



GS 134-10

SPECIAL PURPOSE GEAR UNITS TO API 613

February 1994

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SCOPE AND PURPOSE This Specification covers requirements for special purpose, enclosed, precision single - and double-helical one - and two-stage gear boxes of parallel shaft design, and is primarily intended to apply to equipment for continuous service without installed spare equipment. It supplements the API standard, defining a number of the optional clauses and substituting, modifying or qualifying certain other clauses in the light of BP experience.											
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FOREWORD

Introduction to BP Group Recommended Practices and Specifications for Engineering

The Introductory Volume contains a series of documents that provide an introduction to the BP Group Recommended Practices and Specifications for Engineering (RPSEs). In particular, the 'General Foreword' sets out the philosophy of the RPSEs. Other documents in the Introductory Volume provide general guidance on using the RPSEs and background information to Engineering Standards in BP. There are also recommendations for specific definitions and requirements.

Value of this Guidance for Specification

The API Standard does not cover Special Service duties. This Guidance for Specification incorporates BP's knowledge and experience into the API standard making it applicable for Special Service duties.

Application

This Guidance for Specification is intended to guide the purchaser in the use or creation of a fit-for-purpose specification for enquiry or purchasing activity.

It is a transparent supplement to [API 613](#) Third Edition, dated April 1988, showing substitutions, qualifications and additions to the API text as necessary. As the titles and numbering of the BP text follow those of API, gaps in the numbering of the BP document may occur. Where clauses are added, the API text numbering has been extended accordingly.

Text in *italics* is Commentary. Commentary provides background information which supports the requirements of the Specification, and may discuss alternative options.

This document may refer to certain local, national or international regulations but the responsibility to ensure compliance with legislation and any other statutory requirements lies with the user. The user should adapt or supplement this document to ensure compliance for the specific application.

Specification Ready for Application

A Specification (BP Spec 134-10) is available which may be suitable for enquiry or purchasing without modification. It is derived from this BP Group Guidance for Specification by retaining the technical body unaltered but omitting all commentary, omitting the data page and inserting a modified Foreword.

Principal Changes from Previous Edition

This specification supplements the Third Edition of [API 613](#).



Feedback and Further Information

Users are invited to feed back any comments and to detail experiences in the application of BP RPSE's, to assist in the process of their continuous improvement.

For feedback and further information, please contact Standards Group, BP International or the Custodian. See Quarterly Status List for contacts.



1. GENERAL

1.1 Scope

The requirements of [API 613](#), Third Edition, April 1988 shall be met, except as amplified and modified herein.

This Specification is for use with an API style data sheet to adapt it for each specific application.

(Addition)

1.2 Alternative Designs

Referenced standards may be replaced by equivalent standards that are internationally or otherwise recognised provided that it can be shown to the satisfaction of the purchaser's professional engineer that they meet or exceed the requirements of the referenced standards.

(Substitution)

1.3 Conflicting Requirements

In case of conflict between various documents, their order of precedence shall be:-

- (a) Local Authority or Statutory Regulations
- (b) The Equipment Requisition or Order
- (c) Data Sheets
- (d) This Specification
- (e) Referenced Industry Standards.

(Substitution)

2. BASIC DESIGN

2.1 General

2.1.3 Noise limits will normally be specified on the data sheet. Otherwise, noise levels at or beyond 1 m from the machine surfaces (driving and driven units, transmission and ancillaries) shall not exceed 85 dB(A).

When the above limits cannot be met without the addition of noise attenuation features, the levels with and without these features shall be stated in the proposal.



Enclosure design shall not unduly compromise operation and maintenance. All instrumentation and controls shall be either mounted externally to the enclosure or clearly visible and controllable from outside the enclosure.

(Substitution)

Generally, gearboxes are noisy, i.e. above 96 dB(A). A large proportion of the noise is generated by dynamic loads due to discontinuities in tooth contact, resulting from the inherent manufacturing inaccuracies.

