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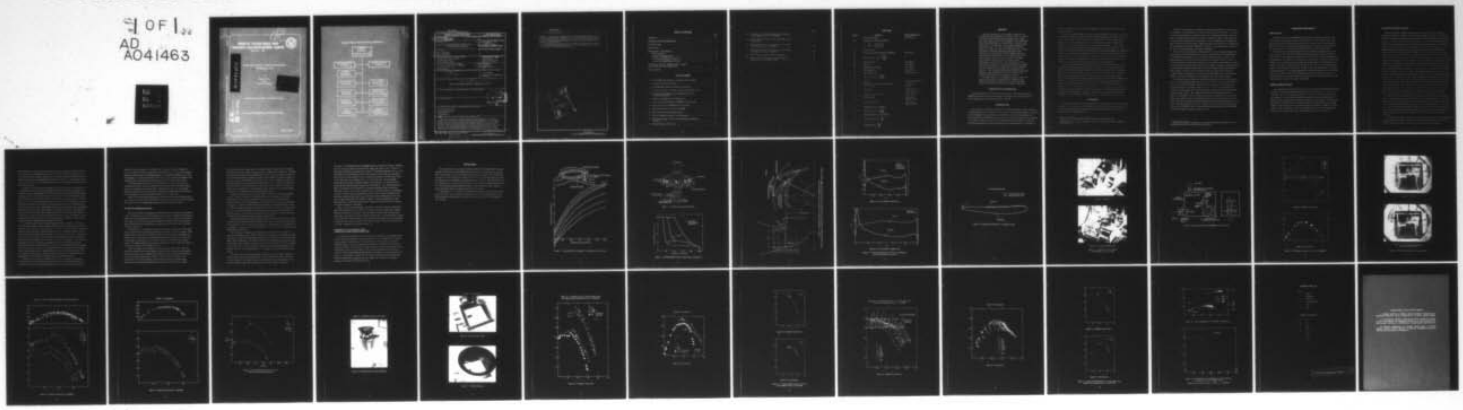

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DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Md. 20084

STATIC EVALUATION OF A CIRCULATION CONTROL CENTRIFUGAL FAN

by

Roger J. Furey

and

Robert E. Whitehead

AVIATION AND SURFACE EFFECTS DEPARTMENT

June 1977

Report 77-0051

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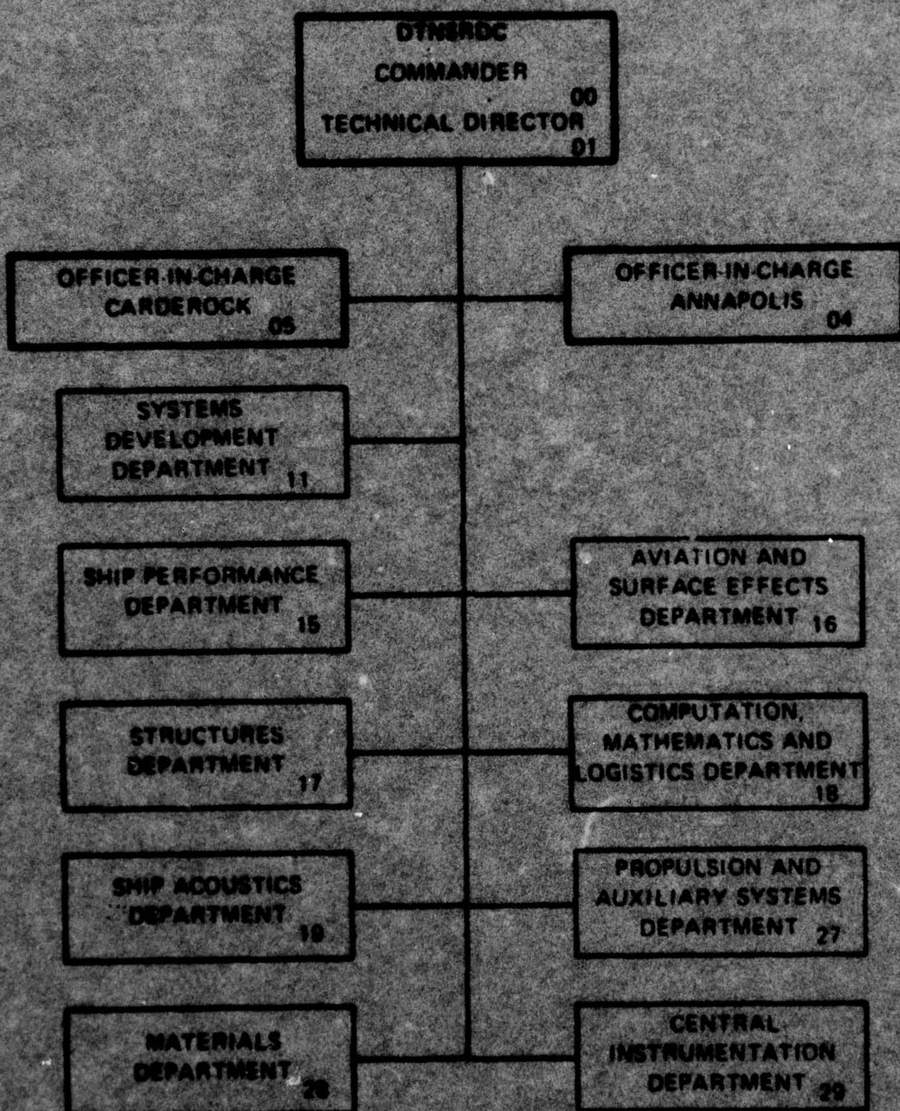

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The static characteristics of a circulation control (CC) fan were determined to demonstrate the feasibility of the CC concept as a means of meeting the lift system requirements of a large, open ocean capable, surface effect ship (SES). These requirements being variable performance, at constant RPM, of sufficient range to provide for heave alleviation when operating at high speeds in advanced sea-states. The scope of the program included two solidity ratios within the model centrifugal impeller and, in effect, two volutes. The better performing combination of these variations was the low solidity ($\sigma = 0.65$) sigma impeller mated with a reduced internal volume volute. This fan demonstrated a flow rate increase of 100 percent over that achieved at the design point, through increasing the flow of control air, while next page

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maintaining a constant head rise. The peak efficiency of this combination was 83 percent. From this peak efficiency, achieved with a moderate amount of control air, the efficiency dropped to a low of 65 percent when operating with a maximum flow of control air. It is shown that the most likely demands of the heave alleviation system would allow for the fan to operate at the highest efficiency possible for the flow rate required.

The high solidity (σ = 1.3) impeller was found to produce an increase in flow rate of 50 percent over that achieved at the design point, through increased control air, and did not achieve as high an efficiency as that of the lower solidity configuration.

Sigma

TABLE OF CONTENTS

LIST OF FIGURES

Page

NOTATION

Symbol	Definition	Units of Measurement
C	Vane chord length	feet (meters)
C_μ	Momentum or blowing coefficient; $C_\mu = \frac{\dot{m} V_j}{q_2 S} = 2\left(\frac{h_j}{C}\right)\left(\frac{V_j}{U_2}\right)^2$	
C_ℓ	Lift coefficient	
b_2	Vane span at outside diameter of impeller	feet (meters)
D_s	Specific diameter; $D_s = \frac{D_2 H^{1/4}}{\sqrt{Q}}$	
D_2	Impeller diameter	feet (meters)
H	Head rise	feet (meters)
$H_{1/3}$	Significant Wave Height	feet (meters)
h_j	Slot height of jet	feet (meters)
N	Fan speed (rpm)	*revolutions/minute*
N_s	Specific speed: $N_s = \frac{N\sqrt{Q}}{H^{3/4}}$	
P	Pressure	pounds/foot^2 (Pascals)
q_2	Impeller exit reference dynamic pressure $q_2 = 1/2\ \rho U_2^2$	pounds/(foot)^2 (Pascals)
Q	Flow rate	cubic feet/second (cubic meters/second)
S	Reference area; $S = Z\ b_2 C$	(feet)^2 (meters)^2
U_2	Fan tip speed	feet/second (meters/second)
V_j	Jet velocity at slot	feet/second (meters/second)
Z	Number of vanes in impeller	
η	Fan efficiency $\eta = \frac{PQ}{550\ HP}$	
ϕ	Flow coefficient; $\phi = Q/\pi D_2 b_2 U_2$	
ψ	Head coefficient; $\psi = \frac{Hg}{U_2^2}$	
σ	Solidity ratio $\sigma = \frac{ZC}{\pi D_2}$	

ABSTRACT

The static characteristics of a circulation control (CC) fan were determined to demonstrate the feasibility of the CC concept as a means of meeting the lift system requirements of a large, open ocean capable, surface effect ship (SES). These requirements being variable performance, at constant RPM, of sufficient range to provide for heave alleviation when operating at high speeds in advanced sea-states. The scope of the program included two solidity ratios within the model centrifugal impeller and, in effect, two volutes. The better performing combination of these variations was the low solidity (σ = 0.65) impeller mated with a reduced internal volume volute. This fan demonstrated a flow rate increase of 100 percent over that achieved at the design point, through increasing the flow of control air, while maintaining a constant head rise. The peak efficiency of this combination was 83 percent. From this peak efficiency, achieved with a moderate amount of control air, the efficiency dropped to a low of 65 percent when operating with a maximum flow of control air. It is shown that the most likely demands of the heave alleviation system would allow for the fan to operate at the highest efficiency possible for the flow rate required.

The high solidity (σ = 1.3) impeller was found to produce an increase in flow rate of 50 percent over that achieved at the design point, through increased control air, and did not achieve as high an efficiency as that of the lower solidity configuration.

ADMINISTRATIVE INFORMATION

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INTRODUCTION

The concept of a circulation control (CC) fan is under development to meet the lift and habitability requirements of a large, open ocean capable, surface effect ship (SES). Circulation control technology has been underway at the David W. Taylor Naval Ship Research and Development Center (DTNSRDC) for some years now and recently culminated in the awarding of a contract to utilize this concept in constructing the full-scale prototype of a helicopter. The features that make this approach promising for helicopter application, i.e., simplicity of operation and rapid response, appear equally promising for meeting the lift system requirements of the SES.

The basic CC airfoil* employs a jet of air blowing tangentially to the airfoil surface near the trailing edge. With a properly shaped and rounded trailing edge, this jet will stay attached to the surface because of the Coanda effect. Varying the flow of this jet will produce lift amplification, as shown in Figure 1. The CC fan studied here is simply a centrifugal fan employing a cascade of such airfoils; the concept is illustrated in Figure 2. Note that the control air passes through the hollow shaft, through passages cut in the impeller base, into individual plenums in each blade, and out through a slot in the blade trailing edge. Modulating the flow of control air to these blades will, in turn, vary the fan output such that a nearly constant head rise can be maintained in actual SES application for a large range of flow rates without changing rpm. This would make it possible to maintain heave acceleration within limits that would not detract from the performance of crew and equipment.

The operational environment of the SES is such that provisions must be made to meet habitability limits during high-speed operations over rough seas. The conventional long-duration habitability limit in heave acceleration is 0.10 g, but that level can be exceeded for short periods. In current SES-type vehicles, the acceleration magnitudes are controlled by heave attenuators which simply expel overboard any excess air in the cushion. Figure 3, shows a typical condition for a large scale SES and indicates the 0.10- and 0.20-g operational limits as a function of sea state and craft velocity.[1] In order to meet these limits, the fan back pressure must be maintained in a range from ± 10 to 20 percent of the nominal operating pressure. It is believed that these demands can be met quite efficiently with a lift system which incorporates the CC fan. The test and data to be discussed here represent the first steps in demonstrating that capability. This first phase is meant only to demonstrate feasibility and not the ultimate performance that can be achieved from a CC fan system.

