DESIGN AND DEVELOPMENT OF A REVERSIBLE PLASTIC BLADE FAN

by

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INTRODUCTION

In most geographical areas, heavy construction equipment is used throughout the year. In summer working conditions the cooling system fan, which is a propeller type fan, is used to cool the engine and the cooling system fluid by pulling air through the radiator. In winter working conditions the fan is used to cool the engine cooling system fluid by pushing the air through the radiator, and the heat from this fluid and the engine is used to keep the equipment operator comfortable.

As much as eight hours working time may be lost to maintenance in changing from summer to winter stationary blade fans on certain types of construction equipment. Therefore it was decided to build a fan that would reduce maintenance and extra equipment cost.

It was the intent of this work to: (1) design a symmetrical plastic fan blade, (2) design a fan hub that would allow the blade pitch to be reversed, and (3) test the reversible blade fan to determine its efficiency.

It was decided to make plastic blades to observe whether they would withstand the same stresses as metal blades.

PROCEDURE

Fan Design

<u>Design Guides</u>. The design of the reversible plastic blade fan was accomplished with the following requirements as guides:

1. The fan was to be an eight-blade fan thirty-four and one-half inches in diameter.

2. The blades were to be symmetrical and made from Polyester Resin reinforced with one layer of fiberglass cloth.

3. The blade reversing mechanism was to be simple and non-automatic.

4. The blade pitch would not exceed twenty degrees because it was desired to design a fan with as high efficiency and stable performance as possible.[1]

5. The fan hub components were to be made of steel and aluminum.

Fan Blade Design. The fan blade shaft was made of SAE 1035 steel (Appendix B, Drawing 32). The maximum external permissible load on the fan shaft was calculated to be 1,585 pounds (Appendix A).

When a body revolves about an axis, some connection must exist capable of applying force enough to the body to constantly deviate it toward the axis. This deviating force is known as centripetal force. The equal and opposite resistance offered by the body to the connection is called a centrifugal force.[2] The centrifugal force, shown in Fig. 1, should not exceed the maximum external tensile force that can be applied to the blade shaft. With this force known, it was determined that the maximum design speed of the fan was 2000 revolutions per minute (Appendix A).

The majority of the stress in the blade shaft was caused by the initial tightening of the blade-holding nut. The centrifugal

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[1] Lionel S. Marks, Mechanical Engineer's Handbook, Text Book Edition, Fifth Edition, p. 1917.

[2] Ibid., p. 204.

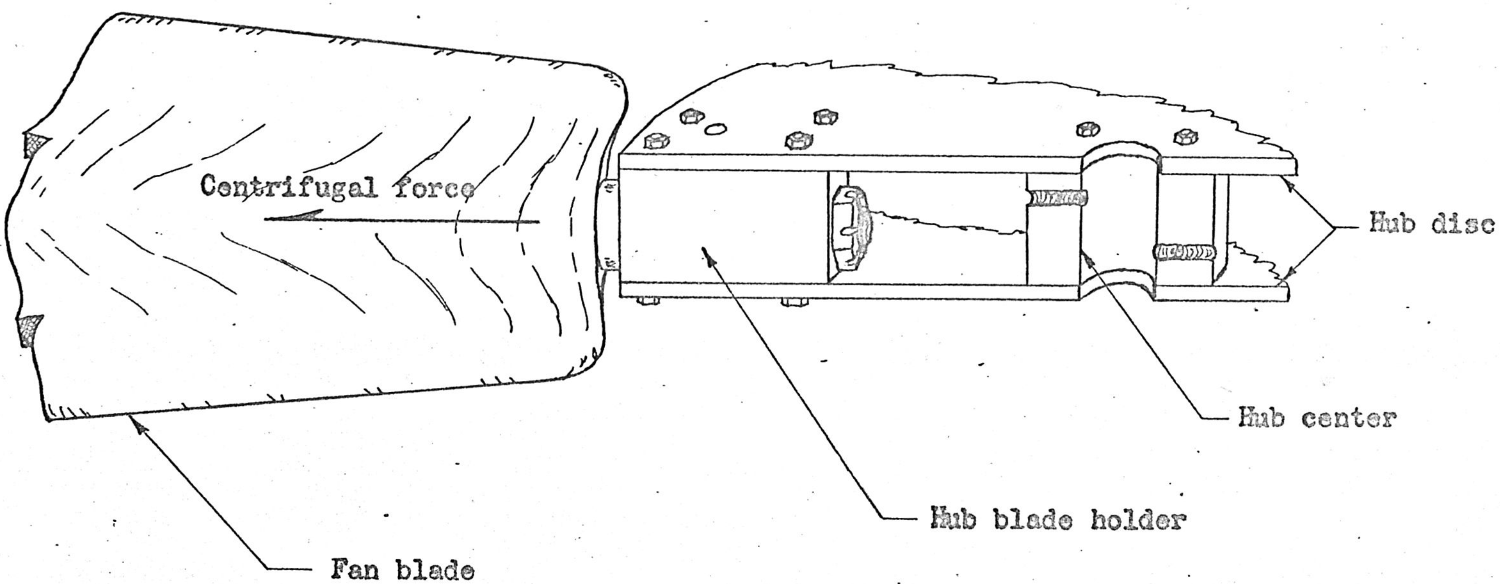

Fig. 1. Schematic drawing of fan blade and hub assembly showing centrifugal force action.

force due to blade rotation caused some stress in the blade shaft. Therefore, in designing the blade shaft, these stresses were the only ones considered.

The fan blade was molded conforming to the design as shown in Appendix B, Drawing 31. Polyester resin and fiberglass cloth were the materials used in the fan blade. The resin used in the blade had the following physical and mechanical properties.[3]

Specific gravity	1.6 to 2.0
Tensile strength	8,000 to 25,000 psi
Flexural strength	40,000 to 75,000 psi
Compressive strength	25,000 to 45,000 psi
Modulus of elasticity in flexure ...	20×10^5 to 38×10^5 psi
Rockwell hardness	M100 to M120

The fiberglass reinforced plastic fan blade was molded in the following manner:

1. A wooden pattern of the blade was constructed. This pattern was made in two parts from balsa wood to conform to the blade design as specified in Drawing 31 of Appendix B.

2. The blade mold was made in two parts, the cope and the drag (Plate I). These parts were equal and opposite and were made by first building a metal frame approximately two inches larger than the blasa wood patterns. The metal frames were centered around the wooden patterns and fastened to the board. The board and patterns were greased to insure the mold releasing from the board after being poured.

3. A compound of approximately 60 per cent sulphur and 40 per cent clay was heated until it became a viscous liquid. With the patterns in place in the metal frames,

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[3] Materials in Design Engineering, Materials Selector Issue, Mid-October 1962, p. 219.

EXPLANATION OF PLATE I

A view of the wooden pattern, fiberglass cloth, polyester resin, sealant, steel blade shaft, metal resin container, connecting rubber tubes, grease mold release, mold open with blade and aluminum foil in place, and three unfinished plastic blades.

PLATE I

the liquid was poured into the mold and allowed to harden. After the molds had cooled, the patterns were removed.

4. A sheet of aluminum foil was placed in each mold and roughly formed to the contour of the blade surface. The aluminum foil was then removed and a thin coating of grease was placed on the surfaces of the mold. The aluminum foil was then replaced in the mold and all wrinkles smoothed out until the foil conformed to the curvature of the blade. The foil was then trimmed to one-quarter of an inch of the blade curvature in the mold and all excess foil and grease removed. (The polyester resin will not set when in contact with grease.)

5. "Alcoa" play dough was placed around the edge of one half of the mold for a sealant. Resin will not adhere to the moist play dough. Metal dowels were placed in the air ports of the mold to insure their being open.

6. The fiberglass cloth was cut to fit the mold. The metal blade shaft with the fiberglass cloth interwoven on it was placed in one half of the mold.

7. The cope and drag were clamped together, the air port dowels removed, and the metal threaded tube placed in the resin input port.

8. The correct amount of resin was placed in a quart metal container. The setting agent was added and the two thoroughly stirred for approximately sixty seconds. The lid was then placed on the quart container and an air hose fastened to one metal tube in the lid. Another hose was fastened to the second metal tube in the lid. The hose was then fastened to the threaded inlet tube at the bottom of the mold.

9. Two to five pounds of air pressure was applied to the container, and the resin was forced into the mold. As the resin ran out, the air ports they were closed by using play dough. When the mold was full the air pressure was shut off and the top air port was plugged. The mold was turned upside down, the inlet tube removed, and the inlet port closed with play dough.

10. The mold was then turned right side up and left to stand for fourteen to sixteen hours to allow curing so the blade would not warp after removal from the mold.

11. The resin container and tubes were cleaned by forcing air through the tubes and cleaning the container with rags. This must be done within twenty to twenty-five

minutes after the setting agent is added to the resin, as the amount added will cause the resin to set in approximately thirty minutes.

12. After the blade was removed from the mold, the aluminum foil was peeled away from the hardened plastic (Plate II). The blade was trimmed and sanded in preparation for applying a thin finish coat of resin to its surface. The mold was cleaned with rags and set up for casting another blade.

13. The finish coat of plastic was applied with a brush. Care was taken to insure a smooth finish surface on the blade.

Fan Hub Design. The fan hub consists of three basic parts, the center, the mounting disc, and the blade mount.

The fan hub center was made from SAE 1035 steel rod to specifications set forth in Appendix B, Drawing 22.

The fan hub blade mounts were made from Cast Aluminum Alloy Type 218 to specifications set forth in Drawing 21 of Appendix B.

The fan hub discs were made from Wrought Aluminum Alloy Type 7178-T6 to specifications set forth in Appendix B, Drawing 23.

The four matching holes for the fan hub center were drilled in the disc and center simultaneously. This was done to avoid drilling larger holes in the disc and center to compensate for variation of hole centers if the pieces were drilled individually.

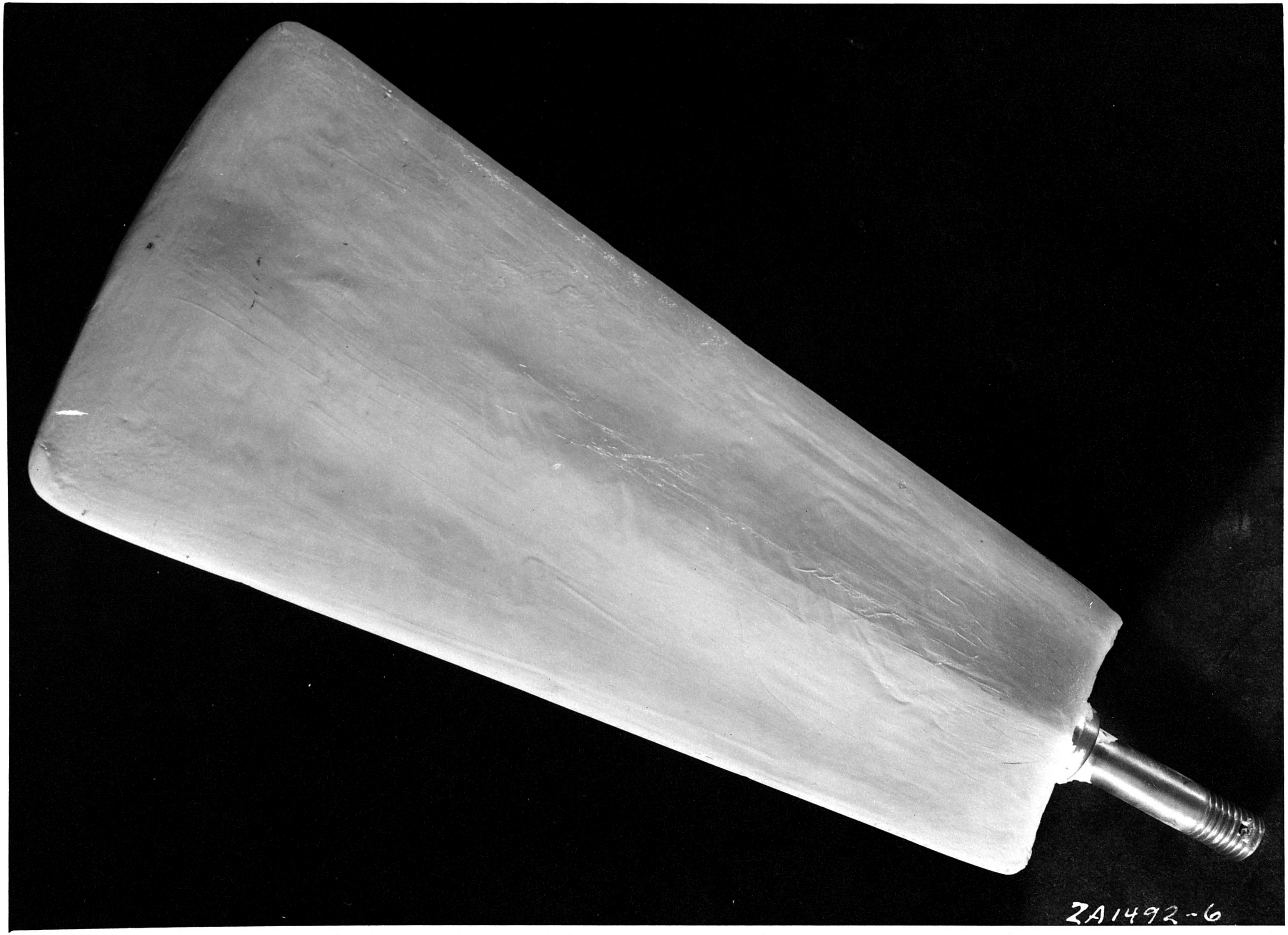
The eight sets of holes in the disc were drilled at the same time the matching holes were drilled in each blade mount. This was also done to avoid drilling larger holes in each piece if drilled separately. The hub specifications were substantiated by calculations as shown in Appendix A.

