

N83 24889

D8

5  
Y

CONTROL OF LARGE THERMAL DISTORTIONS  
IN A CRYOGENIC WIND TUNNEL

John C. Gustafson\*

ABSTRACT

The National Transonic Facility (NTF) is a recently completed research wind tunnel capable of operation at temperatures down to 89K (160°R) and pressures up to  $9 \times 10^5$  Pa (9 atmospheres) to achieve Reynolds numbers approaching 120,000,000. Wide temperature excursions combined with the precise alignment requirements of the tunnel aerodynamic surfaces imposed unique constraints on the mechanisms supporting the internal structures of the tunnel. The material selections suitable for this application were also limited. A general design philosophy of utilizing a single fixed point for each linear degree of freedom and guiding the expansion as required was adopted. These support systems allow thermal expansion to take place in a manner that minimizes the development of thermally induced stresses while maintaining structural alignment and resisting high aerodynamic loads.

Typical of the support mechanisms are the preload brackets used in the fan shroud system and the Watts linkage used to support the upstream nacelle. The design of these mechanisms along with the basic design requirements and the constraints imposed by the tunnel system are discussed in detail.

INTRODUCTION

The National Transonic Facility (NTF) is a recently completed cryogenic and high pressure wind tunnel at the NASA Langley Research Center capable of operating at temperatures in the range of 89K (160°R) and at pressures to  $9 \times 10^5$  Pa (9 atmospheres). This facility is capable of producing Reynolds numbers in the range of 120,000,000 by using cryogenic nitrogen as the test fluid and will provide a marked increase in capability to accurately simulate the full scale aerodynamics of current and future aircraft with small scale models. The tunnel is a recirculating type with a number of internal structures to control and condition the flow through the test section. The internal structures experience wide temperature swings since they are exposed to the cryogenic nitrogen. The tunnel pressure shell is internally insulated and is only exposed to the outside ambient temperature. The design of the mechanisms to support these internal structures from the tunnel pressure shell and still accommodate the thermal and pressure induced relative displacements between the shell and the internals was one of the critical problems to be solved in the facility design.

\*NASA Langley Research Center, Hampton, Virginia

120

This paper discusses the problems associated with the structural design, resulting from thermal expansion, for the entire facility and the general design approach used to deal with it. As typical examples, the design of the supports for two of the major internal structures, the fan shroud system and the upstream nacelle, are examined in detail. The material selection criteria and the constraints imposed by operational considerations are included in the discussion. The final solutions and the major problems experienced in implementing these designs are reviewed in detail.

## BACKGROUND

The primary requirement for dynamic simulation of an aircraft at transonic speeds, by a model in a wind tunnel, is the matching of the Reynolds and Mach numbers. While obtaining the required Mach number has readily been achievable in existing facilities, generating full scale Reynolds numbers on scale models has been a considerably more difficult task.

As a result, over the past three decades existing wind tunnels have been unable to accurately simulate the characteristics of the progressively higher performance aircraft that were being developed, Figure 1.

As early as 1920 it was recognized that the Reynolds number in a wind tunnel could be controlled by varying the temperature of the working fluid.<sup>(1)</sup> Density, viscosity and the speed of sound are all temperature dependent and tend to increase the Reynolds number as the temperature decreases, the effect being most pronounced below 180K (324°R) as shown in Figure 2.

Currently the NASA Langley Research Center is in the process of activating the National Transonic Facility which is designed to achieve full scale Reynolds numbers in the transonic regime and provide a more accurate flight simulation in the wind tunnel testing of the next generation of aircraft. (See Figure 3) The tunnel is a recirculating type operating over a temperature range of 89K (160°R) to 339K (610°R).<sup>(2)</sup> Cryogenic nitrogen is used as a working fluid for the low temperature regime while dry air is used for the ambient case. The general design goals are given in Table 1 with the overall size and configuration shown in Figure 4. To minimize the energy consumption associated with cooling the tunnel system to the cryogenic state and to permit more rapid changes in the working gas temperature, the tunnel insulation system is on the internal surface of the pressure shell, Figure 5. This causes the tunnel pressure shell to remain at near ambient temperature while the temperature of the internal structures, which are supported by the pressure shell, are near the flow stream temperature.

### Accommodating Thermal Expansions

The design of large structures such as NTF must, in general, incorporate some provision for thermal expansion between supports. For

instance it is necessary to accommodate the seasonal and daily temperature induced dimensional changes or stresses. While it appears that large dimensions are the root of the problem, combining the equations defining the modulus of elasticity and the expansion of a material under temperature show otherwise. As an example, the tensile stress generated by changing the temperature of a bar fixed at both ends can be defined by

$$\sigma = E \alpha \Delta T$$

where  $\sigma$  = tensile stress  
E = modulus of elasticity  
 $\alpha$  = linear coefficient of thermal expansion  
 $\Delta T$  = change in temperature from the stress-free state

It follows that for parts of any size, manufactured from mild steel or annealed aluminum alloys, a temperature shift of 50K (90°R) will generate stresses on the order of  $6.89 \times 10^7$  N/m<sup>2</sup> (10,000 psi) or about half of the design allowable stresses. Logically then, it is most efficient to allow the thermal expansion to take place, more or less, unrestrained.

The approach used in the NTF design is the classic one of providing a single fixed point of attachment along a given axis and letting the structure deform as freely as possible away from the anchor point. Typical of this approach is the single point anchoring of the pressure shell at the transfer case, Figure 4, and the use of pivoting supports of the type shown in Figure 6 at other ground anchors.

The internal structures are supported from the inner pressure shell wall. Attachment points between the various internals and the supporting shell are located where a functional requirement dictates a fixed relationship between the two structures. Thermal growth away from these attachment points is guided to maintain critical alignments between the internal components and pressure shell structure. This insures that aerodynamic flow quality is maintained regardless of the operating temperature.

### MATERIALS

Material selection on the NTF project was severely limited. The basic criteria utilized for material screening was a Charpy notch toughness value in excess of 34 Joules (25 foot pounds). The material also had to meet the requirement imposed by the ASME Pressure Vessel Code.

Table 2 is a list of the primary structural materials chosen for use in the NTF. The use of two types of 5000 series aluminum alloys was dictated by an upper service temperature limit of 339K (610°R) for the 5083 alloy. The use of 5083 was desirable because of its superior strength properties in the annealed condition. However, temperatures from the fan to the cooling coils will rise to 366K (659°R) during ambient temperature tunnel runs necessitating the use of 5454 for aluminum

structures in this section of the tunnel. While 5454 has a lower allowable design stress, it is not susceptible to stress corrosion cracking at temperatures above 339K (610°R).

Despite the internal insulation the shell is required to withstand exposure to the cryogenic environment, including possibly a direct contact with liquid nitrogen. Additional considerations of fabrication ease and the requirement for a 50 year fatigue life resulted in the choice of 304 stainless steel for the shell material.

Aluminum was selected over steel for the primary internal structures to reduce weight, increase thermal conductivity and minimize thermal capacity. In order to maintain the thermal stresses within allowable limits, the system cool down time had to be controlled. Even with the optimum material selections, this time could not be reduced to less than approximately 8 hours.

