



GS 134-5

CENTRIFUGAL COMPRESSORS TO API 617

June 1992

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BP GROUP RECOMMENDED PRACTICES AND SPECIFICATIONS FOR ENGINEERING											
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SCOPE AND PURPOSE This BP Group Guidance for Specification covers requirements for Centrifugal Compressors, Excluding fans and blowers that develop less than 0.34 bar pressure rise above atmospheric pressure and secondary packaged, integrally geared air compressors. 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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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 617 Fifth 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 134-5) 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 uses a 'zero-based' approach to define BP's essential requirements.



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 Engineering or the Custodian. See Quarterly Status List for contacts.



1. GENERAL

1.1 Scope

This specification covers BP requirements for centrifugal compressors excluding fans and blowers that develop less than 0.34 bar pressure rise above atmospheric pressure and excluding packaged, integrally geared air compressors.

They shall meet the requirements of [API 617](#), Fifth Edition, dated April 1988 except as amplified and modified herein.

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

(Substitution)

1.2 Alternative Designs

Requirements alternative to those prescribed will be acceptable provided it can be shown to the satisfaction of the purchasers' professional engineer that the required performance and function is 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)



1.4 Definition of Terms

Refer to Appendix A.

(Addition)

1.5 Referenced Publications

Refer to Appendix B.

(Addition)

1.6 Coordination

The compressor vendor shall be responsible for the co-ordination of the design and for the satisfactory functioning of the complete unit, ie, compressor driver, transmission and ancillaries. In cases where the compressor vendor supplies equipment that he has not manufactured, he shall be responsible for ensuring that the designs of these items are compatible with each other and with his own equipment in all respects. In particular, they shall be compatible dimensionally, in performance, in control and in vibration characteristics such that a fully integrated unit is achieved. The satisfactory functioning of the complete unit shall form part of the compressor vendor's contractual guarantee.

(Addition)

For certain installations, particularly gas turbine-driven sets, the coordination might be better undertaken by the gas turbine vendor, reflecting the greater capital cost of the turbine.

2. BASIC DESIGN

2.1 General

2.1.4 For fresh or recirculated water the velocity in the exchanger tubes shall be 0.9 m/s to 1.5 m/s (3 ft/sec to 5.0 ft/sec).

(Qualification)

2.1.9 Noise levels at or beyond 1 m from the machine (plus driver, transmission and ancillaries) surfaces, shall not exceed 85 dB(A) unless an alternative limit is specified on the data sheet.

Noise limits below 85 dB(A) may be required in some countries.

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

Noise-attenuating enclosures 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.

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.

(Substitution)

Noise attenuating enclosures should only be accepted when there is no practical alternative form of noise control.

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 levels to be identified and reduced by locally applied absorption materials.*

(b) *Improved compensation for background and reverberative effects.*

2.1.10 *Liquid injection is required where the process gas contains contaminants which deposit themselves on the impeller causing blockage or unbalance.*

2.1.13 *It is normal practice for vendors to carry out (a) and (c) of the API requirement. The need for a hot alignment check, (b) will depend upon the type of coupling employed, the operating temperatures, the type and construction details of the train equipment and the flexibility of the process piping. It is not normally required where dry type couplings are employed, with centre-line supported casings and well designed process piping.*

In other cases the potential for misalignment needs to be assessed against the misalignment tolerance of the coupling. The dry type coupling's capability is significantly greater than the gear type, typically .0015 in/in and .0001 in/in respectively. The relatively low misalignment tolerance of the gear coupling is the principal reason for the practice of hot alignment checking.

2.1.16 *The vendor shall state possibilities for pre-commissioning field running on air, or inert gas, or under vacuum. Operating limitations such as high discharge temperature, speed, minimum sealing pressure shall be indicated.*

(Substitution)

Pre-commissioning requirements should not be allowed to compromise compressor design but methods should be agreed at an early stage so that any special site facilities can be organised.

Operation on air or nitrogen is often impractical because of:-

(a) *High discharge temperatures resulting from the high ratio of specific heats.*



- (b) *Mismatch of compressor stages due to different molecular weight and temperature, resulting in surge and/or stonewall.*

Operation on air may be impractical due to safety considerations.

Satisfactory procedures can usually be developed using other gases, especially if speed is variable.

A further potential difficulty arises from the use of (high pressure) oil seals at low pressure. Oil starvation can occur resulting in overheating. Special provision may need to be made to maintain oil flows.

2.1.18 The Mach No. at the tip of the impeller eye shall not exceed 0.8.

(Addition)

At flows greater than the design point, the compressor will eventually 'choke' i.e. somewhere within the impeller the Mach No. exceeds unity. This normally occurs first at the tip of the impeller eye, when the relative inlet velocity reaches the local speed of sound.

This is usually only of practical significance on high mol. wt. gases or medium mol. wt. gases at very low temperatures.

In order to give a reasonable operating range this impeller eye Mach number (Me) is limited at the design point to 0.8. However, Mach Nos up to 0.85 have been used.

Data to accurately determine Me is not normally available at the time of initial assessments but it can be estimated by the procedure detailed in Appendix C.1.

The limit of 0.8 quoted for Me may be relaxed, provided the vendor can demonstrate a design methodology for his impellers backed-up by development and field experience, and the compressor has an adequate stable flow range.

2.2 Casings

API 2.2.3 - Compressor discharge pressure will increase with:

- *high suction pressure*
- *high speed*
- *low temperature*
- *high molecular weight*
- *low flowrate*

The worst combination of these will usually result in a pressure well above normal, and for many applications it would be uneconomic to design for such high values. A relief valve will therefore normally be required.

The setting should allow for operation at the surge control point at maximum continuous speed and maximum suction pressure with normal gas composition and temperature.



- 2.2.8 Radially split casings shall be used for flammable or toxic services with maximum allowable working pressures above 40 bar (ga).

