



GS 134-7

SPECIAL PURPOSE STEAM TURBINES TO API 612

November 1994

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SCOPE AND PURPOSE This Guidance for Specification covers requirements for Special Purpose Steam Turbines. It is for use with a data sheet to adapt it for specific application. 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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CONTENTS

Section	Page
FOREWORD	i
1. INTRODUCTION.....	1
1.1 Scope	1
1.2 Alternative Designs.....	1
1.3 Conflicting Requirements.....	1
2. BASIC DESIGN	1
2.1 General.....	1
2.2 Pressure Casing	5
2.3 Casing Appurtenances	5
2.3.1 Nozzles and Diaphragms.....	5
2.3.2 Sentinel Warning Valve.....	6
2.4 Casing Connections	6
2.4.1 See 2.2.10 and 11.	6
2.5 External Forces and Moments.....	6
2.6 Rotating Elements	7
2.6.1 Rotors.....	7
2.6.2 Shafts.....	8
2.6.3 Blading	8
2.7 Shaft Seals.....	10
2.8 Dynamics.....	10
2.8.2 Lateral Analysis	11
2.8.4 Torsional Analysis.....	13
2.8.5 Vibration and Balancing.....	14
2.9 Bearings and Bearing Housings.....	14
2.9.1 Radial Bearings.....	14
2.9.2 Thrust Bearings and Collars	14
2.9.3 Bearing Housings.....	16
2.9.4 Grounding.....	17
2.10 Lubrication and Control Oil Systems.....	17
2.11 Materials.....	18
2.11.1 General	18
2.11.3 Welding	18
3. ACCESSORIES	18
3.1 Couplings and Guards.....	18
3.2 Mounting Plates.....	20
3.2.2 Baseplates.....	20
3.3 Gear Units	21
3.4 Controls and Instrumentation.....	21
3.4.1 General	21



3.4.2	Control Systems.....	21
3.4.3	Protective Devices	22
3.4.4	Instrument and Control Panels	23
3.4.8	Vibration, Position and Bearing Temperature Detectors	24
3.7	Insulation and Jacketing.....	24
3.9	Turning Gear	24
4.	INSPECTION, TESTING AND PREPARATION FOR SHIPMENT.....	25
4.1	General.....	25
4.2	Inspection.....	26
4.2.1	General	26
4.2.2	Material Inspection	27
4.2.3	Mechanical Inspection.....	28
4.3	Testing.....	28
4.3.1	General	28
4.3.3	Mechanical Running Test.....	28
4.3.4	Optional Tests.....	29
5.	VENDOR'S DATA.....	30
5.1	Proposals.....	30
5.2	Contract Data	30
5.2.5	Data.....	30
APPENDIX A.....		32
	CHECK LIST TO API 612 THIRD EDITION - DATA SHEETS.....	32
APPENDIX F		37
	DEFINITIONS AND ABBREVIATIONS	37
APPENDIX G.....		38
	LIST OF REFERENCED DOCUMENTS.....	38



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

This Guidance for Specification defines a number of the optional API clauses and may substitute, add to or qualify other API clauses using BP's knowledge and experience worldwide.

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 612](#) Third Edition, dated November 1987, 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 134-7) 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.



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. INTRODUCTION

1.1 Scope

This document specifies BP requirements for special purpose steam turbines conforming generally to [API 612](#). They shall meet the requirements of [API 612](#), third edition, November 1987, (adopted as [ANSI/API Std 612-1992](#)) as amplified and modified herein.

(Qualification)

1.2 Alternative Designs

Designs alternative to those prescribed will be acceptable provided it can be shown to the satisfaction of the purchaser's professional engineer that the required performance and function are attained.

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

- 2.1.1 Turbines shall be sufficiently robust to withstand without damage the results of occasional abnormal conditions of limited duration, such as surging of compressors, or electrical faults on generators.



(Addition)

Normally turbines to [API 612](#) are for use on continuous running duties. The number of starts experienced during the machine lifetime is therefore relatively small. Starting procedures designed to minimise casing transient thermal stresses are specified by the vendor, therefore casing life is not critical.

However, it is essential to validate these assumptions on specific applications.

2.1.3(b)

[API 612](#) acknowledges that careful consideration needs to be given to the specification of the minimum inlet steam conditions and coincident maximum exhaust conditions, to prevent oversizing of the turbine.

Oversizing can lead to reduced efficiency at the normal operating points, high governing stage stresses at the first control valve point, and possible overloading of blading.

For the governing stage, tangential steam loading and blade heights increase when a reduced nozzle arc is used for normal operation, i.e., when the size of the inlet arc has been increased to accommodate off-design steam conditions.

An increase in tangential load or blade height increases the bending stress.

The tangential load increases on two counts:-

- (i) The number of moving blades exposed to the nozzle arc reduces with reducing arc and hence the load per blade increases.
- (ii) The energy drop across the stage increases as the percentage arc reduces. This happens because the wheel bowl pressure reduces with reducing flow to the wheel bowl.

The blade height increases to compensate for the reduced nozzle arc.

Increased nozzle area permits increased flow to the turbine. Flow will increase in response to increased load or the same load at reduced conditions. Both situations should be reviewed to ensure the blading design is adequate, or any operating limits identified, and appropriate safeguards provided.

Machine costs and complexity can increase if 360 degree arc inlets or by-pass control become necessary to meet these conditions.

A 40% increase in nozzle area over rated duty at normal conditions can generally be accommodated without compromising efficiency, strength, or costs.

Some vendors (European principally) offer by-pass governing to accommodate these requirements.

2.1.3(d)

The performance of condensers new and clean will be better than the design which will be based on the fouled condition. When a condenser with a divided water box is to be available for on-line cleaning or operated with only half the condenser in



service, the turbine duty may have to take this operating condition into account. On-line cleaning methods have gone some way towards reducing the variations in condenser performance over the required period of uninterrupted operation.

Condensing turbines exhausting to condensers which have divided water boxes for in-service cleaning may generate lower pressures when the full surface is in service.

Also, condenser pressure may reduce with falling cooling water temperature.

Lower pressures may lead to overstressing, and increased water erosion of exhaust blading unless the vacuum is controlled by judicious condenser sizing or by bleeding air, or by adjusting the operation of the condenser vacuum equipment.

2.1.3(e) Operation of a compressor at high suction pressure can lead to increased power consumption and torques. The oversizing implicit in 2.1.3(b) may then lead to overstressing of blading.

2.1.3(f) The sizing of the control nozzles for the condensing sections of extraction or induction turbines requires attention to avoid the possibility of overloading the exhaust blading. The sizing of these nozzles should be reviewed with the vendor.

The maximum operating value of the extraction or induction pressure should be established. This, together with the nozzle areas, will determine the maximum possible flow and hence loading on the blading. This needs to be considered in condensing section blade stressing, and any operating limits identified and appropriate safeguards provided.

2.1.11 Noise limits will normally be specified in detail in an enquiry. However, in the absence of such requirements, noise levels at or beyond 1 m (3 ft) from the machine (driver plus driven machine, transmission and auxiliaries) surfaces should not exceed 85 dB(A).

The 85 dB(A) limit is based upon an 8 hour exposure. Where operating shifts exceed 8 hours and maintenance activity may be required in the area of an operating turbine the limit may require to be reduced to 83 dB(A).

The sound intensity method for measuring the noise level of equipment offers significant advantages over conventional sound pressure measurement techniques. These are:-

- (a) Measurement of sound radiated from each surface or area of the equipment. This enables the principal contributors to overall noise level to be identified and reduced by locally applied absorption materials.
- (b) Improved compensation for background and reverberative effects.

When a vendor cannot meet the limits without the addition of noise attenuation features, the levels with and without these features shall be stated in his proposal. Noise attenuation measures proposed by the vendor shall not conflict with the other requirements of this Standard.

A vendor shall provide details of the noise emission in octave bands from his equipment, obtained by the methods of test specified in



EEMUA Publication 140 or a method of test approved by BP. A vendor shall also provide details of any narrow-band noise emitted by his equipment which is noticeable to the ear, and the octave band or bands in which it occurs.

