INVITED PAPER

INDIGENOUS DEVELOPMENT OF A HIGH FORCE RATING ELECTRODYNAMIC SHAKER SYSTEM

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Abstract

In launch vehicles, all systems are subjected to vibration environment during flight. Vibration tests are performed to find out structural integrity and performance of the system for the intended use in a launch vehicle, under vibration environment. The vibration tests are carried out by employing suitable shaker as stand alone unit or with combination of slip table, depending upon the type of vibration levels and axis of vibration. Out of different types of shakers, electrodynamic shakers have wide frequency range so that the sine, shock and random tests are possible.

The force requirement of a shaker is dictated by the mass of the systems to be tested and the test levels. ISRO' future vehicle sub assemblies, demand nearly 250 kN force capacity for the shaker. Indigenous development of such a high capacity system was initiated first time in the country, for self reliance in designing and realizing such high capacity test systems and to meet the schedule critical launch vehicle sub assembly test activities. This paper describes the challenges in the development of such a high capacity system in Vikram Sarabhai Space Centre (VSSC), Trivandrum.

Introduction

A complete electrodynamic vibration test system is comprised of an electrodynamic shaker, electrical power equipment or power amplifier which drives the shaker, slip table system for lateral axis testing, electrical controls and sensors for safety and interlocks, and other auxiliary systems like shaker cooling system, hydraulic power packs etc. The electrodynamic vibration machine derives its name from the method of force generation. The force which causes motion of the table is produced electrodynamically by the interaction between a current flow in the armature coil and the intense DC magnetic field which passes through the coil, as illustrated in Fig.1.

The table is part of the armature assembly, which consist of the armature structure and the armature coil assembly. The armature coil is concentrically located (with radial clearances) in the annular air gap of the dc magnetic circuit. The magnetic circuit is made from soft iron which also forms the body of the vibration machine. The body is magnetically energized, using two field coils as shown in Fig.1. These coils generate a radially directed magnetic field in the air gap, which is perpendicular to the direction of current flow in the armature coil. The generated force in the armature coil is in the direction of the axis of the coil, perpendicular to the table surface. The direction of the force is also perpendicular to the armature current direction and to the air-gap field direction.

The table and armature coil assembly is supported by air suspension, permitting rectilinear motion of the table perpendicular to its surface, corresponding in direction to the axis of the armature coil. Motion of the table in all other directions is resisted by armature guidance system, consisting of a pair of hydrostatic journal bearings and half loop metallic flexures. Table motion results when an ac current passes through the armature coil. The body of the machine is supported by the shaker suspension system with a trunnion shaft centerline passing horizontally through the center of gravity of the body assembly, permitting the body to be rotated about its center, thereby giving a vertical or horizontal orientation to the slip table. The shaker suspension system includes an elastic support of the body, providing vibration isolation between the body and the supporting floor. Because of copper and iron losses in the electrodynamic unit, separate chilled water cooling system is provided to carry off the dissipated heat.

Figure 2 show the components of the shaker system. Designing of the system elements were carried out in VSSC and realisation was done through both VSSC facilities and external agencies.

Major Challenges

The major challenges in the development include

- Electromagnetic design of shaker body elements
- Design and realization of a internally cooled, bonded armature assembly which can withstand 250 kN force continuously
- Design and realization of multi layer, multi turn, electrically single and parallel path cooled field coil assemblies for magnetizing the shaker body
- Design and realization of a high capacity Class D type Modular Switching power amplifier with necessary safety interlock systems
- Realization of shaker body elements, shaker suspension/isolation/tilting system
- Design and realization of armature suspension and guidance systems like half loop metallic flexures and hydrostatic bearings
- Development of technology for welding of 60 mm thick AZ31 Mg alloy plates and realization of 4.9 m x 4.2 m x 50 mm thick Slip plate
- Design and realization of 'T' and 'V' type hydrostatic bearings for slip table etc.

Electro-Magnetic-Mechanical Design

Electro-Mechanical analysis involves computation of the response of the shaker system in terms of current drawn, force developed and acceleration levels experienced when the shaker is driven by a Power Amplifier (voltage source). The design was carried out taking in to consideration of the static, dynamic and thermal requirements.

The magnetic circuit includes the field coils and the shaker body through which the magnetic flux passes. The design of the magnetic circuit aims for a flux density of 2 Tesla in the air gap. The material chosen is pure iron, which has a saturation flux density above 2 Tesla. The magnetic circuit was iteratively trimmed through FE analysis.