(Addition)

Horizontally split casings of adequate strength are available for pressures greater than 40 bar, however, maintaining leak tightness becomes increasingly difficult at high pressure. Problems are most likely to occur at the junction of the main joint with the seal housing because of the 3-dimensional form of the joint, plus the difficulty in providing adequate bolting load at this point. Large temperature gradients, as can occur adjacent to seal housings, or between process sections of a casing, will aggravate the problem.

Prediction of joint behaviour is difficult at the design stage and problems are revealed only on a test bed when remedial actions are limited. For flammable or toxic applications where leakage could have serious consequences, axially split casings should not be used above 40 bar unless the vendor can demonstrate satisfactory experience on a virtually identical design under similar pressure and temperature conditions.

- 2.2.9 *O-ring sealing of main joint faces has been used successfully, but the junction of main joint with the seal housing is a point of weakness. The vendor should be required to justify his design for this region.*

2.3 Interstage Diaphragms and Inlet Guide Vanes

- 2.3.3 Rotating labyrinths may be used if backed by evidence of satisfactory operating experience.

(Qualification)

Rotating labyrinths have the inherent advantage of minimising heat transfer into the shaft in the event of a rub. Thus, the risk of shaft bending and further rubbing, is reduced. This permits slightly tighter clearances and higher efficiencies and should enhance reliability.

They normally take the form of thin strips caulked into grooves in the rotor. A disadvantage is the need for more sophisticated maintenance procedures.

They are preferable to stationary labyrinths provided appropriate overhaul facilities are available.

2.4 Casing Connections

- 2.4.2.1 *An advantage of axially split casings is the ability to access the rotor and other internals without disturbing adjacent equipment. This can be of particular value for inner machines of multiple casing trains. Main process connections on the upper half detract from this advantage, but may need to be considered if they permit significant benefits in plant layout.*

Studded connections on casing nozzles hinder the installation and may hinder maintenance since piping has to be sprung in order to remove casings or spool pieces.



Flanged through bolted connections are therefore preferred. However, on very high pressure barrel casings the use of studded connections simplifies casing construction and offers cost and weight savings. This arrangement has been adopted as standard by many vendors. In these cases easily removable spool pieces should be provided.

- 2.4.3.2 Individual stage drains shall be provided on all compressors fitted with liquid injection facilities, and on those that require periodic washing off-line.

(Qualification)

Drains shall be individually valved. Valves shall be accessible from the operating floor. Drain outlets shall be visible from the drain valves, or other means shall be provided to permit safe monitoring by the operator.

(Addition)

2.6 Rotating Elements

- 2.6.7 Impellers manufactured by electro-erosion and brazing may be used if backed by evidence of satisfactory operating experience.

Rivetted impellers shall not be used on sour service.

(Qualification)

Electro-erosion or brazing is particularly valuable in the manufacture of narrow, closed impellers. Alternatives for such duties are riveting or slot welding. Rivetted designs will be limited by strength and potential corrosion. Slot welding is inherently weakening and quality control is difficult.

- 2.6.18 The compressor vendor shall, jointly with the driver 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 125% of this figure.

(Addition)

With any a.c. drive, a line frequency oscillating torque with a decaying peak typically attaining 3-4 times full load torque (FLT), exists during the run-up period. With synchronous motors, there is in addition a variable frequency torque oscillation of 0.5 - 1.0 times FLT from 2 x line frequency at standstill to zero frequency at full speed.

In addition to high torques experienced during starting, even higher transient torques may occur due to short circuits on the supply system or out of synchronous reconnection of the supply following a transient power failure.



The worst system fault condition from the point of view of the driven equipment is a phase to phase fault near the motor terminals. Depending on the motor design, this condition can produce a transient torque of 4-7 times FLT.

Out of synchronisation reconnection under the worst conditions can produce transient air gap torques of 7-10 times FLT. Such high overloads can be avoided by delaying restoration of the supply for around 1 sec., to allow time for the residual magnetic flux to decay. The transient torque should by then be less than the phase-phase short circuit value.

- 2.6.19 Rectification of machining errors on rotating elements shall be subject to purchasers approval. (Addition)

2.7 Bearings and Bearing Housings

- 2.7.1.3 Radial and thrust bearings shall be capable of withstanding reverse rotation for a short period of time without damage.

(Addition)

Reverse rotation can occur on tripping if the stored energy on the discharge side of the compressor is large compared to the kinetic energy of the rotor system. The maximum permissible stored energy to avoid reverse rotation will depend on the means by which it blows down on tripping. A limit of twice the kinetic energy should minimise the risk.

A further risk remains from leakage through non-return valves. When practicable these should be backed-up by the automatic closure of block valves. Additionally it is desirable that compressors be designed to accept some reverse rotation.

- 2.7.3.3 External forces transmitted through the coupling shall be considered as numerically additive to any internal thrust forces.

(Addition)

- 2.7.3.7 The vendor shall supply to the purchaser a graphic display of speed against maximum load capacity showing the boundaries defined by the criteria below:-

- (i) The minimum oil film 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)

Thrust bearing design can be critical to the successful application of centrifugal compressors. The information requested will provide a rational basis for assessing the suitability of the design. Additional information including guidance on acceptance criteria for oil film thickness and bearing lining temperature is given in Appendix C2.

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.

2.7.4.3 Vertical legs in 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. Drain flow velocity should not exceed $0.03\sqrt{D}$ m/sec where D is drain diameter mm.

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

2.7.4.8 Oil shall not be lost through vents or breathers.

(Addition)

2.7.4.9 The design of shaft and casing shall be such that space is available to fit a shaft earthing brush to overcome bearing problems resulting from electrical discharge between shaft and earth.

(Addition)

2.8 Shaft Seals

2.8.1 Shaft seals and their supporting systems shall be suitable for operation at the maximum suction pressure.

(Addition)

2.8.2 Shaft seals and their sleeves shall be accessible for replacement without removing the top half casing of an axially split compressor or the heads of a radially split unit.

(Substitution)

2.8.3.2 Mechanical contact type seals shall prevent gas leakage when the compressor is not running and the seal oil system is shutdown.



