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Modifications To The 4 x 7 Meter Tunnel For Acoustic Research

Engineering Feasibility Study

DSMA Engineering Corporation
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1. INTRODUCTION

1.1 General

The NASA-Langley Research Center 4 x 7 meter Low Speed Wind Tunnel is currently being used for low speed aerodynamics, V/STOL aerodynamics and, to a limited extent, rotorcraft noise research. The deficiencies of this wind tunnel for both aerodynamics and aeroacoustics research have been recognized for some time. Within the FY-1984 NASA Construction of Facilities (C of F) Program, modifications to the wind tunnel are being made to improve the test section flow quality and to update the model cart systems.

A further modification of the 4 x 7 meter Wind Tunnel to permit rotorcraft model acoustics research has been proposed for the FY-1989 C of F program. As a precursor to the design of the proposed modifications, NASA have conducted both in-house and contracted studies to define the acoustic environment within the wind tunnel and to provide recommendations for the reduction of the wind tunnel background noise to a level acceptable to acoustics researchers. One of these studies by an acoustics consultant, Bolt, Beranek and Newman Inc. (BBN), has produced the primary reference documents (References 1 and 3) that define the wind tunnel noise sources and outline recommended solutions.

As wind tunnel design consultants, DSMA Engineering Corporation has been retained to conduct a conceptual design and feasibility study for the practical application of the modifications recommended in References 1 and 3. This report covers the results of the study.

1.2 Scope of Work

The work is defined in NASA Specification No. 1-14-5627,0236 (Reference 2) and covers the following areas:

- Redesign of the fan to achieve, as a goal, fifty (50) percent fan rotational speed reduction at the operating point.

- Structural considerations to enable installation of acoustic treatment in the settling chamber.

- Acoustic treatment to the test chamber walls, ceiling and floor.

- Acoustic treatment to the turning vanes in corners 1 and 2.

The modifications listed above represent "Scheme B" recommended by BBN in Reference 3 and adopted by NASA for the purpose of this study. The areas of the wind tunnel included in the Scheme B modifications are shown schematically in Figure 1.

The scope of work also included two areas closely connected with acoustic testing.

- Relocation of the control room outside the test chamber.

- Conceptual design of a new Sting and Rotor Drive System.

The overall layout of the 4 x 7 m wind tunnel and of the areas covered by the study, is indicated on Drawing LD - 544301. In each of these areas it is required to develop a feasible design concept, consider its implementation and prepare preliminary cost estimates and preliminary schedule.

The development of suitable design concepts must take into account the following additional requirements important to the facility users:

- The down time of the wind tunnel necessary for implementation of the modifications should be minimized.

- The acoustic treatment on the test chamber floor and the underside of the movable ceiling of the test section, should be removable. This will enable the facility to be converted from acoustic to aerodynamic testing mode and vice versa. The facility down time necessary to accomplish such a conversion must be the shortest possible; one day (two shifts) duration would be desirable.

2. CONCEPTUAL STUDY

2.1 Fan Redesign

The basic requirements for the fan redesign were that when the BBN "Scheme B" (see Figure 1) was implemented, the existing circuit design points would still be achievable. These design "power" points are, for the closed and open test section configuration, 120 and 70 psf dynamic pressure respectively. In addition, an "acoustic" design point with a dynamic pressure of 50 psf in the open test section should be achieved at reduced (halved as a design goal) fan rotational speed, compared to the present situation.

In the present work, no allowance has been made for models in the test section or for the losses associated with the air exchange system (outlet upstream of corner 3 and inlet in the test section diffuser). Estimates of these losses should be included in the final evaluation of circuit performance. Further, since the air inlet is in the test section diffuser, the static pressure at this point will be slightly sub atmospheric and this will modify the circuit pressure levels given in Tables 1, 3 and 6.

The fan redesign procedure in this study included definition of the fan design parameters by calculating the circuit losses, fan aerodynamic design, and development of mechanical/structural concepts.

2.1.1 Circuit Loss Estimates

The operating conditions in terms of the total pressure rise and the mass flow for the fan on which the redesign was based were defined by calculating the circuit losses. All circuit losses were calculated using a DSMA proprietary computer program. This program has been used for the design of a variety of closed return circuit wind tunnels covering a wide range in speed (low subsonic to Mach 1.4), test section size (1.5 to 100m^2), and test section type (closed and slotted wall, and semi-open and open jet); and in all cases where it has been possible to compare the design calculations with measurements in the facility, they have agreed well.

As a first step, the losses and fan requirements at the two power points were calculated for the circuit as it has existed to date. Geometric data was taken from Sanders and Thomas Inc. Drawing No. LD-254369 (March 67), and it was assumed that two 1.0 q screens had been installed in the settling chamber upstream of the contraction. The detailed loss outputs for the two cases are given in Table 1 - part (a) gives the results for the closed test section case and part (b) gives those for the open test section. Since information on the existing fan was not available, the fan efficiencies were estimated by dividing the "air power" from the loss calculations by the maximum drive power of 8,000 hp.

The results for the existing circuit are summarized in Table 2. The closed test section results were compared to detailed experimental data supplied by NASA Langley, and found to be in reasonable agreement. Similar data for the open test section were not available.

The second step was to estimate the losses for the proposed circuit (BBN Scheme B). For this analysis, several modifications were made:

- The loss factors for the turning vanes in corners 1 and 2 were increased. As discussed in Section 2.5.1, a decision was made to use rolled plate turning vanes with acoustic treatment on the inner (pressure) surface as incorporated in the DNW tunnel. The loss increment used was based on experimental data from DNW, Reference 4.

- The cross-sectional area of the "settling chamber" (from corner 3 inlet to corner 4 outlet) was reduced under the assumption that 0.61 m (2 ft) thick acoustic treatment would be internally mounted on the floor, sidewalls, and roof of this section of the wind tunnel. As discussed in Section 2.2.2, the reduction in facility performance due to this area decrease is predicted to be minimal.

- Flow conditioning devices are being installed in the facility to improve the test section flow quality as part of the current program of facility upgrading as described in Reference 5. Losses for these components - 2.0q for the grid upstream of corner 3 and 4.3q for the honeycomb and four screens in the settling chamber (downstream of corner 4) were incorporated in the analysis.

- For the open test section cases, the nozzle area was reduced to account for the 0.61 m thick (2.0 ft) acoustic treatment on the floor of the test section.

With these assumptions, losses were estimated for all three operating conditions defined earlier. The fan efficiencies used were the results of the ongoing fan design analysis. The detailed loss outputs are given in Table 3 - parts (a) and (b) give the results for the "power" point conditions in the closed and open test sections respectively, and part (c) gives the results for the open test section, "acoustic" design condition. The results are summarized in Table 4, and are the basic input data for the aerodynamic design of the fan.

2.1.2 Fan Aerodynamic Design

The principal objective of the fan redesign was to minimize the RPM at the 50 psf operating point in the open test section configuration. The nominal goal was half the speed from the 185 RPM currently required.

As seen from Table 4, the lowest mass flows and highest fan pressure ratios occur for the open test section configuration. Since these operating points will thus be closest to the fan surge line, they represent the critical conditions for the fan design. The first step in the design process was therefore to select the lowest possible RPM which still gave an acceptable surge margin at the 50 psf point. A check was then made that stable operation would be available at the 70 psf open test section point. Finally, an estimate was made of the RPM needed at the 120 psf closed test section point, since this defines the maximum speed needed from the fan drive.

In addition to having the performance objectives described above, the new fan was subject to several aerodynamic and mechanical constraints.

In the first place, it was agreed with Langley personnel that the fan should be of conventional design. Essentially, this involves keeping the fan geometric parameters within the range for which cascade data are available. In this way it will be possible to predict the pressure ratio, surge margin and efficiency of the final design with a high degree of confidence. By comparison, an unconventional design would involve considerable risk and could necessitate expensive model tests. The main constraint arising from these considerations is a maximum blade solidity (chord/blade spacing) of about 2.0 at the hub.

The new fan was not to compromise the aerodynamics of the rest of the circuit, particularly the stability of the fan diffuser. It was found necessary to reduce the fan cross-sectional area in order to obtain a sufficiently high flow coefficient (axial velocity/blade speed), and this in turn increased the area ratio of the fan diffuser. The fan diffuser performance was therefore analyzed in parallel with the development of the fan design. The analysis shows that the diffuser can cope with the 7 m (23 ft) hub diameter of the new fan, particularly since the flow uniformity at the diffuser inlet should be better than at present.

Finally, the new fan should require as little modification of the existing structure as possible. Specifically, the new stators should if possible accommodate the main fan supports which presently pass through three of the seven stators. This objective has been met and the new stators should simply call for the reskinning of the present ones.

The detailed fan geometry is given in Section 2.1.3. The present section describes the method used to predict the fan performance, outlines the aerodynamic rationale for the geometry selected and presents the estimated performance diagram.

The requirements in this particular case necessitated a departure from the usual DSMA fan aerodynamic design procedures. Normally, no attempt is made to predict the off-design performance until the blade angles have been selected during preliminary design. The geometry is then run through the streamline-curvature computer program which predicts the complete performance map, including efficiencies, using cascade correlations. Such detailed calculations are beyond the scope of a feasibility study. The off-design performance has therefore been calculated using a simpler, and necessarily more approximate procedure.

Briefly, the method consists of a through-flow calculation based on simple radial equilibrium neglecting entropy gradients and density changes. To predict the off-design turning performance of the rotor blades it is assumed that the outlet relative flow direction is the same as at design and thus, the change in deviation angle with incidence is neglected. This simplification causes least error near the design point and for this reason the design point was placed close to the critical open test section operating points.

Since the fan inlet flow is known to be non-uniform, this was taken into account in an approximate way by specifying a linear inlet axial velocity variation, with hub and tip axial velocities 120% and 80% of the mean respectively. No attempt was made to vary the degree of non-uniformity with the mass flow rate or as the hub-to-tip ratio was adjusted. Finally, no attempt was made to predict losses. The total enthalpy rise obtained from the through-flow calculations was translated into pressure ratio by assuming an isentropic efficiency. A conservative value of 85% was used at the design point and it was adjusted downward at off-design calculation points.

To achieve maximum performance, the new fan rotor uses the maximum allowable solidity of 2.0 at the hub. However, it was found that the rotor performance now available could not be fully exploited because the stators would be unable to remove the swirl. Using the same chord length as the existing stators, the solidity at the stator hub is already about 2.6 and it was therefore undesirable to try to reduce the loading by increasing the solidity. Instead, a set of inlet guide vanes was added to give the rotor inlet flow 30 degrees of prewhirl. This has the effect of reducing the flow straightening through the stators to about 30 degrees, which should be achievable with reskinned versions of the existing stators. A comparison of the basic geometry of the existing and proposed fan is given in Table 5.

As configured, the three blade rows of the new fan are about equally loaded and two of them are at the allowable geometric limits. There is therefore little scope for further increases in performance. The proposed configuration essentially represents the best that can be done in a single-stage machine.

The approximate performance map for the new fan is shown in Figure 2. It will be noted that the open test section load line lies quite close to the surge line. The occurrence of surge was predicted from a criterion usually employed by DSMA, namely when the diffusion factor (a parameter which quantifies blade loading) reaches 0.6 at any point or 0.4 at the rotor tip. These are very conservative values for a low speed machine. In addition, it may be possible to increase the surge margin slightly during final design, by the choice of an alternative fan design point and with other minor modifications. There is therefore no doubt that stable operation will be available, with a reasonable margin of safety at the points in question.

As seen from the map, 135 RPM will be needed at the 50 psf operating point, whereas the goal was 93 RPM. As outlined earlier, the performance obtained from the new fan is the best that can be achieved in a single-stage machine of conventional design. In short, the 93 RPM goal is not feasible. As to the precise speed needed, the approximate nature of the off-design calculations should be borne in mind. When the design is refined and more accurate performance calculations made, some adjustment in the speed is likely, but at best only a marginal reduction in RPM can be expected.

Although an assessment of the noise characteristics of the new fan was beyond the scope of the present study, the following observations relative to acoustic design of the fan are pertinent.

The DNW wind tunnel, which has a well known excellent acoustical environment, has a fan of very similar diameter to that of the 4 x 7 m wind tunnel with a top speed of about 200 RPM corresponding to an open test section velocity of 85 m/s (about 90 psf dynamic pressure) for the 6 x 8 m nozzle. It thus seems certain that at a given test section dynamic pressure, the proposed fan will be running at a considerably lower tip speed than the DNW fan.

The new 4 x 7 m wind tunnel fan should be designed with noise reduction in mind and due consideration must be given to such questions as the spacing between blade rows in order to reduce the strength of the blade-wake interactions. This, together with the fact that the new fan will be running unstalled, should make it inherently quieter than the existing one. Finally, since the new fan will require a new nosecone, nacelle and tailcone, it would be a relatively easy matter to incorporate acoustical treatment both upstream and downstream of the rotor (as, for example, in NTF).

2.1.3 Mechanical/Structural Concepts

The key consideration in this portion of the study was to develop a feasible mechanical/structural configuration that can be implemented at a reasonable cost.

Based on the aerodynamic considerations discussed in the previous section, a fan geometry was established together with the following design goals aimed at minimizing the costs.

- Retain the fan section outer casing with a diameter of approximately 12. 5 m (41 ft), and modify the casing as required.

- Retain the fan foundation; that is, the location and general size of the fan stator vanes.

- Modify the fan drive system to develop approximately the same power as at present but at reduced speed.

- Replace fan nacelle, nosecone and tailcone.

- Replace fan rotor and blading.

The recommended fan geometry can be seen on the Drawing LD - 544302. Compared to the present design of the fan, several changes may be noted.

The nacelle diameter has been increased from 4.9 to 7 m (16 to 23 ft) and the length has also increased at the tailcone (downstream) end. The short spinning nose cone has been replaced by a stationary semielliptical assembly with an aspect ratio of 2:1, supported by five (5) inlet guide vanes. The fan rotor remains in its original location with its center at station 351'-3" but it is wider than at present to accommodate nineteen (19) large chord rotor blades.

The stator airline profiles will be modified; however, the structural "columns" supporting the fan housing at 8 discrete foundation base plates are unchanged.

Several important components and aspects of the fan redesign were considered in more detail and are discussed in the following sections.

2.1.3 a) Fan Blades and Rotor

The fan blades have the following basic configuration developed in the aerodynamic concept work:

Number of blades	:	19
Hub solidity	:	2
Ratio of tip/hub chord	:	0.75
Taper (chordwise and spanwise)	:	linear
Thickness - hub	:	12% of local chord
- tip	:	8% of local chord
Blade Profile	:	NACA 65 Series, Circular Arc Camber

These values are preliminary and are likely to change somewhat during later design phases.

The relative cost, durability and inherent structural damping of simple, solid wood blades make this construction preferable, if feasible. DSMA has had many years of successful experience with operation of such blades, and design/construction methods are well developed and proven.

A layout of a typical blade geometry and root attachment is shown on Drawing LD - 544303, Sheet 2. The blade is laminated, usually from Sitka spruce. In the spanwise direction toward the root, wood impregnated by phenolic resin (Compreg) is gradually laminated between the spruce sections so that at the root, the full section is made of Compreg. Also, the airfoil shape is changed into a cylindrical root section by a gradual transition, with the portions of the chord overhung outside the root section being lightened. The airfoil section is covered by a thin layer of fiberglass, and the leading edge is protected by a metal (Monel) strip. A pine breakaway section is installed at the blade tip. A steel ferrule with a clamping ring is fastened to the root section, for attachment of the blade to the rotor.

The fan rotor is shown on the Drawing LD - 544303, Sheet 2. It is a steel weldment consisting of a central hub, two discs and radial ribs. Blade ferrules fit in sockets at the rotor circumference and the sockets are connected to the rotor discs by means of short shear tubes.

The blades are fastened in the rotor sockets using clamping rings and high strength bolts. Fairing plates then cover the socket openings as shown on the drawing. This blade attachment design is safe and reliable, and has been proven on a number of low speed wind tunnels. The design also allows small adjustments to the setting angle of the blades and this feature can be used to optimize the fan performance.

The fan blade design was checked by a preliminary stress and vibration analysis. The stress analysis considered the centrifugal and aerodynamic bending loads, and the maximum combined stress at the root was found to be 1.8 Ksi. In the final design, the blade will be tilted to reduce the aerodynamic bending loads so that the maximum stress will decrease. The allowable fiber stress for Compreg is 7.5 Ksi for "infinite" life and therefore, the blade stresses are well within the allowable limits.

Another important aspect of the blade design is the blade vibration. A DSMA blade vibration program was used to calculate the natural frequencies of the baseline blade geometry. The results in the form of a Campbell (interference) diagram are shown in Figure 3. The first two natural frequencies are plotted as a function of the operating speed, with cross-plotted excitation orders (so-called engine orders). This initial evaluation was made without any attempt at optimization of the blade design; it may be seen that there is a possibility of a resonance at close to 130 rpm as a result of excitation of the rotor blade by the pressure field upstream of the stator vanes. Another possible resonance may occur at close to the top speed (180 rpm) due to excitation by the wakes from the inlet guide vanes.

During the design phase, the fan will be optimized to avoid potential resonances at high speeds as they could lead to blade failures. This optimization is normally accomplished by changing the blade section design to alter the natural frequencies, by changing the excitation orders (e.g. number of guide vanes), or by a combination of the two.

As a result of the analyses and previous experience, it is concluded that it will be feasible to design solid wood blades for the new fan.

There are several wood blade manufacturers in the USA and in Europe. During this study, DSMA contacted two of them, to discuss feasibility and obtain pricing of the blade set.

These discussions will be continued during the design phase. An alternative approach will include consideration of hollow blades made of composite materials. The fabrication technology in this area is progressing very rapidly and it is conceivable that, in the future, the composite blades may be less expensive than wooden blades. Hollow blades also offer the potential for increasing the natural frequencies which is desirable from the point of view of vibratory stress levels. On the negative side, hollow blades are more susceptible to foreign object damage and repairs are more costly.

2.1.3 b) Fan Drive System - Initial Concepts

The redesigned fan will absorb approximately the same power as at present (close to 8000 HP) at 158 rpm, about two thirds the present rotational speed.

A completely new drive system was quickly evaluated but the costs would be prohibitive.

The existing drive system consisting of an AC synchronous machine in tandem with a smaller DC drive cannot be modified electrically to provide the required performance. Therefore, a gear reducer appears to be the only effective option. DSMA contacted three suppliers of 'standard' gear reducers with a request for configuration and pricing. All three companies (American Lohmann, Falk Corp., and David Brown Co.) offered their standard line, single stage gearboxes with an offset, despite requests for a coaxial design.

The drive layout shown on Drawing LD-544302, incorporates such a gearbox. It is clear that as a result of this gearbox design, the implementation of the fan redesign becomes more complex than desired:

- fan drive unit must be raised by the amount of the gearbox offset,

- an additional bearing must be added to support the rotor as it cannot be overhung on the gearbox ouput shaft.

These complications bring into focus the work required to modify the fan, and the downtime connected with this activity.

2.1.3 c) Rebuilding the Fan - Initial Concept

The new geometry of the fan and the drive system modifications discussed above will require a substantial amount of work, and entail some risks.

The fan blades and rotor will have to be removed, and all the services to the drive train disconnected. This will be followed by adding new stiffeners and braces whose purpose is to minimize the amount of distortion of the fan housing. An upper portion of the outer casing will be removed, together with the upper stator vanes. Then the nacelle and the drive assembly will have to be removed through the opening in the outer casing. When the lower stator vanes have been modified, the new nose cone and nacelle will be erected, the drive unit with the new gearbox and fan rotor re-installed, new upper stator vanes welded in and the outer casing closed up and re-welded. Finally, the fan blades will be installed and the drive system re-connected prior to the start of the fan tests.

