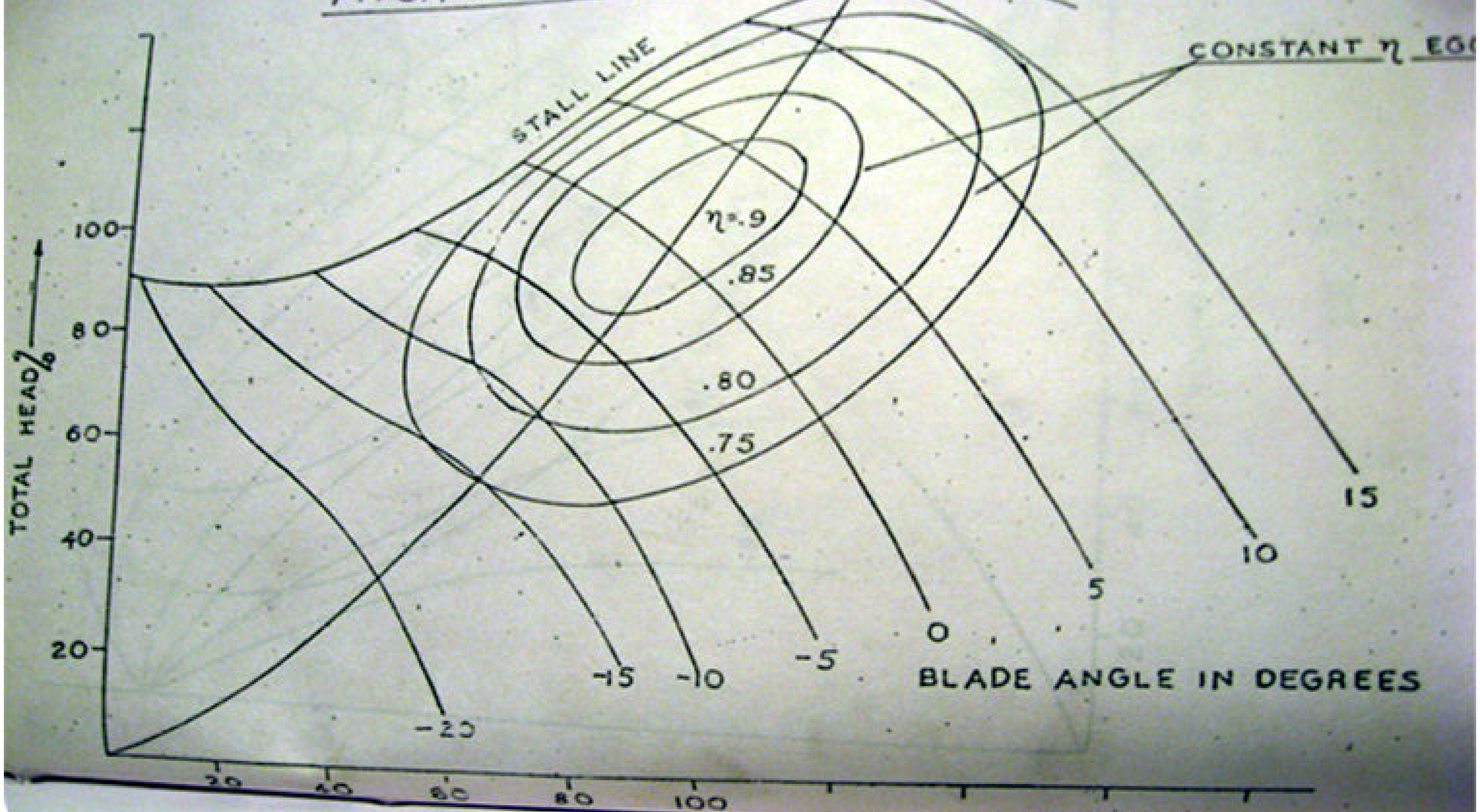


Fig 2.64Water cooled bearing a, b, water circulation connection; c, oil sump; d, spray ring; e, seal; f, oil supply branch.

PITCH CONTROL



CONTROL

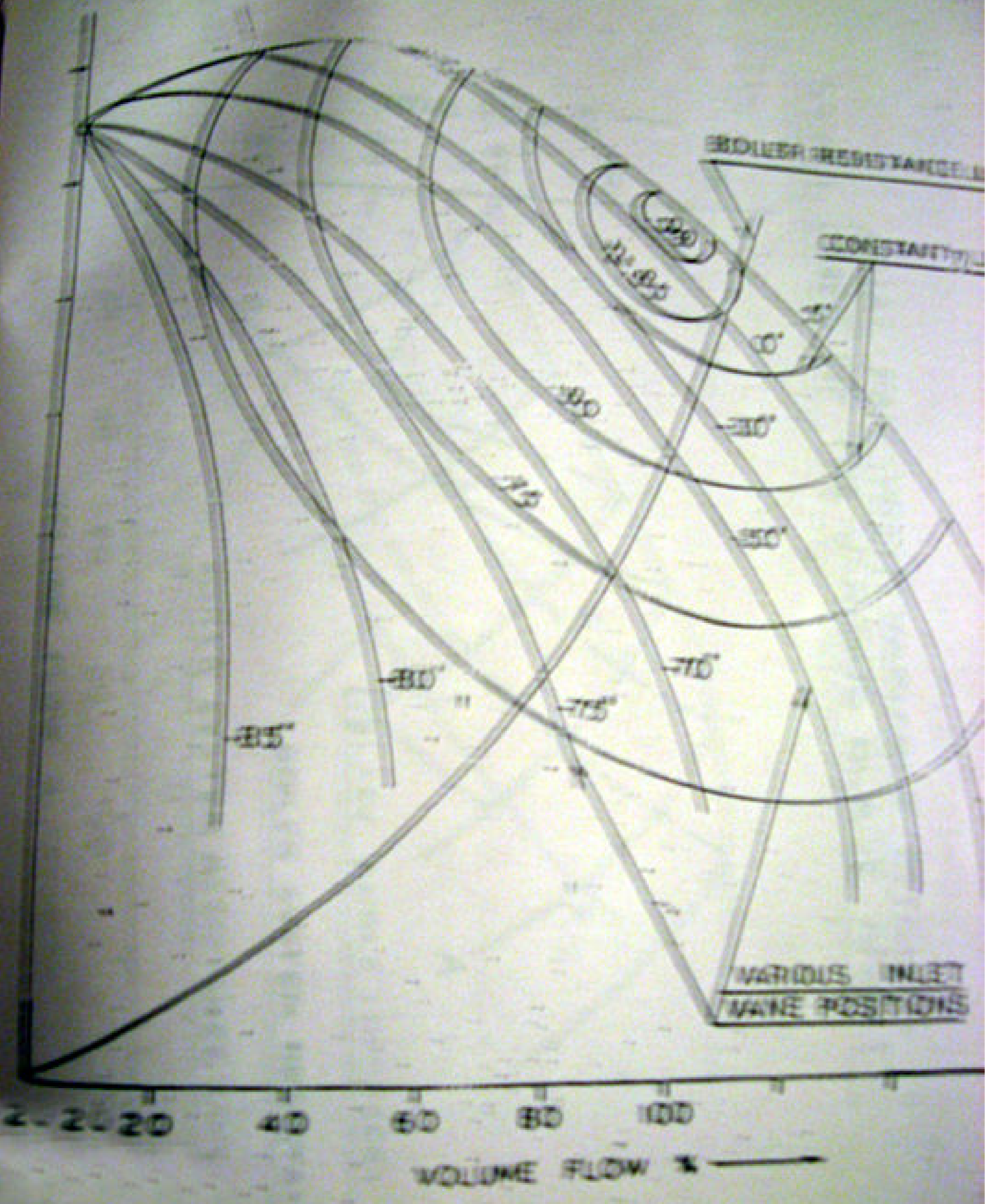
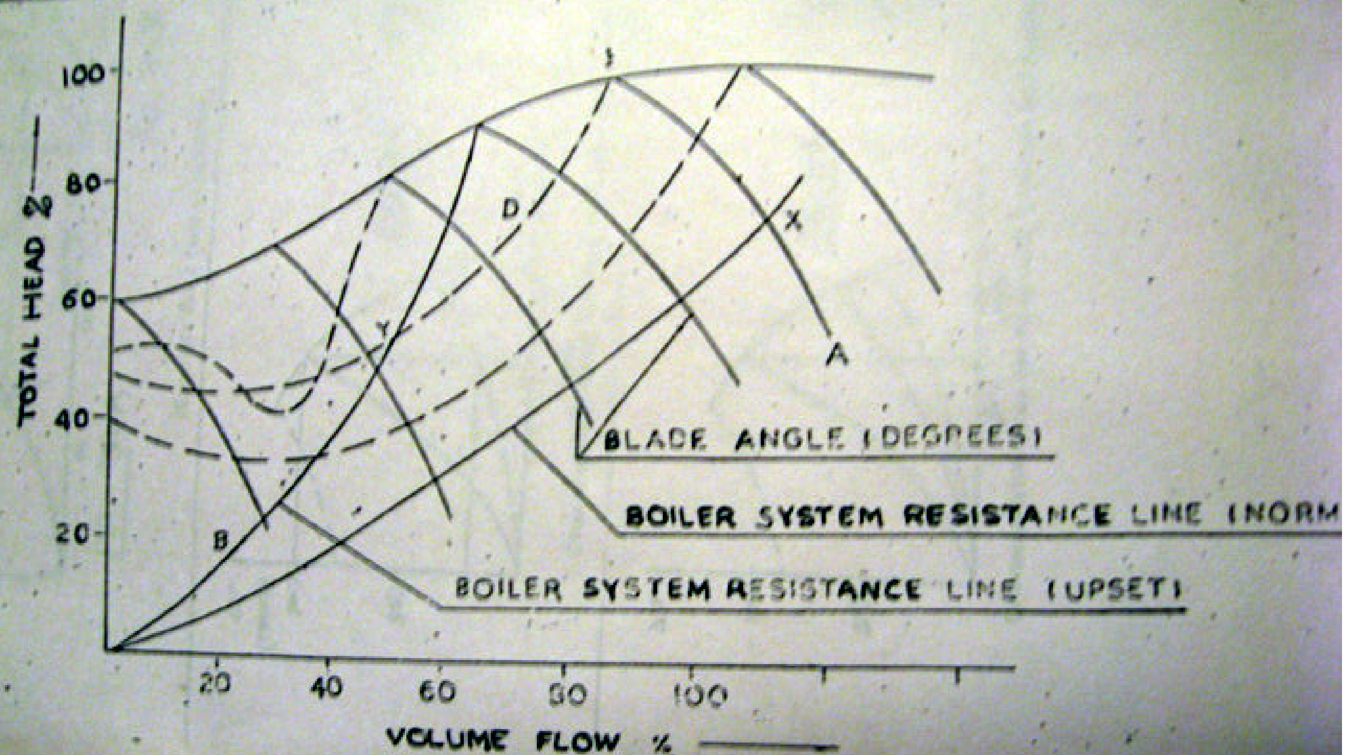


FIG.-32

AXIAL FAN WITH VARIABLE PITCH
STALL LINES OF THE VARIOUS BLADE ANGLES



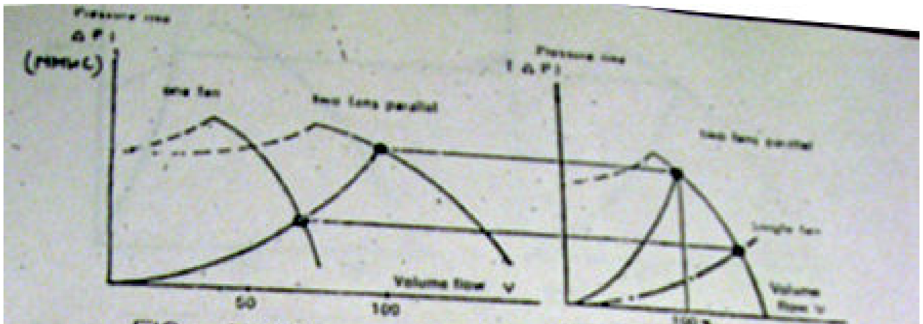


FIG. 3.4 A

FIG. 3.4 B

(Example for 2 fans in parallel)