(Addition)

- 2.8.3.4 Liquid film seals with inward oil flows exceeding 50 litres per day per seal shall not be used on machines handling rich hydrocarbon or gases that are corrosive or toxic. This shall apply even if the seals are normally buffered by a clean non-contaminating gas.

(Addition)

Reclamation of seal oil that has been in contact with rich hydrocarbons or corrosive or toxic elements will usually be difficult. The limit of 50 litres per day means that a fall-back option of discarding the contaminated oil should be economic if reclamation proved to be impractical.

'Rich hydrocarbons' cannot be defined exactly, but above approximately 0.1 mol% C5 and heavier, simple atmospheric degassers become ineffective.

Experience has shown that external buffer gas systems are usually subject to interruption, and hence, whilst a sweet, lean buffer should be used whenever possible to improve the life and reliability of sealing systems, total reliance should not be placed upon them.

- 2.8.3.5 Self-acting gas seals on flammable or toxic duties shall meet the following requirements:-

- (a) There shall be no leakage of flammable or toxic gas to atmosphere local to the machine, or into the bearing housing in normal service nor when the primary seal has failed.

This will normally require the fitting of a secondary seal rated for the full duty.

It may be assumed that leakages can be piped to flare or to atmosphere at a well ventilated location.

The availability of external supplies of nitrogen, air or other gas for purging purposes shall be established for each application.

- (b) Means shall be provided for continuously monitoring the integrity of the primary seal, and secondary seal if fitted, together with any essential buffer or purge gas systems.

(Addition)

Self-acting gas seals have leakage rates typically in range 10-100 L/min dependent on pressure, diameter and speed. If these leakages are of flammable or toxic gas, they need to be directed to a well ventilated location away from the compressor itself or its bearing housings.

Additional information on the design, limitations and application of gas seals is given in Appendix C3.



2.8.7 *Buffering of the shaft seal may be desirable to protect the seal from:-*

- (a) *High or Low Temperatures*
- (b) *Sour Gas*
- (c) *Corrosive Gas*
- (d) *Abrasive Gas*
- (e) *Wet Gas*

2.8.8 For compressors with sub-atmospheric pressure at the shaft end seals, provision shall be made to pressurise these seals with gas at a pressure that is higher than atmospheric.

(Substitution)

2.8.9 Shaft seals shall be capable of withstanding reverse rotating for a short period of time without damage.

(Addition)

The reverse rotation capability may be difficult to achieve with self-acting gas seals. See Appendix C3.

2.9 **Dynamics**

API paragraphs 2.9.1, 2.9.2 and 2.9.3 and Appendix E cover the requirements for analysis of rotor lateral critical speeds.

The procedure improves on earlier API requirements by relating separation margins to amplification factors. It also relates acceptable amplitudes to internal clearance and requires test bay verification of the analysis.

However, the testing is 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 therefore essential to review the source and quality of vendor data and correlations between calculation and test from previous jobs. Particularly where low Amplification Factors and hence reduced separation margins are claimed.

In general, flexural modes with nodes close to the bearings are lightly damped. Calculated critical speeds will therefore be sensitive to the stiffness assumed. On the other hand, Flexural modes with nodes remote from the bearings are likely to be more heavily damped. Calculated amplification factors will therefore be sensitive to the damping assumed.

Bouncing or conical modes entail significant movement at the bearings and are therefore normally heavily damped. The accuracy of analysis is again dependent on the accuracy of damping assumed.

Where satisfactory correlations between calculation and previous tests cannot be demonstrated then the procedures of [API 617](#) Fourth Edition involving fixed values for separation margins should be used.



- 2.9.1.4 A stability analysis shall be performed on machines that have a flexural critical speed less than $0.5 \times$ maximum continuous speed, or handle gas at a density exceeding 70 Kg/cu m.
- The analysis shall demonstrate a positive logarithmic decrement up to trip speed, allowing for all aerodynamic, hydrodynamic and hysteretic cross-coupling forces. The vendor shall provide justification for the values assumed for these forces.
- 2.9.2.3 *A train lateral analysis is required when the elements of the equipment train are solidly coupled.*
- The effects of misalignment on bearing loads and 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.*
- 2.9.2.4(d) An additional plot shall be provided for an unbalance sufficient to cause vibration amplitude at the probe locations at maximum continuous speed equal to the vendor's recommended alarm level. It shall include the amplitude at seal locations along the shaft when the machine is operated through any resonance, including coast down from trip speed.
- (Addition)
- 2.9.2.8 *Guidance on potential sources of self-excited vibration and on methods for their control is given in Appendix C4.*
- 2.9.2.4(e) A stiffness map shall be provided for all analyses.
- (Qualification)
- 2.9.4.5 The vendor shall perform a damped torsional response analysis for all motor driven compressor sets. It shall include excitations arising in the motor due to starting and short circuits plus, if applicable, variable speed control equipment and out of synchronous reconnection of the supply following a transient power failure.
- (Addition)
- Guidance on potential sources of torsional excitation is given in Appendix C5.*
- 2.9.5.2 Balance procedures shall be such that coupling replacement 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)
- 2.9.5.4 High speed balancing at operating speed will be accepted as on alternative to the procedure detailed in API clause 2.9.5.2, if backed by



evidence of satisfactory experience with similar rotors at similar speeds. The vendor shall propose acceptance criteria.

(Substitution)

Information on the advantages and disadvantages of high and low speed balancing is given in Appendix C6.

2.10 Lube-Oil and Seal-Oil Systems

- 2.10.3 (a) Seal oil systems of compressors handling toxic or corrosive gases or heavy hydrocarbons shall be separate from lube oil systems. This requirement shall apply even if seals are normally buffered by a clean non-contaminating gas, or if the sour seal oil is normally degassed and decontaminated before being returned to the system.

