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SPACE SHUTTLE WHEELS AND BRAKES

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ABSTRACT

The Space Shuttle Orbiter wheels have been subjected to a combination of tests which are different than any previously conducted in the aerospace industry. The major testing difference is the computer generated dynamic landing profiles used during the certification process which subjected the wheels and tires to simulated landing loading conditions.

The orbiter brakes use a unique combination of carbon composite linings and beryllium heat sink to minimize weight. The development of a new lining retention method was necessary in order to withstand the high temperature generated during the braking roll. As with many programs, the volume into which this hardware had to fit was established early in the program, with no provisions made for growth to offset the continuously increasing predicted orbiter landing weight.

Both wheels and brakes were designed, manufactured, and certified for Orbiter operational use by B. F. Goodrich Company, Troy, Ohio.

I. INTRODUCTION

The Space Shuttle, as with all space vehicles, requires a minimum weight configuration in order to maximize the payload weight to orbit lift capability. Using this philosophy, and drawing from the years of experience and techniques utilized in commercial and military aircraft applications, the wheel/tire/brake/shock strut configuration was sized on a predicted Space Shuttle Orbiter landing weight established very early in the program.

The main wheel baseline design has many of the same features as a commercial aircraft wheel. An interface with the axle, brake and tire, an over-inflation plug, thermal fuse release plug, inflation valve, bearing housings, grease seals, and pressure seals.

A "live axle" configuration was chosen for the nose gear which differs in design for most current aircraft but has been used in experimental and operational aircraft. With the live axle design, the wheels are splined to the axle and rotate as a unit on bearings mounted in the shock strut. The resulting corotating wheels reduces the tendency for wheel shimmy and, therefore, increases the stability of the assembly.

Preliminary studies by B. F. Goodrich were conducted to evaluate various combinations of materials for both the brake heat sink and friction surfaces in order to minimize the brake weight for both reusable and non-reusable applications. The combinations studied included full carbon composite heat sinks, conventional beryllium heat sinks with mechanically attached sintered metallic friction materials and the combination of carbon composite linings and beryllium heat sink. The carbon composite lining/beryllium heat sink was finally selected for its relatively low peak temperatures (compared to steel and carbon composites) which allowed the minimum weight reusable heat sink. The carbon composite heat sink was the lowest weight for a non-reusable application, requiring new wheels and tires to be installed after every stop. In comparing the levels of brake energies required for a normal landing when the brakes would be usable and a maximum brake energy stop where the brakes and wheels were expendable, the composite lining/beryllium heat sink was the most efficient for the very critical Space Shuttle weight requirements.

II. GENERAL DESCRIPTION OF THE ORBITER MAIN WHEEL

The Orbiter main wheel (See Figure 1) is similar to conventional aircraft wheels in most respects. It is made in two halves machined from 7049-T73 forgings and joined together by eighteen tie bolts and nuts of MP35 multi-phase 240 Ksi material.

The inboard wheel half has a 4340 steel sleeve pressed into the hub and supports the inboard bearing and grease seal. The wheel half, also contains the ports for the three thermal fuse plugs, over-inflation plug and inflation valve. Elevon brake drive lugs which interface with the brake rotors are also part of the inboard half and are protected from wear by 4130 steel hard chrome plated drive channels.

The outboard half contains the outboard bearing, grease seal and has mounting provisions for the hub cap.

The bearings are a standard Timken bearing size with special modifications and processing to help withstand the Orbiter landing speeds and loads.

Tire pressure sealing is accomplished by the compression of an O-ring in a seal cavity between the wheel halves. The O-ring material and seal geometry was established as the result of a separate NASA/B. F. Goodrich study contract. Extremely low leakage rate of the wheel/tire assembly was necessary because of the long "storage" time from gear retraction at vehicle mating to gear extension at landing. Access to the wheel wells for tire pressure check and reinflation during this time period was not possible.

III. GENERAL DESCRIPTION OF ORBITER NOSE WHEEL

The nose wheel (See Figure 2) differs from most conventional nose wheels in that it is mounted on a "live" axle with corotating wheels. That is, the axle rotates with the wheels using bearings mounted in the nose gear strut, thereby, eliminating the bearings from the wheel and replacing them with a spline/bushing configuration.

The nose wheel configuration is similar to the main wheel except for the lack of brake system interface. It is made in two halves of the same material, bolted together by attachments of the same material and has thermal fuse plugs, over-inflation plug and an inflation valve.

Thermal fuse plugs are used as blowout protection against nose wheel well overheat from plasma flow rather than brake heat as in the case of the main wheel. Pressure sealing is accomplished in the same manner as the main wheel.

