

DUAL WOUND DC BRUSH MOTOR GEARHEAD

Barrie W. Henson*

This paper describes the design requirements, the design, development tests and problems, the qualification and life test and the findings of the strip examination of a dual wound DC brushed motor gearhead. At the time of writing it is the only space qualified dual wound DC brushed motor gearhead in Europe.

INTRODUCTION

The dual wound DC brushed motor gearhead which is described herein was developed under the auspices of the ESA Space Telescope Solar Array Project. Moore Reed & Co. Ltd., Andover designed and manufactured the motor and OTM Servomechanism Ltd., Staines was responsible for the gearhead.

Brush motors have been a cause for concern with regard to reliability of the brush/commutator interface. However, they do have advantages over stepper and brushless types, in particular simplicity of electronics and lower costs. This unit has also demonstrated that a long life and reliable performance can be obtained.

REQUIREMENTS

The unit was designed especially to fulfill the requirements and interfaces of a drive unit in the Hubble Space Telescope Solar Array. However, it is suitable for general use on space projects. It was necessary for the unit to be compatible with the existing electronics and, therefore, the motor had to be a direct current "brushed" type. A picture of the unit is shown in Figure 1.

The specification defined by ESA identified the following key aspects :

1. Performance parameters:
The unit should operate at a nominal Voltage of 28 vdc.
The design rating should be 1.5 Nm output torque with a shaft speed greater than 30 r.p.m.
Note: The qualification and life test was performed at a torque of 1.27 Nm (180 oz.in) based on the project requirement.

*European Space Agency, Noordwijk, Holland.

2. Interface requirements:

The mounting interfaces at the front and back should facilitate mounting in a tube with access from one end.

3. Configuration:

To provide electrical redundancy the motor design should comprise a dual wound armature with a commutator at each end.

The motor power should be routed via terminal posts mounted in the rear end cap. Provision should be made to facilitate purging with inert gas if required for the brush material.

4. Special aspects:

The brush material, bearings and lubrication should utilise space qualified materials, parts and techniques.

5. Environment:

The following qualification levels should be demonstrated:

- . Vibration : 10g Sine, 19.5 g(rms) Random
- . Temperature : Survival + 95°C to - 60°C.
Operational + 80°C to - 45°C.

In addition to the basic performance parameters above, the unit had to be capable of being backdriven to enable astronaut manual over-ride.

DESIGN

Motor

The motor is a two pole permanent magnet device in a frame size 18. A fundamental feature of the unit is the dual wound armature with a commutator at each end. This arrangement provides electrical redundancy using a common armature, shaft, housing and magnet system. The configuration is shown in Figure 2.

The principal electrical system is duplicated to provide redundancy in the event of winding or connection failure. The two windings are radially separated and electrically isolated in the armature stack by wrapping the respective windings in Melinex slot liners. The winding layout is shown in Figure 3. The inner winding (redundant) is connected to the front commutator and the outer winding (prime) is connected to the rear commutator.

The armature wire size has been carefully chosen to give the same torque from each winding and the number of turns per winding is identical to ensure the no-load speed is equal. The resulting performance per winding is higher than a normal size 11 motor and the stall torque is approximately double.

Two ceramic magnets are bonded to the stainless steel housing. They are made of barium ferrite which was chosen because it has a good resistance to demagnetisation. This permits the rotor to be removed and replaced without affecting the performance. Although the magnetic characteristics of the material exhibits a knee in the BH curve, it is not possible to demagnetise the motor under normal operations or by reversing the motor whilst running at ambient temperature.

The commutators are made of copper and are comprised of eleven segments set in a moulding. A pair of spring loaded brushes are located in a holder adjacent to each commutator. The wires from the front brush assembly pass through a tube located in the gap between the magnets and exit through the rear end cap where terminal posts are provided for external connections.

The brush material is Boeing compound 046-45. The material is supplied in a compact form and comprises mainly molybdenum disulphide. It has a low rate of wear in vacuum but has the disadvantage that it requires purging when used in air. Because it has a high MoS₂ content the commutator film has a relatively high and variable resistance which influences the performance characteristics at low voltages and loads. In normal operation these effects disappear. For this reason the motor run-in tests are made at 28 Volts and a torque of 100 gcm. The motor shaft is supported on Bartemp dry lubricated bearings. Lubrication is achieved by transfer from a sacrificial cage made of a PTFE/MoS₂/glass fibre composite.

Gearhead

The gearhead is manufactured entirely from stainless steel. It has a conventional spur gear and layshaft configuration and has a ratio of 160:1. There are five layshafts and each one is mounted on miniature ball bearings. This arrangement results in a high overall efficiency (✓90%) and permits the unit to be backdriven. A cross-section of the gearhead is shown in Figure 2. The bearings used are mainly the caged type with shields, but in two cases different bearings had to be used to accommodate the loads and envelope constraints. All bearings are lubricated with Braycote 601 grease.

All the gears and layshafts are hardened by a diffusion process known as "Tufftriding". This is a proprietary type of nitriding process and is used to improve the wear characteristics. It also improves the fatigue life.

