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AERODYNAMIC TESTING OF AN AIRBORNE LIGHTWEIGHT HIGH-EFFICIENCY RADIAL FAN

By

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March 1967

U. S. ARMY AVIATION MATERIEL LABORATORIES FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-454(T)

AEROPHYSICS COMPANY

WASHINGTON, D. C.

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SUMMARY

Under two previous contracts, the Aerophysics Company demonstrated the feasibility of lightweight rotating-diffuser centrifugal fans for internal-flow airborne applications. Aerodynamic efficiency was proved by means of system model tests with the use of a 20-inch-diameter fan. Lightweight fabrication was demonstrated in a program in which two 66-inch overall diameter lightweight fans were built and extensively tested structurally.

The study reported herein represents the last step in the RD fan feasibility program, i. e., the aerodynamic testing of the 66-inch-diameter lightweight unit.

To accommodate the testing of this unit, a testing facility had to be designed and built. Because of the availability of a test pad and high-power variable frequency electric motor equipment, the facility was located on the grounds of the Aerodynamics Laboratory of the U. S. Navy David Taylor Model Basin.

The test stand and its calibration are described. A 150-HP motor was used. Shaft input torque and RPM were measured, so that input power could be measured accurately. Air volume flow and static and total pressure surveys were made, and air horsepower was determined.

Results of the tests indicate a peak total pressure efficiency of 85 percent. This correlates with model fan information, including an experimentally determined scale effect.

A limited noise survey investigation was made. It indicated low noise levels for the RD fan.

FOREWORD

The study presented in this report was undertaken by the Aerophysics Company, Washington, D. C. The experimental part of the work was performed at the Aerodynamics Laboratory of the David Taylor Model Basin (DTMB), Carderock, Maryland. The Project Officer at DTMB was Mr. J. N. Fresh, head of the Subsonic Division. Further support was given by Messrs. C. L. Benedum and C. K. Burdette, of the Aerodynamics Laboratory. The study was sponsored by the U. S. Navy Bureau of Ships*. It was monitored by the U. S. Army Aviation Materiel Laboratories (USAAVLABS), where Mr. William E. Sickles was the Project Engineer.

The study is a continuation of the work reported in USATRECOM Technical Report 64-33 (Reference 5) and USAAVLABS Technical Report 66-60 (Reference 4). The work was performed between June 1966 and January 1967.

*Now, Naval Ship Engineering Center of the Ship Systems Command

ILLUSTRATIONS

CONTENTS

SYMBOLS

BHP	horsepower absorbed by the fan
g	acceleration of gravity, feet per second square
P_o	atmospheric pressure, inches H_2O
P_t	total pressure referred to atmospheric, inches H_2O
Q	fan capacity, cubic feet per minute
RPM	Fan or torquemeter rotational speed
Δ BHP	horsepower absorbed by the reduction gear
ΔP_s	static pressure rise across the fan, inches H_2O
ΔP_t	total pressure rise across the fan, inches H_2O
θ	angle of straightener vane to flow radial direction
η_s	static fan efficiency
η_t	total fan efficiency
ρ	sea level density, slugs per cubic foot
W. G.	inches of water relative to ambient pressure

INTRODUCTION

Under two previous contracts, the Aerophysics Company, in association with Joy Manufacturing Company, demonstrated the feasibility of lightweight rotating-diffuser (RD) centrifugal fans for internal-flow airborne applications. Aerodynamic efficiency was established by means of system model tests using a 20-inch-diameter fan (Reference 5). One of the test configurations is shown in Figure 1. Lightweight fabrication and sturdiness were demonstrated in a program in which two 64-1/2-inch overall diameter lightweight fans were built and extensively tested structurally (Reference 4). An end view and a top view of one of these two fans are shown in Figures 2 and 3, respectively. A patent embodying the RD fan as the prime air mover for ground effect machine applications was recently granted to the Aerophysics Company (Reference 6).

The last step in the RD fan feasibility program is the aerodynamic testing of the full-scale fan, i. e., that shown in Figures 2 and 3. That step was accomplished under Contract DA 44-AMC-454(T), the results of which are discussed in this report.

As the size of the fan gets large, that of the corresponding internal-flow ducting grows accordingly; then, testing on a standard fan dynamometer becomes difficult, since most dynamometers (such as the one used for the testing described in Reference 4) have a horizontal shaft located only 3 or 4 feet from the ground, and the ground may interfere with the proper simulation of the ducting. In addition, to tie up an industrial fan dynamometer for the extended period of time required for the fabrication, calibration, and testing of a large-size internal-flow configuration becomes prohibitively expensive. A study of the adaptability of the dynamometer of the Joy Manufacturing Company, in New Philadelphia, Ohio, to the proposed testing thus led to the conclusion that such testing was impossible in New Philadelphia with anything but a conventional volute. The project was abandoned, since it was felt that some more sophisticated internal ducting simulation was required. It was decided to design and build a special test facility. This was possible at reasonable cost because of the availability on the grounds of the Aerodynamics Laboratory of the David Taylor Model Basin of a concrete pad with a pit at the center with the proper utilities, as well as that of variable-frequency, high-power electric generating equipment, normally used to drive the electric motors of wind-tunnel models. It was also possible to obtain on loan from DTMB a 150-HP electric motor, a strain gage torquemeter and the proper recording equipment. Thus, since the fans were already available, the only piece of capital equipment that had to be procured was the gearbox. Also, since the test pad was not used in any other program, there was no rush in completing the tests. The test setup was designed for maximum flexibility. The main features of the design are discussed in the following pages. Instrumentation, calibration, and method of data reduction are also described. The test stand components were fabricated in the Aerophysics Company shop and were assembled and installed at DTMB by Aerophysics personnel. Actual testing and data reduction were performed by Aerophysics. During the tests, monitoring of the electric power level required for each test was provided by a DTMB technician in telephone contact with the Aerophysics engineer running the test. Actual data from the tests can be found in Reference 1; the significant results are presented in this report in graphical form only. It was found early in the tests that Aerophysics had been provided with the erroneous calibration curve for the DTMB electric motor and that the motor could not furnish 150 HP at the fan design RPM of 1180, because of overheating. All test runs were therefore made at a derated design RPM of 826. This in no way affects the validity of the results.

The main object of the tests was to determine the overall aerodynamic efficiency of fan RD 51-.50-1.3-75^{o} under full-scale conditions. The fan was tested without a volute, and therefore air straightener vanes had to be used. It was found, as had to be expected, that the performance of the fan did depend upon the vane shape and setting. It was desired to compare the performance of the full-scale fan with the predicted performance based on results obtained from tests of a 20-inch fan, plus empirical corrections for scale effect. However, the results on the 20-inch fan were based on tests with a volute and therefore were not directly comparable. Additional difficulties were due to the fact that the discharge duct configuration downstream of the fan, being freely peripheral and radial rather than cylindrical, did not correspond to any of the standard configurations described by the Air Moving and Conditioning Association Standards (Reference 3). It was therefore necessary to do much calibration testing, and to repeat it a number of times, to be sure to obtain meaningful results. Because of the above difficulties, no attempt was made to do much detailed study of the internal flow configuration downstream of the fan. It represented a plenum-chamber configuration for a ground effect machine, but by no means an optimum one. The value of the test comes from the determination of the maximum aerodynamic potential of the fan alone and of its correlation with predicted results.

As a matter of general interest, noise levels of the fan at full power were made and compared with the noise levels of a typical ordinary centrifugal fan designed for identical duty.

DESCRIPTION OF TEST STAND

The test stand was constructed on the GEM pad at the David Taylor Model Basin. The GEM pad is a concrete pad 80 feet in diameter. At the center of the pad is a pit 10 feet in diameter and approximately 2 feet deep. Underground electrical conduits, water and compressed air lines run from the pit to a service bunker at the edge of the pad. A trailer, adjacent to the service bunker, was used to house all of the instrumentation.

The electric motor used to power the fan during the test was controlled from the subsonic model tunnel control room, across the street from the GEM pad. An intercom was installed between this control room and the instrument trailer to provide continuous communication between the test conductor and the control room operator. A sound-powered phone was also installed between the trailer and the test stand.

The test stand was located at the center of the GEM pad. It consisted of a steel framework, covered with 5/8-inch plywood, to form a hexagonal plenum chamber, 8 feet on each side and 39 inches high (Figure 4). At the center of the plenum chamber, extending down into the 10-foot-diameter pit, was the power train. Figures 5 and 6 show the power train held in place by its supporting framework prior to the construction of the plenum chamber.

The electric motor used during the test was supplied by DTMB. It was a split phase, variable speed motor, capable of producing 150 horsepower at 8000 RPM. Cooling water for the motor was provided from the existing water supply line in the pit. A pressure regulator and a gauge were installed in the water line, and the pressure was adjusted to 25 psi. The motor was coupled to the gearbox through a Baldwin Lima Hamilton torque pickup, model SR4, type A-2. The speed reducer was a Link-Belt type 307S, with a reduction ratio of 4.993 : 1. A special shaft was machined to fit in the top of the gearbox upon which the RD fan was mounted.

Inlet air to the RD fan passed through a duct 8 feet high and 27 inches in diameter (see Figure 7). A bellmouth was fitted to the top of the duct to provide smooth air entry into the duct. An egg-crate-type air straightener, approximately 9 inches downstream from the bellmouth, was installed to insure smooth airflow down the length of the duct. Inlet static and total pressures were obtained from 2 Pitot tubes just upstream of the RD fan inlet (see Figure 9). Three radial struts protruded into the duct just above the mouth of the fan. These struts supported a steady bearing at the top end of the fan drive shaft. This bearing was provided to insure proper alignment between the fan inlet and the duct. The base of the inlet duct was designed to allow as little leakage as possible from the plenum back to the inlet of the fan. The labyrinth-type path formed by the top of the plenum chamber, the duct wall, and the lip of the fan inlet can be seen in Figure 5. Inspection holes near the base of the duct were used both to align the duct and to adjust the gap between the duct and the fan inlet. These holes were later sealed with tape.