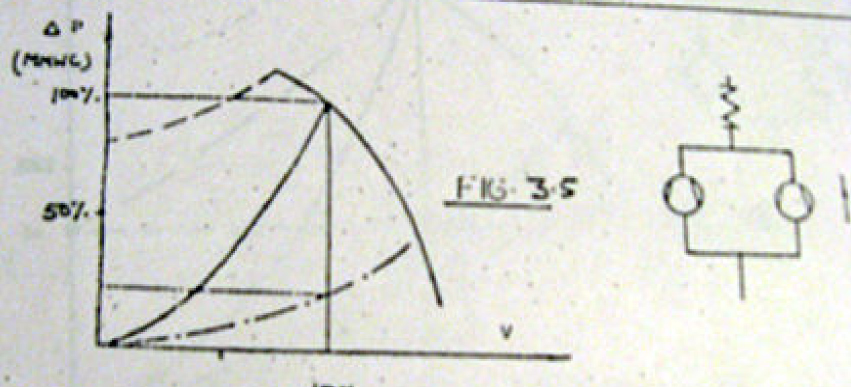


FIG. 3.5

Characteristic Lines for Fans with Common Resistance

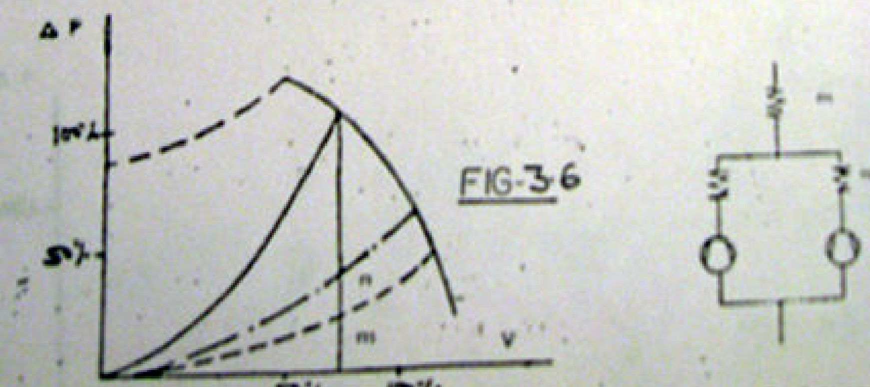


FIG. 3.6

Characteristic Lines for Fans with Common and Separate Resistances

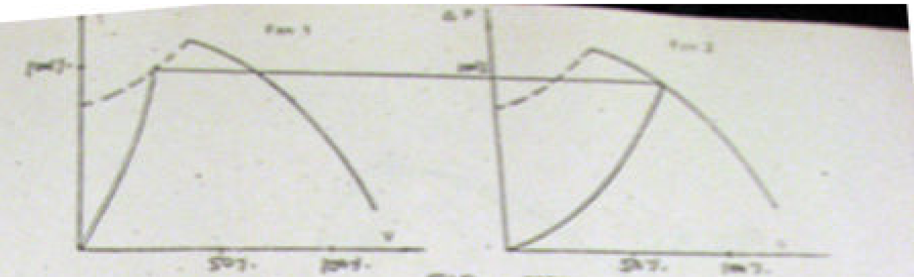


FIG- 37

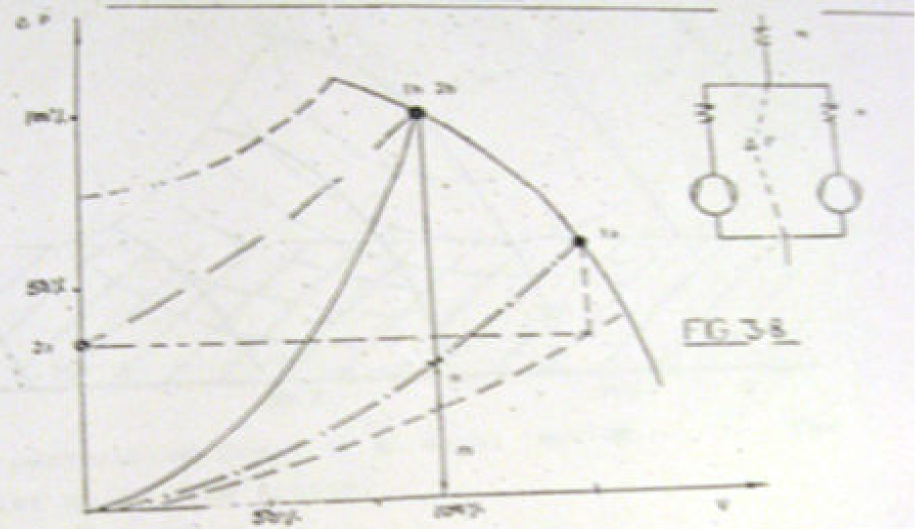


FIG 38

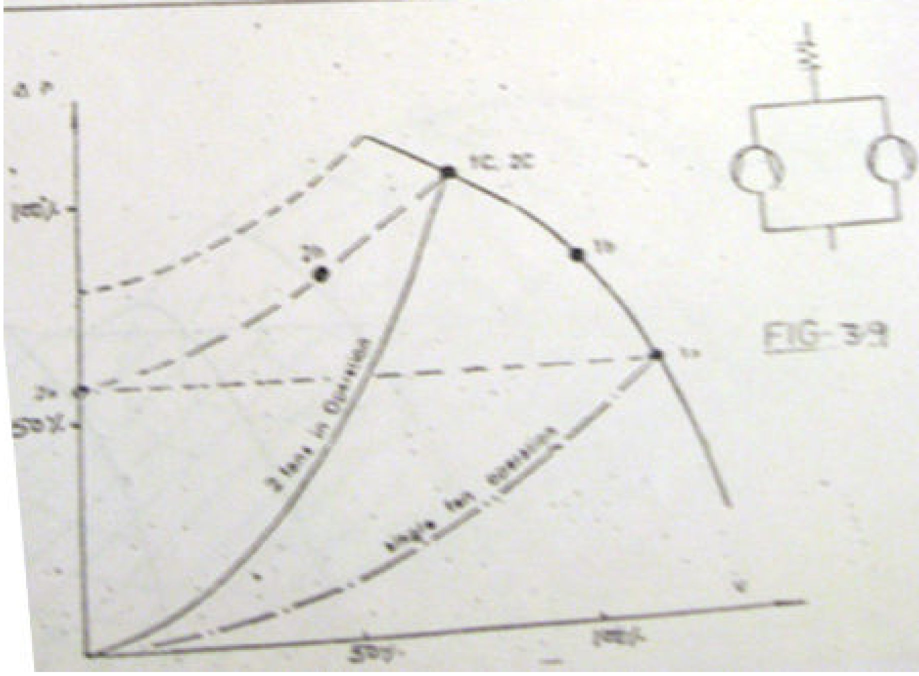
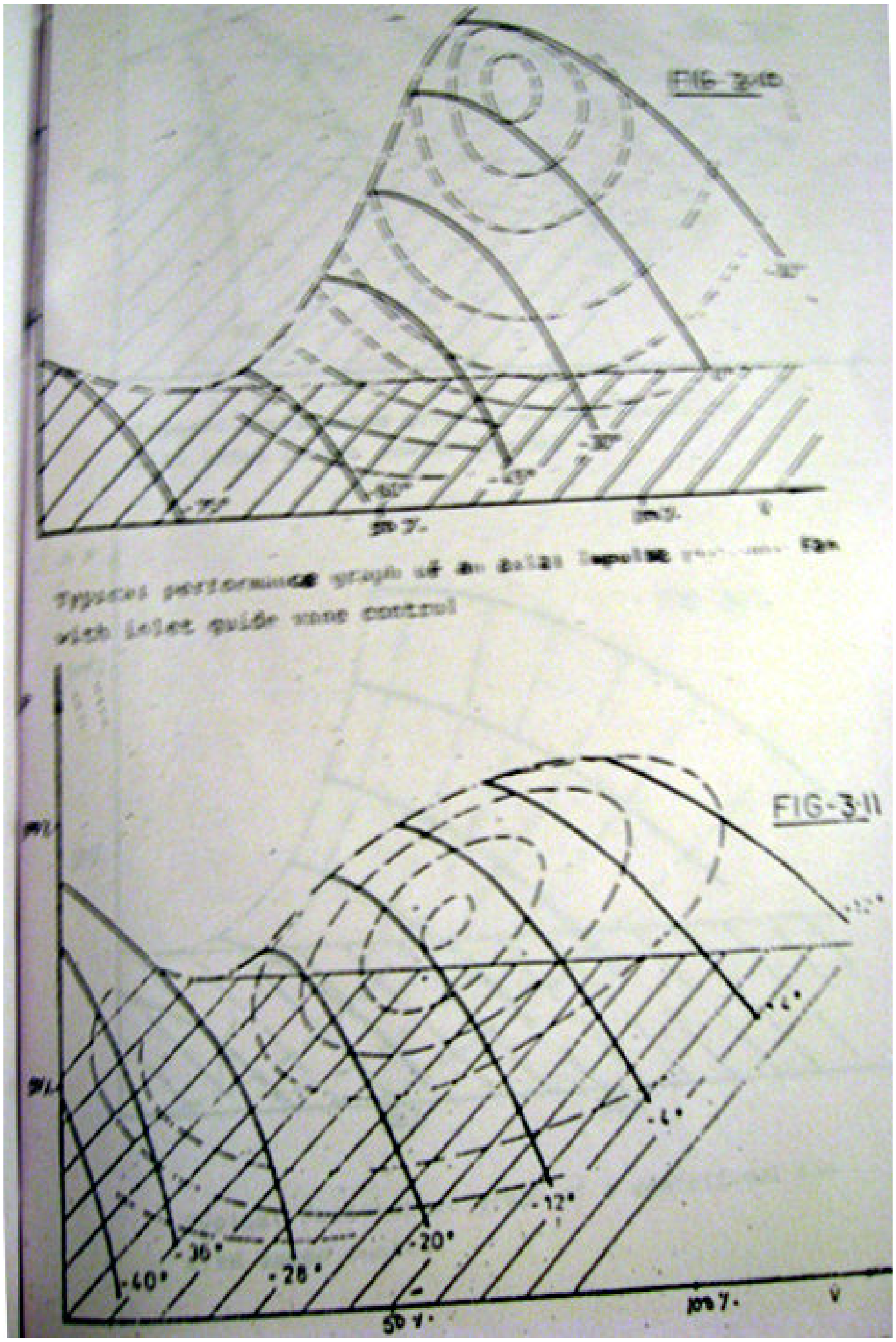
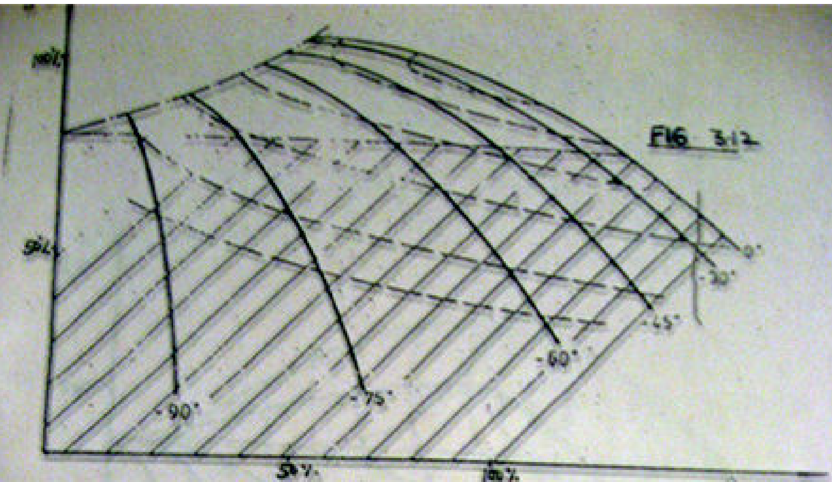


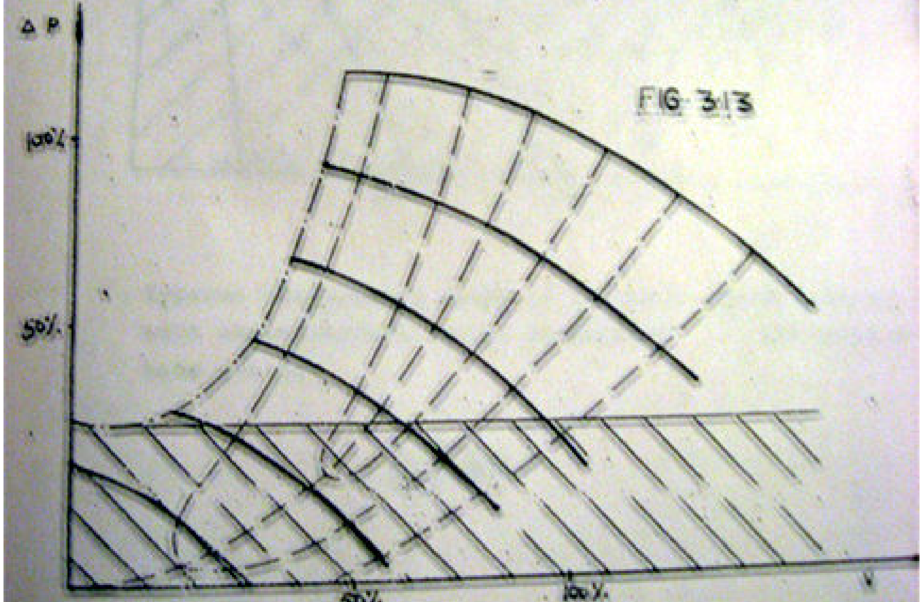
FIG- 39



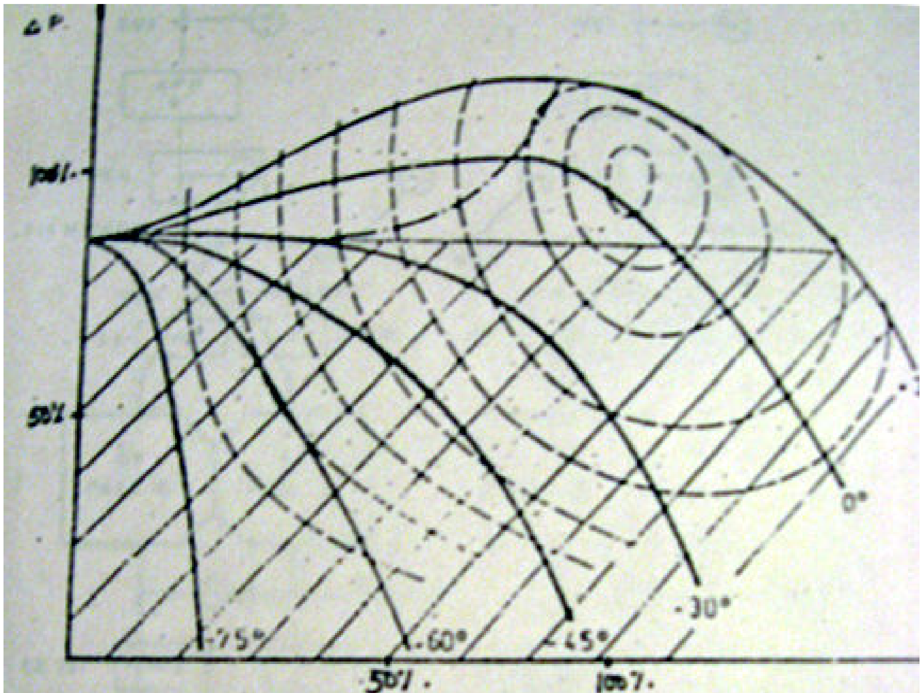
Typical performance graph of an axial fan, reaction type with variable pitch control.