The initial estimate of the inner pole diameter is obtained from the preliminary armature design. From the FE analysis of the magnetic circuit, the dimensions of the shaker body and the field ampere-turns required to attain 2 Tesla in the air gap was iteratively obtained. The results of the FEMM analysis are shown in Fig.3. The actual dimensions of the conductors, the number of turns and current were chosen to meet the ampere-turns in such a way that the coil can be accommodated in the space provided for it and can be cooled adequately. The field coils are made from hollow square copper tubes. Each coil consists of multiple horizontal layers with multiple turns per layer. Current required is computed from FEMM analysis and provided in opposite sense in the field coils, so that the flux will be driven through the air gap.

The structural dynamic analysis of the armature assembly has shown that the mass of the coil dominates the first axial mode. In the preliminary design using copper conductor, the first axial mode was at around 1100 Hz, which was below the specified minimum of 1400 Hz. To bring this frequency above 1400 Hz, the mass of the conductor had to be reduced. A study was carried out on the typical properties of conductor grade tube materials. Even though ETP copper has the best conductivity, the specific conductivity index (specific conductivity/ density) is less when compared to most of the common Aluminium alloys. The other important property for armature coil material is the fatigue limit, which is 90MPa for the present case. Based on these, Al alloy has been selected as the coil material for the armature.

The field coils and body are designed to achieve a flux density of 2 Tesla in the air gap. But the armature has been designed assuming that only 1.6 Tesla is available in the air gap. An iterative design approach has been followed to arrive at the final conductor material, dimensions and current such that the axial frequency is above 1400 Hz, cooling water flow is approx 5 m/s, temperature rise is

~30^{o}C, air gap flux density is 1.6T and adequate margins are available for static load and dynamic envelope. Each step in the iteration covered independent Magnetic circuit analysis, Thermal analysis, Structural Dynamic analysis and the final Electro Mechanical analysis.

The shaker characteristics essentially are the dynamic behavior in terms of force, acceleration, current and power dissipated when the shaker is driven by a Power amplifier. This involves an interaction of the electrical and mechanical system dynamics. A method was developed wherein these interactions were also accounted for. This interaction alters the system characteristics in the entire operating frequency range and most significantly it gives rise to a mode (resonant peak) itself, which is captured by the combined model, but completely missed in the separate analyses. This happens because, out of the two reactive elements that give rise to the resonance, one is from the mechanical system (Mass) and the other from the electrical system (Inductance).

It is the result of this integrated analysis that provided the updated shaker characteristics, against which the design parameters were checked and matched for firming up the design. These results also provided the specifications of the Power Amplifier and the cooling system.

Realization of Armature Assembly

The armature assembly consists of the armature structure and the armature coil assembly as shown in Fig.4.

The configuration and dimensions of these elements were finalized based on the electro magnetic, static, dynamic and thermal analyses. The armature structure, being only limited requirements in number, machining route was suggested, rather than the casting route.

Realisation of a defect free, 860 mm diameter, 720 mm height AA2014 casting through forged route was a major challenge. Special toolings were developed for the intricate machining of the forging. Fig.5 show the machining operations for the armature structure.

The armature conductor required for fabrication was around 60 m long AA 1050 hollow rectangular single length tube. Special cold extrusion dies and eddy current based inspection techniques were developed for realization of defect free armature conductor.

The armature coil is resin bonded sandwich construction with special armour plates provided on either side of the helical armature coil. The force is generated in the coil and has to be transferred to the armature structure suitably. Since tension bond alone can not transfer the force, parallel load paths were provided for shear transfer of force generated from successive coils to the armature structure, using armour plates. Optimal thickness armour plates and thickness of the resin were finalized based on parametric studies and static analysis.

The coil dimensions and the concentricity with the armature structure was critical, as the gap provided between the armature coil and the stationery body elements were only 1.5 mm.

Split winding fixtures were developed in-house for this purpose. In order to maintain the critical dimensions of the assembly, a two stage bonding process was finalized and the parameters were finalized through specimen studies. Studies were also carried out on the effect of two stage curing on the strength of the joint. Various options for the bonding resin were studied and a suitable film adhesive was finally selected. Fig.6 shows the armature coil, split mould and armature structure during trial assembly.