A large circular disk, approximately 10 feet in diameter, was located just under the fan so that a radial delivery duct was formed at the fan exhaust (the top of the plenum chamber being the top of the delivery duct).

At the downstream end of the delivery duct, adjustable air straightener vanes were installed. Three sets of vanes were designed; two were built, installed, and tested. The precise location of the vanes can be found from the plan view drawing on Figure 9 (page 7), where the general arrangement drawing of the test stand is also shown. The vanes can also be seen on the schematic in Figure 10 (page 9). The coordinates of the profile of the second set of vanes, which were used in the final runs, are shown in the table below.

Adjustable flaps at the lower edge of the plenum chamber provided a means for varying the fan air delivery. The fan openings were adjustable from 0 to 10 inches. The adjustable flaps are shown in the partially open position in Figure 8 on page 6.

TABLE OF ORDINATES AND PROFILE OF SERIES 2 AIR STRAIGHTENER VANES, % CHORD.
(Installed Chord Length: 15.10 In.)

Station	Lower	Upper
0	1.85	1.85
2.5	0.13	4.90
5.0	1.00	6.10
7.5	1.85	7.20
10	2.58	8.14
20	[illegible].70	11.52
30	5.75	13.32
40	6.16	13.83
50	5.87	13.32
60	5.23	11.78
70	4.24	9.53
80	3.11	6.42
90	1.79	3.58
95	1.13	1.99
100	0	0

L. E. Radius: 1.72 %

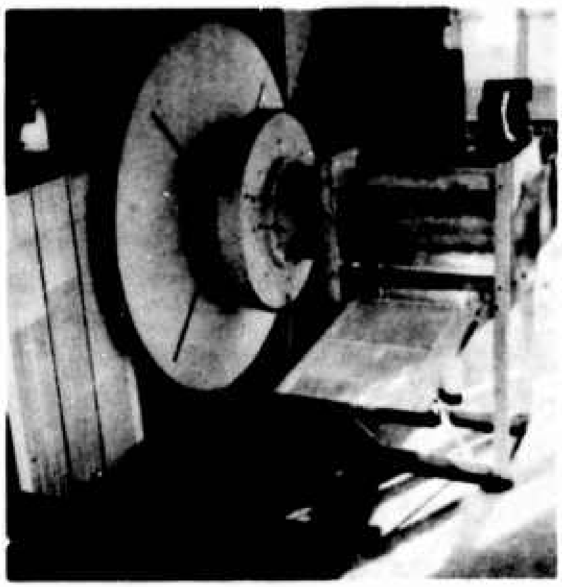


Figure 1. Overall View of Test Setup - 1963 Program of 20-Inch-Diameter RD Fan Model Testing, Circular Configuration.

Figure 2. End View of Full-Size RD 51-.50-1.3-75° Fan, 64.50-Inch Overall Diameter.

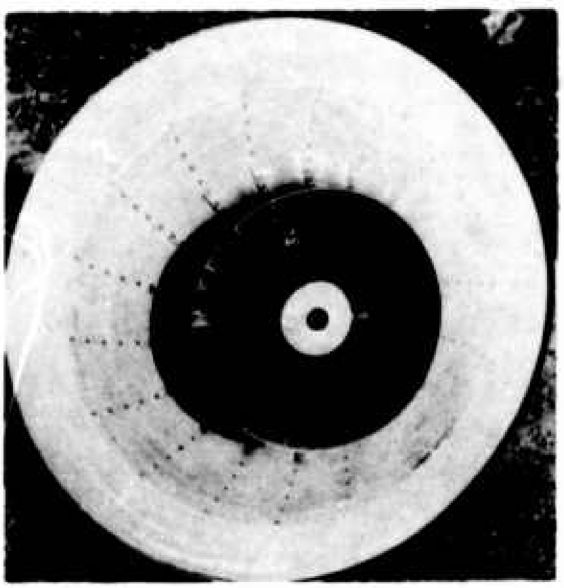


Figure 3. Top View of Full Size RD 51-.50-1.3-75° Fan.

Figure 4. View Showing Complete Installation of RD Fan Test Stand on the David Taylor Model Basin GEM Pad.

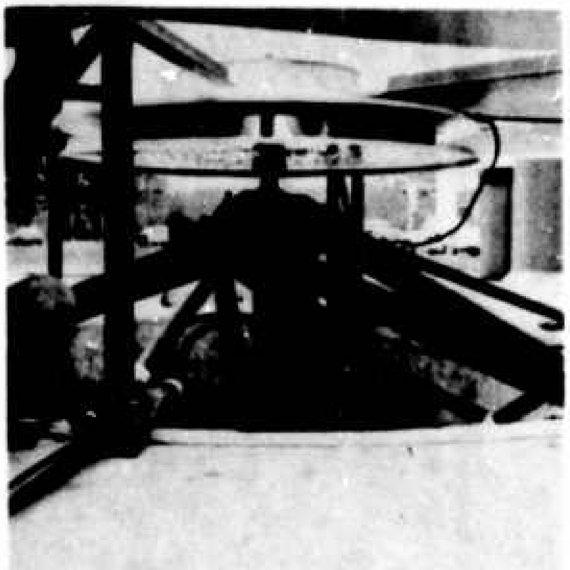


Figure 5. Fan RD 51-.50-1.3-75^{o} Installed on Power Train at David Taylor Model Basin GEM Pad. Surrounding Structure Partially Completed.

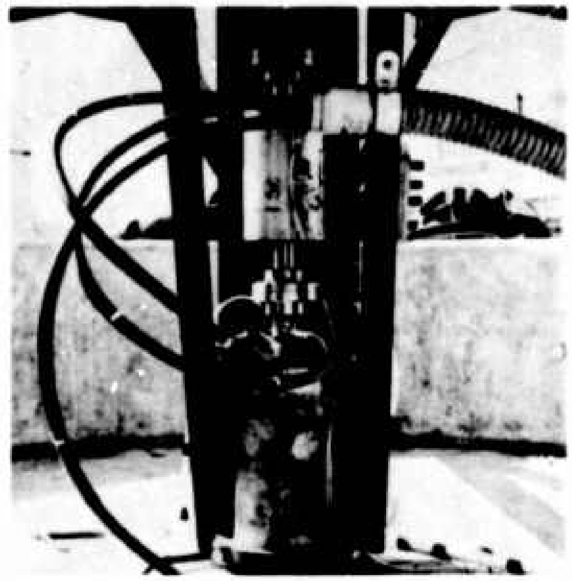


Figure 6. Close-Up of Power Train Showing 150-Horsepower Electric Motor, Torque Pickup, Flexible Couplings, Electric and Water Lines.

Figure 7. Calibrated Inlet Duct and Air Straightener Assembly Before Installation.