### INTERNAL STRUCTURE TO SHELL ATTACHMENTS

As noted earlier, for operational cost considerations, the tunnel pressure steel shell is insulated from the flow stream and the internal structures stabilize at the flow stream temperature. In operation this results in a temperature differential between the shell and internals of apparently 205K (370°R) and can result in large relative motions between initially adjacent points on the two structures.

In general, all connections between the internal structural elements and the shell provide fixity in only one local linear direction. Moments on the gross structures are restrained by couples generated by opposing supports on either side of the structure. Local rotations at the supports will occur due to uneven temperature distributions as the system transitions from ambient to its initial operating setpoint. These rotations are simply allowed to occur since they will essentially disappear as thermal equilibrium is reached prior to system start up.

This approach does not directly address the problem of stresses internal to the structure caused by non-uniform temperature profiles during thermal transitions. These are controlled operationally by a five hour structural conditioning cooldown to approximately 117K (210°R). Operational temperature variations of the flowstream are then restricted to a change in temperature of 48K (89°R). Since these operational swings cannot be predicted, the design calculations assume that an instantaneous change in the gas temperature equal to the full operational temperature swing can occur at anytime within the limits.<sup>(3)</sup> (See Figure 7)

This design approach of letting the structure deform freely reduces both the complexity of the thermal stress analysis and the magnitude of the tunnel stresses. The transient cases of cooldown and the 48K (86°R) operational temperature swing became the dominant concern in this area.

Determination of the final stress levels was done by superimposing the mechanically generated stress distribution over the thermal transient stress distributions.

### FAN CONTAINMENT SHROUD ASSEMBLY

The fan containment shroud shown in Figure 8 is one of three sections of a sixty foot shroud assembly located in the fan region of the tunnel. This particular section forms the outer surface of the flow annulus, provides the outboard attachments for the fan inlet and exit flow control vanes and, as the name implies, provides protection for the tunnel pressure vessel in the event of separation of a fan blade from the drive disc. Table 3 summarizes the pertinent dimensions, loads, and design requirements of this structure.

The shroud is supported from the tunnel wall which is reinforced for the high loads in this region by external rings. To deal with the combined thrust and torsional operating loads on the shrouds and concurrently allow thermal expansion of the shroud assembly, independant radial and axial support systems are utilized. Each system provides local support along one linear axis and allows relative freedom of motion in the other two orthogonal axes.

#### Thrust Support

In operation the absolute pressure inside the shroud assembly increases from the upstream to downstream end. To prevent recirculation between the shroud and the shell wall this flow path was sealed off. It is this seal requirement which generates the majority of the force on the shroud and results in a net force tending to drive the shroud upstream. A single bulkhead seal is used on the upstream end and this provides the single fixed axial support point for the shroud, Figure 9.

To insure the alignment during non-operating conditions a set of four preloaded brackets are placed around the circumference of the shell. Each bracket provides an initial force of 89 kN (20,000 lbs) against the seal, and the shallow angle of the line of action precludes any appreciable change in loads due to a radial displacement of the shroud relative to the shell.

#### Radial Support

In the cold condition the clearance between the rotating blades and the shroud is designed to be less than 1.8 mm (.070 in). This requirement dictated a support design that:

1. Provides for radial adjustment of the structure position at installation in extremely small increments.

2. Is sufficiently rigid to maintain alignment under operational loads.
3. Allows radial thermal expansion but maintains the alignment of the shroud centerline to the blade rotational axis as the expansion takes place.

The rotational and dead loads from the shroud assembly are transferred by three sets of radial keys on the shroud into matching preload brackets bolted to pads fabricated on the pressure shell. Each set is composed of four assemblies, Figure 10, located symmetrically around the circumference of the shroud. The two brackets on the horizontal react only vertical forces and the two on the vertical react only horizontal forces.

The keys on the shroud are 11.43 cm (4.5 in) thick aluminum plates fitted between adjacent reinforcing rings on the outside of the shroud assembly. The plates are fitted into 2.54cm (1 in) deep slots machined into the ring webs and maintained in position by through bolts.

The brackets, Figure 11, are a welded structure fabricated from 9 percent nickel steel. The loads are transmitted into the shell primarily through a 15.2 cm (6 in) diameter shear pin. The configuration is unsymmetrical with the heavier side positioned to resist the dead loads of the shroud assemblies. Vertical and horizontal alignment of the shroud is provided by opposing A286 stainless steel screws threaded into the bracket arms. The screws react into the key through self aligning pucks. The motion of the pucks over the key surface is restrained only by sliding friction and allows the radial and axial deformation of the shroud to occur relatively unimpeded. To minimize the sliding friction and provide a semblance of lubrication the pucks have a reinforced fluorocarbon pad bonded to the sliding surface.

The bracket itself is exposed to the gas temperature of the tunnel. To minimize local cold spots on the shell, the bracket is separated from the shell pad by machined fiberglass reinforced polyester spacers. The spacers are selectively machined to correct for the relatively coarse tolerances on the "as welded" shell pad and are used as an accurate reference surface for the bracket load bearing assemblies. The shear pin is also thermally isolated from the shell by being totally encased in a load bearing fiberglass envelope.

The design, while straightforward in appearance, proved to be more difficult to implement than expected. The alignment of the shroud to the centerline of the rotating fan, as noted earlier, is critical because of the limited clearance to the blade tips. Once aligned, the support system must maintain this alignment over all of the thermal excursions and operating loads. These criteria were met by preloading the bracket arms to a level in excess of in-service loads. This insures that the entire bracket is fully involved and the effective stiffness of each bracket is maintained at a required value in excess of  $1.4 \times 10^8$  N/m ( $8 \times 10^5$  lb/in).

Initial closed form analysis indicated that the bracket stiffness would be on the order of  $3.5 \times 10^9$  N/m ( $20 \times 10^6$  lb/in). The results were highly questionable since the geometry of the bracket does not meet the length to depth ratios assumed in the standard beam equation. A finite element analysis was also made which predicted a much lower stiffness primarily due to flexing of the bottom plate. The difficulty here was correctly modeling to account for the contribution of the mounting bolts, the shear pin, fiberglass spacers and the shell mounting pad to the overall stiffness.

Preload relief during cooldown resulting from the differential expansion between the aluminum key and the steel bracket was also a concern. This imposed a minimum deflection of .25 mm (.01 in) between the two bracket legs before any preload was applied. Depending on the bracket stiffness, the total preload could increase to the point that, under certain loading conditions, stresses in the weld root could approach design allowables.

This prompted a change in weld filler alloy from AWS specification ENiCrFe-3 (normally used for 9 percent nickel on NTF) to specification ENiCrMo-3 which has an ultimate strength in excess of the base material. A rigorous program of NDT weld inspection and post weld machining of the fillets to remove potential stress risers was also imposed.

Finally a test program was initiated on the tunnel site to determine the actual load, deflection, and root area strain relationship in the bracket, Figure 12. The results showed an effective spring constant of  $1.2 \times 10^9$  N/m ( $7 \times 10^6$  lb/in), which was adequate but considerably lower than expected. A large part of the deflection occurred in the base as predicted in the finite element analysis.