The whole bonding process and design was validated through realization of a proto armature coil and static testing the same to 150 % of the rated load (i.e 250 * 3 * 1.25 * 1.5 = 1400 kN). Fig.7 shows the test setup for static testing of 21 turn proto armature coil assembly. Tension bonding was initially carried out under controlled temperature in a curing oven with the coil wound on the split mould (along with armature structure), film adhesive provided between the coils and vacuum bagging. Fig.8 shows the armature assembly after bonding of the coil alone (tension bonding).

After through inspection, cleaning and pressure testing of the assembly, bonding of the armour plates was carried out on inside and outside of the coil. Special processes were developed to avoid electrical contact of the armor plates with the coil. Fig.9 shows the armature assembly after bonding of the armour plates also.

Design and Realisation of Field Coils

For generating the high magnetic flux density in the air gap, the body elements have to be magnetized by the field coils. The configuration of the field coils, i.e: inner and outer dimensions, number of turns required, conductor material and size etc were finalized through the electro magnetic analysis. The specific challenge for this system was configuring a electrically single coil having 240 turns, having parallel paths for water cooling. Based on various iterative studies, the configuration finalized was 24 layers of wound coils in each field coil assembly with 10 turns per layer. Each twin layer coil assembly forms one cooling circuit, connected to the inlet and outlet manifolds for cooling water supply, through electrically isolated conduits.

The conductor used was 8 mm x 8 mm ETP copper coil with 2 mm wall thickness. Electrical and chemical analyses were conducted to ensure material properties. The conductors realized were of varying lengths and were not sufficient for realization of a one layer of the field coil. Hence special brazing techniques were developed in house for joining the coils. Further single length rolls of OFHC conductors (Oxygen Free High Conductivity), were realised. Special winding tapes were used for isolating conductors in each layer. Brazing of each twin layer coils assemblies were carried out for providing the electrical continuity and making the coil assembly electrically single winding. Fig.10 shows the field coil assembly during fabrication (with one of the inlet power / cooling water manifold).

In order to ensure that the required specifications are met, the electrical resistance of each twin layer assemblies was measured prior to final assembly. Ball testing was conducted on individual coil layers, after joining, to ensure 'Zero' block. The twin layer coils sets are pressure tested to ensure leak tightness. Fig.11 shows the top field coil assembly.

Realisation of Switching Power Amplifier

The power amplifier system is the major component of the total system, which provides the required power to the shaker. The major challenge was designing modular Class D switching power module and paralleling such modules to generate the required power to drive the shaker.

The system realized was a modular design with a Class D switching module having a capacity of 20 kVA. By paralleling the modules, required power could be generated. Eight such modules were configured as one rack with independent bay interface unit and high current DC power supply. The bay interface unit provided the necessary control feedback for the power modules in the rack and monitors the health of each module. The modules were arranged in three racks with a total capacity of 480 kVA. Necessary feedback controls were also provided for equal sharing of the current between the modules when paralleled.

The total system was designed and two proto modules were initially developed. Fig.12 shows the MOSFET based proto module for the power amplifier. Performance evaluation of these modules was carried out using water load and a low force rated shaker available. MOSFETs were used in the module as power devices.

Specially designed output transformer with a 5 kHz operational frequency range was provided to meet the high voltage requirements for random tests. Copper wire caging ('Faraday cage') was provided for the total system and earthed to dedicated special earthing system, to reduce the switching noise. The system was initially tested with resistive load and further connected to the shaker system and tuned. Fig.13 shows one of the power amplifier rack with 8 power modules and its power supply unit.

Configuring the Shaker Suspension, Isolation and Tilting System

Specific requirements for the shaker suspension system were isolation of the shaker body from the support system and to facilitate tilting of the shaker (a) 90 degrees for attaching to slip table during lateral tests and (b) 180 degrees during assembly/ maintenance operations.

The major challenges in the realization of the system were (a) Configuration of the system meeting design requirements and fabrication feasibility, (b) Finalization of specifications and procurement for various sub systems elements like air springs, linear ball bush bearings, chain drives, motorized gear box etc and (c) Identification of agency for realization. Fig.14 shows the configuration finalized after various iterations.

Due to the complexity of the design and various integration requirements, a heavy machine fabricator was selected for the realization and assembly of the system. All the input materials were procured and assembly operations were carried out at the factory of the builder. Fig.15 shows the assembly operations for shaker suspension system and Fig.16 shows the system after realisation.