FAN DESIGN

The primary consideration for this first-generation, centrifugal-type, CC fan was simplicity of design for ease of fabrication. Fan dimensions were governed by the fan test facility to be used and the availability of an existing volute which had been employed in a recent test of a scale model of the 100B SES lift fan.

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*Reported informally by R.J. Englar in NSRDC Technical Note AL-201 (May 1972).

[1] Luscher, W.P. et al., "*A Study of Characteristics of Lift Fans for Surface Effect Ships*," ALRC Report 9725-234 (Apr 1973).

The specific speed range for a centrifugal fan suitable for application to a large size SES is considered to be $150 \leq N_s \leq 200$. A specific diameter range of $0.95 \leq D_s \leq 1.1$ is found to offer the best efficiency for a specific speed of 170. The head rise required at the design point for a large size SES would be on the order of 300 psf (14.7 KPa). These restrictions, along with those associated with CC requirements - such as suitable ratios for trailing edge radius-to-blade chord and slot height-to-trailing edge radius were provided to Dr. G. Wislicenus* who subsequently produced the design shown in Figure 4. Some modifications, pertinent to the needs of circulation control, were made to this design.

Since the CC concept depends on boundary-layer control, it is extremely important for satisfactory and efficient performance that the flow remain attached on the impeller blades. The geometrical requirements for the centrifugal fan application leads to a blade section profile dissimilar to those that form the experimental technology base of the concept. To ensure that the present blade design would perform as expected, it was decided to investigate the fan flow field by using the best theoretical tools available. A group of computer programs was obtained from NASA Lewis Research Center for this purpose. When used collectively, these programs can predict the fan internal flow field for a given set of fan operating conditions within the assumptions of an inviscid ideal fluid. Blade surface velocity profiles similar to those shown in Figure 5 can be obtained as part of the problem solution.

Several blade section modifications were made based on the results of this theoretical study: (1) the leading edge radius of the blade was increased to reduce velocity gradients in this area and lessen the possibility of leading edge flow separation, (2) the suction side thickness distribution was modified slightly to reduce an adverse velocity gradient in the vicinity of the slot location, and (3) the location of the slot was based on the theoretically predicted velocity profiles. The modifications made to the blade design are shown in Figure 6 as dashed lines. The most effective jet turning for minimum power was obtained with the slot located just aft of the minimum pressure point on the aft suction side of the blade. The slot was located at 97-percent chord based on the theoretical predictions discussed above. With these modifications, the fan of Figure 4 was subjected to detailed design and fabrication at DTNSRDC. Figure 7 shows the finished product mounted in the DTNSRDC fan test facility prior to being encased in the volute.

*Recently retired as Head of the Department of Aeronautical Engineering and the Water Tunnel of the Applied Research Laboratory at the Pennsylvania State University.

PROCEDURES AND RESULTS

TEST FACILITY

The basic features of the DTNSRDC fan test facility are illustrated in Figure 8. The 860-ft^3 plenum is positioned above the test section of the 8- x 10-ft subsonic wind tunnel. The fan itself is enclosed within the plenum, and the fan inlet is open to the tunnel below. Flow rate and head rise are determined through pressure and temperature measurements at the fan inlet and the exhaust portion of the volute (Stations 3 and 4, respectively).

The orifice plate, located downstream of the 30-in.-diameter discharge valve, was used only as a check on the mass flow calculations as determined at the fan inlet and exhaust. (For most of the test program, the orifice plates were not in place.) Under normal operation the supplementary air supply valve is closed and fan back pressure is varied by sequencing the discharge valve in small increments from fully open to fully closed. The supplementary air supply is used only to simulate back flow through the fan. Control air for the individual fan blades is provided through a hollow shaft, rotating with its base protruding into the control air plenum (Figure 7b), and passages drilled in the impeller base leading to each of the blades. The usual test sequence called for establishing fan rpm and control air flow rate and for monitoring both static and total pressure and temperature at the fan inlet and exhaust, along with fan torque, at each of the exhaust valve settings.

DESIGN CONSIDERATIONS

The original design point for the CC fan was aimed at meeting the following requirements: a head rise of 300 psf (14.4 KPa) and, at this scale, a flow rate of 260 cfs (7.8 m^3/s) at 5300 rpm. The cost for design and fabrication of a fan capable of operating at that speed was so high that it was considered justifiable to establish a less severe performance goal inasmuch as the test objective was merely to demonstrate concept feasibility. With operating speed restricted to 4000 rpm, the fan could be fabricated by using conventional machining methods and without the need for composite materials as required for a higher performance machine. Other considerations, such as the available flow rate of control air and vibration in the drive belt at high torque, led to conducting most of the test at 2200 rpm.

CC FAN WITH ORIGINAL VOLUTE

Figure 9a shows the head/flow characteristics obtained at 2200 rpm. (The head coefficient and efficiency of Figure 9 and all subsequent figures are based on total pressure.) It indicates the significant range in performance that is possible through increased control air blowing. The starred symbol represents the design point to be attained without blowing. As indicated, the actual performance without blowing was just under this point. The difference between goal and achievement may be attributable to the fan volute used in this test. The original volute was too large for the size and specific speed of the CC impeller, but this handicap was felt to be acceptable since the objective of this phase was simply a proof-of-concept for the CC approach. However, a number of features in the data can be attributed to this mismatch. First and foremost is that related to fan efficiency. The maximum efficiency achieved at 2200 rpm was less than 70 percent. To check whether the oversized volute would contribute to considerable recirculation within it, tufts were taped along the inside walls of volute and observed during fan operation: see Figure 10. The view in Figure 10a is looking down through the exhaust opening of the volute with no flow. The impeller is to the upper left of the opening and the inside contour of the bellmouth, leading to the inlet, is seen to the right. Tufts were attached to the bellmouth, along the walls of the volute, and just above the "cutwater" or tongue of the volute. Figure 10b shows the arrangement while operating at 1000 rpm, no control air ($C_\mu = 0.0$), and back pressure such that the fan operated at its point of best efficiency. Note that tufts attached to the bellmouth and along the cutwater all streamed toward the inside of the volute. For the most part, those attached to the wall, at about the same plane as the "cutwater," also streamed toward the inside of the volute. All of this indicates an excessive amount of recirculation within the volute and thereby reduced efficiency, even at what was the most efficient operating point of the system.

Another factor which contributed to the low efficiency was Reynolds number effects. Figure 11 shows fan performance data at two speeds. The maximum efficiency achieved at 1500 rpm was about 62 percent whereas that at 2500 rpm was on the order of 70 percent. The belt vibration mentioned earlier prevented operation at higher blowing rates than shown for 2500 rpm. The trend of these figures emphasized that some penalty was involved in the restriction to 2200 rpm when obtaining the full range of performance through blowing. The maximum efficiency of the fan in this configuration is at least the 70 percent observed at 2500 rpm.

As determined for Figures 9 and 11, fan efficiency includes the effects of the compressor power required for the control air. However, it should be noted that the regions of low efficiency are, for the most part, out of the range of interest for SES operation.

The requirement of a nearly constant head rise through the full range of flow rate limits the area of interest. A meaningful load line would be one passing through the design point at a near-constant head rise, as shown in Figure 9a. The shaded symbols in the efficiency data of Figure 9b indicate the corresponding load line in terms of efficiency. The operating efficiency is seen to be at or very near the maximum efficiency for that particular flow rate throughout the range of operation.

A feature of the CC fan in this configuration is that peak efficiency is maintained over a considerable range of flow rate and through moderate levels of blowing. This suggests that the design or normal operating point can be met at different speed and blowing rates without sacrificing efficiency. The asterisk on Figure 9a represents a point at which the requirement of 300 psf (4.2 Pa) and 260 cfs (7.8 m^3/s) can be met with an operating speed of 5100 rpm and $C_\mu = 0.005$. The load line through this point shows that the fan is capable of increasing its flow rate by better than 50 percent of normal operating capacity and of maintaining its head rise all the way to cutoff. The flagged symbols of Figure 9b represent the corresponding efficiencies and indicate there is little penalty for operating in this manner. Operating along this load line, rather than through the original design point, allows more positive control in that the flow of control air can be reduced when attempting to reduce fan outflow. Its static characteristics indicate that the CC fan has a sufficient range of operation to cope with the demands imposed by high-speed SES operations in rough seas.

Two additional points need to be made regarding the data of Figure 9a. It has been shown that the compressor power required to maintain blowing has little effect on efficiency within the range of interest for SES operations. Nevertheless, it is apparent that a considerable reduction is associated with the higher blowing rates. This is considered partly due to flow separation within the impeller and enhancement of the separation by blowing. A tufted wand was used to examine the flow at the periphery of the rotating impeller. It was noted that when the tuft was brought close to the shroud or inlet side of the impeller, it was sucked into the impeller. This was attributed to local separation of the flow from the shroud as it negotiated the turn from the axial direction at the inlet to the radial direction in the impeller. The Coanda jet of the control air, at the outside trailing edge of the blade, was discharging into this separated region; and at higher blowing rates, that caused further separation in this area, and reduced the efficiency. (The lower solidity configuration combined with a volute modification, to be discussed later, eliminated this region of flow separation.)

Another point of interest is the limiting effect of increased blowing. The minimum cross-sectional area in the control air circuit occurred at the 0.75-in.-diameter throat of the venturi meter. There was a choked flow condition at this point when a mass flow rate of about 0.25 lb/sec was reached. At 2200 rpm, this occurred at a blowing coefficient of $C_\mu = 0.05$. Beyond this point, shock losses occurred within the system and the increment in performance improvement with blowing decreased. At the highest blowing rate, $C_\mu = 0.06$ corresponding to a control air mass flow of 0.31 lb/sec, sonic conditions were reached at the trailing edge slots and performance did not appear to be enhanced beyond this point.

The trailing edge Coanda surface can be so designed that a supersonic jet will stay attached and not shock down until well around to the pressure side of the blade; however, this was not attempted in the current design. Although more mass flow can be pushed through the system after sonic flow has been reached at the trailing edge slots, this appears to be a limiting point for increased blade circulation in the current design.

Figure 12 shows the performance range for no blowing and maximum blowing in terms of actual head rise and flow rate. Note that in the region of the best efficiency point, between 90 to 140 cfs, the use of trailing edge blowing more than doubled the head rise.

CC FAN WITH MODIFIED VOLUTE

Since the first phase of this program was intended merely to demonstrate the feasibility of the CC-centrifugal fan concept, some shortcuts were taken in assembling the demonstration model. Foremost among these was the utilization of an available casing for the CC impeller. This casing, or volute, was oversized for the impeller and although head/flow characteristics proved sufficient, the associated efficiency reached only a moderate level (near 70 percent), as shown in Figure 11b.

The significance of an efficiently operating fan prompted a modification to the original volute to bring it more in keeping with the specific speed and size of the CC impeller. These modifications consisted simply of inserting a plywood panel to decrease volute volume and mahogany sections to reduce the exhaust opening and bring the tongue of the volute closer to the impeller. Additionally, it was necessary to make a new bellmouth for the volute exhaust. The original bellmouth and volute have already been shown in Figure 10. Figure 13 presents various aspects of the modified version.

Figure 14 compares fan performance at 2200 rpm with the two volutes. Note from Figure 14a that the flow rate, proceeding horizontally from the design point represented by the starred symbol, can be increased by better than 50 percent of that attained at design, as was the case with the original volute. Figure 14 enables a direct comparison of the characteristics of the two configurations. The impeller was designed to deliver design conditions with no blowing and should therefore operate efficiently for that condition.

The asterisk indicates the design points to be attained without blowing. With the modified volute, the fan was able to meet this point whereas the original volute undershot it. Moreover, the stall was more gradual with the modified volute. Both of these conditions are indicative of a more efficient mating of impeller and casing. The remaining characteristics of Figure 14 are those concerned with maximum control air for the respective configurations; Figure 14b shows the efficiencies associated with these characteristics. For no-control air, fan efficiency with the modified volute peaked about five points higher (about 72 percent) and this peak efficiency was maintained over a somewhat greater range. Differences in maximum blowing rate for the two configurations also favored the modified volute.