Noise attenuating enclosures will only be accepted when there is no alternative form of noise control. The design of these enclosures shall be such that normal operation and maintenance are not unduly compromised. All instrumentation and controls shall be either mounted externally to the enclosure or shall be clearly visible and controllable from outside the enclosure.

Enclosures shall be adequately purged and cooled. Instrumentation, sensors and cables installed inside enclosures shall not be subjected to an environment which causes the component to be operated outside the vendor's specified ambient temperature limits.

Where the area classification would otherwise be Zone 2, equipment within an enclosure will normally need to be suitable for a Zone 1 area.

(Substitution)

- 2.1.15 For electric power generation applications, the turbine vendor shall be responsible for the co-ordination of the design and for the satisfactory functioning of the complete unit, i.e. turbine, generator, transmission and ancillaries. In cases where the turbine vendor supplies equipment that they have not manufactured, they shall be responsible for ensuring that the designs of these items are compatible with each other and with their own equipment in all respects.

The satisfactory functioning of the complete unit shall form part of the turbine vendor's contractual guarantee.

For other applications, the driven equipment vendor will normally be responsible for overall co-ordination. The turbine vendor shall co-operate with the driven equipment vendor to facilitate the development of the complete unit.

(Substitution)

The responsibilities defined in this Specification are generally valid. However, the allocation of overall responsibility for the train may vary on a case to case basis depending upon the application, technical complexity and relative value of the driving and driven equipment.

The turbine vendors responsibility should in any case include the condenser and its auxiliaries.

2.2 Pressure Casing

- 2.2.3 For turbines with dedicated condensers, a relief valve mounted on the condenser shell is required, as part of the condenser vendor supply. (Sized to H.E.I. requirements).



For other applications ensure that relief valves are included in the lines upstream of block valves.

- 2.2.9 Each drain shall be individually valved and piped to the edge of the turbine base. Valves shall be accessible from the operating floor.

(Addition)

- 2.2.10 & 11 *Studded connections on casing nozzles hinder installation and maintenance since piping has to be sprung in order to remove casings or pipe spools.*

Above circa 413°C bolting is subject to creep stress relaxation and may require periodic replacement.

During dismantling if a nut or tapped thread binds, replacement of studs becomes problematic compared to through bolting, which can be sawn or burnt off.

- 2.2.12 Bolt tensioning systems using electric heaters should preferably be avoided on machines to be installed in hazardous areas.

(Addition)

Bolting, particularly at the main casing horizontal joint may be large; spanner tightening becomes impractical and hydraulic stretching or electric heating becomes essential.

Hydraulic tensioning is preferred to electric heating in hazardous areas. However, vendor standard designs may not have access for hydraulic systems, and electric heating may have to be accepted subject to working with a hot work permit.

2.3 Casing Appurtenances

2.3.1 Nozzles and Diaphragms

- 2.3.1.1 *The need to replace nozzles arises from:-*

(a) *Abrasive wear on high pressure drop governing stages (particularly part load nozzle banks).*

(b) *Need to re-rate turbines.*

Abrasive wear may be caused by passage of debris (pipe scale, weld spatter and slag, shot etc.) or wet steam. The effects can be minimised by careful pre-commissioning and pipe descaling. The use of in-line vortex separators can remove debris and free water.

Non-welded nozzles can be replaced in the field. Nozzles welded to a casing will require removal by machining.



Non-welded nozzles may experience steam cutting where they are located in casing grooves, especially on high pressure drop or wet applications. Vendor experience needs to be reviewed where a deviation is requested.

Where nozzles are mounted in nozzle belts which can be removed piece-small to a local workshop, then the welded type may be acceptable. In general this arrangement is used on high pressure (Class 900 plus) applications, where pressure drop and hence need for welding is greatest.

2.3.1.2 *Impulse turbines employ diaphragm nozzle blade carriers set into individual grooves in the main casing.*

Generally, reaction turbines employ a stationary blade carrier in the form of an inner casing. The blades are set into grooves in this casing. However, some reaction designs do employ a diaphragm style nozzle carrier, which is still housed in a carrier.

2.3.2 **Sentinel Warning Valve**

Sentinel warning valves shall be supplied.

(Qualification)

Sentinel warning valves may be considered of use only where operators are consistently within audible distance. If this is not the case they may not be specified.

For non condensing steam turbines the exhaust temperature may be higher than the temperature limit for some brasses and bronzes. The valve material must be suitable for the maximum steam temperature.

2.4 **Casing Connections**

2.4.1 *See 2.2.10 and 11.*

2.4.6.4 The vendor shall furnish mating parts of connections larger than those covered by ANSI.

(Qualification)

2.5 *External Forces and Moments*

The pipe force and moment limits permitted by [API 612](#) are lower than for other types of machinery (pumps and compressors).

Steam turbines have bearing housings which support the steam casing, whereas with pumps and compressors the casing supports the bearing housing.

As a consequence, the steam turbine rotor to stator alignment is more sensitive to pipe loadings. Steam pipe and casing thermal growth may be an order of magnitude greater than on pumps and compressors. Steam pipe designs need to be flexible and the positions of anchors, guides and spring hangers carefully considered.



The use of twin connections can aid by balancing pipe loads, and may be justified on larger machines.

Passout, backpressure or induction lines may be more difficult to design because of their greater size and stiffness. Here again, twinning of pipes may be justified.

The design of piping systems should be carefully reviewed with the contractor and vendor.

If after an exhaustive investigation of pipe runs, dual connections, anchor supports, cold draw etc. it proves impossible to achieve mutually acceptable pipe loadings, then on low pressure systems (ANSI 30 0lbs.) the use of flexible bellows may be considered. It is anticipated that this concession will only need to be applied to low pressure systems.

2.6 Rotating Elements

2.6.1 Rotors

2.6.1.3 Built-up rotors are most common in American designed turbines. European designs only use built-up rotors where size precludes the use of solid forged rotors (800/1000 MW).

Potential difficulties with built up rotors are:-

- (a) Stress raisers due to keys, and/or shrink fits. The latter can be very severe.*
- (b) Discs loosening due to centrifugal and/or thermal gradients, particularly during run-up.*
- (c) Introduction of cross-coupling from shrink fits contributing to sub-synchronous whirl.*
- (d) Fretting at the shrink interface.*
- (e) Thermal bending of rotors caused by differential heat transfer at shrink interfaces.*

It is essential to ensure adequate interference fits to eliminate looseness and fretting, but not so severe as to induce rotor instability, or high stress concentrations.

The number and rate of start-up influences thermal gradient loosening.

This type of rotor should only be accepted where there are strong economic incentives and existing designs, (identical steam conditions, power, speeds) with 5 or more years satisfactory operating experience.

2.6.1.5 For machines operating above the first flexible mode, balancing planes shall be provided between bearings to allow field balancing after installation. Provision shall be made for ready external access (not requiring removal of the case) at the balancing planes.



(Substitution)

In-place balancing offers significant advantages in reducing plant downtime, and provision should always be specified.

A minimum of two planes is required. Ideally these should be outside the casings; however, rotor dynamics may dictate that they are within the casings.

The need for, and location of other planes within the casing depends upon the rotor mode shape(s) in the operating speed range. These should be studied by forced response calculations.

In general, the first and second flexural modes can be two plane balanced, higher modes may require additional planes.

2.6.2 Shafts

- 2.6.2.7 The turbine vendor shall, jointly with the driven equipment vendor, establish the maximum transient torques that will occur in the shafting system under startup, running, and fault conditions. All components, including the coupling, and the fit of the coupling hub on the shaft, shall be suitable for at least 115% of this figure. See 2.8.4.1 and 2.8.4.2.

(Addition)

2.6.3 Blading

- 2.6.3.1 For each blade row, the vendor shall verify by Campbell diagrams or their equivalent (corrected to actual operating temperatures and speeds) that excitation of in-phase tangential, out-of-phase tangential, axial, torsional, and any other high-response modes by multiples of up to 15 times running speed, by nozzle passing frequency, and by twice nozzle passing frequency does not occur within the range 90% minimum speed to 110% maximum continuous speed.