Realisation of Armature Suspension and Guidance System

The armature inside the shaker is freely suspended. The suspension system supports the static load on the armature and provides options for centering the armature coil in the magnetic flux path. For static load support, shaker is sealed and pressurised. Special seal boots were realized to connect the moving armature and stationary shaker body part. An Automatic Armature Positioning System (AAPS) was realized using optical sensing elements and solenoid valves for automatic centering of the armature.

Guidance of the armature is critical to avoid rubbing of the armature against the shaker body. Two hydrostatic journal bearings as shown in Fig.17 were provided to balance the overturning moments acting on the armature. The journals are attached to the armature, while the housing is attached to the inner pole piece. Test rigs were developed to test the overturning moment capability of the bearings.

Half loop metallic flexures were realized in house for armature guidance. This consisted of layers of Beryllium - Copper sheets formed in 'U' shape. The stiffness of these springs is dictated by the configuration and heat treatment process. Being a multi layered component, analytical approach was not feasible in finalizing the design. Hence a 'fabricate and test' approach was used. Various configurations were realized and static tested prior to finalization of the configuration. Fig.18 shows the test setup for stiffness evaluation of half loop metallic flexures.

Technology Development for Welding of Mg Alloy Plate

The slip table system is required for testing of test articles in the lateral axis. This consist of a base plate, a set 'T' and 'V' type hydrostatic thrust bearings and a Mg alloy slip plate. Fig.19 shows the configuration of the slip table assembly.

The slip plate configured was one of the largest slip plates in the world, a Mg Alloy AZ 31 plate, 4.4 m x 4.92 m size and 50 mm thick, to meet the testing requirements of 4 m class sub assemblies. The input material available was 5 m long, 1.5 m wide, 60 mm thick rolled Mg alloy plates. Hence technology for welding 60 mm thick Mg alloy plates had to be developed in house for realizing the slip plate. For this a TIG welding process was developed using AZ61 filler wire, a multi pass welding scheme with stage gauging and X ray inspection was finalized. Process evaluation was carried out on samples and NDT carried out for evaluating the porosity, tungsten inclusion etc. Mechanical tests were also carried out for weld strength evaluation.

The technology was transferred to industry for welding of the plates. The main challenge was controlled cooling of the molten metal to avoid residual stresses and warping of the plate. This was achieved by heating the plates using heating pads. Even with such precautions, the warping of the plate after the two welds were around 40 mm. Through controlled heat treatment with simultaneous application load through dead weights, the warping was brought down to less than 10 mm. The machining of the plate was carried out in-house and the 50mm thick plate was realized successfully. Fig.20 shows machining of the welded Mg Alloy plate.

Conclusion

The realisation of a high capacity shaker system was a challenging task, as it involved design of complex system meeting the static, dynamic, electromagnetic and thermal requirements. Many technologies were developed in-house for realisation of the shaker system elements. The system has been installed and commissioned, meeting the design specifications. The slip table system is one the largest in the world. The indigenous shaker system developed by VSSC would contribute largely in meeting the test requirements of ISRO's future launch vehicle systems.

Acknowledgements

Realisation of this system was possible with contributions by members from various Entities/ Divisions of VSSC. The team acknowledge the valuable contributions by Materials, Fabrication, Avionics, Control and Actuation Systems, Structural Analysis, Thermal Analysis, Structural Testing, Dynamic Testing, System Reliability Agencies and GSLV MKIII Project. Suggestions and guidance by the Shaker Review Committee are also acknowledged which were valuable in solving many of the issues.

Reference

1. "Harris Shock and Vibration Hand Book", Ed: Allen G. Piersol and Thomas. L. Paez, Chapter 25 , Vibration Testing Machines, David O Smallwood.