The fan parts were assembled by first setting the blades in the blade mounts at a pitch angle of fifteen degrees, tightening

EXPLANATION OF PLATE II

A view of a symmetrical plastic fan blade before sanding and applying a finish coat of polyester resin.

PLATE II



the blade holding nut, locking the nut into position with a cotter pin, and tightening the blade set-screw. The blade mounts and the hub center were placed in position on one of the hub discs, the steel bolts were placed in all holes, and the second disc put in place. The bolts were then tightened. Thus, the fan was assembled as shown in Plate III and Appendix B, Drawing 30.

Fan Testing

Testing Guides. The testing of the fan was accomplished with the following requirements as guides:

1. The fan blades were not to be reversed. Since the blades were symmetrical, it was assumed that reversing would give the same test results.

2. The fan blades were set at a pitch angle of fifteen degrees and this angle was not changed.

3. The fan was treated as an exhauster fan.

4. The air inlet area to the test duct was changed by using the standard ASME nozzles available in the laboratory.

General Testing Information. After assembly, the fan was statically balanced by placing it on a bearing mounted shaft and adding weight to the hub until balanced. There was no convenient means for dynamic balancing, therefore this was not done.

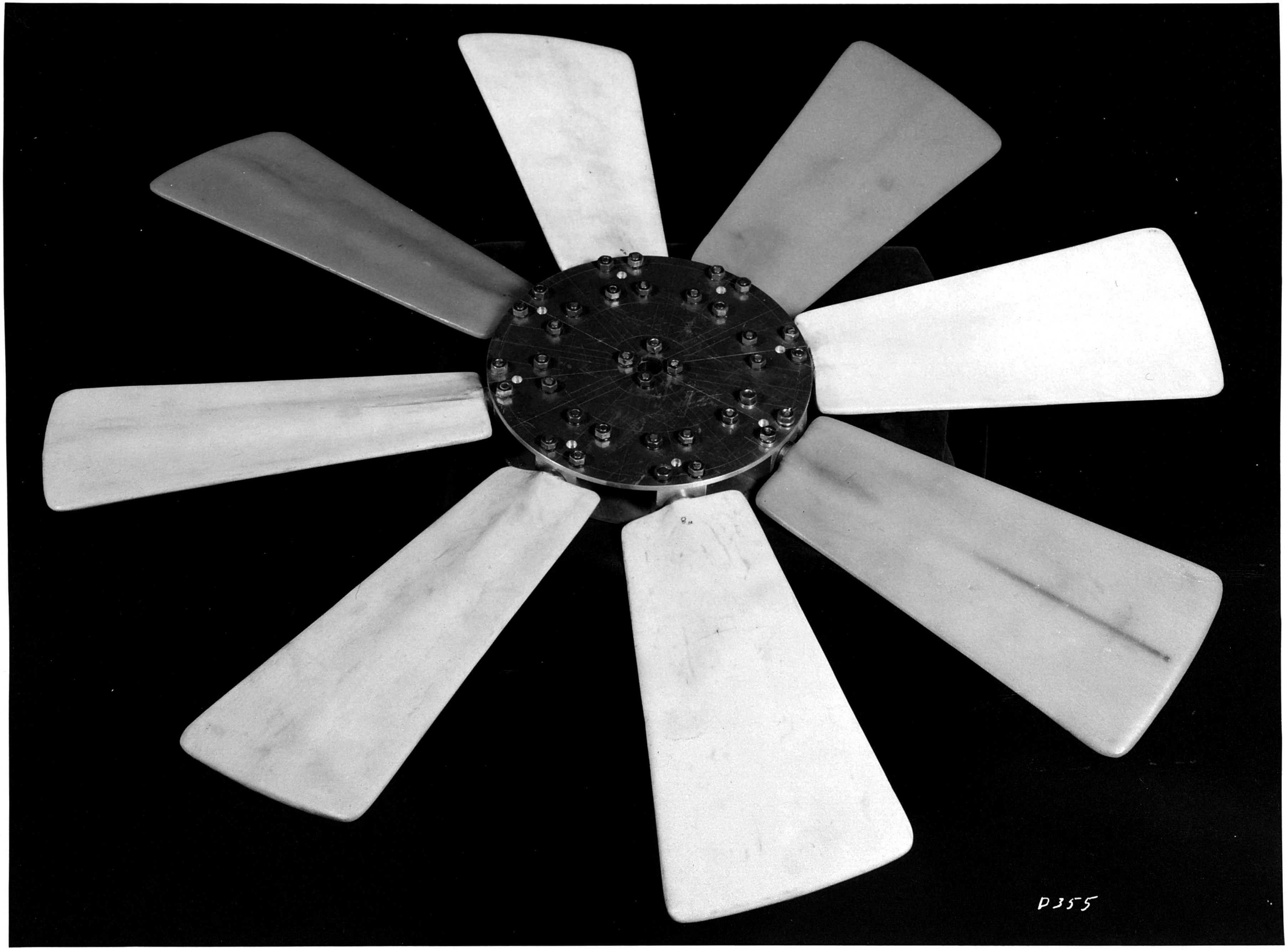
After static balancing, the fan was mounted in the test tunnel as shown in Plate IV. The test equipment was constructed specifically for testing this fan.

Twenty-four tests were run on the fan. Four standard RPM values were arbitrarily selected (250, 500, 750, and 1000) with six tests conducted at or near each of these selected speeds.

EXPLANATION OF PLATE III

A view of the assembled reversible plastic blade fan.

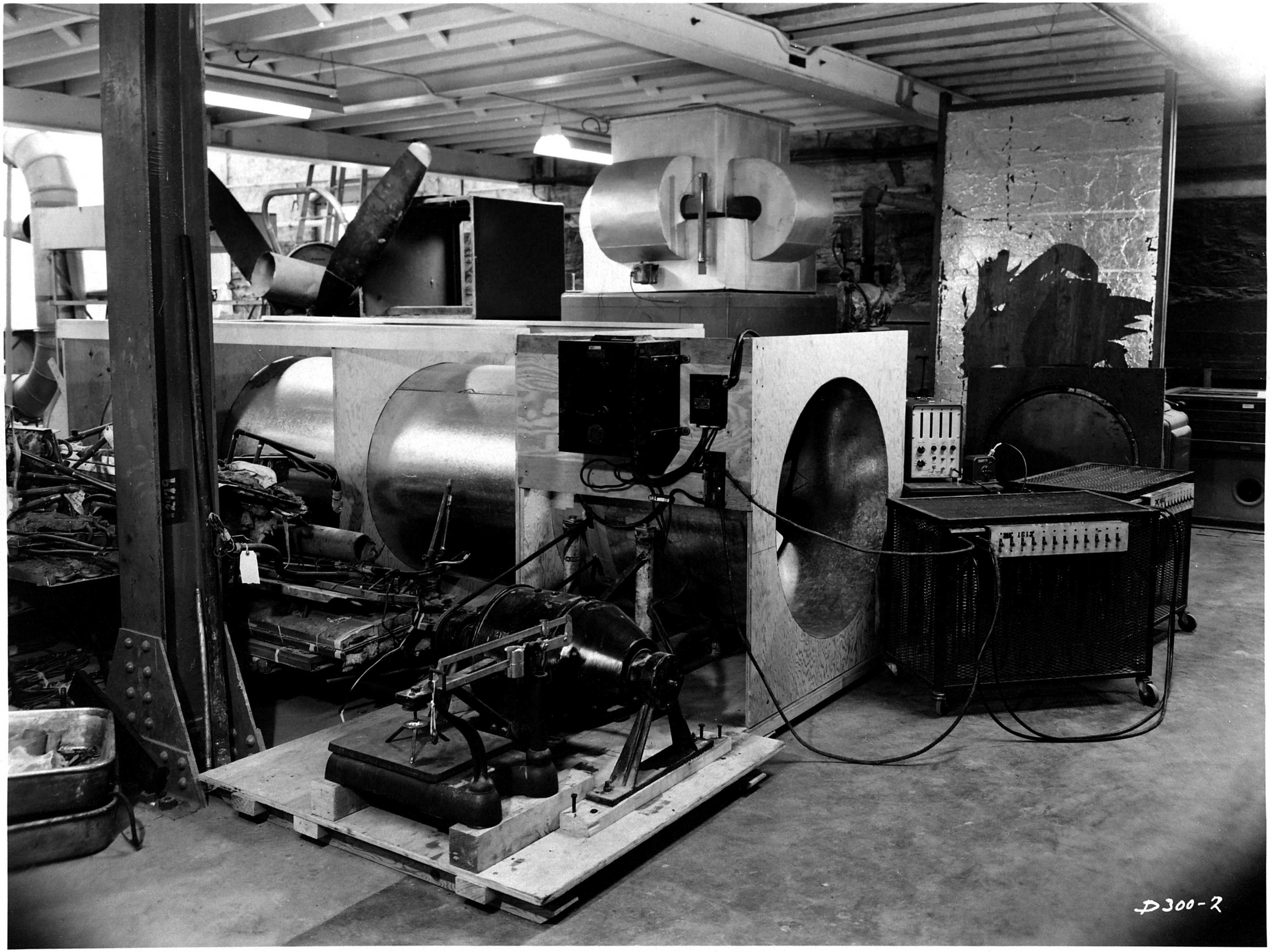
PLATE III



EXPLANATION OF PLATE IV

A view of the test facilities showing the thirty-six-inch duct, the five-horsepower dynamometer and scales, the wire resistance racks, and the revolution counter.

PLATE IV



Tests were run with the inlet duct closed and completely open as well as using the 7, 10, 14, and 17-inch standard ASME nozzles. Plate V shows the duct with the 7-inch nozzle installed.

The speed of the fan for each test was controlled by wire resistance racks as shown in Plate IV. The racks were connected in series in the electrical circuit of the five-horsepower direct current dynamometer used for supplying power to the fan. The actual fan speed was determined with the Beckman electrical revolution counter also shown in Plate IV.

A ten-point traverse was made for each test run with the ASME nozzles and the open duct. A total pressure tube was used in making the traverse. Static pressure was measured at four equally spaced holes on the circumference of the thirty-six-inch duct.

Pressure measurements were made with a Dwyer Number 1420 Portable Hooke Gage. The gage accuracy is one-thousandth of an inch of water.

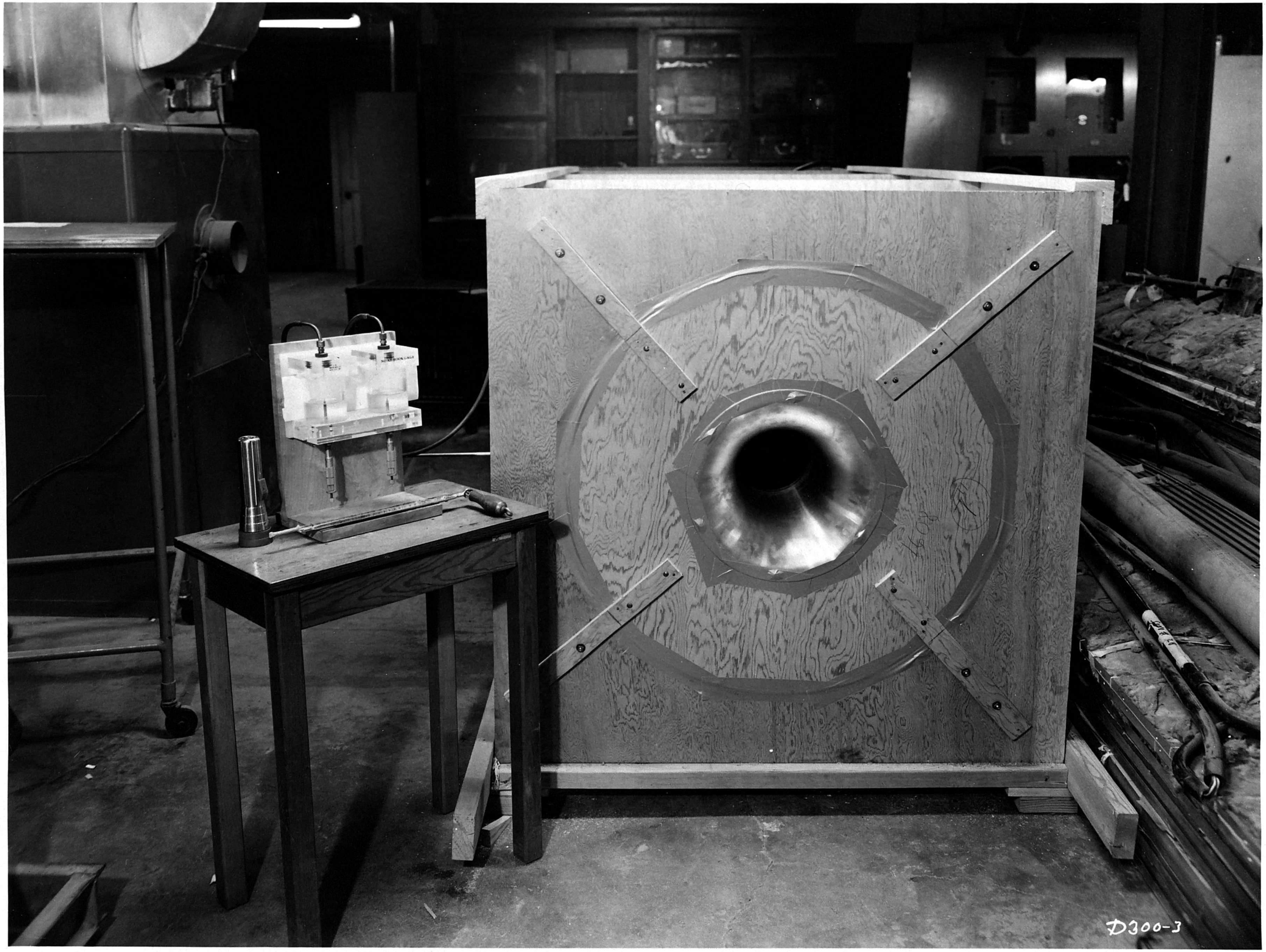
The wet and dry bulb temperatures were taken with a sling psychrometer. The barometric pressure was recorded from a mercury column barometer in the laboratory.