The data discussed to this point were all obtained at 2200 rpm. It had been intended to repeat a number of these runs at a higher speed in order to determine any Reynolds number effects. That was not possible, however, because the teflon seal between the stationary inlet and the rotating impeller was lost near the end of the first run at 2800 rpm. The single run completed at 2800 rpm did however produce some significant results. Figure 15 shows the head-flow characteristic and the associated efficiency for the no control air case ($C_\mu = 0.0$). The head-flow behavior was much the same as that achieved at 2200 rpm, but the peak efficiency was improved about 4 percent over that achieved at the lower speed and was about 8 percent higher than reached with the original volute.

The aforementioned modifications made it necessary to remove the entire volute in order to replace the seal, rather than simply removing only the inlet, as could have been done prior to the modification. Because of the time entailed in accomplishing this, it was considered to be opportune to configure a lower solidity impeller ($\sigma = 0.65$ rather than 1.3 and half as many impeller blades at this time).

The low solidity configuration was subjected to the same series of tests. Figure 16 indicates the performance range achieved. Note that the no-blowing case fell below the design point and, as would be expected, did not develop as much of a head rise as obtained with the higher solidity configuration. The flow rate, however, was slightly greater than obtained with higher solidity and the efficiency reached a level of 76 percent. However, the (b.e.p.) shifted to a higher flow rate, as can be observed by noting the location of the design point (starred symbol). The b.e.p. for the high solidity ($\sigma = 1.3$) impeller occurred at approximately that location.

The range in flow rate through blowing attained for the lower solidity configuration is much more suitable for SES requirements. It can be increased by nearly 100 percent over that achieved at the design condition while maintaining the required head rise. This is nearly double the range that was possible with the higher solidity configuration.

The penalty for increased performance through blowing, as measured in terms of efficiency, was less for this low solidity case than for the original high solidity version. Figure 16b shows that the efficiency remained above 70 percent for blowing coefficients up to $C_\mu = 0.03$. A desirable mode of SES operation which could capitalize on this condition is depicted in Figure 16a; the asterisk represents a point at which the requirements of 300 psf and 260 cfs can be achieved at a flow coefficient of $C_\mu = 0.02$ and 5100 rpm. The characteristic represented by this blowing coefficient attained a peak efficiency of 74 percent just two percentage points less than that achieved in the no-blowing case. The cross-hatched area of Figure 16a represents the ±10 percent of the nominal operating pressure that must be maintained in order to achieve the 0.1-g limit in heave acceleration required as a habitability constraint. The flow rate can be double that required at the nominal operating point and still remain within the cross-hatched acceptable pressure range. Operating from this point allows, at least theoretically, the possibility of going to zero flow through the fan while remaining within the prescribed pressure limits. (Dynamic evaluation of the fan is required to establish its behavior in this region.)

Figure 17 indicates the performance characteristics for the low solidity configuration at higher speed (2800 rpm) and no control air. As was noted with the high solidity impeller, a Reynolds number effect was present. The peak efficiency of 83 percent is the measured value with no corrections to account for losses in the belt drive. (The torque meter is located on the same shaft as the drive motor. Control air passage required in the fan shaft precluded locating the torque meter where belt losses could be bypassed.) The power required for the low solidity configuration is indicated in Figure 18.

FEASIBILITY OF THE CONCEPT FOR A LARGE SCALE OPEN OCEAN CAPABLE SES

The head/flow characteristics of a CC fan for a large scale SES can be determined from results of the present model tests; the lower solidity ($\sigma = 0.65$) configuration is considered more promising for this application. The peak efficiency of 83 percent, as measured at 2800 rpm, and $C_\mu = 0.0$, can conservatively be projected to 85 percent for the full-scale version on the basis of test observations. Figure 19 indicates that the peak efficiency would be expected to occur at some moderate level of control air blowing rather than with no control air. (The loss associated with the belt drive was included in the measured value of 83 percent.) This observation, coupled with the usual expected improvements resulting from Reynolds number effects in going to full scale, suggest that 85 percent is a moderate projection of efficiency for the full-scale CC fan.

CONCLUSIONS

The static evaluation of a circulation control centrifugal fan has demonstrated the unique capability of this concept in meeting the heave alleviation needs of a large open ocean capable SES. A low solidity ($\sigma = 0.65$) seven bladed, two foot diameter, impeller model demonstrated the ability to increase its flow rate by 100 percent over that achieved at the design point, while maintaining head rise and RPM, by increasing the flow of control air. The peak efficiency of this configuration was 83 percent with a drop to 65 percent when utilizing maximum control air. The most likely operating mode when used in a heave alleviation system is shown to deliver the highest efficiency attainable for the flow rate required.

A high solidity ($\sigma = 1.3$) version of the fan demonstrated the ability to increase its flow rate by 50 percent over that achieved at the design point, while maintaining head rise and RPM, through the increased flow of control air. The peak efficiency of this configuration was 78 percent.