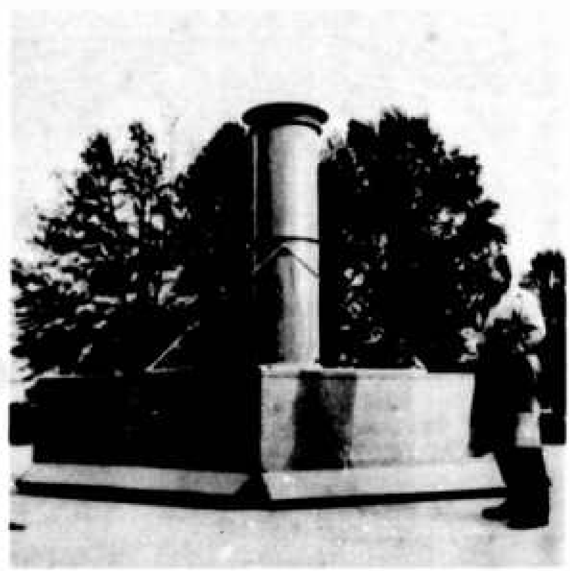
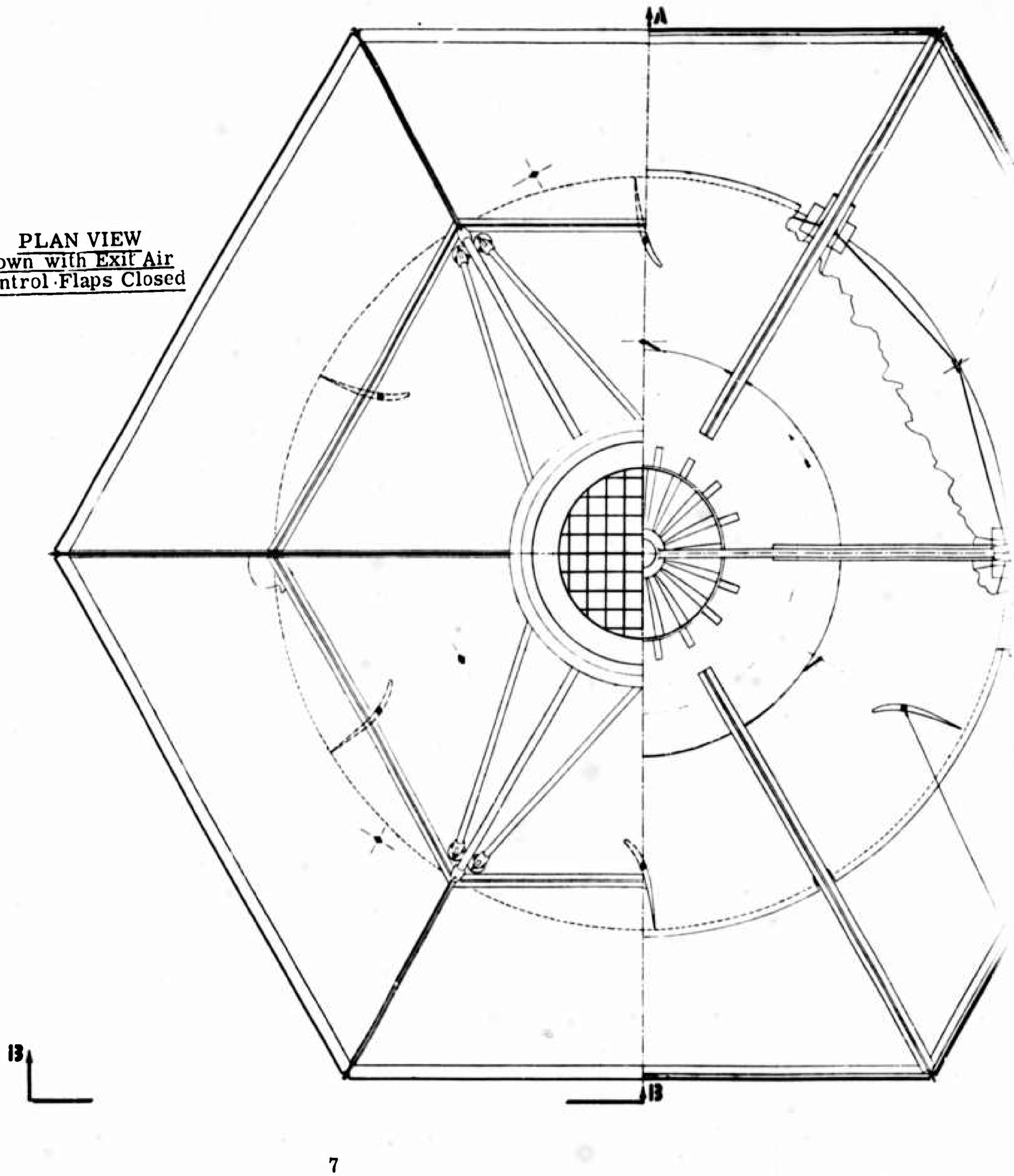
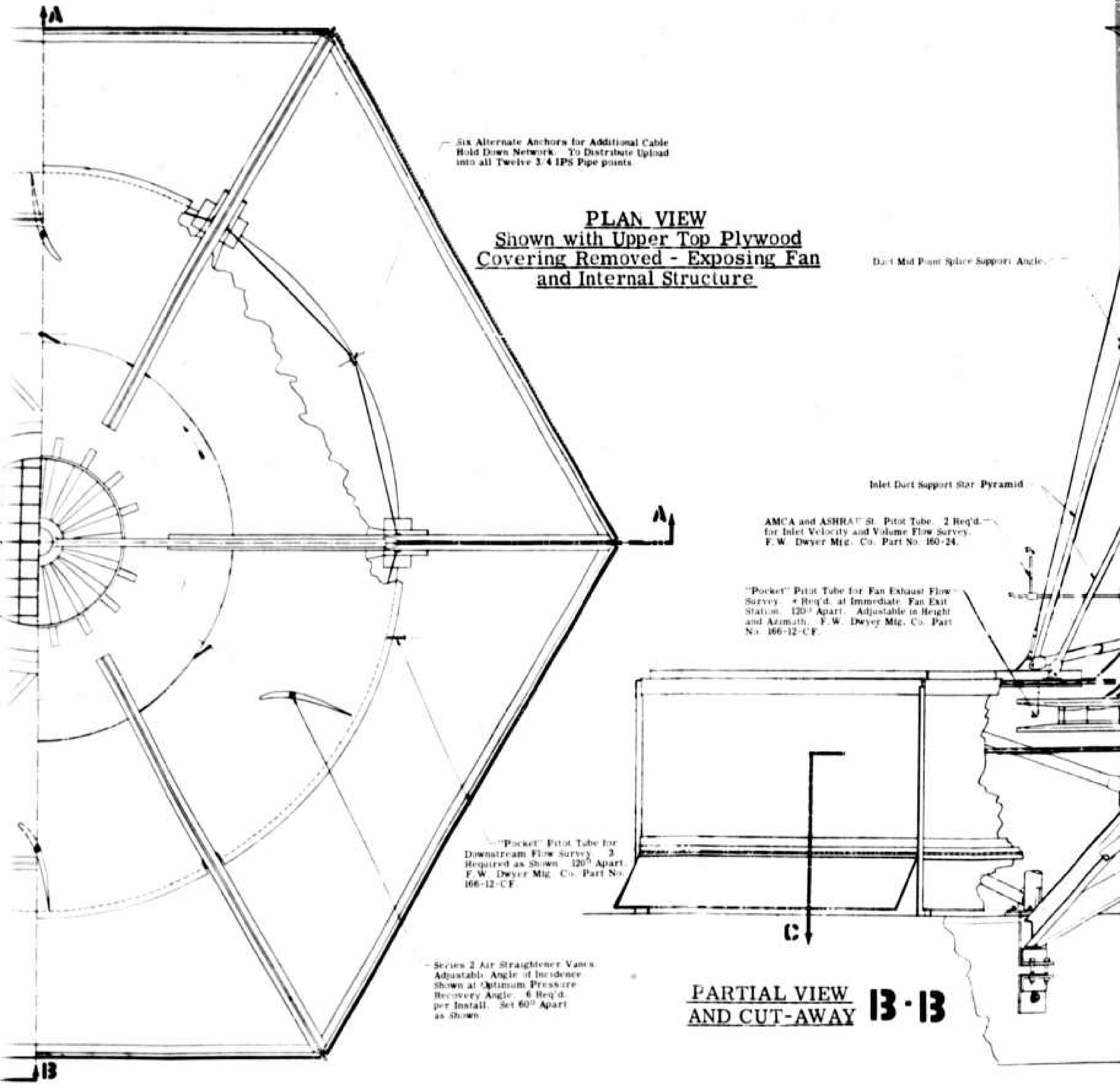


Figure 8. Partially Open Flaps at Lower End of Plenum Chamber.

PLAN VIEW
Shown with Exit Air
Control Flaps Closed



A


A
Six Alternate Anchors for Additional Cable Hold Down Network. To Distribute Upload into all Twelve 3/4 IPS Pipe points.
PLAN VIEW
Shown with Upper Top Plywood Covering Removed - Exposing Fan and Internal Structure
Duct Mid Point Splice Support Angle.
Inlet Duct Support Star Pyramid
AMCA and ASHRAE St. Pitot Tube. 2 Req'd. for Inlet Velocity and Volume Flow Survey. F.W. Dwyer Mfg. Co. Part No. 160-24.
"Pocket" Pitot Tube for Fan Exhaust Flow Survey. 4 Req'd. at Immediate Fan Exit Station. 120° Apart. Adjustable in Height and Azimuth. F.W. Dwyer Mfg. Co. Part No. 166-12-CF.
A
"Pocket" Pitot Tube for Downstream Flow Survey. 3 Required as Shown. 120° Apart. F.W. Dwyer Mfg. Co. Part No. 166-12-CF.
C
Series 2 Air Straightener Vanes. Adjustable Angle of Incidence Shown at Optimum Pressure Recovery Angle. 6 Req'd. per Install. Set 60° Apart as Shown.
PARTIAL VIEW AND CUT-AWAY
B - B
B