Typical Performance graph of a centrifugal fan with inlet louver damper control



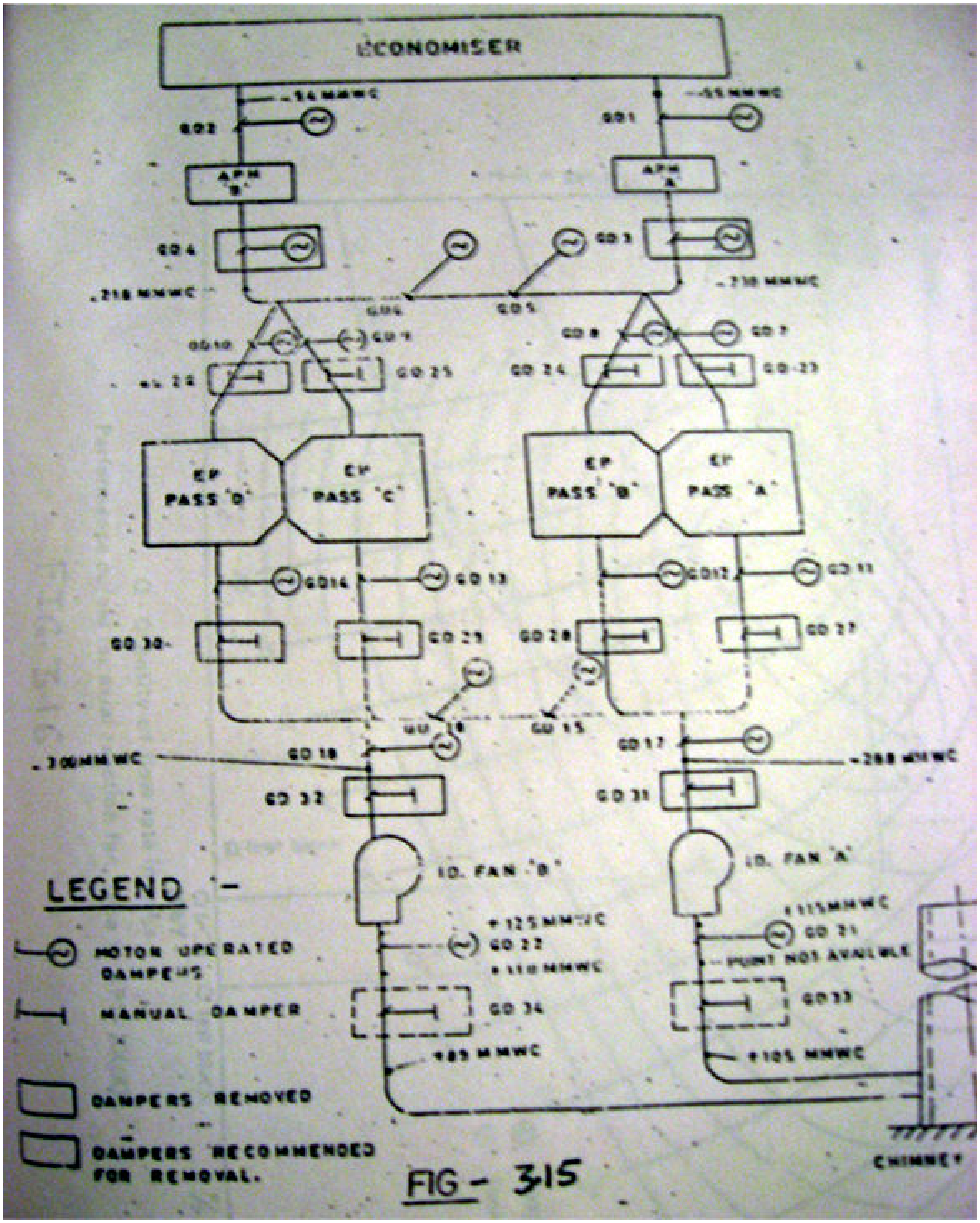
Typical Performance graph of a centrifugal fan with speed control

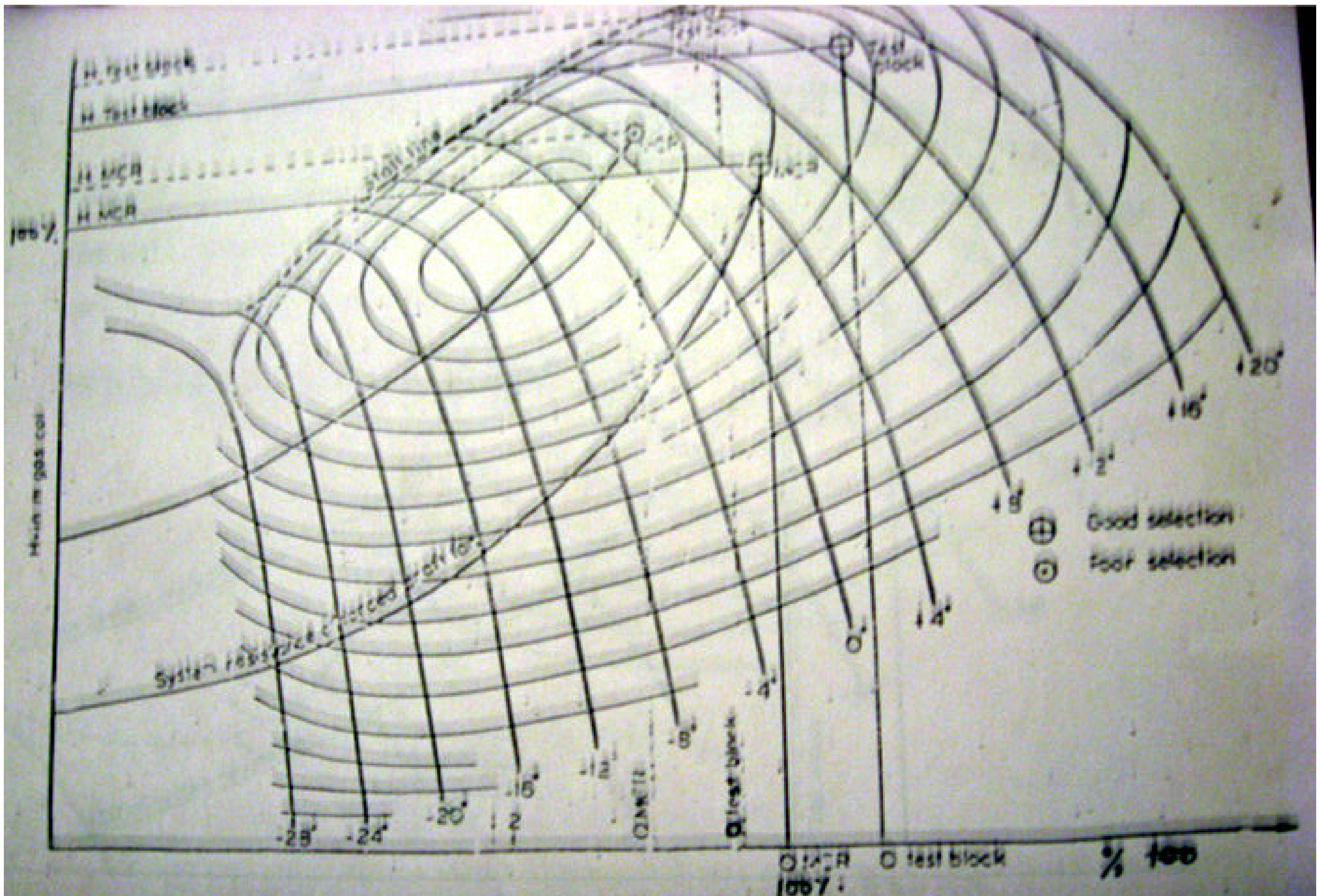


. Typical performance graph of an axial impulse p
with characteristic curve stabilizer and
vane control

FIG- 3.14

STATUS OF FULE GAS DAMPERS

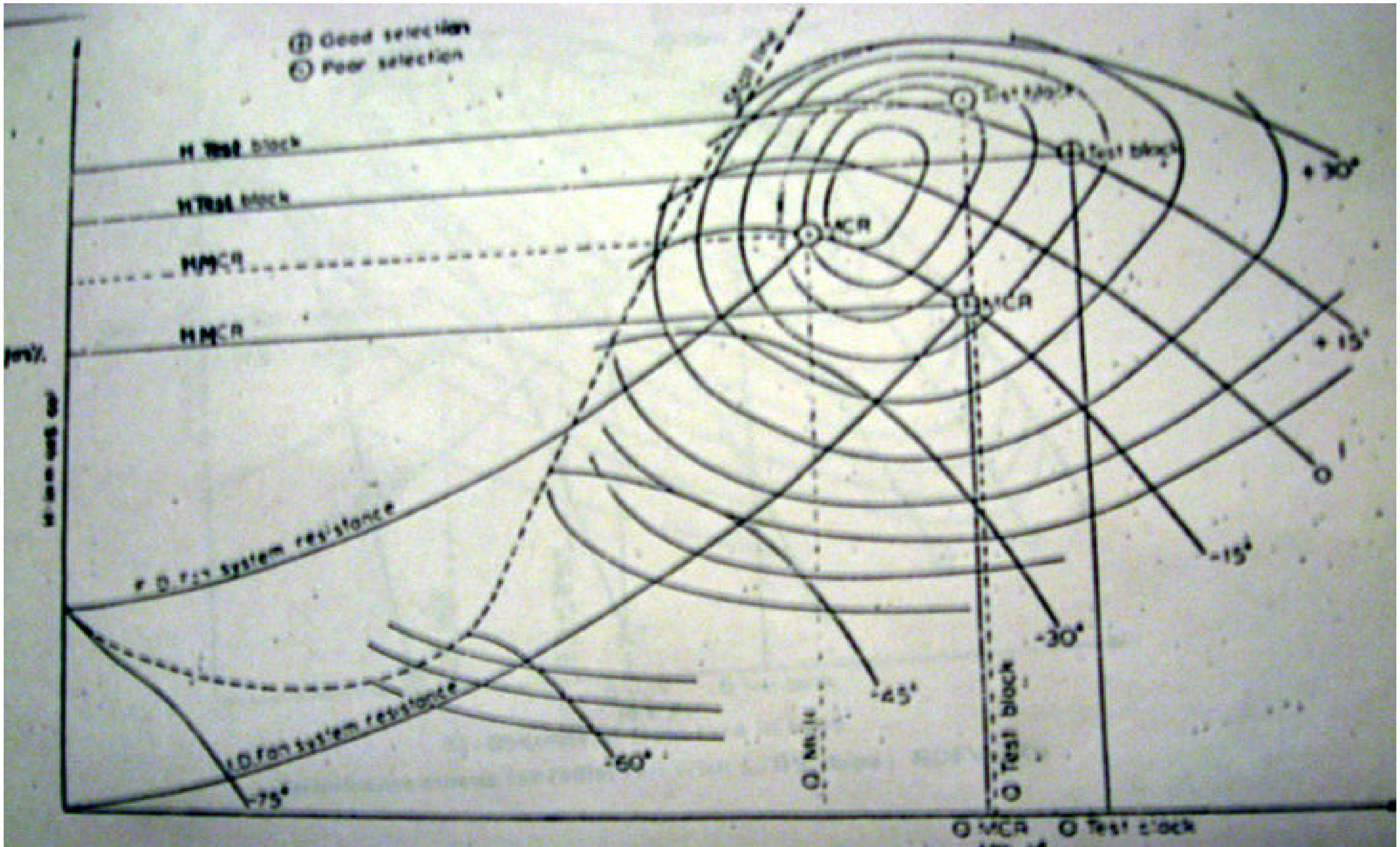




Q - Quantity of flow rate in m³/s

Performance curves for axial reaction fan, type : AP1 - XX/XX.