All test data were corrected to standard atmospheric conditions and standard revolutions per minute. Performance calculations were made for each standard speed selected (Appendix C, tables and sample calculations).

EXPLANATION OF PLATE V

A view of the test facilities showing the seven-inch ASME nozzle installed for the air inlet. The Hooke Gage and sling psychrometer are also shown.

PLATE V



DISCUSSION

Fan Design

The axial flow, or propeller, type fan is best adapted to handling large volumes of fluid against low resistance. In its simpler forms, the propeller fan is most often used for ventilation service, such as being mounted in an opening in a wall, with no duct work.

The outstanding desirable feature of the propeller fan is that it will deliver large volumes of fluid when not restricted by guide vanes, elbows, or inlet and/or outlet duct diameters smaller than the fan diameter. Perhaps the main deficiencies of the propeller fan are that at certain rates of flow it tends to pulsate badly and that, due to its high rotative speed, it tends to be excessively noisy.

The fan blades were designed symmetrical so they could be reversed in pitch; thereby reversing the direction of fluid flow. The fan blades were molded using Polyester Resin, a thermosetting plastic, because it was desired to determine if the blades could be made of plastic and perform as satisfactorily as metal blades. Information about the bending strength between Polyester Resin and metal was unobtainable, therefore it was assumed that the blades would perform properly when tested at fan speeds up to 2000 revolutions per minute.

Using a factor of safety of 1.5, the blade shaft was designed to withstand stresses at a speed of 2000 revolutions per minute. The individual parts of the fan hub were also designed to withstand stresses at a speed of 2000 revolutions per minute.

The fan was statically balanced, but was not dynamically balanced because facilities for dynamic balancing were not available.

Fan Testing

A fan is a pump or volumetric machine which can move a given quantity of gas by imparting to it enough energy to set it in motion and also overcome all resistances to movement that are in its path of flow. The power expended on the gas by the fan depends on the volume of gas moved and on the total head against which it is moved.

The power expended on the air by the fan is expressed as Air Horsepower. The work done on the fan is the Shaft Horsepower used to drive the fan. The fan efficiency is the ratio of the output of the fan (Air Horsepower) to the input of the fan (Shaft Horsepower).

Energy of the fan is imparted to the air in two different forms: (1) kinetic or velocity energy, and (2) potential or pressure energy. Either of these forms can be changed to the other by a gradual change in the area of the channel through which the gas is flowing. If the fan discharges the gas at a velocity higher than is required for flow through the nozzle, and at a lower pressure than is required to overcome the system resistance, part of the velocity energy will be changed to pressure energy.

The test facilities as shown in Plates IV and V were set up to measure the quantities required for determining the Air Horsepower and Shaft Horsepower for the purpose of determining the fan efficiency.

Static pressure, total pressure, barometric pressure, fan revolutions per minute, scale beam weight, wet bulb temperature, and dry bulb temperature were the quantities measured during each test.

Figure 2 is a line diagram of the test facilities with respect to the fan position, air flow, and the volume considered as the system. The fan is located at the exit of the duct and is therefore considered as an exhauster or an induced draft fan.

From Fig. 2, consider the volume contained by points 1 and 2 to be infinite as compared to the volume contained from point 2 to 3. The shaft work on the fan is applied at point 3 of the system. The shaft work can be calculated after determining the speed of the fan and the scale beam weight reading.

If the system is considered to be the volume contained from point 1 to 2, it is observed that the pressure at point 1 is approximately equal to the pressure at point 2. The accuracy of the Hooke Gage used to measure total and static pressure at point 2 of the system is one-thousandth of an inch of water. So for the purpose of determining the power expended on the air by the fan and consequently the fan efficiency, the assumption was made that the pressures at points 1 and 2 are equal. From this assumption it can be said that the only energy imparted to the incompressible air by the fan is the kinetic or velocity energy

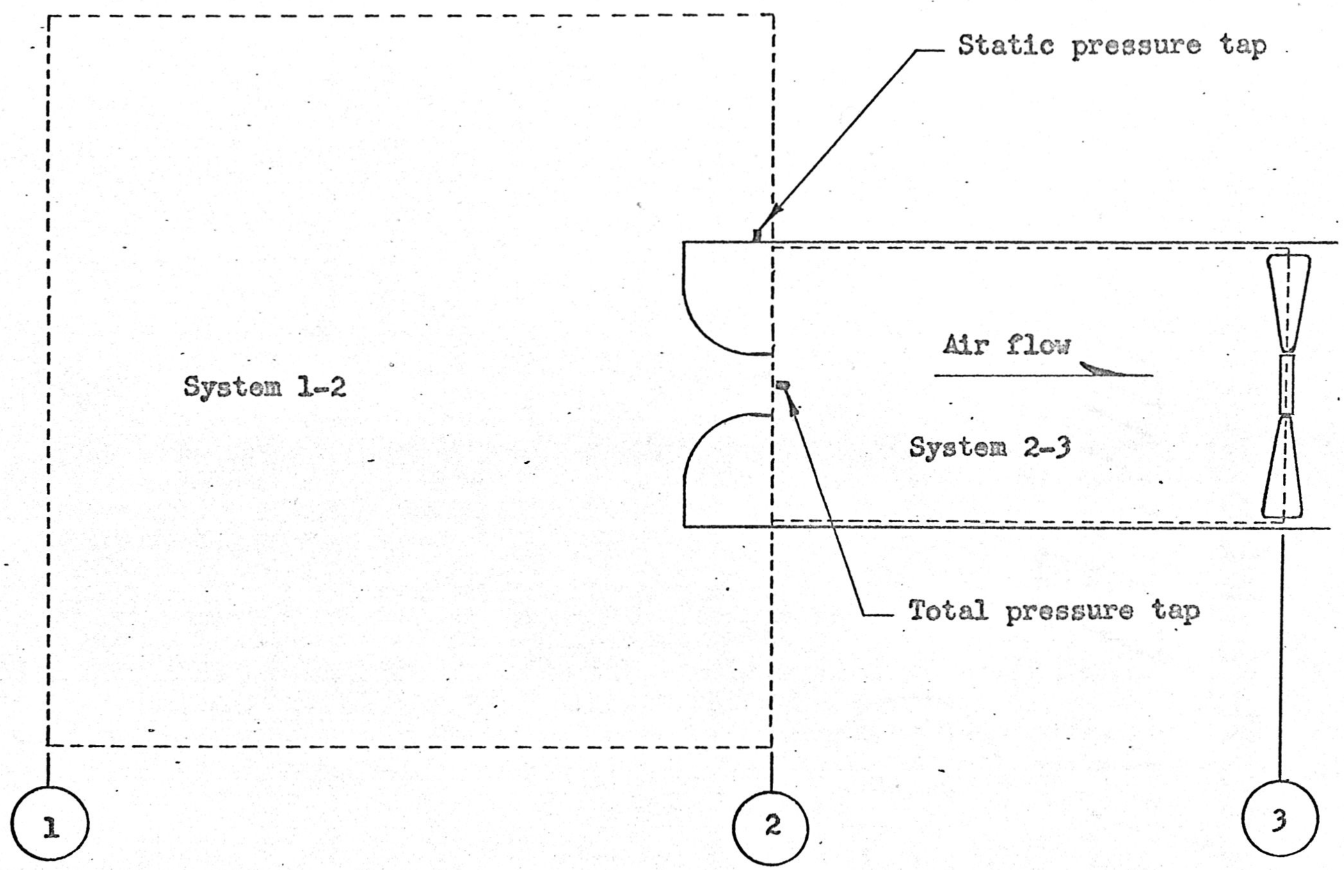


Fig. 2. A line diagram of the thermodynamic system.

form. This increase in kinetic energy is observed by the negative static pressure reading at point 2 as the air crosses the 1-2 system boundary at point 2.

As shown in the sample calculations, the volume flow rate can be calculated using the velocity pressure reading (corrected to standard air conditions). The air horsepower was calculated using the static pressure reading corrected to standard air conditions. After these calculations were made, the fan efficiency was calculated.

From the curves plotted in Fig. 3, it was noted that the fan operation was very inefficient for all tests except the open duct test. The efficiency improved with increase in speed and inlet area as shown in Figs. 4, 5, and 6. This observation further substantiates the statement that the propeller type fan is primarily used in open duct or without duct as a ventilating fan, operating at high speeds and moving large volumes of fluid.

The curves of Figs. 3, 4, 5, and 6 also indicate that the test method should be improved to test the fan in the area for which this type fan is designed. A series of larger nozzles or orifices could be used at the inlet to give a closer observation of the fan performance nearer the open duct conditions.

CONCLUSIONS

In comparing the experimental plastic blade fan with similar propeller type fans on the market, it was found that it compared favorably with an adjustable pitch blade propeller type fan manufactured by the Joy Manufacturing Company. The experimental fan