The combined preload (thermal and operational load) was set at  $3.3 \times 10^3$  N (73,000 lb) and the strain measured at the weld root for this preload. Strain gages were then mounted at the same relative location on the remainder of the brackets and the strain measurement duplicated to set the preload as the final installation step.

#### UPSTREAM NACELLE

The upstream nacelle, Figure 13, forms the inner aerodynamic surface of the flow annulus. It houses the main radial bearing for the fan shaft, the bearing lubrication supply lines, the inlet guide vane actuator and its hydraulic system. To avoid cryogenic temperatures on the lubrication and hydraulic systems, the internal volume is insulated and maintained at 292 K (525°R) or above.

The upstream nacelle is an L shaped tubular shell structure beginning as a conical shape just downstream of the nitrogen injection assembly, Figure 14, and terminating at the aerodynamic fairings on the fan blades.

The general dimensions and loads are given in Table 4. Here also, the nacelle loads are resisted by independent support systems each providing a single fixed point for the three linear axes.

### Thrust Loads

The thrust loads are the main aerodynamic forces and are reacted in two places. Thrust in the X direction, Figure 4, is towards the fan in the fan leg and is reacted at the transfer case structure by pivoting brackets which allow radial motion. Thrust in the Y direction is removed at the corner by an "Invar" thrust link reacted by the tunnel pressure shell, Figure 13.

### Radial Loads

Radial support is provided in three locations. At the transfer case a set of brackets similar in concept to that discussed for the shroud are used. Additional vertical supports were needed near the corner and near the nose to eliminate the overturning moment at the transfer case. The supports must cross the flow stream in this area that requires a design having a low flow blockage and an aerodynamic shape to minimize flow disturbance. In addition the supports must provide sufficient rigidity to raise the nacelle first mode frequency above 15 hz and allow independent thermal expansion of the shell and nacelle.

The first two requirements are most efficiently met by tension members between the shell and the nacelle but rigid attachment would not meet the requirement for free thermal expansion. The solution was to use a Watts linkage for a support as shown schematically in Figure 15. At the corner the mechanism is built into the turning vanes. In the upstream section of the nacelle the legs have leading and trailing edge fairings attached to reduce turbulence. Only vertical support is provided since the nacelle structure in this area is sufficiently rigid to react against any horizontal bending moments.

The Watts linkage is attached to the nacelle by a pivot through a support beam at the nacelle centerline. This allows the nacelle to freely expand and contract radially. Seals are incorporated around the vertical members where necessary to prevent intrusion of the cold flow stream into the heated environment inside the nacelle. With the pivot at the center, the tunnel shell is allowed to expand/contract independent of the nacelle with no effect on the alignment of the nacelle to the centerline. Maintenance of alignment is predicated on a uniform radial change of the tunnel shell. This condition is reasonably well met by having this section of the tunnel enclosed in a high bay equipment area. Horizontal motions of the nacelle relative to the tunnel wall are accommodated by the use of a spherical self aligning bearing at each end of the vertical leg attachments, Figure 16.



Material selection is based solely on strength and stiffness since the radial freedom of the system will obviously accommodate changes in leg length. The entire Watts linkage structure is 304 stainless steel except for the pivot pins which are A286 stainless steel. The aerodynamic fairings are 6061 aluminum attached to the legs with slip fittings to accommodate the thermal expansion.

During installation a slight preload is built in to take up fabrication looseness in the various pivot points.

#### Concluding Remarks

Support systems utilizing preloaded brackets and Watts linkage mechanisms have been applied to a large highly loaded structure operating over a wide temperature range. These systems have been configured to allow relatively unimpeded thermal expansions and still meet close alignment requirements over the operational temperature and load range.

### References

1. Kilgore, Robert A.: Evolution of the Cryogenic Wind Tunnel and Experience With Langley 0.3 Meter Transonic Cryogenic Tunnel. Paper #1, Cryogenic Technology Conference, NASA Langley Research Center, November 1979.
2. Howell, Robert R.: Overview of Engineering Design and Operating Capabilities of the National Transonic Facility. Paper #2, Cryogenic Technology Conference, NASA Langley Research Center, November 1979.
3. Ramsey, James W. Jr.: Design for Thermal Stress. 6th Paper, Cryogenic Technology Conference, NASA Langley Research Center, November 1979.

Table 1. General Design Goals

Maximum Reynolds Number	120,000,000 at Mach 1
Mach Number Range	.1 to 1.2
Operating Pressure Range	$5.72 \times 10^4$ to $8.27 \times 10^5$ Pa (8.3 to 130 psia)
Operating Temperature Range	78°K to 339°K (140°R to 610°R)
Input Power	$9.7 \times 10^7$ Watts (130,000 Hp)
Operating Medium	Nitrogen
Test Section Size	2.5 m x 2.5 m (8.2 ft. x 8.2 ft)

Table 2. Primary Structural Materials

<u>Ferrous</u>	<u>Non-Ferrous</u>
304 Stainless Steel	5083-0 Aluminum
A286 Stainless Steel*	5454-0 Aluminum
9% Nickel Steel	2024-T4 Aluminum*
Invar 36	6061-T6 Aluminum*

\*Primarily used as fastening materials

Table 3. Fan Containment Shroud Dimensions and Loads

	<u>SI Units</u>	<u>English Units</u>
Length	4.44 m	14.56 ft
Inside Diameter	6.0 m	19.70 ft
Wall Thickness	5.09 cm	2.0 in
Material	5454 aluminum	
Weight	196000 N	44,000 lb
Diameter Change Over Temp	2.64 cm	1.04 in
Minimum Fan Blade Clearance	1.8 mm	.07 in
Maximum Thrust Load	$1.1 \times 10^6$ N	246,420 lb
Maximum Torque about axis	$4.26 \times 10^6$ Nm	$3.13 \times 10^6$ ft lb
Maximum Differential Pressure	$1.52 \times 10^5$ Pa	22 psia

Table 4. Upstream Nacelle Dimensions and Loads

Overall Length Along Centerline	13.7 m	45 ft
Maximum Diameter	4.63 m	15.2 ft
Dead Load On System	$2.5 \times 10^5$ N	55,240 lb
Thrust Load Z Direction	$1.08 \times 10^6$ N	242,650 lb
Thrust Load X Direction	$1.56 \times 10^6$ N	350,600 lb
Thermal Motion Z Axis	3.47 cm	1.37 in
Thermal Motion X Axis	3.88 cm	1.53 in
Moments X Axis	$2.79 \times 10^3$ Nm	$2.05 \times 10^3$ ft lb
Y Axis	$5.22 \times 10^5$ Nm	$3.84 \times 10^5$ ft lb
Z Axis	$5.36 \times 10^4$ Nm	$3.94 \times 10^4$ ft lb
Material	5083-0 Aluminum	