Motor Gearhead Assembly and Mounting Arrangements

The motor-gearhead interface comprises a spigot location with a screw fixation and the motor shaft pinion meshes directly with the first layshaft gear in the gearhead. Locking of the interface is achieved by four small spots of adhesive.

Performance of the unit with one winding powered at 28 Volts is shown below and is illustrated in Figure 4.

No load speed (r.p.m.)	:	38 - 43
No load current (mA)	:	50 - 100
Speed at 1.5 Nm (r.p.m.)	:	30 - 34
Current at 1.5 Nm (mA)	:	300 - 420

The performance with both windings powered is very similar to Figure 4 but the speed at rated torque is slightly higher and of course the total current is shared between the two windings. The torque required to backdrive the unit is less than 0.7 Nm (100 oz.in.).

The purpose of the two windings is to provide redundancy rather than to increase the torque capability. The gearhead is designed for the rated torque of 1.5 Nm and has a factor of safety of 2 based on static loads. The motor gearhead should not be operated above the rated torque.

In the specific application for which it was designed the motor gearhead is mounted inside a tube. This is achieved by using a special collar at the front end and a Delrin spider at the rear end as shown in Figure 5. Various alternative methods of mounting may be considered but they should always provide support at the front and rear.

DEVELOPMENT PROGRAMME

Choice of Lubricant

Previous experience indicated that Braycote 601 grease (formerly 3L38RP) was the preferred lubricant for the gearbox bearings. However, for the gears, it was decided to make a comparison test with Braycote Y Vac 3 grease, because of its reported good behaviour in an industrial application.

Two identical gearheads, size 18 with a 100:1 ratio, fitted to a size 11 motor were evaluated at ambient temperature and in vacuum over a temperature range of + 85°C to - 45°C with a torque load of 0.7 Nm (100 oz.in).

The main difference in the performance of the two gearheads was at low temperature (-45°C) when the gearhead containing Y Vac 3 grease required a significantly higher motor current than the one with 601 grease for the same output torque. This was attributed to the increased viscosity of the base oil (Fomblin Y 140) in the Y Vac 3 which has a pour point of - 25°C and therefore was semi-solid in this test. Thus considering the required environmental range Braycote 601 was chosen because its low pour point ensured consistent performance even at - 45°C.

Choice of Brush Material

Many types of material have been considered by engineers for use in DC brushed motors. The ideal solution has still to be found as most materials have some disadvantage when operation in both air and vacuum is required. Normally the most critical problems occur in vacuum (often at low temperature) when high wear rates are observed.

The motor used in the gearhead lubrication tests was fitted with brushes made from a material designated Sperry 318 E. The motor gearhead performed very well but after 95 hours (33 hours in vacuum) the motor stopped. Investigation revealed that the brushes were completely worn out, a length of some 6 mm being consumed. Consequently the commutator area was full of powder. This result persuaded us to return to a known material, namely Boeing compound 046-45, because previous experience indicated that it had good wear resistance.

Bearing Failures

The first 160:1 gearhead tested was coupled to a size 11 motor. The unit was subject to thermal vacuum testing and a life test. After 39 hours operation a gearhead failure was observed. Investigation revealed that a bearing on the third layshaft had collapsed although the rest of the gearhead was in good condition. This was an unexpected failure mode and from a brief examination it was considered to be a random failure.

The second 160:1 gearhead was coupled to a dual wound size 18 motor. The unit was subject to sine and random vibration at qualification levels. Subsequent derivation of Q factors due to fixture design and mounting resonances indicated that the unit was subject to random levels of the order of 100 g(rms). To improve this situation for future vibration tests a new fixture was designed. The unit was then subject to thermal vacuum testing and life testing. A total of 82 hours running time had been accomplished when the unit failed. Investigation revealed that the identical bearing on the third layshaft had collapsed. The gearhead was generally in good condition, but the appearance of the grease had deteriorated and some evidence of debris existed. This failure eliminated the "random" bearing failure theory previously supposed.

A detail design assessment of the gearhead was initiated. This resulted in the following conclusions:

- drawing dimensions allowed axial pre-load to occur under worse case tolerance conditions.
- dimensions of the third shaft resulted in the bearing being under an axial pre-load and induced a small angular misalignment of the bearing.
- the failed bearing had higher loads than all other identical bearings in the gearhead and these were not conducive to a long life.

Resolution of these difficulties was achieved by implementing the following design changes: -

- Revision of axial dimensions to ensure clearance existed under all tolerances.
- Inclusion of a double bearing in the failed location on the third shaft.
- Application of the Tufftride process to the gears and shaft. This change was made to reduce wear debris which could promote breakdown of the grease and become a source of contamination in the bearings.

To ensure that these changes had eliminated causes of the bearing failure a modified gearhead was subject to a 200 hour life test. This test was completely successful.