(Modification)

The origin of blade natural frequency data should be reviewed. Frequently this is calculated from test results, and as such subject to considerable scatter. This scatter leads to the probability of resonance, particularly at high engine orders.

At low engine orders (2-5) it may be possible in theory to avoid resonances. However, this will probably require static frequency checking of individual blades to control scatter to acceptable levels.

In general therefore it is safest to assume that resonance will occur at the closest lower engine order in the case of long blades and between the nozzle passing frequency or twice nozzle passing frequency and the closest natural vibration mode in the case of short blades. The vendor should demonstrate that in these circumstances the vibration stresses are safe.



Vendors will employ their own evaluation techniques, data for which may be proprietary.

The methodology employed by the vendor should be reviewed to ensure that as a minimum it takes account of the following:-

- (a) Steady steam bending stresses at the most severe operating condition.
- (b) The vibration mode and form of shrouding or lacing employed e.g. free standing, integral shroud free standing, integral shroud firm contact, riveted shroud (No. of shroud or lacing packets and blades per packet).
- (c) The effects of irregularities in the steam flow created by blade wakes and other non-uniformities such as:-
 - (i) Variations in blade to blade pitch
 - (ii) Variations in axial location of blade trailing edges
 - (iii) Variations in pressure caused by control valve or nozzle groupings, extraction flows, ribs up or downstream in the flow path and non-uniform velocities in exhaust casings
 - (iv) Variation in energy resulting from the use of nozzle groupings and by-pass overload control
- (d) Damping at the resonant condition.
- (e) Stress concentrations, particularly at the junction of aerofoil and root platform.
- (f) Operation in the transition zone.

The transition point is not fixed on the turbine expansion line. It will vary with steam inlet temperature and the flow rate to the LP section, particularly for extraction turbines. This creates a 'zone' in which the transition point can occur.

The possibility of the presence of damaging chlorides at the transition point makes it essential that all stages which may fall into the transition zone are designed to be safe in a chloride atmosphere.

The turbine expansion line should be drawn for the range of powers and inlet steam conditions specified.

2.6.3.3 Blading that could be in the transition zone between the dry and wet regions under any operating conditions shall be conservatively designed with low working stresses. It shall be assumed that chlorides could be present, and the fatigue strength of blading materials shall be selected accordingly. See 2.11.1.6.

(Addition)

See commentary 2.6.3.1(f).



2.6.3.4 All blading shall have been proven in service, and the vendor shall demonstrate satisfactory operating experience with each blade profile and height.

The vendor shall state whether the response frequencies have been substantiated by static or rotating tests.

(Addition)

2.6.3.5 The vendor shall advise any operating constraints imposed by blading e.g. limits on extraction, induction and exhaust pressures, temperatures and flowrates.

(Addition)

2.7 Shaft Seals

2.7.1 *Outer labyrinths of the spring loaded carbon packing type have the advantage of close clearance running and hence are able to reduce steam leakage to atmosphere. This reduces the risk of steam/condensate escaping to atmosphere and possible ingress into bearing housings (see 2.7.5 of [API 612](#)). However, they are subject to wear which may eliminate this advantage.*

Carbon ring packings are generally limited to rubbing speeds up to 44 m/sec.

2.8 Dynamics

[API 612](#) paragraphs 2.8.1, 2.8.2 and 2.8.3 and Appendix C comprehensively cover the requirements for analysis of rotor lateral critical speeds.

The procedure improves on earlier API requirements by relating separation margins to amplification factors, acceptable amplitudes to internal clearance and test bay verification of calculation assumptions.

However, the testing will be time consuming, and final verification is left until the end of the design manufacturing cycle when the scope for remedial action is limited.

It is essential to review the source and quality of vendor data and correlations between calculation and test from previous jobs.

In general, flexural modes with nodes close to the bearings are lightly damped with the critical speed heavily dependent on accurate stiffness predictions. Flexural modes with nodes remote from the bearings are likely to be more heavily damped with a satisfactory result dependent on accurate damping predictions.

Bouncing or conical modes entail significant movement at the bearings and are therefore normally heavily damped and dependent on accurate damping predictions.

Reaction turbine rotors, because of their drum design, tend to be stiffer than impulse turbine rotors. As a result their flexural critical speeds are higher than those of impulse turbines.



For power generation duties the first flexural critical speed of reaction turbines will be above running speed, with the heavily damped bouncing and conical modes below running speed. Although some instances exist in old designs where these modes are at running speed; no problems have been reported. The stiffer design is also less sensitive to thermal bending with consequent easier run-up characteristics.

The more flexible impulse turbine rotor runs above its first critical speed and below or at its second critical speed, although some impulse turbines in the 1-4 MW range have 'stiff' rotors.

For many mechanical drive applications these comments are also valid. For higher power and higher speed condensing and passout condensing applications, reaction turbine rotors are designed to run between their first and second flexural critical speeds. The equivalent impulse design may operate above its second critical speed and possibly close to its third.

The greater stiffness of the reaction turbine shaft gives it greater resistance to cross coupling. Hence, bearings with poorer cross coupling and better damping characteristics can be employed. (See 2.8.2.8 below).

In summary, it can be stated that reaction turbines because of their stiffer rotors present an easier rotor dynamics design problem than impulse turbines.

2.8.2 Lateral Analysis

2.8.2.3 *A train lateral analysis is required when the shafts of elements of the equipment train are solidly coupled.*

The effects of misalignment on bearing loads, stiffness and critical speeds must be evaluated.

The effects of foundation settlement, solar heating, pipe loading, alignment errors etc. need to be reviewed and evaluated.

2.8.2.4b *The units of the SI version of equation (1) are incorrectly stated. They should be micrometers not millimetres.*

2.8.2.4e *The vendor shall supply a stiffness map as specified in API 612.*

(Qualification)

2.8.2.8 *The vendor shall perform a stability analysis to demonstrate that self-excited vibration will not occur under any loading condition. Assumptions regarding the magnitude of excitations from oil film, aerodynamic and other sources shall be stated.*

(Addition)

Self-excited vibrations result from cross coupling forces applied to the rotor. These originate in:-



- (a) The journal bearings
- (b) Shaft labyrinths
- (c) Blade tips
- (d) Shrink fits
- (b) and (c) are density dependent, the effects therefore increase with steam pressure.

They are also a function of the tangential velocity of the steam in the close clearances of labyrinths and shrouded blade tips. The effects can be significantly reduced by destroying the tangential components of velocity at entry to the clearance space. This is done with radial vanes fitted at the entrance.

The measure of sensitivity to instability is the logarithmic decrement of vibration amplitude. This increases with rotor stiffness, as does the stability.

The journal bearing geometry has a major influence on the logarithmic decrement, due to its influence on cross coupling force magnitude. However, the bearings which suppress cross coupling best have the lowest damping against lateral vibrations, and therefore increase the rotor sensitivity to out of balance.

The following table shows these effects qualitatively on a scale in which 1 is the best:-

Type of Journal	Cross Coupling Suppression Ranking	Damping Ranking
Cylindrical	5	1
Lemon bore (2 lobe)	4	2
3-lobe	3	3
4-lobe	2	4
Tilting pad	1	5

Not listed is the off-set halves type of bearing which has cross coupling suppression second only to the tilting pad, with damping second only to the cylindrical. Unfortunately it is uni-directional and not suited to mechanical drive applications, if reverse rotation is a possibility.

Tilting pad journal bearings are required on impulse turbines where the flexible rotors lead to poor damping and low logarithmic decrements. For reaction turbines with stiffer shafts, lemon bore, 3 or 4 lobe bearings are adequate because reaction turbines have lower sensitivity to out of balance as noted in 2.8.

As noted in 2.6.1.3 shrunk-on discs contribute to cross coupling and experience with this type of machine needs to be carefully reviewed.

Second order effects on stability limits are:-

- (1) Bearing clearance, reduce to increase stability

- (2) Oil viscosity, increase to increase stability
- (3) Pre-load, increase to increase stability
- (4) Bearing specific load, increase to increase stability

These factors need to be reviewed in the detail analysis of rotor stability, for the selected bearing type since they are dependent on manufacturing tolerances, operating conditions, and alignments.