IV. GENERAL DESCRIPTION OF THE ORBITER MAIN WHEEL BRAKE

The Orbiter brake (See Figure 3) is a four rotor, multiple disk brake using beryllium as the heat sink. The friction surfaces are carbon composite linings mechanically attached to the beryllium heat sink surfaces. The brake assembly consists of a pressure plate, back plate, rotors, stators, support torque tube and a hydraulic actuation piston housing. The piston housing has two separate hydraulic circuits for redundant actuation. The brake assembly is bolted to the landing gear shock strut with a bracket register to the axle for support of the outboard end of the brake assembly.

V. WHEEL DESIGN CHALLENGE TO OPERATIONAL CERTIFICATION

Modification to the wheel/tire testing philosophy, that is, the addition of the dynamic landing load profiles had little effect upon the original nose wheel design. The nose wheel loading is not impacted by crosswinds and only slight changes occur as the result of total vehicle weight change. This is because the nose gear is a semi-free castor design and can react a limited side load and the nose gear carries such a small percentage of the total vehicle static weight. In addition, the pitch over rate and resulting nose gear impact load or "slap down" has not changed. Therefore, the remaining portion of the text will address the challenges which had to be solved to certify the main wheel and brake.

As with many programs, requirements change over a period of time, and the Orbiter was certainly not different. Not only did the vehicle landing weight increase dramatically from the initial levels, but the methods of dynamic landing load profile simulation testing currently in use had never been performed on an aircraft wheel/tire assembly in the past. This advancement of the state-of-the-art caused several failures during the certification process.

The original or baseline wheel assembly design (See Figure 4) is similar to conventional aircraft wheels used throughout the aerospace industry. The baseline wheel was used during five Approach and Landing Tests (ALT) where the Orbiter was released at altitude from the 747 carrier aircraft to prove the flying and landing ability of the Orbiter. However, the Orbiter vehicle weight increased from approximately 150,000# for ALT to 227,000 pounds for an operational abort condition. The weight was subsequently increased again to the present 240,000 pound level. What are now called dynamic landing load profile tests were added to the baseline conventional MIL specification requirements at this time in the program. Because of the dramatic increase in loading conditions due to these changes, failures in the wheel bearings began to occur.

The dynamic load profiles (See Figure 5) are generated by computer programming that "lands" the Orbiter under given conditions of weight, velocity, aero surface configuration, c.g. location, tire sideload resistance capability, angle of attack, etc. Using these conditions, values for tire/wheel radial load, tire velocity, yaw angle and lateral load are obtained and used as inputs to the Wright Patterson Air Force Base (WPAFB) automated dynamometer.

In the baseline wheel configuration, the inboard bearing is located at approximately the tire centerline and is considerably larger than the outboard bearing. With the advent of the dynamic profiles the increased lateral load and corresponding increased moment on the wheel resulted in bearing failures.

At this point, a major redesign and development testing of the wheel was undertaken. The final wheel configuration (See Figure 1) which is still, as of this writing, being tested under straight roll, maintained the same interfaces between the wheel, brake and tire but the bearing configuration was drastically changed. By adding a steel sleeve pressed into the inboard wheel half, the distance between the bearings was increased and thereby equalized the load distribution on the bearings.

The use of the steel sleeve was required to carry the large cantilevered inboard bearings load and still fit within the inner diameter of the brake torque tube. Installation of the insert, however, provided to be a major design problem During initial tests, it was discovered that the insert rotated in the inner wheel half when installed with approximately 0.007 inch interference. The interference was increased in steps until it reached the presently used whopping 0.022 inch interference.

During the testing to solve the sleeve rotation problem, a parallel program of bearing configuration, grease and bearing axial preload test series was being conducted. The tests showed that by "manicuring" and adding a phosphate process to a standard tapered roller bearings and installing them with a high axial preload, dramatically increased the load capability. The high preload is contrary to instructions in automobile maintenance manual for front wheel bearing installation.

The preload is applied as a measure of wheel rolling resistance and requires approximately 1500 fott pounds torque on the axle nut.

The initial redesigned or "heavy duty" wheel, flown on STS-1, had a second or redundant O-ring added at the split line to help reduce pressure loss during mated and orbit operations (See Figure 6). The O-ring groove placement and configuration proved to be a stress riser causing a complete circumferencial crack in the outboard wheel half during the straight roll test.

The next configuration flown on STS-2, 3, 4, and 5 removed the second O-ring groove but failed in two areas at approximately 800 miles of straight roll. Cracks occurred in the inboard half originating from the fuse plug hole and in the outboard half in the tie bolt hole at the wheel split line.

In the third configuration, flown on STS-6, the fuse plug holes were repositioned to a reduced stress area and the outboard wheel half web thickness was increased and a larger radius added in the tie bolt hole at the split line. At the 500 mile point of straight roll for this configuration, non-destruction inspection revealed a crack in the steel sleeve at the outer surface of the bearing cup housing. The failure was traced to an inclusion in the 4340 steel material just below the outer diameter surface.