This overall procedure can be described by few sentences; however, in reality the construction will be difficult, time consuming and expensive. In addition, the extensive amount of cutting and re-welding on the 1 inch thick outer casing plate will likely result in large distortions. It may then be very difficult or even impossible to bring the casing shape to within the limits acceptable for the running track of the fan blades.

Therefore, this concept although considered feasible, was not satisfactory and work continued on development of a more suitable concept.

2.1.3 d) Fan Drive System - Proposed Concept

The disadvantages of the 'standard' offset gearbox approach resulted in an in-house development of another approach - a coaxial gear reducer built within the fan rotor assembly. This compact unit is laid out on Drawing LD - 544303, Sheet 3. It is of a solar gear type which has a central sun gear, 4 planets meshing with and spaced uniformly around the sun, and a ring gear meshing with the planets. The sun gear is stationary and is mounted in a rigid support able to resolve the reaction torque. The ring gear as the input member is doweled and bolted to a heavy sleeve mounted on the existing shaft of the drive unit using the same mounting as the present fan rotor. The total transmitted torque is divided among the planet gears mounted on needle bearings and heavy precision ground shafts in a planet carrier which is the output member.

Since the torque is divided among 4 planets the size of the gear tooth is approximately 4 times smaller than in a standard gearbox, enabling design of a compact unit that will easily fit on the existing shaft.

The fan rotor will be mounted on heavy Timken or Torrington tapered roller bearings with all the loads being transfered to the drive motor shaft.

Lubrication and cooling of the gears is done by splashing and a forced feed lubrication system, employing a self-contained lubricating unit.

In designing a planetary gearbox of this kind, particular attention must be directed to the following design issues:

- high bearing loads on the planet pins; high capacity needle bearings or roller bearings (space permitting) will be used, mounted on precision-ground shafts.

- balance and vibration of the rotating cage; high precision and tight tolerances will overcome this problem.

- load sharing between planets; a free floating sun gear will be considered to help ensure equal load distribution among the gear meshes.

- epicyclic gears require high accuracy and precision; therefore, heat treated alloy steel gears will be employed, the planet and sun gear will probably be carburized, surface hardened to 60 Rc and then ground to quality class AGMA 10, and the ring gear through-hardened to 36 Rc.

- noise generated by the gearbox (and the fan drive motor assembly); the noise level estimates will have to be made, and necessary internal acoustic treatment defined in the design phase.

DSMA has performed a preliminary design analysis of the gearbox arrangement. The stress levels in the gears and planet gear shafts are well within the acceptable limits (20 Ksi was considered as the limit for this conceptual design stage), and bearings with the required static and dynamic capacities are readily available.

The detail design may result in some changes mainly due to lubrication requirements, and detailed consideration of component sizing and manufacturing. The lubrication unit will be located inside the fixed nose cone as shown on Drawing LD - 544303, Sheet 1.

The design of a coaxial planetary gearbox integrated in the fan rotor assembly is feasible and is also very advantageous for the fan redesign considered in this study.

2.1.3 e) Fan Housing - Proposed Concept

With the successful solution to the fan drive problem, a suitable concept for the fan housing redesign logically followed:

- The fan drive unit can remain essentially in the same location as in the existing fan.

- It should not be necessary to remove the existing (small diameter) nacelle.

- Consequently, there is no need to cut open the fan outer casing, and the risk of distortion during refabrication is eliminated.

The proposed fan layout based on this concept development is shown on Drawing LD - 544303, Sheet 1.

The fan drive unit is not disturbed and neither are the services to the unit. The existing nacelle is modified by removing the tailcone downstream of the drive unit and by adding stiffeners and mounting brackets.

The new nacelle and tailcone is a "fairing" of a lightweight construction, fastened to the existing "structural" nacelle. This lightweight construction will probably incorporate additional acoustic treatment as required, to reduce both the external and internal (drive system) noise.

The same applies to the stator vanes, and the new nose cone assembly.

The new fan rotor/blade assembly is mounted on the drive shaft through the coaxial planetary gear reducer. The lubricating unit for the reducer is mounted inside the nose cone.

This concept is considerably simpler and more economical than the one initially considered. Very little structural modification of the fan housing is required. New components can be tailor-made in sections based on the "as built" measurements of the existing fan thereby reducing the installation time and the fitting problems. Access into the wind tunnel shell can be made relatively easily during the construction through the upstream transition for the nose cone, rotor and blades, and through the fan diffuser for the nacelle, tailcone and stator vane fairings.

The cost estimates in Section 3 are based on this concept for the fan and fan drive redesign.

2.2 Settling Chamber Acoustic Treatment

2.2.1 Concept Development

The "settling chamber" structure is not unlike a large building made of structural steel. The floor is reinforced concrete.

External steel columns and beams suitably braced, support the roof trusses. The airflow surfaces on the walls and ceilings are formed by corrugated steel sheeting fastened to the inside of the steel structure.

Reference 1 recommends that the settling chamber be lined by 2 ft deep "bulk absorber treatment" without changes to the airline dimensions. The treatment basically consists of perforated sheet at the airflow side and 2 feet of mineral wool or fiberglass.

The requirement to preserve the existing airline dimensions means that the corrugated steel sheets must be replaced by the flat perforated sheets of the acoustic treatment, with the corrugated sheet relocated to the outside flanges of the structure.

Clearly, this task will be lengthy and expensive for the following reasons:

- The corrugated sheet will have to be removed and presumably not all of it can be re-used.

- Additional stiffeners will have to be provided to support the more flexible perforated sheet on the airflow side, and to create an effective grid of panels to be filled with the insulation.

- With the outer corrugated wall completed, the acoustic insulation material will be installed in the non-standard "panels" followed by installation of the perforated sheets.

Since this recommended solution has severe cost and schedule deficiencies, an alternative approach was considered that would place the acoustic treatment inside the existing settling chamber. An analysis of this approach and potential performance penalties is discussed in the next section.

2.2.2 Aerodynamic Considerations

The efect on wind tunnel performance of mounting the treatment on the existing inner surface of the settling chamber shell, and thus reducing the flow area in this section, was investigated.

Loss calculations described in Section 2.1.1 above were also performed for the closed and open test section "power point" conditions with the settling chamber area as it has existed to date (the area reduction due to mounting 0.6 m (2 ft) thick acoustic treatment on the floor, sidewalls, and ceiling of the settling chamber is on the order of 12%).

The detailed loss outputs for these two cases are given in Table 6 - the results for the closed and open test section configurations are given in the (a) and (b) parts respectively.

Comparison of these results with those of Table 3 (a) and (b) shows that the differences are extremely small - well within the accuracy limits of the calculations. This is due to the fact that the settling chamber cross-sectional area is large compared to that of the test section, and the losses in this section of the wind tunnel are quite small - on the order of 1.5 and 0.5% of the total circuit losses for the closed and open test section configurations respectively. Thus, small changes in this section of the wind tunnel have a small effect on the total losses.

Based on these results, it was concluded that the acoustic treatment in the settling chamber can be mounted on the inner surface of the existing shell. The resultant negative effect on facility performance is minimal, if not negligible; but the positive effects on ease of installation and cost of this treatment are significant.

There is very little space between the outlet from corner 4 and the honeycomb in the settling chamber. Care will have to be exercised in fairing out the acoustic treatment in this region to avoid an adverse effect on flow quality.

2.2.3 Proposed Concept

The basic concept of the acoustic treatment in the settling chamber is shown on Drawing LD - 544304. The acoustic material is placed on the inner surfaces of the existing settling chamber in a form of 0.6 m (2 ft) deep flat panels. The design and construction of the flat panels are standard and will be also used for the sound attenuation in the test chamber.

Mineral wool or fiberglass (density 4 lb/ft^3) is used as an acoustic material, filling a galvanized steel enclosure. At the side facing the airstream the steel sheet is perforated (30% open area) and the acoustic material is covered with a fiberglass cloth and a wire mesh. The panels are attached to the inner flanges of the wall columns and trusses trough the corrugated steel sheets by means of channels, battens and couplings. The panels on the floor are connected together with battens and are bolted to the concrete of the floor in several places.

In order to avoid excessive turbulence, tapered fairings are provided at both the upstream and downstream end of the settling chamber to cover the steps between the inner surfaces of the panels and the surfaces of neighboring elements of the wind tunnel.

A brief review of the existing structure of the settling chamber has indicated that no large-scale strengthening will be required to support the new acoustic treatment particularly at the roof level; however, minor local reinforcements may be necessary. These reinforcements will be configured in the design phase.

The bulk absorber concept recommended in Reference 1 should also be reviewed during the design. As an alternative, a 0.6 m (2-ft) deep treatment with approximately 0.15 m (6-inch) thick panels at the airflow side and 0.45 m (18-inch) airspace between the panels and the corrugated sheet shell, should be carefully evaluated. Such a treatment may provide acceptable noise reduction at lower cost than the bulk absorber.

2.3 Test Chamber Acoustic Treatment

2.3.1 Treatment Concept

The test chamber acoustic treatment is based on the recommendations of Reference 2 (Attachment 2) and Reference 3.

Two types of treatment are used:

- 0.6 m (2 ft) deep flat panels installed on the floor within the air flow area and to the left of it looking downstream, and also on the adjacent (left-hand) wall of the test chamber parallel with the airstream.

- 0.9 m (3 ft) deep panels with wedges installed on the remainder of the floor, the remaining walls, the underside of the roof trusses and the underside of the movable ceiling.

As the wind tunnel is to be convertible between acoustic and aerodynamic operation, the treatment on the floor and on the underside of the movable ceiling is removable.

The general arrangement of the acoustic treatment is shown on Drawing LD - 544306, Sheet 1 and 2.

Design of the flat panels consists of a galvanized sheet metal enclosure covered on the airflow side by perforated steel sheet with 30% open area. The enclosure is filled with mineral wool or fiberglas with a density of 4 lb/ft^3; to prevent release of the fill material into the airstream, a covering of fiberglas cloth and fine wire mesh is used below the perforated cover sheet. Construction of these removable floor panels is sturdy to allow (possibly frequent) handling, and has provisions for lifting with a fork-lift truck.

Tapered fairings are installed on the floor at the inlet and the outlet of the test section to smooth-out the 2 ft steps between the surface of the flat panels and the original wind tunnel floor. The fairings are made in sections (eight each at the upstream and downstream end) and are built of aluminum, to ease handling. Each section consists of an upper plate and a set of longitudinal and transverse stiffeners that are welded to the plate. Rubber seals are provided around the perimeter of each section. Before they are finally fastened to the wind tunnel floor, the individual sections must be properly aligned. To facilitate this task, omnidirectional casters are installed on the underside of each section.

Construction of the panels with wedges is similar to that of the flat panels. The wedges are attached to a sheet metal base and are made of mineral wool or fiberglas, covered with fiberglas cloth and wire mesh (22 GA wire, 0.5" x 1" spacing). The wedges are placed within the panels in perpendicular groups of 3 or 4, (to improve the acoustic performance). The removable panels again have sturdier design compared to the permanent installation, and have lifting provisions similar to the flat panels.

The permanent acoustic treatment panels are attached to the test chamber structure using channels, battens and bolts as indicated in detail Z of Drawing LD - 544306, Sheet 1. The panels attached to the walls are largely self-supporting since the lower panels support the weight of the upper panels; loads transferred to the test chamber structure are not large and can be accommodated without major structural modifications.

A preliminary estimate of the allowable extra loads on the test chamber roof structure has been done by NASA. It appears that no major modifications of the roof will be required when the wedge panels are installed (the wedge panels will impose a load of about 12 psf as compared to the estimated allowable extra load of 20 psf). However, detail analyses of the roof structure will be required, to define all the necessary local reinforcements.

The removable floor panels will be bolted to the existing floor; here, consideration will be given to interlocking the panels so that the number of fasteners penetrating into the floor can be minimized (as the floor openings must be plugged when converting to the aerodynamic testing mode).

Design of the removable acoustic treatment panels must take into account the requirement for a quick conversion, that is, installation or removal of the panels. A design concept to accomplish this task has been developed and is discussed in the following section.

2.3.2 Installation and Removal

The installation or removal of the acoustic panels will be a fairly complex task because the area to be covered is large and rather irregular; also the panels, especially the wedges, will have to be handled carefully so as not to damage them.

Therefore, the task will have to be well organized to even approach the conversion time of two shifts desired by NASA.

The installation concept takes into account the susceptibility of the panels to damage, and the fact they will be stored outside the test chamber.

The panels are stored in special storage racks and each rack is lifted into the test chamber through the open floor area (see Drawing LD 544306, Sheet 1) by the existing overhead crane.

For the installation of the wedge panels on the underside of the movable ceiling, a special portable hoist will be permanently located on the top of the ceiling. The panels are individually lifted from the storage racks using this hoist. When a panel has been fastened to the ceiling, the hoist is disconnected and moved to an adjacent location for installation of another panel.

The floor panels are withdrawn from the racks one by one and placed in their proper location, using a fork-lift truck. Each panel is identified and has its assigned location which must not change. Adherence to this simple rule in conjunction with a fixed sequence of installation will improve the installation time.

Once the regular panels have been installed, the fork lift truck is removed from the test chamber. Then small panels (some of them irregular) are brought in and placed by hand in the area of the collector and close to the floor opening.

It should be noted that the fork lift truck is brought in and removed from the test chamber using the existing overhead crane. Initial enquiries to the manufacturers of fork-lifts have shown that the smallest trucks weigh in excess of 10,000 lbs and this is well over the capacity of the crane. For the purpose of this study it has been assumed that the crane capacity will be increased; however, further investigations should be made to see if lighter fork lifts can be supplied.

The storage of the acoustic panels and floor fairings outside the test chamber will require construction of a storage building, since there is no storage room anywhere within the existing building.

In addition, it will be necessary to provide an area for preparation and checkout of the acoustic models.

Therefore, it is proposed to build a new small building for these two purposes. The building size has been estimated at 15 x 17 m (50 x 55 ft) and 5.5 m (18 ft) high, and its location is shown on Drawing LD - 544301. A large door in this building is located in line with the access door into the "basement" of the test chamber.

The building construction is prefabricated steel, of the type supplied by Butler Manufacturing and other companies. The building is insulated and heated. The area allocated for the model preparation and checkout is approximately 7 x 6 m (23 x 20 ft).

Transportation of the storage racks with acoustic panels between this building and the test chamber basement is acomplished by a second fork-lift truck.

This procedure will obviously require further refinements and detail consideration. However, it is simple in concept, and feasible.

2.4 Turning Vanes Acoustic Treatment

2.4.1 Aeroacoustic Considerations

In Reference 3, the recommended concept for the acoustic turning vanes in corners 1 and 2 was profiled vanes with a chord and thickness of approximately 5.0 and 0.5 m respectively; the interior consisting of variable geometry (or depth) cavities with acoustic absorptive material, and covered with perforated sheet metal facing on both airflow surfaces. Reasonable acoustic performance was claimed for these vanes; however, they would be expensive to manufacture and install, and they would interfere with the flow control "choke flaps" downstream of corner 1 due to their long chord. As an alternative, DSMA investigated the use of a simpler vane consisting of rolled plate with acoustic material mounted on only one side (pressure surface). Vanes of this type are used in the DNW tunnel in The Netherlands, Reference 4.

From the aerodynamic point of view, the optimum chord length for turning vanes in corners 1 and 2 would be 1.6 m (5.2 ft) based on normal DSMA design procedure. The acoustic performance of these vanes was estimated from data supplied by DNW, and compared with the predicted performance for the vanes recommended in Reference 3. At frequencies of 500 Hz and above, both types of vanes had equivalent performance; but in the 125 and 250 Hz bands, the rolled plate vanes gave significantly less attenuation than the profiled design of Reference 3.

The low frequency attenuation of the rolled plate vanes can be increased by increasing the chord length of the vanes. The performance improvement was estimated for chord lengths of 2.5 m (8.2 ft) and 4 m (13.1 ft), and is compared with the 1.6 m vane and the profiled vane performance predictions, in Table 7.It can be seen that rolled plate vanes with chord lengths of 1.6 and 2.5 m do not achieve as much attenuation at the low frequencies (125 and 250 Hz bands) as the profiled vanes, but a roller plate vane with a chord length of 4.0 m gives equivalent acoustic performance over the whole frequency range.

Based on these results, it was concluded that rolled-plate vanes with a chord length of 4.0 m (13.1 ft) could be used in corners 1 and 2 to achieve the required attenuation; and the cost of these vanes would be lower than that for the profiled vanes recommended in Reference 3.

2.4.2 Structural Concept

A layout of corner 1 with the new 4 m chord, rolled plate turning vanes, is shown on the Drawing LD - 544307. There are 8 complete vanes and an incomplete vane in the outer corner. Overall, the interference with the flow control vane assembly downstream of the corner is minimal. The spacing of the turning vanes may be slightly altered in the design phase to place the second vane (from the inner corner) in line with the flow control vane. This will remove the small misalignment seen on the drawing, and improve the flow through the corner. The turning vanes in corner 2 will be identical.

The design concept for a typical turning vane including the acoustic treatment is also shown on the drawing. A standard steel plate vane is the principal structural member. On its pressure side, fairings and continuous flanges are attached at the leading and trailing edges. Flanges are also located at the center chord. Modular acoustic treatment panels are bolted to the flanges.

The panel design consists of a perforated sheet at the airflow side, supported by an "eggcrate" grid of stiffeners. Each grid spacing is filled with acoustic absorption material, mineral wool or fiberglas, sewn into a cover mat of fiberglas cloth. The panels are self-supporting, and will be delivered to site completely assembled. Their installation onto the steel plate vanes will be straightforward and, compared to modifications in other areas of the wind tunnel, relatively short.

2.5 Relocation of Control Room

The control room is at present built inside the test chamber. When the test chamber is transformed into a semi-anechoic chamber during the modifications covered by this study, the control room obviously must be relocated.

DSMA discussed with Langley personnel the possible options for location of the new control room, and finally selected the location adjacent to the present control room, outside the test chamber walls.

A layout was developed and is shown on drawing LD - 544305. The size of the new control room was initially specified as 12 x 4.5 m (40 x 15 ft); however, during discussions at NASA Langley prior to the Design Review Meeting on 7 November, 1984, it was agreed that the size was rather marginal. It was decided to increase the width to 7.6 m (25 ft) so that sufficient flexibility for further upgrades is built in.

The new control room layout features simple access into the test section - a door with an airlock is situated next to the contraction outlet. When the wind tunnel is configured in the acoustic testing mode, personnel requiring access to the model can step out almost directly onto the flat acoustic panels located in the flow area.

A second means of access or egress is provided directly into the wind tunnel building outside the test chamber.

The control room has a large window area for model observation, and the control consoles can be placed in front of the windows as schematically shown on the drawing. In any case, the windows will be covered with acoustic treatment during acoustic testing.

The construction of this control room is a conventional steel structure with steel cladding. No design problems are expected here and no detail consideration was given to the design in this conceptual phase.

The major design issue associated with the control room relocation will be the re-routing of all the existing power, control and instrumentation lines from the present to the new location. The concept proposed by DSMA is as follows:

- Most of the cables enter into the present control room at the north-east corner of the test chamber. The new control room is placed adjacent to and directly across the test chamber east wall.

- An electrical termination cabinet will be placed inside the new control room, very close to this existing cable entry point.

- In general, the construction of the new control room can be almost completed before the wind tunnel shut-down so that the termination cabinet can be ready at the point of the shut-down.

- When the wind tunnel has been shut down for the modifications, all the cables will be disconnected and tagged, enabling the removal of the control consoles, and demolition of the structure of the present control room.