ORIGINAL PAGE IS  
OF POOR QUALITY

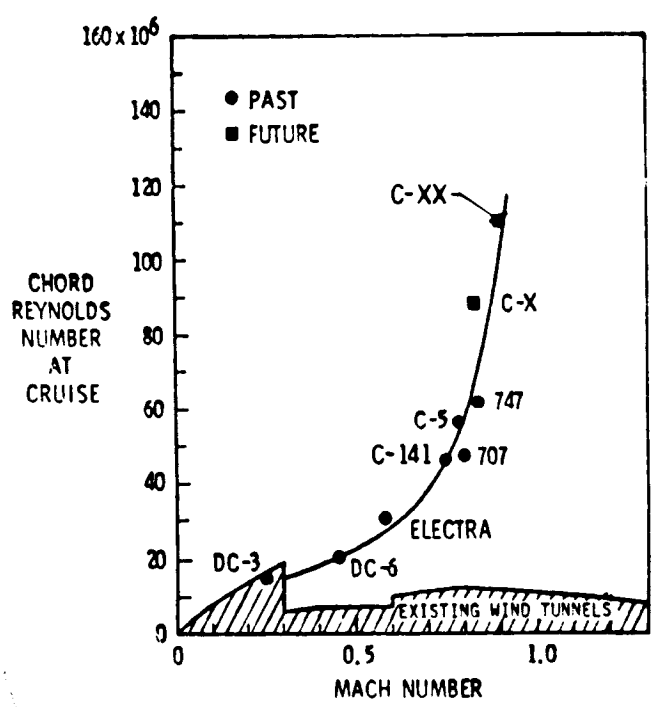


Figure 1.- Flight Reynolds numbers for existing and projected aircraft compared with the capability of existing wind tunnels.

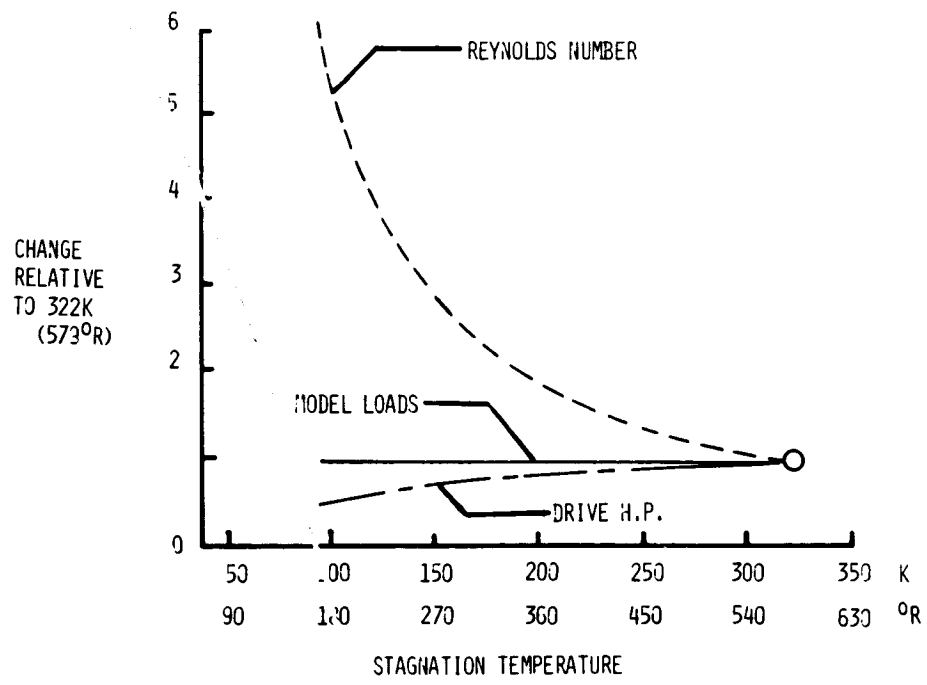


Figure 2.- Cryogenic tunnel concept

ORIGINAL PAGE IS  
OF POOR QUALITY

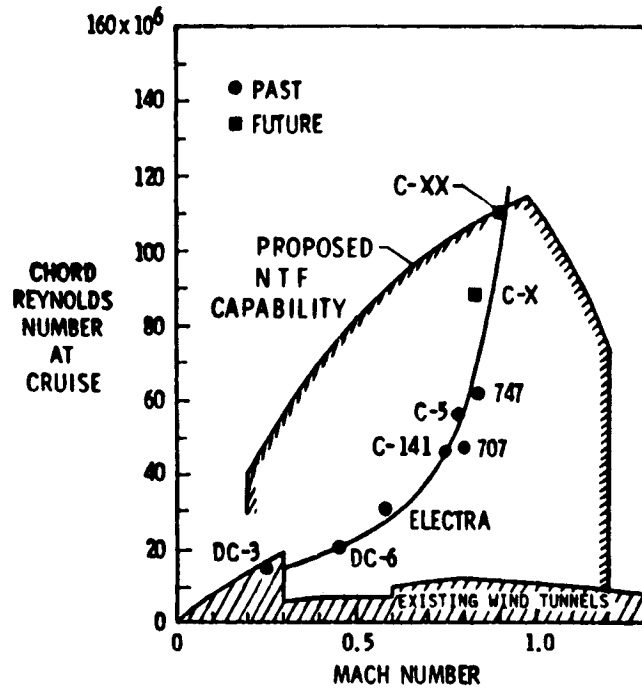


Figure 3.- Flight Reynolds numbers for existing and projected aircraft compared with the capability of the National Transonic Facility.

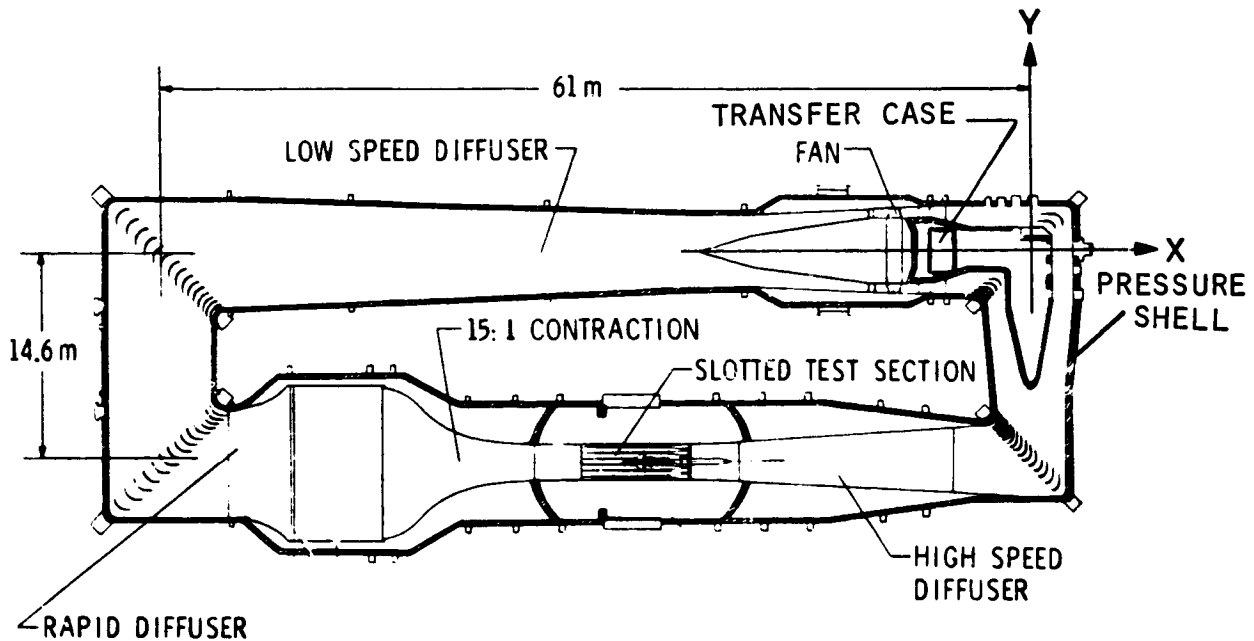


Figure 4.- Plan view of tunnel circuit showing pertinent dimensions and components - National Transonic Facility.