Buffer gas supplies are commonly subject to interruption, and oil clean-up systems are commonly not completely effective. The objective of this requirement is to avoid contamination of lube oil systems without placing reliance on such sub-systems. An advantage of combined systems is a reduction of compressor shaft centres by eliminating the lube to seal oil shaft separation device. This leads to improved rotor dynamic behaviour and may be essential for high density gas applications to guarantee a stable rotor system.

- (b) Seal oil systems of compressors handling flammable gases shall be separate from the lube oil system of gas turbines.

The objective of this requirement is to avoid flammable gas in the gas turbine lube oil system where ignition could occur, without placing reliance on degassing systems.

- (c) Seal oil systems of compressors handling flammable gases shall be separate from the lube oil system of HV motors of 3 kV and above unless of Ex d or Ex p construction.

HV motors of 3 kV and above are currently considered as potentially sparking in service. The risk of gas accumulation via the oil system must therefore be minimised unless the motor is purged (Ex p) or is capable of containing an internal explosion (Ex d). The limiting 3 kV level is true now but work is underway in order that this voltage level can be better defined.

- (d) When separate seal and lube oil systems are used, positive separation of the seal and bearing housings shall be provided to ensure that cross flow of seal oil into the lubricating system and vice-versa cannot occur. This shall be achieved without an external purge gas. Separation of the oil streams shall be demonstrated during the works test.



(Substitution)

- 2.10.5 Lubricating and seal oil systems shall be in accordance with BP Group GS 134-3.

(Substitution)

- 2.10.6 Oil seals of compressors handling sour gases shall be suitable for operation with high quality straight mineral oils.

(Addition)

Lubricating oil for turbo-machinery normally contains additives to minimise oxidation, foaming, emulsification and to enhance EP properties. These additives can react with non-hydrocarbon contaminants in seal oil systems resulting in deposits or plating on seals and high seal oil leakage.

In particular, zinc based additives (commonly used to enhance EP Properties) should not be used in sour gas duties containing H₂S.

Similarly, phosphor based additives, also used to improve EP properties should not be used in applications where temperatures exceed 45°C.

Straight mineral oils which contain no additives have been successfully used but must be restricted to seal systems, and the risk of oxidation of the oil minimised. The use of N₂ blanketing of seal oil reservoirs with reservoir and seal chamber vents manifolded has proved an effective barrier to oxidation.

Where these arrangements cannot be made then the use of an anti-oxidation additive may be necessary. These are stable with temperatures well in excess of those seen in bearings and seals. However, commercially available straight mineral oils do not include anti-oxidation additives, and a special formulation would be necessary.

- 2.10.7 The vendor shall state in his proposal the method(s) to be used to de-gas and clean contaminated seal oil to restore flash point, viscosity and other properties.

Such systems shall include storage facilities for at least 3 days normal consumption.

(Addition)

A number of methods are available for the de-gassing of contaminated seal oil, these are:-

- (a) Simple atmospheric degassers
- (b) 'Vacuum' Degasser
- (c) Air Stripper
- (d) Steam Stripper.

Simple atmospheric degassers cannot remove components heavier than C₅. Hence heavier gases will stay in solution and the seal oil viscosity and flash point will be reduced. Many existing heavy gas installations operate with these systems, which can, with careful operation, maintain flash points above 80°C. However, following a number of safety incidents, many offshore operators consider any reduction in



flash point below 120°C as unacceptable and on existing installations are either dumping the contaminated seal oil or retrofitting more effective degassing devices.

Vacuum degassers have been applied on many installations in the N. Sea. The equipments is complex and expensive and requires careful monitoring in operating.

The air stripping column was developed in an endeavour to overcome the shortcomings of the vacuum degasser. The principle is simple ie. the oil cascades down a column in counterflow to air. The single pass operation proved very effective. However, the final design proved to be equally complex and expensive because of the need to monitor and control the gas levels in the exist air below the LEL and provide purging on loss of the normal air blower.

The steam stripper works on the same principle as the air stripper and because the steam is inert, the need to monitor the LEL is eliminated. However, the unit requires a steam generator and potable water supply. The units has proved very effective in trials.

The more severe degassers all to some degree strip additives from the oil. Where a separate seal oil system is installed this may not have any practical import, but with a combined lube and seal oil systems it is essential that the correct additive levels are maintained, necessitating their regular monitoring and replenishment.

Sour gas can be removed by simple atmospheric degassing, and the recovered oil recycled to a separate seal oil reservoir. If the reservoir is purged by N₂ this will alleviate the possible accumulation of toxic H₂S levels.

2.11 *Materials*

2.11.4.1 *The NDT requirement for piping should be based on a criticality concept based on size, pressure and temperature rating, materials and service conditions.*

3. ACCESSORIES

3.1 Drivers

3.1.3 *Process conditions at start-up have a significant effect on the compressor run-up speed-torque curve, as does the ratio of suction side to discharge side volumes when starting blocked-in on recycle.*

It is normal to assume start-up from settle-out pressure is required in order to minimised re-start intervals following a pressurised shutdown. If this leads to excessive motor sizing then a depressurised start may be necessary.

3.1.4 *Steam turbines shall be sized to deliver continuously at least 112% of the maximum power (including gear, fluid coupling, or other losses, as applicable) required for the purchaser's specified operating conditions while operating at a corresponding speed with the specified steam conditions. They shall also be capable of delivering continuously 102% of the maximum power (as above) with the worst steam conditions.*

(Substitution)



Special purpose steam turbines should conform to BP Group [GS 134-7](#) which supplements [API 612](#).

- 3.1.5 For motor-driven units, the motor nameplate rating (exclusive of service factor) shall be at least 112% of the greatest power (including gear, fluid coupling, or other losses, as applicable) required for any of the specified operating conditions.

(Substitution for 1st sentence)

- 3.1.6 Gas turbine shall be selected to have a site base load rating of at least 115% of the greatest power (including gear, fluid coupling, or other losses, as applicable) required for any of the specified operating conditions.