- All the cables will be brought into the electrical termination cabinet and fastened to the allocated terminal strips.

- During the installation and wiring of the control consoles in the new control room, new cables will be installed (within the computer floor provided) between the termination cabinet and the consoles.

This procedure will ensure orderly re-wiring and a minimum of interface problems.

2.6 Sting and Rotor Drive

Conceptual design of this model support equipment has been included in the scope of work for the following reasons:

- The present cranked sting support was designed for aeronautical testing and it is larger and longer than desirable for rotorcraft aeroacoustic testing. The design requirements for the new sting were defined specifically for the aeroacoustic testing.

- The existing rotor drive can be operated only in an upright position while the aeroacoustic test requirements necessitate rolling the rotor +/-180^{0}. Also, the rotor drive power on the new drive should be increased from 35 to 60 HP.

DSMA had several discussions with the users of the system and the designers of the existing rotor drive. As a result, the concepts described in the following sections were developed.

2.6.1 Sting Support System

A brief evaluation was carried out to determine the suitability of a "cranked rotary sting" versus a double-articulated type such as is used at DNW.

The cranked rotary sting utilizes relatively simple rotary actuators to achieve the various combinations of roll, pitch and yaw and is hence simpler to maintain and less expensive to construct. Furthermore, due to the nature of its design, the pivot point is fixed in space. The articulated sting type, however, will always have some relative motion of the pivot point although this is usually minimal. Actuation is achieved by linear hydraulic or electro-mechanical actuators which can be costly. Furthermore, the articulated sting designed to the same stiffness and freedom from backlash as the cranked rotary type will likely present more blockage to the airflow. For these reasons and given the fact that a cranked sting is already in use in the 4 x 7 m tunnel, a version of the cranked sting design was selected for study.

Drawing No. LD - 544308, Sheet 1, illustrates the general arrangement of the rotor drive and the cranked rotary sting. The sting has been designed to provide +/-360^0 roll and +/-20^0 pitch and yaw. An internal passage provides for supply of 30 lbs/sec of air at 5000 psi in addition to oil and water supply and return lines and electrical conductors. The length of each rotary joint has been minimized to allow the sting to be compacted to a relatively clean configuration immediately behind the model. This can be achieved with an overall distance of 3.7 m (12.25 ft) between the model pivot point and the support mast centerline. The inset in the bottom left hand corner of the drawing illustrates the possible use of a "jogged" mast extension which shifts the rotary actuators further downstream from the model should it be necessary to accommodate a larger model or have a cleaner configuration behind the model. This would however, limit the amount of vertical translation which could be generated from the model support cart.

Sheet 2 of Drawing LD - 544308 illustrates a cross-sectional view of rotary joint No. 1. Control and instrumentation cables are routed through the center of the rotary joint while the air, oil, and water supply and drain lines are connected via drillings to annular spaces in the non-rotating portion of the joint. These annular spaces are separated by seals and vent spaces as appropriate. The air passage (5000 psi) is sealed by glass reinforced U cups. Rotary motion is achieved by a DC gear motor with an integral brake and a multi-stage planetary gear head which is arranged below a harmonic drive that it drives through an intermediate spur gear.

Rotary joint No. 1 incorporates a removable model support section which forms the connecting link between the model and the joint. Provision for removal is necessary in order to fit longer or shorter stings or stings with angles built-in or with different balance support structures. Attachment to the rotary joint is achieved through a keyed taper held in place by a locking ring.

Prior to the design review meeting held at Langley on November 7, 1984, this tapered connection was located within the body of the rotary joint and carried the oil and water services and electrical conductors as well as the air supply. These services could then be carried within the sting right up to the balance attachment point. This resulted in a rotary joint that was fairly bulky and which presented more blockage to the air flow than was desirable. Accordingly, the design was revised to the arrangement presently shown.

Model position control would be achieved via a position controller designed to:

- transform roll, pitch and yaw commands into the appropriate angular orientations for the three rotary joints,
- host communications (master/slave) from an external test automation system and provide command/status interfacing,
- provide a local command interface as well as a local position display,
- provide output signals of current position for use by the test automation system or elsewhere,
- input appropriate control constants as required for any modifications of the sting geometry.

The sting support control system will be configured to be compatible with the existing facility control system.

2.6.2 Rotor Drive System

The general concept for the rotor drive shown in Drawing LD - 544309, Sheet 1, is based on the U.S. Army 2 m Rotor Test System with modifications made to permit operation with the roll, pitch and yaw inputs indicated above. The majority of the design work carried out as part of this study centered on the gear box and particularly on its lubrication system which will be described in more detail below. The rotor is driven by an Able Corporation 75 HP, water cooled electric motor (design requirements called for a 60 HP motor while the closest Able design is capable of 75 HP). The rotor head is a four-bladed design with adjustable viscous dampers on the lead/lag pivot point and potentiometers to resolve the flap and lead/lag angles. Cyclic and collective inputs are fed to the blades through the rotating and non-rotating swash plates from three electro-mechanical actuators mounted on top of the gear box. Flap and droop stops are provided so that the model can be stopped in any attitude.

The entire drive system is supported on a gimbal mount where any vibration is reacted by springs and adjustable viscous dampers. The springs and dampers will be selected to locate the resonant frequencies of the rotor drive system outside the normal operating speed range.

Separate balances are provided for the fuselage and rotor drive as well as a total balance which connects to the sting.

Further detail of the gear box design is shown on Sheet 2 of Drawing LD - 544309. A pair of spiral bevel gears carry the drive input from the motor to the intermediate shaft and achieve a reduction ratio of 2:1. Final reduction is achieved by a pair of helical gears also with a ratio of 2:1 (or greater if required). Final selection of tooth geometry will be based on available manufacturing equipment in order to minimize costs.

Lubrication and cooling of the gears and all the bearings is achieved by individual spray feeds. Two nozzles spray oil on the mesh points of the two gear pairs. An additional two nozzles spray the intermediate gear shaft bearings from the side of the bearing cap. This allows the oil to be carried through by the pumping action of the bearing. Similarly, oil is spray-fed from between the pair of tapered roller bearings on the output shaft. The pumping action of the bearings carries the oil either through the drain holes provided or onto the helical gear from which it is flung to the side of the gear box. Oil fed to the cylindrical roller bearing of the output shaft will drain by gravity through the drain holes or, with the gear box inverted, will fall back down onto the gear.

Feeding the oil to the gears and bearings is relatively simple compared with the scavenging system required to insure that the box does not fill up. A brief review of accessory gear boxes on aircraft engines which may be called upon to operate in an inverted attitude shows that

- either two scavenge pump pick-up points are used with selection made by gravity acting on a suitable switchover device,

- or the scavenge points are located at the bottom of the gear box only for operation in a normal attitude and when inverted or in unusual attitudes, the gear box is allowed to fill with oil.

The second case is considered acceptable for an aircraft application. Oxidation of the oil caused by aeration when in contact with the gearing is minimal due to the short time spent in the unusual attitude, and the power lost by the action of churning the oil is insignificant when compared with the amount of power available from the engine. In the case of the subject rotor drive system, degradation of the oil could be tolerated to a certain extent as a relatively large reservoir is available externally, but a large power loss could not be tolerated as only limited power is available within the constraints of the model envelope.

The number of oil scavenge points is largely dependent upon the size of the gear box casing - a large casing can incorporate considerable sump capacity which will allow the oil to collect regardless of model attitude and can be scavenged before the level reaches the gears. A compromise between gearbox size and excessive drain connections resulted in the incorporation of eight scavenge points, one at each corner of the box. A rotary selector valve is used to connect the drain line with the appropriate scavenge point. The rotary selector is driven by a stepper motor connected through a gear set. Gravity and an eccentric weight on the rotary valve could have been used to achieve the appropriate selection; however, it was felt that more positive results would be obtained by a motor drive with input from the sting control system. This also results in a more compact arrangement.

The selection of a pressurized gear box versus one operated with some degree of vacuum is based mainly on considerations of the effects of leakage. With this in mind and given the availability of an existing vacuum oil lubrication system, vacuum scavenging was adopted. The degree of vacuum in the gear box is determined by the pressure loss at the vent fitting. This loss is minimized to insure that sufficient pressure is still available to push the oil through the drain line and sting back to the vacuum reservoir. Within the gear box, a small trap is provided on the air vent line to prevent any oil loss when the model is stopped in an inverted attitude.

3. COST ESTIMATES

3.1 General

The budgetary cost estimates presented in this section have been based on the designs developed in the study, shown on the layout drawings and described in the report.

The estimates include material, shop fabrication, assembly, erection and the required checkout testing. Costs of engineering, procurement and construction management are not included. No allowance has been made for taxes or custom duties. An optimum fabrication and erection schedule was assumed; thus, no allowance was made either for compressing or stretching the program.

The estimates have been based on costs developed in-house from weight estimates and cost data from other similar projects, and on cost estimates obtained from suppliers of the proprietary items. In particular, the fan nacelle and fan rotor costs were estimated on the basis of weight estimates and unit costs from similar recent DSMA projects.

The cost estimate for the fan blades was based on budgetary cost estimates received from two potential suppliers, Hoffmann Co. in Germany and Permali Co., in England.

The acoustic treatment in the settling chamber and in the test chamber was costed in-house, and confirmatory estimates were obtained from Eckel Industries, a supplier of anechoic chambers and acoustic treatment panels. Transport and installation equipment costs (for the removable panel handling) were obtained from equipment suppliers.

The construction costs for the new control room were prepared for DSMA by a civil engineering company while the costs for re-routing the electrical cabling were estimated by the DSMA electrical department.

The acoustic model preparation building costs were based on a budgetary estimate from Butler Manufacturing, a supplier of prefabricated buildings.

DSMA designed the acoustically treated corner vanes on DNW wind tunnels and, more recently, on a low speed wind tunnel (of similar size to the 4 x 7 m) presently under construction in Europe. The cost data from these two projects was used to estimate the cost of the new acoustically treated vanes in corners 1 and 2.

The sting and rotor drive system costs were based on discussions with personnel of NASA Langley, and Sikorsky Aircraft (for the rotor drive system). It should be noted that the costs of the control system hardware and software for these two systems are excluded.

3.2 Cost Estimate (1984 Dollars)

		US Dollars (000)
Fan		3,300
New Fan Centerbody (Nosecone, nacelle, tailcone)	1,000	
Fan Drive - Modified (coaxial gearbox)	400	
Fan Rotor	400	
Fan blades	1,500	
Settling Chamber Acoustic Treatment		1,100
Relocation of Control Room		250
Test Chamber		1,050
Acoustic Treatment	900	
Transport and Installation Equipment	150	
Acoustic Model Preparation Building.		100
Corner Vanes		550
Removal of existing vanes	50	
New vanes with acoustic panels	500	
Sting and Rotor Drive System		1,200
Sting Support	450	
Rotor Drive	750	
TOTAL		7,550

4. PROJECT SCHEDULE

The project schedule is shown in Figure 4. It covers only the construction phase of the project; the engineering and procurement (tendering and contract award) activities are excluded.

An optimum schedule has been assumed as already mentioned in the previous section. Duration of the individual activities were discussed both internally within DSMA and with potential suppliers (fan blades, acoustic treatment). The total duration of the construction phase up to the point of the aeroacoustic performance verification (commissioning) is 16 months, and the estimated shutdown of the facility is 6 months. It is felt that the fan modifications can be done faster than shown, especially when the proposed concept with coaxial gearbox is adopted. However, it is not likely that the facility shutdown can be reduced below 5 months.

The sting support and rotor drive are shown as requiring 16 months to completion. However, this activity is not necessarily connected with the facility shut down; furthermore, it could be initiated earlier than the remainder of the work.

5. CONCLUSIONS

As a result of this study, the following has been concluded:

- Mechanical feasibility has been established for implementation of BBN "Scheme B" for the modification of the 4 x 7 m tunnel for aeroacoustic research.

- The design goal for 50 percent reduction in fan speed has not been achieved and is not feasible with a conventional fan design.

- The test section noise level specification may well be achievable with the fan design proposed in the study. However, the substantiation of the acoustic performance of the fan assembly was not within the scope of this study and will be assessed by NASA.

- The closed test section performance of the facility with the new fan will be improved over the present configuration due to the improvement in fan efficiency. .

- The proposed modifications can be accomplished within a reasonable time and at a reasonable cost.

REFERENCES

(1) Hayden, R. E., and Wilby, J. F., "Sources, Paths, and Concepts for Reduction of Noise in the Test Section of the NASA Langley 4 x 7 m Wind Tunnel", NASA CR-172446-1, September, 1984.

(2) NASA Request for Proposal 1-14-5627,0236 Architect-Engineer Services for Feasibility Study for Modifications to the 4 x 7 Meter Tunnel for Aeroacoustic Research, Bulding 1212C. July 24, 1984.

(3) Hayden, R. E., "Addendum and Executive Summary: Sources, Paths, and Concepts for Reduction of Noise in the Test Section of the NASA Langley 4 x 7 m Wind Tunnel", NASA CR-172446-2, September, 1984.

(4) van Ditshuizen, J. C. A., and Ross, R., "Aerodynamic and Aeroacoustic Design Aspects" in "Construction 1976-1980", M. Seidel, Editor, D.N.W., May 1982.

(5) Applin, Z. T., "Flow Improvements in the Circuit of the Langley 4- by 7- Meter Tunnel", NASA TM-85662, December 1983.

TABLES

LANGLEY 4 X 7 M. TUNNEL - EXISTING, CLOSED T/S, OCT.30/84.

TEST SECTION CONDITIONS

MACH NUMBER	-	0.2850
TOTAL PRESSURE	-	1.0721 BARS
TOTAL TEMPERATURE	-	293.7000 DEG.K.
DYNAMIC PRESSURE	-	0.0576 BARS
CHORD REYNOLDS NUMBER	-	3.5796 MILLIONS

FAN

LOSS FACTOR(DPTF/QTS)	-	0.2493
TOTAL PRESSURE RISE	-	0.0144 BARS
AIR POWER	-	5898.3991 KW
EFFICIENCY	-	0.6700
(PTFO/PTFI)	-	1.0135
(RMF*SQRT(TRFI)/PRFI)	-	3346.2277
CIRCUIT TRANSIT TIME	-	12.0265 SEC.
EQUIV. CONTRACTION L.	-	9.0000 M.
PLENUM BLOCKAGE	-	0.0000
TUNNEL MASS FLOW	-	3478.4118 KGR./SEC.
FAN INLET UNIT RE.	-	1.7007 MILLIONS/M.
FAN OUTLET BLOCKAGE	-	3.0000 PERCENT
FAN DIFFUSER BLOCKAGE	-	8.1803 PERCENT

	AREA	M	PT	PS	TT	TS	U	LOSS FACTORS	
	SQ.M.		BARS		DEG.K.		M./SEC.	LOCAL	T/S
TEST SECTION	29.305	0.2850	1.0721	1.0133	293.700	289.005	97.061	0.0000	0.0000
T/S DIFFUSER	29.614	0.2817	1.0721	1.0146	293.700	289.113	95.942	0.0775	0.0758
CORNER 1	79.008	0.1018	1.0677	1.0600	293.700	293.093	34.898	0.1500	0.0200
CROSSLEG 1	79.008	0.1019	1.0666	1.0589	293.700	293.092	34.936	0.0775	0.0103
CORNER 2	86.304	0.0932	1.0660	1.0595	293.700	293.191	31.973	0.1500	0.0168
FAN INLET	86.304	0.0933	1.0650	1.0586	293.700	293.190	32.002	0.1975	0.0221
FAN	112.615	0.0714	1.0637	1.0600	293.700	293.401	24.511	-3.8046	-0.2500
FAN TAILCONE	112.615	0.0706	1.0781	1.0744	294.831	294.537	24.276	0.0473	0.0031
FAN DIFFUSER	141.448	0.0562	1.0780	1.0756	294.831	294.645	19.314	0.2024	0.0083
AIR OUTLET	263.329	0.0301	1.0775	1.0768	294.831	294.777	10.367	0.0000	0.0000
CORNER 3	263.329	0.0301	1.0775	1.0768	293.700	293.647	10.328	0.1500	0.0018
CROSSLEG 2	263.329	0.0301	1.0774	1.0767	293.700	293.647	10.328	0.0008	0.0000
CORNER 4	263.329	0.0301	1.0774	1.0767	293.700	293.647	10.328	0.1500	0.0018
SETTLING CHAMBER	263.329	0.0301	1.0773	1.0766	293.700	293.647	10.328	4.0031	0.0474
CONTRACTION	263.329	0.0302	1.0746	1.0739	293.700	293.647	10.355	0.0092	0.0001
TEST SECTION	29.305	0.2842	1.0745	1.0159	293.700	289.030	96.800	0.0412	0.0411

Table 1. Losses, Existing Circuit, Power Point
a) Closed Test Section

LANGLEY 4 X 7 M. TUNNEL - EXISTING, OPEN T/S, OCT. 30/84.

TEST SECTION CONDITIONS

MACH NUMBER	-	0. 2170
TOTAL PRESSURE	-	1. 0471 BARS
TOTAL TEMPERATURE	-	291. 7000 DEG. K.
DYNAMIC PRESSURE	-	0. 0334 BARS
CHORD REYNOLDS NUMBER	-	2. 7259 MILLIONS

FAN

LOSS FACTOR(DPTF/QTS)	-	0. 7192
TOTAL PRESSURE RISE	-	0. 0240 BARS
AIR POWER	-	5859. 7333 KW
EFFICIENCY	-	0. 8700
(PTFO/PTFI)	-	1. 0232
(RMF*SQRT(TRFI)/PRFI)	-	2607. 5574
CIRCUIT TRANSIT TIME	-	15. 6274 SEC.
EQUIV. CONTRACTION L.	-	9. 0000 M.
PLENUM BLOCKAGE	-	0. 0000
TUNNEL MASS FLOW	-	2648. 5909 KGR. /SEC.
FAN INLET UNIT RE.	-	1. 3015 MILLIONS/M.
FAN OUTLET BLOCKAGE	-	3. 0000 PERCENT
FAN DIFFUSER BLOCKAGE	-	8. 1803 PERCENT

	AREA	M	PT	PS	TT	TS	U	LOSS FACTORS	
								LOCAL	T/S
	SQ. M.		BARS		DEG. K.		M. /SEC.		
TEST SECTION	29. 305	0. 2170	1. 0471	1. 0133	291. 700	288. 978	73. 899	0. 000	0. 0000
COLLECTOR	32. 620	0. 1938	1. 0471	1. 0200	291. 700	289. 525	66. 067	0. 329	0. 2641
CORNER 1	79. 008	0. 0792	1. 0383	1. 0337	291. 700	291. 334	27. 082	0. 150	0. 0204
CROSSLEG 1	79. 008	0. 0793	1. 0376	1. 0330	291. 700	291. 334	27. 100	0. 077	0. 0105
CORNER 2	86. 304	0. 0725	1. 0372	1. 0334	291. 700	291. 393	24. 806	0. 150	0. 0171
FAN INLET	86. 304	0. 0726	1. 0367	1. 0329	291. 700	291. 393	24. 820	0. 197	0. 0225
FAN	112. 615	0. 0556	1. 0359	1. 0337	291. 700	291. 520	19. 016	-10. 734	-0. 7188
FAN TAILCONE	112. 615	0. 0545	1. 0599	1. 0577	293. 616	293. 441	18. 708	0. 048	0. 0032
FAN DIFFUSER	141. 448	0. 0434	1. 0598	1. 0584	293. 616	293. 505	14. 888	0. 203	0. 0085
AIR OUTLET	263. 329	0. 0233	1. 0595	1. 0591	293. 616	293. 584	7. 994	0. 000	0. 0000
CORNER 3	263. 329	0. 0232	1. 0595	1. 0591	291. 700	291. 669	7. 943	0. 150	0. 0018
CROSSLEG 2	263. 329	0. 0232	1. 0595	1. 0591	291. 700	291. 669	7. 943	0. 001	0. 0000
CORNER 4	263. 329	0. 0232	1. 0595	1. 0591	291. 700	291. 669	7. 943	0. 150	0. 0018
SETTLING CHAMBER	263. 329	0. 0232	1. 0594	1. 0590	291. 700	291. 669	7. 943	4. 003	0. 0479
CONTRACTION	263. 329	0. 0233	1. 0578	1. 0574	291. 700	291. 668	7. 955	0. 010	0. 0001
TEST SECTION	29. 305	0. 2147	1. 0578	1. 0244	291. 700	289. 035	73. 127	0. 325	0. 3217

Table 1. Losses, Existing Circuit, Power Point
b) Open Test Section

Assumed Conditions

$P_{S\infty} =$ 2116 PSF

$T_{S\infty} =$ 60°F

$P_M =$ 8,000 HP

2x1.0Q Screens in Settling Chamber

Quantity	Test Section	
	Closed	Open
Mach Number	0.285	0.217
Dynamic Pressure (PSF)	120	70
Fan Speed (RPM)	275	220
Loss Factor	0.25	0.72
Fan Efficiency (%)	0.67	0.87

Table 2. Loss Summary, Existing Circuit

LANGLEY 4X7 TUNNEL - FAN DESIGN CASE, CLOSED T/S, 120 PSF, OCT. 30/84.