2.8.4 Torsional Analysis

- 2.8.4.1 Potential sources of torsional excitation to be considered shall include surge of compressors and electrical faults and mal-synchronisation on electric generators.

(Addition)

- 2.8.4.2 For generator drive applications the lowest torsional resonance of the complete train shall be at least 30% below the frequency of the electrical system.

(Addition)

Surge is a potential source of excitation of torsional vibrations. It will certainly excite free vibration, which will probably not be damaging. However, no analysis is known which predicts the magnitude and frequency of a surge, hence its effect cannot be quantified. The vendor and contractors experience should be reviewed.

Electrical fault excitation can occur from short circuits or mal-synchronisation of generators.

The most severe short circuit condition is phase to phase; this will produce air gap torques at system frequency.

180 degree mal-synchronisation is potentially more severe. Protection against mal-synchronisation should be provided. If on review this can be shown to be reliable, then this case may be ignored.

The 30% margin specified in 2.8.4.2 should provide an adequate safety margin against short circuit or mal-synchronisation. If this margin is not practicable then a forced damped response analysis should be performed in order to verify that torques will be acceptable.

- 2.8.4.5 The turbine vendor shall perform a torsional vibration analysis of the complete coupled train if they have responsibility for overall co-ordination. See 2.1.15.

(Substitution)

2.8.5 Vibration and Balancing

- 2.8.5.2 Balancing procedures shall be such that rotor and coupling interchangeability can be achieved without the need for rebalancing.



This will require rotors to be first balanced without couplings, and then to be check balanced with coupling hubs mounted.

(Addition)

2.8.5.4 *High speed balancing is an acceptable method for balancing rotors. It should be preceded by a low speed balance.*

The technique gives a greater certainty of meeting final test acceptance limits because it shakes out blade path settlement.

Rotors subjected to only a low speed balance may require balancing adjustment following initial test bed trials.

2.9 Bearings and Bearing Housings

2.9.1 Radial Bearings

For mechanical drive applications, radial bearings shall be capable of withstanding reverse rotation for short periods of time without damage.

Radial bearings shall be fabricated so that it is not possible to install them in any but the correct way.

(Addition)

2.9.2 Thrust Bearings and Collars

2.9.2.1 For mechanical drive applications, thrust bearings shall be capable of withstanding reverse rotation for short periods of time without damage.

Thrust bearings shall be fabricated so that it is not possible to install them in any but the correct way.

(Addition)

2.9.2.6 The vendor shall submit a graphic display of speed against maximum load capacity showing the boundaries defined by the criteria below:-

- (i) The minimum oil thickness for continuous operation.
- (ii) The maximum bearing lining temperature for continuous operation.
- (iii) The fatigue or mechanical limit for the bearing or its lining material.

This graph shall also indicate the maximum continuous and transient loads applied to the bearings.

(Addition)



The oil film thickness can be determined by calculation. It is a function of pad geometry (shape), dimensions, speed, viscosity, bearing oil supply (i.e. directed or flooded lubrication), pivot location (central or offset) and load.

The minimum acceptable oil film thickness for continuous running is empirically determined and usually conservative.

Typical limits as a function of pad radial length published by Michell Bearing Co. are tabulated below for centre and offset pivots

Radial length	mm	25	50	75	100	125
Centre pivot	microns	10	13	15	17	19
Offset pivot	microns	8	11	13	14	15

The load applied should be 50% of that to produce the above film thickness.

Film thickness is increased by increasing viscosity either by higher index oil or lower oil film temperature. Lower oil film temperatures result from using directed lubrication, higher conductivity pad materials, or reduced oil supply temperatures.

The bearing lining metal temperature is a function of the same parameters as oil film thickness. In addition offset pivots reduce oil temperature. The maximum pad temperature is taken to occur at the 75/75 position i.e. 75% of pad width back from the leading edge and 75% of pad radial length up from the pad internal diameter.

The ability of offset pivots to run backwards must be reviewed. Some vendors claim adequate capability for offset pivots.

Temperature failure can occur from two modes:-

- (a) Melting. This can result from severe overload or loss of oil, and failure is instantaneous.
- (b) Surface metal deterioration. This results in cracking and spalling of the lining material, and failure occurs over a period of hundreds of hours.

The temperature limit depends upon the lining metal. However for the white metals commonly used in industry, an upper limit of 140°C is generally accepted. This leads to a continuous operating limit of circa 120°C, which is the temperature resulting from applying a load 50% of that to produce 140°C.

However, temperatures of this magnitude can produce problems of oil lacquering of pad surfaces due to water contamination evaporating from the surface. For steam turbine applications the highest metal temperature in operation should be below 100°C (say 95°C) and the design should take account of any uncertainties which might lead to the design temperature being exceeded.

Lacquering of surfaces above 100°C has occurred with lubricating oils, with high levels of EP additives and additive constituents with a phosphorous base. Oils of the BP Energol THB type with a zinc based additive constituent rather than phosphor based have proved satisfactory up to 120°C.

Pad temperatures can be reduced as outlined earlier for oil film temperature reduction, in addition, load reduction may be possible. However, in certain limit designs, it may not be possible to reach these temperatures; the vendor and specific



site experience, together with water ingress prevention and removal measures should be reviewed before accepting higher temperatures.

The fatigue or mechanical limit is defined by the static load capability of the pad or pivot - deflection and indentation, or the fretting of pivots. This load generally occurs at 35 to 40 bar specific pad load.

- 2.9.2.8 When integral collars are provided, the design shall permit replacement with a shrunk on collar in the event of severe damage.

The ability to replace integral collars in the event of a catastrophic failure is desirable. However, with certain designs, e.g. where for rotordynamic reasons the thrust collar is in-board of the journal bearing, this may not be practical.

2.9.3 Bearing Housings

- 2.9.3.1 Bearing housings shall be fitted with connections for breathers or vents.

(Addition)

- 2.9.3.2 *Air or nitrogen buffering shall be designed to purge the oil labyrinths with an axial velocity of 4.5 to 6.0 metres per second. Allowance should be made for wear to 2x design clearance.*

These measures combined with gland condensers and possibly carbon labyrinths (2.7.1) combine to minimise ingress of steam vapour into bearing housings. In addition, careful design of any vertical drain legs is required. (See 2.9.3.8).

- 2.9.3.5 *Accelerometers are not normally necessary if shaft proximity probes are fitted.*

- 2.9.3.7 Thrust bearings and radial bearings shall be fitted with bearing metal temperature sensors in accordance with clause 3.4.4 of [API 670](#).

(Substitution)

Bearing metal temperature sensors give the most accurate indication of the bearing temperature. However, the sensors are not totally reliable and installed spares are recommended. These can be independent or dual sensors. They should be hooked-up to the instrument junction box or otherwise suitably terminated.

The sensor installation proposed by the vendor should be reviewed as follows:-

- (i) *Check that thrust sensors are in a 75/75 position (see 2.9.2.6).*
- (ii) *Check that journal sensors are close to the point of maximum temperature. The position of this point will vary with bearing design.*
- (iii) *Check the method of securing the sensors in place. This should ensure that the sensor is in intimate contact with the white metal, and secure. Spring loading or epoxy embedding have proved successful. Embedding in the white metal is not essential. Avoid loose fitting sensors since these can lose contact with the white metal and give false readings.*



Sensors enable the machine design basis to be verified and are particularly useful on thrust bearings, where increasing temperatures (at a given speed) indicate increasing load from fouling, balance drum wear, or bearing lacquering. Other faults such as inadequate lubrication or ingress of abrasives will also increase temperature.

On journal bearings monitoring temperature can aid in diagnosing misalignment, lubrication problems, lacquering or ingress of abrasives.

2.9.3.8 Vertical legs in a lube oil drain line causes the falling oil to entrain air. This causes two problems:-

- (a) Excessive oil vapour in the lube reservoir.
- (b) Moist air to be drawn into the bearing housing either via the housing breather or the shaft oil seal.

Correct sizing of the drain eliminates this effect i.e. drain flow velocity not greater than $0.97D^{1/2}$ m/sec where D = pipe diameter in metres ($1.76D^{1/2}$ ft/sec where D = pipe diameter in ft). These criteria are based on a maximum Froude Number of 0.31 for a pipe to be self venting in vertical downflow.