The manufacturing quality, or accuracy and finish, of the gear together with tooth geometry and pitch line velocity have a strong influence on the noise levels. In general, lowering quality, increasing pitch line velocity and tooth size, will tend to increase the noise levels.

Manufacturers strive to control noise levels by careful design and manufacture of the teeth. Casings are commonly thick walled cast iron which provides good acoustic attenuation. Nevertheless, to meet this requirement an additional acoustic enclosure will probably be required.

2.1.5 Equipment shall be capable of full speed reverse rotation for limited periods of time without damage.

(Addition)

Gears should be compatible with any requirements for reverse rotation capability of the driver and driven equipment.

The precise reverse speed and power will be dependent on the machinery train and the system in which it is installed. Provision of this data will be the responsibility of others. In the absence of this data the provisions of 2.1.5 must be satisfied. Any limitations on speed, load, or duration should be stated by the vendor.

Reverse rotation most commonly occurs following the tripping of a machinery train, when the driven machine can temporarily become the driver. In these circumstances the direction of torque is usually unchanged, the normal gear tooth flanks remain in contact and hence there is little likelihood of problems at the gear teeth in the short term. In the longer term, lubrication or cooling could be compromised, as gear tooth lubrication systems are often directional.

Careful consideration may need to be given to journal bearings, as their design may already be a compromise between load capacity and stability, even in the normal direction of rotation. See 2.7.2 for additional guidance.

Tilting pad thrust bearings with offset pivots have a very limited load capacity in reverse. To cater for significant loads in reverse, central pivots may be necessary although this would somewhat reduce the forward capacity.

2.2 Rating

2.2.5 [AGMA 421.06](#) rating formulae for pitting and bending will be considered as an acceptable alternative to the API method, provided the following conditions are satisfied:-



- (a) Gears are case hardened and ground.
- (b) The vendor can validate the rating method by reference to satisfactory operating experience.
- (c) All loadings, both steady and oscillating, have been catered for. This applies particularly to the abnormal torques that can arise during startup, short circuit, re-acceleration or stalling.
- (d) The vendor can demonstrate that scuffing will not be a problem.

(Addition)

There is a widespread feeling amongst European manufacturers that the API rating formulae are too conservative, even for applications in the petroleum and petrochemical industries. They believe that [AGMA 421.06](#) would be a more relevant standard and many can support this view by reference to good operating experience.

The use of API rating formulae which are primarily concerned with tooth bending and surface pitting will generally lead to gears which are larger, less efficient, and noisier than those to AGMA. This results in higher capital and operating costs, higher operating temperatures and greater environmental disturbance. These effects become more significant as powers and pitch line velocities increase and in the ultimate, bearing and tooth temperatures may become more limiting than tooth bending or surface pitting. In such cases the reduction in sizes permitted by the AGMA rating formulae may be essential to reliability.

Specialist European manufacturers produce case hardened and ground gears. The case hardening (gas carburized or nitrided) increases the bending strength and the resistance to surface pitting and scuffing. Grinding not only improves the surface finish but also facilitates correction of the involute profile at tip or root, and helix for any elastic deformation of the tooth and its supports. Grinding also facilitates end relief or barrelling of the teeth to prevent concentrated end loading.

Nitrided gears have similar bending and pitting resistance to carburized gears. However they have a significantly lower shock load resistance than carburized gears. Therefore particular attention should be paid to nitrided gears when these are proposed for applications where shock loads are likely, e.g. machinery trains with an electric motor or generator. The scuffing resistance of nitrided gears is generally recognised as superior to that of carburized gears.

API and AGMA consider the principal failure modes as bending and pitting. However as powers, speeds and load per unit face width (unit load) increase, a third criteria (scuffing) becomes significant.

Hardened and ground gears permit the use of much higher unit loads than soft gears. Also AGMA permits higher unit loads than API. Therefore the risk of scuffing becomes significant with hardened and ground gears and gears rated to AGMA will be at greater risk than those rated to API.