TEST SECTION CONDITIONS

MACH NUMBER	-	0.2850
TOTAL PRESSURE	-	1.0721 BARS
TOTAL TEMPERATURE	-	293.7000 DEG. K.
DYNAMIC PRESSURE	-	0.0576 BARS
CHORD REYNOLDS NUMBER	-	3.5796 MILLIONS

FAN

LOSS FACTOR(DPTF/QTS)	-	0.2659
TOTAL PRESSURE RISE	-	0.0153 BARS
AIR POWER	-	5114.4664 KW
EFFICIENCY	-	0.8200
(PTFO/PTFI)	-	1.0144
(RMF*SQRT(TRFI)/PRFI)	-	3343.1790
CIRCUIT TRANSIT TIME	-	11.4852 SEC.
EQUIV. CONTRACTION L.	-	9.0000 M.
PLENUM BLOCKAGE	-	0.0000
TUNNEL MASS FLOW	-	3478.4118 KGR./SEC.
FAN INLET UNIT RE.	-	2.2069 MILLIONS/M.
FAN OUTLET BLOCKAGE	-	3.0000 PERCENT
FAN DIFFUSER BLOCKAGE	-	8.1803 PERCENT

	AREA	M	PT	PS	TT	TS	U	LOSS FACTORS	
	SQ. M.		BARS		DEG. K.		M./SEC.	LOCAL	T/S
TEST SECTION	29.305	0.2850	1.0721	1.0133	293.700	289.005	97.061	0.0000	0.0000
T/S DIFFUSER	29.614	0.2817	1.0721	1.0146	293.700	289.113	95.942	0.0775	0.0758
CORNER 1	79.008	0.1018	1.0677	1.0600	293.700	293.093	34.898	0.1700	0.0227
CROSSLEG 1	79.008	0.1019	1.0664	1.0587	293.700	293.092	34.941	0.0775	0.0103
CORNER 2	86.304	0.0932	1.0658	1.0594	293.700	293.190	31.978	0.1700	0.0190
FAN INLET	86.304	0.0933	1.0647	1.0583	293.700	293.189	32.011	0.0058	0.0007
FAN	86.833	0.0928	1.0647	1.0583	293.700	293.196	31.816	-2.4013	-0.2656
FAN TAILCONE	86.833	0.0916	1.0800	1.0737	294.900	294.406	31.489	0.0817	0.0089
FAN DIFFUSER	141.448	0.0561	1.0795	1.0771	294.900	294.715	19.290	0.2024	0.0083
AIR OUTLET	263.329	0.0301	1.0790	1.0783	294.900	294.847	10.354	0.0000	0.0000
CORNER 3	231.799	0.0341	1.0790	1.0781	293.700	293.632	11.716	0.1500	0.0023
CROSSLEG 2	231.799	0.0341	1.0789	1.0780	293.700	293.632	11.718	0.0058	0.0001
CORNER 4	231.799	0.0341	1.0789	1.0780	293.700	293.632	11.718	0.1500	0.0023
SETTLING CHAMBER	263.329	0.0300	1.0787	1.0781	293.700	293.647	10.314	6.3031	0.0745
CONTRACTION	263.329	0.0302	1.0744	1.0738	293.700	293.647	10.355	0.0092	0.0001
TEST SECTION	29.305	0.2842	1.0744	1.0158	293.700	289.030	96.800	0.0412	0.0411

Table 3. Losses, Proposed Circuit
a) Closed Test Section, Power Point

LANGLEY 4X7 TUNNEL - FAN DESIGN CASE, OPEN T/S, 70 PSF, OCT. 30/84

TEST SECTION CONDITIONS

MACH NUMBER	-	0.2170
TOTAL PRESSURE	-	1.0471 BARS
TOTAL TEMPERATURE	-	291.7000 DEG. K.
DYNAMIC PRESSURE	-	0.0334 BARS
CHORD REYNOLDS NUMBER	-	2.5310 MILLIONS

FAN

LOSS FACTOR(DPTF/QTS)	-	0.7126
TOTAL PRESSURE RISE	-	0.0238 BARS
AIR POWER	-	5444.2288 KW
EFFICIENCY	-	0.8000
(PTFO/PTFI)	-	1.0230
(RMF*SQRT(TRFI)/PRFI)	-	2247.1059
CIRCUIT TRANSIT TIME	-	17.1963 SEC.
EQUIV. CONTRACTION L.	-	9.0000 M.
PLENUM BLOCKAGE	-	0.0000
TUNNEL MASS FLOW	-	2283.3927 KGR./SEC.
FAN INLET UNIT RE.	-	1.4556 MILLIONS/M.
FAN OUTLET BLOCKAGE	-	3.0000 PERCENT
FAN DIFFUSER BLOCKAGE	-	8.1803 PERCENT

	AREA	M	PT	PS	TT	TS	U	LOSS FACTORS	
	SQ. M.		BARS		DEG. K.		M./SEC.	LOCAL	T/S
TEST SECTION	25.260	0.2170	1.0471	1.0133	291.700	288.978	73.899	0.0000	0.0000
COLLECTOR	28.118	0.1939	1.0471	1.0200	291.700	289.524	66.081	0.3462	0.2781
CORNER 1	79.008	0.0683	1.0378	1.0344	291.700	291.428	23.343	0.1700	0.0172
CROSSLEG 1	79.008	0.0683	1.0372	1.0339	291.700	291.428	23.358	0.0773	0.0078
CORNER 2	86.304	0.0625	1.0370	1.0341	291.700	291.472	21.381	0.1700	0.0144
FAN INLET	86.304	0.0625	1.0365	1.0337	291.700	291.472	21.392	0.0058	0.0005
FAN	86.833	0.0622	1.0365	1.0337	291.700	291.475	21.261	-8.5107	-0.7125
FAN TAILCONE	86.833	0.0610	1.0603	1.0575	293.598	293.380	20.920	0.0848	0.0070
FAN DIFFUSER	141.448	0.0374	1.0600	1.0590	293.598	293.516	12.830	0.2036	0.0063
AIR OUTLET	263.329	0.0201	1.0598	1.0595	293.598	293.574	6.890	0.0000	0.0000
CORNER 3	231.799	0.0227	1.0598	1.0594	291.700	291.670	7.778	0.1500	0.0017
CROSSLEG 2	231.799	0.0227	1.0598	1.0594	291.700	291.670	7.778	0.0059	0.0001
CORNER 4	231.799	0.0227	1.0598	1.0594	291.700	291.670	7.778	0.1500	0.0017
SETTLING CHAMBER	263.329	0.0200	1.0597	1.0594	291.700	291.677	6.846	6.3033	0.0560
CONTRACTION	263.329	0.0200	1.0578	1.0575	291.700	291.677	6.859	0.0098	0.0001
TEST SECTION	25.260	0.2148	1.0578	1.0244	291.700	289.034	73.141	0.3250	0.3218

Table 3. Losses, Proposed Circuit
b) Open Test Section, Power Point

LANGLEY 4X7 TUNNEL - FAN DESIGN CASE, OPEN T/S, 50 PSF, OCT. 30/84

TEST SECTION CONDITIONS

MACH NUMBER	-	0.1840
TOTAL PRESSURE	-	1.0375 BARS
TOTAL TEMPERATURE	-	291.0000 DEG. K.
DYNAMIC PRESSURE	-	0.0240 BARS
CHORD REYNOLDS NUMBER	-	2.1454 MILLIONS

FAN

LOSS FACTOR(DPTF/QTS)	-	0.7133
TOTAL PRESSURE RISE	-	0.0171 BARS
AIR POWER	-	3338.6595 KW
EFFICIENCY	-	0.8000
(PTFO/PTFI)	-	1.0166
(RMF*SQRT(TRFI)/PRFI)	-	1914.7184
CIRCUIT TRANSIT TIME	-	20.1294 SEC.
EQUIV. CONTRACTION L.	-	9.0000 M.
PLENUM BLOCKAGE	-	0.0000
TUNNEL MASS FLOW	-	1935.8910 KGR./SEC.
FAN INLET UNIT RE.	-	1.2359 MILLIONS/M.
FAN OUTLET BLOCKAGE	-	3.0000 PERCENT
FAN DIFFUSER BLOCKAGE	-	8.1803 PERCENT

	AREA	M	PT	PS	TT	TS	U	LOSS FACTORS	
	SQ. M.		BARS		DEG. K.		M./SEC.	LOCAL	T/S
TEST SECTION	25.260	0.1840	1.0375	1.0133	291.000	289.043	62.668	0.0000	0.0000
COLLECTOR	28.118	0.1646	1.0375	1.0181	291.000	289.431	56.108	0.3453	0.2777
CORNER 1	79.008	0.0581	1.0308	1.0284	291.000	290.803	19.861	0.1700	0.0172
CROSSLEG 1	79.008	0.0582	1.0304	1.0280	291.000	290.803	19.870	0.0774	0.0078
CORNER 2	86.304	0.0532	1.0302	1.0282	291.000	290.835	18.189	0.1700	0.0144
FAN INLET	86.304	0.0533	1.0299	1.0278	291.000	290.835	18.196	0.0058	0.0005
FAN	86.833	0.0529	1.0299	1.0279	291.000	290.837	18.085	-8.4863	-0.7125
FAN TAILCONE	86.833	0.0522	1.0470	1.0450	292.373	292.214	17.873	0.0861	0.0071
FAN DIFFUSER	141.448	0.0320	1.0468	1.0461	292.373	292.313	10.964	0.2041	0.0064
AIR OUTLET	263.329	0.0172	1.0467	1.0464	292.373	292.356	5.889	0.0000	0.0000
CORNER 3	231.799	0.0195	1.0467	1.0464	291.000	290.978	6.658	0.1500	0.0017
CROSSLEG 2	231.799	0.0195	1.0466	1.0463	291.000	290.978	6.659	0.0059	0.0001
CORNER 4	231.799	0.0195	1.0466	1.0463	291.000	290.978	6.659	0.1500	0.0017
SETTLING CHAMBER	263.329	0.0172	1.0466	1.0464	291.000	290.983	5.862	6.3033	0.0566
CONTRACTION	263.329	0.0172	1.0452	1.0450	291.000	290.983	5.869	0.0100	0.0001
TEST SECTION	25.260	0.1826	1.0452	1.0212	291.000	289.072	62.206	0.3250	0.3227

Table 3. Losses, Proposed Circuit
c) Open Test Section, Acoustic Design Point

Assumed Conditions

$P_{S\infty} =$ 2116 PSF

$T_{S\infty} =$ 60°F

Grid at Corner 3
Honeycomb and 4 Screens in Settling Chamber
New Fan
"Acoustic" Turning Vanes in Corners 1 and 2
Acoustic Lining in Crossleg 2 (Settling Chamber)
and Test Section

Quantity	Test Section		
	Closed	Open	
Mach Number	0.285	0.217	0.184
Dynamic Pressure (PSF)	120	70	50
Fan Speed (RPM)	182	158	135
Loss Factor	0.266	0.713	0.713
Fan Efficiency (%)	0.82	0.80	0.80
Fan Power (HP)	6,900	7,300	4,500
Fan Pressure Ratio	1.0144	1.0230	1.0166
Mass Flow (kg/s) $\dot{m}\sqrt{\theta}/\delta$ *	3343	2247	1915

* $\theta = T_o / 288$ K, $\delta = P_o / 101325$ Pa

Table 4. Loss Summary, Proposed Circuit

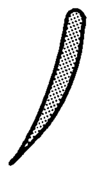
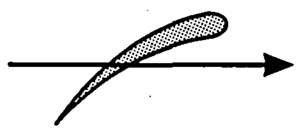
	Existing	Proposed
Tip Diameter	12.5 M	12.5 M
Hub Diameter	4.9 M	7.0 M
Inlet Guide Vanes 	None	5
Rotor Blades 	9	19
Stators 	7	7

Table 5. Comparison of Existing and Proposed Fan Geometries

LANGLEY 4X7 TUNNEL - CLOSED T/S, 120 PSF, ORIGINAL XLEG 2, OCT.30/84.

TEST SECTION CONDITIONS

MACH NUMBER	-	0.2850
TOTAL PRESSURE	-	1.0721 BARS
TOTAL TEMPERATURE	-	293.7000 DEG.K.
DYNAMIC PRESSURE	-	0.0576 BARS
CHORD REYNOLDS NUMBER	-	3.5796 MILLIONS

FAN

LOSS FACTOR(DPTF/QTS)	-	0.2653
TOTAL PRESSURE RISE	-	0.0153 BARS
AIR POWER	-	5114.4664 KW
EFFICIENCY	-	0.8200
(PTFO/PTFI)	-	1.0144
(RMF*SQRT(TRFI)/PRFI)	-	3343.1790
CIRCUIT TRANSIT TIME	-	11.8220 SEC.
EQUIV. CONTRACTION L.	-	9.0000 M.
PLENUM BLOCKAGE	-	0.0000
TUNNEL MASS FLOW	-	3478.4118 KGR./SEC.
FAN INLET UNIT RE.	-	2.2069 MILLIONS/M.
FAN OUTLET BLOCKAGE	-	3.0000 PERCENT
FAN DIFFUSER BLOCKAGE	-	8.1803 PERCENT

	AREA	M	PT	PS	TT	TS	U	LOSS FACTORS	
	SQ.M.		BARS		DEG.K.		M./SEC.	LOCAL	T/S
TEST SECTION	29.305	0.2850	1.0721	1.0133	293.700	289.005	97.061	0.0000	0.0000
T/S DIFFUSER	29.614	0.2817	1.0721	1.0146	293.700	289.113	95.942	0.0775	0.0758
CORNER 1	79.008	0.1018	1.0677	1.0600	293.700	293.093	34.898	0.1700	0.0227
CROSSLEG 1	79.008	0.1019	1.0664	1.0587	293.700	293.092	34.941	0.0775	0.0103
CORNER 2	86.304	0.0932	1.0658	1.0594	293.700	293.190	31.978	0.1700	0.0190
FAN INLET	86.304	0.0933	1.0647	1.0583	293.700	293.189	32.011	0.0058	0.0007
FAN	86.833	0.0928	1.0647	1.0583	293.700	293.196	31.816	-2.4013	-0.2656
FAN TAILCONE	86.833	0.0916	1.0800	1.0737	294.900	294.406	31.489	0.0817	0.0089
FAN DIFFUSER	141.448	0.0561	1.0795	1.0771	294.900	294.715	19.290	0.2024	0.0083
AIR OUTLET	263.329	0.0301	1.0790	1.0783	294.900	294.847	10.354	0.0000	0.0000
CORNER 3	263.329	0.0300	1.0790	1.0783	293.700	293.647	10.312	0.1500	0.0018
CROSSLEG 2	263.329	0.0300	1.0789	1.0782	293.700	293.647	10.312	0.0058	0.0001
CORNER 4	263.329	0.0300	1.0789	1.0782	293.700	293.647	10.312	0.1500	0.0018
SETTLING CHAMBER	263.329	0.0300	1.0788	1.0781	293.700	293.647	10.312	6.3031	0.0745
CONTRACTION	263.329	0.0302	1.0745	1.0738	293.700	293.647	10.352	0.0092	0.0001
TEST SECTION	29.305	0.2842	1.0745	1.0159	293.700	289.032	96.781	0.0412	0.0411

Table 6. Losses, Proposed Circuit, Original Settling Chamber Area
a) Closed Test Section

LANGLEY 4X7 TUNNEL - OPEN T/S, 70 PSF, ORIGINAL XLEG 2, OCT.30/84.