The site base-load rating shall be determined at:-

- (a) Average site ambient pressure.
- (b) That ambient temperature that is exceeded for only 5% of the year.
- (c) Design (clean) inlet and exhaust pressure losses, including waste heat recovery systems if applicable.

(Substitution)

Gas turbines should conform to BP Group [GS 134-12](#) which supplements [API RP 11 PGT](#).

- 3.1.7 *Speed increasers and reducers should conform to BP Group [GS 134-10](#) which supplements [API 613](#).*

- 3.1.8 *Motor drives should conform to BP Group [GS 112-4](#).*

3.2 Couplings and Guards

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

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

The distance between coupling faces may be less than the 18 inches specified in clause 2.1.3 of [API 617](#), provided that the resulting distance is of sufficient length to allow removal of coupling hubs and maintenance of adjacent bearings and seals without removal of the shaft or disturbance of the equipment alignment.

(Qualification)



Gear couplings suffer from a number of operational problems:-

- (a) Wear caused by fretting resulting in vibration.*
- (b) Relatively small radial misalignment capability which, if exceeded, may cause fretting, transmission of vibration and increased axial loading.*
- (c) 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.*

Gear couplings have advantages over flexible membrane types as they are lighter, which may be of value when rotordynamic design is difficult. Also they have greater axial movement capability.

When these couplings are used they should be of a design which permits the inspection of the teeth without disturbing hubs. They shall also be of anti-sludging design and be lubricated via dedicated 2 x 100% 2 micron filters.

Flexible membrane coupling avoid the shortcomings of gear types but they are normally of relatively large diameter necessitating careful design of the coupling housings to minimise windage heating and oil mist or oil vapour generation. These problems are more severe with diaphragm than metallic element couplings.

To minimise windage problems clearance between coupling flange and housing needs to be adequate and there must be a path for cooling air to sweep the interior of the housing without entraining oil mist or vapour remembering that the coupling flanges act like impellers drawing air in at the internal diameter and expelling it outwards. Additionally, couplings should be shrouded to minimise bolt windage.

The vendor should provide windage and heat balance calculations to demonstrate the safe level of air and guard surface temperatures, and experience should be carefully reviewed.

Personnel protection guards should be provided if (as is likely) the guard surface temperatures are greater than 60°C. (A perforated screen set 40-50 mm off the surface will suffice).

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

(Addition)

All moving parts for UK applications shall be guarded in accordance with BS 5304.

- 3.2.7 Spacers of flexible element couplings shall be positively contained from flying out in the event of failure of the flexible membranes.

(Addition)

- 3.2.8 When turbine drivers are specified couplings shall incorporate means for the continuous monitoring of torque.



(Addition)

Torque meter couplings can be an important aid to the condition monitoring of driving and driven equipment on turbine driven applications.

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

3.3 Mounting Plates

3.3.2.2 Leveling pads are required.

(Qualification)

3.3.2.8 A single continuous base plate shall be provided for compressor(s), gear and driver, unless impractical for shipping reasons.

(Substitution)

3.4 Controls & Instrumentation

3.4.1.1 [API 617](#) places the responsibility for the compressor control system including the anti-surge system with the purchaser. The vendor's responsibilities are to supply the purchaser with the necessary information to design the system. The vendor may at the purchasers option review the system for compatibility with vendor supplied equipment.

These requirements acknowledge that the anti-surge system design is strongly influenced by factors outside the control of the vendor. For example the process parameters and their variation in operation as well as the sizing and configuration of the process equipment. However in practice the responsibilities are distributed to suit the requirements of the application.

The system needs to be configured to satisfy the specific functional requirements of the application which may include:-

- *Variation in process parameters*
- *Side streams*
- *Parallel operation*
- *Minimising surge margins to reduce power losses.*

These demands may require a level of expertise not available to the purchaser or the vendor and a specialist control vendor may then be appointed either by the purchaser or the vendor to supply the system.

All systems must have the following characteristics:-

- *Rapid response especially when operating at the surge control point*
- *Be stable when operating in recycle*
- *Protect against surging on tripping of the unit.*

The severity of a surge is increased as the head, and densities increase and as the discharge volumes upstream of the check and anti-surge valve increase. The



response time required by the system (controllers and valves) to protect against surge reduces as the discharge volumes upstream of the check and anti-surge valves increase.

The process parameters are fixed by the application. However the purchaser has some control of the discharge volumes and may need to work closely with the system designer to optimise these volumes and the system response times. This may require the use of computer simulations of the process and anti-surge systems. The system design expertise required for these applications may not be available to the purchaser or the vendor and a specialist control vendor may be appointed by the vendor or the purchaser.

In other cases the process parameters, sizing and configuration of the process equipment are commonplace and permit the use of the vendor's or purchaser's standard system with a minimum of routine design effort.



3.4.1.3 Instrumentation shall be in accordance with BP Group [GS 130-2](#).

3.4.2.1 VIGVs (Variable Inlet Guide Vanes) offer a control means for fixed speed machines comparable in range and efficiency to variable speed control. The effectiveness of VIGVs decreases with the number of impellers.

VIGVs have not found wide spread application and are available only from European vendors. Their principal application has been on multi-stage refrigeration duties, in sizes up to 900/1000mm.

The perceived complication of the device and its limitation to clean duties have restricted wider adoption. The improved efficiency over suction throttling, and relative compactness compared to throttle valves on large volume flow application may make them attractive where proven design exist. They are widely used on axial compressors.

3.4.3.1 Typical Instrumentation required for the compressor is listed here for reference. These requirements are in addition to those called for by BP Group [GS 134-3](#).