Wear processes, such as scuffing, are influenced by several factors. These include the geometry of the gear teeth, and the quality of manufacture as well as the power

transmitted and the gear speeds. The type and specification of the lubricant will also affect the gear contact conditions and the susceptibility to wear.

Specialist European manufacturers routinely check their designs for scuffing risk. Figure 1 gives guidance on the transmitted power and speed levels where manufacturer's assessments of the risk of scuffing should be verified, but it is recommended that all gear trains in critical service are assessed.

The short term loading of gears above their rated values is quite common. The overload capability of gears rated to AGMA will be lower than those rated to API; hence if AGMA is used, greater care will be necessary to ensure that all significant abnormal loadings have been identified and can be accommodated.

2.5 Gear Elements

2.5.1 General

When specifying gear sets, the choice between single or double helical gear units will often arise. There may be no overriding reason to select one as opposed to the other, but the following points should be considered:-

Advantages of double helical:-

- (a) The need for a thrust bearing on the high speed pinion line is eliminated.
- (b) Tilting forces on the gears are eliminated. Journal bearing radial loads are equalised with symmetrical bearing and shaft deflections.
- (c) The relatively free choice of helix angle resulting from thrust balance enables the gear designer to have a freer choice of tooth numbers and module whilst retaining the 'hunting tooth combination' for a specific ratio.
- (d) The use of larger helix angles generally results in a smoother and quieter drive for a given gear quality.
- (e) By dividing the total tooth face width into two sections, temperature gradients across the face are reduced.
- (f) Greater tolerance to torsional and bending shaft distortions permit wider uncorrected face widths and hence higher powers for a given gear box.

Disadvantages of double helical:-

- (a) Because of the inherent inaccuracies in gear manufacture, the opposing helices, if extended towards each other, do not coincide on the same circumferential line. In fact they wander from side to side about this line. This so-called 'apex wander' results in axial shuffling of the gear, pinion, or both. API limits this to .0015 inches, which requires gears of a very high accuracy, viz. [AGMA 390.03](#) Quality 14/15 ([DIN 3960](#) Quality 2/3). The ability to meet these stringent requirements, together with experience at the power and pitch line velocity, would need to be checked with any potential vendor.



- (b) *The equilibrium of tangential loads on the flanks is upset if external axial thrust loads are transmitted through the gears. The resultant increase in load on one side could cause overloading and a reduction in life. Hence double helical gears are not suitable for applications where the transmission of significant axial loads is necessary as can arise with some gear, solid, or rubber insert type couplings. The axially flexible metal diaphragm or element type couplings do not normally transmit significant thrust forces; however they must be capable of accepting any axial shuffling from the gear set.*

2.5.1.4 *The scuffing resistance of gears has been found to increase significantly once the gears are 'run in'. In some cases, prior to running-in, the gear may fail due to scuffing (so-called 'initial scuffing' failure).*

For many applications it may not be possible or desirable to follow a prolonged and carefully controlled running-in programme for the gears. In these cases, experience has shown that 'initial scuffing' can be avoided by plating the flanks of the gears with a soft metal, such as copper or silver. Once plated, the gears will then run in at normal loads and speeds without scuffing.

2.5.2 **Quality Assurance**

The quality standard required will generally be to [AGMA 390.03](#), [DIN 3960](#), or [BS 1807](#). The required quality grade will be a function of the pitch line velocity and tooth size for a given gear configuration i.e. single or double helical.

A vendor's recommendations should be justified by the vendor, based on his experience.

2.5.2.4 The manufacturing dimensional accuracy standard shall be stated in the proposal.

(Addition)

2.5.4 **Shafts**

2.5.4.1 Shafts shall be sized to accommodate the maximum unidirectional and oscillatory torques associated with the driving, driven equipment, or both. The design shall cater for, but not be limited to, the torques generated during startup, short circuit, re-acceleration and stall.