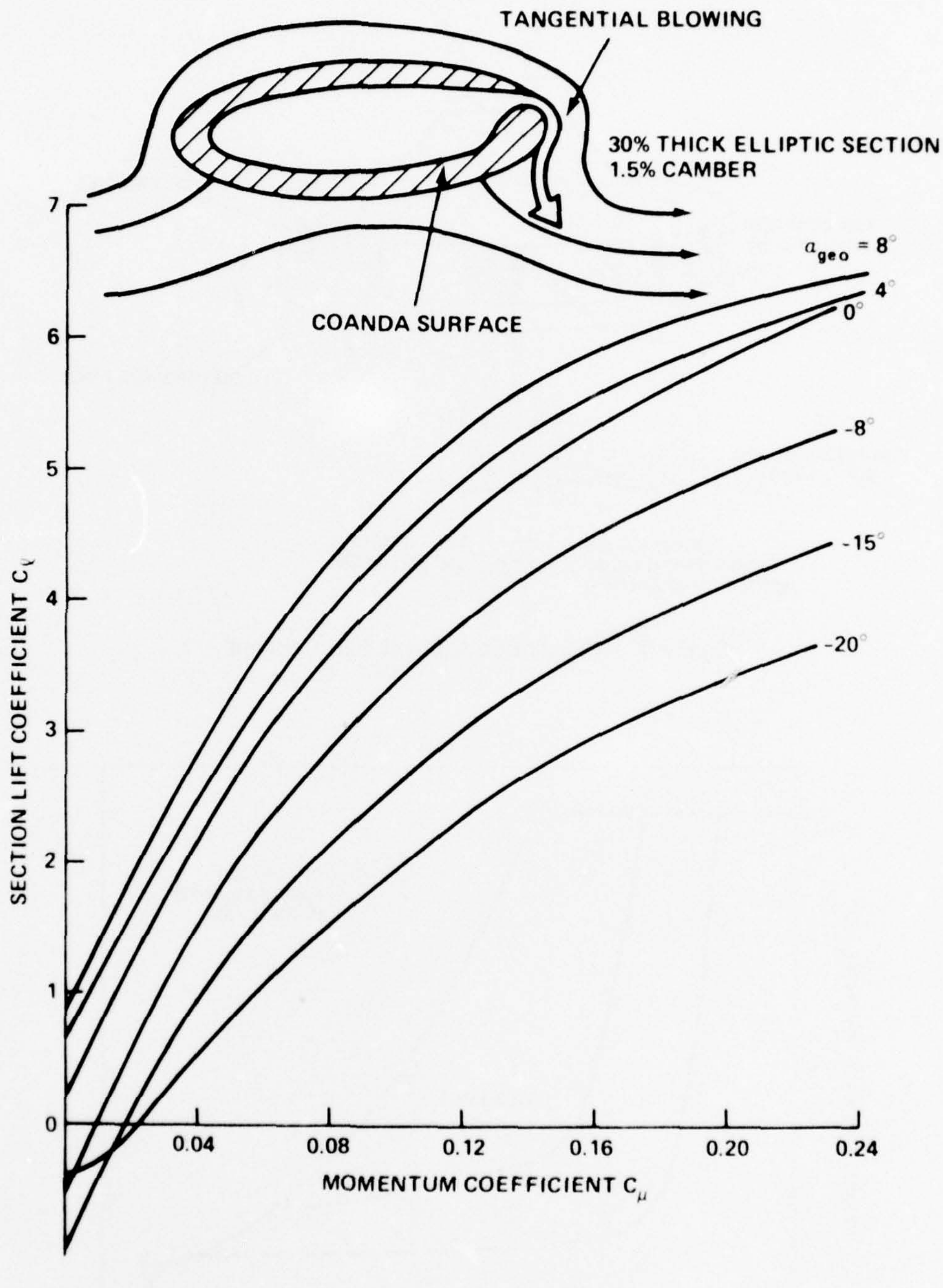

Figure 1 – Typical High Lift Capability of Circulation Control Airfoil

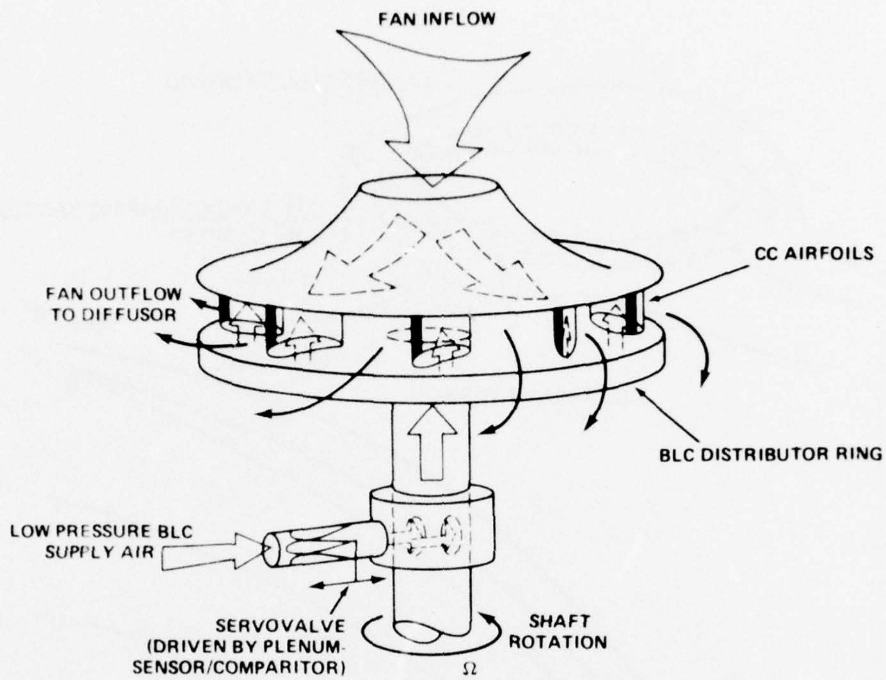

Figure 2 – Circulation Control Fan Concept

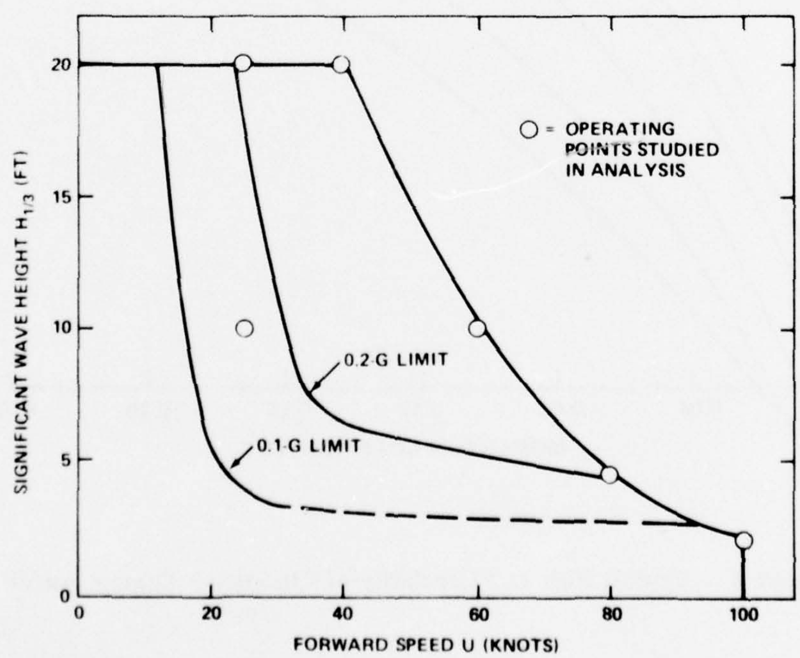

Figure 3 – SES Habitability Limits without Heave Attenuation

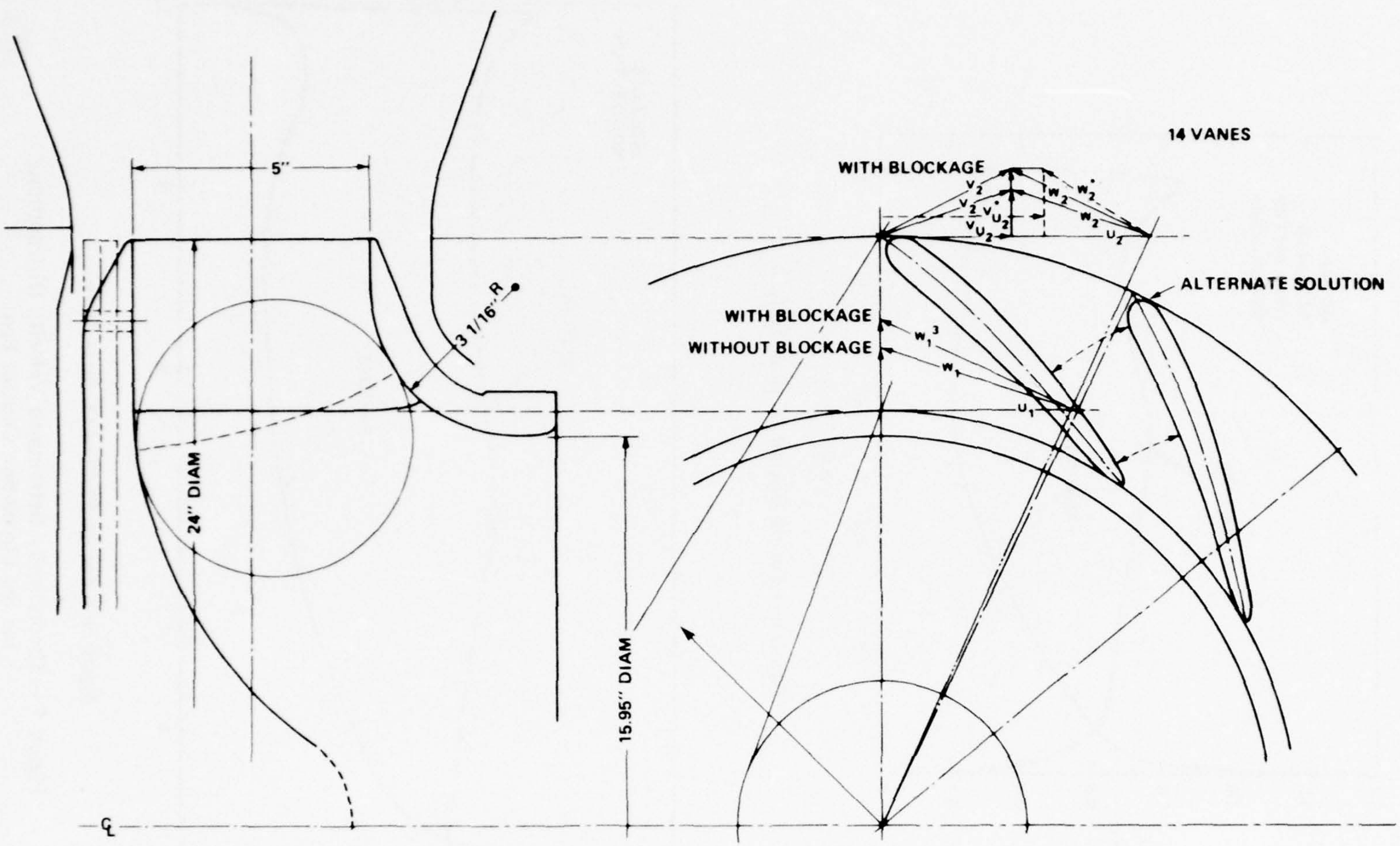

Figure 4 – Preliminary Design for Circulation Control Centrifugal Flow Fan

(Tentative (Wislicensus) design of radial-flow blower with circulation control; full scale for 260 cps, 4000 ft head at 5320 rpm)

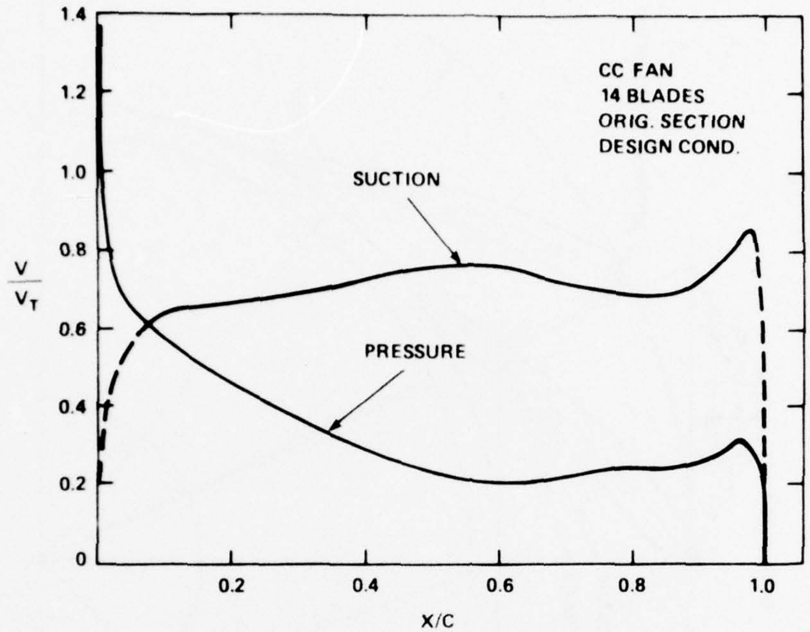

Figure 5a – For the Original CC Blade Section

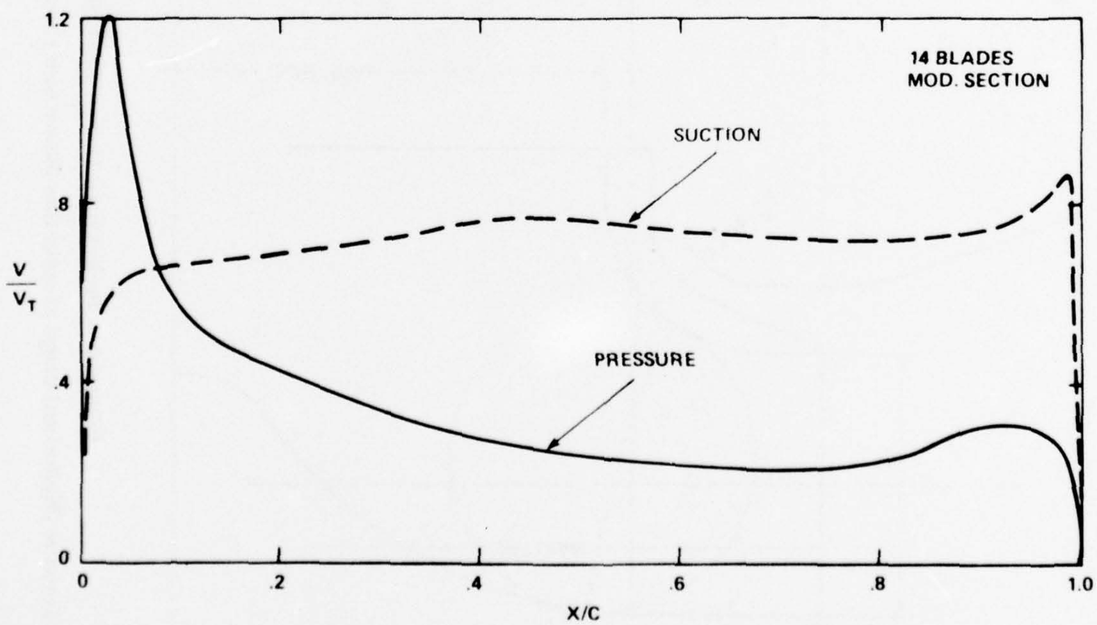

Figure 5b – For the Modified CC Blade Section

Figure 5 – Theoretically Determined Velocity Distribution for the Circulation Control Fan