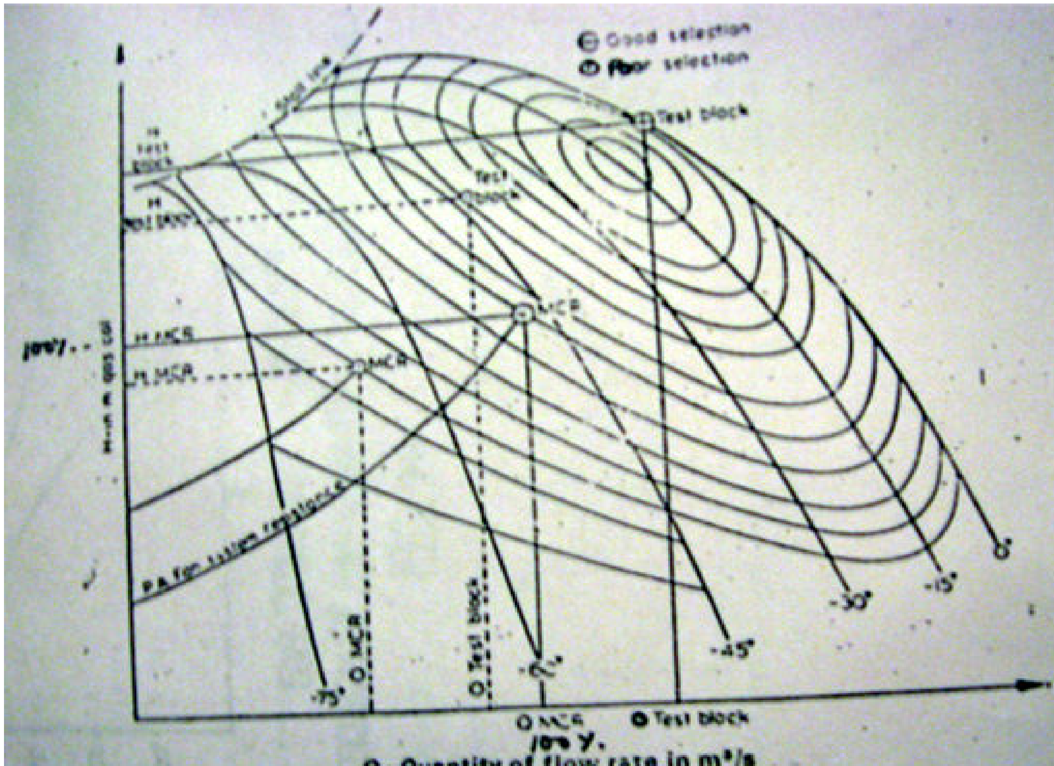
FIG-3.16



G - Quantity of flow rate in m³/s

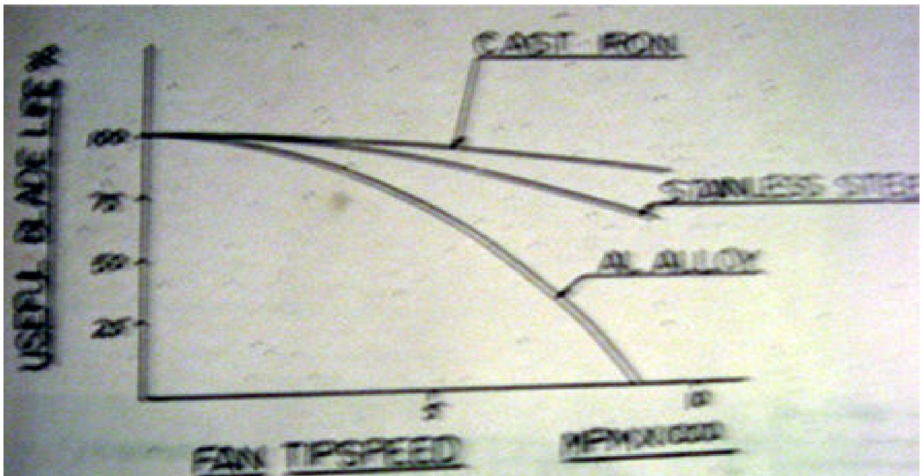
Performance curves for axial impulse fan, type : AN - XX e6.

FIG. 3.17



Performance curves for radial fan with L. GV. type: NDFV-XXb

FIG-3.18



EROSION OF FAN BLADES (ASH SIZE -2 MICR

FIG 3-9

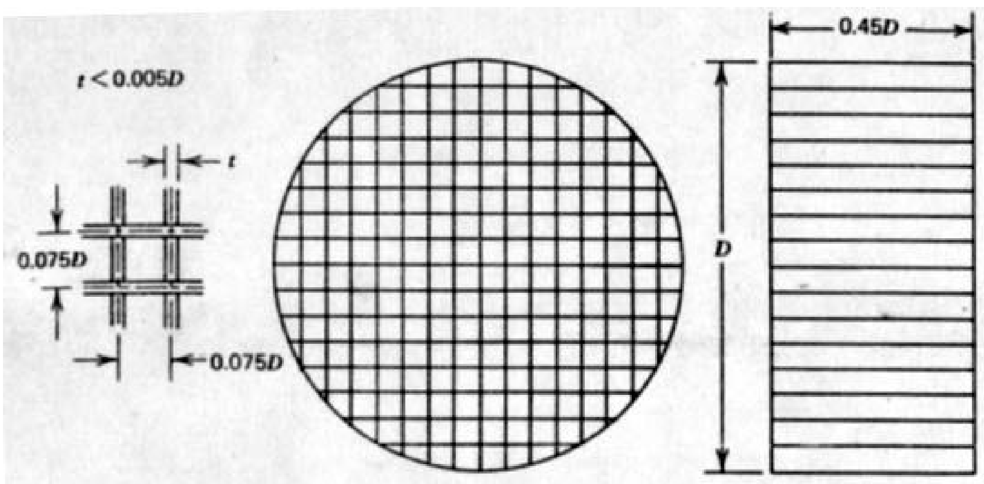


Fig 4.1 Airflow straightener, (21.3). (Reproduced by permission of AMCA and ASHRAE from AMCA standard 210-74, published by the Air Movement and Control Assoc. Inc. Arlington Heights, I11., USA.)

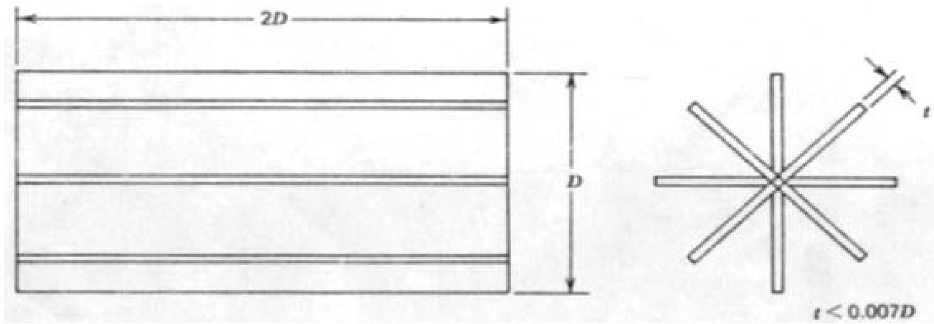


Fig 4.2 Airflow straightener, (21.4)

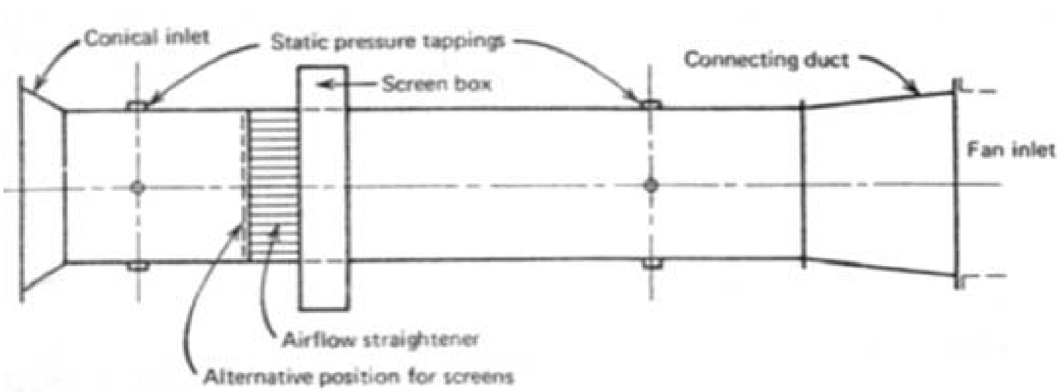


Fig 4.3 Inlet ducting arrangement.

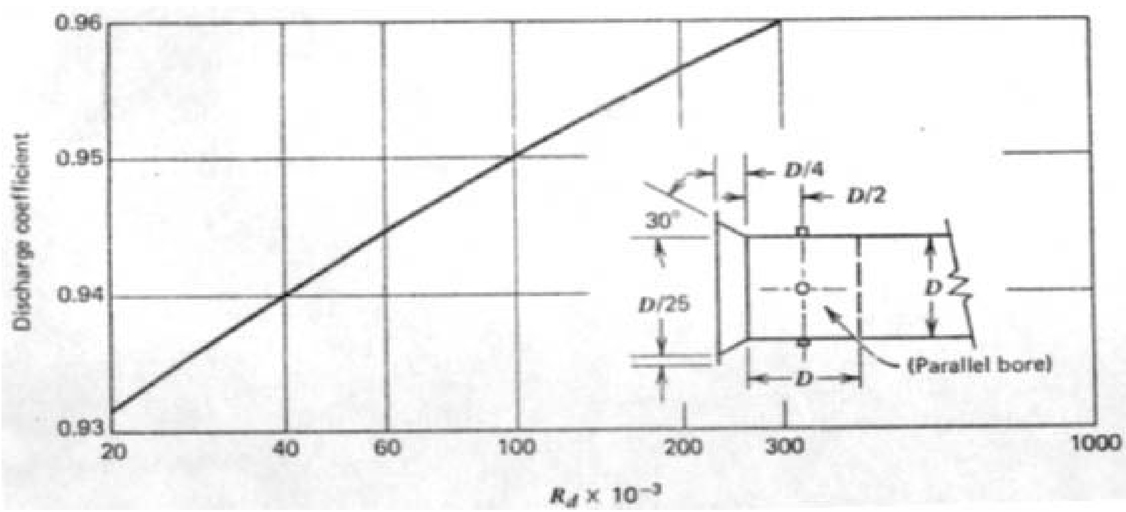


Fig. 4.4 Conical inlet for volume flow measurement, (21.4). (Reproduced by permission from BS 848: Part 1: 1980, published by the British Standards Institution 2 Park Street, London W1A 2BS, from whom complete copies may be obtained.)

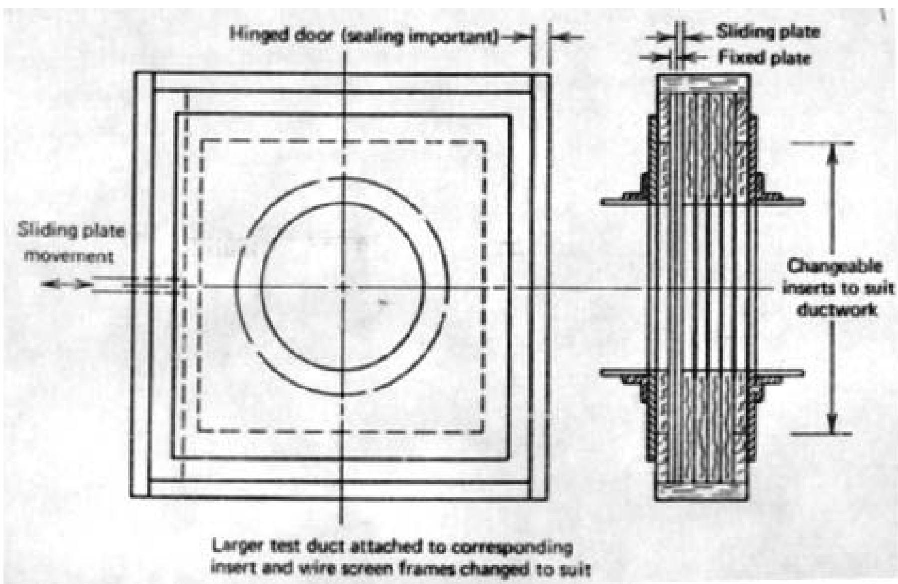


Fig. 4.5 Schematic illustration of combination box for wire screen and sliding perforated plates.

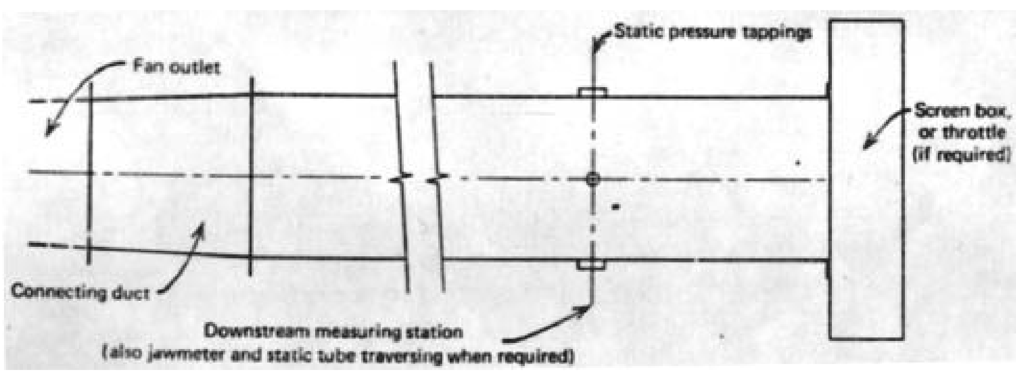


Fig. 4.6 Outlet test ducting arrangement.

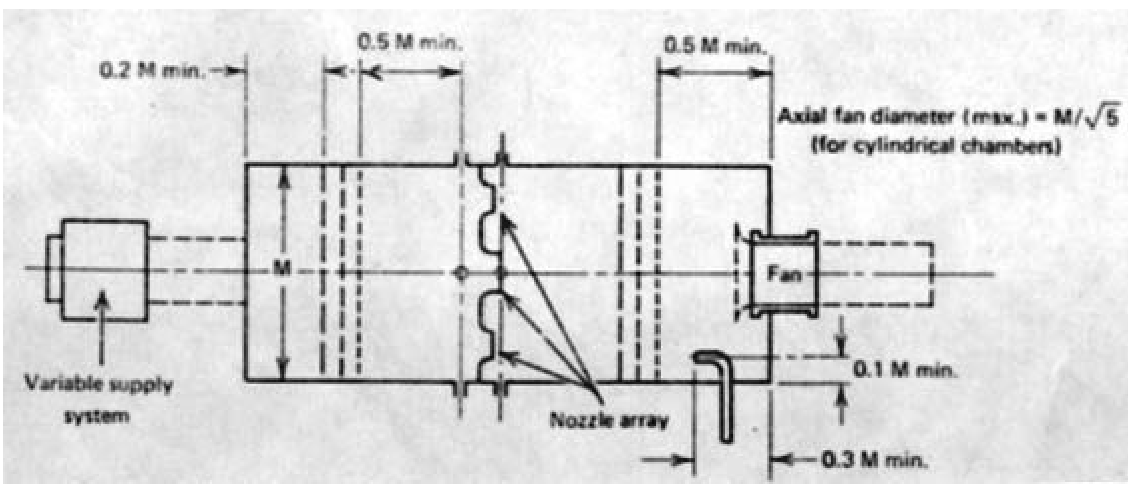


Fig. 4.7 Inlet chamber test equipment, (Reproduced by permission of AMCA and ASHRAE from AMCA standard 210-74, published by the Air Movement and Control Assoc. Inc. Arlington Heights, III. USA.)