(Qualification)

The requirements of API ignore driver stall and starting torques together with any oscillatory effects produced during start-up, short circuits or re-acceleration.

These requirements should be calculated by the main manufacturer and relayed to the gear vendor, so that he may allow for them in evaluating shaft stresses.



2.6 Dynamics

2.6.1 Critical Speeds

2.6.1.4 Critical speeds, as determined by the fluid film bearings, depend on the load and speed. If the presence of a critical speed within the speed ranges (API) is unavoidable, then reference to the vendors experience is required. If the vendor can illustrate that the response at the critical speed is heavily damped and if time spent at the particular load and speed is likely to be small then the critical speed should be accepted.

2.6.2 Balance

2.6.2.1 The maximum allowable residual unbalance in the plane of each journal shall satisfy [ISO 1940](#) : G.1.0.

Balancing procedures shall be such that rotor and coupling interchangeability can be achieved without the need for re-balancing. This will require rotors to be first balanced without couplings and then to be check balanced with coupling hubs mounted.

(Substitution)

The API requirement results in very low values of permissible out-of-balance at the high speeds that may be found with pinion gears. Such pinions are light in weight and the principal loads on them and their bearings result from gear tooth reactions, not out of balance. The [ISO 1940](#) criteria is more logical. A comparison is shown in Figure 2.

2.7 Bearings

2.7.2 Radial Bearings

The journal bearings of gearboxes operate under far higher loads than those of compressors, turbines and pumps. Hence it is important that their selection be given closer consideration.

Bearing operation must be stable. As one progresses away from plain cylindrical through elliptical and multi-lobe to tilting pad bearings, stability increases but load capacity, damping and tolerance to reverse rotation and reverse torque decrease.

In general, 'stiff' shafts such as pinions or gear wheels should not require the highest stability of a tilting pad bearing, yet a plain cylindrical bearing might well have insufficient stability at light loads.

The manufacturer should show, with test data, calculations or experience, that the bearings can safely carry full loads, will be stable when unloaded and can accommodate any reverse rotation and reverse torque requirements.

2.7.2.2 The bearing-metal temperature shall not exceed 105°C.

(Addition)

2.7.3 Thrust Bearings



- 2.7.3.1 The vendor may offer a double helical design without a low speed thrust bearing, as specified in [API 613](#), Appendix C.

A thrust collar at the Pitch Circle Diameter of a single helical gear may be used if supported with experience at similar loads and speeds.

(Qualification)

Brown Boveri has designed and licensed to certain manufacturers a device known as a thrust collar. Thrust is absorbed by collars fixed on either side of the pinion tooth flanks, bearing on a mating face machined on the side of the gear wheel. This enables a single helical gear to be designed such that it exerts no external axial thrust thus eliminating the need for an external thrust bearing.

This device has been used extensively on Sulzer compressor trains where it forms part of a special Sulzer shaft/coupling system (including successful applications for BP). It has also been applied in conventional shaft/coupling systems by the various licensees. It may prove attractive for single helical applications where the reduced losses associated with this device could be attractive (lower power, smaller oil system etc.).

The vendor would need to produce data on the application to show it was well within his experience and test envelope. In particular the comparative sliding velocities and Hertzian stresses should be reviewed.

The thrust loads are minimal when flexible-element couplings are used in conjunction with double helical gears. Consequently, a taper land thrust bearing for the low speed shaft of a double helical design is acceptable. This results in a cheap and simple configuration.

- 2.7.3.2 Integral thrust collars shall be designed so that they may, if necessary, be replaced by a separate shrunk-on thrust collar.

(Addition)

The intent is to be able to reclaim an expensive gear set following a thrust failure(s) which has eliminated the collar thickness margin. This requirement is desirable but should not be insisted upon if it cannot be incorporated in a manufacturer's standard design.