For the latest sleeve configuration, flown on STS-7, as of this writing is again in the straight roll test phase; the material specification was changed which limits the allowable size and number of inclusions. In addition, a shot peening process was added to the highly stressed bearing housing area.

VI. BRAKE DESIGN CHALLENGE TO OPERATIONAL USE

The main wheel brake design (See Figure 3) is an extension of the technology developed by B. F. Goodrich on other aircraft programs but with special emphasis on weight savings and performance. The beryllium heat sink is used on the military C-5A and F-14 aircraft. However, the combination of beryllium heat sink and mechanically attached carbon composite linings is used for the first time on the Orbiter.

As the program progressed, changes were made to the design requirements from a single stop and replacement to a multiple stop and refurbishment. Results of development tests on various combination of heat sink and friction lining materials as well as the design requirements changes led to the selection of beryllium as the heat sink and carbon composite for the linings. Beryllium was chosen for its light weight and efficient heat absorption capability and carbon composite was chosen because of its great wear resistant, light weight, high strength, and high temperature capability.

The major problems encountered in the development and certification testing were the method of mechanical attachment and physical configuration of the linings (See Figure 7). Several iterations of attachment, materials, and processing were tried before the final configuration was established. Included in the assembly are 1722 Rhodium plated steel for the "T" clips used to react the braking torque; Columbium rivets with molybdenum washers to attach the lining segment; and monel rivets to attach the "T" clips to the beryllium disks.

Design changes were also made in the attachment of the back plate to the torque tube and to the hydraulic piston seal configuration.

Certification testing included those normally associated with conventional aircraft brakes along with hydraulic burst and pressure cycle endurance tests. Because there was not a method to conduct the normal rejected takeoff test on the Orbiter, a simulated landing roll brake test was conducted using a dynamometer. The specification requirement stated that the brake assembly shall be capable of absorbing 36.5 million foot-pounds of energy in five separate "normal" stops and 55.5 million foot-pounds in one return to launch site (RTLS) stop, commonly referred to as an abort landing without failure. The certification test program verified these requirements could be met.

It must be pointed out at this time that the brake assembly certification document was dated August 1977. This was well over 3-1/2 years before the flight of STS-1. And because of the increased landing weights, changes in wheel and axle configuration, impact of crosswinds and other factors, certain brakes have been operated in the energy range which exceeds the reuse/refurbishment capability during the STS flights. For example, the brakes were designed for use on five landings with the maximum Orbiter landing weight of 188,000 pounds (32K pound payload) and refurbishable, followed by an emergency landing weight of 227,000 pounds (65K payload) with no reuse. Under these requirements all landings above 188,000 pounds were considered an emergency and, therefore, could result in the loss of the brakes. To date and for future planning all Orbiter landings have been above 188,000 pounds.

VIII. BRAKE PERFORMANCE DURING STS FLIGHTS

Brake pressure data obtained after the STS-1 landing indicated a considerably higher pressure was present at the right inboard brake than for the right outboard brake. This discrepancy was traced to a faulty connection in the brake skid control box. However, inspection of the right inboard brake showed that damage to the No. 3 rotor drive slot had occurred. The damage was determined to be lack of complete engagement between the wheel drive lug channels and the rotor drive slot face. Because of axle deflection relative to the brake assembly centerline, there is axial movement between the wheel lug and rotor slots as the wheel rotates. This axial in and out relative motion caused the end of the channel to displace the beryllium material at the rotor drive slot face outboard.

New longer channels were installed for STS-2 and post landing inspection revealed no further brake damage.

Initial inspection of brakes assemblies used on STS-3 and STS-4 and subsequent analysis revealed that the relative deflection between the wheel/brake/axle combination allow contact between rotating and stationary parts of the brakes and wheels. There was contact between the rotor I.D. and the torque tube lugs and between the stator O.D. and wheel lugs. Also, there was wear and displaced metal at some rotor drive slots. Subsequent inspection made by removing the carbon linings disclosed that beryllium carbide not previously visible, had formed in localized areas beneath the linings on both the rotors and stators (See Figure 8).

In this same time frame, the landing of STS-5 took place. During the last 60 feet of the landing roll, the left inboard tire skidded for 50 feet and then rolled the last 10 feet. When the brake assembly was disassembled, it was found that the No. 3 stator was broken into 5 segments.

An in-depth analysis of all the available data from STS-1 through -5 and the qualification test report established these conclusions.

1. Beryllium carbide formations are occurring in local "Hot spots" on rotors and stators of brake assemblies that are approaching or exceeding the reusability energy limit of 36.5 million foot-pounds. These formations occur when beryllium reaches its melting point of approximately 2400°F.