	Indicator	Alarm	Shutdown
Inlet pressure for each section	x		
Inlet temperature for each section	x		
Discharge pressure for each section	x		
Discharge temperature for each section	x	x	
Reference gas pressure	x		
Balance drum differential pressure	x		
Buffer gas differential pressure	x	x	
Recycle flow for each section	x		
Compressor speed	x		x
Shaft vibration at each bearing	x	x	
Rotor axial position	x	x	x
Bearing drain oil temperature	x		
Thrust bearing metal temperature	x	x	
Radial bearing metal temperatures	x	x	
Manual local shutdown			x
Remote shutdown			x



Notes:

- (a) *For side stream machines, sufficient flow indicators shall be provided to allow the flows through each compressor section to be deduced.*
- (b) *Tapping points for section inlet and outlet pressures and temperatures shall be sufficiently removed from the compressor to ensure accurate readings.*

3.4.7.2 Vibration and axial position monitors shall be supplied and calibrated in accordance with [API Standard 670](#).

(Substitution)

3.4.7.3 Bearing temperature monitors shall be supplied and calibrated in accordance with [API Standard 670](#). Each sensor shall have an installed spare. Sensors shall be securely fixed in intimate contact with the bearing metal and located at the bearing 'hot spot'.

(Substitution)

3.4.7.3 *Bearing metal temperature sensors give the most accurate indication of the bearing temperature. They are particularly useful on thrust bearings, where increasing temperatures (at a given speed) indicates increasing load from fouling, balance drum wear, or bearing lacquering. Other faults such as inadequate lubrication or abrasive ingress will also increase temperature. On journal bearings monitoring temperature can aid in diagnosing misalignment, lubrication problems, lacquering, or abrasive ingress.*

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 installation proposed by the vendor should be reviewed to ensure that thrust sensors are in a 75/75 position and journal sensors are on the hot spot of the bearing. Sensors should be in intimate contact with the white metal, and secure to avoid false readings. Spring loading or epoxy embedding have proved successful. Embedding in the white metal is not essential.

3.5 Piping and Appurtenances

3.5.1.5 The piping requirements of BP Group [GS 134-3](#) shall apply to all lubricating-oil, seal-oil and control-oil piping provided by the vendor.

(Substitution)

4. INSPECTION, TESTING & PREPARATION FOR SHIPMENT

4.1 General

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

- 4.1.5 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)

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 as might arise from statutory or insurance reasons should be taken into account.

Typical inspection activities pertinent to centrifugal compressors 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, impellers and shafts should normally be examined. They should be readily available being a requirement of API 4.2.1.*

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 impellers 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 offshore or otherwise onerous or if long term storage is required.*

4.2 *Inspection*

4.2.3 *The type and extent of non-destructive examination should be agreed in detail with the vendor.*

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

The following guide lines are given for the purpose of assessing the vendor's proposals:-

- | | | |
|------------------|---|---|
| <i>Shafts</i> | - | <i>100% Ultrasonic</i> |
| <i>Impellers</i> | - | (a) <i>100% Ultrasonic on shroud and hub forging.</i> |
| | - | (b) <i>100% Magnetic Particle on Welds.</i> |
| | - | (c) <i>100% Liquid Penetrant on welds.</i> |
| | - | (b) & (c) <i>Before and after overspeed test.</i> |

Casings:

- | | | |
|---------------------------|---|--|
| <i>Cast Casings</i> | - | <i>100% Magnetic Particle at cast intersections.</i> |
| | - | <i>100% Radiography of welds.</i> |
| <i>Fabricated Casings</i> | - | <i>100% Radiography of welds.</i> |
| | | <i>100% Magnetic Particle of welds.</i> |
| <i>Forged Casings</i> | - | <i>100% Radiography of welds.</i> |
| | | <i>100% Magnetic Particle welds.</i> |

4.3 **Testing**

4.3.1 The following tests are required:-

- (a) Hydrostatic test in accordance with 4.3.2.
- (b) Impeller overspeed test in accordance with 4.3.3.
- (c) Mechanical running test in accordance with 4.3.4.
- (d) Assembled compressor gas-leakage test in accordance with 4.3.5.



- (e) Performance test in accordance with 4.3.6.1 for the first off each type.
- (f) Complete-unit test in accordance with 4.3.6.2 for all offshore compressors.
- (g) Helium test in accordance with 4.3.6.5 for all cast pressure containing parts for gases containing hydrogen at a partial pressure of 5 bar abs or higher.
- (h) Sound level test for the first off each type.
- (i) Post-test inspection at hydraulically fitted couplings, in accordance with 4.3.6.10.

(Substitution)

4.3.4.2.4 Shaft seals not subjected to maximum pressure during the performance test shall be pressurised with a suitable gas to the maximum pressure against which they may have to operate (see 2.8.1), to check the integrity of the shaft seals and their ancillary systems. Checks shall include measurement of seal oil flow rates both inward towards the process and outward towards atmosphere. The shaft shall be rotated at the maximum practical speed during the test.

This test may be combined with the leak test of clause 4.3.5 if pressure levels are compatible.

(Qualification)

4.3.4.2.5 Lube-oil and seal-oil temperatures shall be held for at least half an hour at the value corresponding with 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 4.3.4.3.2 checking in particular for oil film instabilities.

(Qualification)

4.3.4.3.2 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)

4.3.4.3.6 &
4.3.4.3.7

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.

- 4.3.4.4.1 On completion of testing, sufficient dismantling will be required to permit complete inspection of all bearings and gears. Additional dismantling will be required for inspection of other components such as shaft seals and internal labyrinths, if the testing has given rise to doubts on the integrity of such components. If this additional work involves dismantling pressure containing parts, the leak test of API 4.3.5 shall be repeated after final reassembly.

(Substitution)

- 4.3.6.1.1 The [ASME PTC 10](#) Reynolds Number correction method shall not be used.

For variable speed compressors, a minimum of 5 test points shall also be taken at maximum continuous speed and again at minimum operating speed.

(Qualification)

A Reynolds Number correction method based on the work of the International Compressed Air and Allied Machinery Committee (ICAAMC) will normally be acceptable.

- 4.3.6.1.3 For variable speed compressors, a speed other than the normal speed may be used if necessary to achieve the specified performance and performance tolerances, provided the following conditions are met:-

- (a) The adjusted speed and those of all specified operating conditions meet the criteria specified in 2.9.
- (b) Maximum continuous speed meets the requirements of 1.4.5.
- (c) Trip speed meets the requirements of 1.4.7.