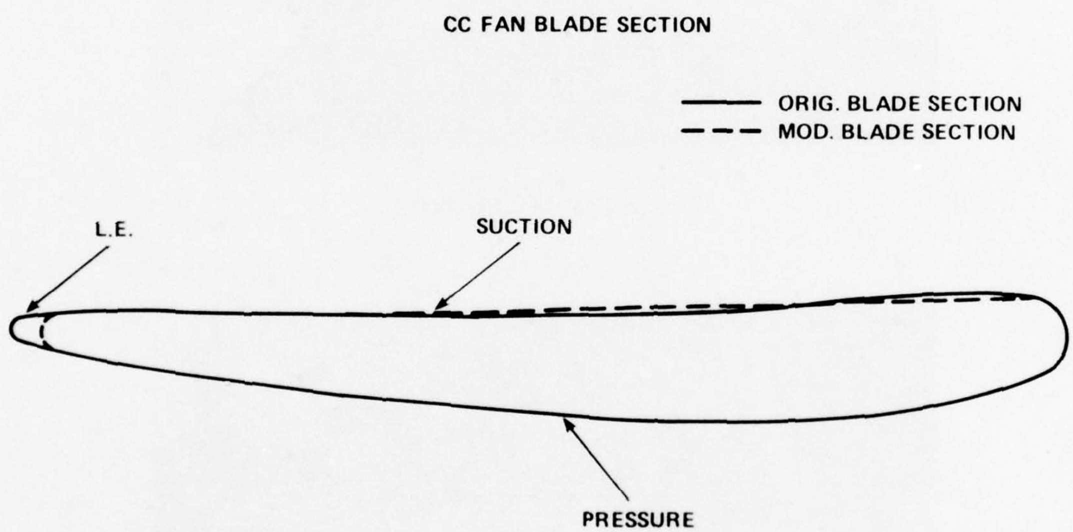

Figure 6 – Original and Modified CC Fan Blade Sections

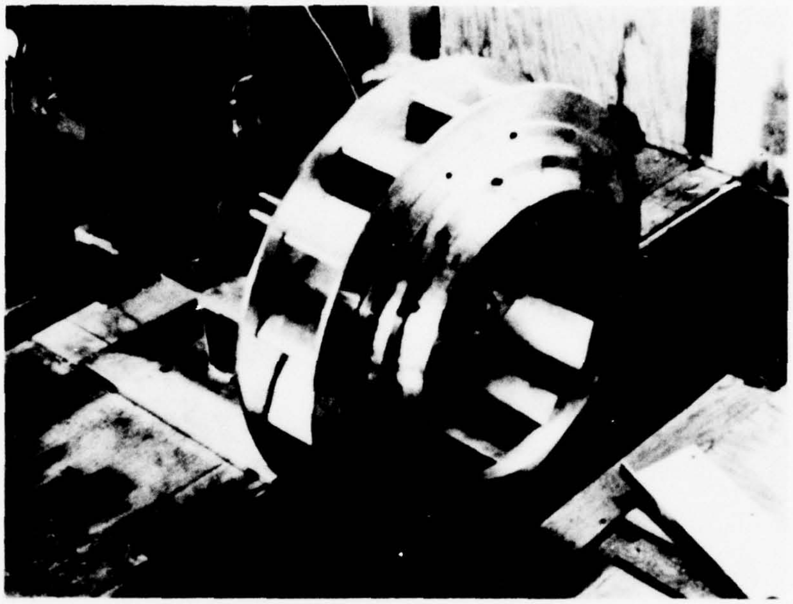

Figure 7a – CC Impeller

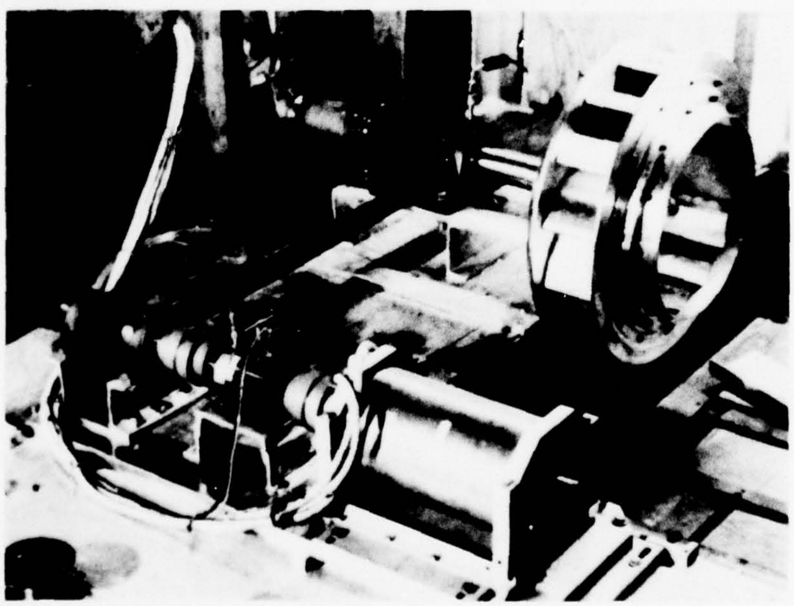

Figure 7b – CC Impeller and Drive Train

Figure 7 – Circulation Control Impeller
in DTNSRDC Fan Test Facility

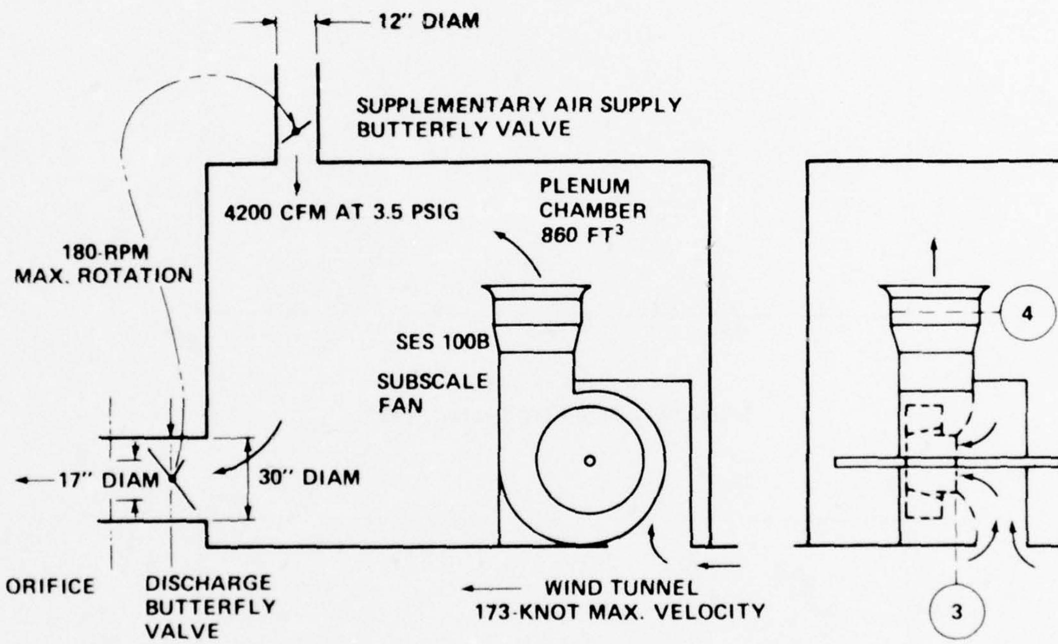

Figure 8 – Basic Features of the DTNSRDC Fan Test Facility

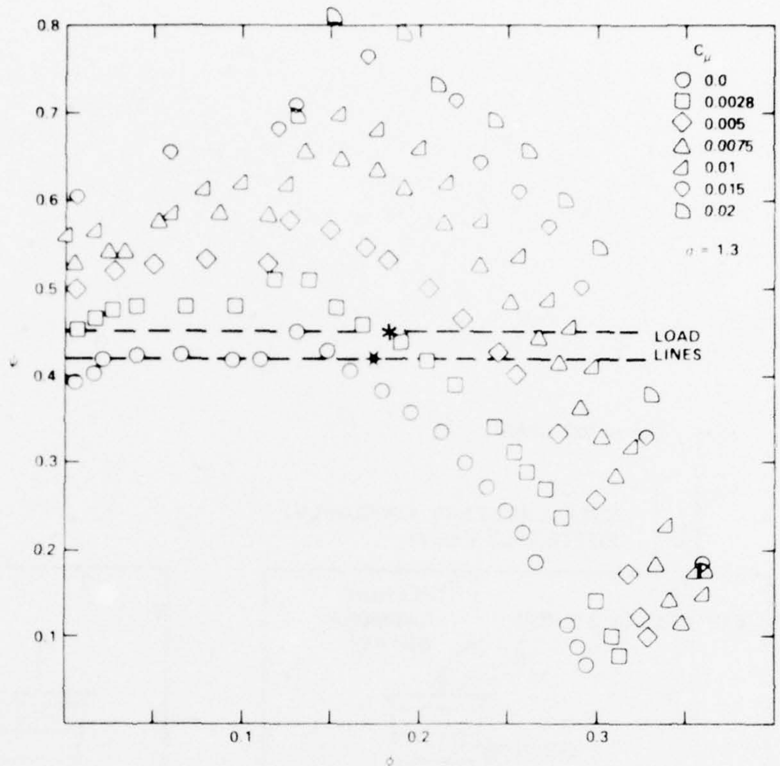

Figure 9a – Head/Flow Characteristics

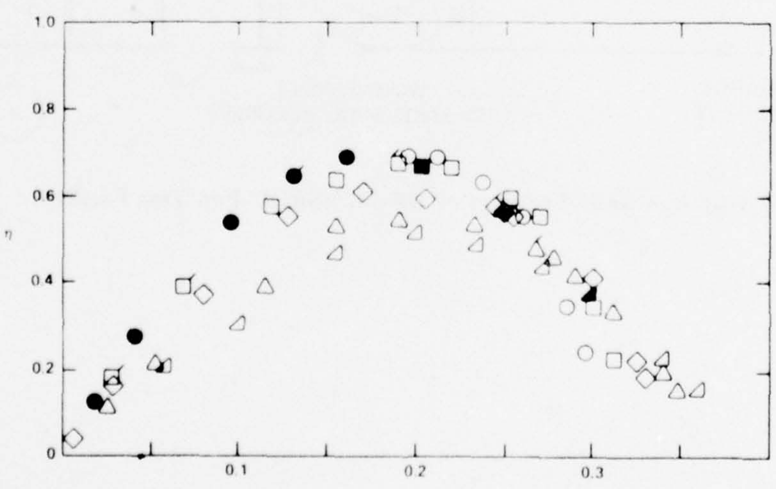

Figure 9b – Fan Efficiency

Figure 9 – Performance Data for CC Fan at 2200 RPM

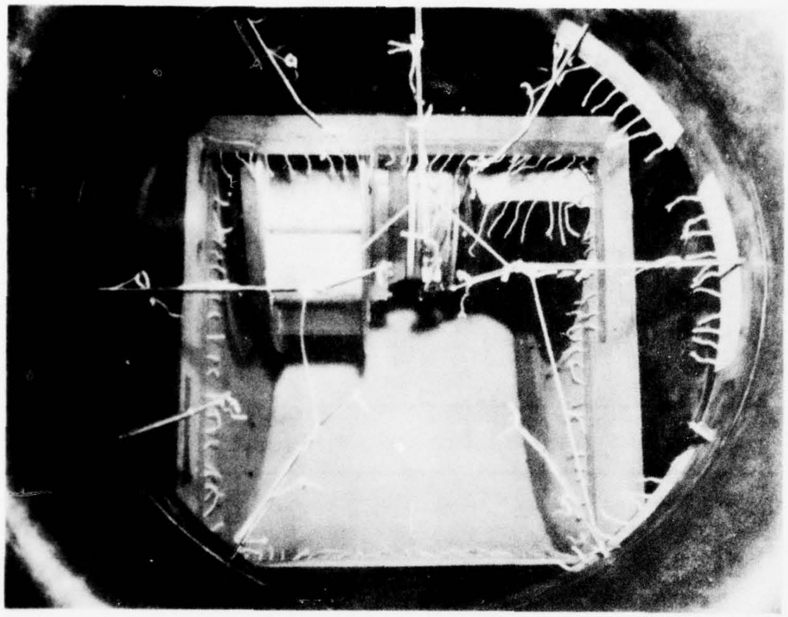

Figure 10a – No Flow