- 2.7.3.3 The bearings shall be selected at 50% of the maximum load capacity for continuous operation under all specified conditions. 100% of the maximum load capacity shall be the smaller of the following loads:-

- (a) That to produce the minimum oil thickness without inducing failure in continuous operation.
- (b) That to produce the maximum permitted bearing lining temperature without inducing failure in continuous operation.
- (c) The fatigue or mechanical limit for the bearing or its lining material.



(Substitution)

The API selections using ultimate load could be misleading, particularly with large bearings. The BP paragraph represents a more rational basis for thrust bearing selection, taking account of the important criteria such as oil film thickness and pad lining material temperature.

The oil film thickness may be more limiting than the fatigue or mechanical limit at low speeds. The lining material temperature may be more limiting than the fatigue or mechanical limit at high speeds.

Generally, the fatigue or mechanical limit specific loadings as set by European manufacturers will be higher than the corresponding API value. This is usually acceptable in the context of a gearbox for which the normal loads can be predicted with some confidence, unlike the situation with thrust load within most turbomachinery.

2.8 Lubrication

Oil systems are often integrated to supply the needs of the whole machinery package.

3. ACCESSORIES

3.2 Couplings & Guards

3.2.1 *API implies removable hubs. If the vendor standard is an integral hub, these should be used, provided it does not compromise replacement of parts and maintenance.*

3.2.2 *Removable coupling hubs shall be non-keyed, tapered bore and hydraulically fitted.*

(Addition)

3.4.2 Instrumentation

The use of journal and thrust bearing metal temperature sensors is a more precise, sensitive and responsive means of bearing monitoring than bulk drain temperature sensors. The need for metal temperature detectors should be reviewed in relation to the application i.e. importance of the duty and the bearings' limiting load criteria

Typically for a low power (say up to 1500 kW) or physically small gearbox, where such temperatures are modest, the vendor's standard instrumentation is acceptable.

For a medium power (say up to 7500 kW) and medium size gearbox, where such temperatures are significant, consideration should be given to fitting bearing metal temperature sensors to the pinion journal and thrust bearings of single helical gears.

For large power units (above 7500 kW) on critical duties where the bearing rating is limited by metal temperatures, bearing metal temperature detectors should be fitted to pinion journal and thrust bearings of single helical gears. Also, fitting metal temperature detectors on wheel journal and thrust bearings should be considered.



3.4.3 Vibration and Position Detectors

3.4.3.1 *The need for X-Y probes on a gearbox should be reviewed in relation to the application i.e. importance of duty, power, speed and size.*

Typically for a low power (say up to 1500 kW) or physically small gearbox on sites where portable vibration monitoring equipment is regularly used, no permanent equipment need be fitted. Alternatively it may be more convenient to have a permanent monitor, in which case a casing mounted accelerometer(s) with local monitor would suffice.

For a medium power (say up to 7500 kW) and medium size gearbox where on-line vibration monitoring is required, a casing mounted accelerometer(s) with vibration probes fitted adjacent to couplings (when practical) would suffice. In addition, an axial position probe(s) on the pinion line of single helical gears should be considered.

For large power units (above 7500 kW) on critical duties, casing accelerometer(s), vibration probes on all bearings and axial position indicators adjacent to thrust bearings would be recommended.

Casing mounted accelerometers are commended as the means to monitor the condition of the tooth mesh. The tooth mesh frequencies are generally high, hence proximity type monitors cannot detect tooth mesh deterioration because of their poor response at high frequencies.

4. INSPECTION, TESTING & PREPARATION FOR SHIPPING

4.1 General

4.1.3.3 *The following list is the recommended extent of participation in inspection and testing by the purchaser:-*

- (a) Mechanical running tests.*
- (b) Full load running tests.*
- (c) Sound level tests.*
- (d) Post test inspection of components.*
- (e) Any proposed repairs.*
- (f) Cleanliness.*

4.1.4 The vendor shall operate a quality system to ensure that the technical requirements of this Specification are achieved. The vendor shall be prepared to demonstrate the quality system to the purchaser.