TEST SECTION CONDITIONS

MACH NUMBER	-	0.2170
TOTAL PRESSURE	-	1.0471 BARS
TOTAL TEMPERATURE	-	291.7000 DEG.K.
DYNAMIC PRESSURE	-	0.0334 BARS
CHORD REYNOLDS NUMBER	-	2.5310 MILLIONS

FAN

LOSS FACTOR(DPTF/QTS)	-	0.7122
TOTAL PRESSURE RISE	-	0.0238 BARS
AIR POWER	-	5444.2288 KW
EFFICIENCY	-	0.8000
(PTFO/PTFI)	-	1.0230
(RMF*SQRT(TRFI)/PRFI)	-	2247.1059
CIRCUIT TRANSIT TIME	-	17.7033 SEC.
EQUIV. CONTRACTION L.	-	9.0000 M.
PLENUM BLOCKAGE	-	0.0000
TUNNEL MASS FLOW	-	2283.3927 KGR./SEC.
FAN INLET UNIT RE.	-	1.4556 MILLIONS/M.
FAN OUTLET BLOCKAGE	-	3.0000 PERCENT
FAN DIFFUSER BLOCKAGE	-	8.1803 PERCENT

	AREA	M	PT	PS	TT	TS	U	LOSS FACTORS	
	SQ.M.		BARS		DEG.K.		M./SEC.	LOCAL	T/S
TEST SECTION	25.260	0.2170	1.0471	1.0133	291.700	288.978	73.899	0.0000	0.0000
COLLECTOR	28.118	0.1939	1.0471	1.0200	291.700	289.524	66.081	0.3462	0.2781
CORNER 1	79.008	0.0683	1.0378	1.0344	291.700	291.428	23.343	0.1700	0.0172
CROSSLEG 1	79.008	0.0683	1.0372	1.0339	291.700	291.428	23.358	0.0773	0.0078
CORNER 2	86.304	0.0625	1.0370	1.0341	291.700	291.472	21.381	0.1700	0.0144
FAN INLET	86.304	0.0625	1.0365	1.0337	291.700	291.472	21.392	0.0058	0.0005
FAN	86.833	0.0622	1.0365	1.0337	291.700	291.475	21.261	-8.5107	-0.7125
FAN TAILCONE	86.833	0.0610	1.0603	1.0575	293.598	293.380	20.920	0.0848	0.0070
FAN DIFFUSER	141.448	0.0374	1.0600	1.0590	293.598	293.516	12.830	0.2036	0.0063
AIR OUTLET	263.329	0.0201	1.0598	1.0595	293.598	293.574	6.890	0.0000	0.0000
CORNER 3	263.329	0.0200	1.0598	1.0595	291.700	291.677	6.846	0.1500	0.0013
CROSSLEG 2	263.329	0.0200	1.0598	1.0595	291.700	291.677	6.846	0.0058	0.0001
CORNER 4	263.329	0.0200	1.0598	1.0595	291.700	291.677	6.846	0.1500	0.0013
SETTLING CHAMBER	263.329	0.0200	1.0597	1.0594	291.700	291.677	6.846	6.3033	0.0560
CONTRACTION	263.329	0.0200	1.0579	1.0576	291.700	291.677	6.859	0.0098	0.0001
TEST SECTION	25.260	0.2148	1.0579	1.0244	291.700	289.034	73.141	0.3250	0.3218

Table 6. Losses, Proposed Circuit, Original Settling Chamber Area
b) Open Test Section

TC Hz	Attenuation, DB						
	Turning Vane Proposed By BBN (Profiled Vane)	Turning Vane Concept Similar To DNW (Rolled Plate Vane)					
		1.6 M Chord		2.5 M Chord		4.0 M Chord	
			Δ		Δ		Δ
125	8	0	8	4	4	8	0
250	12	6	6	10	2	13	–
500	10	14	–	13	–	10	–
1,000	10	10	–	10	–	10	–
2,000	10	10	–	10	–	10	–
4,000	5–10	10	–	10	–	10	–
8,000	5–10	9	–	10	–	10	–

Table 7. Comparison of Acoustic Turning Vane Concepts

FIGURES

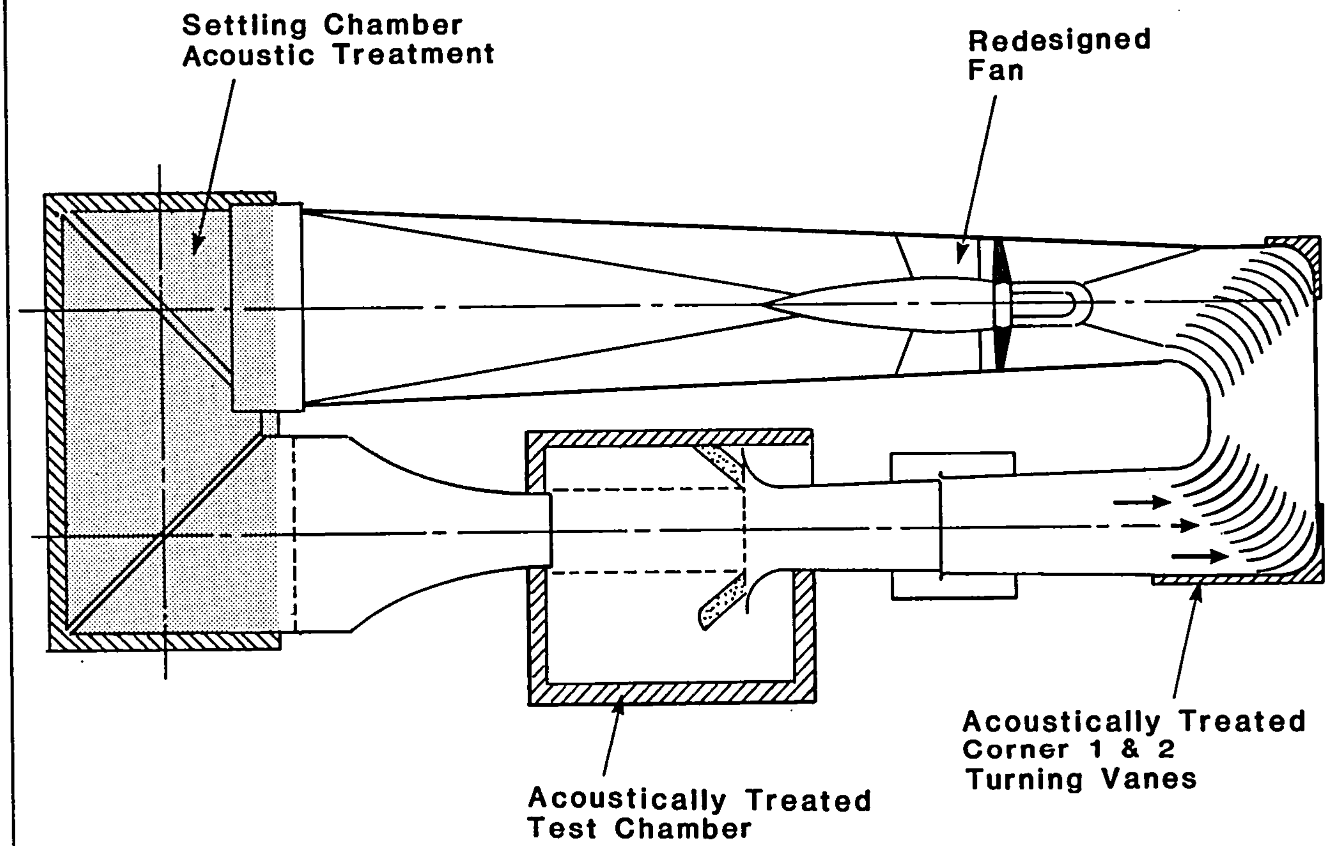


Figure 1. Scheme B (Significant Fan Redesign)
Acoustic Treatment of the 4x7 m Tunnel Circuit
From Reference 3

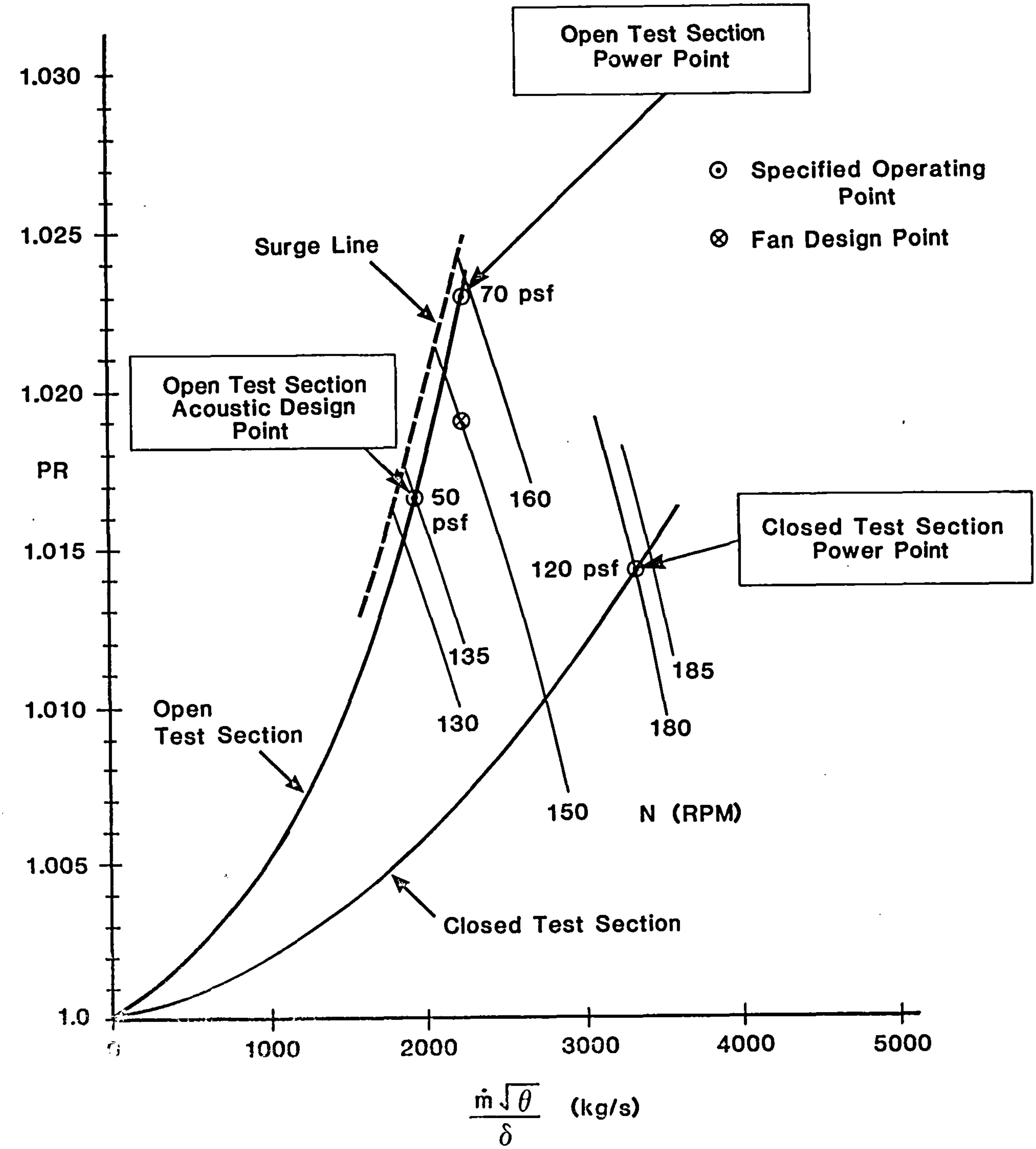


Figure 2. Approximate Fan Performance

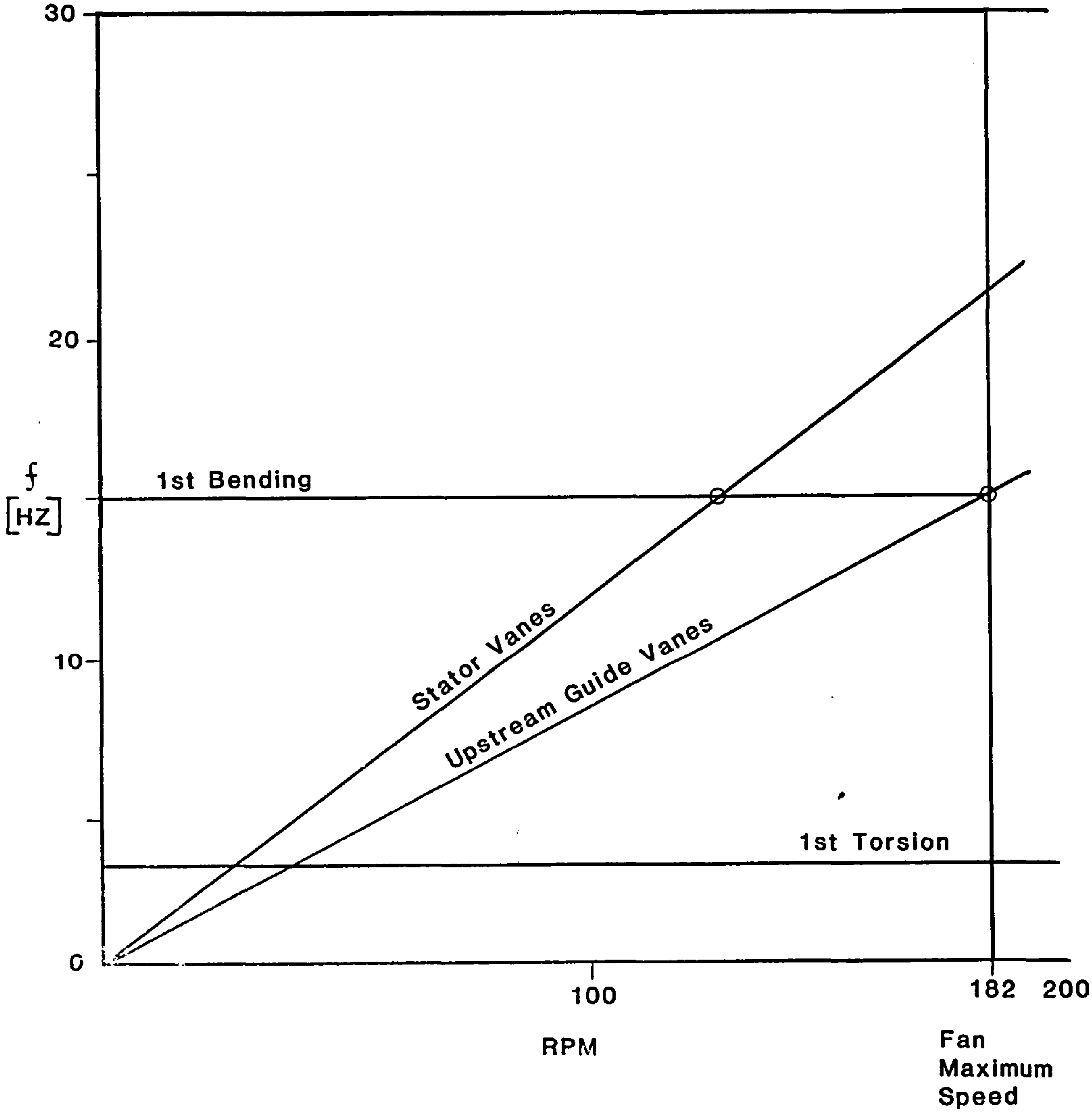


Figure 3. Fan Blade Interference Diagram

DSMA Engineering Corporation

Orlando, Florida

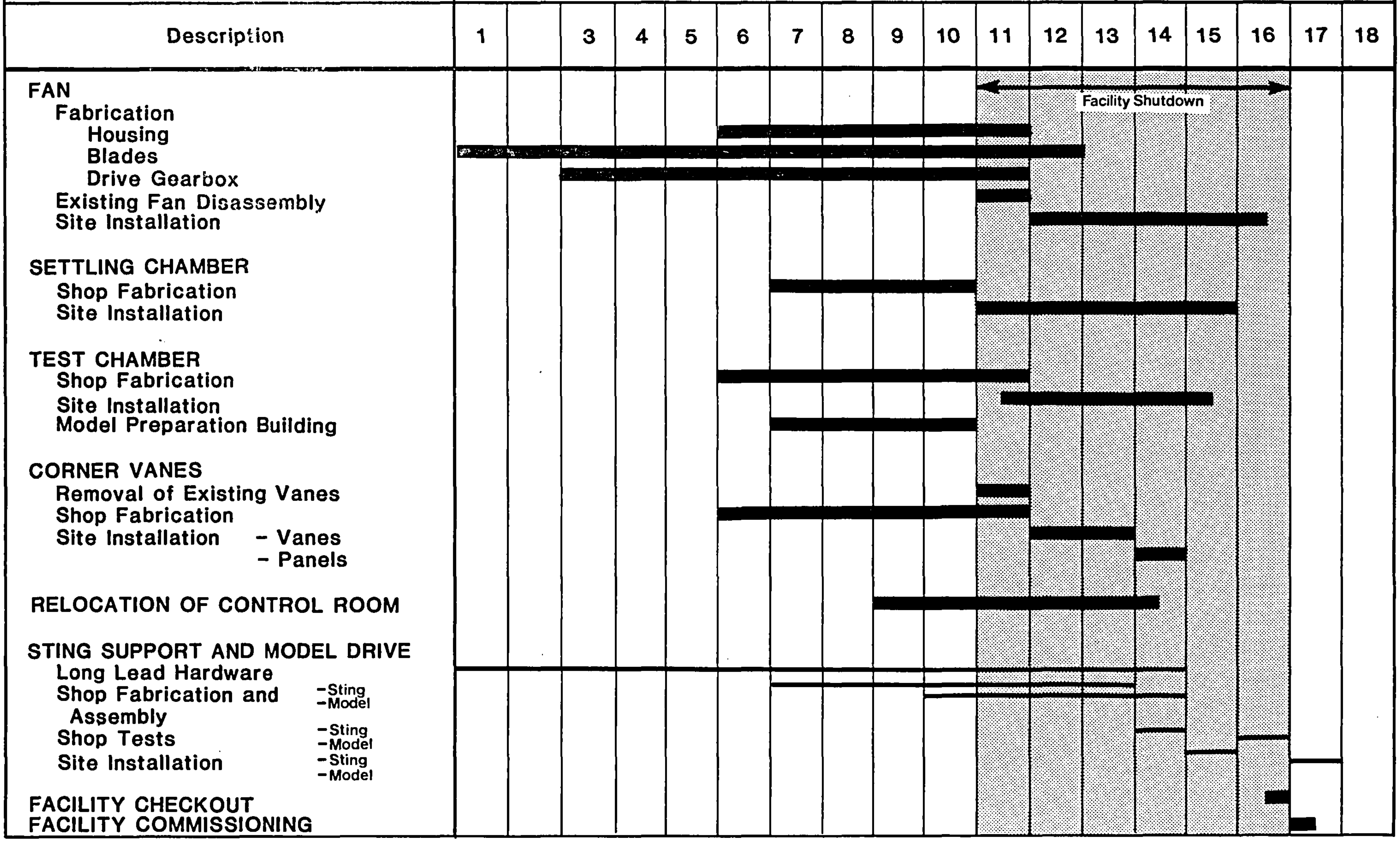
Figure 4. Project Schedule - Construction

Project No. 4054

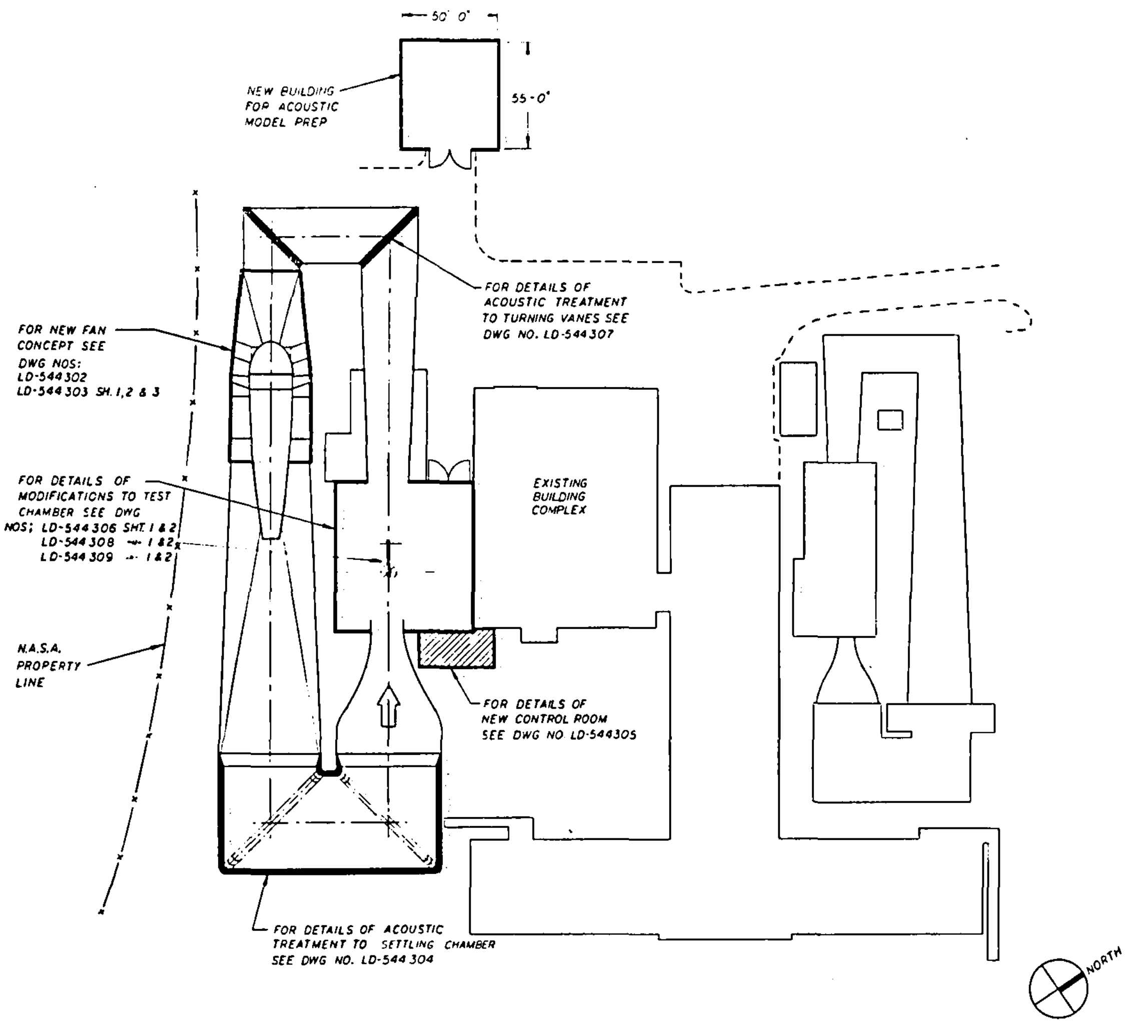
Project Title: NASA Feasibility Study Modifications To The 4x7 Meter Tunnel For Acoustic Research