ORIGINAL PAGE IS  
OF POOR QUALITY

S  
Y

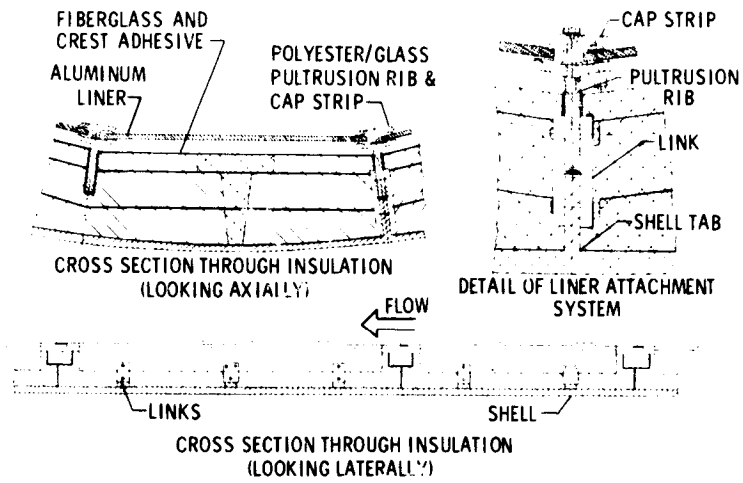


Figure 5.- Typical sections of the internal insulation system.

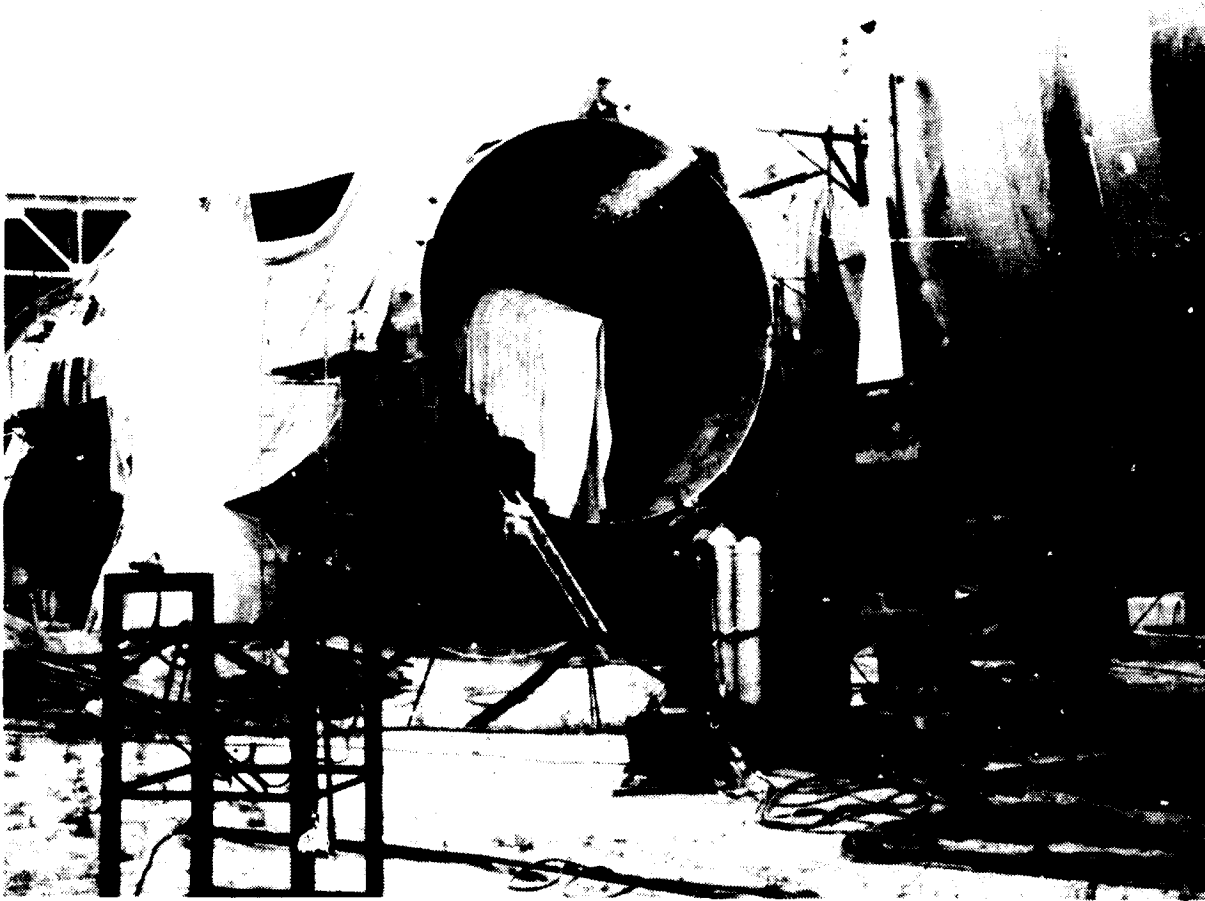


Figure 6.- Tunnel shell support legs.

ORIGINAL FILED  
OF PCR QUALITY

5  
Y

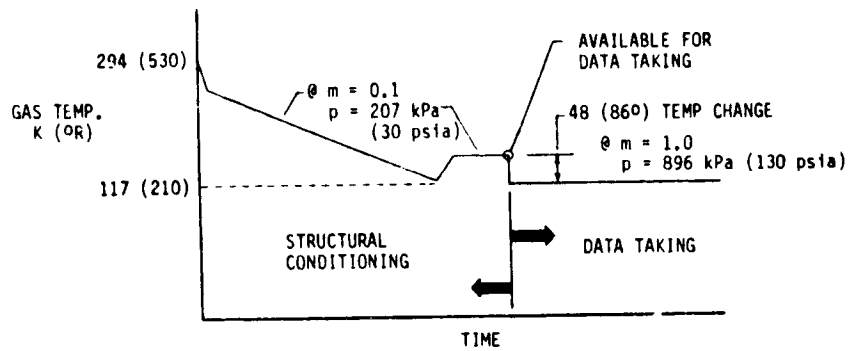


Figure 7.- Thermal forcing function cooldown and operating temperature change.