2. Heat generation is not distributed evenly across the lining surfaces in a radial direction, thereby causing local "hot spots" and results in carbide formations.

3. Carbide formation causes a bulge in the lining thus increasing the localized contact pressure and thereby self-perpetuates the temperature increase and carbide formation.

4. Heavy braking at high velocity and extended braking causes the "hot spots" to increase in size and spread to include the structural load path sections, resulting in stator failure.

5. Detection of beryllium carbide requires removal of linings. Beryllium carbide, upon cooling, is a porous material and leaves a pocket in the disk at the "hot spot." If the brake is reused, a new "hot spot" can form in a new location radially under the lining face.

6. Heat generation is not distributed evenly along the length of the brake assembly. Rotors and stators closer to the back plate show larger carbide formations than those close to the pressure plate. The formations are slightly larger at the bottom of the stator than at the top.

7. Brake on time during qual test reusability stops did not exceed 22 seconds. Brake on time for the flights has ranged from 26 seconds (STS-2) with no damage to 52 seconds (STS-5) with the failed stator.

8. Differential braking in a crosswind increases the potential for brake damage and reduces the total brake energy capability available for stopping.

9. Continued use of the brakes close to or exceeding the refurbishment limit will reduce the reusability capability of the assembly.

10. The stator failure on STS-5 was the result of loss of stator strength due to high heat build-up from "extended" medium to heavy braking.

11. Failure of a stator or rotor is not a safety issue. Loss of one or all stators reduces braking capability but does not cause complete loss. The brake would act as a single rotor brake in the case of an all stator failure.

Prior to STS-6, a support bracket was added to the brake assembly which fits between the torque tube I.D. and the axle O.D.. The purpose of the bracket is to reduce the relative deflection between the axle and brake assembly during landing rollout.

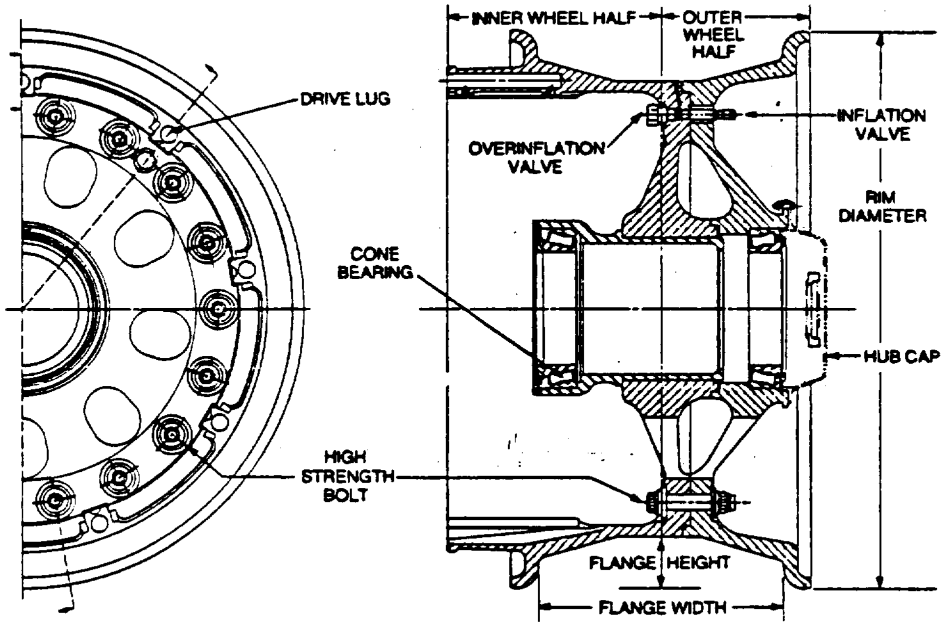
Inspection of the brakes after STS-6 landing did, indeed, show that the bracket eliminated the contact noted on previous flights. However, two brake assemblies approached the refurbishable energy level (34.7 and 32.6 million, respectively) resulting in carbide formations. In addition, three stators in one other brake had cracks in the beryllium. These cracks were traced to the forced interference fit of the "T" retainer spacer which caused scratches in the beryllium and subsequently caused crack propagation during brake use.

This condition has been corrected by rework on the brake assemblies installed on STS-7 as well as the addition of steel clips on the rotor and stator drive slot faces. The clips were added to eliminate the galling of the beryllium at slot faces caused by the relative motion of the drive keys and disks. An evaluation of the clip performance will be made after the STS-7 landing.

B. F. Goodrich is presently performing a study to determine the feasibility of increasing the brake energy capacity within the present limited space that is available, thereby increasing the refurbishment capability.