(Addition)



4.3.3

Optional Tests

The specification of any of these optional tests will depend upon the circumstances of the particular contract, e.g.:-

- (a) Criticality of the duty.*
- (b) Gearbox rating in relation to vendor's experience or limiting criteria such as scuffing.*
- (c) Whether a full load string test is to be made in the main manufacturer's works.*
- (d) Availability of a sufficiently large loading device.*
- (e) Estimated pinion bending and torsional distortion.*
- (f) Availability of an identical second unit.*
- (g) Cost of test.*

4.3.3.1

Full-Speed/Full- or Part-Load Test

Full load tests are always desirable but are usually only possible by means of a back-to-back test (See API) and this implies the availability of a second identical or similar unit. String tests of complete machinery trains may provide some reassurance but such tests rarely achieve full load.

4.3.3.2

Full-Torque/Slow-Roll Test

This test may be valuable as a compromise where a full load trial cannot be made. It would be useful for those cases where the elastic deflections of the pinion and teeth have necessitated significant profile and helix corrections. The full torque will reproduce these deflections due to the elasticity of the gears and casing. However, it would not create the deflections due to oil film flexibility nor any effects due to temperature.

The possibility of wear of the bearings during the proposed slow roll, needs to be considered. The duration should be fixed through discussion with the manufacturer. Agreement will have to be reached on criteria for acceptable contact patterns. The recommendation of Lloyds Register of Shipping Rules, BS 1807 or American Bureau of Shipping are suggested.

4.3.3.3

Full-Torque/Static Test

The slow roll test will give an indication of average contact condition resulting from successive contact between pinion and wheel teeth.

The static test will be somewhat less satisfactory since contact patterns will only be established if the teeth are rocked backwards and forwards. In any case these will only represent random tooth to tooth patterns.

This type of test should only be adopted if the slow roll test cannot be performed.



4.3.3.4 Back-to-Back Locked Torque Test

This is the most practical form of testing for full load trials on gearboxes. However it is only possible when a second identical unit is available.

The pinion and wheel shafts of gearboxes are coupled together and the full load torque is applied by twisting, then bolting the shafts together with the gearbox stationary. Thus full tooth and bearing loads are applied during starting and the design must be checked to ensure that bearing or tooth damage will not occur as a result. The possibility of employing on-line torquing should be investigated. One technique uses a modified epicyclic gearbox for this purpose, and others a hydraulic device.