Shaft Failure

The gearhead which had accomplished 200 hours operation was still running smoothly and it was decided to continue the test to obtain more confidence. At 240 hours a catastrophic failure occurred. The third layshaft fractured at a change in the cross-section from a gear face to the bearing spigot. This formed an almost square corner, less than 0.125 mm (0.005") radius to facilitate seating of the bearing.

A thorough metallurgical examination revealed that during final machining the Tufftride layer had been ground away on the bearing spigot. Thus at a critical change of section with a sharp radius attracting a stress concentration factor of about 4 the material properties also changed. The section and surface structure are shown in the scanning electron microscope picture, Figure 6, and the fracture line is shown in Figure 7.

The running time equated to approximately 3 million revolutions of the shaft, i.e. some 6 million stress reversals. Analysis of the load/fatigue curve indicated that the 'expected' life had been reached. Whilst the running time was some 5 times more than required for the intended application, it was considered essential to initiate design and process changes to improve the product in view of the wide variations associated with fatigue failures. The changes required were quite straightforward, viz.:

- 1) increase the blend radius from 0.125 mm (0.005") to 0.250 mm (0.010"), and thus reduce the stress concentration factor from 4 to 1.5.
- 2) finish grinding the bearing spigot prior to applying the Tufftride process because the growth in size due to the Tufftride process is insignificant.

These changes were consequently incorporated in the Qualification and Life Test model for formal testing.

QUALIFICATION AND LIFE TESTS

The planned test programme comprised:

- performance tests at motor level.
- performance tests at motor gearhead level.
- vibration tests.
- performance tests at ambient and in thermal vacuum.
- a 100 hour life test at ambient and in thermal vacuum.

The life tests were extended to 250 hours because performance remained satisfactory and time permitted.

The performance tests at motor and motor gearhead level confirmed that the unit complied with its specification.

Vibration

The motor gearhead was mounted using the collar and spider shown in Figure 5.

The qualification level vibration test comprised:

a. Sinusoidal

- 5 - 20 Hz : 20 mm peak to peak
- 20 - 100 Hz : 16.3 g.
- Duration : 1 sweep per axis at 2 octaves/min.

b. Random

- Level : 19.5 g(rms)
- Spectrum : See Figure 8
- Duration : 3 minutes per axis

During these tests the actual level on the motor was also monitored. It was noted that a substantial degree of cross-coupling occurred even with the new fixture. Q factors up to 3 resulted in levels up to 60 g(rms) on the motor. Post vibration function tests confirmed that the unit was still in good working order.

Thermal Vacuum

The thermal vacuum and life tests were performed at the European Space Tribology Laboratory, Risley U.K.

The motor gearhead was subject to function tests at a torque of 1.27 Nm (180 oz.in) at ambient temperature with a nitrogen purge and in vacuum over a temperature range of + 80°C to - 45°C. The unit was also subject to temperature survival tests at + 95°C and - 60°C. Performance was nominal throughout the tests and all motor configurations were demonstrated.

Life Tests

The planned life test comprised of operations on each winding at a torque of 1.27 Nm (180 oz.in) at ambient and in vacuum over a temperature range of + 50°C to - 30°C. During the initial phase of the ambient life test the winding 1 current trace became noisy and the shaft speed reduced. After some brief checks of no-load speed, start voltage and breakaway torque measurements, the performance reverted to nominal. It was concluded that the temporary anomalous performance was caused by a brush fragment. The motor gearhead continued to function well for the planned duration of 100 hours, 50% of which were in vacuum.

Extended Life Test

Due to the satisfactory performance obtained, the life test was extended until a total running time of over 250 hours had been accomplished. Fifty percent of this testing was in vacuum and the motor gearhead was still giving a nominal performance.

STRIP EXAMINATION AND ANALYSIS

The motor gearhead was progressively stripped and subject to a full examination. The main observations were:

- the motor condition and general appearance was good.
- the gearhead condition and general appearance was clean and good with adequate grease coverage.
- the brushes were in good condition but had worn slightly unevenly, at least 50% of their length remaining.
- the commutators were showing slight wear as expected.
- some brush debris was evident in various places, obviously having been blown around by the gaseous purge used when operated in air.
- insulation resistance had reduced to values of 10 K Ω and 80 K Ω . It is believed this was due to tracking across debris paths possibly formed by centrifugal deposition of MoS₂.
- all gears were in good condition and there was no evidence of wear.

SUMMARY

The motor gearhead has performed extremely well and completed a life test far in excess of that originally planned. The development problems encountered were very instructive and indicate the care and detail consideration that needs to be taken in the construction of reliable motor gearheads.

ORIGINAL PAGE IS
OF POOR QUALITY

ACKNOWLEDGEMENT

The author wishes to thank the following personnel for their contributions towards this successful project.