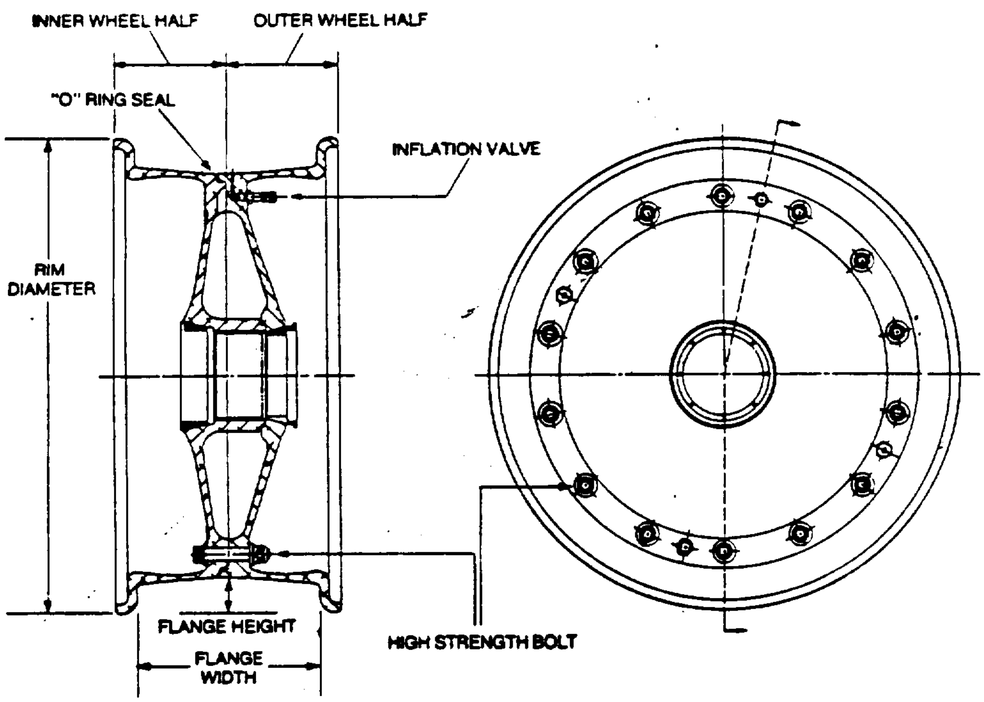
VIII. SUMMARY

The Orbiter wheels and brakes have been subjected to an extremely rigorous test program which far exceeds those used for any wheel and brake presently used in the aerospace industry. Changing requirements, increasing load conditions, and the use of landing load profiles has been the major contributor to the problems which occurred during the certification program and the Space Shuttle STS flights.