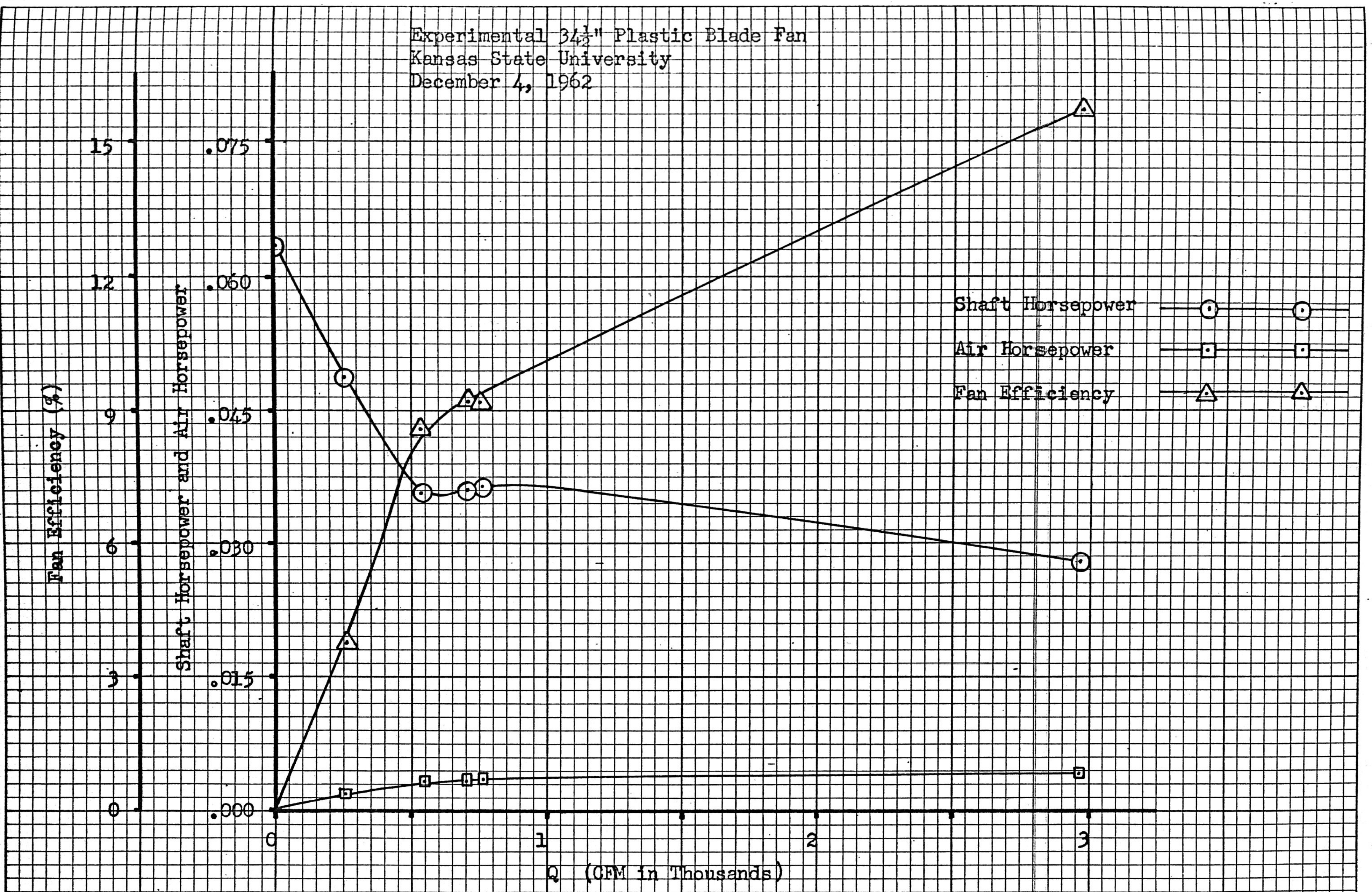


Fig. 3. Plastic blade fan characteristics at 250 RPM

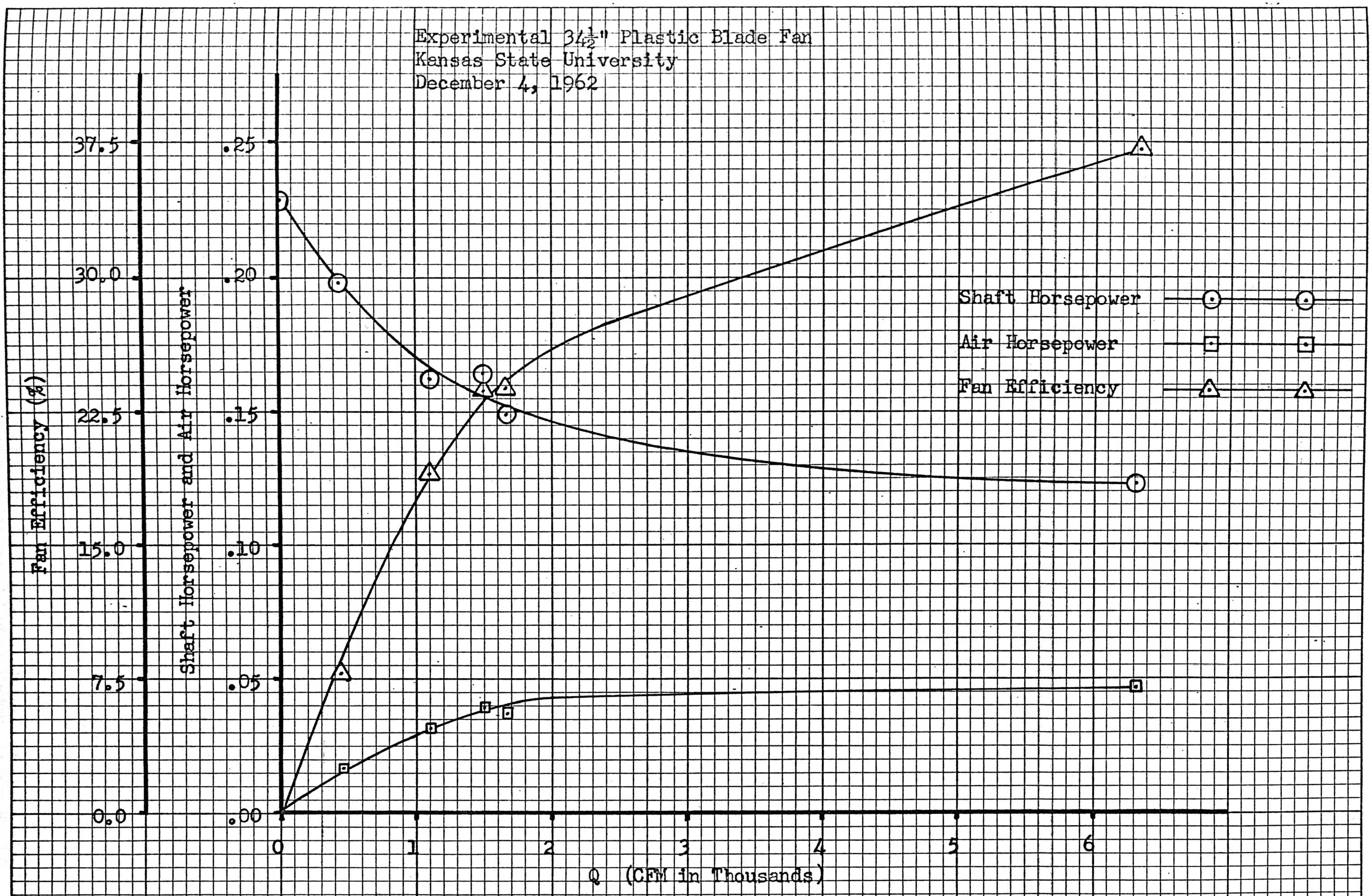


Fig. 4. Plastic blade fan characteristics at 500 RPM

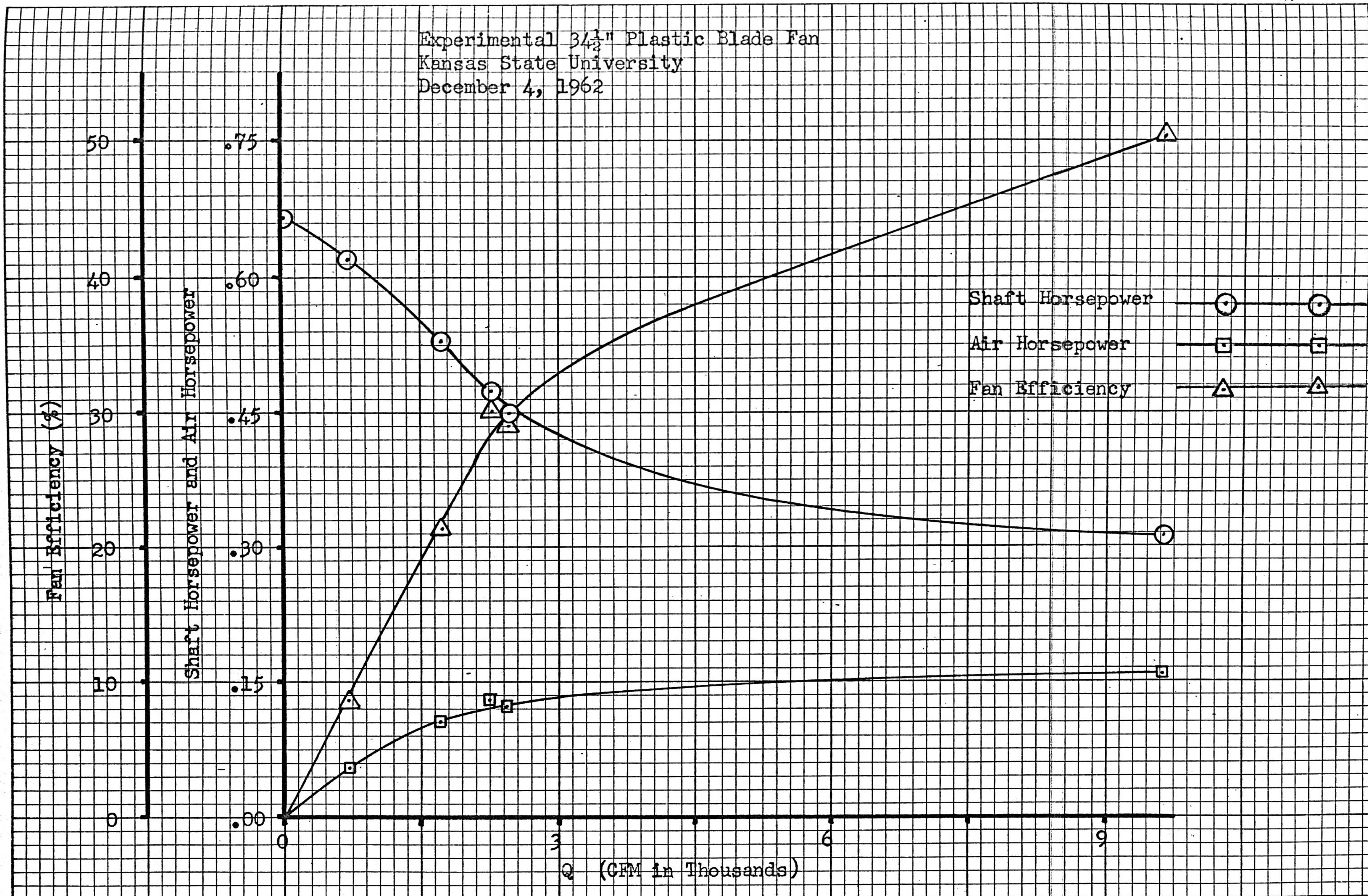


Fig. 5. Plastic blade fan characteristics at 750 RPM

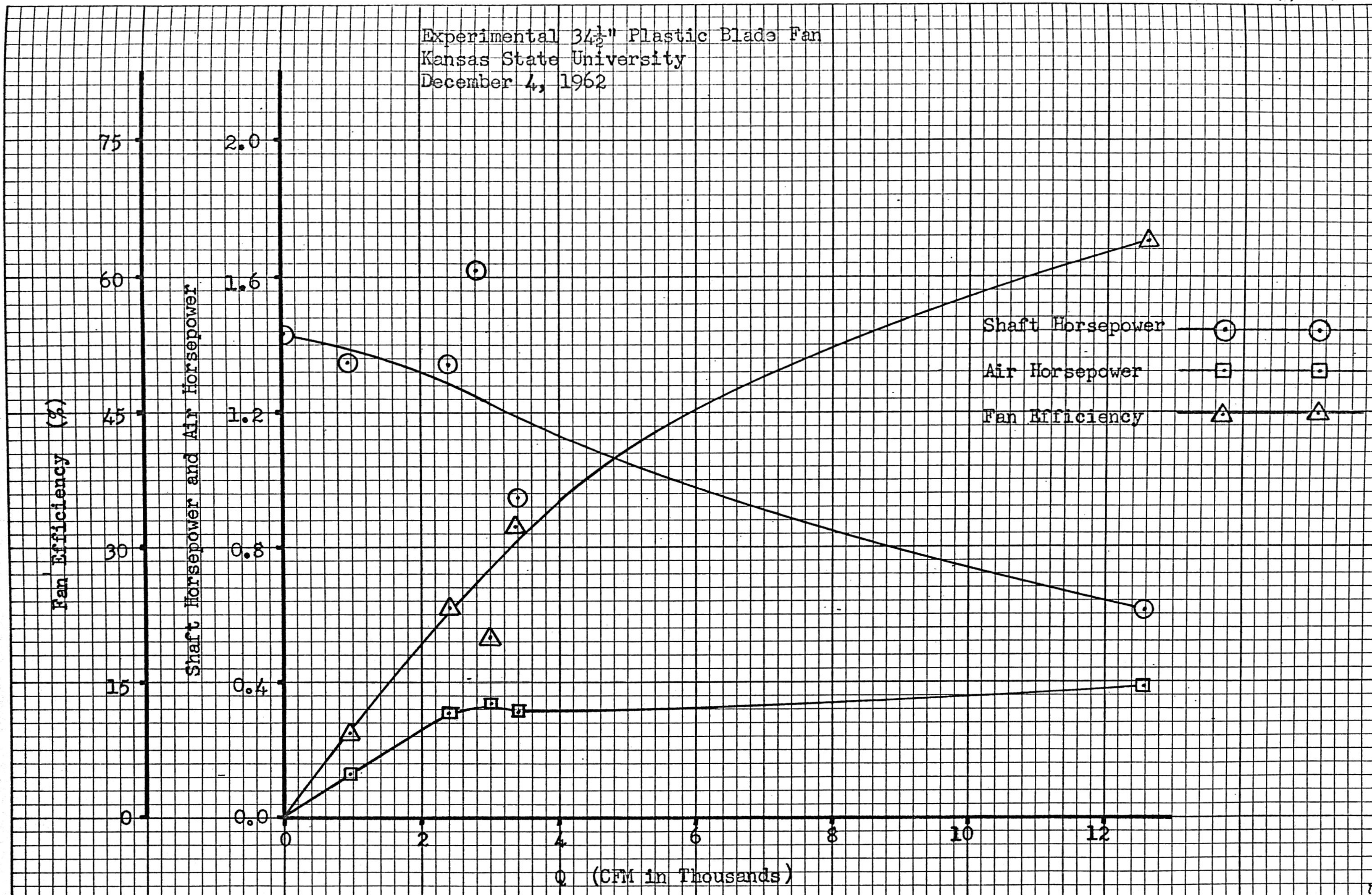


Fig. 6. Plastic blade fan characteristics at 1000 RPM

operating at 1000 revolutions per minute with open duct was compared to the Joy 1000 series Axivane Fan model 36-14-1150. This model is a 36-inch fan with a 14-inch hub operating at 1150 rpm.

Comparison:

Experimental fan	Brake Horsepower	0.608
	Air Horsepower	0.391
	Fan Efficiency	64.2%
Joy model 36-14-1150	Brake Horsepower	1.8
	Air Horsepower	1.11
	Fan Efficiency	61.8%

The calculations for the Joy fan were made using curves published by the Joy Manufacturing Company.[4]

The fan blade and hub design calculations were made for fan speeds up to 2000 revolutions per minute. The fan was not tested at speeds greater than 957 revolutions per minute because the fan had not been dynamically balanced, and a slight vibration was detected at this test speed.

The fan was tested primarily in the "stall region" for propeller type fans. The inlet area was so small the air flow was restricted to the point that the fan could not operate in the flow range for which this type fan is designed. Low flow rate and low efficiency are indicative of the "stall region" for propeller type fans.

The bonding strength between polyester resin and steel is acceptable for propeller type fans operating at speeds up to 957 revolutions per minute.

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[4] Joy Manufacturing Company, <u>Joy Axivane Catalogue</u>, Performance curve C-1252.

The test equipment and methods should be re-evaluated and re-designed.

Further testing after re-design should be done to give substantial proof of the fan performance.

ACKNOWLEDGMENT

The writer wishes to express his sincere appreciation to Dr. John C. Lindholm, Associate Professor of Mechanical Engineering, for guidance in the preparation and organization of this report.

BIBLIOGRAPHY

ASA Standard, B 1.1, 1949.

Faires, Virgil Moring. Design of Machine Elements, 3rd edition, 1955.

Joy Axivane Catalogue, Joy Manufacturing Company, 1961.

Machinery Handbook, 15th edition, 1957.

Marks, Lionel S. Mechanical Engineers' Handbook, 5th edition, Text Book edition, 1951.

Materials in Design Engineering, Materials Selector Issue, Mid-October, 1962.

APPENDICES

APPENDIX A

Fan Design Calculations

Fan Blade Shaft Design. The fan blade shaft was made from cold drawn SAE 1035 steel conforming to the following specifications.

Shaft length .... 2 and 9/16 inches.

Shaft diameter .... 5/8 inch.

Maximum of 7/8 inch of 5/8 - 11 UNC - 3A threads.

Tensile strength[5] .... 92,000 pounds per square inch.

Yield point[6] .... 78,200 pounds per square inch.

Maximum external permissible load on the fan blade shaft.

$$F_t = F_i + \left(\frac{k_{steel}}{k_{aluminum} + k_{steel}} \right) F_e$$

Where: F_t is the total load on the shaft (pounds).

F_i is the initial load due to tightening of the blade holding nut on the shaft (pounds).

F_e is the maximum external permissible load on the fan blade shaft (pounds).

$F_i = 16{,}000\ D$[7]

Where D is the shaft diameter (0.625 inches).

$F_i = 10{,}000$ pounds

$$s_i = \frac{F_i}{A_s}$$

s_i is the initial tensile stress in the shaft and A_s is the stress area. The stress area is that area corresponding to a

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5 Lionel S. Marks, op. cit., Table 17, p. 554.