A site fix for this problem is to install an air recycle line from the bottom to the top of the vertical drain pipe.

Notwithstanding the measures outlined here and in 2.9.3.2 to eliminate moisture, it is still recommended that the oil system be provided with facilities for removal of moisture. (2.10.4).

2.9.4 Grounding

Brushes suitable for use in oil mist environments are available. Consideration must be given to in-service replacement of worn brushes or ensure adequate brush length is provided to last between planned overhauls.

Brushes are essential on condensing turbines because of the potential build-up of static electricity caused by wet steam flowing through the exhaust blading.

2.10 Lubrication and Control Oil Systems

2.10.4 Pressurised oil systems shall conform to the requirements of BP Group [GS 134-3](#).

(Substitution)

The type of water removal device specified shall be mutually agreed with the vendor, to maintain moisture levels within his specified limits.

The effect of moisture removal on lube oil additive depletion must also be considered, together with facilities and procedures for on-line additive replenishment.

In addition, for vacuum systems, additive depletion may be inherent in the vacuum process.



2.11 Materials

2.11.1 General

2.11.1.6 *For the purposes of evaluating the vibration stresses in blading, it shall be assumed that steam contains < 1% NaCl in the transition zone. However, since chloride concentrations can build up at the wet dry interference in the transition zone the concentration of NaCl in the steam is only one consideration; the selection of blading materials and their allowable stress values should take account of the likelihood of chlorides.*

The presence of corrosive agents in the environment can influence the choice of materials for components exposed to the atmosphere e.g. high salinity environments affect choice/use of austenitic stainless steels, sulphur dioxide, and ammonia affect the use of non-ferrous alloys. These agents will be project dependent.

2.11.3 Welding

2.11.3.6.1 *The NDT requirement for piping should be based on a criticality concept.*

This would define a criticality rating (usually 1, 2, 3 or 4) based on size, pressure and temperature rating, materials and service conditions.

The specification should be consistent with the requirements for piping elsewhere on the plant, forming part of the same project.

The NDT requirements for casings (wall and weld) shall be based on the vendors normal practice reviewed against the recommendations of [API 612](#) 4.2.2 and any defined criticality ratings.

2.11.3.6.4 *Connection designs for class 900 and above should be submitted for approval.*

3. ACCESSORIES

3.1 Couplings and Guards

The material of guards shall be sheet steel unless conditions specifically require an alternative. Coupling guards shall be sufficiently robust to resist distortion under typical industrial maltreatment with consequent risk of sparking by contact with moving parts.

(Addition)

3.1.2 Coupling hubs shall be designed for hydraulic mounting and dismounting.

All moving parts shall be guarded in accordance with national standards and national statutory regulations.

Gear couplings shall not be used without specific approval of the purchaser.



If used their design shall permit inspection of the teeth without disturbing the hubs.

When these couplings are used they shall be of anti-sludging design and be lubricated via 2, dedicated, parallel, 100%, 2 micron filters suitable for on-line maintenance.

(Addition)

For all applications flexible membrane couplings are preferred to gear couplings.

Gear couplings suffer from a number of operational problems:-

- (i) *Wear caused by fretting resulting in vibration.*
- (ii) *Relatively small radial misalignment capability which, if exceeded, may cause fretting, transmission of vibration and increased axial loading.*
- (iii) *Transmission of axial loads due to inherent friction, which may become excessive if, as happens, the teeth become clogged by sludge centrifuged from the lubricating oil.*

This locking has resulted in thrust bearing failures.

These shortcomings do not afflict the flexible membrane coupling. However, careful design of the coupling housings are required to minimise windage heating and oil mist or oil vapour generation.

In extreme cases of overheating fires have been induced. These problems are more severe with diaphragm than metallic element couplings.

Points to watch are:-

- (i) *Clearance between coupling flange and housing needs to be adequate (avoid close fits).*
- (ii) *There must be a path for cooling air to sweep the interior of the housing without entraining oil mist or vapour. The coupling flanges act like impellers drawing air in at the internal diameter and expelling it outwards.*
- (iii) *Couplings should be shrouded to minimise bolt windage.*
- (iv) *The vendor should provide windage and heat balance calculations to demonstrate that air and guard surface temperatures are safe.*

A method for predicting coupling and guard temperatures is given in 'Design of Coupling Enclosures, M. M. Calistrat, R. E. Munyon, Texas A & M 14th. Turbomachinery Symposium 1985.

Vendor experience and proposals should be carefully reviewed.

- (v) *Personnel protection guards should be provided if (as is likely) the guard surface temperatures are greater than 60 C. (A perforated screen set 1.1/2 to 2 in. off the surface will suffice).*



Gear couplings do have two advantages over flexible membrane types i.e.:-

- (i) They are lighter, which may be of value when rotordynamic design is difficult.*
- (ii) They have greater axial movement capability.*

3.1.7 Spacers of flexible membrane couplings shall be positively constrained from flying out in the event of failure of the flexible membranes.

(Addition)

3.1.8 Consideration should be given to incorporating a means for the continuous monitoring of torque.

(Addition)

A torque meter coupling is a useful aid to the condition monitoring of driving and driven equipment on mechanical drive applications.

Reliable non-contacting, inductive pickup direct reading types are available and are preferred to slip ring or radio transmitting types.

3.2 Mounting Plates

3.2.2 Baseplates

3.2.1.2.3 Horizontal jacking screws may also be required when the equipment supported weighs less than 1000 lb. (453 kg)

(Addition)

3.2.2.1 *It is preferable to mount the driving and driven unit on the same baseplate. This facilitates shop packaging and shipment and minimises site work.*

However, this concept becomes impractical as equipment dimensions increase, when manufacturing or shipping limitations dictate the maximum dimensions. These may therefore be vendor dependent.

3.2.2.2 The baseplate shall be provided with levelling pads or targets protected with removable covers.

(Qualification)

3.3 Gear Units

Gear units shall conform to BP Group [GS 134-10](#).

(Substitution)



Gears to BP Group GS 134-10 may be rated to AGMA 421.06 as an alternative to API 613.

API 613 specifies very conservative surface and bending stress factors which lead to larger wheel centres, face widths and tooth size, and consequently more costly and less efficient equipment. For high pitch line velocities (above 100 metre/sec) the reliability of the gears is reduced by oversizing due to increased scuffing risk and higher face width thermal gradients.

The use of alternative rating codes (often vendors own) should be limited to those vendors who can demonstrate satisfactory service experience with these codes.

3.4 Controls and Instrumentation

3.4.1 General

3.4.1.1 Electrical equipment and instrumentation shall conform to BP Group GS 130-2.

(Addition)

3.4.2 Control Systems

3.4.2.1 *Electronic governors are preferred. The governor vendor should be able to demonstrate a negligible probability of a turbine shutdown due to failure of the electronic control system.*

The system should be reviewed for adequate redundancy and have self-diagnostic facilities and be specifically designed for and well proven in steam turbine applications.

Proprietary units are available with a range of features which should be reviewed for their applicability to a specific project. These include configurable software, interfacing with plant computers, startup control, speed control options, extraction/admission control, process control, safety and protection features.

3.4.2.11 *Hand valves on single throttle valve turbines enable, when open, operation at minimum steam conditions whilst retaining efficient operation (minimum throttling) with normal conditions when closed.*

They may also be used to improve efficiency by closing them at part load and opening them at full or overload.

Automatic operation of these valves may be required if unexpected or frequent changes of load and steam conditions occur.

3.4.3 Protective Devices

3.4.3.3 A facility to partially stroke the trip and throttle valve is required.

(Qualification)



'Manual Exercisers' are used to stroke the emergency stop valve to demonstrate that it is free to move. Prolonged operation in a fixed position can result in build-up of deposits from the steam, or build-up of corrosion in hydraulic systems.

Regular exercising can remove these deposits or demonstrate the need for remedial action.

However, the stroke will not cover the full lift of the valve, leaving the possibility that deposits on the unstroked length will cause jamming.