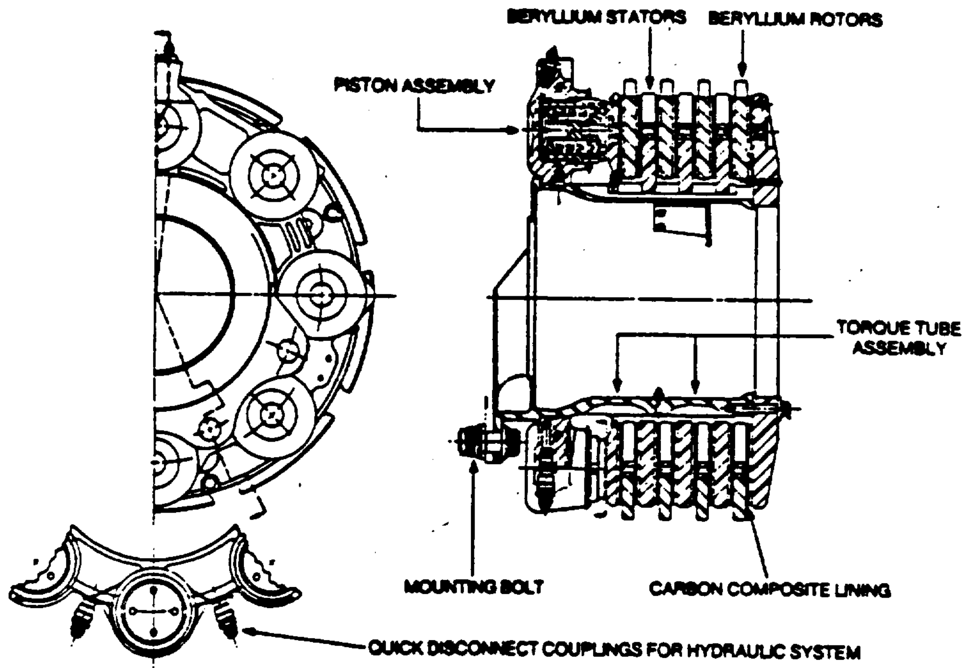
MAIN WHEEL OPERATIONAL CONFIGURATION

FIGURE 1



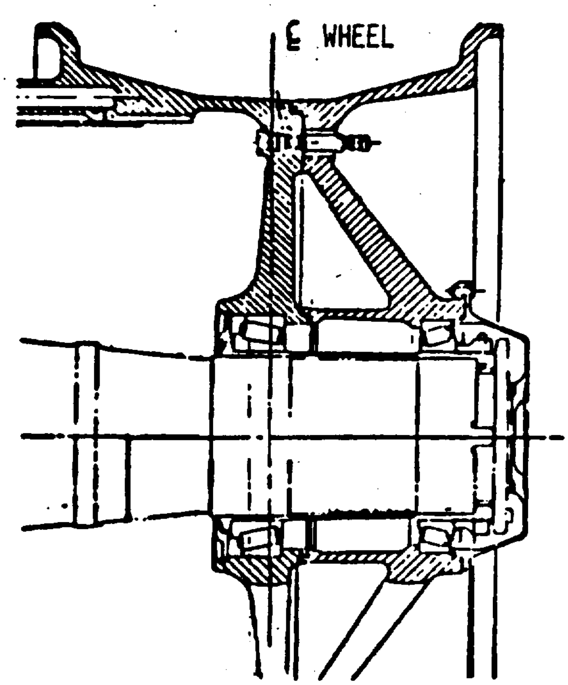
NOSE WHEEL OPERATIONAL CONFIGURATION

FIGURE 2



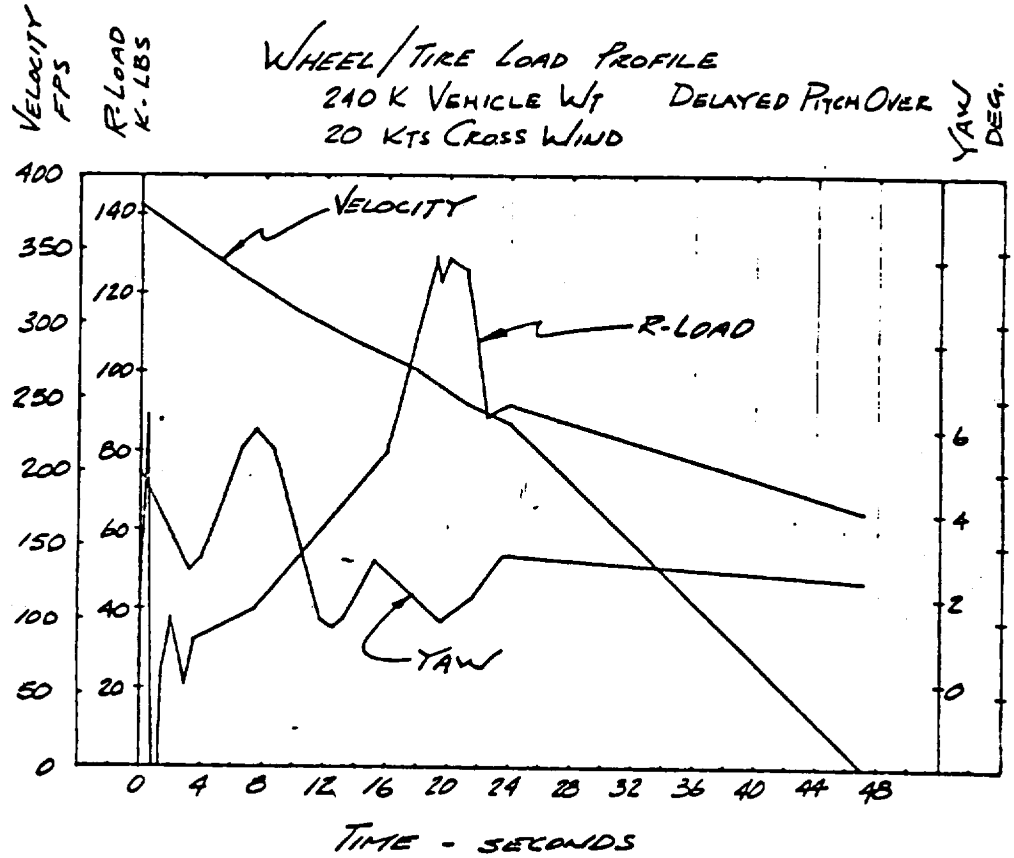