6 Ibid., p. 554.

7 Virgil Moring Faires, Design of Machine Elements, p. 135.

diameter which is the average of the pitch and minor diameters for threads of class three tolerance.[8]

$$s_i = \frac{10,000}{0.2256} = 44,000 \text{ pounds per square inch}$$

Cross-sectional area of steel blade shaft.

$$A_{steel} = \frac{3.14(0.625)^2}{4} = 0.306 \text{ square inch}$$

Cross-sectional area of aluminum blade mount.

$$A_{aluminum} = (1.375 \times 1.375) - A_{steel} = 1.584 \text{ square inches}$$

$$k = \frac{(\text{cross-sectional area})(\text{modulus of elasticity})}{(\text{length})}$$

$$k_{aluminum} = \frac{(1.584)(10 \times 10^6)}{(1.875)} = 8.45 \times 10^6$$

$$k_{steel} = \frac{(.306)(30 \times 10^6)}{(1.875)} = 4.89 \times 10^6$$

The total load which would have the joint on the point of opening when the belt has been tightened to a value of F_i is F_o.

$$F_o = F_i \left(\frac{k_{steel} + k_{aluminum}}{k_{aluminum}} \right) = 10,000(1.58) = 15,800 \text{ pounds}$$

The maximum allowable load on the shaft must be less than F_o. The actual load on the shaft is given by:

$$F_{t_{max}} = F_i + \left(\frac{k_{steel}}{k_{aluminum} + k_{steel}} \right) F_e = 10,000 + (.366)F_e$$

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8 ASA Standard, B1.1 - 1949.

Assume a factor of safety of 1.5

Then[9] $\frac{1}{N} = \frac{s_m}{s_y} + \frac{K_f\ s_v}{s_n}$

s_y = yield stress = 78,200 pounds per square inch.

s_m = mean stress for mean load (F_m).

s_v = varying stress for varying load (F_v).

s_n = endurance stress[10] $= \frac{s_u}{2}(.7)(.85) = \frac{92,000}{2}(.7)(.85)$

$= 27,350$ pounds per square inch.

K_f = actual stress concentration factor[11] = 1.8

$$\frac{1}{1.5} = \frac{s_m}{78,200} + \frac{1.8\ s_v}{27,350}$$

$$52,200 = s_m + 5.15\ s_v$$

A_s = 0.2256 square inch

$$s_m = \frac{F_m}{A_s} \quad ; \quad s_v = \frac{F_v}{A_s}$$

$$11,750 = F_m + 5.15\ F_v$$

$$F_m = \frac{F_{t_{max}} + F_i}{2} \quad ; \quad F_v = \frac{F_{t_{max}} - F_i}{2}$$

F_i = 10,000 pounds

$$23,500 = 6.15\ F_{t_{max}} - 41,500$$

$$F_{t_{max}} = \frac{65,000}{6.15} = 10,580 \text{ pounds}$$

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9 Faires, loc. cit., p. 147.

10 Ibid., p. 146.

11 Ibid., p. 116.

$10,580 = 10,000 + .366\ F_e$

$F_e = \frac{580}{.366} = 1,585$ pounds

Maximum tensile force on fan blade shaft due to centrifugal force must not exceed the external force of 1,585 pounds. The center of gravity (blade and shaft) was experimentally determined to be 7.1875 inches from the threaded end of the shaft.

The weight of the blade and shaft was experimentally determined to be 1.405 pounds.

The weight of the steel shaft was experimentally determined to be 0.485 pound.

Centrifugal Force (CF) = $\left(\frac{W}{g}\right)\left(\frac{2\pi N}{60}\right)^2 X$

W is the weight of the rotating mass (pounds).

g is the gravitational constant (32.2 feet per second squared).

N is the rotational velocity (revolutions per minute).

X is the distance from center of rotation to center of gravity (feet).

$\frac{2\pi N}{60}$ is the angular velocity (radians per second).

X = 0.76 foot

W = 1.405 pounds

Then CF = $0.0331\ (0.1046\ N)^2$

Case A .... N is 500 RPM

CF is 90.6 pounds

Case B .... N is 1000 RPM

CF is 362 pounds

Case C .... N is 1500 RPM

CF is 813.9 pounds

Case D .... N is 2000 RPM

CF is 1,450 pounds

Case E .... N is 2500 RPM

CF is 2,160 pounds

2000 RPM was selected as the design speed.

Fan Blade Hub Design.

Total force on hub blade mount belts.

Volume of blade mount = Cross-sectional area x length

$= (1.375)(1.375) - [(.785)(.625)^2 + (.785)(.25)^2 (1.375)(4)] =$

2.537 cubic inches.

Weight of blade mount = 2.537 cubic inches x 0.093 pound per cubic inch = 0.236 pound.

Centrifugal force due to hub blade mount when fan is rotating at 2000 RPM.

The center of gravity of the blade mount was determined to be 3.75 inches from the center of rotation of the fan. C. G. = 0.313 foot.

$$CF = \frac{0.236}{32.2} \times \frac{(2 \times 3.14 \times 2000)^2}{(60)^2} \times 0.313 = 101.3 \text{ pounds}$$

The total force on the hub belts at 2000 RPM is the sum of the centrifugal force of the blade and blade holder.

$CF_{total} = 1.450 + 101.3 = 1551.3$ pounds

Calculations for bolt shear.

$$P = \frac{1551.3}{2} = 775.6 \text{ pounds}$$

$A = (4)(.785)(.25)^2 = 0.196$ square inch

$$S = \frac{P}{A} = \frac{775.6}{0.196} = 3{,}961 \text{ pounds per square inch}$$

Shear stress for SAE 3150 steel bolts is 113,000 pounds per square inch.(12)

Assume a factor of safety of 4.

Then the design stress $(s_d) = \frac{s_{shear}}{N} = \frac{113{,}000}{4} = 28{,}250$ pounds per square inch. The design stress is greater than the actual stress, therefore the steel bolts will not shear at 2000 RPM.

Failure due to crushing of hub disc. The compressive strength of Type 7178-T6 Aluminum is assumed to be one half of its tensile strength.(13)

$$s_{comp} = \frac{88{,}000}{2} = 44{,}000 \text{ pounds per square inch}$$

Assume a factor of safety of 4.

$$\text{Design stress } (s_d) = \frac{44{,}000}{4} = 11{,}000 \text{ pounds per square inch}$$

$P = 1{,}450$ pounds

$A = 8(.25 \times .1875) = .3750$ square inch

$$\text{Actual stress} = \frac{1{,}450}{.3750} = 3{,}870 \text{ pounds per square inch}$$

The design stress is greater than the actual stress, therefore the hub disc will not fail due to crushing when the fan is operating at a speed of 2000 RPM.

Failure of hub blade mount due to crushing.

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12 Faires, op. cit., p. 34 and 35.

13 Materials in Design Engineering, op. cit., p. 108.

The compressive yield strength of Type 218 Aluminum Alloy is 27,000 pounds per square inch.[14]

Assume a factor of safety of 4.

$$\text{Design stress due to crushing} = \frac{27,000}{4} = 6,750 \text{ pounds}$$

$$\text{Actual stress} = \frac{1,450}{4(.25 \times 1.375)} = 1,055 \text{ pounds per square inch.}$$

The design stress is greater than the actual stress, therefore the blade mount will not fail due to crushing when the fan is operating at a speed of 2000 RPM.

Hub Design

The size set-screw required to hold the hub center on the three-quarter inch shaft when the fan is operating at 2000 RPM was determined as follows:

Assume that the power supplied to the fan shaft will be five horsepower. This is the rated horsepower of the dynamometer. The torque required to hold the fan on the three-quarter inch shaft is:

$$T = \frac{63,000 \times HP}{n} = \frac{63,000 \times 5}{2000} = 157.5 \text{ inch pounds}$$

If one set-screw is used to hold the fan on the shaft, the set-screw size must be determined from the following equation.[15]

$$(d)^{2.3} = \frac{T}{1250\ D} = \frac{157.5}{1250(.75)} = .46 \text{ inch}$$

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[14] Materials in Design Engineering, op. cit., p. 112.

[15] Machinery Handbook, 15th edition, p. 976.

Therefore a one-half inch diameter set-screw should be used.

It was arbitrarily decided to use five-sixteenths inch set-screws.

$$P = \frac{1 \times 1000}{50}(.3125)^{2.3} = 20(.069) = 1.38$$

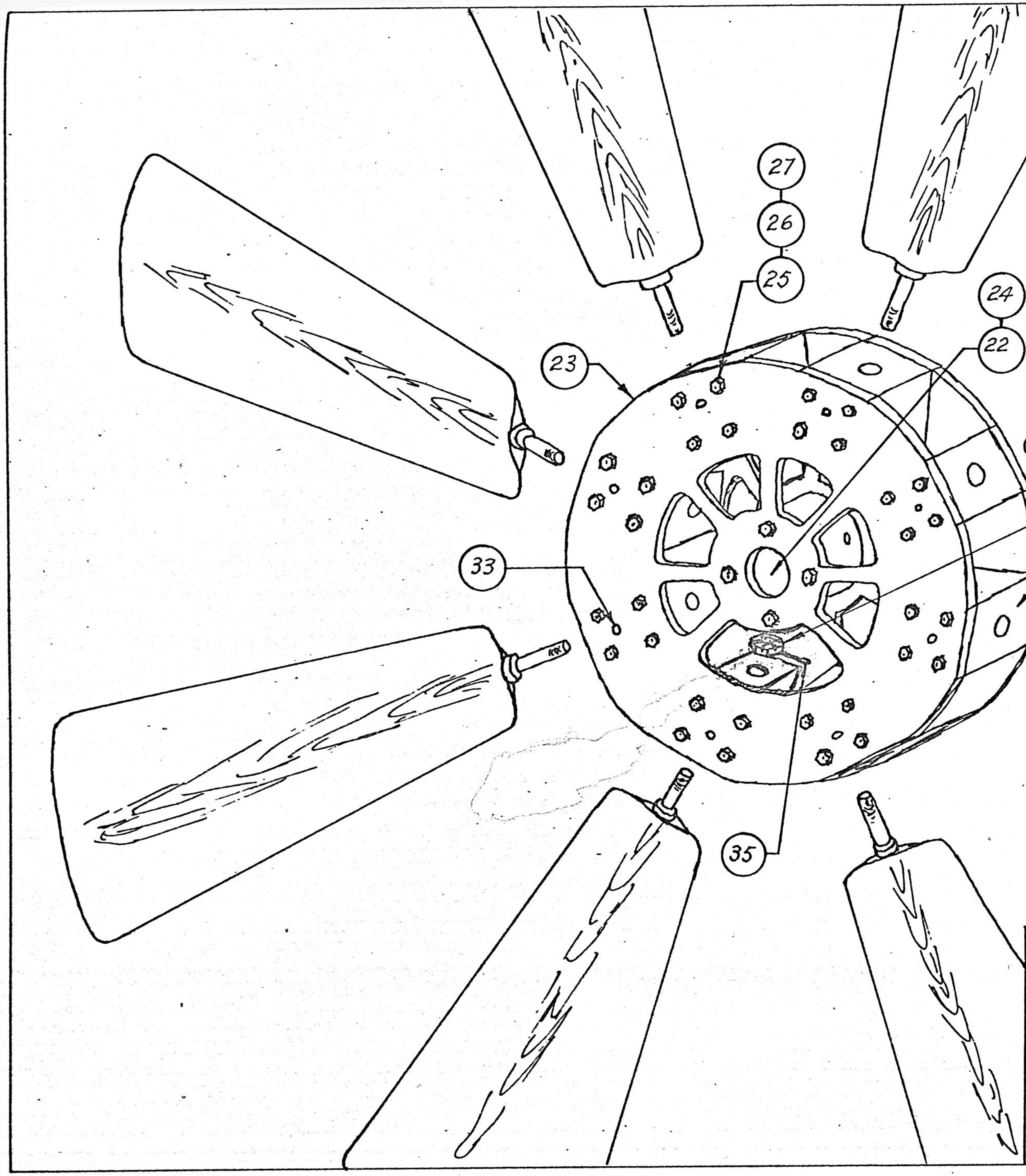
$\frac{5}{1.38} = 3.63$; therefore four five-sixteenth inch diameter set-screws must be used in the hub center to hold the fan when it is operating at 2000 RPM.