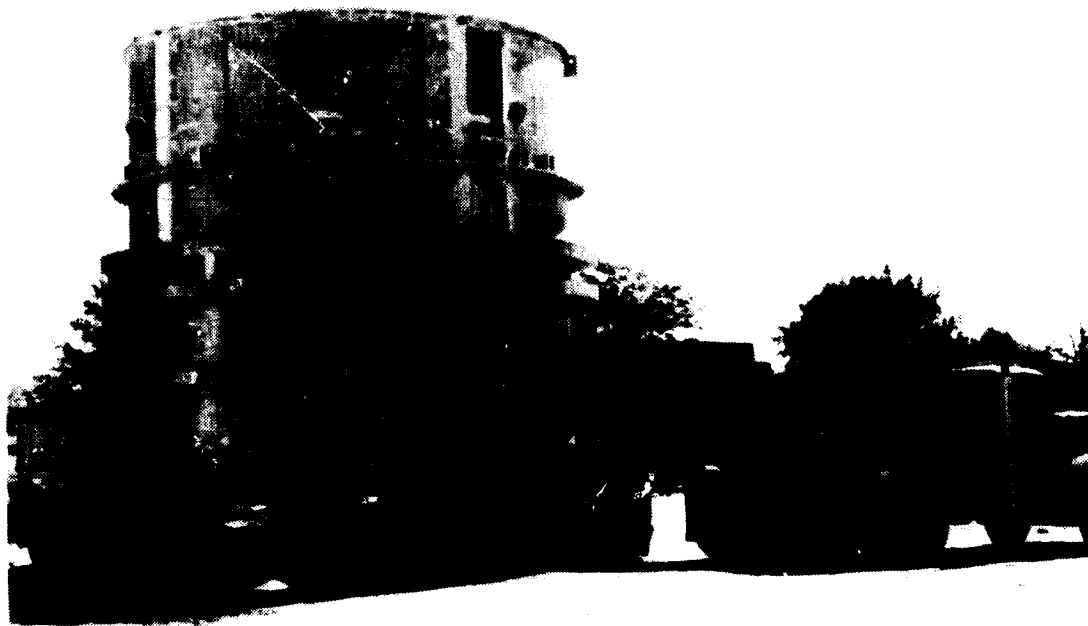


Figure 8.- Fan Containment Shroud



ORIGINAL PAGE IS  
OF POOR QUALITY

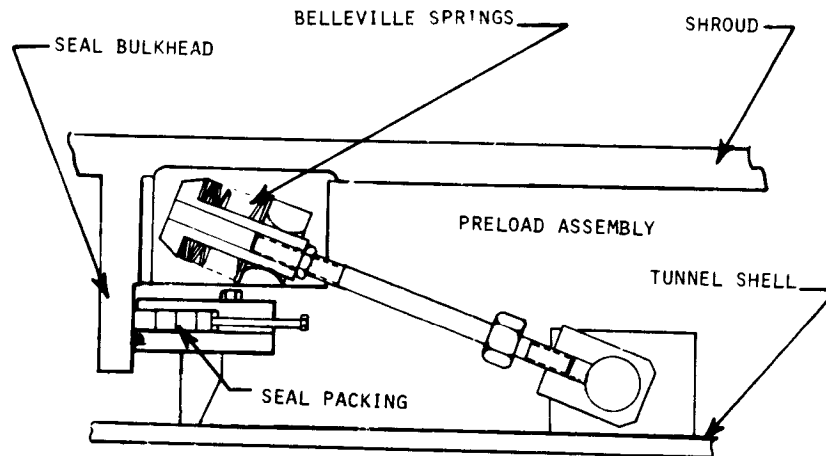


Figure 9.- Seal assembly and thrust brackets for the fan contamination shroud.

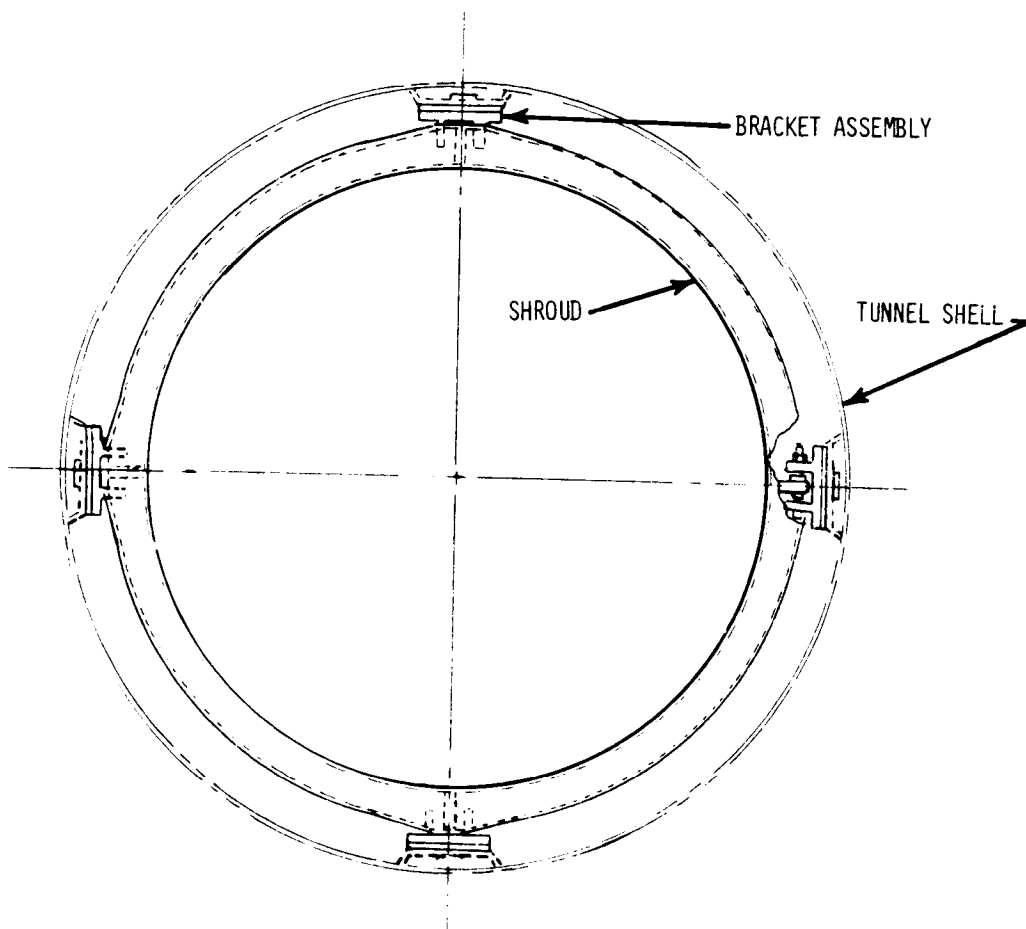


Figure 10.- A fan shroud radial support set.