MAIN WHEEL BRAKE OPERATIONAL CONFIGURATION

FIGURE 3



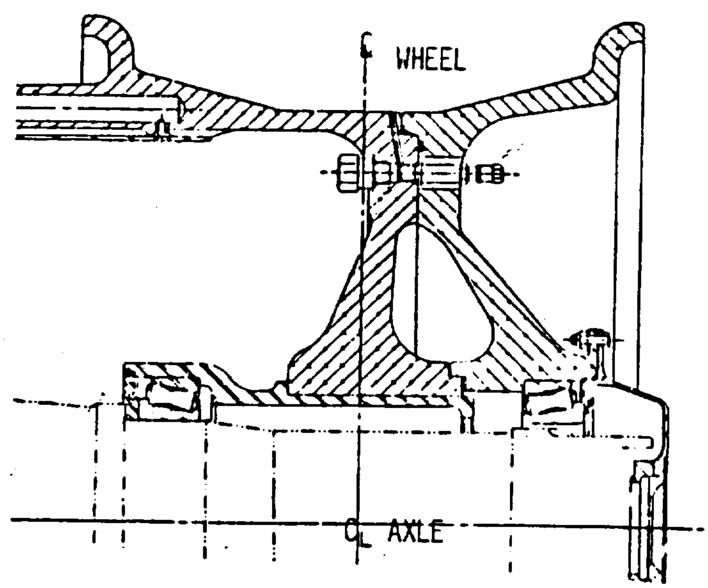
MAIN WHEEL BASELINE CONFIGURATION

FIGURE 4



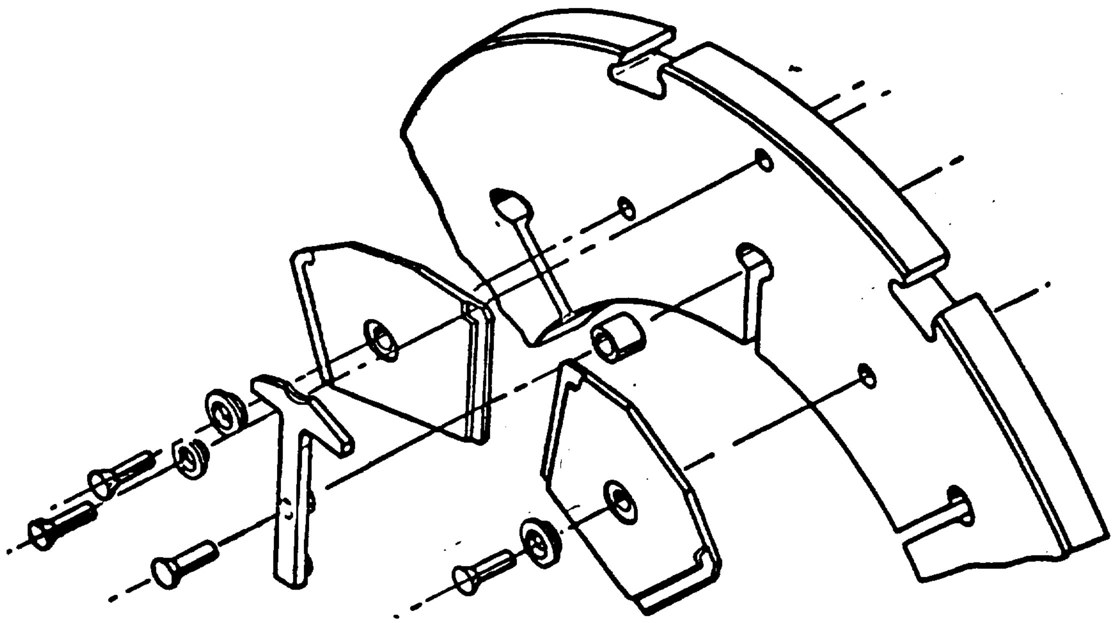
TYPICAL LANDING LOAD PROFILE

FIGURE 5

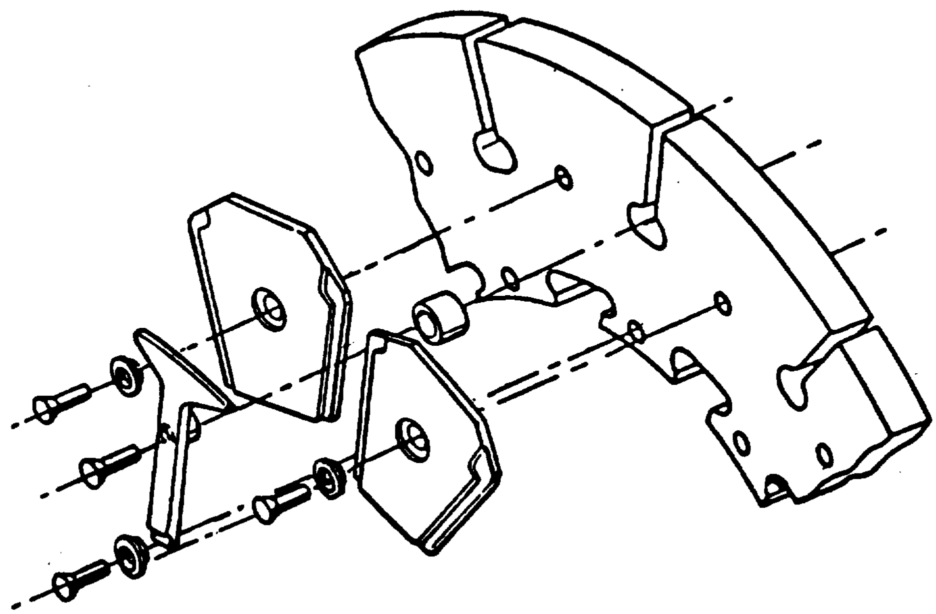