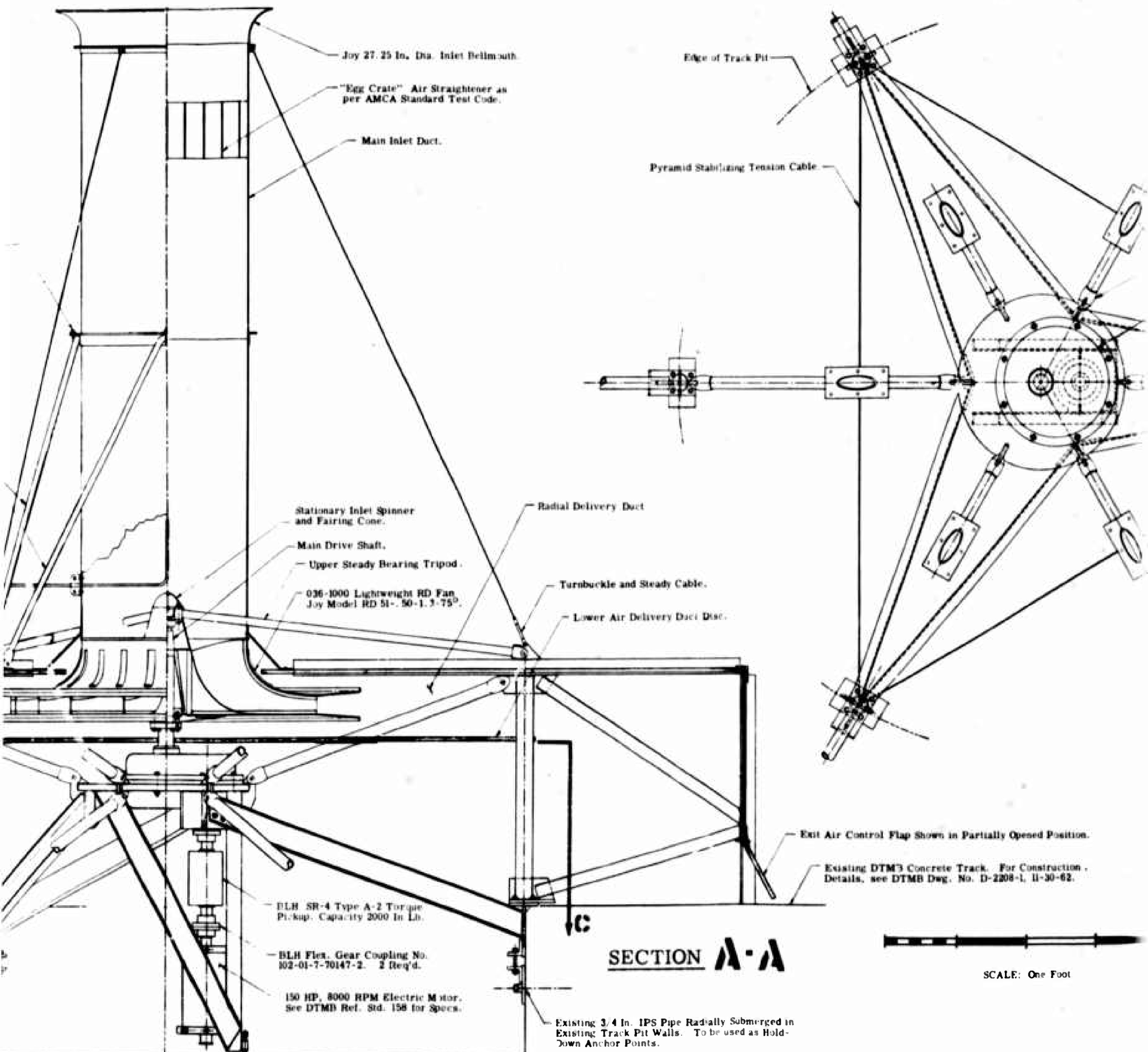


Figure 9. General Ar
Test Stand - Rotating

C

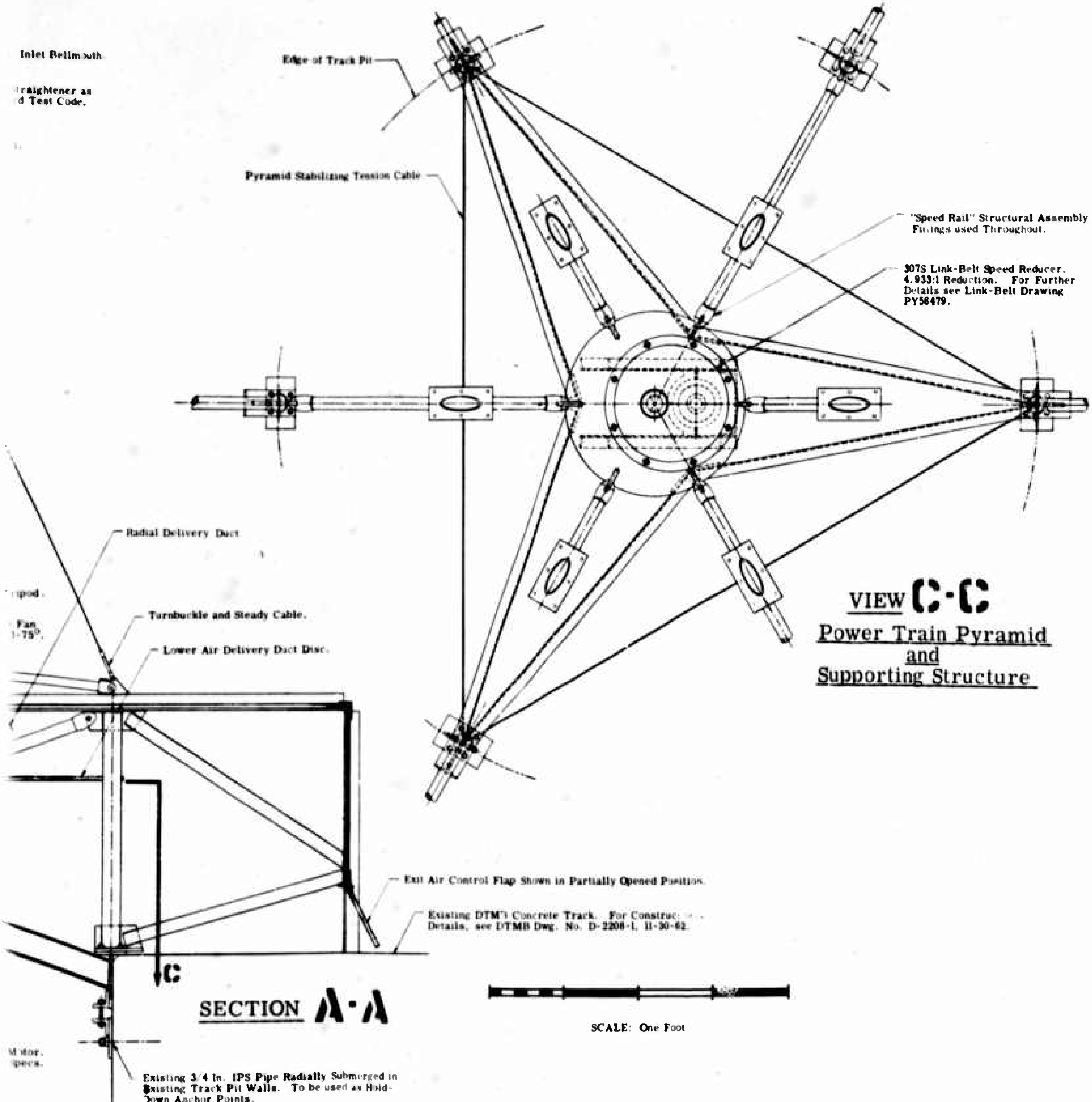


Figure 9. General Arrangement of Test Stand - Rotating Diffuser Fan

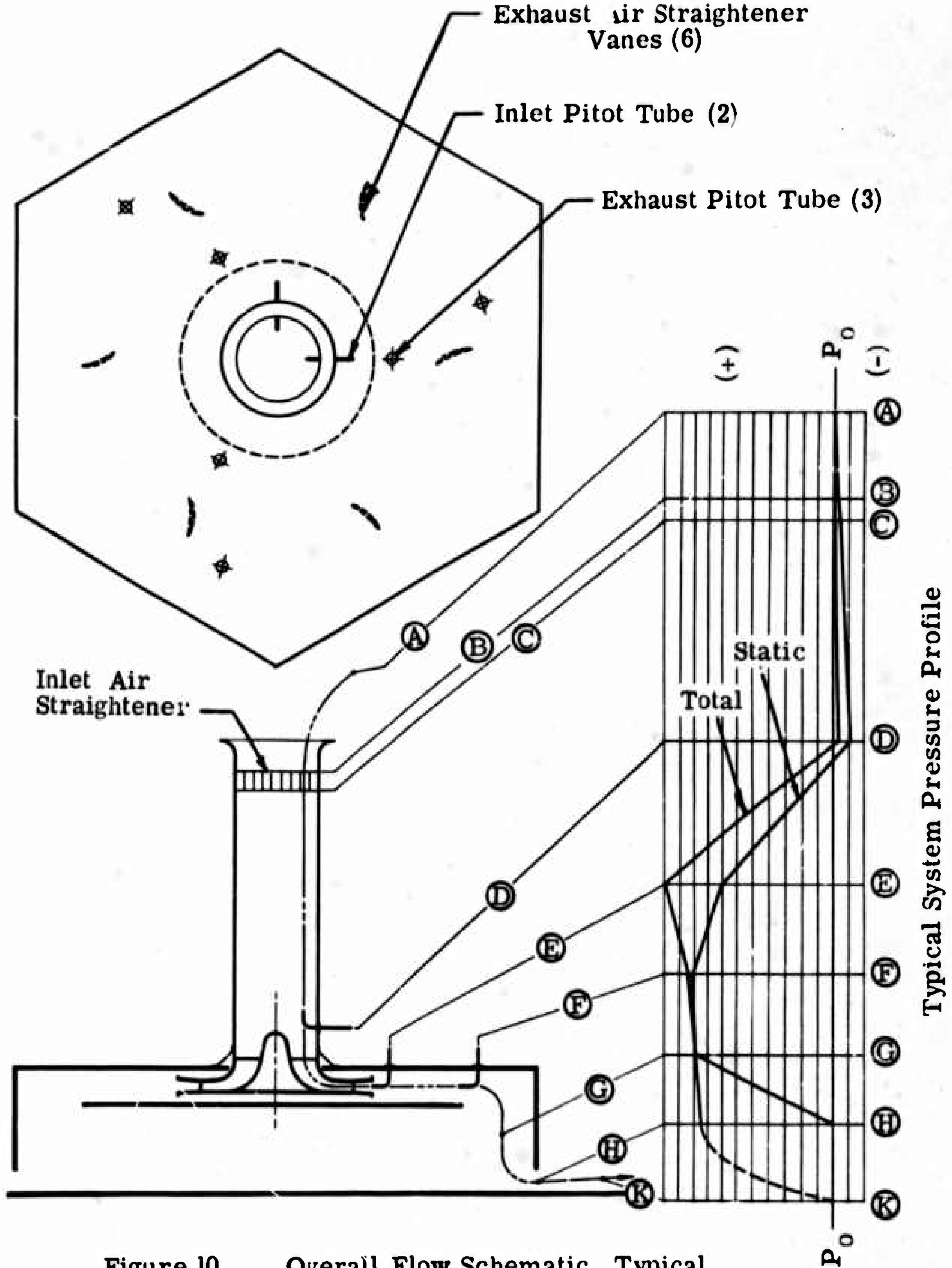


Figure 10. Overall Flow Schematic, Typical System Pressure Distribution and Pitot Locations.

DESCRIPTION OF INSTRUMENTATION, CALIBRATION, AND SUPPORT EQUIPMENT

All of the measurements taken during these tests were made with the highest quality instruments available, many of which were furnished by the Instrumentation Branch of the Aerodynamic Laboratory of DTMB. On-site installation and arrangement of the test equipment generally followed the procedures described in the Laboratory Guide (Reference 2) of the Air Moving and Conditioning Association, Inc. (AMCA). Basic testing, calibration, observation, and calculating procedures, as standardized by the AMCA Test Code (Reference 3), were used throughout this test when applicable. These standards were used mainly for test procedure guidance, as the geometry and arrangement of the RD fan do not adapt themselves directly to this code. Great care was taken to insure that the installation and use of all the equipment were in accordance with the manufacturer s instructions and specifications. A discussion of the use of the test measuring equipment follows.