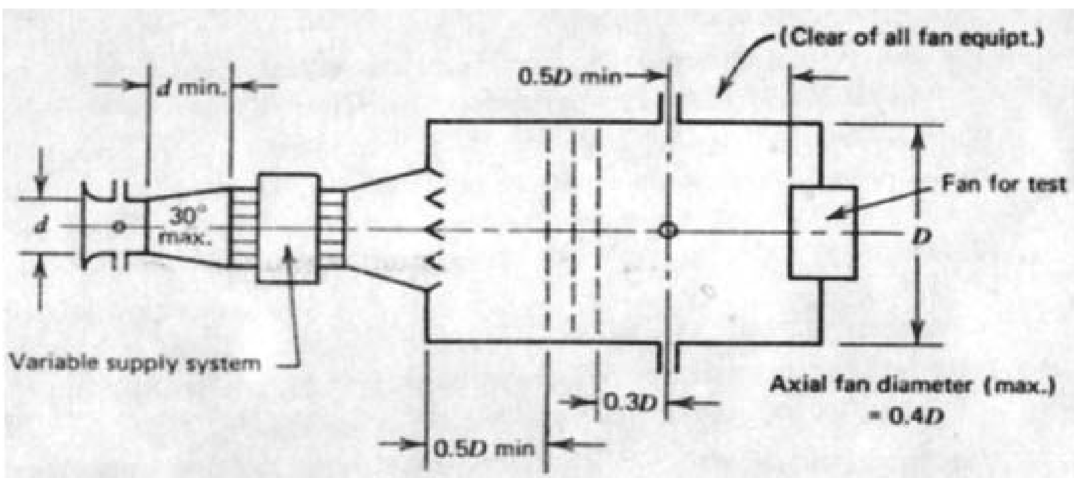


Fig. 4.8 Inlet chamber test equipment, (21.4). Reproduced by permission from BS 848: Part 1: 1980, published by the British Standards Institution 2 Park Street London, W1A 2BS, from whom complete copies may be obtained.

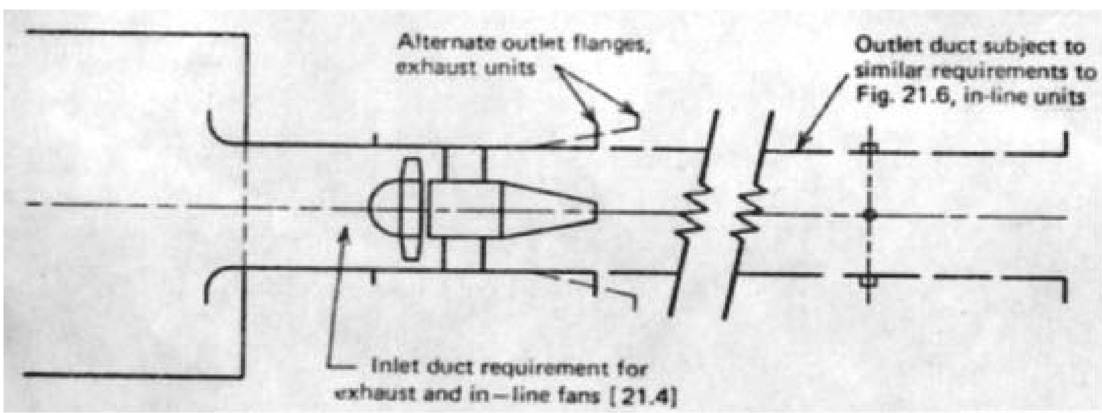


Fig. 4.9 Axial fan arrangement inlet chamber test.

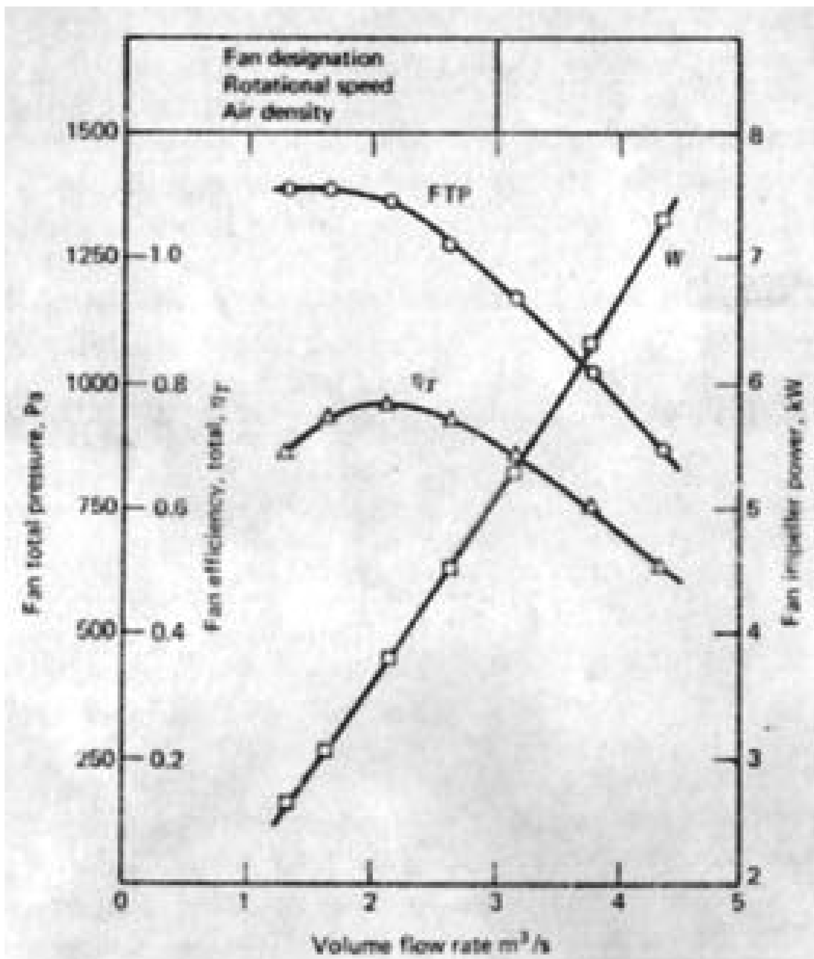


Fig 4.10 Typical fan characteristic curves.

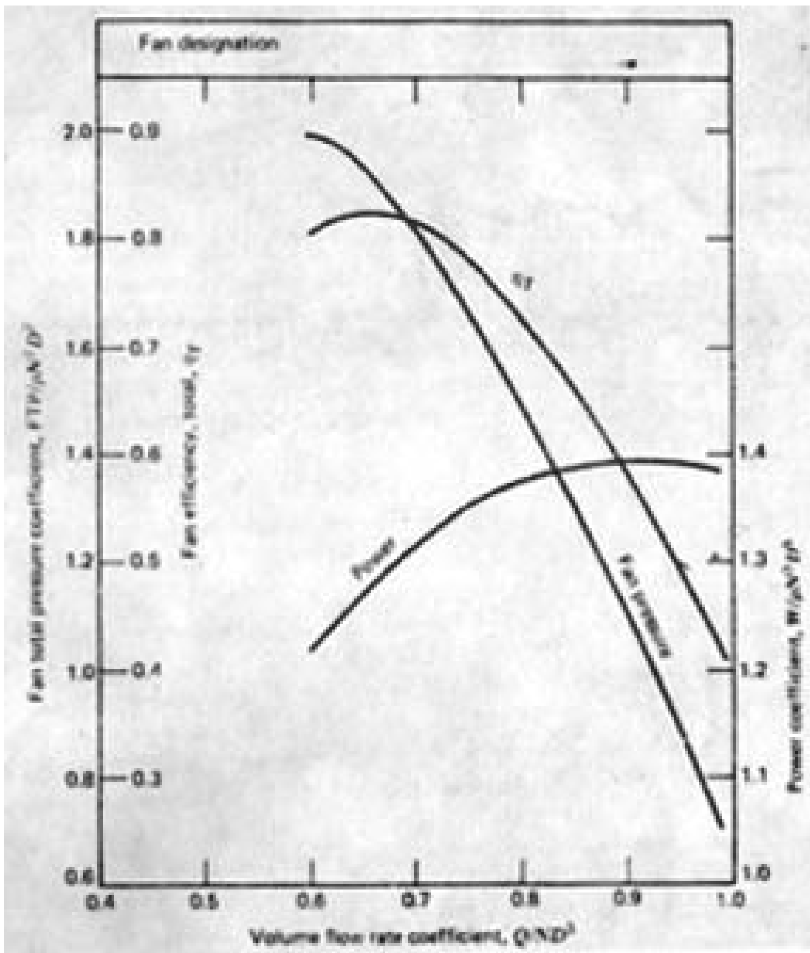


Fig. 4.11 A nondimensional presentation of test data.