APPENDIX B

Part No.	No. Req'd.	Part Name	Material	Next Assem.
35	8	Cotterpin	Steel $\frac{5}{32} \times 1\frac{1}{2}$ Extended Mitre End	30
34	8	Blade Holding Nut	Steel Regular Hexagonal Slotted Nut $\frac{5}{8}$ - 11UNC-2A	30
33	8	Blade Set-Screw	Steel Hexagonal Socket Type Cone Point - $\frac{5}{16}$" Length #10-24UNC-2A	30
32	8	Blade Shaft	SAE 1035 Steel Rod	31
31	8	Blade	Polyester Resin and Fiberglass Cloth	30
27	36	Lock Washer	Carbon Spring Steel $\frac{1}{4}$ inch Medium	30
26	36	Nut	Steel Regular Semi-Finished Hexagonal Head $\frac{1}{4}$ - 20UNC-2A	30

Part No.	No. Req'd	Part Name	Material	Next Assem.
25	36	Bolt	Steel $\frac{1}{4} \times 2$ Machine Hexagonal Head $\frac{1}{4}$ - 20 UNC-2A	30
24	4	Hub Set-Screw	Steel Hexagonal Socket Type Cup Point - $\frac{1}{2}$" Length $\frac{5}{16}$ - 24 UNF-2A	30
23	2	Hub Disc	Wrought Aluminum Alloy Type 7178-T6	30
22	1	Hub Center	SAE 1035 Steel Rod	30
21	8	Hub Blade Mount	Cast Aluminum Alloy Type 218	30

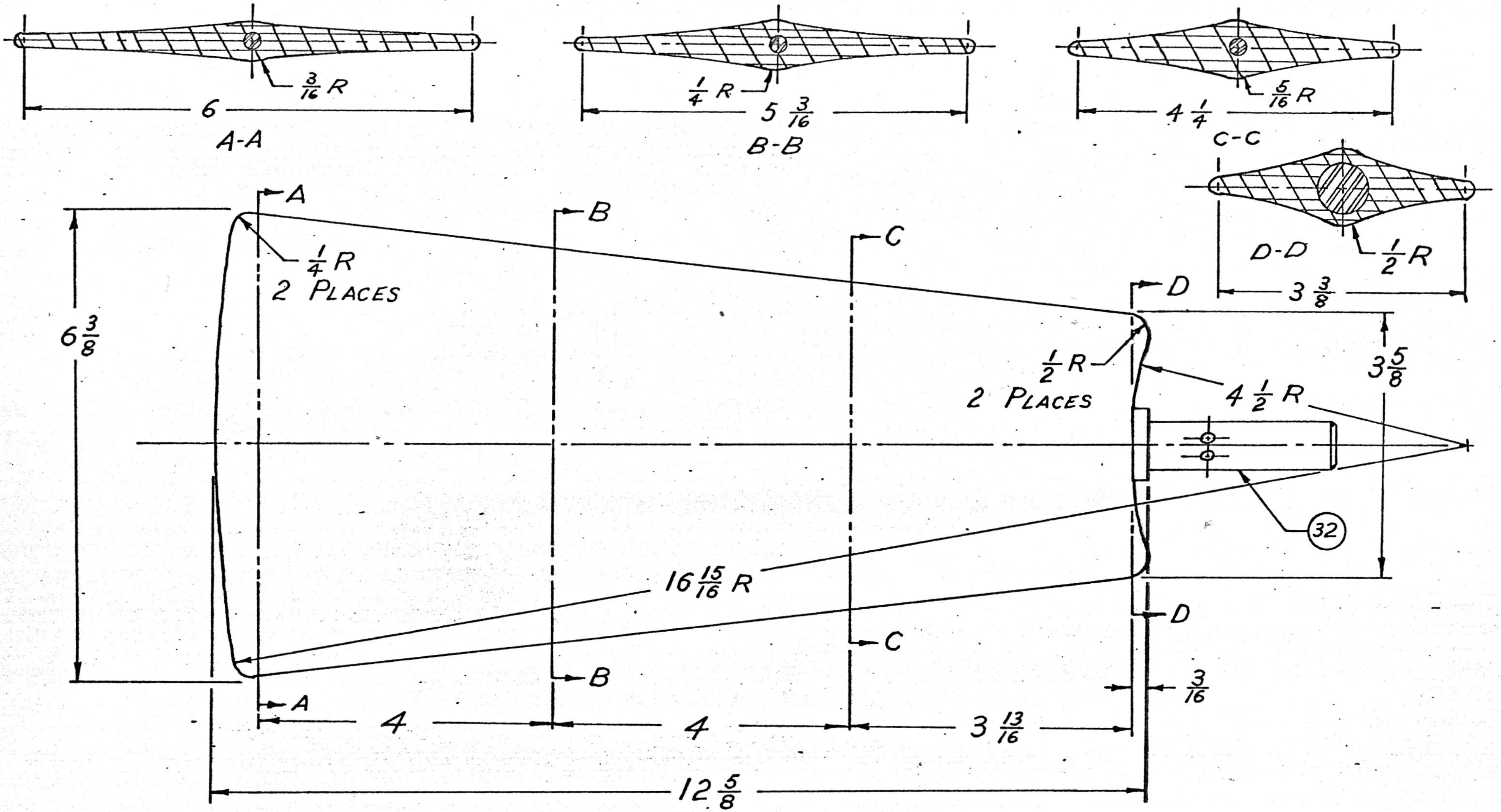
REVERSIBLE BLADE FAN		
PARTS LIST		
DFTMN: H.M.N. 9/22/62	DATE: 9-22-62	NEXT ASSEM. —
APPROV: H.M.N. 9/22/62	SCALE: NONE	DWG. 20



REVERSIBLE BLADE FAN

FAN ASSEMBLY

DFTMN: H.M.N. 9/20/62	DATE: 9-20-62	NEXT ASSEM. ~
APPROV: H.M.N. 9/21/62	SCALE NONE	DWG 30



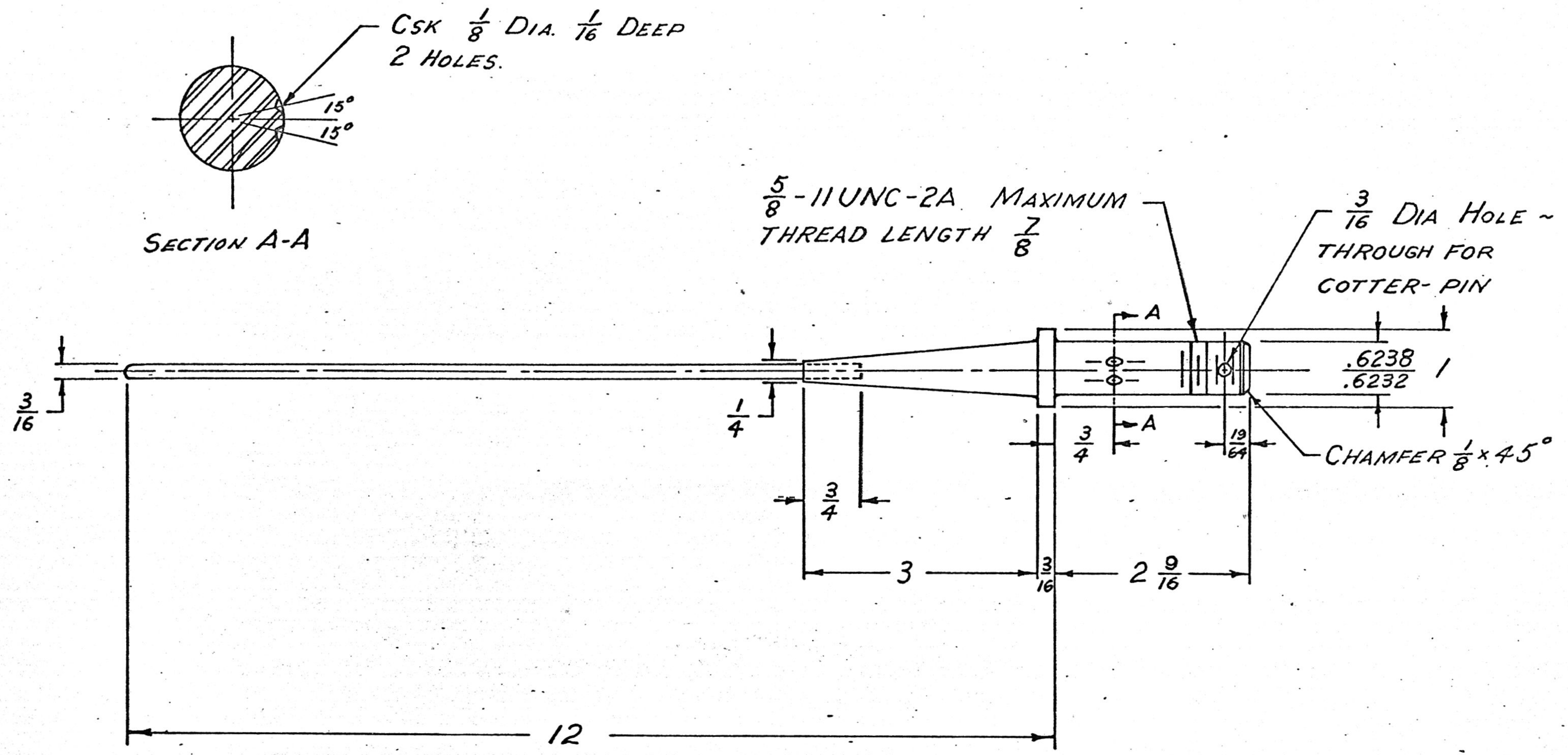
NOTE: 1- TOLERANCE UNLESS OTHERWISE NOTED:
FRACTIONAL $\pm\frac{1}{64}$; DECIMAL $\pm$0.005; ANGULAR $\pm 1°$

2- BLADE MADE FROM POLYESTER RESIN WITH ONE LAYER OF FIBERGLASS CLOTH IN CENTER OF BLADE.

3- MOLD SHAFT AND BLADE WITH SET-SCREW HOLES IN POSITION AS SHOWN.

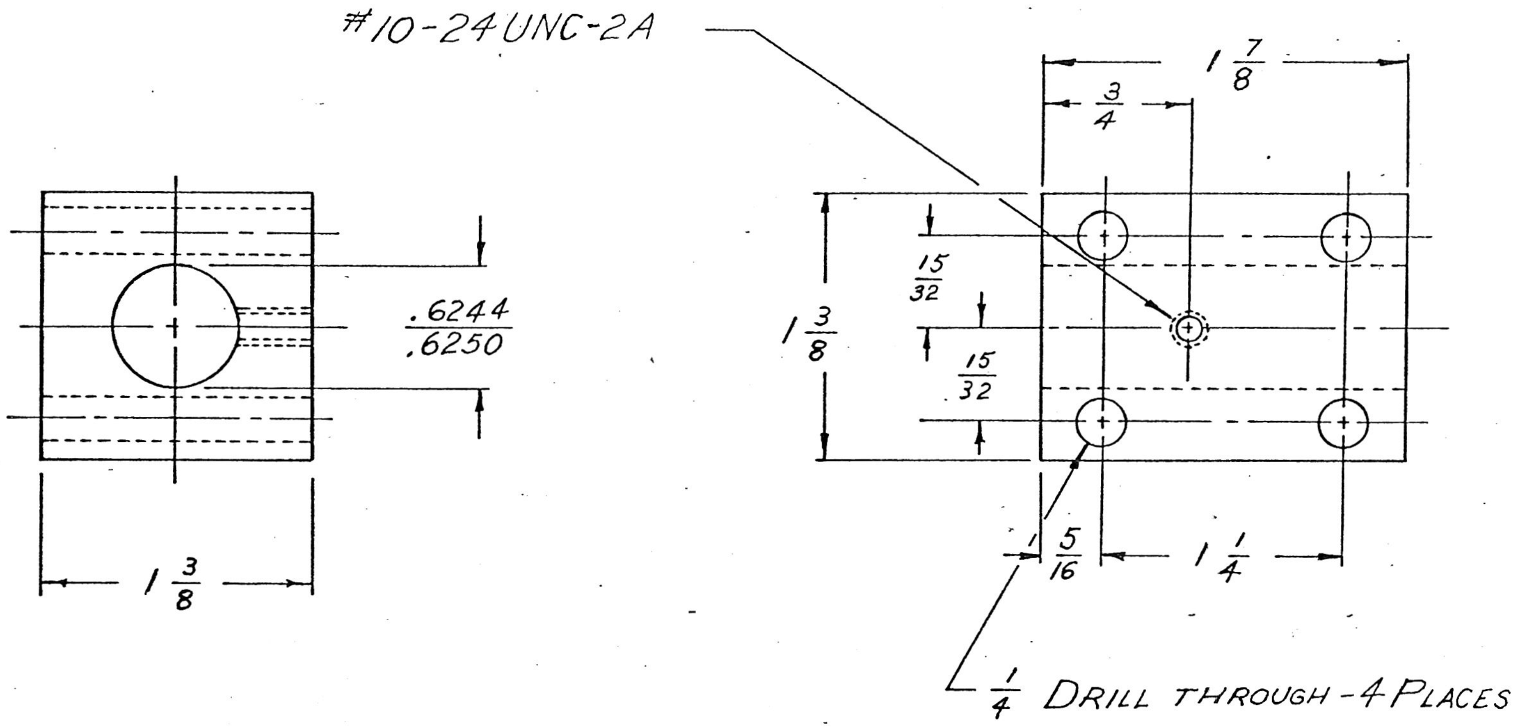
4- BLADE EDGE $\frac{3}{16}$ DIA. TYPICAL.