5. VENDOR'S DATA

5.1 Proposals

5.1.1 The following information is required in addition to that listed in API 613:-

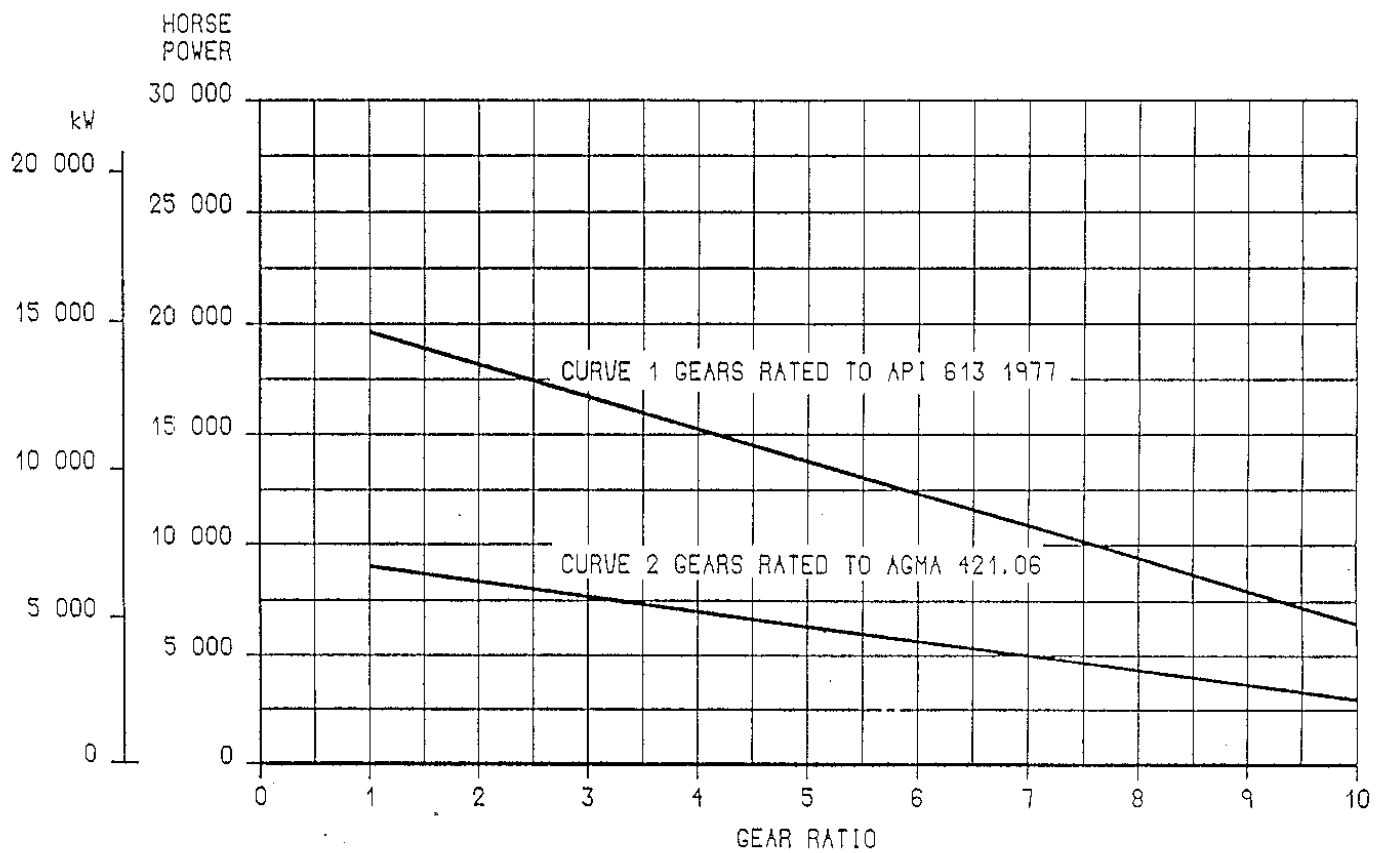
- (p) Noise levels with and without attenuation features, where applicable, as required by 2.1.3.
- (q) Details of rating methods and supporting data as required by 2.2.5.
- (r) A statement of the proposed dimensional accuracy standard.

(Addition)