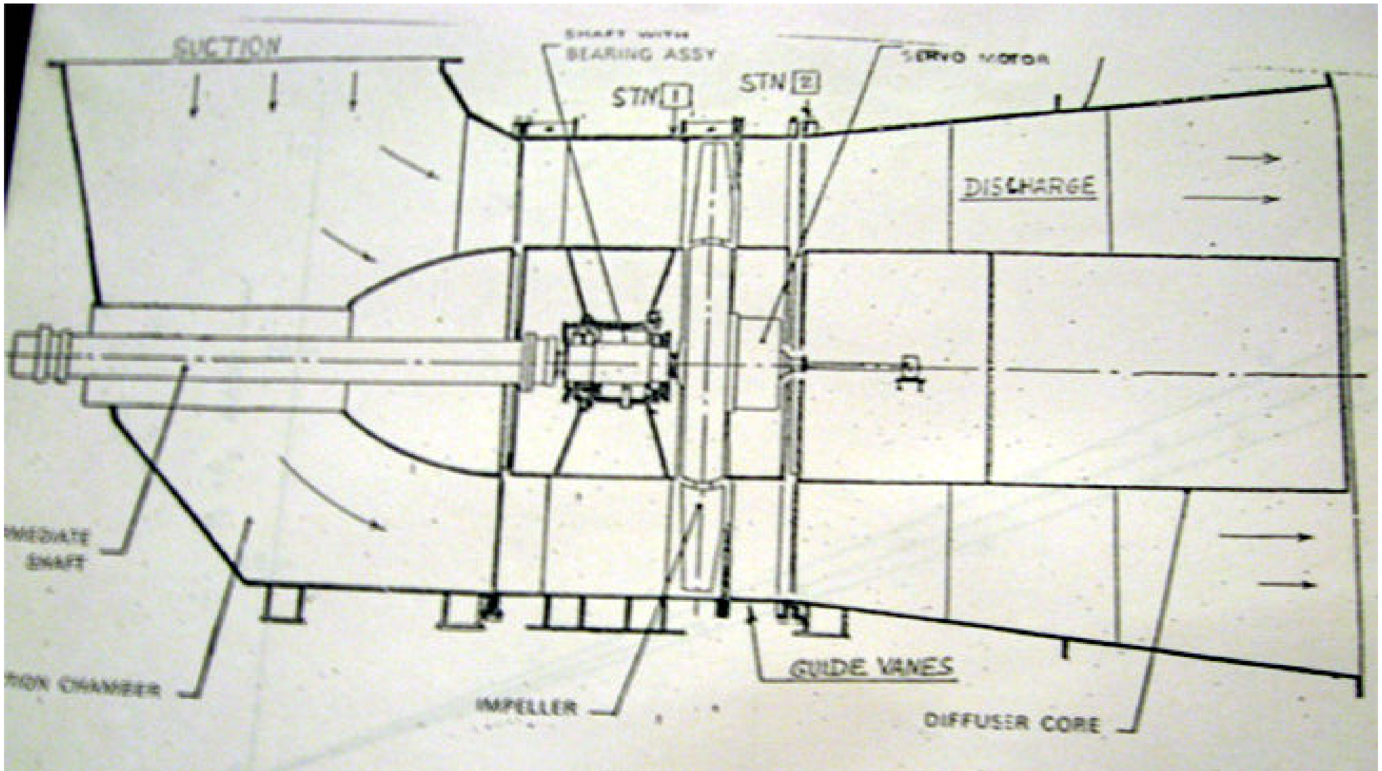


FIG-4.12

210 MW

AXIAL REACTION
FAN — ARRANGEMENT

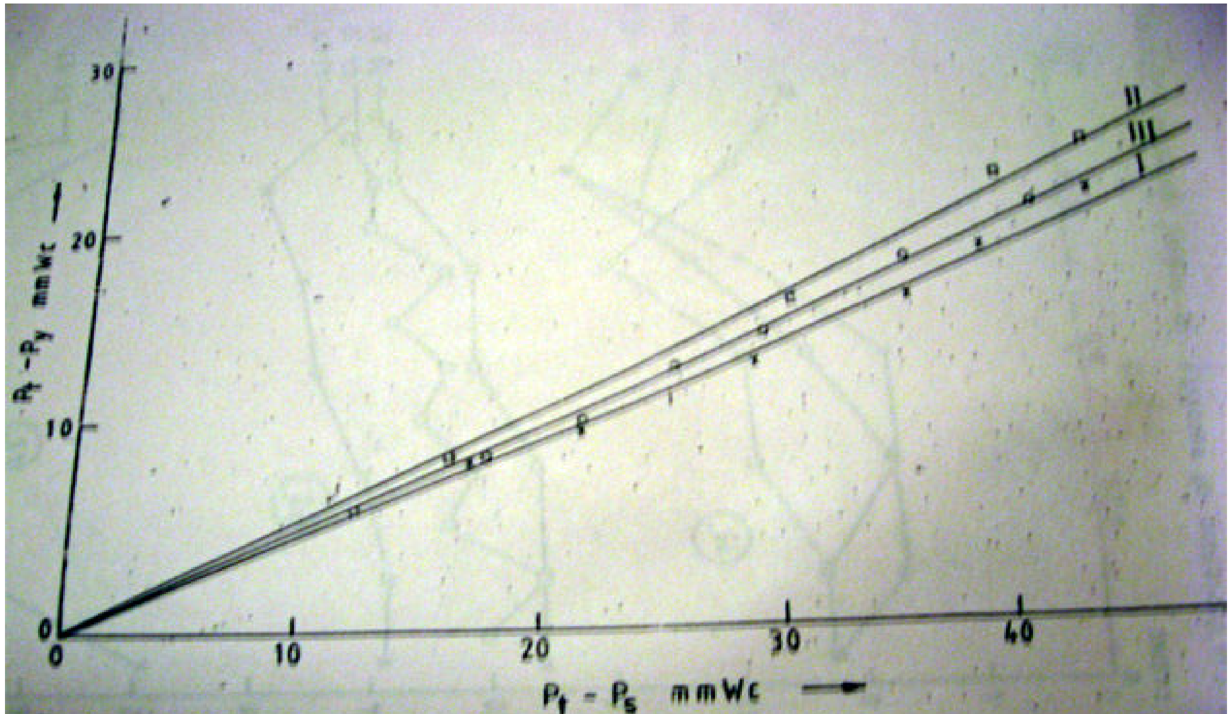


Fig.4.13 Calibration curve for YAW probe

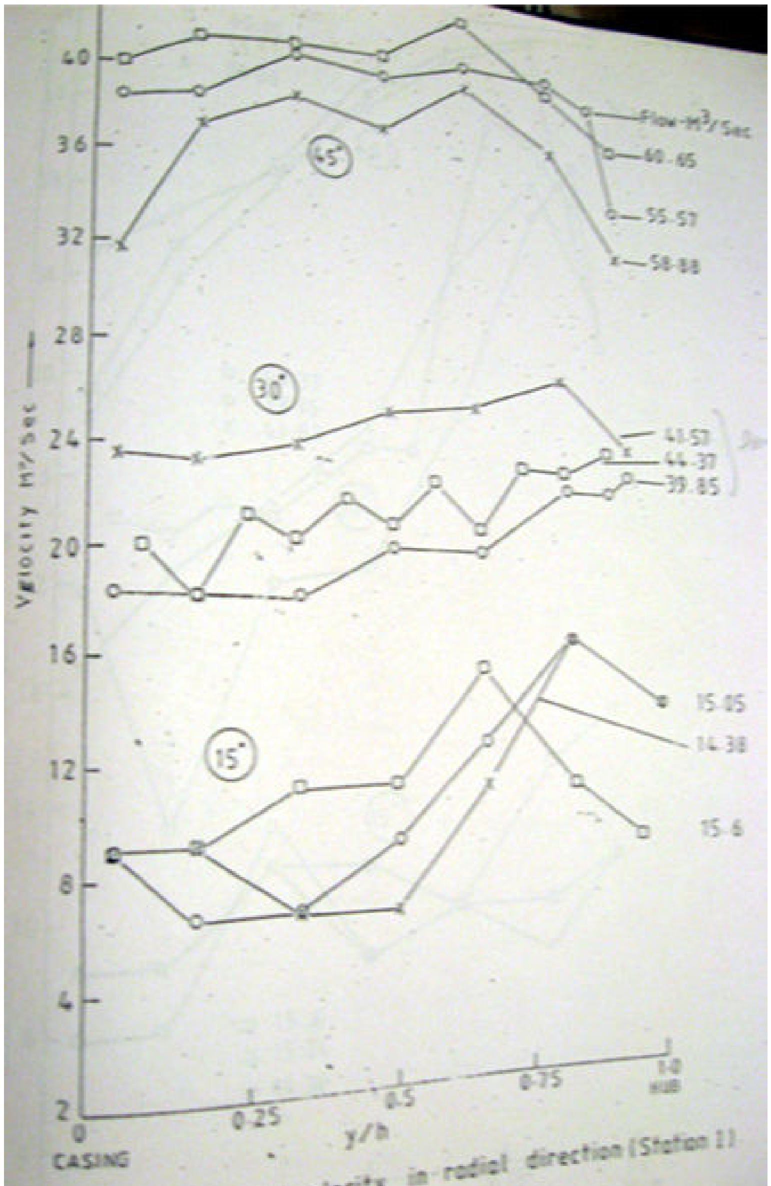


Fig. 4.16 Variation of velocity in radial direction (Station 1)

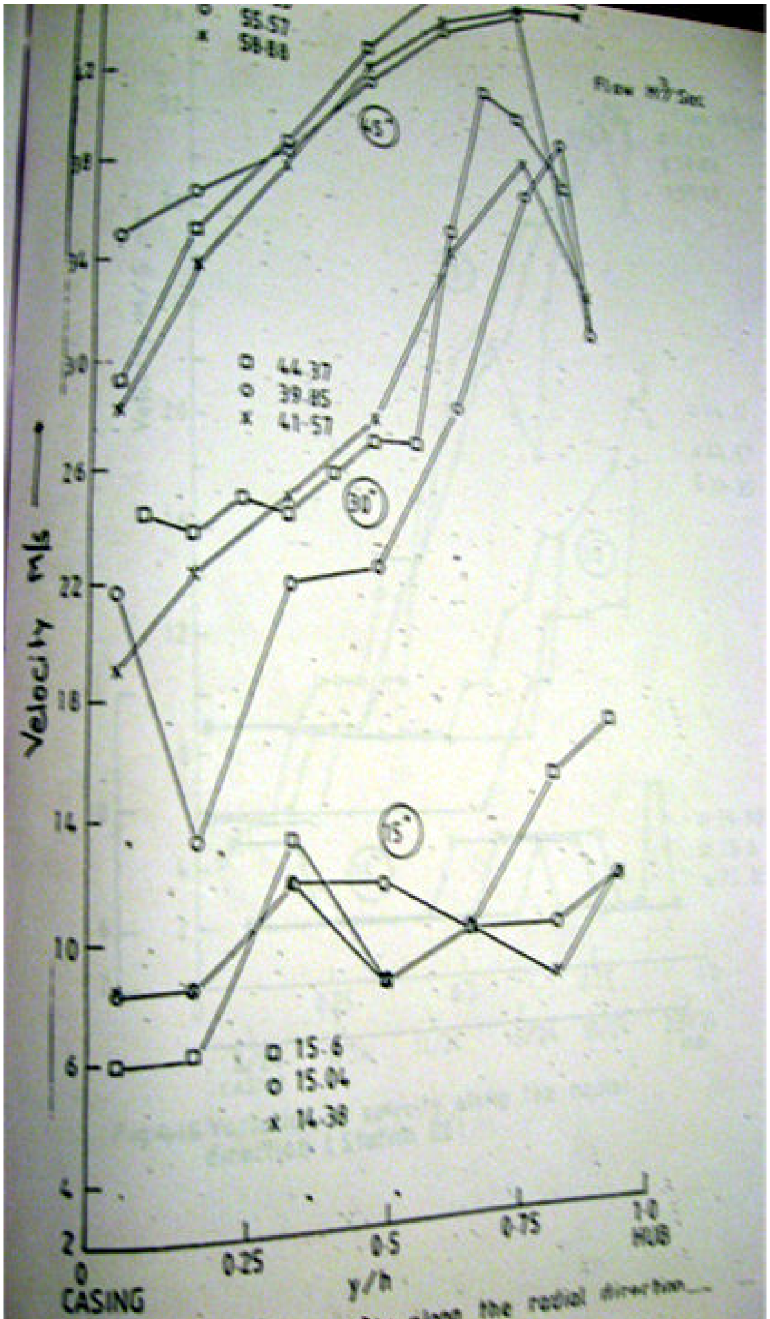


Fig.4.15 Variation of velocity along the radial direction

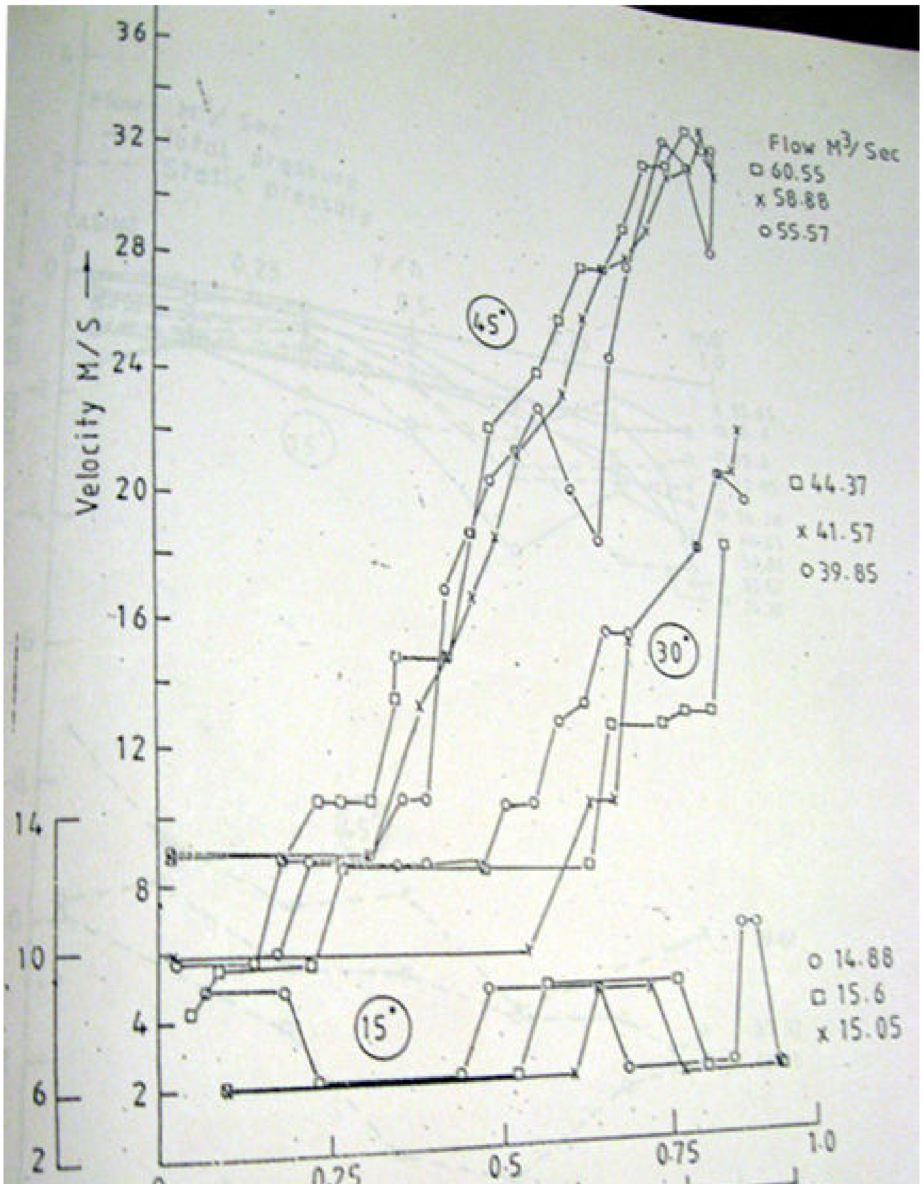


Fig 4.16 Variation of velocity along the radial direction (Station III)

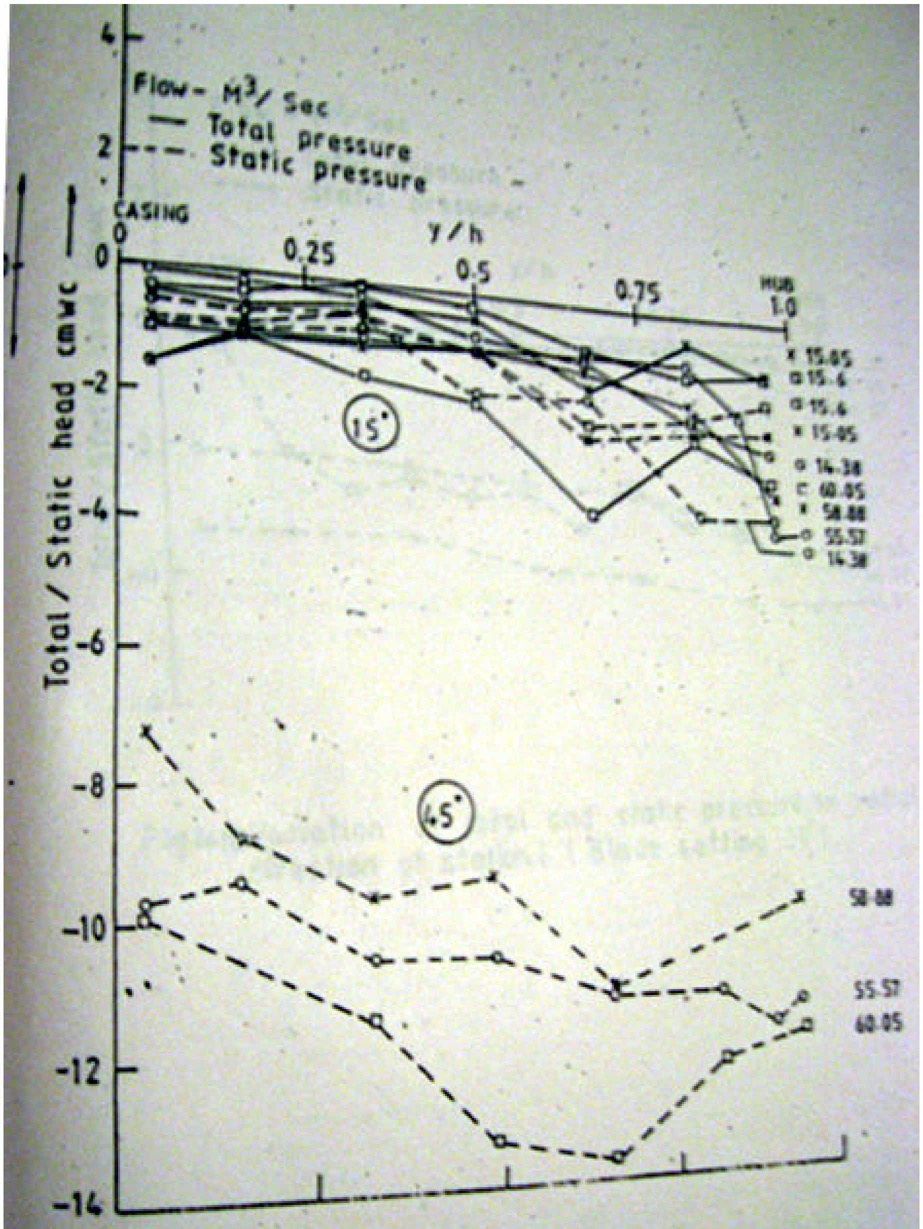


Fig 4.17 Variation of total & static pressure in radial direction at station I (Blade settings 15° & 45°)