A twin inlet and emergency valve arrangement will permit complete stroking of individual valves, and also aids with pipe loadings (see 2.5).

For generating sets where full load can easily be rejected, the use of twin inlets should be considered. This feature is normally available on high flow turbines only.

3.4.3.6 The mechanical overspeed trip shall be mounted on the main shaft.

Any electronic overspeed system should be capable of on-line testing.

(Addition)

Mechanical overspeed trips are the traditional means for overspeed tripping of steam turbines.

The mechanical systems suffer from a number of potential disadvantages:-

- (a) *Jamming due to build-up of sludge or corrosion in the bolt or ring and striker mechanisms.*
- (b) *Jamming open of devices with on-line testing facilities.*
- (c) *Poor repeatability.*

Electronic devices safeguard against the failure of the mechanical system.

Consideration should be given in conjunction with the vendor to removing the mechanical overspeed trip and substituting a high integrity 3 channel, 2 out of 3 voting electronic system.

The system should have three independent channels, an uninterruptible power source, and self-diagnostic facilities. This should be backed up by extensive field experience.

3.4.3.7 An electrically operated tripping system, set at a lower speed than the mechanical overspeed trip, is required.

(Qualification)

3.4.3.9 Vacuum breakers are not normally provided.

Their spurious operation could lead to a plant shutdown.



The design and operational experience with throttle and emergency stop valves should be reviewed.

The use of the upstream hand block and bleed valve should enable the operator to stop the turbine.

3.4.4 Instrument and Control Panels

- 3.4.4.1 A single local control panel shall be provided for the turbine, driven unit and ancillaries. It shall meet the requirements of BP Group [GS 130-2](#) and shall include the equipment referred to in BP Group [GS 130-2](#) including the following minimum functions for the turbine itself:-



Function	Indicator	Alarm	Shutdown
Steam inlet pressure	x		
Steam inlet temperature	x		
Steam flow at inlet	x		
Steam pressure at first stage	x		
Steam pressure at extraction (if applicable)	x		
Steam flow at extraction (if applicable)	x		
Steam pressure at induction (if applicable)	x		
Steam temperature of induction (if applicable)	x		
Steam flow at induction (if applicable)	x		
Steam pressure at exhaust	x	x	
Control oil pressure	x		
Speed	x	x	x
Torque (if required)	x		
Shaft vibration	x	x	
Shaft axial position	x	x	x
Lub oil pressure	x	x	x
Bearing (metal) temperatures	x	x	
Manual shutdown	x		
Extraction and back pressure temperature measurements (if applicable)	x		

(Qualification)

3.4.8 Vibration, Position and Bearing Temperature Detectors

Complete monitoring systems comprising sensors, monitors and all intermediate equipment shall be supplied for shaft vibration, shaft axial position, and bearing temperature. They shall be supplied and calibrated in accordance with [API 670](#).

(Substitution)

3.7 Insulation and Jacketing

3.7.1 Insulation and jacketing should be supplied and fitted by the turbine vendor.

(Qualification)



3.9 Turning Gear

- 3.9.1 It shall be possible to quickly restart the turbine without delay after a shutdown of any duration. The vendor shall supply turning gear if necessary to meet this requirement. Slow roll to remove shaft bow prior to restart will not be an acceptable substitute for turning gear.

(Substitution)

Turning gear is generally required on machines which have their first flexural critical speed below the running speed range. This type of rotor, especially on condensing duties, is particularly susceptible to thermal bending, and turning gear is recommended.

Stiff rotors i.e. where the first flexural critical speed is above the maximum continuous speed, are the least susceptible to thermal bending and turning gear may, subject to a review of vendor experience, be omitted.

- 3.9.2.3 Automatic engagement shall not occur in the event of reverse rotation following a trip.

(Addition)

Mechanical drive turbines may sometimes be driven in reverse for a limited period of time following a trip. The turning gear should be designed to prevent automatic engagement under these conditions, in particular overrunning clutches should be fitted with a lock-out device.

4. INSPECTION, TESTING AND PREPARATION FOR SHIPMENT

4.1 General

- 4.1.4 The vendor shall table his internal inspection and test plan as the basis of discussion to agree the extent of purchaser participation in the inspection and testing.

(Substitution)

Verification of the vendor's quality system is normally part of the pre-qualification procedure, and is therefore not specified in the core text of this specification. If this is not the case, clauses should be inserted to require the vendor to operate and be prepared to demonstrate the quality system to the purchaser.

Further suggestions may be found in the BP Group RPSEs Introductory Volume.

Purchaser participation will need to be agreed on an individual job basis recognising factors such as:-

- *The maturity of the design.*
- *The criticality of the machine with respect to operation and safety.*



- *Experience in the production and test facility where the machine will be built and tested.*
- *Previous experience with the vendor.*

Any requirement for inspection by an Independent Authority such as might arise from statutory or insurance reasons should be taken into account.

Typical inspection activities pertinent to special purpose steam turbines are listed below together with guidance on their importance. It is assumed that the competence of the vendor has been established as satisfactory by previous experience or by audit.

- (a) *Material certification: certificates for major items such as casing, rotors and shafts should normally be examined. They should be readily available being a requirement of API 4.2.1.1.*

Additionally, it should be established that satisfactory systems exist for material traceability.

- (b) *Repairs: those justifying purchaser involvement would normally be limited to through-thickness weld repairs, repairs of cracks in casings (to establish the cause) and repairs to rotating elements.*
- (c) *Overspeed tests: witnessing of these will not normally be necessary unless the rotors are exceptionally heavy with a very high kinetic energy such that a failure might be unconfined.*
- (d) *Balancing: witnessing will not normally be justified as the state of balance will be demonstrated during the mechanical test(e) Pressure tests: witnessing of main casing tests is necessary as a check of functionality. Witnessing of tests on ancillary systems is not normally justified.*
- (f) *Dimensions and layout: checking of dimensions will not normally be necessary but layout of customised (non-standard) packages should be examined to ensure adequate access for operation and maintenance.*
- (g) *Mechanical and performance: all tests should be witnessed. They demonstrate the essential functionality of the machine.*
- (h) *Packaging: this may justify attention if shipment is onerous or if long term storage is required.*

4.2 Inspection

4.2.1 General

- 4.2.1.1e Records of final-assembly, maintenance and running clearances shall be retained for at least 5 years.

(Qualification)



4.2.1.3 The type and extent of non-destructive examination should be agreed in detail with the vendor. See 4.2.3.5

The vendors normal procedures should be acceptable unless National Codes or Statutory Requirements overrule.

The following guidelines are given for the purpose of assessing the vendors proposals:-

(a) Class 1500 and 900 Casings:-

(i) Welds: as for pipe welds see 2.11.3.6.1

(i) Casing walls: 100% ultrasonic

(ii) Cast Intersections: 100% ultrasonic
plus 100% magnetic particle inspection
plus Visual examination

(b) Class 600 Casings and below:-

(i) Welds as for pipe welds see 2.11.3.6.1

(ii) Casing walls: 100% MPI

(iii) Intersection: 100% MPI

(c) Rotors, discs and shafts as [API 612](#)

(d) Blades:-

(i) Forged: 100% radiography

(ii) Bar stock: 100% dye penetrant

4.2.1.4 Quality Assurance

Verification of the vendor's quality system is normally part of the pre-qualification procedure, and is therefore not specified in the core text of this specification. If this is not the case, clauses should be inserted to require the vendor to operate and be prepared to demonstrate the quality system to the purchaser. The quality system should ensure that the technical and QA requirements specified in the enquiry and purchase documents are applied to all materials, equipment and services provided by sub-contractors and to any free issue materials.

Further suggestions may be found in the BP Group RPSEs Introductory Volume.

4.2.2 Material Inspection

4.2.2.5 The vendor shall supply details of procedures and acceptance limits for thermal stability tests on rotors.

(Substitution)

Thermal stability checks may be required for forged rotors above 610 mm (24 inches) wheel diameter.



The checks are designed to eliminate asymmetric residual stresses arising from the forging process and check that the rotor is free from temperature dependent bending resulting from permanent asymmetric circumferential variation in properties.