ORIGINAL PAGE IS  
OF POOR QUALITY

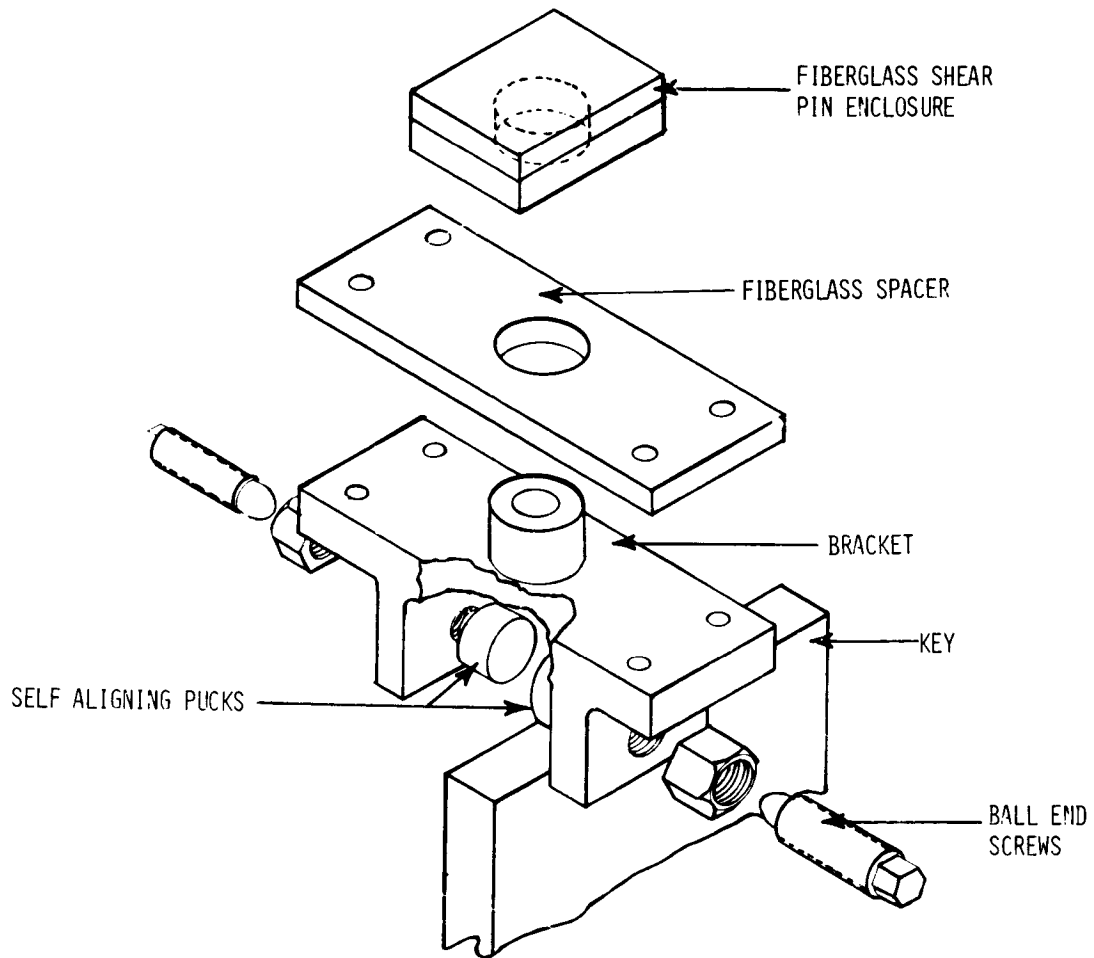


Figure 11.- A fan shroud bracket assembly.

ORIGINAL PAGE IS  
OF POOR QUALITY.

5  
Y

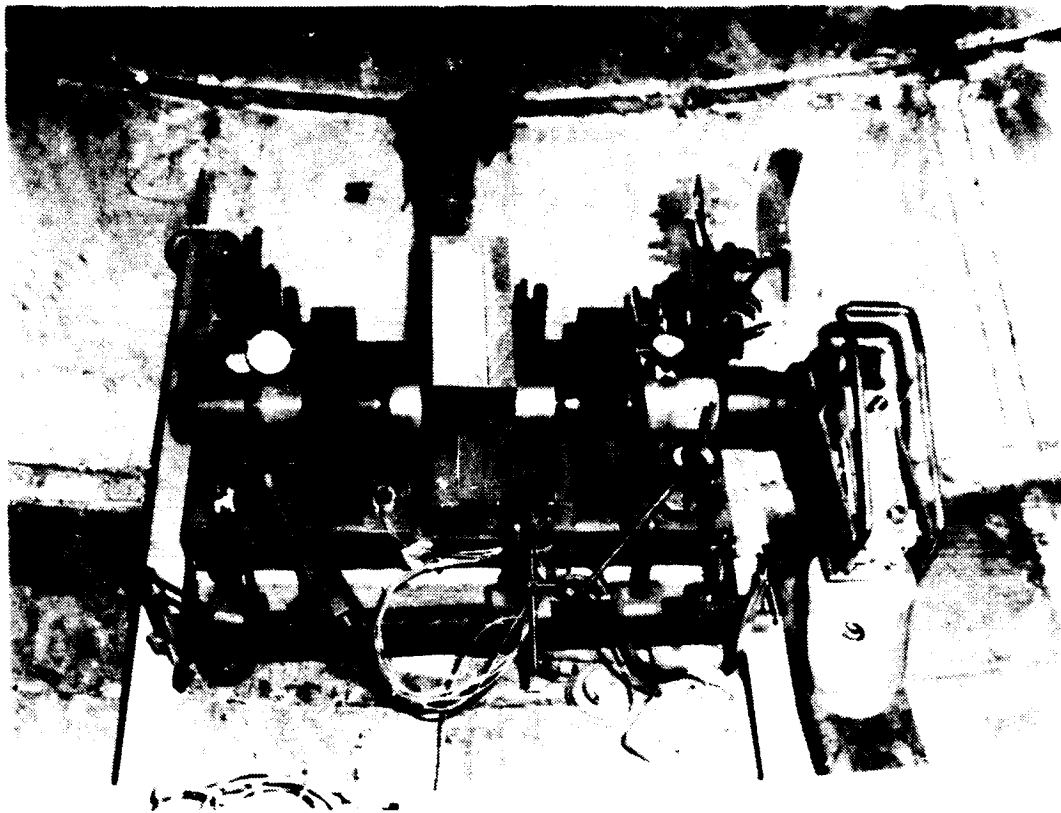


Figure 12.- Fan shroud bracket preload test setup.