REVERSIBLE BLADE FAN		
BLADE DETAIL		
DFTMN: H.M.N. 8/21/62	DATE: 8-20-62	NEXT ASSEM. 30
APPROV: H.M.N. 8/24/62	SCALE: $\frac{1}{2}$" = 0'-1"	DWG. 31



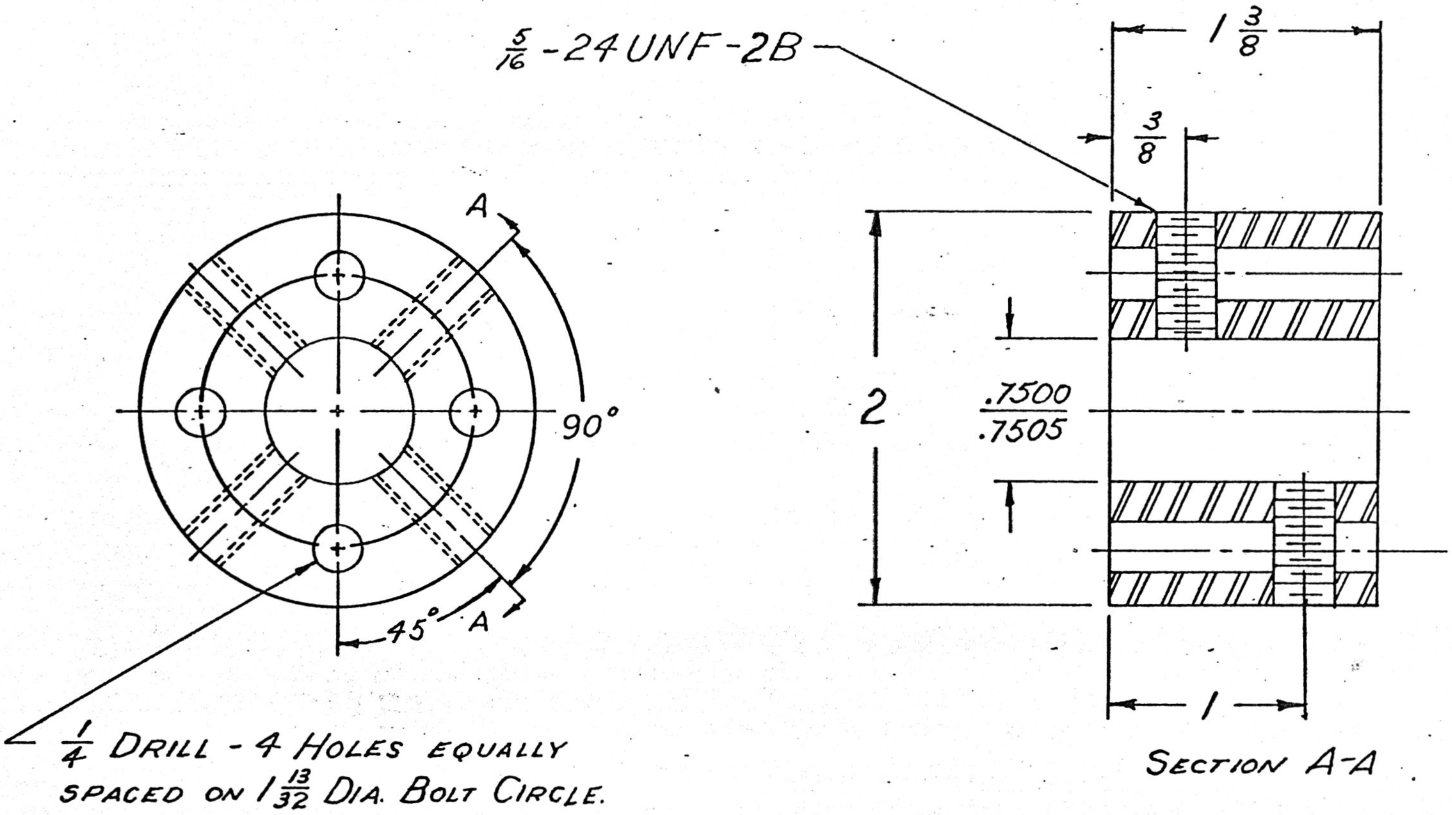
NOTE: 1- TOLERANCE UNLESS OTHERWISE NOTED:
FRACTIONAL $\pm\frac{1}{64}$; DECIMAL $\pm$ 0.005; ANGULAR $\pm$ 1°
2- SHANK MADE FROM SAE 1035 STEEL BAR.
3- STIFFENER ROD MADE FROM SAE 1035 STEEL ROD.
4- SILVER SOLDER SHAFT SHANK AND STIFFENER.
5- FILLETS $\frac{1}{16}$ R, ROUNDS $\frac{1}{16}$ R UNLESS OTHERWISE NOTED.

REVERSIBLE BLADE FAN		
BLADE SHAFT DETAIL		
DFTMN: H.M.N. 8/21/62	DATE: 8-21-62	NEXT ASSEM. 31
APPROV: H.M.N. 8/21/62	SCALE: $\frac{1}{2}$" = 0'-1"	DWG. 32



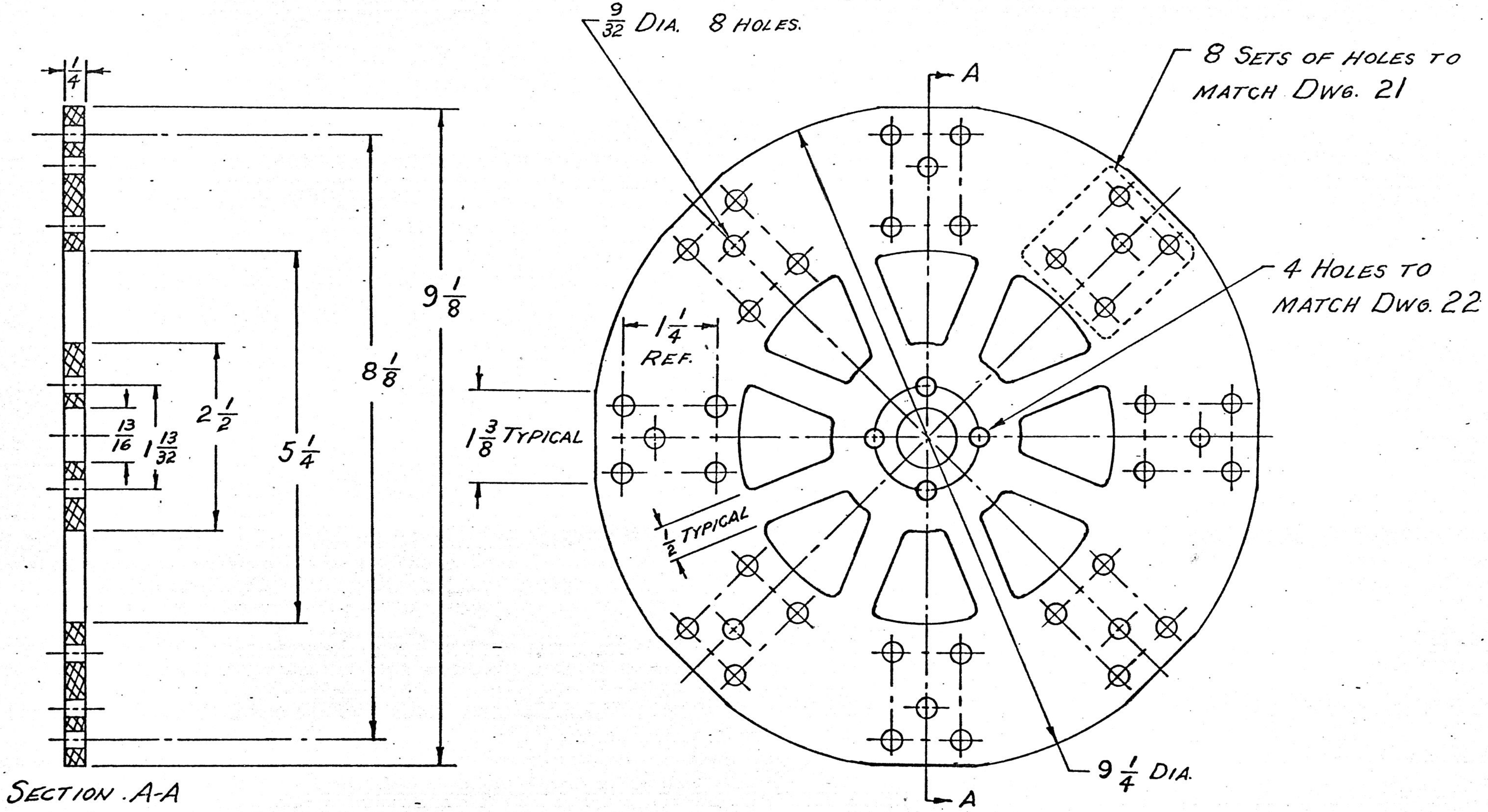
NOTE: 1- TOLERANCE UNLESS OTHERWISE SPECIFIED:
FRACTIONAL $\pm\frac{1}{64}$; DECIMAL $\pm$0.005; ANGULAR $\pm 1°$
2- MAKE HUB BLADE MOUNT FROM - CAST ALUMINUM ALLOY TYPE 218
3- 8 BLADE MOUNTS REQUIRED.

REVERSIBLE BLADE FAN		
HUB BLADE MOUNT		
DFTMN: H.M.N. 9/10/62	DATE: 9-10-62	NEXT ASSEM. 30
APPROV: H.M.N. 9/12/62	SCALE: 1" = 0'-1"	DWG. 21



NOTE: 1- TOLERANCE UNLESS OTHERWISE SPECIFIED:
FRACTIONAL $\pm\frac{1}{64}$; DECIMAL $\pm$.005; ANGULAR $\pm 1°$
2- MAKE HUB CENTER FROM SAE 1035 STEEL ROD.
3- ONE CENTER REQUIRED.

REVERSIBLE BLADE FAN		
HUB CENTER		
DFTMN: H.M.N. 9/12/62	DATE: 9-12-62	NEXT ASSEM. 30
APPROV: H.M.N. 9/12/62	SCALE: 1" = 0'-1"	DWG 22



SECTION A-A

NOTE: 1- TOLERANCE UNLESS OTHERWISE SPECIFIED:
FRACTIONAL $\pm\frac{1}{64}$; DECIMAL $\pm$ 0.005; ANGULAR $\pm 1^\circ$
2- MAKE HUB DISC FROM - WROUGHT ALUMINUM ALLOY TYPE 7178-T6
3- TWO DISC REQUIRED.
4- FILLETS $\frac{1}{16}$R, ROUNDS $\frac{1}{16}$R UNLESS OTHERWISE NOTED.

REVERSIBLE BLADE FAN		
HUB DISC		
DFTMN: H.M.N. 9/15/62	DATE: 9-15-62	NEXT ASSEM. 30
APPROV: H.M.N. 9/17/62	SCALE: $\frac{1}{2}$" = 0'-1"	DWG. 23

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APPENDIX C

Nomenclature and Symbols

A Air inlet area (square feet).

C Coefficient of discharge for Standard ASME nozzles used in this test (dimensionless).

C_1 Coefficient of discharge of duct (assumed to be 1.0 for the span duct, dimensionless).

D Diameter of duct (feet).

d Air density at point of pressure measurement and at dry bulb temperature (pounds per cubic foot).

d_a Standard air density (0.075 pound per cubic foot).

e_f Fan efficiency. The ratio of the static air horsepower to the shaft horsepower (per cent).

F Scale beam reading (pounds).

h Velocity head (feet of air).

Air HP Horsepower imparted to the air by the fan.

K_1 Standard air conditions correction factor for pressures and horsepower when speed is constant (dimensionless).

K_2 Standard air conditions correction factor for volume (dimensionless).

K_3 Standard air conditions correction factor for pressures (dimensionless).

K_4 Standard air conditions correction factor for horsepower (dimensionless).

L Length of duct (feet).

l Brake arm length (feet).

P_a Standard barometric pressure (pounds per square inch).

P_s Static air pressure at standard conditions (inches of water).

P_t Total air pressure at standard conditions (inches of water).

P_v Air velocity pressure at standard conditions (inches of water).

P'_v Vapor pressure at standard conditions (pounds per square inch).

p_s Average of static pressure readings (inches of water).

p_t Average of total pressure readings (inches of water).

p_v Average of velocity pressure readings (inches of water).

p'_s Average static pressure reading corrected for air density (inches of water).

p'_t Average total pressure reading corrected for air density (inches of water).

p'_v Average velocity pressure reading corrected for air density (inches of water).

P_w Pressure of saturated vapor at wet-bulb temperature (pounds per square inch).

Q_{RPMx} Flow rate at standard conditions and actual RPM (cubic feet per minute).

Q_{RPM} Flow rate at standard conditions and standard RPM (cubic feet per minute).

R Gas constant for air (0.370 for pressure in pounds per square inch).

RPM Standard revolutions per minute. Arbitrarily chosen for each test series.

RPM_x Actual revolutions per minute for each test run.

SHP_{RPM} Fan shaft horsepower at standard revolutions of fan.

SHP_{RPMx} Fan shaft horsepower at actual average revolutions of fan.

T_d Dry bulb temperature (degrees Rankine).

t_d Dry bulb temperature (degrees Fahrenheit).

t_w Wet bulb temperature (degrees Fahrenheit).

V_{RPM} Average air velocity at standard RPM (feet per second).

V_{RPMx} Average air velocity at actual RPM (feet per second).

Formulae

P_a = P (actual reading) $\pm$ Temperature Correction $\pm$ Gravity Correction $\pm$ Elevation Correction

$$P_v' = P_w - \left[\frac{P_a}{30} \times \left(\frac{t_d - t_w}{90}\right)\right]$$

$$d = \frac{P_a - 0.38\ P_v'}{RT_d}$$

$$K_1 = \left(\frac{d_a}{d}\right)$$

$$p_s' = K_1 p_s \ ; \quad p_t' = K_1 p_t \ ; \quad p_v' = K_1 p_v$$

$$K_2 = \left(\frac{RPM}{RPM_x}\right) \ ; \quad K_3 = \left(\frac{RPM}{RPM_x}\right)^2 \ ; \quad K_4 = \left(\frac{RPM}{RPM_x}\right)^3$$

$$h = p_v' \left(\frac{62.4}{12\ d}\right)$$

$$V_{RPM_x} = C\ (8.02)\ \sqrt{h} \ ; \quad V_{RPM} = V_{RPM_x} K_2$$

$$A = \frac{\pi D^2}{4}$$

$$Q_{RPM} = (60)(A)V_{RPM}$$

$$\text{Air HP} = 15.73 \times 10^{-5}\, Q\, P_s$$

$$e_f = \frac{\text{Air HP}}{\text{SHP}}$$

Sample Calculations

The sample calculations are for Test Number 3-7".

$$P'_v = .2749 - \left[\frac{14.39}{30} \times \left(\frac{71 - 62}{90}\right)\right] = .2270 \text{ pounds per square inch}$$

$$d = \frac{14.39 - 0.38(.2270)}{.37(521)} = .0740 \text{ pounds per cubic foot}$$

$$h = .490 \left[\frac{62.4}{12(.0740)}\right] = 34.4 \; ; \sqrt{h} = 5.87 \text{ feet of air}$$

$$K_1 = \left(\frac{.0750}{.0740}\right) = 1.012 \; ; K_2 = \left(\frac{750}{763}\right) = .984 \; ; K_3 = \left(\frac{750}{763}\right)^2 = .967 \; ;$$

$$K_4 = \left(\frac{750}{763}\right)^3 = .951$$

$p'_s = -.4820 \times 1.012 = -.4880$ inch of water; $p'_t = .0006 \times 1.012 =$ .006 inch of water.