Mr. R. Smith, Mr. A. Flew
Mr. A. Yardley
Mr. J. Anderson

- Moore Reed and Company Ltd.
- OTM Servomechanism Ltd.
- European Space Tribology Laboratory,
(ESTL), Risley, U.K.
at the European Space Agency, Noordwijk

My colleagues

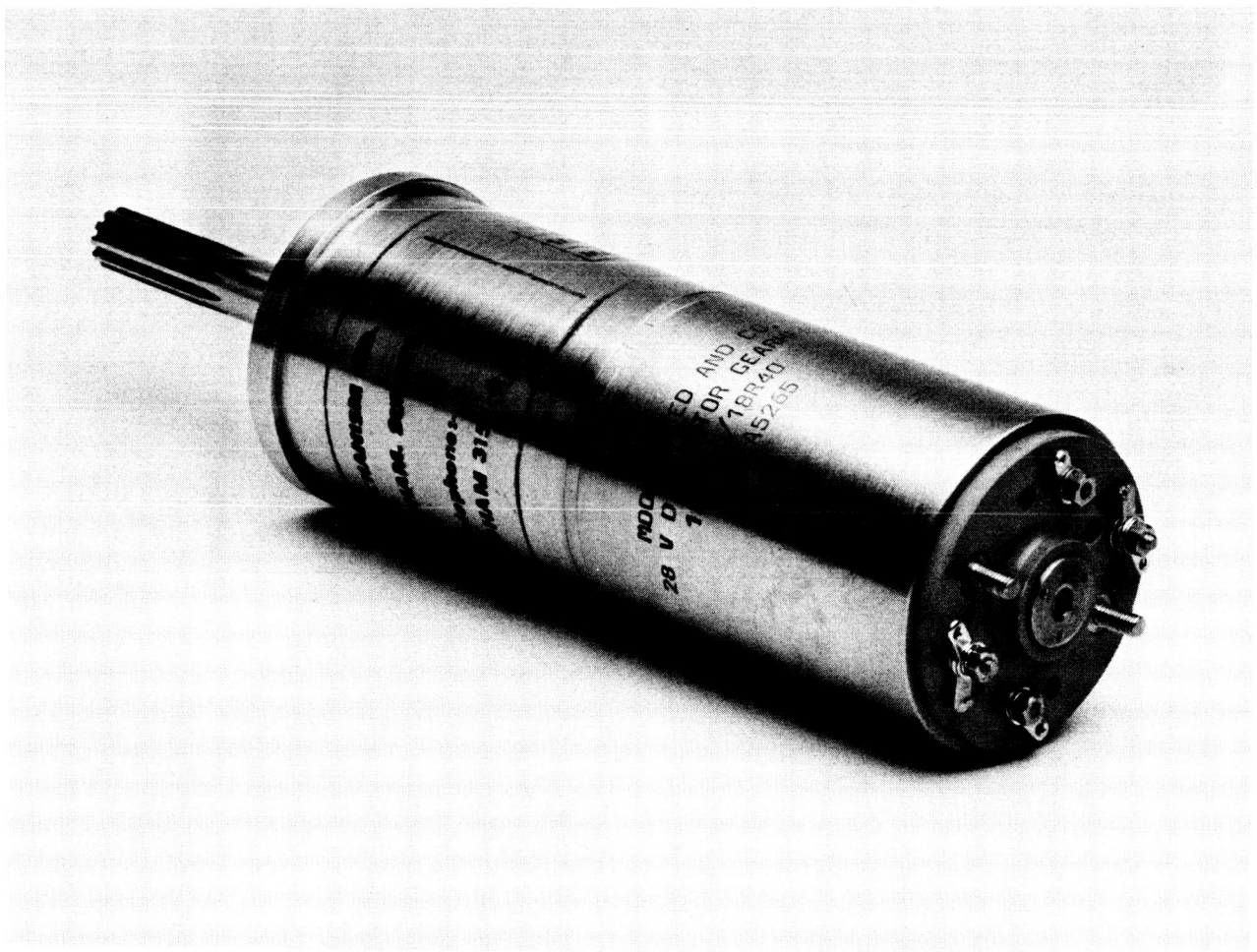


Figure 1. - Dual wound motor gearhead.