Date: Nov. 2/84

Sheet 1 Of 1



DRAWINGS



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Engineers and Constructors
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NAME	ORGANIZATION	DATE	NAME	ORGANIZATION	DATE

MATERIAL
SCALE 1" = 30'-0"

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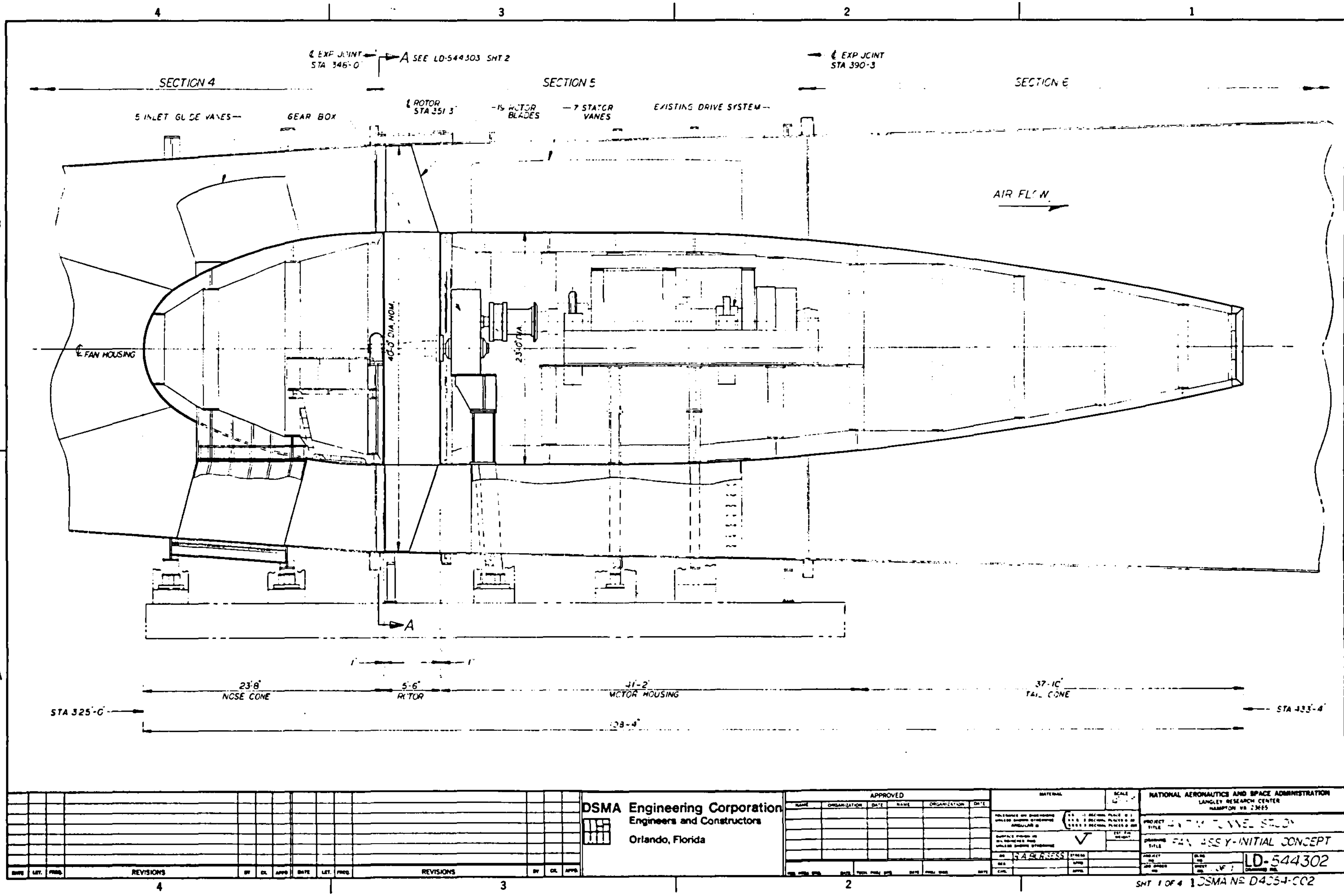
LANGLEY RESEARCH CENTER
HAMPTON VA 23665

PROJECT TITLE: 4 x 7 M. TUNNEL STUDY
DRAWING TITLE: KEY DIAGRAM FOR PROPOSED MODIFICATIONS
DR: V. HIGGINS
DRAWING NO. LD-544301

1 DSMA Nº. 04054-001



¢ EXP JOINT STA 346-0
A SEE LD-544303 SHT 2
¢ EXP JOINT STA 390-3
SECTION 4
SECTION 5
SECTION 6
¢ ROTOR STA 351 3"
-15 ROTOR BLADES
-7 STATOR VANES
EXISTING DRIVE SYSTEM
5 INLET GUIDE VANES
GEAR BOX
AIR FLOW
¢ FAN HOUSING
40-0" DIA. NOM.
23-0" DIA
A
23-8" NOSE CONE
5-6" ROTOR
41-2" MOTOR HOUSING
37-10" TAIL CONE
STA 325-0"
STA 433-4"
108-4"
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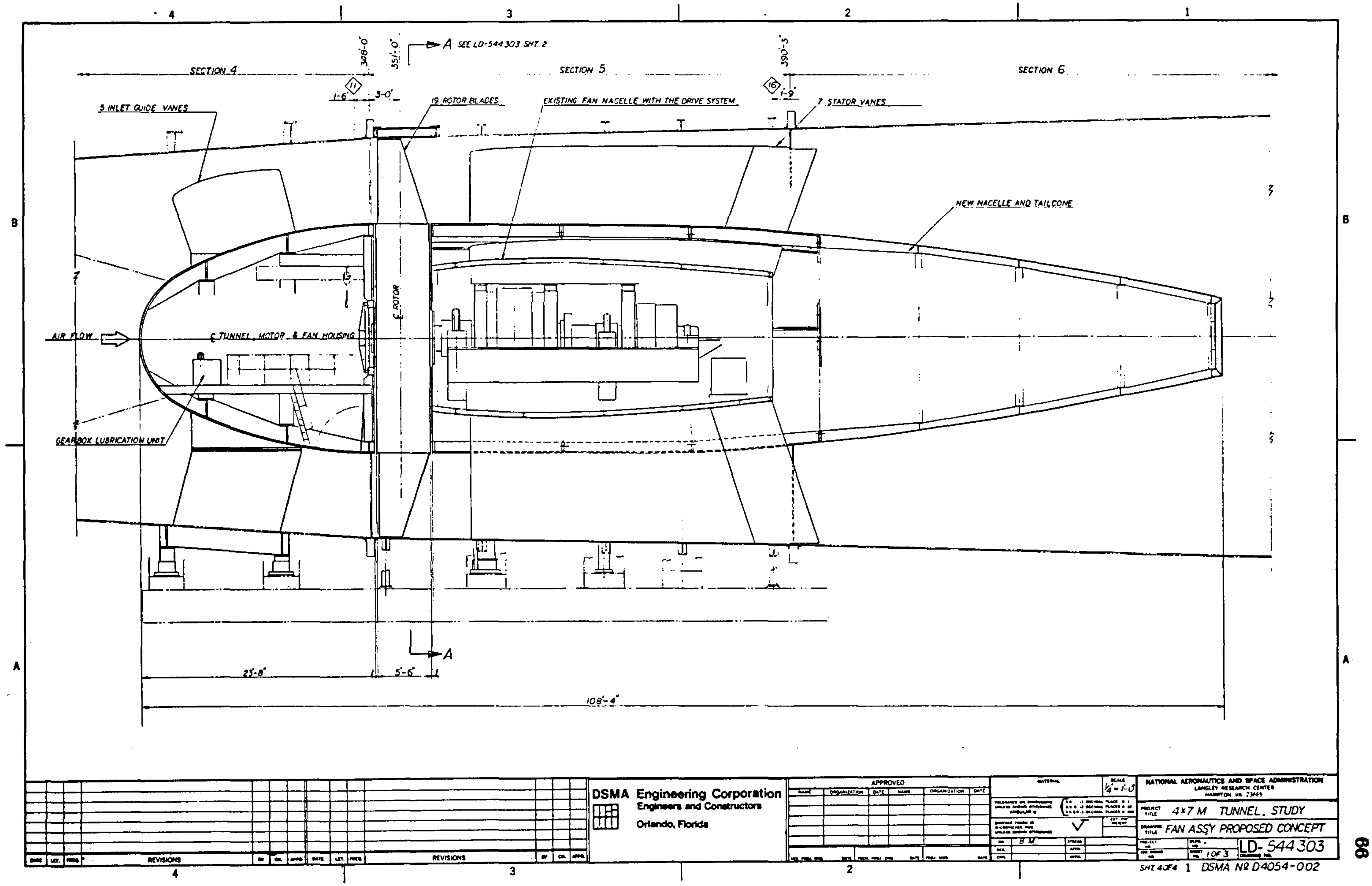
MATERIAL
SCALE
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LANGLEY RESEARCH CENTER
HAMPTON VA 23665
FAN ASSY-INITIAL CONCEPT
G A BURGESS
LD-544302
SHT 1 OF 4
DSMA Nº D4054-002



SECTION 4
SECTION 5
SECTION 6
A SEE LD-544303 SHT. 2
348'-0"
351'-0"
390'-3"
1'-6"
11
3'-0"
16
1'-9"
5 INLET GUIDE VANES
19 ROTOR BLADES
EXISTING FAN NACELLE WITH THE DRIVE SYSTEM
7 STATOR VANES
NEW NACELLE AND TAILCONE
℄ ROTOR
AIR FLOW
℄ TUNNEL, MOTOR & FAN HOUSING
GEARBOX LUBRICATION UNIT
A
25'-8"
5'-6"
108'-4"
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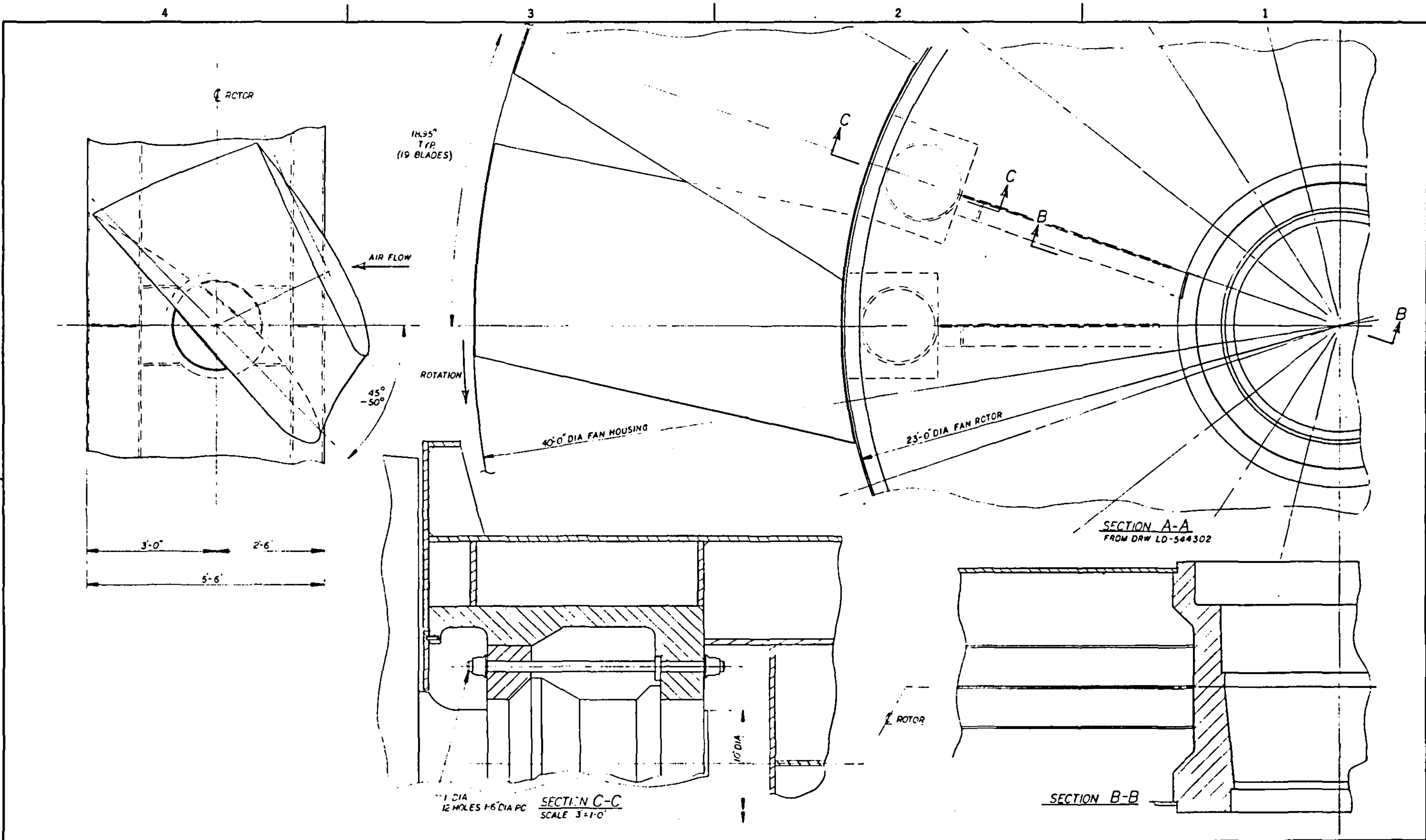
MATERIAL
SCALE 1/4" = 1'-0"
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LANGLEY RESEARCH CENTER
HAMPTON VA 23665
PROJECT TITLE 4x7 M TUNNEL. STUDY
DRAWING TITLE FAN ASSY PROPOSED CONCEPT
B M
SHEET NO. 1 OF 3
LD-544303
DRAWING NO.
SHT. 4 OF 4 1 DSMA Nº D4054-002



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SCALE: 3"=1'-0" NOTED

PROJECT TITLE: 4 x 7 M. TUNNEL STUDY

DRAWING TITLE: FAN ROTOR/BLADES PROPOSED CONCEPT

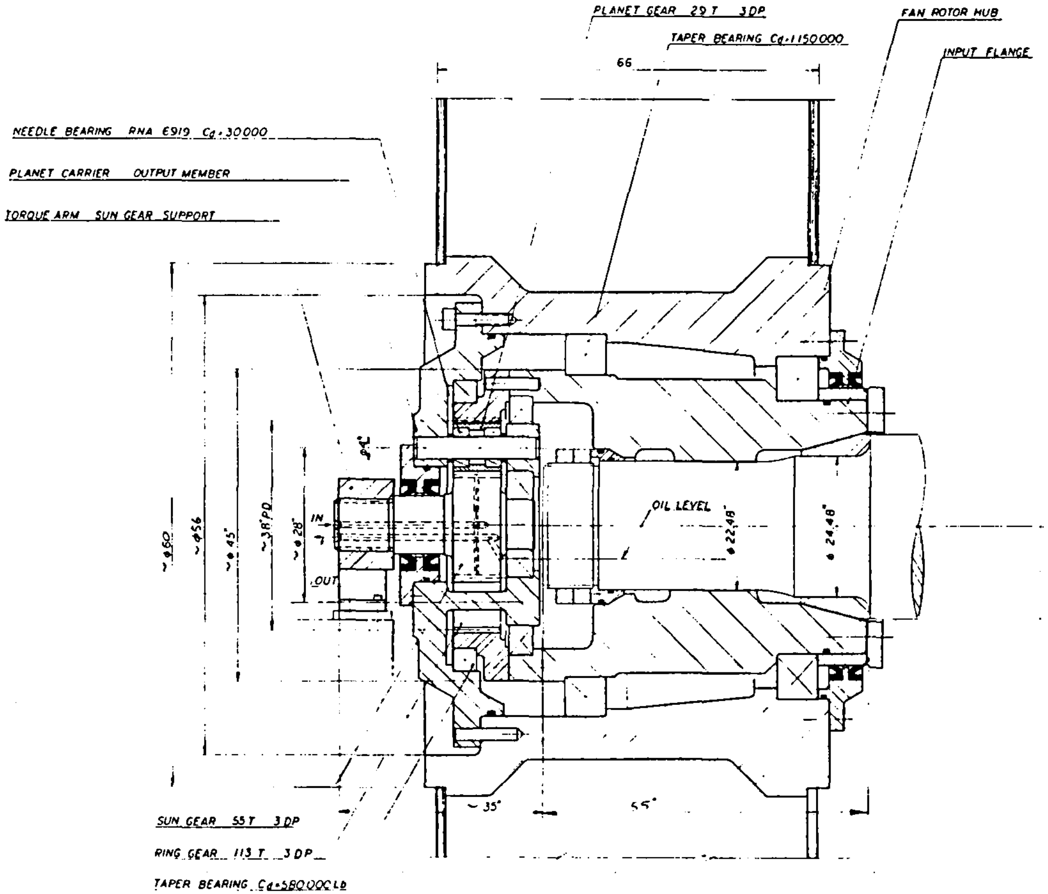
DR: R.A. BURGESS

12I2C

SHEET NO. 2 OF 3

LD- 544303

1 DSMA Nº D4C54-003



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SCALE 1:10

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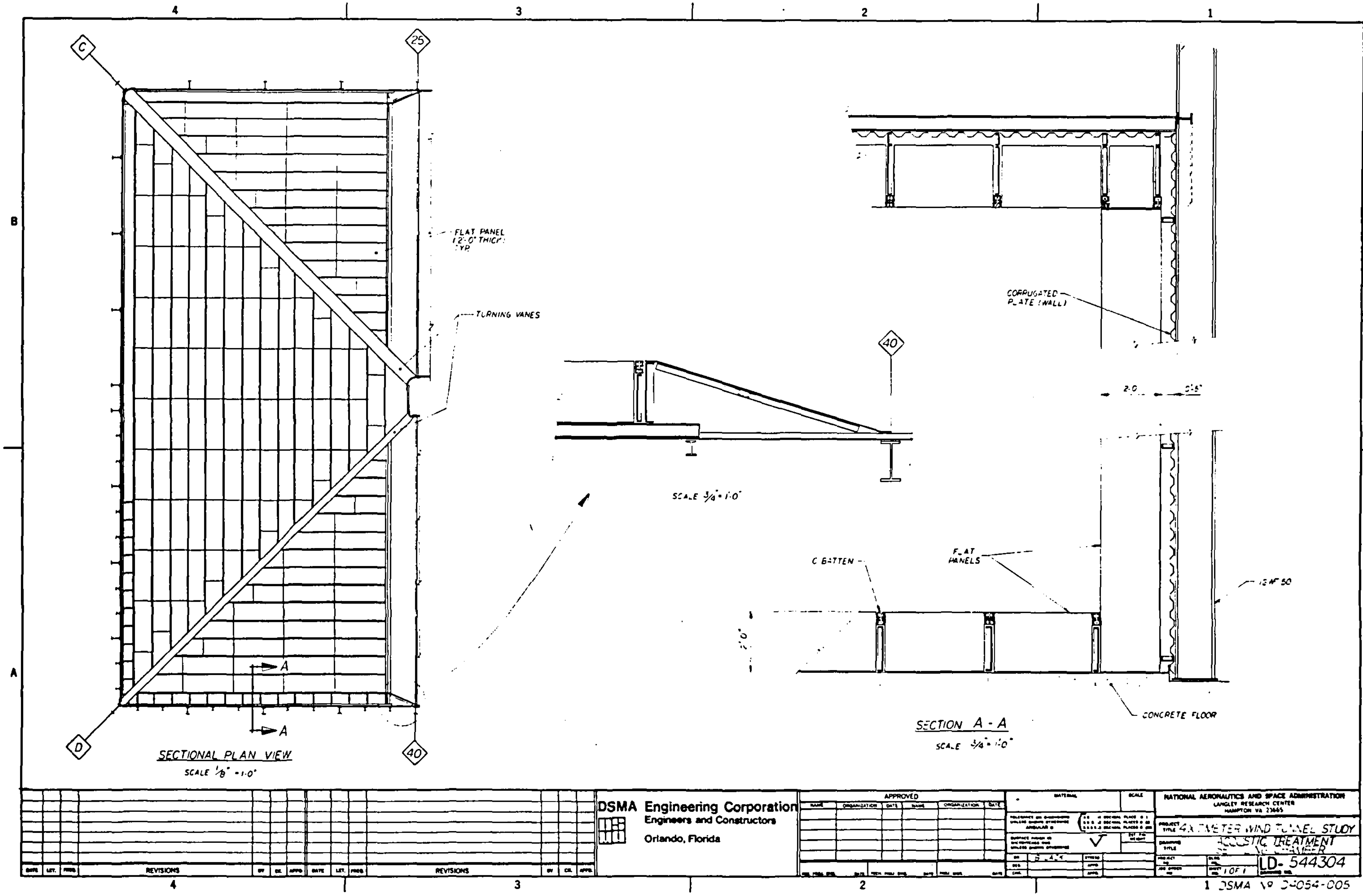
PROJECT TITLE: [illegible] TUNNEL STUDY