SPEED MEASURING EQUIPMENT

Fan RPM is read out on a Hewlett Packard Model 5512A frequency counter. This frequency counter receives its sensing signal directly from a four-pole generator which is an integral part of the fan drive motor. By knowing the frequency count and the speed reducer ratio, one may accurately compute the fan RPM. The accuracy of the frequency counter is positively assured and is stabilized by a built-in and fixed-time-based crystal-controlled oscillator.

POWER MEASURING EQUIPMENT

Torque being transmitted from the drive motor to the RD fan is dynamically sensed by a Baldwin Lima Hamilton (BLH) SR4 type A-2 torque pickup. This unit is, in effect, an electronic transducer which produces an electrical signal proportional to the applied torque. The torque pickup is essentially a short length of shaft which, when loaded in torsion, translates that torsion into changes in electrical output. The torque sensitive element in the pickup is a system of strain gages; these are bonded to a shaft in such a position and are connected into a Wheatstone bridge circuit in such a way that the effects of bending and thrust strains are canceled, while the effects of torsional strains are added. The relation between bridge unbalance and torsional strain is linear.

Slip rings on the shaft and a nonrotating brush assembly allow the strain gage bridge to be energized, and the resulting resistive unbalance is measured by and read out on Aeres Instrument Company Model T-1C strain indicator.

On-the-site daily calibration was conducted on the torque pickup-strain indicator loop. This procedure was carried out in accordance with BLH specifications for calibration of the A-2 unit. Essentially, this calibration consists of applying a dummy or simulated torque into the bridge circuit in the form of a pickup-matching, factory-calibrated load resistor and of adjusting the strain indicator to read this load. Each torque pickup load resistor is carefully selectively matched at the factory. This initial calibration insures that the pickup unit conforms to specification MIL-C-45662A - "Calibration System Requirements".

All of the wiring for the RPM and torque measurements was shielded, and all shields were terminated at a single point ground in the instrument trailer in accordance with NEMA standards.

PRESSURE MEASURING EQUIPMENT

At the beginning and end of each test day and at the beginning of each significant final determination or run, the ambient barometric pressure was recorded on a Wallace and Teirnan aneroid barometer, Model FA126. This was done to insure that the final data could be reduced to a common standard day performance comparison.

All pressure measurements made in the test system were made with fully adjustable Pitot tubes, located as shown in Figures 9 and 10. The two AMCA standard inlet Pitot tubes surveyed the volume flow and 20 traverse points, as described in Reference 3. The immediate RD fan exhaust flow is surveyed by three "pocket" Pitot tubes which are a reduced version of the AMCA standard unit. Another set of three "pocket" Pitot tubes is located at the outer edge of the air delivery duct. Also, many static pressure taps were located throughout the plenum box and along the ground. Although pressure surveys were made at the latter locations, the results only served as a performance check and are not presented in this report.

Pressures originating at these Pitot tubes are collectively read on a multiple vertical tube manometer board whose reference reservoir is open to the atmosphere. All pressure measurements were made directly in inches of water and are presented directly as such in the text of this report. The vertical manometer board was initially checked against a precision micromanometer. Daily checks were made of the pressure lines between the Pitot tubes and manometers for leakage and stoppage.

TEMPERATURE MEASURING EQUIPMENT

At the beginning and end of each test day and at the beginning of each significant final determination or run, the dry and wet bulb temperatures were measured at the fan inlet bellmouth with an H-B Instrument Company sling psychrometer to determine ambient temperature and humidity.

Incidental to the actual accumulation of test data, temperatures of the electric drive motor field windings and bearings, torque pickup bearings and speed reducer oil were sensed by thermocouples and were recorded on a Leads and Northrop type G Speedomax multichannel temperature recorder. This was done to insure that all of the equipment operated within predetermined safe temperature units.

SOUND LEVEL EQUIPMENT

Several brief sound level surveys were made at the test site with a portable General Radio Sound Level Meter, type 759-B. Readings were made at several distances from the test rig, both with air control flaps full open and closed.

EXPERIMENTAL PROCEDURES

FAN ALONE

The main quantity that must be measured accurately is the total efficiency of the fan, i. e., the ratio of the air horsepower delivered by the fan to the shaft horsepower.

The shaft horsepower is the product of the fan RPM and the fan torque. The fan RPM is measured very accurately by measuring the motor RPM and by knowing the speed reducer gear ratio. The torque could be measured very accurately by using a torque pickup located in line with with the electric motor. However, as shown clearly in Figure 9, for a number of design considerations, it was found necessary to locate the gearbox between the fan and the torque meter. Thus, it becomes necessary to calibrate the power losses of the gearbox and to subtract them from the measured power. Ideally, this should be done by replacing the fan by a known load, for example, an electric dynamometer or eddy-current brake, the output of which can be measured. Since such was not available, investigations were made of the possible use of standard loads, as recommended in Marks' Handbook; these consist of flat plates at the ends of long rotating arms, in the form of a fan brake dynamometer. It was found that this procedure would result in calibration errors of the order of 20 percent; it was therefore abandoned. Finally, the Link-Belt speed reducer was calibrated without load. Compared with the losses under load, this is considered to be conservative.

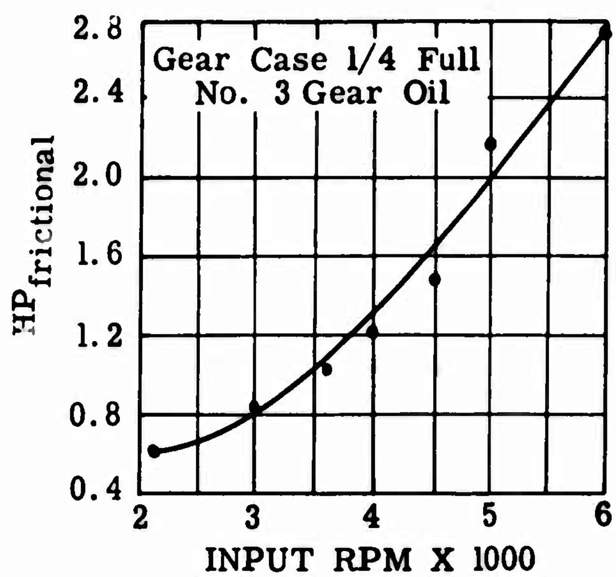


Figure 11. Calibration Curve for Frictional Horsepower Absorbed by Link-Belt Speed Reducer No. 307S.

The air horsepower is the product of the volume flow and of the total pressure rise across the fan.

The volume flow was measured in a conventional manner, in accordance with the AMCA Code (Reference 3). This was made possible by measuring static and total pressure at the downstream end of the calibrated inlet duct, as shown in Figure 10, at the radial locations specified in Reference 3 and shown in Figure 10.

As to the total pressure rise across the fan (and the static pressure rise as well), its measurement was much more complex. Since what was desired was the pressure rise for the fan alone, not the fan-plenum chamber combination, the pressures were measured just downstream of the rotating diffuser. As can be seen in Figures 13 and 14, the pressure was very far from being uniform in the fan axial direction. Pitot stations 1 and 7 were taken right at the diffuser edge. In Figures 13 and 14, the exhaust Pitot survey was carried out at seven equally spaced vertical stations. Station 1 was located at the lower immediate edge of the fan diffuser ring, while station 7 was at the upper diffuser edge. Station 4 located the Pitot at the center line of the fan diffuser discharge. For each run, it was therefore necessary to measure the total pressure at seven stations and graphically integrate the area under the curve, to obtain a mean pressure, which is used in the calculation of air horsepower.

Precautions must be taken to determine the direction of the flow first in order to read the correct pressure. This was done by trial and error for each position, with an operator standing on top of the plenum chamber and changing the direction of the Pitot tube until the operator in the trailer signaled over the phone that a maximum reading had been obtained.

To determine the direction of the flow in the plenum chamber, an alternate method was used, using tufts or flags free to rotate in the airstream.

It was now necessary to do a similar upstream pressure averaging, to account for the presence of the chimney, hence the negative pressures at the fan inlet, evidenced in Figure 10. This was not too much additional work, since a pressure survey there was needed to calculate the volume flow.

An independent check on inlet losses was provided in Reference 5 by making two successive measurements for the same fan operating condition, with and without the inlet duct respectively. This was not possible here, because dismantling of the chimney would have been a major operation. However, the losses could be calculated theoretically, and the results agreed well with the test data. However, in this manner, the drag losses due to the three struts just upstream of the fan eye could not be accounted for. Therefore, measured inlet values are likely to be too small rather than too large.

PLENUM CHAMBER

The only quantities that can be measured in the plenum chamber are static pressure, total pressure, and flow velocity direction. Pressures were measured either by using single Pitot tubes or by using rakes to determine the average pressure or velocity through a section. Surveys such as that shown on Figure 10 were made. It was not the intent of the project to make a detailed study of the plenum chamber or to modifv it to minimize the pressure losses. This is reserved for a future study.

DISCUSSION OF RESULTS

EFFECT OF STRAIGHTENER VANES

A rotating-diffuser fan is normally used in conjunction with a volute. This is not convenient when essentially radial flow distribution is desired. This problem was considered earlier by Aerophysics as part of the system studies reported in Reference 5. The solution that was proposed and found experimentally to work, for a circular planform configuration, consisted of adding curved vanes in the far downstream peripheral nozzle, thus leaving the radial duct uncluttered. Since, in the present study, it was not intended to simulate an annular jet ground effect machine configuration, it was felt worthwhile to devise a set of air straightener vanes that would work well for a plenum chamber type of configuration.