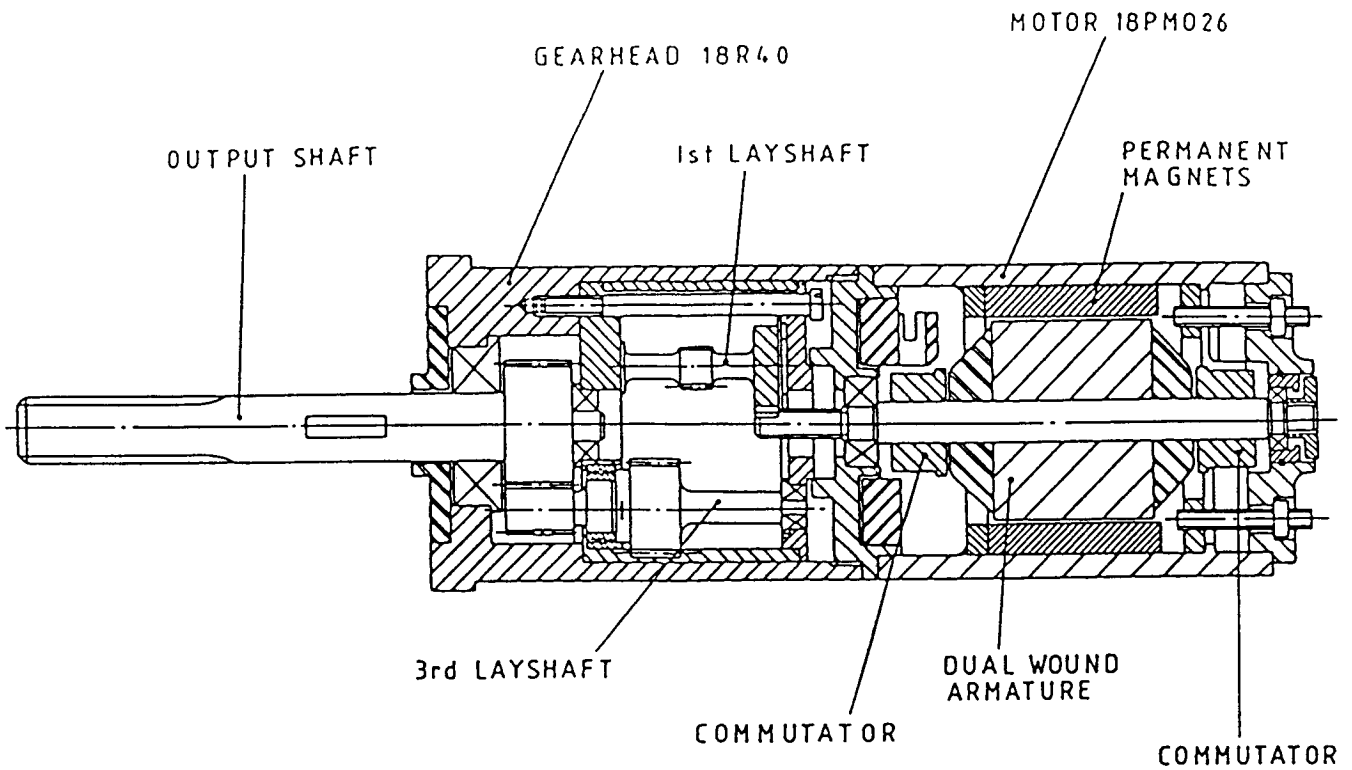


Figure 2. - Section view of dual wound motor gearhead. Note second layshaft omitted for clarity.