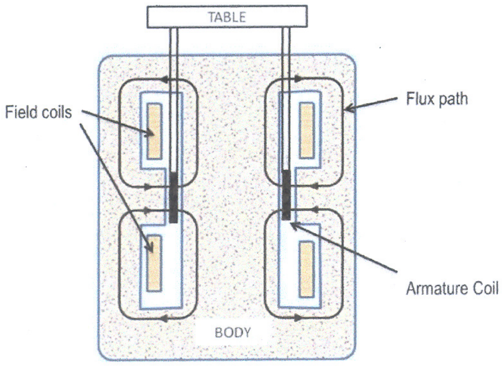

Fig.1 Shaker Configuration

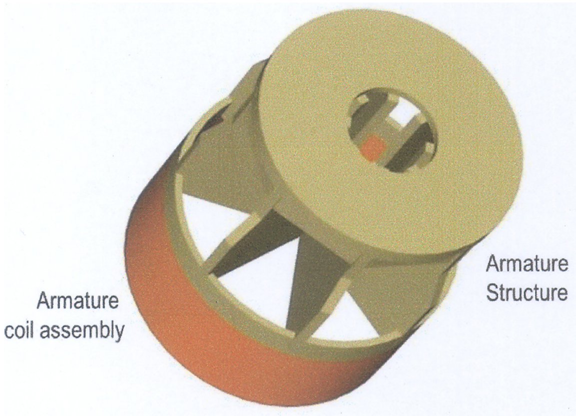

Fig.4 Armature Assembly

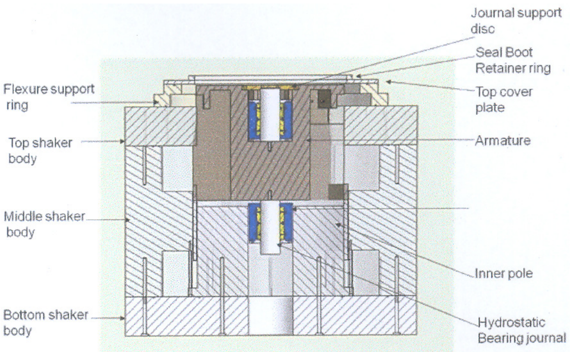

Fig.2 Major Components of the Shaker

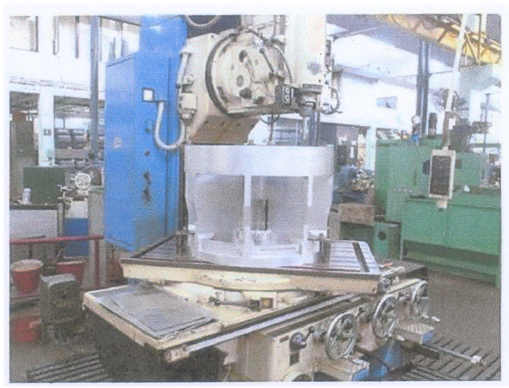

Fig.5 Fabrication of Armature Structure

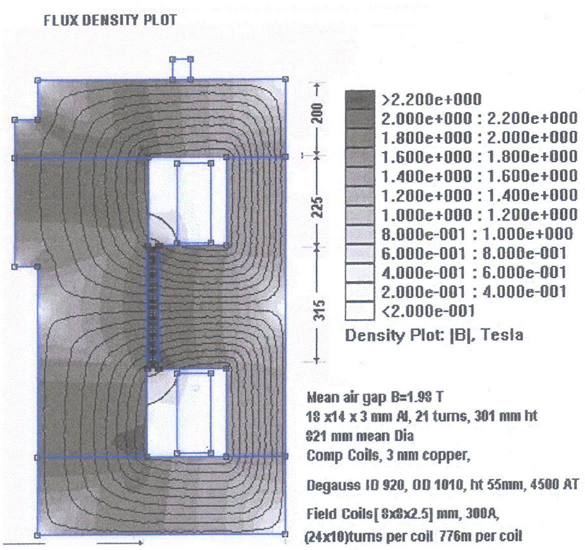

Fig.3 Electro Magnetic Design Details

Fig.6 The Armature Coil, Split Mould and Armature Structure

Fig.7 Loading Testing of 21 Turb Armature Coil Assembly

Fig.8 Armature, After Tension Bonding

Fig.9 Armature Assembly

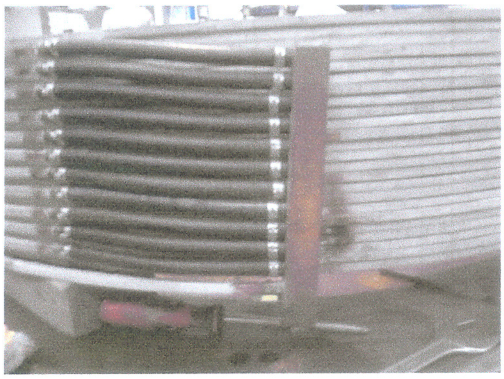

Fig.10 Field Coil with Inlet Manifold

Fig.11 Top Field Coil Assembly