(Substitution)

- 4.3.6.1.4 For constant speed compressors, the capacity shall be as in 4.3.6.1.2. The head shall be within the range of 100-105% of the normal head. The horsepower at the specified capacity and actual head shall not exceed 104% of the specified value.

(Substitution)

- 4.3.6.2 & 4.3.6.9 *Complete Unit Test and Full-Pressure/Full-Load/Full-Speed Test.*

Complete unit tests will normally be at limited load. However for high pressure compressors where the aerodynamic cross coupling forces can be important (see Appendix C4) full-pressure/full-load/full-speed tests are valuable in verifying stable operation of the compressor rotor.



There is an increasing trend towards full-pressure/full-load/full-speed testing of all compressor trains where the location is isolated e.g. Off-shore and remote on-shore areas. The cost and delivery extension for these tests may be justified by forstalling site problems and hence expediting commissioning. Areas verified by full-pressure/full-load/full-speed test not covered by no load tests are:

- (a) Complete rotordynamic behaviour.
- (b) Compressor Thrust Bearing Loads
- (c) Gear box: Bearing temperatures, noise, and tooth contact.
- (d) Hot alignment.
- (e) Transient behaviour of seal oil systems during starting, tripping, pump change-over etc.
- (f) Accurate surge detection (if Class I test).

Full-pressure/full-load/full-speed tests should not be confused with [ASME PTC10](#) Class 1 tests. Full-pressure/full-load/full-speed tests are intended to simulate the aerodynamic and mnechanical conditions experienced in service. Alternatives to the service gas may be used, e.g. inert mixtures may be substituted for flammable. Class 1 tests are intended to establish the thermodynamic perofrmance using the service gas at conditions very close to those in service. Class 1 tests will of necessity closely match the requirements of full-pressure/full-load/full-speed tests but the converse in not true.

For multiple unit orders full-pressure/full-load/full-speed testing is unlikely to be justifiable for more than one train.

5. VENDOR DATA

5.1 Proposals

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

- (a) Equations of state and thermodynamic procedure used in the estimation of compressor performance on hydrocarbon duties.
- (b) Justification for the use of combined lubricating and seal-oil systems.
- (c) Method proposed for degassing and cleaning contaminated seal-oil.

(Addition)

5.2 Contract Data

- 5.2.3.9 Drawings shall be provided of main casings and other pressure containing parts, together with information detailing the vendor's previous experience with components of similar design, subject to



similar temperatures and pressures. These drawings and data shall be sufficiently detailed to provide assurance that components will safely withstand design and test pressures.

Where previous experience is insufficient to provide this assurance, detailed stress calculations or alternatively hydrostatic test experience data on components of similar design will be required.

(Addition)

- 5.2.4.1 All curves, both estimated and test, shall show all operating points and limits of stable operation from minimum operating speed to trip speed for each gas composition handled. Pressures, flows and temperatures shall be based on conditions at casing nozzles. Flows shall be net figures after allowance for balance piston and other recycle flows. For sidestream machines, the balance piston flow and nozzle to impeller pressure losses that have been used, shall be stated.

(Addition)

- 5.2.5.2(b) Certified copies of test data for all shop tests shall be provided prior to shipment.

(Substitution)

- 5.2.5.8 The vendor shall provide bearing performance data as detailed in clause 2.7.3.7 of the Specification.

(Addition)

- 5.2.5.9 Detailed test schedules for mechanical running tests, performance tests, and all other shop tests shall be supplied prior to the tests. These schedules shall list all test activities with durations, measurements to be made, instruments to be used with associated calibration procedures, inspections to be carried out, driver and coupling provision.

Performance tests and schedules shall include a statement of objectives, class of test, operating conditions, test gas, definition of all performance points, piping and driver arrangement, instrumentation, limitations on test and deviations from tests code rules, methods of computation, and estimates of possible errors.

(Addition)

- 5.2.5.10 The vendor shall provide information on pre-commissioning methods and limitations as required in clause 2.1.16 of this Specification.

(Addition)

- 5.2.7 Installation and Instruction Manuals.

- 5.2.7.3(g) Instructional manuals shall include a schedule of alarm and trip settings, with procedures for checking these.



(Addition)



DEFINITIONS AND ABBREVIATIONS

Definitions

Standardised definitions may be found in the BP Group RPSEs Introductory volume

purchaser: a contractor acting on behalf of BP, or BP itself in the case of a direct purchase.

vendor: the main supplier of the machinery to which this Specification applies including items designed and manufactured by others.

Note: Any specific application of the terms and the responsibilities of the parties defined above is a matter for the relevant Conditions of Contract.

sour service: as defined in NACE MR-0175 plus all applications with more than 10 mol% H₂S.

Abbreviations

API	American Petroleum Institute
ASME	American Society of Mechanical Engineers
NACE	National Association of Corrosion Engineers



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 612	<i>Special-Purpose Steam Turbines for Refinery Services</i>
API 613	<i>Special-Purpose Gear Units for Refinery Services</i>
API RP 11 PGT	<i>Recommended Practice for Packaged Combustion Gas Turbines</i>
API 617 , Fifth Edition, April 1988	Centrifugal compressors for general refinery services
API 670	Vibration axial - position, and bearings - temperature monitoring systems
API 671	Special-Purpose Couplings for Refinery Service
NACE MR-0175	Standard material requirements - sulphide stress cracking resistant metallic materials for oilfield equipment
BS 5304	<i>Code of Practice for Safety of Machinery</i>
BP Group GS 112-4	<i>High Voltage Induction Motors</i> (was BP Std 220)
BP Group GS 130-2	Instrumentation and electrical equipment for rotating machinery (was BP Std 128)



BP Group GS 134-3	Lubrication, shaft sealing and control oil systems for special purpose applications to API 614 (was BP Std 190)
BP Group GS 134-7	<i>Special Purpose Steam Turbines to API 612</i> (was BP Std 198)
BP Group GS 134-12	<i>Packaged Gas Turbines to API RP 11 PGT</i> (replaces BP Std 204)
BP Group GS 134-13	<i>The Packaging of Rotating Machinery for Offshore Use</i> (was BP Std 205)
BP Group GS 136-1	<i>Materials for Sour Service to NACE Standard MR 017590</i> (was BP Std 153)



SUPPLEMENTARY COMMENTARY

C1 Procedure to Determine Impeller Eye Mach No.