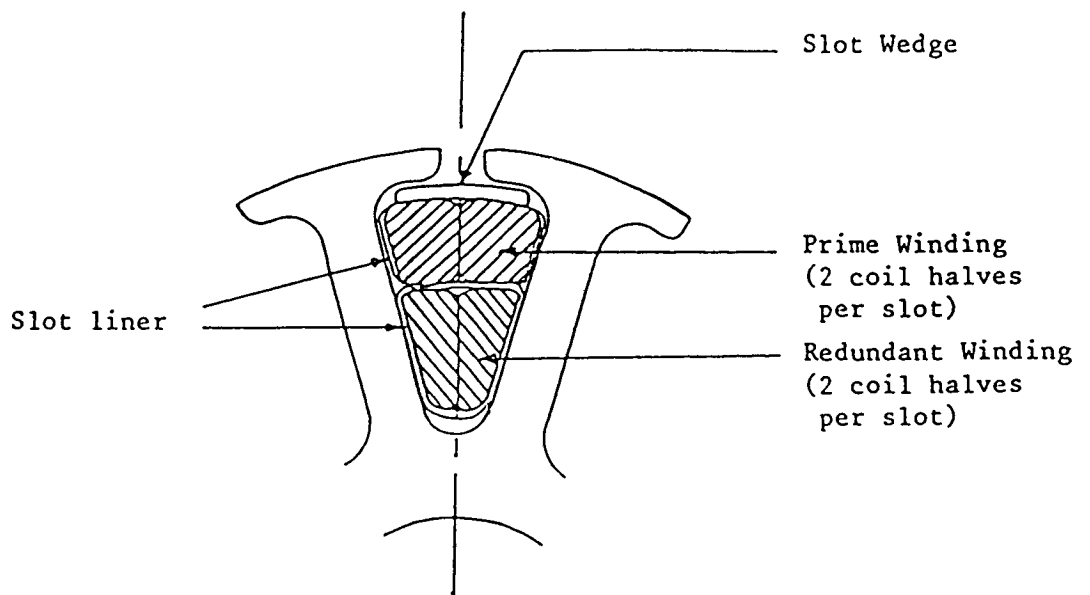
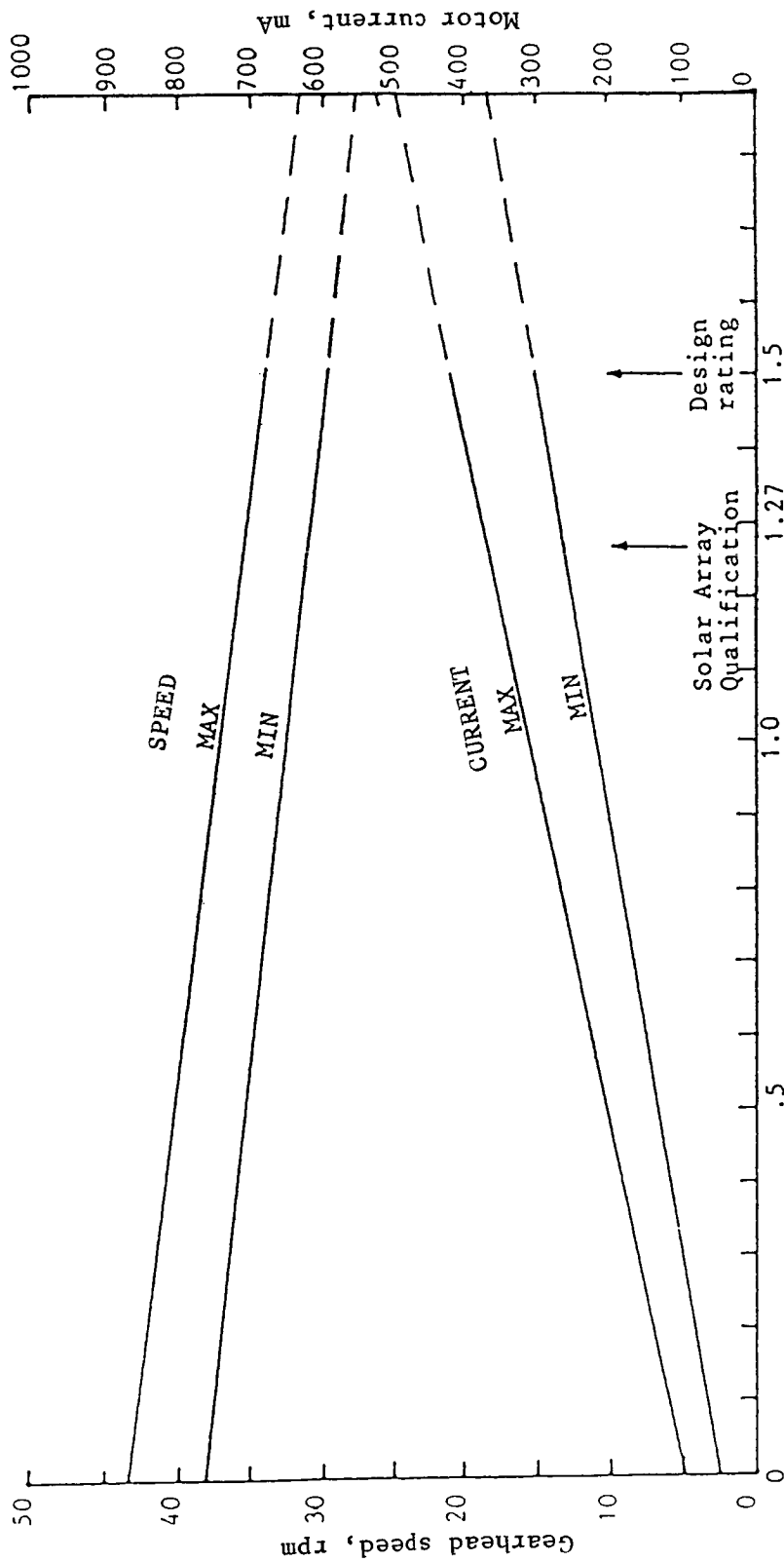


Figure 3. - Winding layout.



Gearhead torque, N-m

Figure 4. - Gearhead torque as function of gearhead speed and motor current. One winding powered at 28 volts.