$p'_v = .4830 \times 1.012 = .4900$ inch of water

$$SHP_{RPM_x} = \frac{2(3.14)(1.73)(2.62)(763)}{33{,}000} = .329 \times 10^{-3} \times 1997 = .6560$$

$$SHP_{RPM} = .6560 \times .951 = .6240$$

$V_{RPM_x} = (.99)(8.02)(5.87) = 46.6$ feet per second

$V_{RPM} = (46.6)(.984) = 45.9$ feet per second

$P_s = -.488 \times .967 = -.473$ inch of water

$P_t = .0006 \times .967 = .0006$ inch of water

$P_v = .490 \times .967 = .4724$ inch of water

$$A = \frac{(3.14)(.584)^2}{4} = .266 \text{ square foot}$$

$$Q_{RPM} = (60)(.266)(45.9) = 732 \text{ cubic feet per minute}$$

$$\text{Air HP} = 15.73 \times 10^{-5} (.732 \times 10^{-3})(4.73 \times 10^{-1}) = .0545$$

$$e_f = \frac{.0545}{.624} \times 100 = 8.74 \text{ per cent}$$

Table 1. Tabulated test data and calculations.

Test Number	l	$\frac{2\pi l}{33,000}$	F	RPM_x	SHP_{RPM_x}	RPM	K_4	SHP_{RPM}	d
1-Closed	1.73	$.329 \times 10^{-3}$	.853	263	.0737	250	.860	.0634	.0737
1-7"	1.73	$.329 \times 10^{-3}$	.782	288	.0740	250	.656	.0485	.0730
1-10"	1.73	$.329 \times 10^{-3}$	.625	300	.0617	250	.580	.0358	.0734
1-14"	1.73	$.329 \times 10^{-3}$	.625	299	.0615	250	.585	.0360	.0739
1-17"	1.73	$.329 \times 10^{-3}$	.625	297	.0610	250	.597	.0364	.0746
1-Open	1.73	$.329 \times 10^{-3}$	.625	347	.0748	250	.374	.0280	.0740
2-Closed	1.73	$.329 \times 10^{-3}$	1.75	560	.3230	500	.711	.2295	.0740
2-7"	1.73	$.329 \times 10^{-3}$	1.78	608	.3560	500	.556	.1980	.0730
2-10"	1.73	$.329 \times 10^{-3}$	1.50	618	.3045	500	.531	.1618	.0734
2-14"	1.73	$.329 \times 10^{-3}$	1.44	611	.2995	500	.548	.1640	.0734
2-17"	1.73	$.329 \times 10^{-3}$	1.38	616	.2790	500	.535	.1492	.0739
2-Open	1.73	$.329 \times 10^{-3}$	1.00	578	.1902	500	.647	.1231	.0729
3-Closed	1.73	$.329 \times 10^{-3}$	2.68	740	.6460	750	1.042	.6730	.0740
3-7"	1.73	$.329 \times 10^{-3}$	2.62	763	.6560	750	.951	.6240	.0740
3-10"	1.73	$.329 \times 10^{-3}$	2.13	739	.5160	750	1.049	.5411	.0734
3-14"	1.73	$.329 \times 10^{-3}$	2.00	768	.5060	750	.933	.4730	.0734
3-17"	1.73	$.329 \times 10^{-3}$	1.75	739	.4260	750	1.049	.4470	.0734
3-Open	1.73	$.329 \times 10^{-3}$	1.28	753	.3165	750	.991	.3140	.0727
4-Closed	1.73	$.329 \times 10^{-3}$	3.91	942	1.2100	1000	1.198	1.4500	.0740
4-7"	1.73	$.329 \times 10^{-3}$	3.44	910	1.0300	1000	1.330	1.3700	.0730
4-10"	1.73	$.329 \times 10^{-3}$	2.75	816	.7380	1000	1.840	1.3600	.0734
4-14"	1.73	$.329 \times 10^{-3}$	3.77	873	1.0820	1000	1.502	1.6260	.0737
4-17"	1.73	$.329 \times 10^{-3}$	2.25	890	.6580	1000	1.420	.9350	.0737
4-Open	1.73	$.329 \times 10^{-3}$	1.69	957	.5320	1000	1.144	.6080	.0721

Table 1 (cont.)

Test Number	p_v	p_s	p_t	K_1	p'_v	p'_s	p'_t	V_{RPM}	A	Q_{RPM}	K_3
1-Closed	0	-.0607	-.0607	1.019	0	-.0619	-.0619	0	0	0	.904
1-7"	.0610	-.0590	.0007	1.028	.0627	-.0606	.0007	15.95	.266	255	.755
1-10"	.0532	-.0527	.0005	1.022	.0544	-.0540	.0005	13.55	.654	532	.695
1-14"	.0526	-.0489	.0037	1.016	.0535	-.0497	.0038	12.88	.916	709	.699
1-17"	.0414	-.0398	.0016	1.006	.0414	-.0398	.0016	11.32	1.11	755	.708
1-Open	.0205	-.0180	.0022	1.014	.0208	-.0183	.0022	6.99	7.06	2,960	.519
2-Closed	0	-.2770	-.2770	1.014	0	-.2810	-.2810	0	0	0	.786
2-7"	.2950	-.2940	.0003	1.028	.3030	-.3020	.0031	30.30	.266	484	.676
2-10"	.2647	-.2643	.0004	1.022	.2710	-.2700	.0004	28.20	.654	1,106	.656
2-14"	.2438	-.2430	.0006	1.022	.2490	-.2480	.0006	27.30	.916	1,500	.670
2-17"	.2114	-.2107	.0009	1.016	.2140	-.2130	.0009	25.00	1.11	1,667	.659
2-Open	.0630	-.0600	.0030	1.030	.0649	-.0618	.0003	14.92	7.06	6,320	.748
3-Closed	0	-.4750	-.4750	1.014	0	-.4820	-.4820	0	0	0	1.028
3-7"	.4830	-.4820	.0006	1.014	.4900	-.4880	.0006	45.90	.266	732	.967
3-10"	.4048	-.4034	.0014	1.022	.4120	-.4120	.0016	43.75	.654	1,715	1.032
3-14"	.4006	-.3997	.0009	1.022	.4100	-.4080	.0009	41.80	.916	2,300	.955
3-17"	.3069	-.3064	.0005	1.022	.3140	-.3139	.0005	38.05	1.11	2,540	1.032
3-Open	.1090	-.1020	.0070	1.031	.1125	-.1050	.0072	22.70	7.06	9,620	.994
4-Closed	0	-.7740	-.7740	1.014	0	-.7850	-.7850	0	0	0	1.128
4-7"	.6660	-.6630	.0028	1.028	.6820	-.6780	.0029	60.80	.266	972	1.210
4-10"	.5485	-.5475	.0010	1.022	.5610	-.5600	.0010	61.30	.654	2,400	1.504
4-14"	.5164	-.5128	.0036	1.018	.5260	-.5210	.0037	55.40	.916	3,050	1.311
4-17"	.4510	-.4490	.0022	1.018	.4590	-.4570	.0022	50.80	1.11	3.390	1.264
4-Open	.1770	-.1730	.0037	1.040	.1840	-.1800	.0039	30.20	7.06	12,600	1.094

Table 1 (cont.)

Test Number	P_t	P_v	P_s	P_w	P_a	t_d	t_w	P_v'	h	$\sqrt{h}$		Air HP
1-Closed	-.0558	0	-.0558	.2302	14.38	67	57	.1770	0	0	0	0
1-7"	.0005	.0452	-.0457	.2749	14.39	71	62	.2270	4.46	2.110	16.75	.0018
1-10"	.0003	.0372	-.0375	.1716	14.29	63	49	.1243	3.86	1.964	15.60	.0031
1-14"	.0026	.0272	-.0298	.1652	14.29	62	48	.0909	3.77	1.940	15.40	.0033
1-17"	.0011	.0271	-.0282	.1588	14.29	57	45	.1152	2.88	1.698	13.45	.0034
1-Open	.0011	.0084	-.0095	.2219	14.38	64	56	.1793	1.46	1.210	9.70	.0043
2-Closed	-.2210	0	-.2210	.2219	14.38	64	56	.1793	0	0	0	0
2-7"	.0021	.2019	-.2040	.2749	14.39	71	62	.2270	21.60	4.650	36.82	.0155
2-10"	.0003	.1767	-.1770	.1716	14.29	63	49	.1243	19.20	4.380	34.80	.0308
2-14"	.0004	.1656	-.1660	.1716	14.29	63	49	.1243	17.65	4.200	33.35	.0392
2-17"	.0006	.1396	-.1402	.1592	14.29	62	47	.0797	15.08	3.880	30.80	.0368
2-Open	.0002	.0460	-.0463	.2561	14.38	72	60	.1923	4.64	2.152	17.25	.0460
3-Closed	-.4950	0	-.4950	.2219	14.38	64	56	.1793	0	0	0	0
3-7"	.0006	.4724	-.4730	.2749	14.39	71	62	.2270	34.40	5.870	46.60	.0545
3-10"	.0016	.4234	-.4250	.1716	14.29	63	49	.1243	29.40	5.420	43.10	.1150
3-14"	.0009	.3891	-.3900	.1716	14.29	63	49	.1243	29.10	5.390	42.75	.1410
3-17"	.0005	.3235	-.3240	.1592	14.29	63	47	.0744	22.30	4.720	37.50	.1300
3-Open	.0072	.0971	-.1043	.2660	14.38	73	61	.2022	8.05	2.840	22.79	.1580
4-Closed	-.8850	0	-.8850	.2219	14.38	64	56	.1793	0	0	0	0
4-7"	.0035	.8165	-.8200	.2749	14.39	71	62	.2270	48.60	6.970	55.30	.1250
4-10"	.0015	.8405	-.8420	.1716	14.29	63	49	.1081	39.80	6.310	50.00	.3180
4-14"	.0049	.6791	-.6840	.1780	14.29	64	50	.1037	37.10	6.090	48.40	.3280
4-17"	.0028	.5712	-.5740	.1650	14.29	64	48	.0804	32.40	5.690	45.20	.3070
4-Open	.0043	.1927	-.1970	.2749	14.38	74	62	.2111	13.26	3.640	28.90	.3910

Table 1 (concl.)

Test Number	Fan Efficiency
1-Closed	0
1-7"	3.78%
1-10"	8.78%
1-14"	9.24%
1-17"	9.21%
1-Open	15,80%
2-Closed	0
2-7"	7.85%
2-10"	19.05%
2-14"	23.90%
2-17"	24.70%
2-Open	37.30%
3-Closed	0
3-7"	8.74%
3-10"	21.20%
3-14"	30.00%
3-17"	29.00%
3-Open	50.30%
4-Closed	0
4-7"	9.15%
4-10"	23.40%
4-14"	20.20%
4-17"	32.80%
4-Open	64.20%

DESIGN AND DEVELOPMENT OF A REVERSIBLE PLASTIC BLADE FAN

by

HENRY MASON NEELY, JR.

B. S., Kansas State University, 1956

AN ABSTRACT OF A MASTER'S REPORT

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1963

The axial flow, or propeller, type fan is best adapted to handling large volumes of fluid against low resistance. In its simpler forms, the propeller type fan is most often used for ventilation service, such as being mounted in an opening in a wall, with no duct work.

Most propeller fan blades are made of metal, and are inherently of the backwardly curved blade type.

The purpose of this work was to design a symmetrical plastic fan blade, assemble it on a fan hub also designed for this particular blade, then test the fan in order to determine its performance.

The reversible plastic blade fan designed and tested is a thirty-four and one-half inch diameter, eight blade propeller type fan.

Each blade is twelve and five-eights inches long, three and five-eighths inches wide at the base, and six and three-eighths inches wide at the tip. Each blade is made of Polyester Resin reinforced with one layer of Fiberglass cloth interwoven on a steel support rod in the center of the blade and extending to the edges of the blade. Each blade is symmetrical in front and back, with the longitudinal center thicker than the blade edges. The blades were designed symmetrical in order to be reversible so that the direction of fluid flow could be reversed without changing the direction of rotation of the motor driving the fan.

Each blade was made by forcing the viscous polyester resin into a mold under two to five pounds pressure.

Before casting the blade, the mold surfaces were coated with grease. A sheet of aluminum foil was placed in each half of the mold and smoothed to conform to the blade contour. The metal blade shaft with the fiberglass cloth interwoven was placed in one half of the mold. A synthetic clay was placed around the edge of one half of the mold as a sealant, and the mold halves were bolted together.

The resin was forced into the mold and allowed to harden and cure for twelve to sixteen hours. This curing time was required to be sure the blade did not warp when removed from the mold.

After the blade was taken from the mold, the aluminum foil was removed, the blade trimmed of excess fiberglass, gently sanded to remove any small imperfections, and painted with a finish coat of polyester resin. After the finish coat of resin had cured, the blade was ready to install in the fan hub.

The fan hub consists of three basic parts: the hub blade mounts, the hub center, and the hub mounting disc. The hub is nine and one-quarter inches in diameter and one and seven-eighths inches wide. The hub blade mounts are made from Type 218 Cast Aluminum. The mounting discs are made from Type 7178-6T Wrought Aluminum. The hub center is made from SAE 1035 steel.

All parts of the fan hub were designed to withstand stresses produced by the fan when rotating at 2000 RPM.

The plastic blade fan was assembled with the blade pitch angle at fifteen degrees.

The fan was placed in the outlet end of the test duct, and the inlet duct area was varied during the testing by using a series of ASME nozzles.

From the test results, the following conclusions were made.

The fan was tested primarily in the "stall region" for propeller type fans. The inlet area was so small that the air flow was restricted to the point that the fan could not operate in the air flow range for which this type fan is designed.

When tested with the inlet duct area the same as the outlet duct area, the fan was within its operational region and performed favorably to similar fans on the market today.

The test equipment and methods should be re-evaluated and re-designed.

Further testing after re-design of the test methods should be done to give substantial proof of the fan's performance.