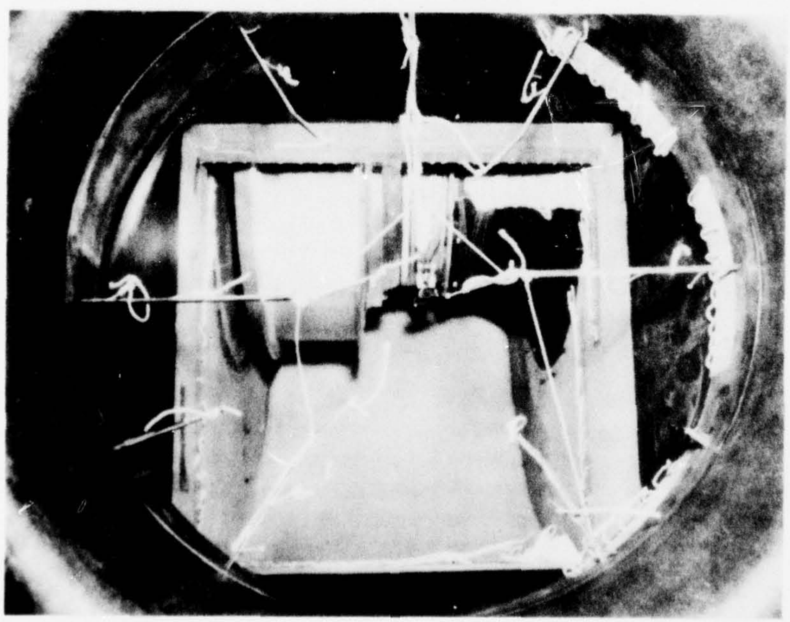

Figure 10b – Fan Operating at 1000 RPM
and Point of Best Efficiency

Figure 10 – Recirculation within the Original Volute

Figure 11 – Effect of Reynolds Number on Fan Performance

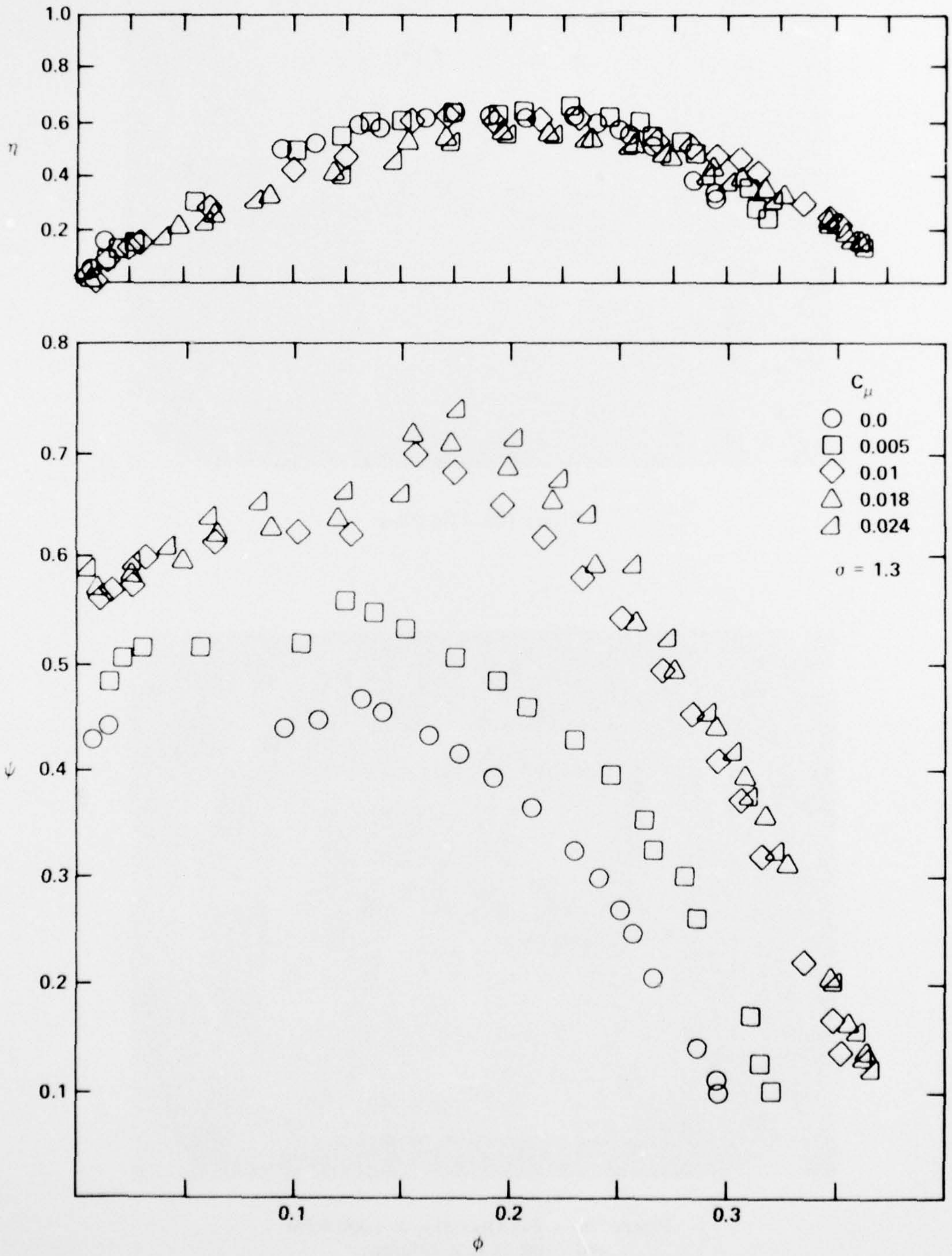

Figure 11a – Head/Flow Characteristics at 1500 RPM

Figure 11 (Continued)

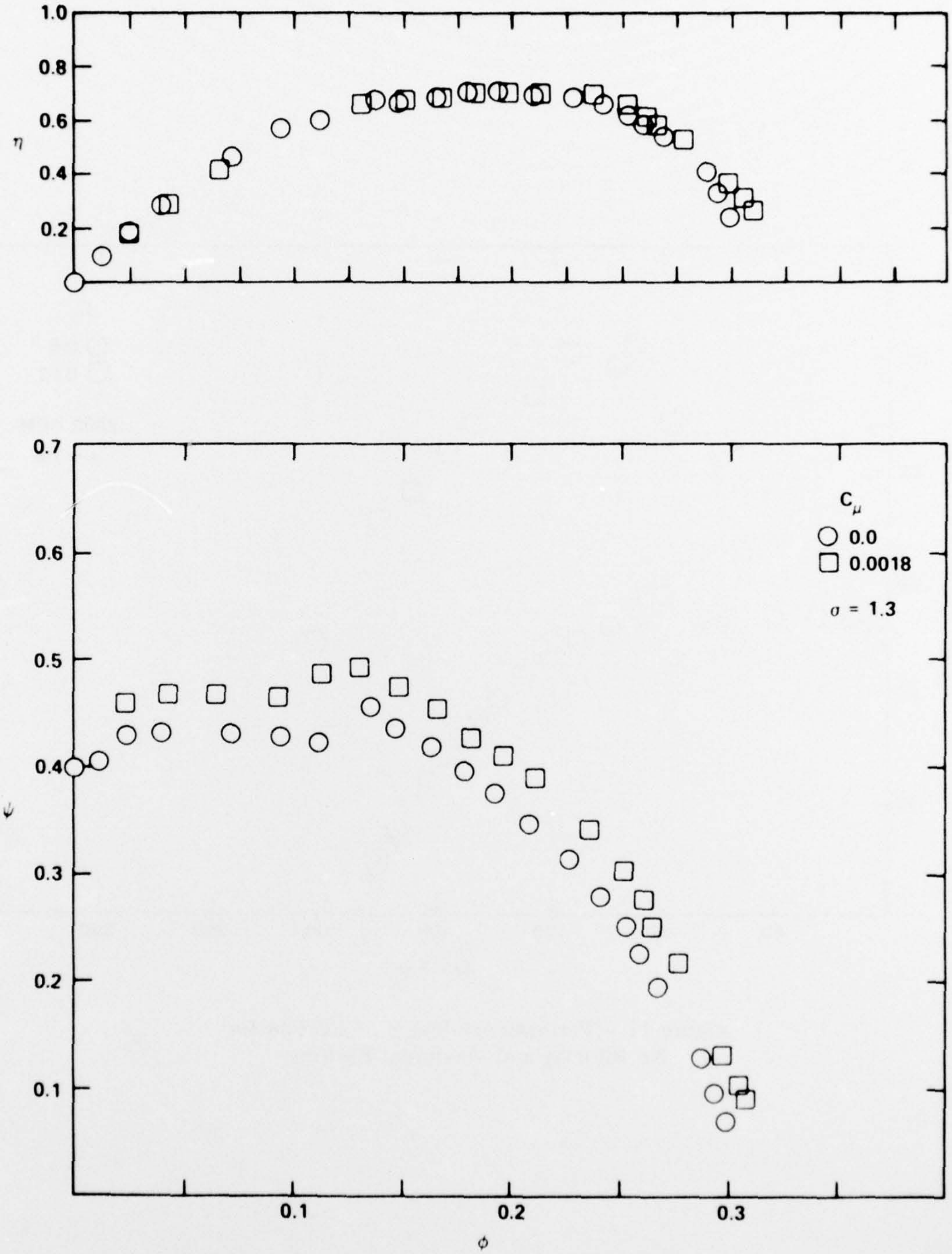

Figure 11b – Head/Flow Characteristics at 2500 RPM

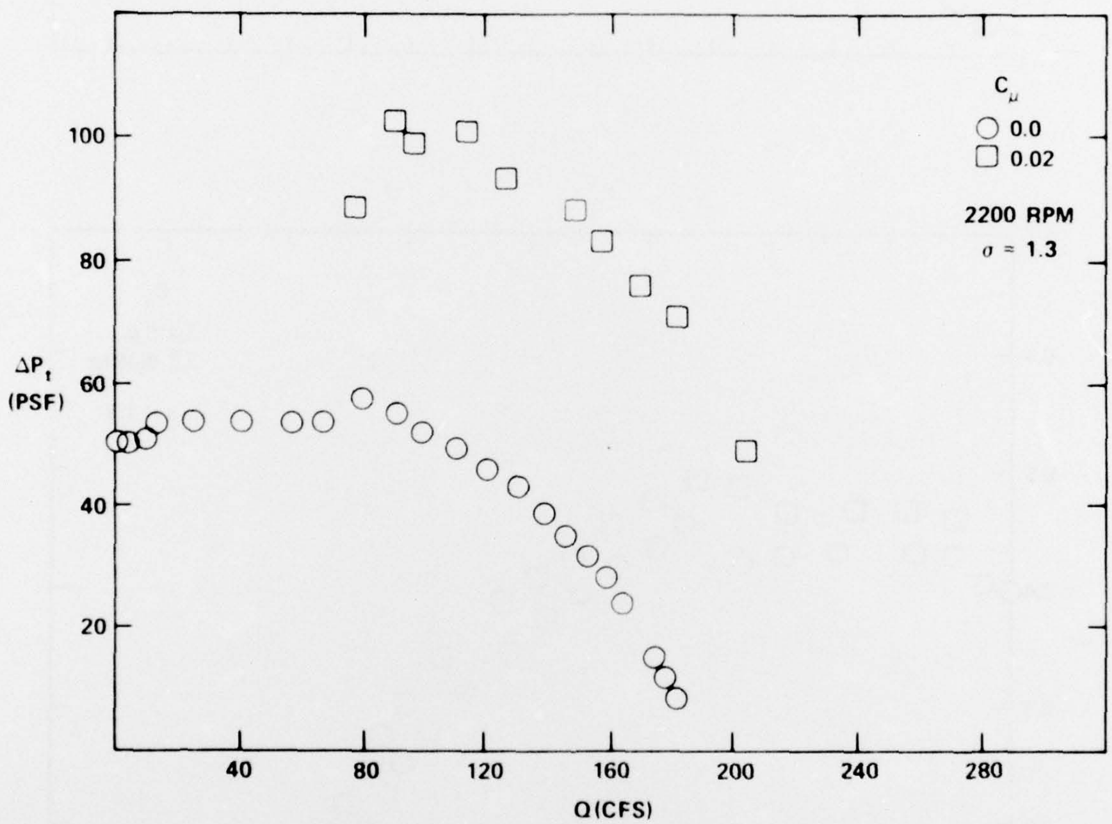

Figure 12 – Performance Range of CC Fan for No Blowing and Maximum Blowing

Figure 13 – Modified Version of the Volute

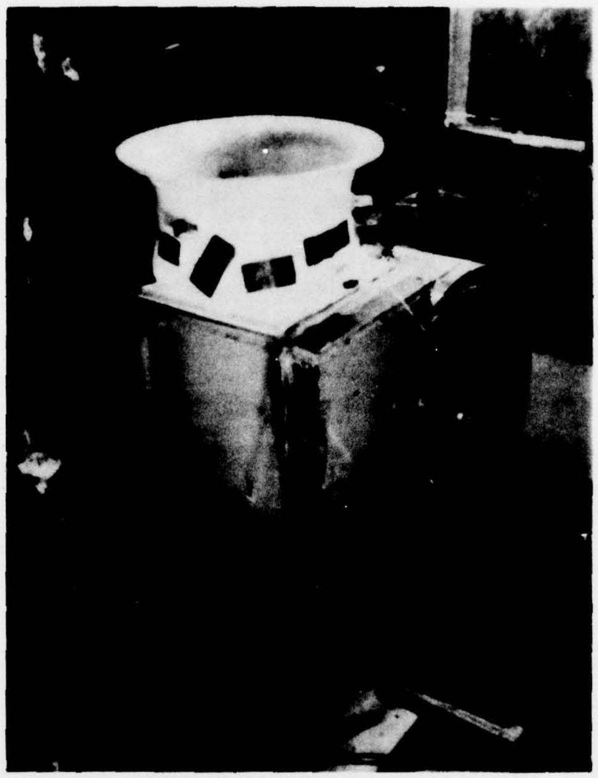

Figure 13a – Reduced Exhaust Opening and Bellmouth

Figure 13 (Continued)