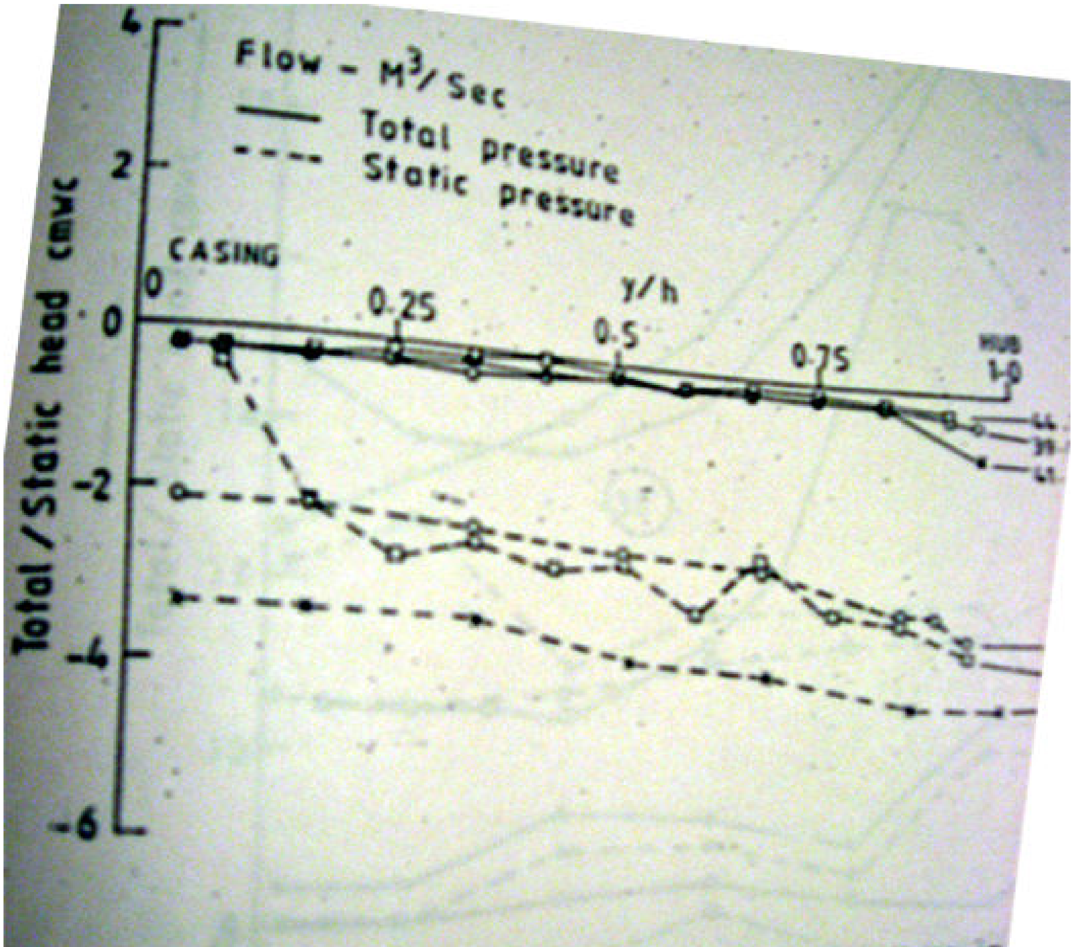


Fig 4.18 Variation of total and static pressure in direction at station 1 (Blade setting 30°)

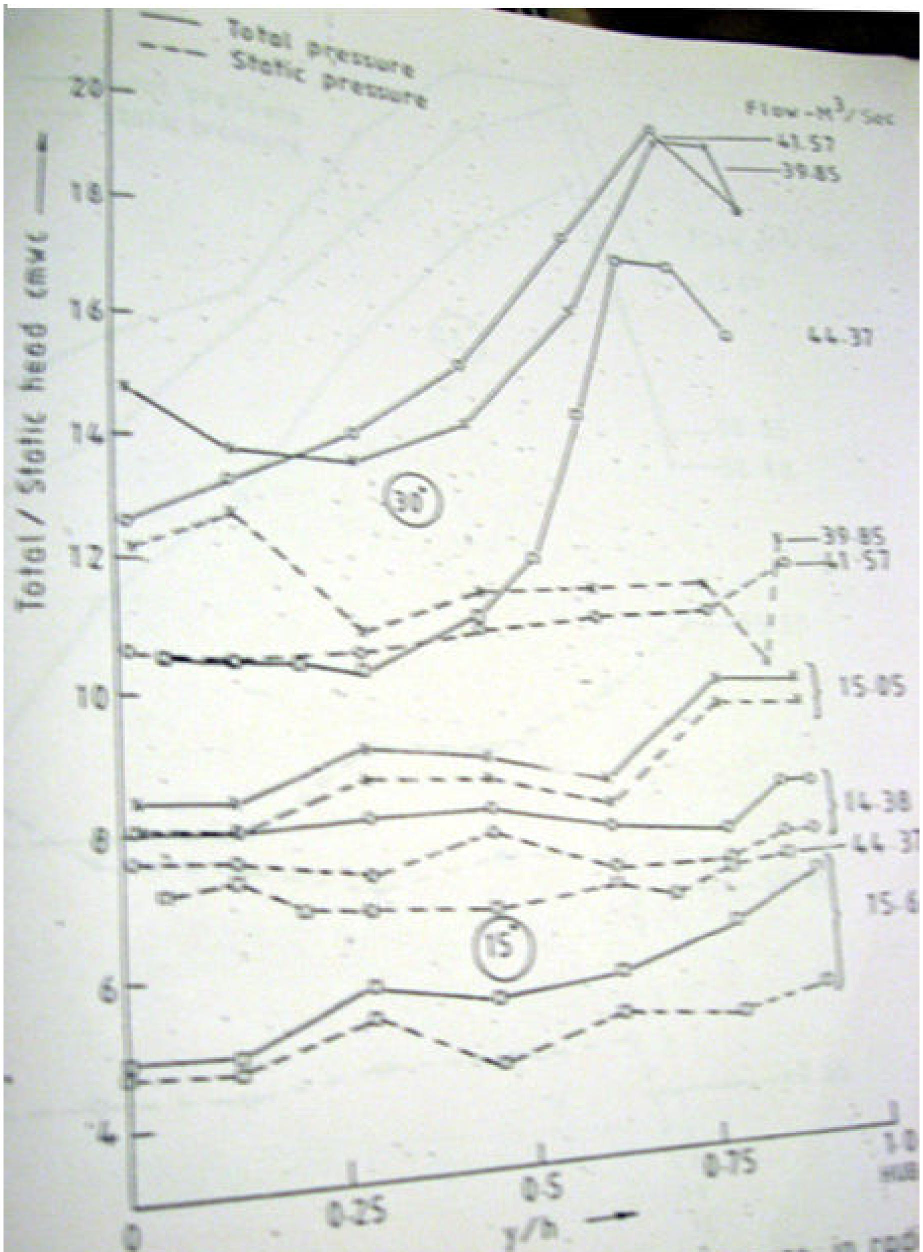
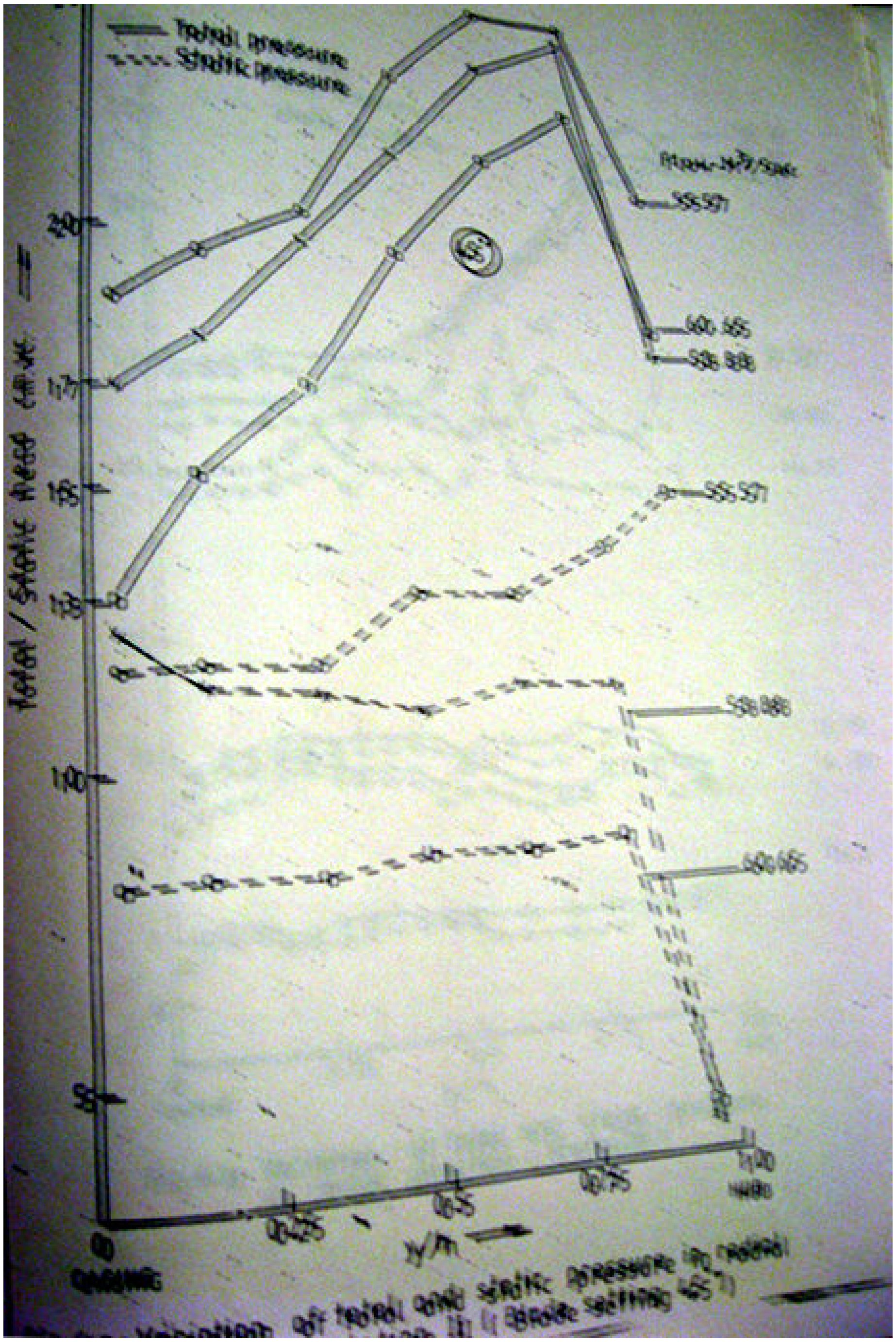


Fig.4.19 Variation of total and static pressure in radi
direction and station II (Blade setting 15° & 3



Total pressure
Static pressure
Total / Static head cm wc
45°
555.597
600.665
588.888
y/h
0.0
casing
0.25
0.5
0.75
1.00
hub