The checks are done by rotating the rotor in an oven at a temperature just below its tempering temperature. The rotor eccentricity is measured at various points and the check continued until no further changes in eccentricity are detected.

The check is repeated at a lower temperature (but above the maximum operating temperature). The cold to cold eccentricity change is noted and must not exceed a prescribed limit. The check is repeated until the limit is satisfied. This should ideally be zero.

The cold to hot eccentricity change is also noted. This is a particularly critical parameter as it indicates the presence of temperature dependent bending, which, depending upon the rotor parameters and operating speeds, may lead to its rejection. Ideally this change should be zero. The changes must not be confused with effects caused by differing surface emissivities, which should be transient and disappear once the rotor has soaked for a sufficient time. The check must be repeated to demonstrate this effect has corrected itself.

4.2.3 Mechanical Inspection

4.2.3.5 BP may require demonstration of the quality system, but this may be waived if the system has been verified recently by an accreditation scheme acceptable to BP.

4.3 Testing

4.3.1 General

4.3.1.1 The following tests are required:-

- (a) Hydrostatic test in accordance with 4.3.2. of [API 612](#).
- (b) Mechanical running test in accordance with 4.3.3.

See 4.3.4 for requirements for Optional Tests

(Substitution)

4.3.3 Mechanical Running Test

4.3.3.1.9 Shaft runout at the location of each proximity probe shall be measured immediately before the mechanical running test.

(Addition)

4.3.3.3.2 The sweep of vibration amplitudes versus frequencies shall additionally be carried out at the minimum operating speed and at the normal operating speed.



Journal orbits shall be recorded at maximum continuous speeds. Vibration phase readings shall be related to the fixed shaft phase reference.

(Addition)

4.3.3.3.5 & 4.3.3.3.6

Tape recordings enable detail analysis of phase, amplitude and spectrum to be made subsequent to the testing, and also to capture transient events, e.g. runup, coast down, or any unscheduled happening during the tests.

Copies of these tapes are not normally requested.

- 4.3.3.3.7 During the 4 hour test, lube-oil and control oil temperatures shall be held for at least half an hour at the value corresponding to the minimum allowable viscosity and half an hour at the values corresponding to the maximum allowable viscosity. Under both conditions shaft vibrations shall be measured in accordance with 4.3.3.3.2 checking in particular for oil film instabilities.

(Addition)

The purpose of varying the oil supply temperature and hence the oil viscosity is to:-

- (a) *Ensure that the rotor is stable at the minimum viscosity (high temperature oil) and also that the temperatures are acceptable.*
- (b) *Ensure that bearing temperatures (particularly the thrust bearing) are acceptable with maximum viscosity (low temperature oil).*

4.3.4 **Optional Tests**

- 4.3.4.1 *A shop performance test is possible when there is steam available at the required quality and in the required quantity, together with the means to load the turbine.*

Where steam consumption is important and guarantees, with penalties, have been given by the vendor, then a test is essential. In the absence of shop facilities site testing can be arranged. This requires provision for the accurate measurement of steam conditions, flow and load. This will need to be considered at the design stage.

- 4.3.4.2 *Wherever the shop facilities permit, a complete unit test should be specified. The complete unit test may be performed in place of separate tests. However, this may not always be justified; the time required for an additional complete unit test may be more valuable during plant pre-commissioning on site provided that the driven unit can be operated (say the compressor running on air). There is therefore a need to evaluate the benefit of a complete unit test in addition to the usual mechanical tests. Different constraints will be added for offshore activity.*

Torsional tests are required only where analysis leads to concern and the need to determine natural frequencies and damping. When these are required they should be carried out in the manufacturers works on the complete train.



4.3.4.3 *Shop sound level tests have diminishing validity as steam conditions and load deviate from site conditions.*

Where a valid test can be made the 'sound intensity' technique is recommended. (See 2.1.11). The services of a specialist consultant is usually required for this purpose.

4.3.4.4 *Auxiliary systems should be tested wherever shop facilities permit.*

4.3.4.5 *Internal inspection should only be made if the testing has given rise to concern for the internals.*

4.3.4.6 *The requirement to inspect coupling hubs has not arisen on past BP contracts. The fact that API notes this as an option implies there have been problems elsewhere.*

The vendor's experience should be examined or integral coupling flanges used.

4.3.4.7 *Additional tests/checks may include:-*

- (a) *Coupling windage checks*
- (b) *Thrust load checks*
- (c) *Governor response checks*
 - i) *Injecting a signal into the speed control loop and measuring the response of control valves.*
 - (ii) *Full load rejection.*
- (d) *Trip valve response checks*
- (e) *Rotor to stator differential expansion check*
- (f) *Casing expansion checks*
- (g) *Pipe load distortion checks*
- (h) *Hot run-up check*

5. VENDOR'S DATA

5.1 Proposals

The following information shall be provided in addition to that listed:-

- (a) *Blading limitations referred to in 2.6.3.5.*
- (b) *Potential maximum power.*
- (c) *Approximate values for lateral critical speeds.*



- (d) Typical sound pressure levels.
- (e) A description of the quality system required by clause 4.2.1.4.

(Addition)

5.1.g.3 *Failure can originate at the trailing edge of blading, especially if the edge or the transition to the root is sharp. The vendor should be required to give details of blade configuration in these areas.*

5.2 **Contract Data**

5.2.5 **Data**

5.2.5.2d The vendor shall submit certified copies of test data to the purchaser before shipment.

(Qualification)

5.2.5.2f The vendor shall supply calculated rotor response reports.

(Substitution)



CHECK LIST TO API 612 THIRD EDITION - DATA SHEETS

DATA SHEET REFERENCE			API PARA	SUBJECT	REQUIREMENT
PAGE	LINE	COL.			
Performance					
1	11	1-9	2.1.1	Normal operating point	Process Engineering
1	13	1-9	2.1.3(e)	Specify lowest speed & maximum torque with minimum inlet and ma. exhaust conditions	Process Engineering
Steam Conditions					
1	20-22	1-8	2.1.3 (a)-(h)	Specify min., normal, steam conditions	Process Engineering & BP Group GS 134-7
Site and Available Utility Data					
1	24	2	2.1.3	Location	Project Specific, also see 3.4.7.5
1	30	2	2.11.1.6 2.11.1.9	Atmospheric Corrosion agents	Project Specific
1	31	-	2.1.12	Area Classification	Project Specific, also see 3.4.7.5
1	32-38	1-4	3.4.7.1	Specify electrical supply characteristics	Project Specific
1	32-39	5-6	2.1.10	Specify cooling water system design parameters	Process Engineering
Reference Specifications					
2	4	1		Turbine Specification	Specify BP Group GS 134-7
2	4	1	2.11	Noise Specification	BP Group GS 134-7 or Project requirements
Casings, Nozzles and Diaphragms					
2	15	2	2.3.1.1	Welded nozzle ring	BP Group GS 134-7
Casing Connections					
2	23-27	1	2.11.3.6.4	Submit casing connection design for approval	BP Group GS 134-7
2	23-27	4	2.4.1	Inlet and outlet nozzle position	Project Specific, also see 2.2.11
2	23-27	5	2.2.11	Use of studded connections	Undesirable, see BP Group GS 134-7
2	23-27	6	2.4.6.4	Mating flange and gaskets for sizes >ANSI	This feature is required
Allowable Piping Forces and Moments					
2	28-35	Rmks	2.5	Allowable external forces and moments	BP Group GS 134-7
Materials - Casings and Appurtenances					
2	37	1	2.11.1.6 2.11.1.9	Steam contaminants	API, BP Group GS 134-7 and Process
2	38	1	2.11.1.9	Steam path hardness	Process Engineering and BP Group GS 136-1
2	39	1	2.11.1.15	Low temperature service materials	Project Specific to ASME