Three sets of vanes were designed (Reference 1). The first set used a symmetrical airfoil, the second a moderately cambered airfoil, and the third a strongly cambered airfoil. The first and second sets were built and tested under similar flow conditions, using six blades. Under a broad range of flow conditions, it was found that the second set of vanes performed much better than the first. Intuitively, this made a great deal of sense, since the flow leaving the fan is closer to the tangential than to the radial direction; however, at the end of the radial delivery duct, i. e., at the trailing edge of the vanes, the flow must be essentially radial. Lack of time prevented the testing of the third set of blades, but it is speculated that the third set would have given still better results.

It is difficult to rely on theoretical arguments to optimize the straightener vanes design. To complicate things further, the radial delivery duct was far from being aerodynamically clean, because of the presence of the struts which served as foundation for the hexagonal plenum box. Optimization of the straightener vanes is a trial-and-error process, in which various airfoil chords, curvatures, and gaps should be tried. This was beyond the scope of the present test. However, very significant information was gained from the test, and it will be discussed now.

The blades could be rotated about the 33-percent chord point. The effect of blade angle on total pressure recovery was systematically studied. Obviously, the optimum blade angle depends upon the flow delivery. At large capacities, the angle θ between the blade reference line and the radial direction is smaller than at small capacities. In other terms, the resultant velocity is more tangential for small capacities than for large ones. If one wishes to optimize near the capacity for maximum efficiency, a larger curvature than was used in the test is also probably desirable.

The flow condition for which the vane angle θ was optimized was dictated by practical considerations. Specifically, to minimize leakage losses, no provision was made to build an access door into the plenum chamber. Entry into the chamber was made by crawling under the peripheral flap in the full-open position. Since dozens of runs were made with different blade settings, it was found convenient to do the optimization with the flap fully open, thus avoiding the necessity to reset the flap door at the same precise setting every time. When final runs were made, the optimum blade angle determined in the full-open position was used. This fact will be noted in the discussion of the results made in a following paragraph.

Figure 12 shows the variation of a typical reference total pressure (total pressure at the center line at the rotating diffuser outer edge station) as a function of blade angle. The main conclusion is that the fan performance is extremely dependent upon the blade angle. As will be seen later, the other conclusion is that a system of 6 blades of type II (see table on page 4) gives very acceptable efficiencies, in spite of the fact that no strong attempt was made to optimize them.

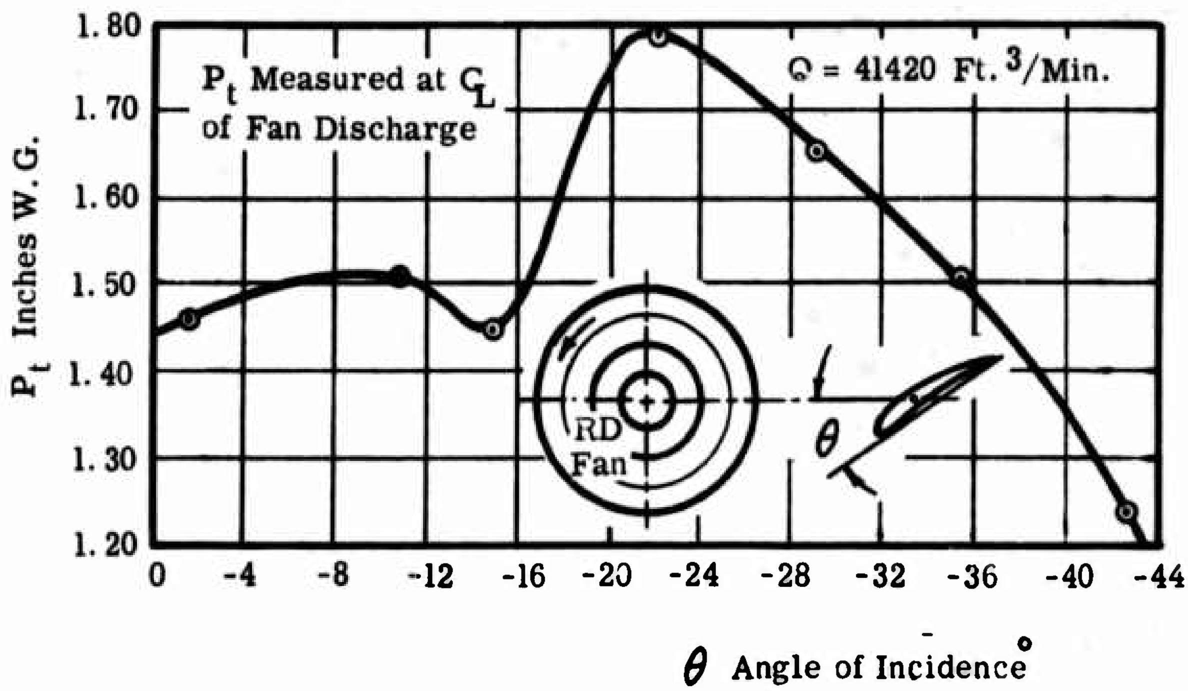


Figure 12. Effect of Air Straightener Vane Angle of Incidence (θ) on Total Pressure Recovery (P_t) at Fan Exhaust.

STATIC FAN EFFICIENCY

The static fan efficiency is defined as:

$$\eta_s = \frac{Q \times \Delta P_s}{550 \times BHP}$$

As indicated earlier and as shown on Figure 10, the volume flow is calculated from a static and total pressure survey made just ahead of the fan (Station D) in accordance with AMCA Standards. The brake horsepower is obtained as indicated earlier. Some care must be exercised in the determination of the static pressure rise across the fan.

The static pressure rise ΔP_s is obtained as the difference of static pressures between station D and station E, none of which is uniform. The static pressure distribution at D and the corresponding average pressure were already calculated for the determination of the volume flow. The static pressure distribution at E (referred to atmospheric) across the rotating diffuser width is shown in Figure 13, at 826 RPM, for the 8 volume flows of interest. The average static pressure P_s was obtained by integration of the area under the curve with a planimeter and dividing by the rotating diffuser depth. Then, $\Delta P_s = \Delta P_{s_E} - \Delta P_{s_D}$

Static pressure rise, horsepower, and static efficiency are plotted in Figure 15, at 826 RPM, as a function of volume flow. These results will be discussed after presentation of the corresponding results for total pressure.

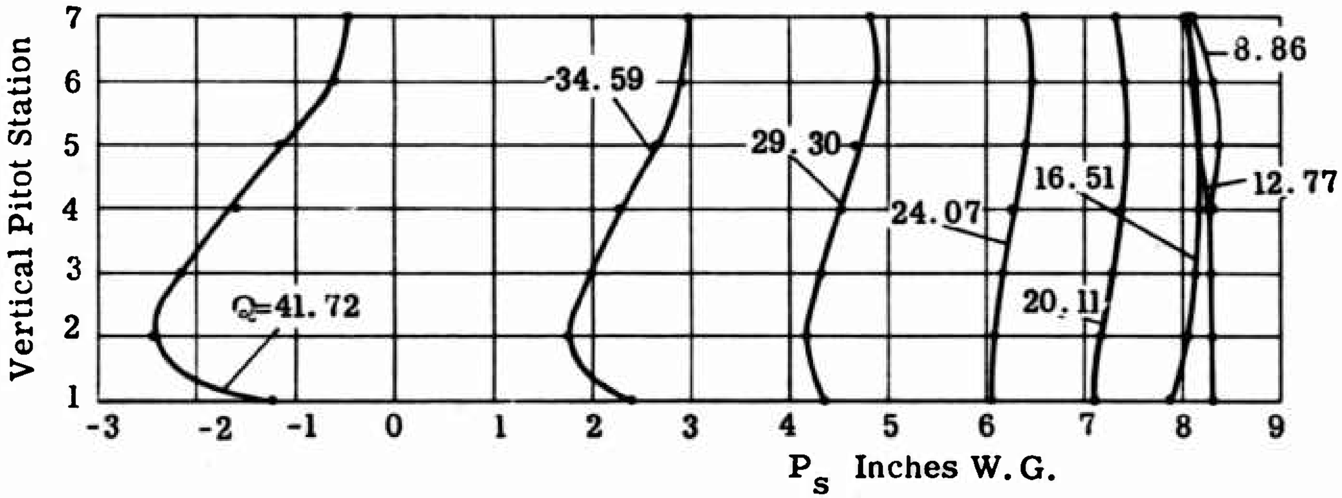


Figure 13. Fan Exhaust Static Pressure Profiles at Various Volume Flows (Q). Ft.^{3}/Min. x 1000.

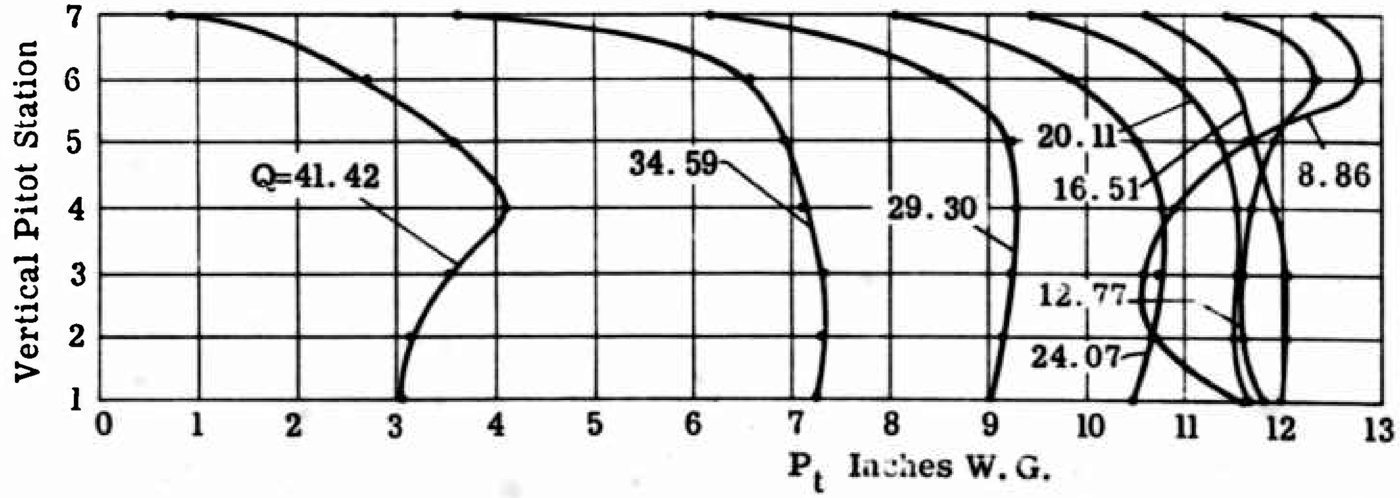


Figure 14. Fan Exhaust Total Pressure Profiles at Various Volume Flows (Q). Ft.^{3}/Min. x 1000.