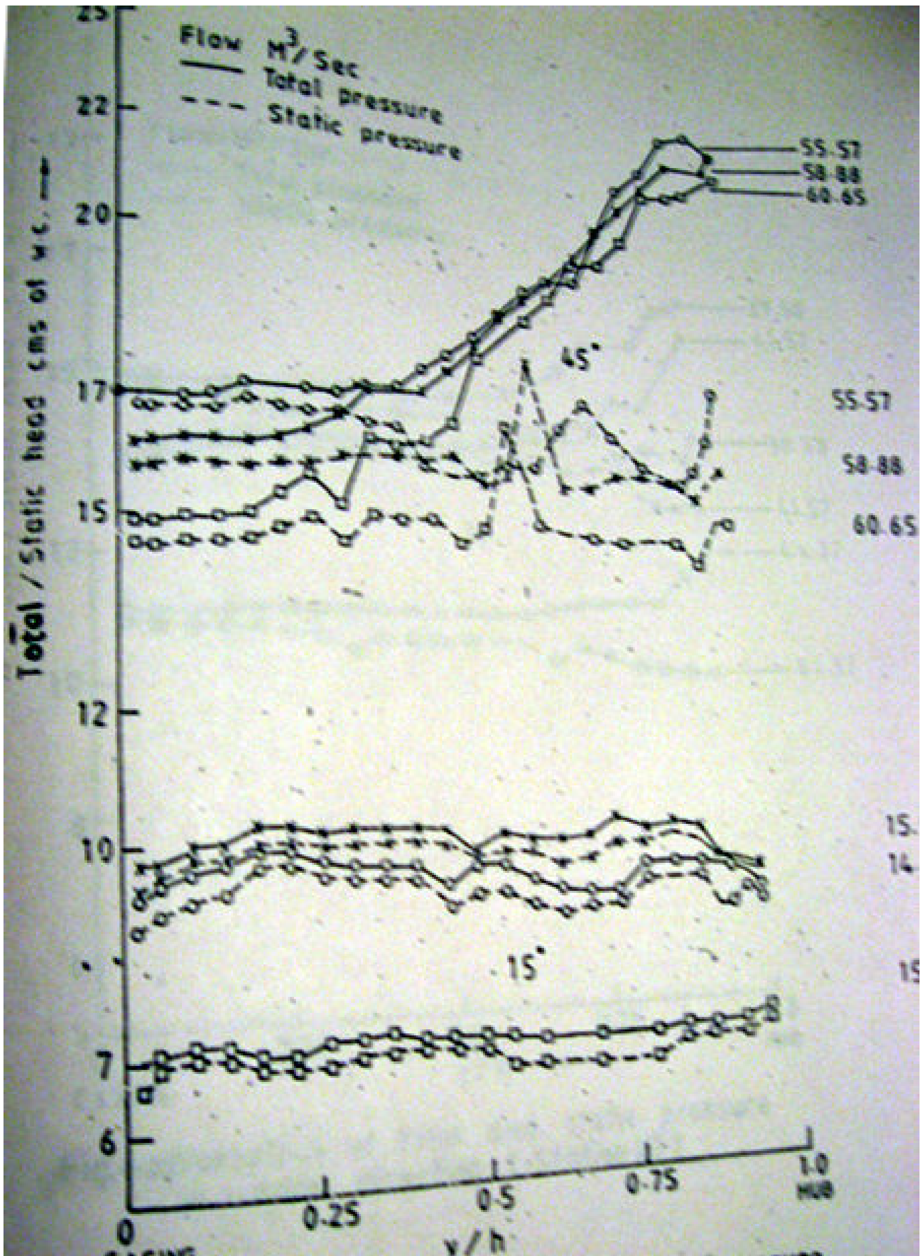
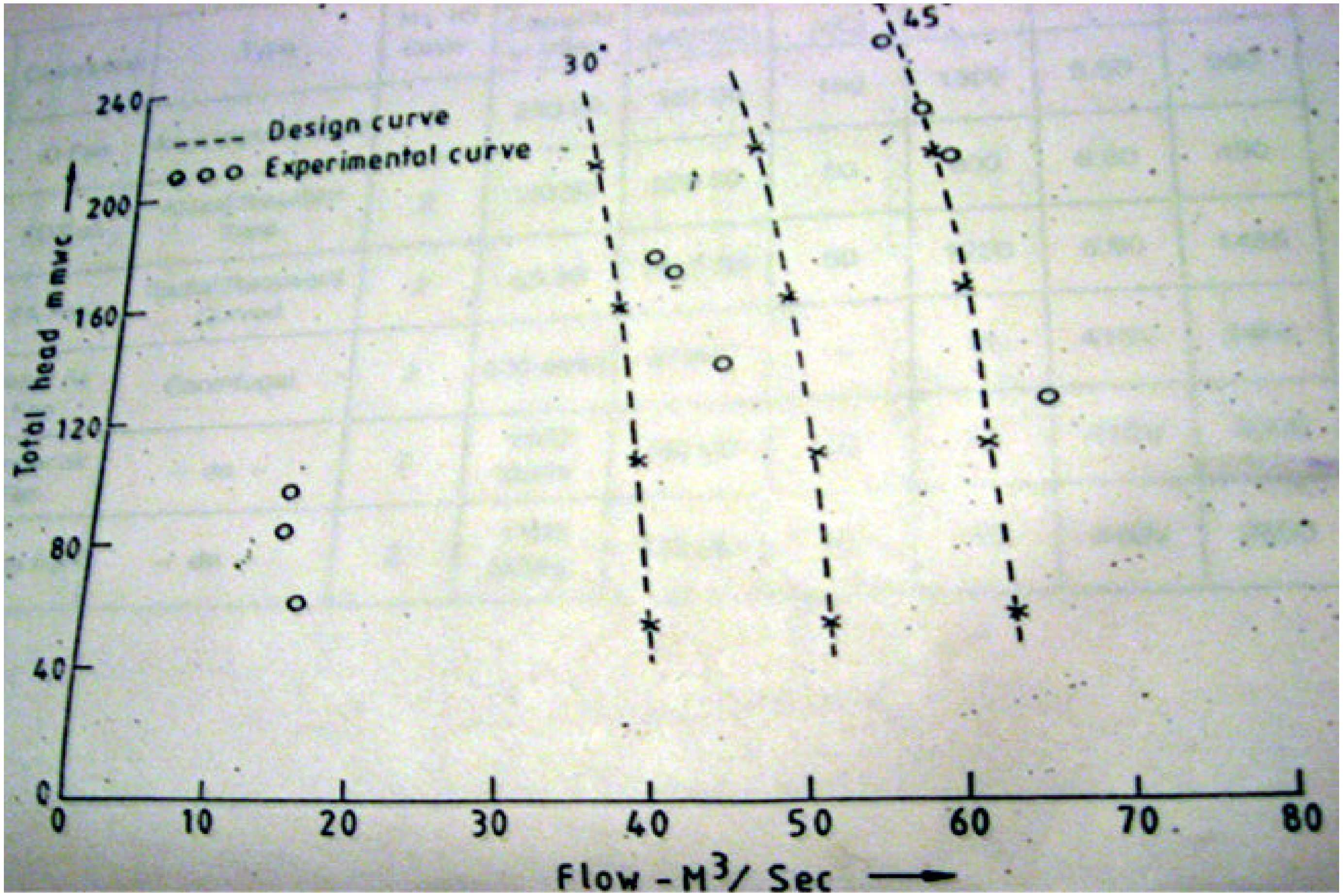


Fig.4.21 Variation of total and static pressure in radial direction (Station III)



Total head mmwc
240
200
160
120
80
40
0
---- Design curve
ooo Experimental curve
30°
45°
0
10
20
30
40
50
60
70
80
Flow - M3/ Sec

TABLE : 1

FANS FOR 210 MW UNIT IN THERMAL POWER STATIONS

S/n	Description	Type	No. off Boiler	Fan			KW	Voltage (KV/V)	RPM
				Capacity M3/Sec.	Pressure (MMWC)	Temp (oC)			
A	ID Fan	Axial Impulse Type	2	230.90	387.00	150	1300	6.60	990
B	FD Fan	Axial Reaction Type	2	103.80	520.50	50	800	6.60	490
C	FA Fan	Radial Backward Curved	2	65.30	1345.50	50	1250	6.60	1485
D	Ignitor Air Fan	Centrifugal	2	400 scfm	4" WG	--	11	415V	1460
E	Scanner air Fan	-- do --	2	1100 M3/hr	160.00	50	1.1	415V	3000
F	Seal Air Fan	-- do --	2	1100 M3/hr	150.00	50	10	440V	2850

TABLE-2

DETAILS OF BOILER FANS ON INDIAN SCENE

S/n	***Capacity of unit (MW)***	***Fan service***	***Type of fan***	***Type of control***
1	100/110	Forced draft	Axial impulse	Inlet regulating vane
		Induced draft	Axial Impulse	Inlet regulating vane
			Centrifugal fan	Inlet regulating vane
		Primary air	Radial single suction	Inlet regulating vane
2	210	Forced draft	Axial impulse	Inlet regulating vane
			Axial reaction	Blade pitch control
		Induced draft	Axial impulse	Inlet regulating vane
			Axial reaction	Blade pitch control
			Radial double suction	Variable speed (Hydraulic coupling)
		Primary air	Radial single suction, backward curved blade	Inlet regulating vane
3	500	Forced draft	Axial reaction	Blade pitch control
		Induced draft	Radial backward curved	Inlet vane and speed control (Hydraulic coupling)
		Primary air	Axial impulse	Inlet guide vane
			Axial reaction	Blade pitch control

TABLE-3

GENERAL GUIDELINES FOR APPROPRIATE SELECTION OF FANS FOR VARIOUS APPLICATIONS

Type of fan		***Application***				
		Forced Draught		***Induced Draught***		***Primary Air***
Axial reaction Single-stage	I.	500 MW 200 MW 110MW				
Axial reaction Double stage			III.	200 MW 500 MW	I.	500 MW
Axial impulse	II.	500 MW 200 MW 110MW	I.	500 MW 200 MW 110 MW		
Radial – single Inlet (overhang) Radial single Inlet (simply supported)	III.	110 MW			II.	200 MW
Radial Double inlet	III.	200 MW 500 MW	II.	110 MW 200 MW 500 MW	II.	500 MW

I, II and III denote the preferences in the descending order.

TABLE-4

BREAK UP OF FORCED OUTAGES IN 200/210 MW UNITS FOR THREE DIFFERENT YEARS

		Ist year (18 Units)			IInd year (22 Units)			IIIrd year (34 Units)		
S/n	Couse of outage	No. of outages	Energy Loss Gwh	% of Gwh loss to the total F.O. loss in 200/210 MW group	No. of outages	Energy Loss Gwh	% of Gwh loss to the total F.O. loss in 200/210 MW group	No. of outages	Energy Loss Gwh	% of Gwh loss to the total F.O. loss in 200/210 MW group
1	Boiler	628	2882	37.57	640	2510.01	25.95	480	4102.23	26.36
2	Boiler Aux	69	551	7.18	142	1090.24	11.27	93	1035.37	6.64
3	Boiler & Boiler Aux	697	3433	44.75	782	3600.25	37.22	573	5137.60	33.00
4	Tuboine	210	2343	30.54	146	2688.99	27.80	134	2498.73	16.05
5	Turbine Aux	59	477	6.22	71	422.07	4.36	74	758.19	4.88
6	Turbine &Turbine Aux	269	2820	36.76	217	3111.06	32.16	208	3256.92	20.93
7	Generator	73	471	6.15	71	789.10	8.16	89	3255.84	20.92
8	Others	213	947	12.35	307	2171.75	22.45	265	3913.89	25.15
9	Total	1252	7671	100.00	1377	9672.16	100.00	1135	15564.26	100.00

TABLE-6

PARTIAL LOSS OF GENERATION IN GWH IN ID, FD AND PA FANS OF 200/210 MW UNITS DUE TO VARIOUS LONG AND SHORT DURATION CONSTRAINTS FOR TWO DIFFERENT YEARS

		Ist year (22 Units)		IInd year (34 Units)	
S/n	Cause of Auxilary trouble	Gwh loss	Equivalent full load outage in days	Gwh loss	Equivalent full load outage in days
1	I.D. Fan	155.59	32.42	188.93	39.00
2	F.D. Fan	42.00	8.75	28.26	5.80
3	P.A. Fan	197.46	41.14	405.77	84.00

TABLE-7A

INDUCED DRAFT FAN

	Initial cost	***Extra operating cost (Million rupees)***	***Loss due to outages/unit/ year (Million rupees)***	***Total (Million rupees)***
Centrifugal (Vane control)	---	42.70	---	---
Centrifugal (Speed control) hydraulic coupling	26.50	19.71	10.60	56.80
Axial impulse (Inlet-vane control)	22.20	13.97	12.602	48.80
Axial reaction (Blade pitch control)	26.20	Base	10.622	36.02

TABLE-7B

FORCED DRAFT FAN

	Initial cost	***Extra operating cost (Million rupees)***	***Loss due to outages/unit/ year (Million rupees)***	***Total (Million rupees)***
Axial Impulse (Inlet Vane control)	16.20	8.556	12.80	47.556
Axial Reaction (Blade pitch control)	18.70	Base	10.60	29.300

TABLE – 8

TECHNICAL DATA

VARIABLE PITCH FORCED DRAFT FAN

Particulars	Fan Data
Fan Orientation	Horizontal
Medium handled	Atmospheric Air
Location	Ground level
No of fans/boiler	Two
Fan design rating	

a)	Capacity , M3/sec	:	108.4
b)	Total head developed, mm of W.G.	:	520.0
c)	Temperature of medium, C	:	50
d)	Sp. Wt. Of medium, Kg/M3	:	1.090
e)	Speed, RPM	:	1480

Drive Motor	750 KW, 6.6 KV 60 C.P.S., 1500RPM
Regulation	Variable pitch

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6	DEEPROSE, W.M.	“The Usefulness of BS 848 in establishing the installed performance of a fan” – Mech. E. Conference Publications
7	R Parker	“Pressure Fluctuations due to interaction between blade rows in axial flow compressors”
8	BARD, H.	“The Stabilization of axial fan performance”, Mech. E. Conference Publications.
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11	J H Horlock	“Axial flow turbines”
12	R A Wallis	“Design procedure for optimal axial flow fans”
13	D S Whitehead	“Aerodynamic aspects of blade vibrations”
14	TRAXEL, W.	“Parallel Operations of Fans” Seminar on “Fans”
15	M Turner	“All you need to know about Fans”
16	PARTHASARTHY. J.	“Some experience and recommendations of A.P.S.E.B. on Fans “
17	ECK, B.	“Fans”
19	Central Electricity Authority	“Annual Report of Central Electricity Authority”
20	DALY, B.B.	“Woods Practical Guide to Fan Engineering.”
21	R A Wallis	“Sheet metal blades for axial flow fans”