DRAWING TITLE: FAN DRIVE COAXIAL GEARBOX CONCEPT

SHEET 3 OF 3

LD-544303

SHT. 3 OF 4 1 DSMA № 04054-002



FLAT PANEL 12'-0" THICK TYP
TURNING VANES
SECTIONAL PLAN VIEW
SCALE 1/8" = 1'-0"
SCALE 3/4" = 1'-0"
CORRUGATED PLATE (WALL)
C BATTEN
FLAT PANELS
CONCRETE FLOOR
SECTION A - A
SCALE 3/4" = 1'-0"
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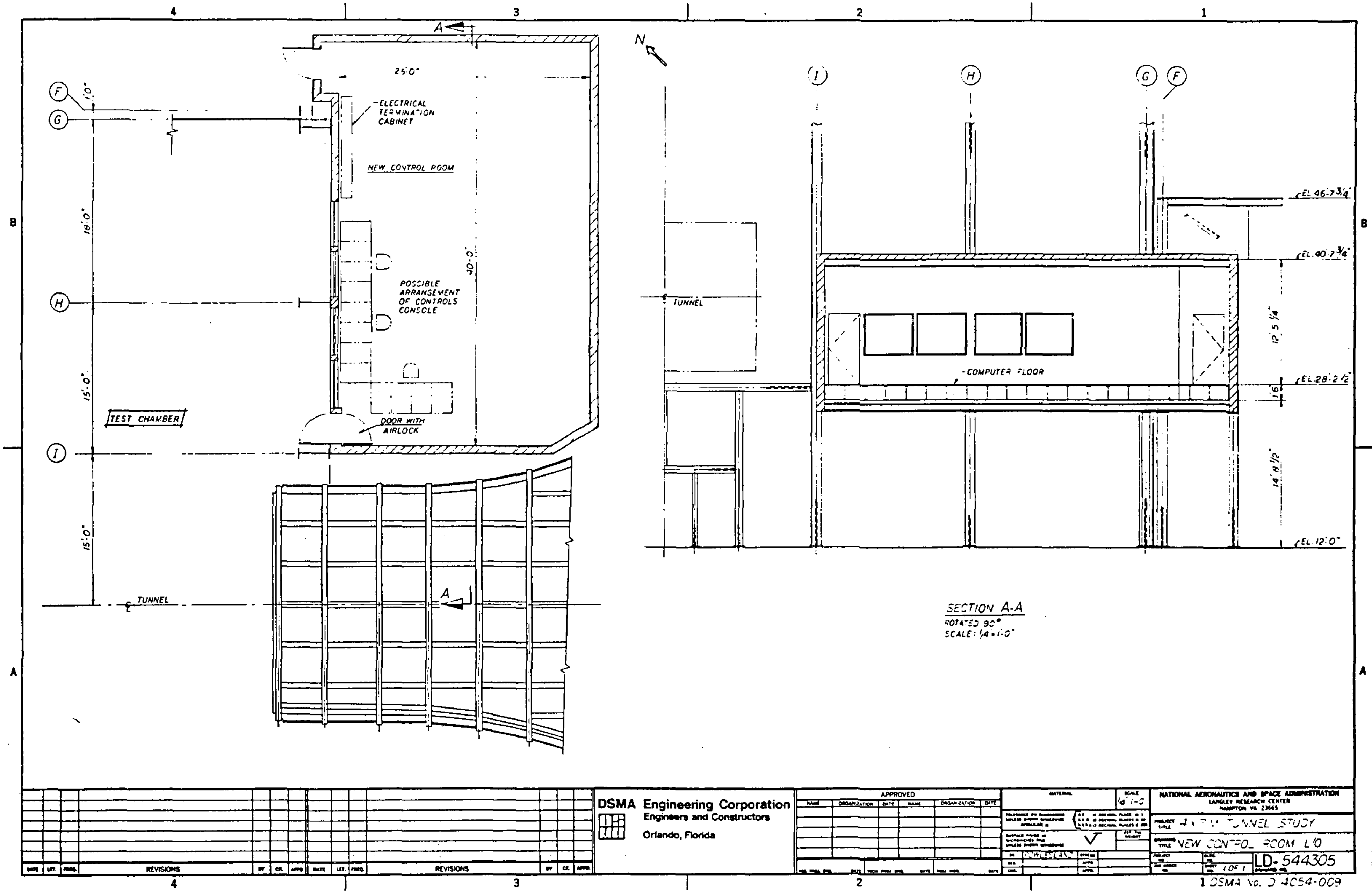
APPROVED
MATERIAL
SCALE
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LANGLEY RESEARCH CENTER
HAMPTON VA 23665
PROJECT TITLE
DRAWING TITLE
ACOUSTIC TREATMENT
1 OF 1
LD- 544304


A
N
25'0"
1'0"
18'0"
15'0"
15'0"
40'0"
ELECTRICAL TERMINATION CABINET
NEW CONTROL ROOM
POSSIBLE ARRANGEMENT OF CONTROLS CONSOLE
DOOR WITH AIRLOCK
TEST CHAMBER
TUNNEL
F
G
H
I
EL. 46'-7 3/4"
EL. 40'-7 3/4"
EL. 28'-2 1/2"
EL. 12'0"
12'-5 1/4"
1'6"
14'-8 1/2"
COMPUTER FLOOR
SECTION A-A
ROTATED 90°
SCALE: 1/4"=1'-0"
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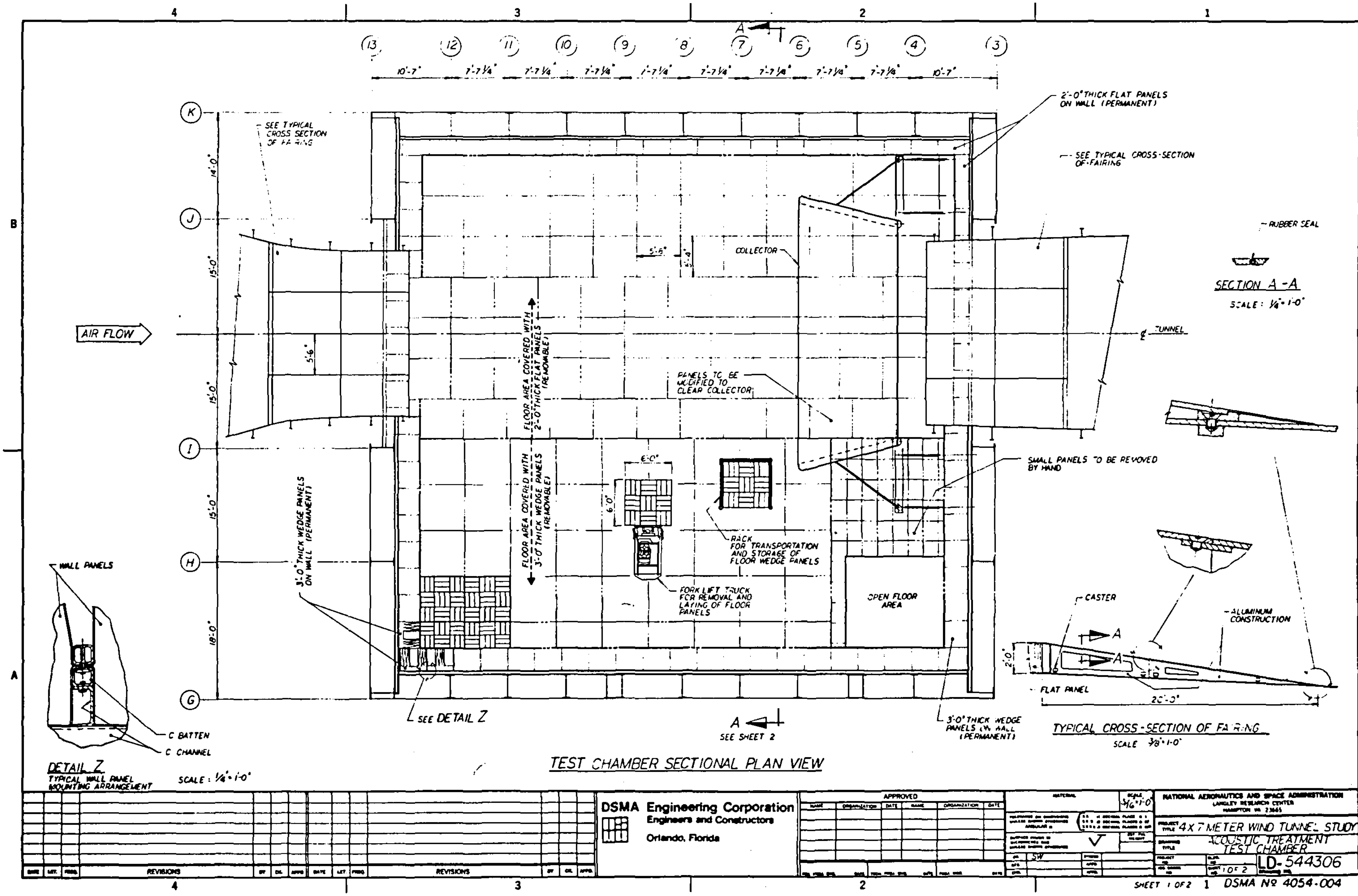
NEW CONTROL ROOM L/O
1 OF 1
LD-544305
1 DSMA No. D 4054-009



TEST CHAMBER SECTIONAL PLAN VIEW
AIR FLOW
10'-7" 7'-7 1/4" 7'-7 1/4" 7'-7 1/4" 7'-7 1/4" 7'-7 1/4" 7'-7 1/4" 7'-7 1/4" 7'-7 1/4" 10'-7"
14'-0" 15'-0" 15'-0" 15'-0" 18'-0"
SEE TYPICAL CROSS SECTION OF FAIRING
2'-0" THICK FLAT PANELS ON WALL (PERMANENT)
SEE TYPICAL CROSS-SECTION OF FAIRING
COLLECTOR
5'-6"
5'-4"
FLOOR AREA COVERED WITH 2'-0" THICK FLAT PANELS (REMOVABLE)
FLOOR AREA COVERED WITH 3'-0" THICK WEDGE PANELS (REMOVABLE)
PANELS TO BE MODIFIED TO CLEAR COLLECTOR
SMALL PANELS TO BE REMOVED BY HAND
6'-0"
RACK FOR TRANSPORTATION AND STORAGE OF FLOOR WEDGE PANELS
FORK LIFT TRUCK FOR REMOVAL AND LAYING OF FLOOR PANELS
OPEN FLOOR AREA
3'-0" THICK WEDGE PANELS ON WALL (PERMANENT)
SEE DETAIL Z
A
SEE SHEET 2
3'-0" THICK WEDGE PANELS ON WALL (PERMANENT)
℄ TUNNEL
RUBBER SEAL
SECTION A-A
SCALE: 1/4"=1'-0"
CASTER
ALUMINUM CONSTRUCTION
FLAT PANEL
2'-0"
20'-0"
TYPICAL CROSS-SECTION OF FAIRING
SCALE 3/8"=1'-0"
WALL PANELS
C BATTEN
C CHANNEL
DETAIL Z
TYPICAL WALL PANEL MOUNTING ARRANGEMENT
SCALE: 1/4"=1'-0"
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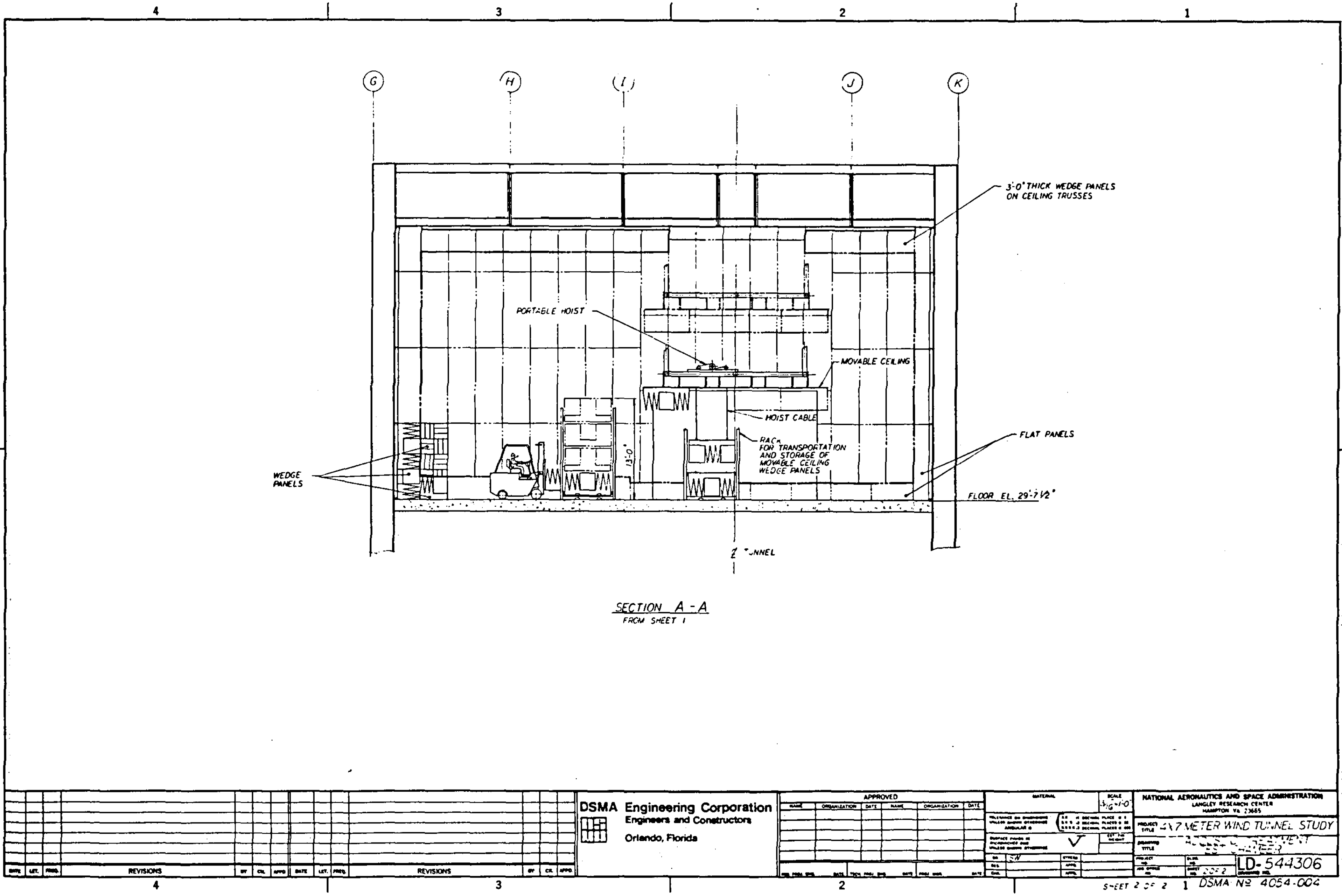
HAMPTON VA 23665
3/16"=1'-0"
4 X 7 METER WIND TUNNEL STUDY
ACOUSTIC TREATMENT TEST CHAMBER
SW
1 OF 2
LD-544306
SHEET 1 OF 2
DSMA Nº 4054-004



G
H
I
J
K
3'-0" THICK WEDGE PANELS ON CEILING TRUSSES
PORTABLE HOIST
MOVABLE CEILING
HOIST CABLE
RACK FOR TRANSPORTATION AND STORAGE OF MOVABLE CEILING WEDGE PANELS
FLAT PANELS
WEDGE PANELS
13'-0"
FLOOR EL. 29'-7 1/2"
℄ TUNNEL
SECTION A-A
FROM SHEET 1
DSMA Engineering Corporation
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MATERIAL
SCALE 3/16"=1'-0"
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
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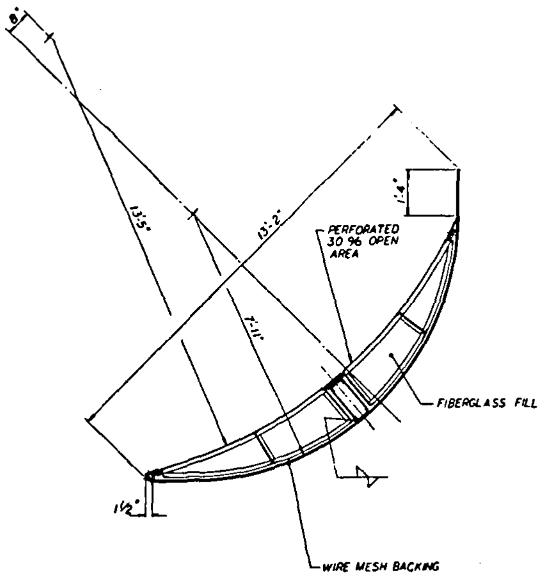
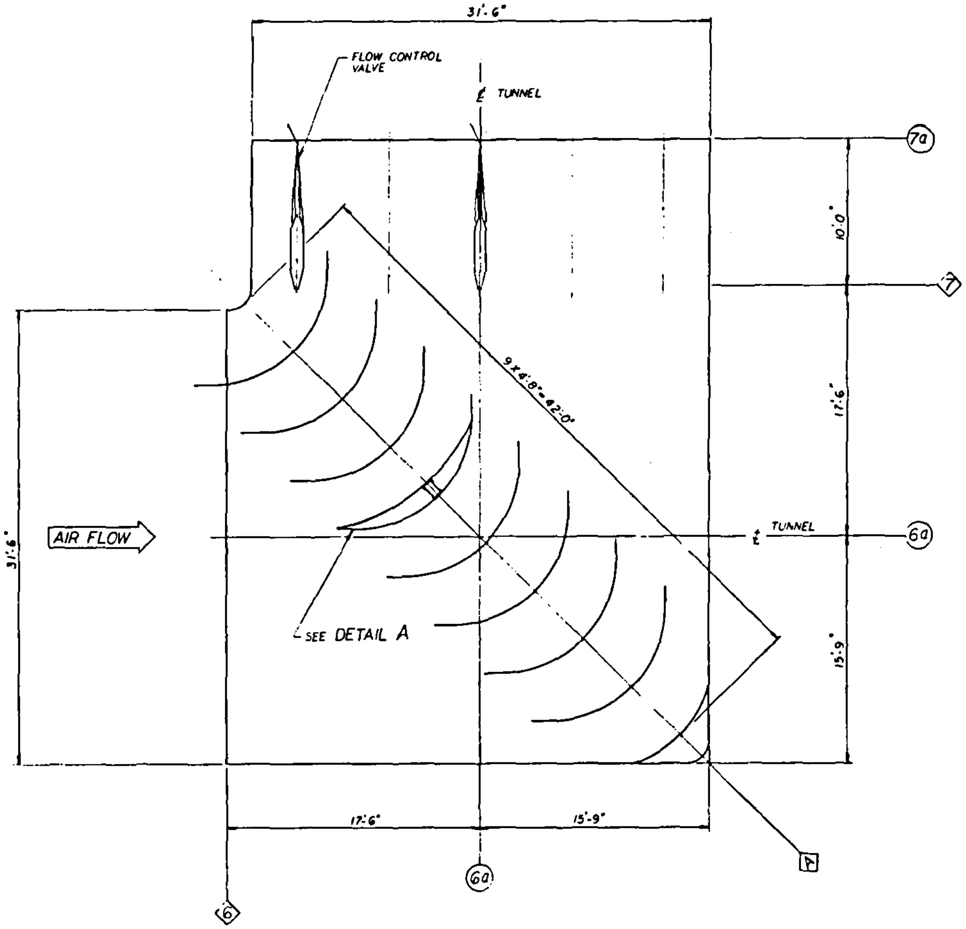
PROJECT TITLE 4X7 METER WIND TUNNEL STUDY
DRAWING TITLE
LD-544306
SHEET 2 OF 2
DSMA Nº 4054-004