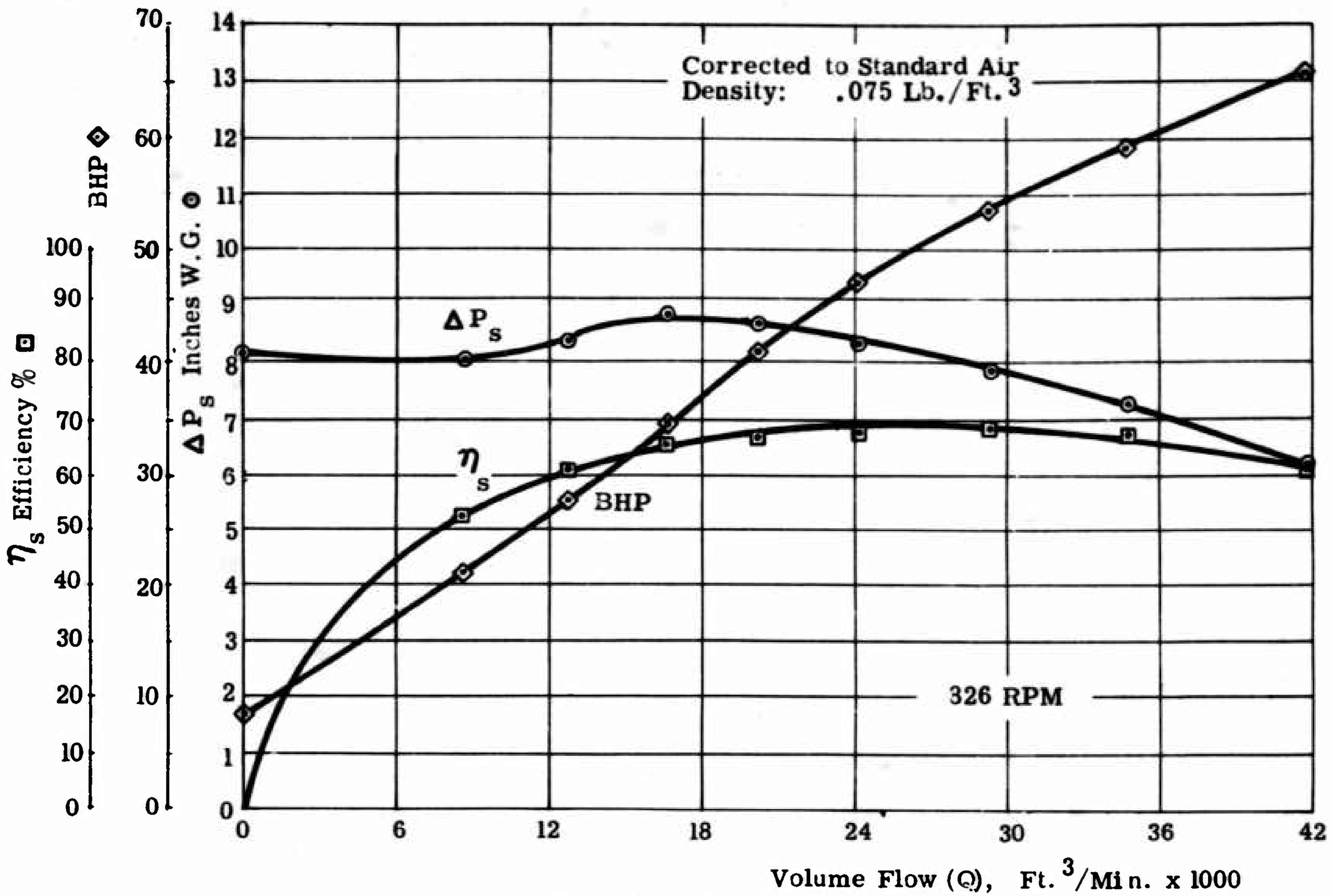


Figure 15. RD 51-.50-1.3-75° Fan - Measured Performance Based on Static Pressure Rise.

TOTAL FAN EFFICIENCY

The total fan efficiency is defined as:

$$\eta_t = \frac{Q \times \Delta P_t}{550 \times BHP}$$

The determination of the average total pressure rise ΔP_t across the fan is similar to that of the static rise ΔP_s; i. e., the total pressure rise ΔP_t is the difference between the average total pressures at stations D and E of Figure 10. The total pressure distribution at station E at 826 RPM, is shown on Figure 14 for the 8 volume flows of interest. It can be seen that the total pressure distribution is very far from being uniform. An average total pressure at station E can be determined in the same manner as for the static pressure, hence ΔP_t.

The variations of total fan pressure rise, horsepower, and total efficiency with volume flow, at 826 RPM, and for the straightener vane setting described earlier are plotted on Figure 16.

When the maximum potential of the fan as an air mover for an annular-jet type of ground effect machine is determined, the total pressure rise and the total efficiency are significant. Actually, in Reference 5, only the total pressure is plotted. For a plenum-chamber type of machine, static efficiency may be more representative of the fan performance. However, even with the worse possible internal-flow design, some dynamic pressure recovery would take place, so that the true efficiency would lie somewhere between static and total efficiency. It is thus felt that, in the absence of a specific internal-flow design, total efficiency is much more representative of the fan performance than static efficiency.

It is seen from Figure 16 that the maximum total efficiency of the fan is 84.2 percent. It is further seen that a significant surge of the fan is observed at low volume flows. On the contrary, there was no surge at all in the model tests reported in Reference 5. The reason was that the fan of Reference 5 was equipped with modulating vanes and the present one was not. An additional reason, and that point will be discussed again later, is that the straightener vanes were set and optimized for maximum volume flow, and the result is a penalty at zero volume flow.

PREDICTED FAN PERFORMANCE

The Joy-Neu proprietary method of prediction of the performance of a rotating diffuser fan is described in Reference 5. It is based on the proper use of test data on a 20-inch-diameter wheel, to which is added a pseudo-empirical correction for scale effect. The Neu method was applied to the RD 51-.50-1.3-75° fan, at 826 RPM. Performance

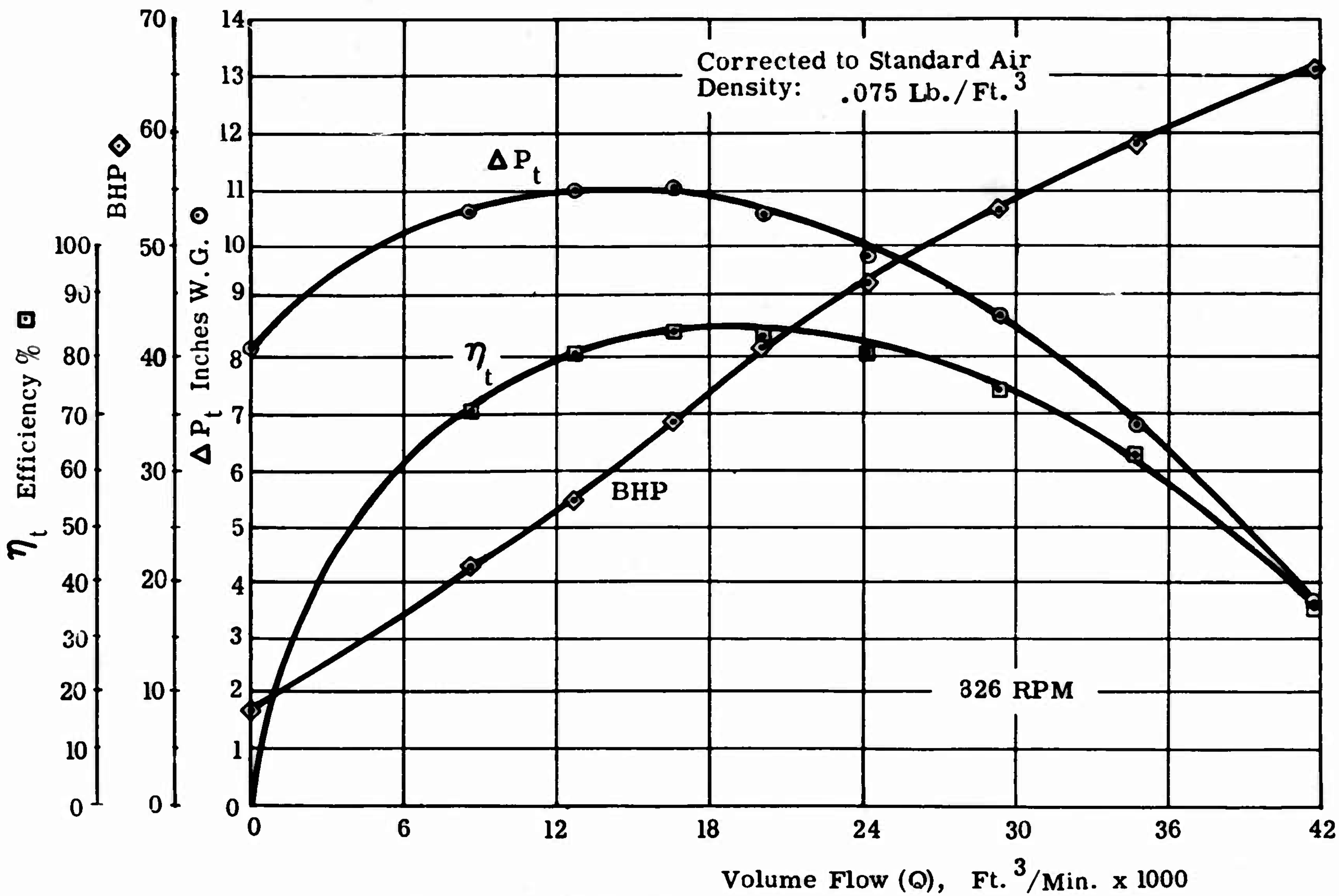


Figure 16. RD 51-.50-1.3-75° Fan - Measured Performance Based on Total Pressure Rise.

for the 20-inch wheel was calculated first; performance for the 51-inch wheel was calculated next. It was found that the scale effect resulted in an efficiency increase. The increase was not very large at the point of maximum efficiency, 1.8 percent, but it reached nearly 6 percent at the point of maximum volume flow.