FIGURE 1
MAXIMUM HORSEPOWER BEFORE SCUFFING v GEAR RATIO



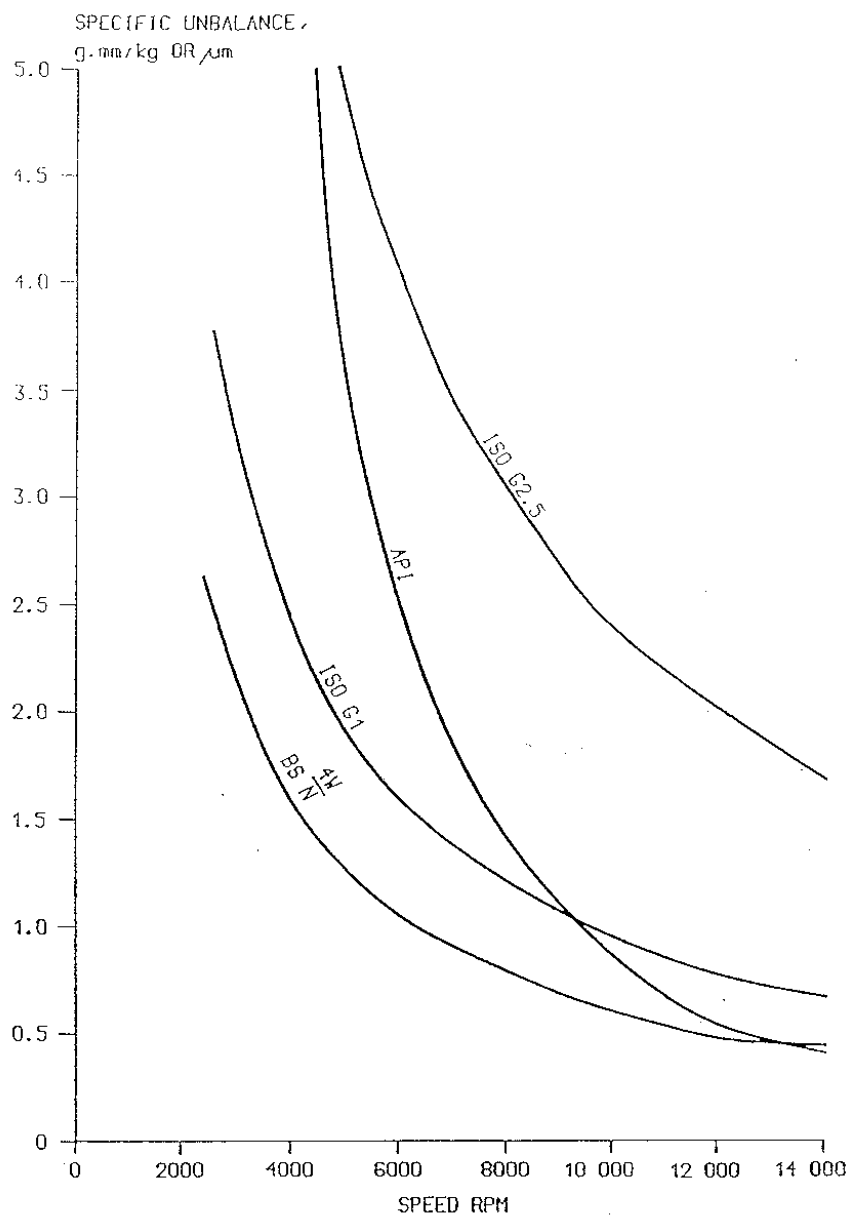


FIGURE 2

**COMPARISONS OF PERMITTED OUT OF BALANCE VALUES v SPEED FOR
VARIOUS RECOGNISED STANDARD AUTHORITIES**



DEFINITIONS AND ABBREVIATIONS**Definitions**

Standardised definitions may be found in the BP Group RPSEs Introductory Volume.

Abbreviations

AGMA	American Gear Manufacturers Association
API	American Petroleum Institute
BS	British Standard
ISO	International Organisation for Standardisation.



LIST OF REFERENCED DOCUMENTS

A reference invokes the latest published issue or amendment unless stated otherwise.

Referenced standards may be replaced by equivalent standards that are internationally or otherwise recognised provided that it can be shown to the satisfaction of the purchaser's professional engineer that they meet or exceed the requirements of the referenced standards.

API 613 3rd Edition April 1988	Special-Purpose Gear Units for Refinery Services.
AGMA 390.03	<i>Gear Handbook Gear Classification, Materials and Measuring Methods for Bevel, Hypoid, Fine Pitch Wormgearing and Racks only as Unassembled Gears.</i>
AGMA 421.06	Practice for High Speed Helical and Herring-Bone Gear Units.
BS 1807:1981	<i>Specification for Marine Main Propulsion Gears and Similar Drives</i>
DIN 3960 : 1987	<i>Definitions, Parameters and Equations For Involute Cylindrical Gears and Gear Pairs.</i>
ISO 1940-1	Mechanical Vibration - Balance Quality Requirements of Rigid Motors - Part 1: Determination and Permissible Residual Unbalance.