ORIGINAL PAGE IS
OF POOR QUALITY

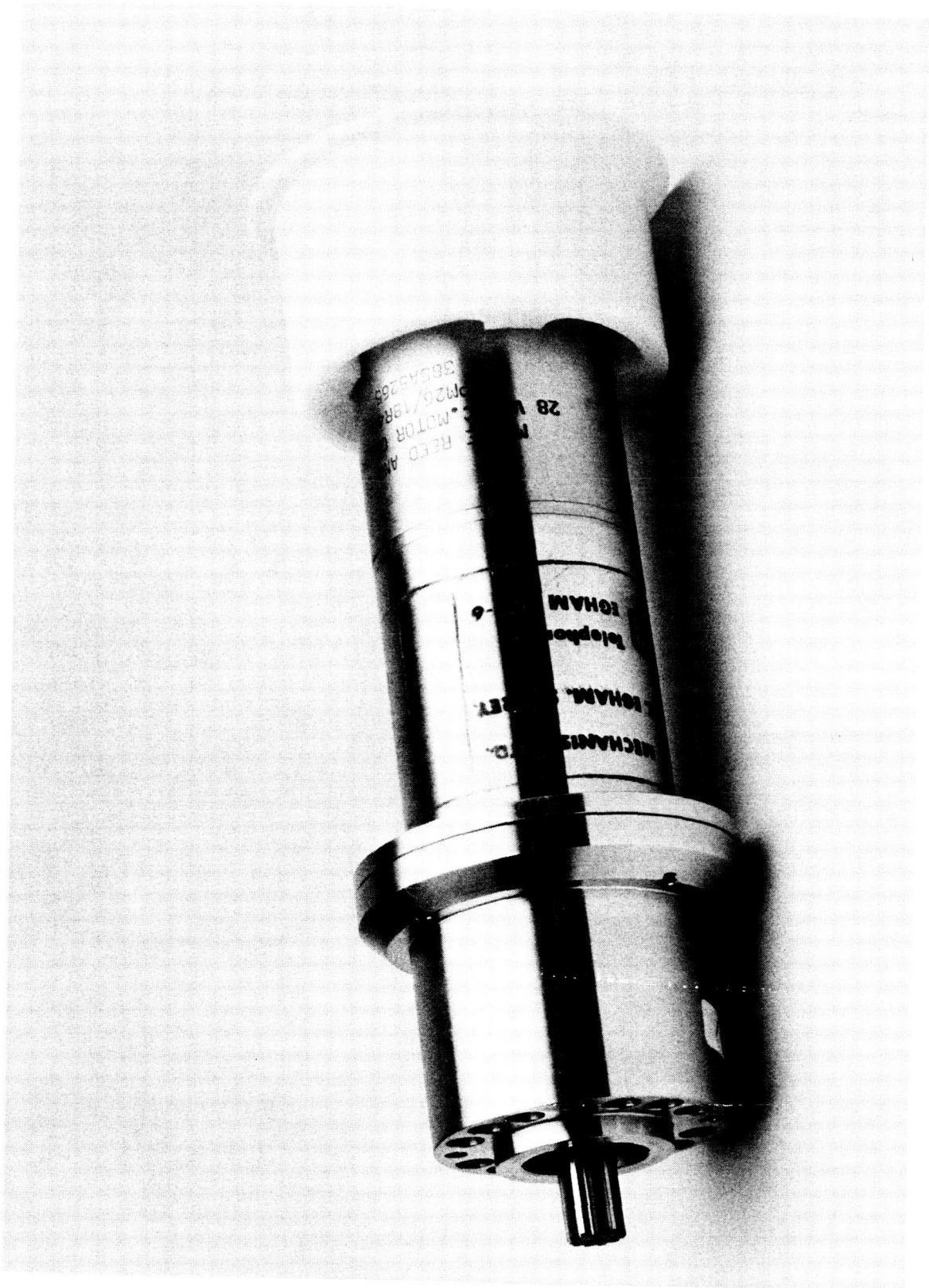


Figure 5. - Dual wound motor gearhead mounting arrangement.

ORIGINAL PAGE IS
OF POOR QUALITY

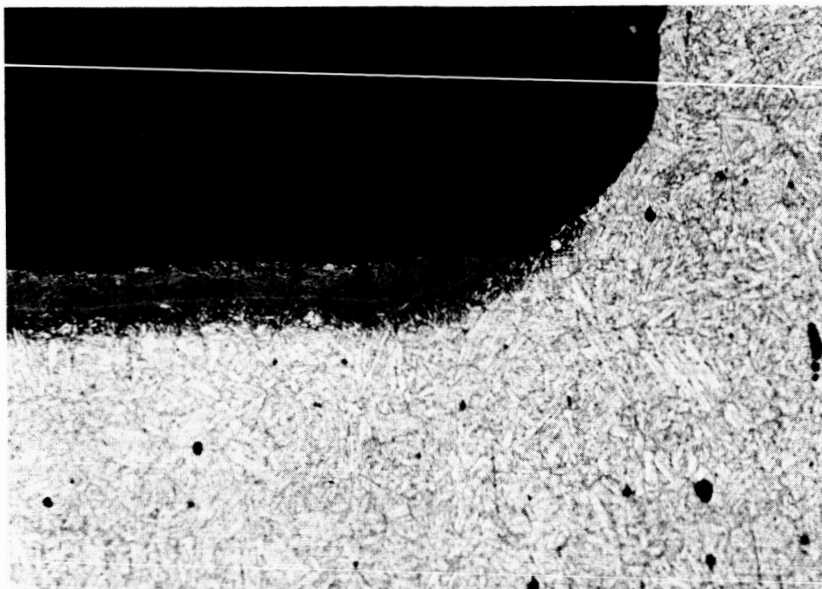


Figure 6. - Cross section of gearhead shaft.