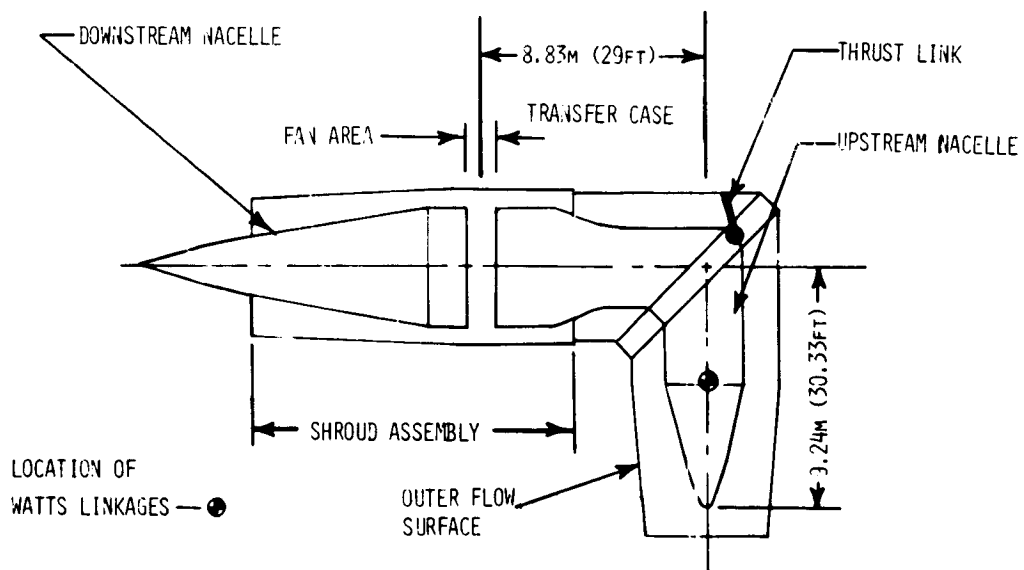


Figure 13.- Upstream Nacelle Configuration.

ORIGINAL PHOTO IS  
OF POOR QUALITY

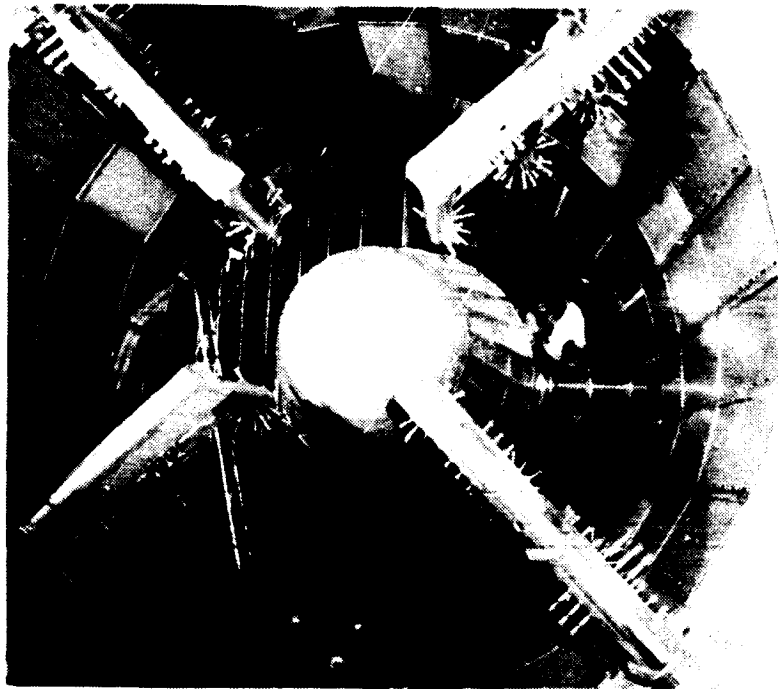


Figure 14.- The upstream nacelle viewed from the  $N_2$  injector assemblies.

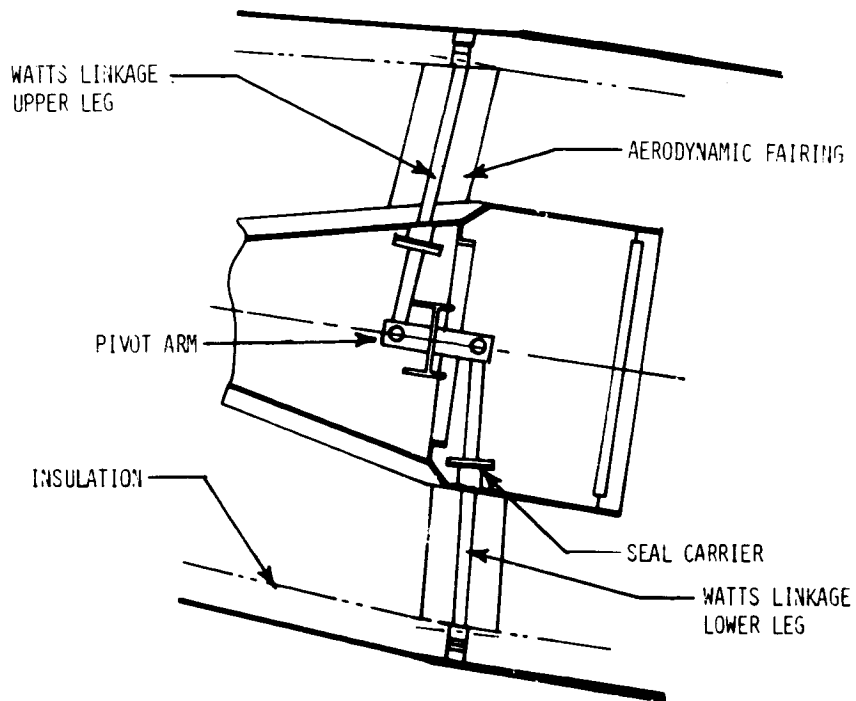


Figure 15.- Watts linkage schematic.

ORIGINAL FACE IS  
OF POOR QUALITY

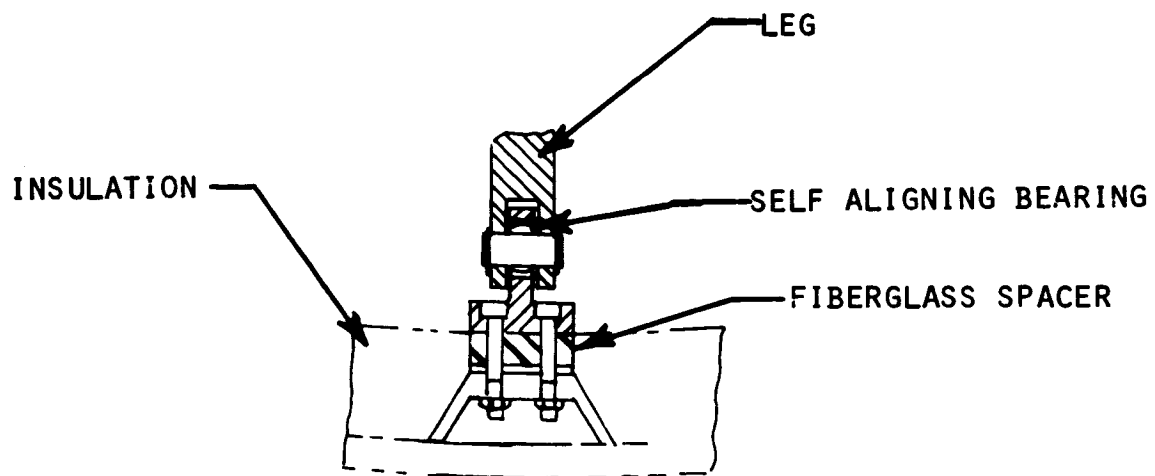


Figure 16.- The Watts linkage leg attachment  
to the tunnel shell.