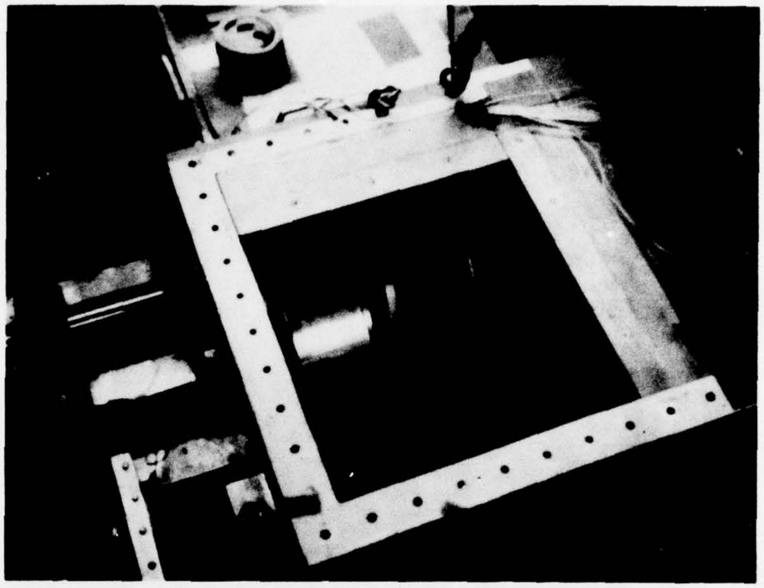

Figure 13b – Reduced Internal Volume

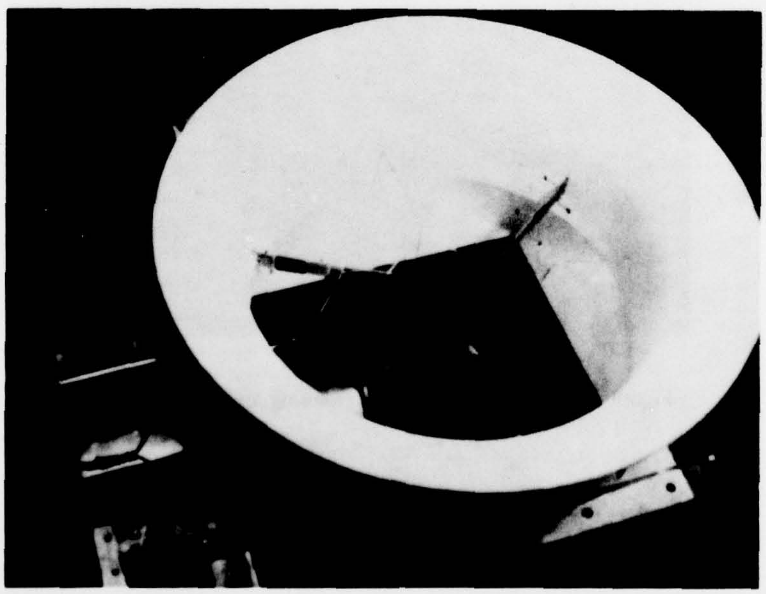

Figure 13c – Matching Bellmouth

Figure 14 – Comparison of CC Fan Performance with the Original and the Modified Volute at 2200 RPM

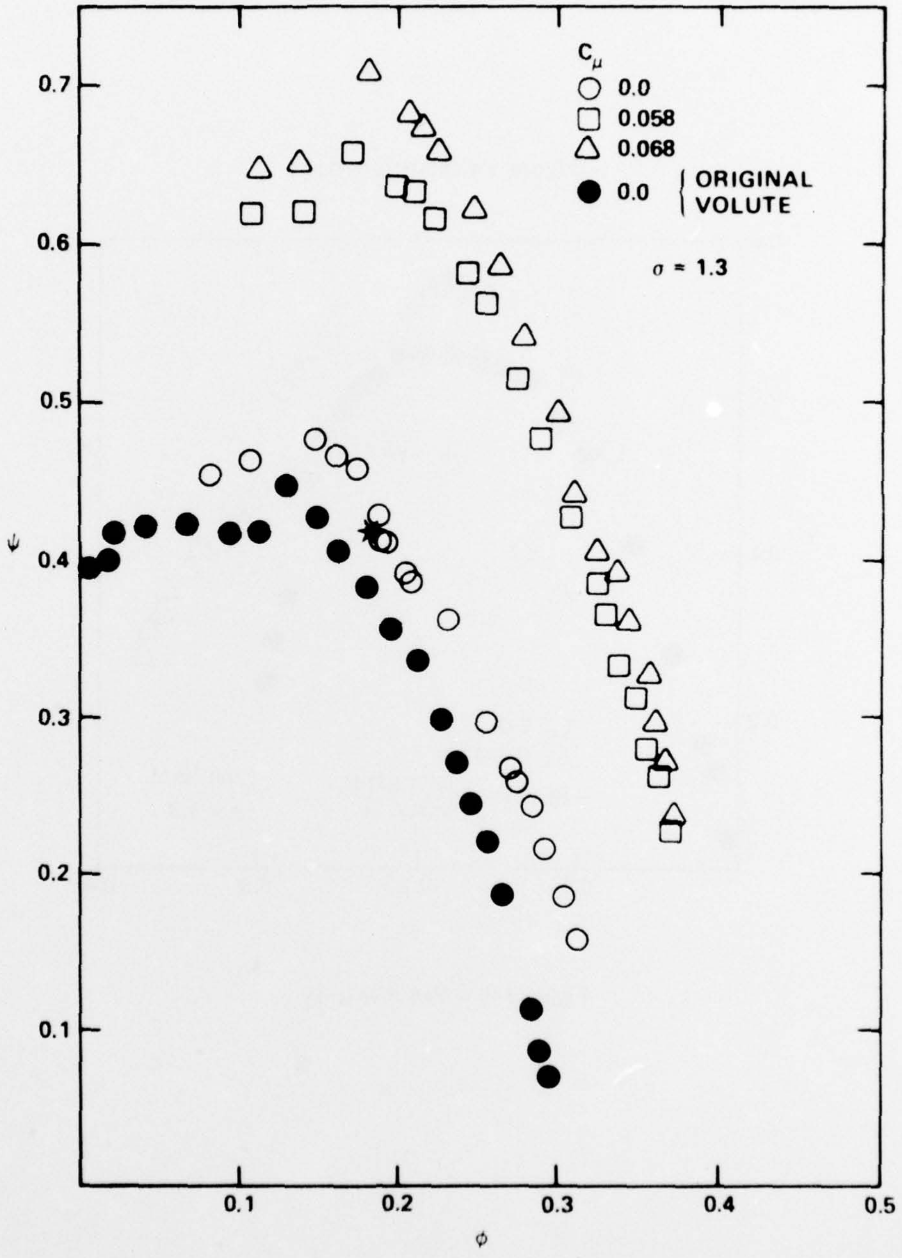

Figure 14a – Head/Flow Characteristics

Figure 14 (Continued)

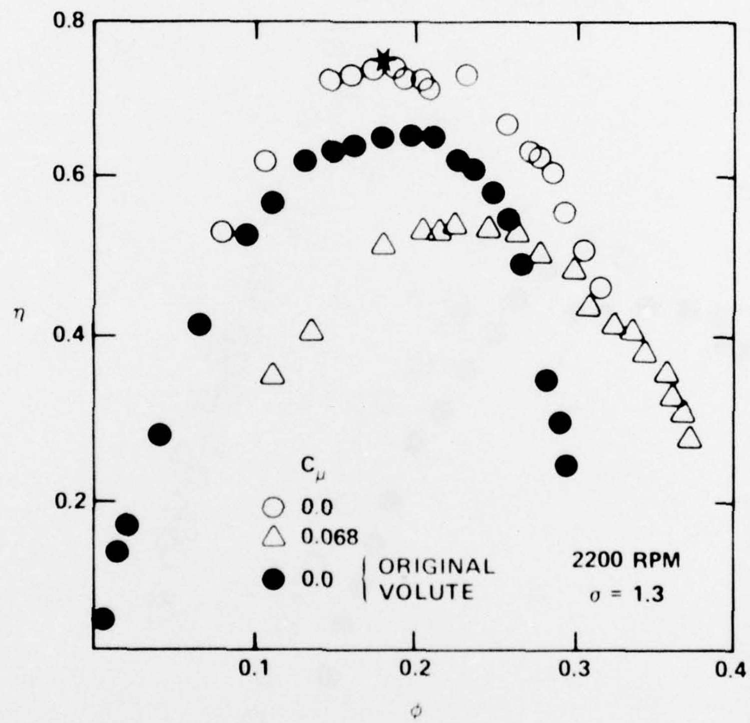

Figure 14b – Fan Efficiency

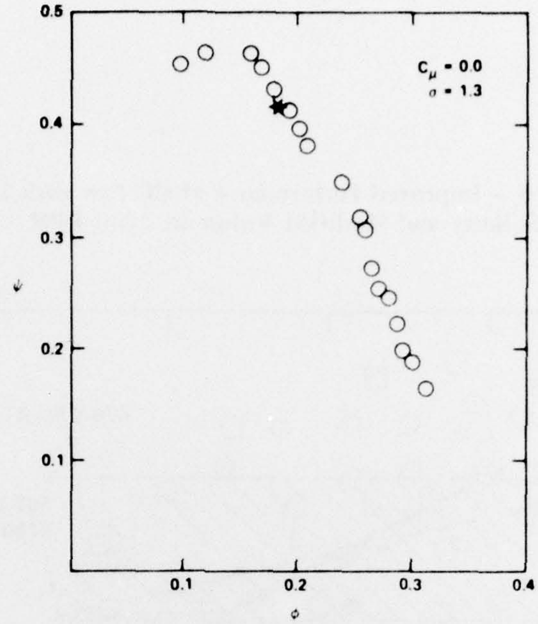

Figure 15a – Head/Flow Characteristics

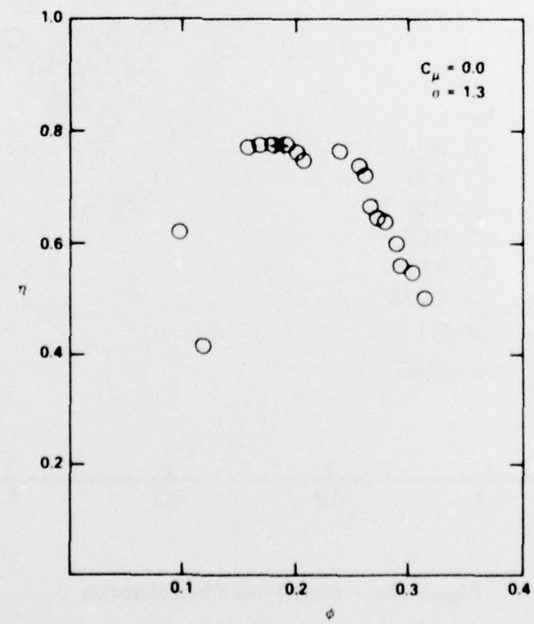

Figure 15b – Fan Efficiency

Figure 15 – Improved Performance of CC Fan with Modified Volute at 2800 RPM

Figure 16 – Improved Performance of CC Fan with Lower Solidity and Modified Volute at 2200 RPM