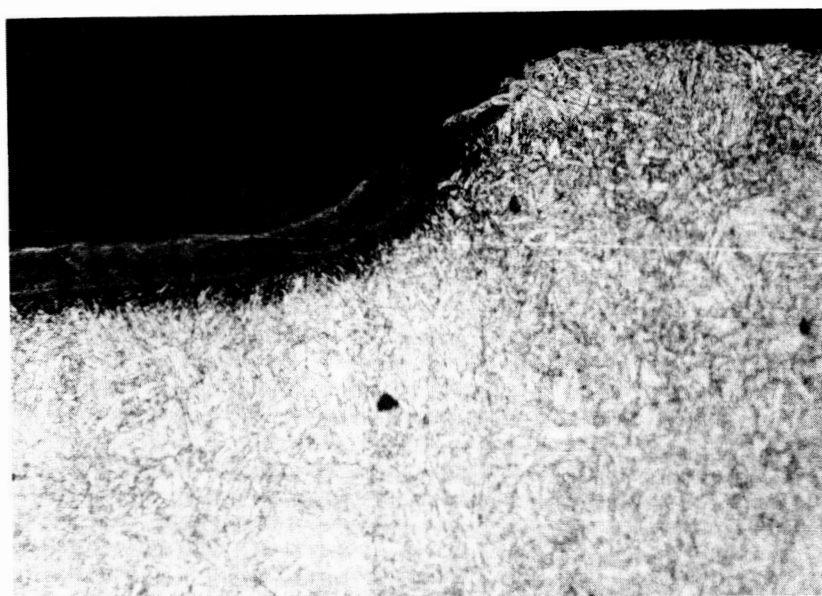


Figure 7. - Cross section of fracture line.

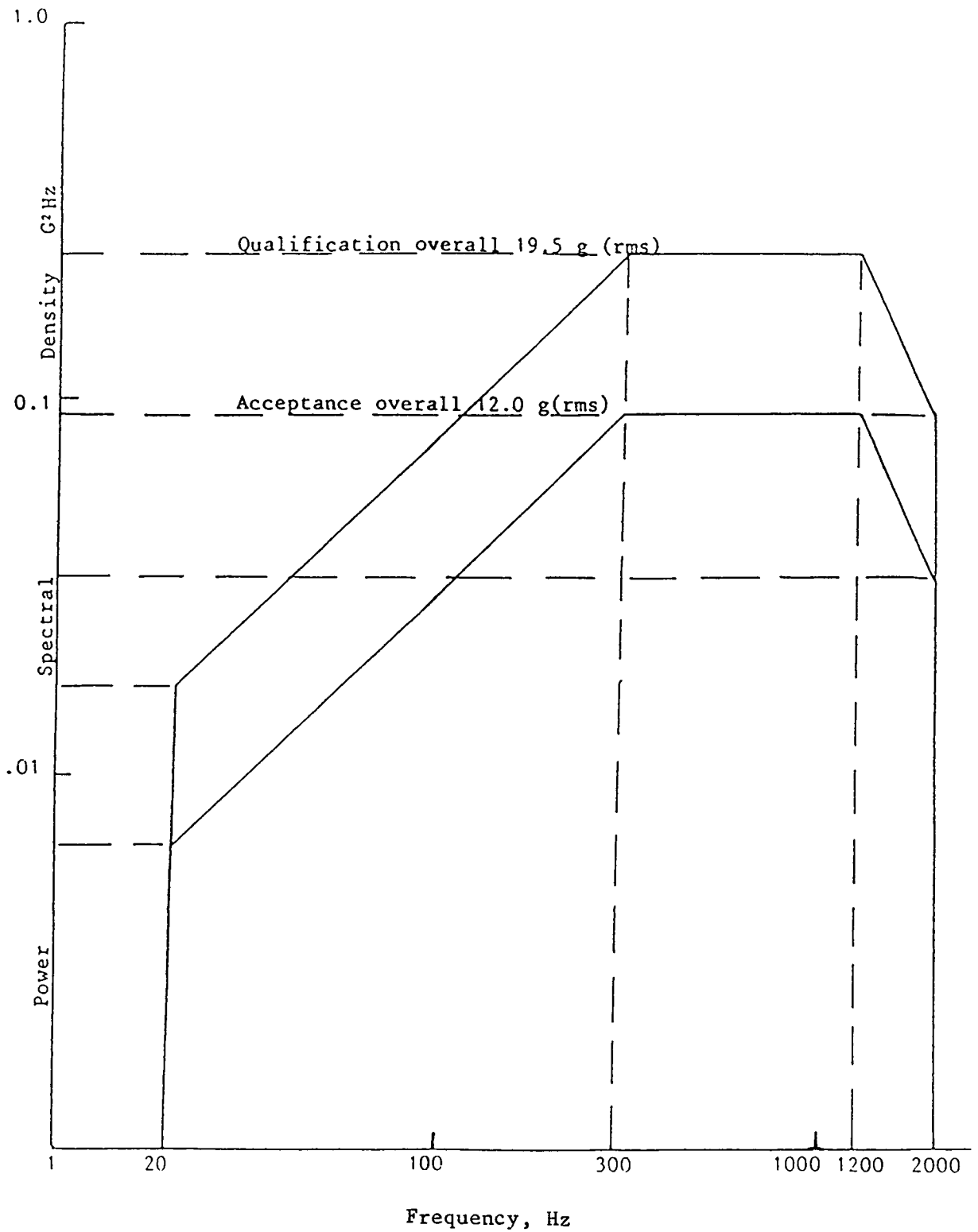


Figure 8. - Random vibration test requirements.