DETAIL A

TYPICAL ACOUSTICALLY TREATED TURNING VANE

SCALE: NONE



FIRST CORNER SECTIONAL PLAN VIEW

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SCALE $\frac{1}{4}" = 1'-0"$

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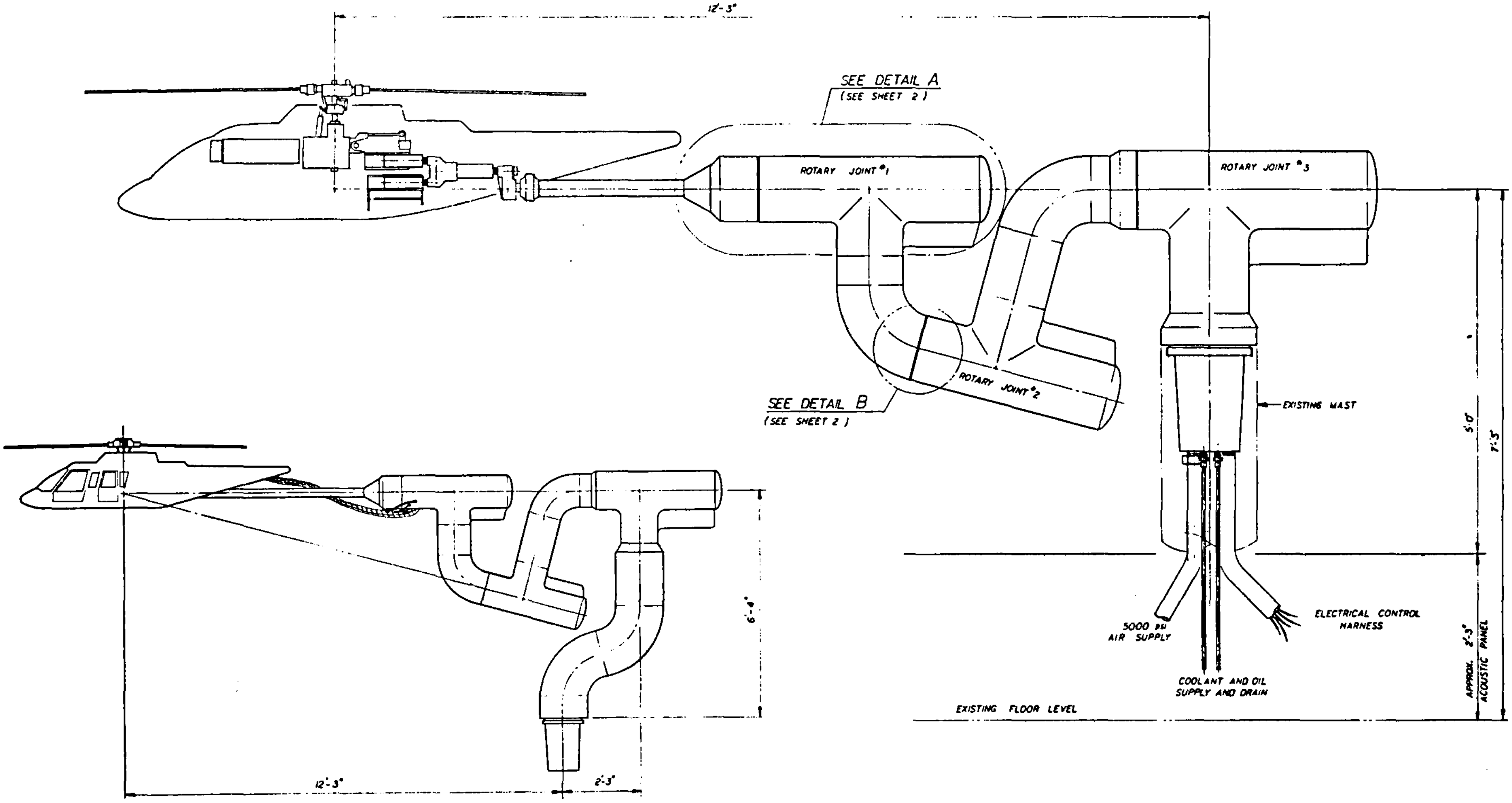
PROJECT TITLE: 4 x 7 METER WIND TUNNEL STUDY

DRAWING TITLE: ACOUSTIC TREATMENT TURNING VANES

SHEET 0F 1

DRAWING NO. LD-544307

DSMA № 4054-006



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SCALE 1/8 & 1/16

PROJECT TITLE 4x7M TUNNEL STUDY

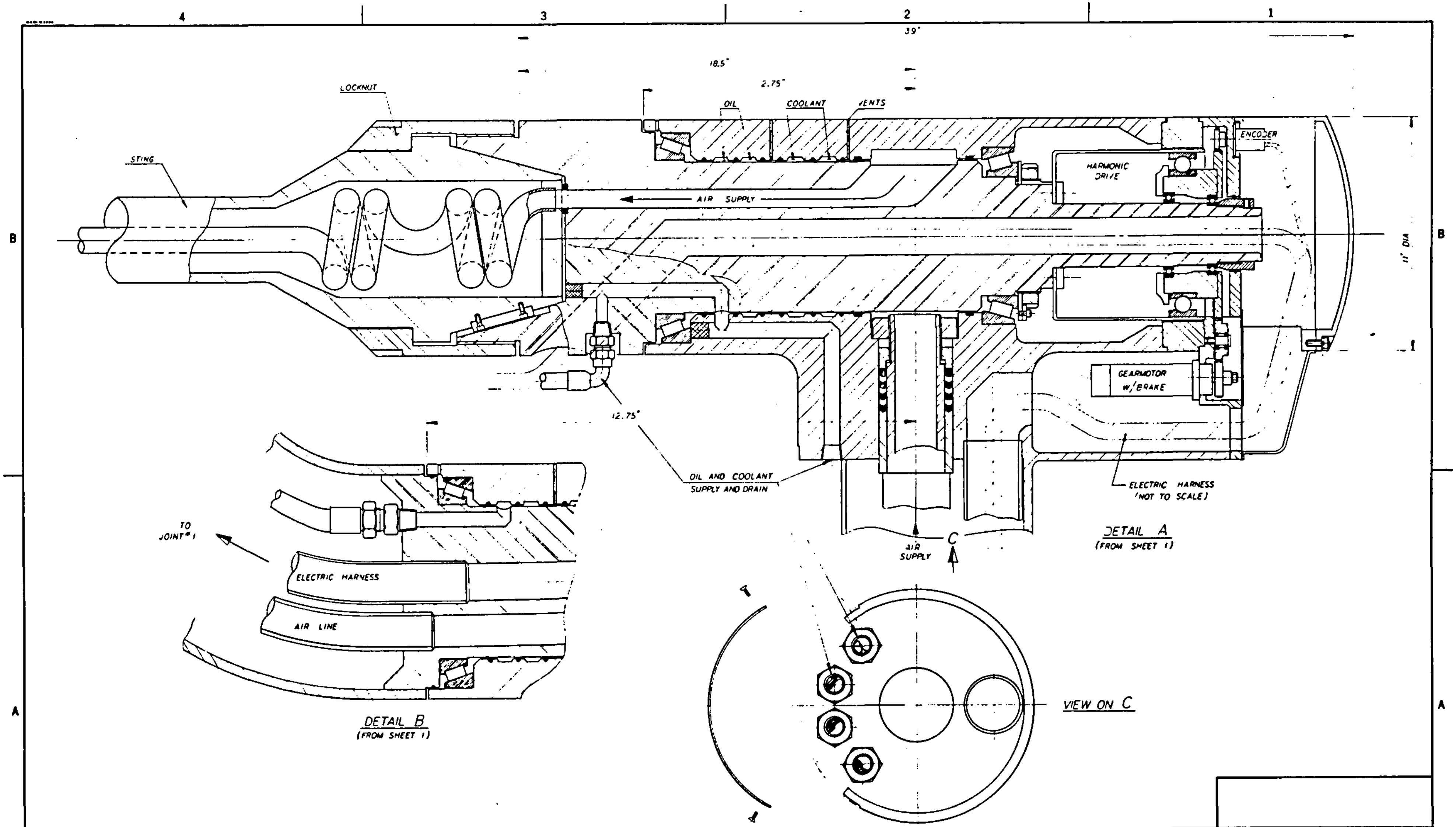
DRAWING TITLE ROTARY STING ARRANGEMENT

DR. B. JACK

SHEET 1 OF 2

LD-544308

SHEET 1 OF 3 1 DSMA Nº D4054-007



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MATERIAL

SCALE 1/2

DR B. JACK

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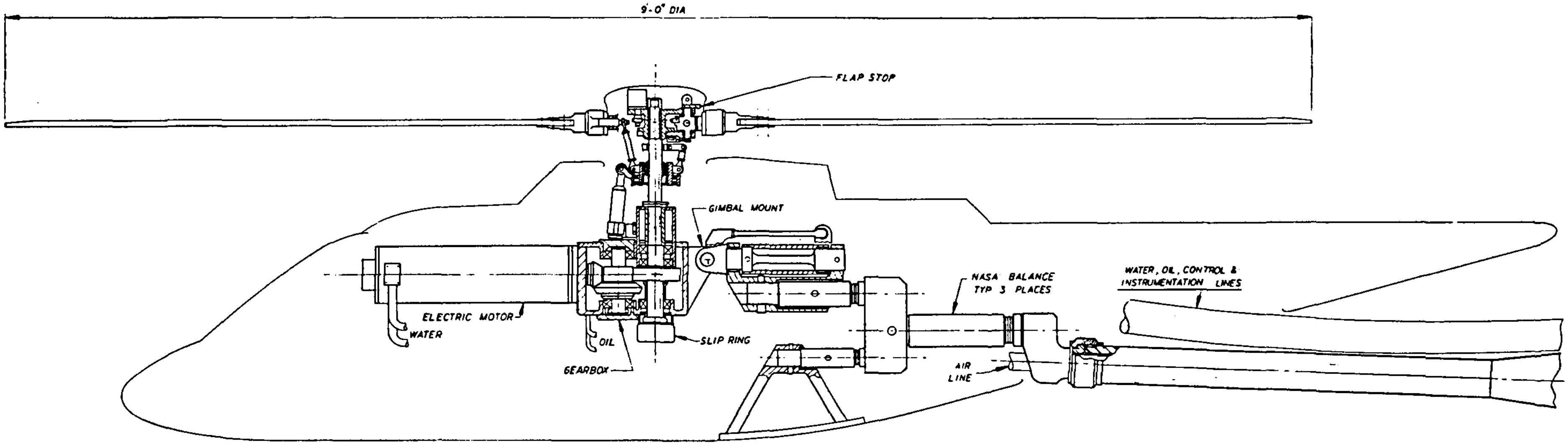
PROJECT TITLE 4 x 7M TUNNEL STUDY

DRAWING TITLE Nº 1 ROTARY JOINT

SHEET NO. 2 of 2

LD-544308
DRAWING NO.

SHT 3 OF 3 1 DSMA D 4058 007



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SCALE 3" = 1'-0"

PROJECT TITLE 4x7 M TUNNEL STUDY

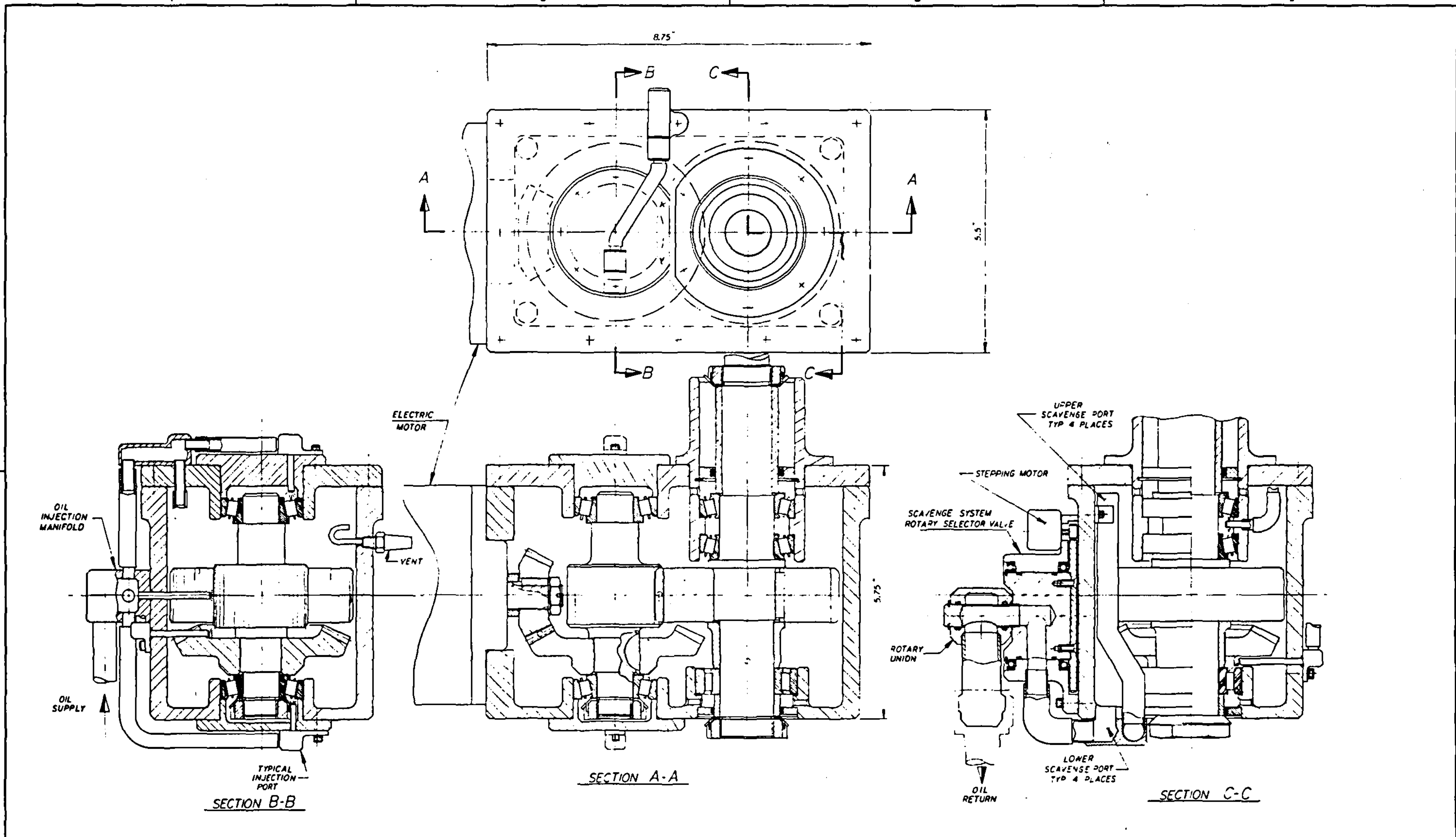
DRAWING TITLE ROTOR DRIVE ASS'Y CONCEPT

DR. R.A. BURGESS

SHEET 1 OF 2

LD-544309

SHEET 1 OF 2 1 DSMA Nº D4054-008



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PROJECT TITLE: 4X7M TUNNEL STUDY
DRAWING TITLE: ROTOR DRIVE GEARBOX CONCEPT
DR: H.A. BURGESS
SHEET 2 OF 2
LD-544309

SHEET OF 1 DSMA NS D4054-006

77

Standard Bibliographic Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA CR-178079		

4. Title and Subtitle	5. Report Date
Modifications to the 4x7 Meter Tunnel for Acoustic Research--Engineering Feasibility Study	March 1986
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No. 4054/R128
	10. Work Unit No.
9. Performing Organization Name and Address: DSMA Engineering Corporation, Orlando, Florida	11. Contract or Grant No. NAS1-17892
	13. Type of Report and Period Covered: Contractor Report
12. Sponsoring Agency Name and Address: National Aeronautics and Space Administration, Washington, DC	14. Sponsoring Agency Code: 534-03-12-02

15. Supplementary Notes

Final Report
Langley Technical Monitor: S. R. Barringer

16. Abstract

The NASA-Langley Research Center 4 x 7 Meter Low Speed Wind Tunnel is currently being used for low speed aerodynamics, V/STOL aerodynamics and, to a limited extent, rotorcraft noise research. The deficiencies of this wind tunnel for both aerodynamics and aeroacoustics research have been recognized for some time. Within the FY 1984 NASA Construction of Facilities (CofF) Program, modifications to the wind tunnel are being made to improve the test section flow quality and to update the model cart systems.

A further modification of the 4 x 7 Meter Wind Tunnel to permit rotorcraft model acoustics research has been proposed for the FY 1989 CofF program. As a precursor to the design of the proposed modifications, NASA have conducted both in-house and contracted studies to define the acoustic environment within the wind tunnel and to provide recommendations for the reduction of the wind tunnel background noise to a level acceptable to acoustics researchers. One of these studies by an acoustics consultant, Bolt, Beranek and Newman Inc. (BBN), has produced the primary reference documents (Refs. 1 and 3) that define the wind tunnel noise sources and outline recommended solutions.

As wind tunnel design consultatns, DSMA Engineering Corporation has been retained to conduct a conceptual design and feasibility study for the practical application of the modifications recommended in Refs. 1 and 3. This report covers the results of this study.

17. Key Words (Suggested by Authors(s))	18. Distribution Statement
wind tunnel design wind tunnel fans acoustic treatment	Unclassified - Unlimited Subject Category 71

19. Security Classif.(of this report)	20. Security Classif.(of this page)	21. No. of Pages	22. Price
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