MAIN WHEEL DUAL O'RING CONFIGURATION

FIGURE 6



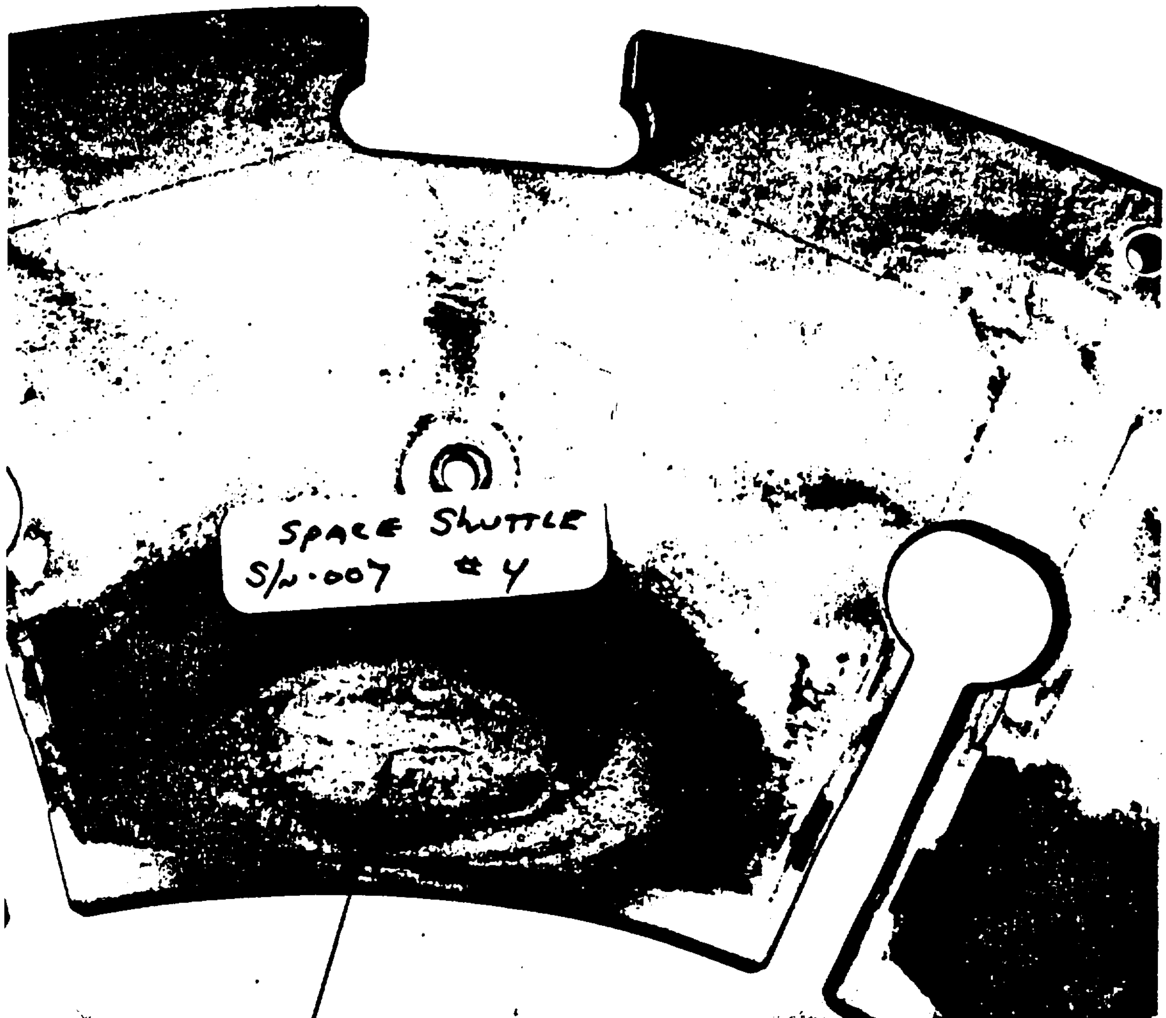
ROTOR



STATOR

ROTOR AND STATOR LINING ATTACHMENT METHODS

FIGURE 7



BERYLLIUM CARBIDE POCKET

BERYLLIUM CARBIDE FORMATION ON ROTOR FACE

FIGURE 8