The calculated variations of total pressure, horsepower, and total efficiency with volume flow are plotted in Figure 17 for the 51-inch (64-1/2-inch overall diameter) fan. The corresponding curve for the 20-inch model fan, showing the difference due to scale effect, was not plotted because, for most of the curve, it would have been difficult to distinguish from the other curve.

COMPARISON OF CALCULATED AND MEASURED PERFORMANCE

On Figure 17, the measured performance is shown as a solid line and the calculated performance as a dashed line. The following can be seen:

- The maximum total pressure rise is the same.
- The maximum measured efficiency falls short of the calculated one by 1.6 percent.
- The measured total pressure rise is significantly larger, at large volume flows, than the calculated one. This is compensated by the fact that, at and near zero flow, the measured pressure is smaller than the calculated one.
- Correspondingly, the measured efficiency is larger, at larger volume flows, than the calculated one (in spite of the fact that the calculated efficiency was raised by 6 percent because of scale effect).
- Measured horsepower is always larger than calculated horsepower.

These results lead one to the following conclusion: There is reasonable agreement between predicted performance and measured performance. The predicted performance is based on data obtained with a volute. Measured performance was obtained with a radial delivery duct and a set of nearly radial straightener vanes. The key point is that, in the results presented herein, the straightener vanes were optimized for maximum flow delivery, hence the higher pressure and higher efficiency at high volume flows and the corresponding penalty at low volume flows. There seems to be little doubt that, as one would decrease the angle θ of Figure 12 further away from the radial direction, one would lose total pressure at large volume flows but would trade it off for larger pressures at lower volume flows. Specifically, a change of the straightener vane setting of about -20 degrees would reduce the total measured pressure (Figure 12) by the proper amount to bring measured and predicted performance into coincidence. The dashed pressure curve of Figure 17 would rotate about point A, and this would

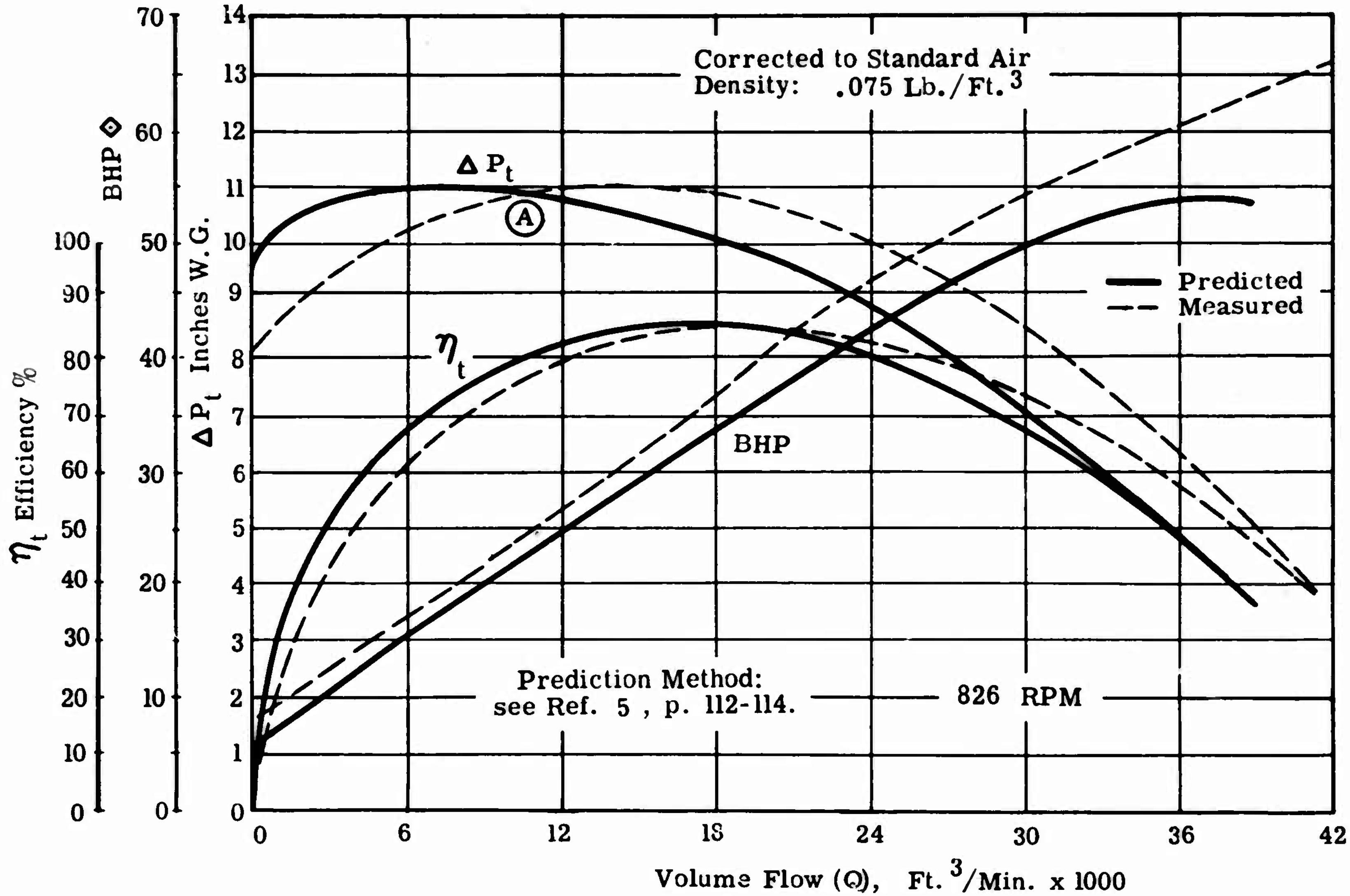


Figure 17. RD 51-.50-1.3-75° Fan - Predicted Fan Performance and Comparison with Measured Performance.

insure good agreement between predicted and measured pressures through the whole range of volume flows. At the same time, the efficiency would be expected to decrease at large volume flows and to increase at the low flows, also giving a better agreement on the efficiencies.

When one goes back to the static performance plot of Figure 15, the influence of optimizing the straightener vanes at large volume flows can be seen in the fact that the maximum static efficiency takes place at much larger volume flow values than one would normally expect, and the static pressure rise at large volume flows is also exceptionally high.

NOISE LEVEL STUDY

At the end of the tests, a limited noise survey was performed; standard General Radio sound pressure equipment was used. Sound level measurements were made directly on top of the hexagonal plenum chamber. With the B weighting curve, a maximum sound pressure level of 90 decibels was recorded. The corresponding noise level of the backward-curved centrifugal fan designed for the same duty is 110 decibels. It is concluded that the RD fan is significantly quieter than the ordinary centrifugal fan. The reason is that the RD fan eliminates the major causes of fan noise: blade-to-blade turbulence and close clearance between blade tips and cutoff.

CONCLUSIONS

1. A test facility for fans up to 6 feet in diameter was designed and built on the GEM test pad of the David Taylor Model Basin. The facility permits the full-scale testing of internal-flow air delivery systems such as are used in ground effect machines of the annular-jet, plenum-chamber, or captured-air-bubble types.

2. A 64.5-inch overall diameter rotating-diffuser centrifugal fan was tested in the DTMB facility for a total time of about 35 hours. Information on static and total pressure rises and efficiencies was obtained. It was found that the maximum total efficiency of the fan was approximately 85 percent. It was also found that the maximum performance was in very good agreement with the predicted performance. Sound measurements showed a maximum sound pressure level of 90 decibels.

3. There is no agreement at the present time about the relative value of the rotating-diffuser fan and of the radial fan for ground effect machine applications. This matter could easily and cheaply be settled by making the same measurements on a radial fan as were reported here on the rotating-diffuser fan, if the DTMB facility were used.

4. In the present tests, little attention was paid to the properties of the internal-flow system downstream of the fan. The DTMB facility is ideally suited to the development of efficient GEM internal-flow systems. The entire internal ducting system configuration can be simulated at close to full scale, and its performance can be optimized by trial and error modifications of the ducting components.

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13. ABSTRACT
The results of the aerodynamic testing of a 66-inch diameter rotating-diffuser centrifugal fan for internal flow airborne applications are presented.
To accommodate the testing of this unit, a testing facility had to be designed and built. Because of the availability of a test pad and high-power variable frequency electric motor equipment, the facility was located on the grounds of the Aerodynamics Laboratory of the U. S. Navy David Taylor Model Basin. The test stand and its calibration are described, and the results are presented. Results of the tests indicate a peak total pressure efficiency of 85 percent. This correlates with model fan information, including an experimentally determined scale effect.
The RD fan is thus found to be an efficient, robust, stable air mover for air cushion vehicles and for other lightweight airborne applications.
A limited noise survey investigation was made. It indicated low noise levels for the RD fan.

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