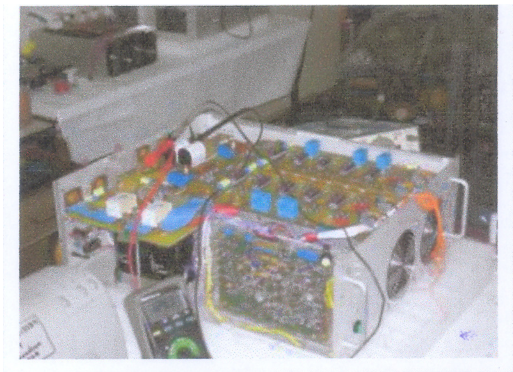

Fig.12 MOSFET Based Proto Module

Fig.13 Power Amplifier Rack with Power Supply

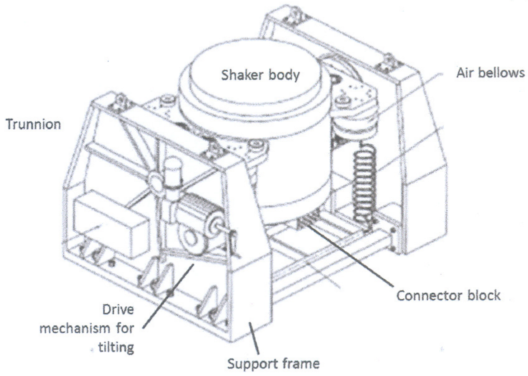

Fig.14 Shaker Suspension System

Fig.15 Assembly Operations at for Shaker Suspension System

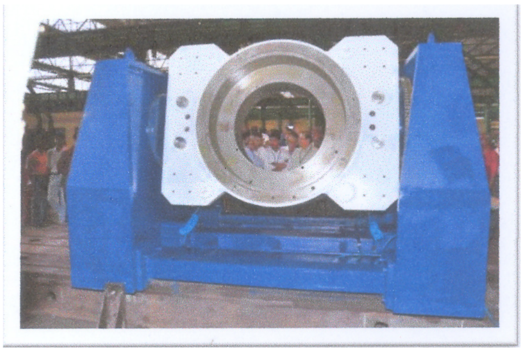

Fig.16 Shaker Body with Suspension System

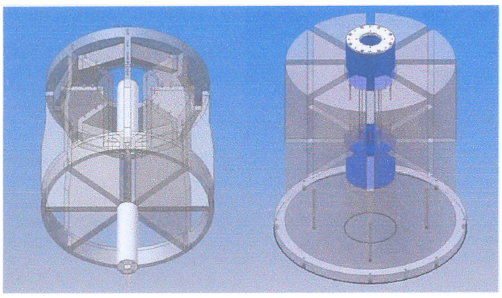

Fig.17 Bearing Assembly Inside the Shaker

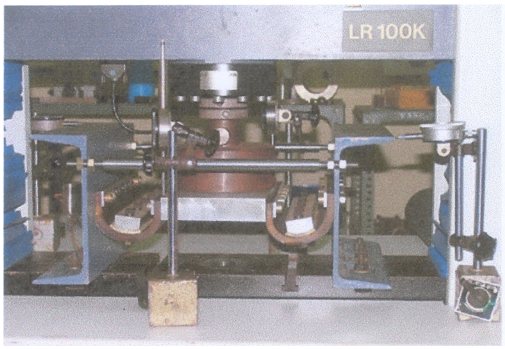

Fig.18 Static Testing of U Type Metallic Flexure

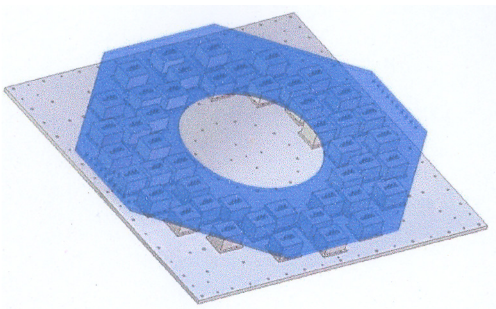

Fig.19 Slip Table Assembly

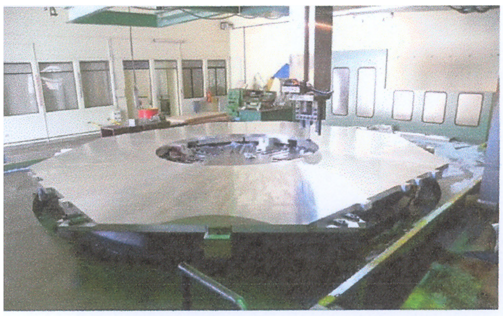

Fig.20 Machining of Welded Mg Alloy Plate