DATA SHEET REFERENCE			API PARA	SUBJECT	REQUIREMENT
PAGE	LINE	COL.			
Rotating Elements					
3	8	2	2.6.15	Provision for field balance	Required BP Group GS 134-7
Shaft Seals					
3	26	1	2.7.1	Carbon ring packings	Not above 44m/sec BP Group GS 134-7
Bearings and Bearing Housings					
3	48	2	2.9.2	Thrust bearing lubrication	BP Group GS 134-7
3	49	2	2.9.2	Thrust collar assembly Integral collar/replaceable	Integral plus replacement capability BP Group GS 134-7
3	50	2	2.9.2	Thrust pad materials	BP Group GS 134-7
Lubrication and Control Oil Systems					
4	2	1	2.10.4	Lube system reference specifications	BP Group GS 134-3
4	3-12	1	2.10.2 2.10.5	Lube system supplier and scope	Prime vendor
Coupling and Guards					
4	14-29	1-2	3.1.2	Coupling type and details	BP Group GS 134-7
4	17	1	3.1.1	Coupling supplier	Prime vendor
Mounting Plates					
4	27	1 or 2	3.2.1.1	Baseplate or soleplate	Baseplate where practical. See BP Group GS 134-7
4	28	2	3.2.2.1	Extended baseplate	Required unless manufacture, transport or lifting overrule
4	29	2	3.2.2.6	Non-skid decking	Required
4	30	2	3.2.2.2	Levelling pads	Required
4	31	1	3.2.2.3	Column mounting	Not normally required
4	32	1	3.2.2.5	Sub-sole plates	Required
4	33	1	3.2.1.2.7	Epoxy grout	Required Add note to Data Sheet
Gear Unit					
4	38	2	3.3	Gear unit spec.	BP Group GS 134-10
Remarks					
4	40	1	3.1.2	Torque meter	Required for mechanical drive See 3.1.9 Add note to Data Sheet
Controls and Instrumentation					
5	2	2	3.4.4.1 Supplier	Local control panel	Prime vendor Add note to Data Sheet
5	2	2	3.4.4.4	Air purging of panels	BP Group GS 130-2 Add note to Data Sheet
5	2	2	3.4.8.2	Vibration and axial position monitor	Prime vendor Add note to Data Sheet
5	2	2	3.4.8.4	Bearing temp. monitors supplier	Prime vendor Add note to Data Sheet
5	8	1	3.4.1.1.	Instrumentation spec.	BP Group GS 130-2



DATA SHEET REFERENCE			API PARA	SUBJECT	REQUIREMENT
PAGE	LINE	COL.			
Protective Devices					
5	12-21	1 & 2	2.2.5	Exhaust relief valves (Exhaust & Extract)	Supplier - Project Data - process Also see 2.2.4 and 3.4.5.7
5	12-21	3	2.3.7	Sentinel warning valve	Required
5	12-21	4	3.4.3.9	Vacuum breaker	Not normally required
5	12-21	5	3.4.3.8	Non-return valve	Supplier - Project Data - Process Engineering
Trip and Throttle Valves					
5	31	1 or 2	3.4.3.3	Trip valve manual exer.	Required (see 3.4.3.3)
Turning Gear					
6	33-44	1	3.9.1	Turning gear	BP Group GS 134-7
Miscellaneous					
6	34	2	2.1.6	Vendor review of piping and foundation drg.	Required
6	36	2	2.1.6	Vendor witness initial alignment check	Required
6	39	2	3.2.2.2	Optical alignment flats	Required
Governor					
7	1	1	3.4.2.6	NEMA class of governor	API 612
7	1	2	3.4.2.2	No. of governor speed pickups	2 per API 612
7	2-30 2-6	1 1	3.4.2.1 3.4.2.2	Speed governor type and characteristics	Type-Electronic to BP Group GS 134-7 Characteristics Project
7	8	2	3.4.2.11	Hand Valves	Process Engineering & BP Group GS 134-7
Overspeed Trap					
7	10-22	2	3.4.3.6	Overspeed trip	BP Group GS 134-7
Gland Sealing and Vacuum System					
7	42-50	1	3.7.1	Insulation and jacketing	BP Group GS 134-7
Inspection and Testing General					
8	306	1	4.1.4	Extent of shop inspect	BP Group GS 134-7 and this checklist
Inspection and Material Testing					
8	8	1	4.2.1.1e	Final assembly clearance records	Required
8	9-20	1	2.11.36 4.1.3 4.2.1.3	Special material inspec. and testing requirements	BP Group GS 134-7
8	21	1	4.2.2.5	Heat stability	BP Group GS 134-7
8	22	1	4.2.3.3	Cleanliness check	Required Witness
8	23	1	4.2.3.4	Hardness check	For NACE mat only. Test certs required
8	24	1	4.3.2	Hydro tests	Required Witness



DATA SHEET REFERENCE			API PARA	SUBJECT	REQUIREMENT
PAGE	LINE	COL.			
Inspection and Material Testing (Cont'd)					
8	25	1	None	Blade shaker tests	Not normally required - Test certs. (see BP Group GS 134-7 2.6.3.1)
8	27	1	2.8.5	Low speed balance	Required Witness
8	28	1	1.8.6.4	High speed balance	Required where facilities exist Witness
8	29	1	4.4.3	Final surface inspection	Required
8	30	1	4.4.1	Crating inspection	Required Witness
8	31	1	None	Spare rotor fit	Required Witness
Painting					
8	32	1	None	Painting specification	Vendor std. to Project approval
Shipment					
8	36-42	1	4.4.1 4.4.3.10	Shipment	Project Specific
Mechanical Running Tests					
8	4	2	4.4.3	Contract rotor	Required Witness
8	5	2	4.4.3	Spare rotor	Required Witness
8	6	2	4.4.3	With job coupling	Required
8	7	2	4.4.3	Test tape required	Required
8	8	2	4.4.3	Test tape to purchaser	Project Specific
Optional Tests					
8	11	2	4.3.4	Optional tests	BP Group GS 134-7
8	13	2	4.3.4.1	Performance test	Required when facilities permit - see BP Group GS 134-7 Witness
9	14	2	4.3.4.2	Complete unit test	Required when facilities permit - see BP Group GS 134-7 Witness
8	15	2	4.3.4.2	Torsional measurement	Not normally required (see BP Group GS 134-7)
8	16	2	4.3.4.3 2.1.11	Sound level	Required Witness
8	18	2	4.3.4.4	T and T valve	Required Witness
8	19	2	4.3.4.4	Gland sealing system	Required Witness
8	20	2	4.3.4.4	Gland vacuum system	Required Witness
8	21	2	4.3.4.4	Lube oil system	Required Witness



DATA SHEET REFERENCE			API PARA	SUBJECT	REQUIREMENT
PAGE	LINE	COL.			
Optional Tests (Cont'd)					
8	22	2	4.3.4.4	Relief valves	Bench test Witness
8	25	2	4.3.4.5	Casing internal inspect	Required if test causes concern BP Group GS 134-7
8	26	2	4.3.4.6	Coupling to shaft fit	Match marking required Witness
8	27	2	None	Turning gear	Required when supplied Witness
8	28	2	4.3.4.7	Additional tests	BP Group GS 134-7



DEFINITIONS AND ABBREVIATIONS**Definitions**

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

Abbreviations

AGMA	American Gear Manufacturers Association
ANSI	American National Standards Institute
API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
BS	British Standard
EEMUA	Engineering Equipment and Materials Users' Association
EP	Extreme Pressure
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.

American

AGMA 421.06	<i>Practice for High Speed Helical and Herring Bone Gear Units</i>
API 612	Special-Purpose Steam Turbines for Refinery Services (to be replaced by ISO 10437)
API 613	Special Purpose Gear Units for Refinery Services (to be replaced by ISO 13711)
API 614	Lubrication, Shaft-Sealing and Control-Oil Systems for Special Purpose Applications (to be replaced by ISO 10438)
API 670	Vibration, Axial-Position, and Bearing-Temperature Monitoring Systems.

BP Group Documents

BP Group GS 130-2	Instrumentation for Rotating Machinery. (replaces BP Std 128)
BP Group GS 134-3	Lubrication, Shaft Sealing and Control Oil Systems for Special Purpose Applications to API 614 (replaces BP Std 190)
BP Group GS 134-10	Special Purpose Gear Units to API 613. (replaces BP Std 202)

EEMUA

EEMUA Publication No. 140	Noise Procedure Specification (replaces OCMA specification NWG-1).
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