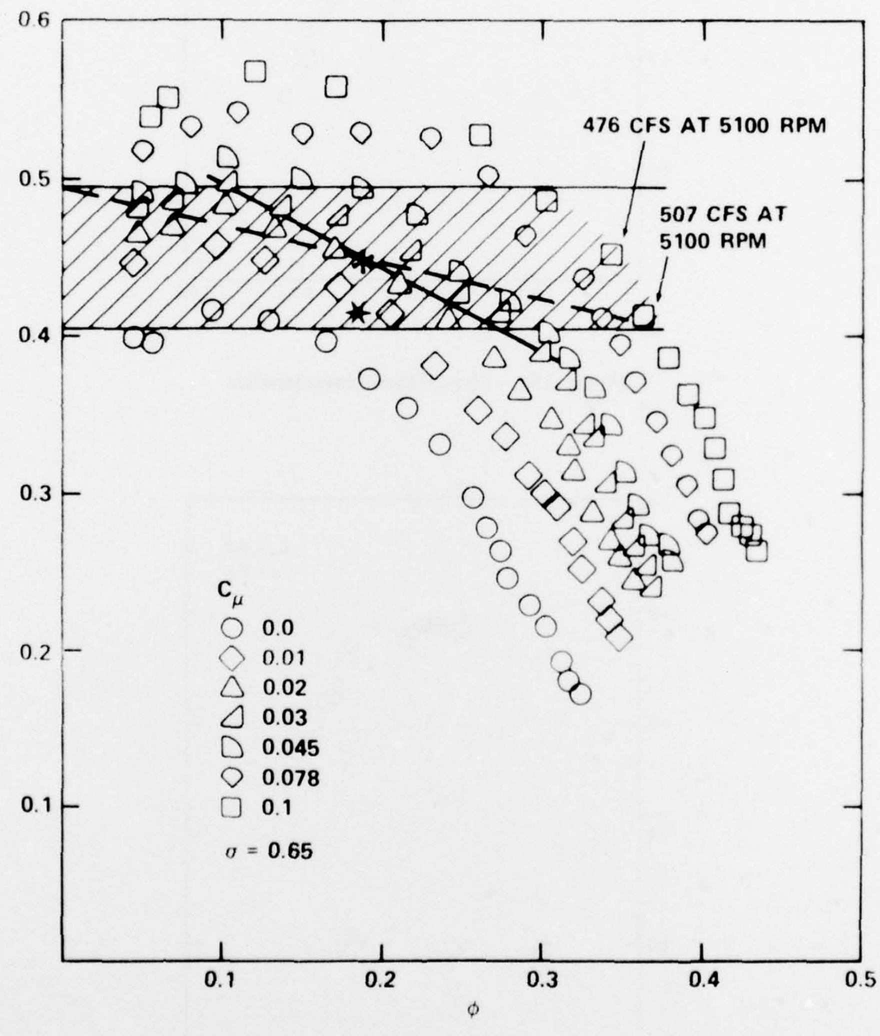

Figure 16a – Head/Flow Characteristics

Figure 16 (Continued)

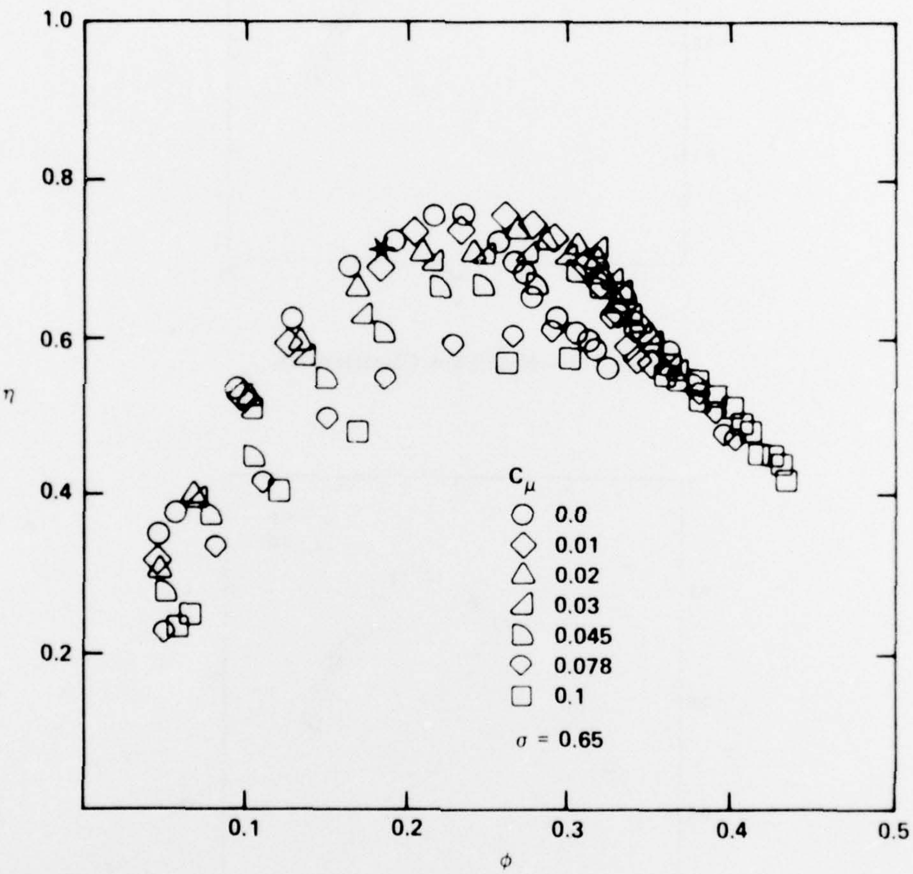

Figure 16b – Fan Efficiency

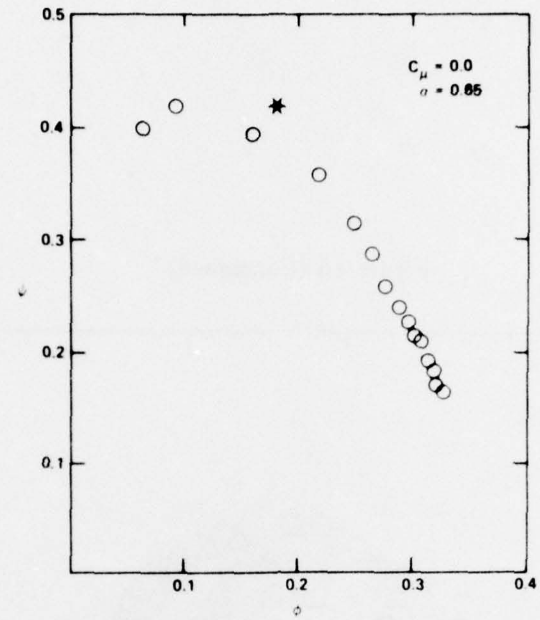

Figure 17a – Head/Flow Characteristics

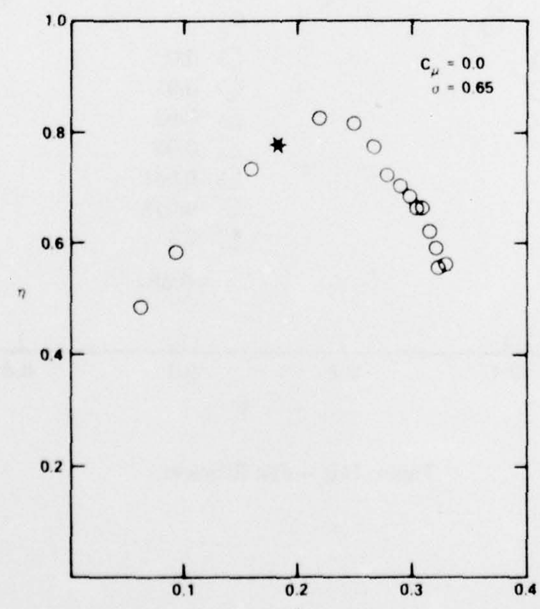

Figure 17b – Fan Efficiency

Figure 17 – Improved Performance of CC Fan with Lower Solidity and Modified Volute at 2800 RPM

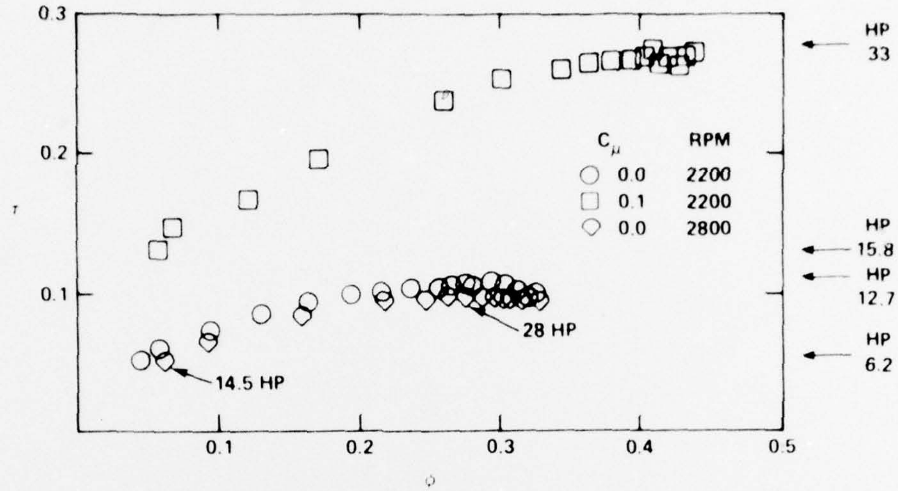

Figure 18 – Power Required for Low Solidity Configuration

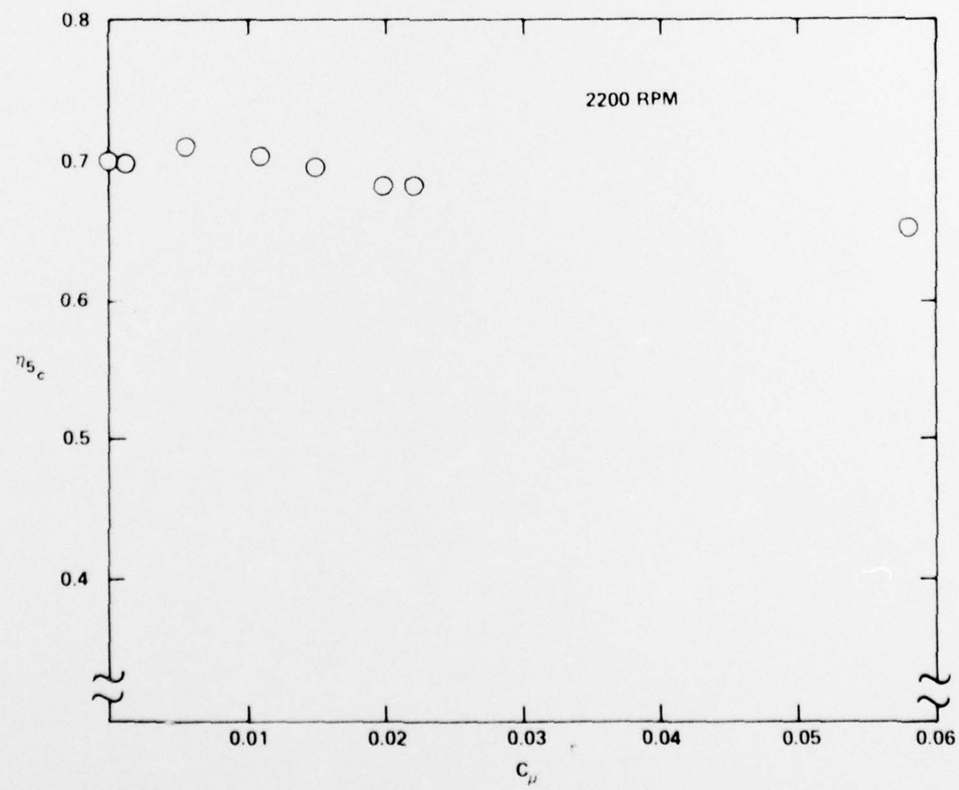

Figure 19 – Projected Effect of Moderate Control Air Blowing on Efficiency of Fan for 2000-Ton SES

(Exhaust valve at 20 percent open, $\phi \cong 0.188$, $\sigma = 1.3$, 2200 rpm)

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