This Commentary relates to clause 2.1.18.

The Impeller Tip Mach No. (M_t) is readily calculated:-

$$M_t = U_2/A_o$$

where

$$U_2 = \text{Impeller Tip Speed}$$

$$A_o = \text{Speed of Sound for the Compressor Inlet Nozzle Conditions}$$

M_t can be related to M_e by the flow coefficient f the impeller to shaft diameter ratio K and the inlet blade angle b where:-

$$f = \frac{\text{Inlet Volumetric Flow}}{p/4 U_2 D_2^2}$$

$$K = \frac{\text{Impeller Tip Diameter}}{\text{Shaft Diameter}}$$

$$D_2 = \text{Impeller Tip Diameter}$$

The relationship between M_t , M_e , f and K is shown in Figure C1. The inlet blade angle b has been assumed at 60° , which is close to the optimum value for 3-dimensional inducer impellers as used in these services. Reducing b to 50° has only a second order effect. Below 50° the effect becomes increasingly pronounced.

Impeller tip to shaft diameter ratios (K) of 2.5 to 4.0 are shown. These cover the range commonly encountered on actual machines.

For preliminary estimates a value of 3 can be assumed for K . For an eye Mach No of 0.8, this leads to the following impeller tip Mach Nos depending on flow coefficient:-

f	0.15	0.12	0.11	0.10 or less
M_t	1.00	1.06	1.08	1.11

For flow coefficients less than 0.10 a nominal limit on M_t of 1.11 should be applied, since the inlet angle b may be reduced below 50° .



C2 Thrust Bearing Design

This Commentary relates to clause 2.7.3.7.

- (i) *The oil film thickness can be determined by calculation. It is a function of pad shape, dimensions, speed, viscosity, bearing oil supply (ie. 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 are tabulated below for centre and offset pivots.

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

The load applied should be limited to 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.

- (ii) *The bearing lining metal temperature is a function of the same parameters as oil film thickness. In addition offset pivots reduce oil temperature.*

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 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 temperature limit depends upon the lining metal. For 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 and also from the degradation of phosphorous based anti-oxidants and EP additives. For these applications the pad temperature should therefore be limited to less than 100°C (say, 95°C).

Pad temperatures can be reduced as outlined earlier for oil film temperature reduction, in addition load reduction may be possible. Higher temperatures should not be accepted without a review of specific site experience including water ingress prevention and removal measures.

C3 Gas Seals

This Commentary relates to clause 2.8.3.5.

The requirement for a gas tight system even on failure of the primary seal stems from a desire for the same degree of sealing integrity as has been customarily available from systems using oil seals. The primary elements of oil seals, oil lubricated bushings or contacting faces, are effectively backed-up by the seal oil system itself. If the primary element fails, gas leakage is prevented (for a limited period of time) by an increase in seal oil flow. Failure of the seal oil supply itself is unlikely as critical components are normally spared.

Self-acting gas seals are more likely to suffer a major failure as they incorporate highly stressed brittle materials such as tungsten carbide and silicon carbide. A number of catastrophic failures of these components has occurred. They are typical of problems that arise during the development of new designs, and should not be assigned undue importance. Lessons are being learned and the incidence of such failures in the future should be low. however, experience is still relatively limited and the conservative specification is justifiable at the present time for systems handling hazardous gases.

Some limitations of self-acting gas seals are well known. In particular, pressure, diameter, speed limits are dictated primarily by strength and are usually clearly stated by the seal vendor.

Temperature limits are generally dictated by the material of the dynamic secondary seal, usually an elastomeric O-ring behind the stationary face. Cooling of gas seals requires special attention as, unlike oil seals, there are no large sealant flows to take away heat conducted along shafts or seal housings.



O-rings can impose additional limitations of hang-up at low differential pressures, and explosive decompression from high pressures.

Reverse pressure differentials cannot normally be accommodated. They will either overload seal faces causing damage, or force the faces apart causing high leakage.

Most seal designs incorporate spiral grooves and are uni-directional. Reverse rotation capability will be very limited, and this places increased importance on positioning non-return valves and block valves such that the risk of reverse rotation is minimised (See also 2.7.1.3).

Taking into account the points made above, a typical installation for a flammable gas duty within the pressure limits of a single seal, would include the following:-

- (a) Tandem arrangement, with each seal having full pressure capability.*
- (b) Clean process flush at the lowest practical temperature, to the inboard seal, with low flow alarm.*
- (c) Interspace between seals vented via an orifice to atmosphere or flare. Vent orifice sized to develop a pressure differential of approximately 1 bar.*
- (d) Alarm and trip on high interspace pressure.*
- (e) Outboard purge between lube oil system and seal, with low flow alarm.*
- (f) Outboard purge preferably N₂, otherwise air.*
- (g) Outer seal gas leakage (and purge) vented to atmosphere; via a flame arrester if the purge is air.*

C4 Self-Excited Vibration

This Commentary relates to clause 2.9.2.8.

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

- (a) The journal bearings*
- (b) Shaft labyrinths*
- (c) Impeller Tips*
- (d) Impeller Shrouds*
- (e) Shrink fits*
- (f) Liquid film shaft seals*

Particular attention needs to be paid to rotors that are very flexible, ie. the ratio of first flexural critical speed to maximum continuous speed is low.



Additionally, applications involving high densities require analysis because (b), (c) and (d) above are density dependent. (The 70kg/m^3 limit approximates to 100 bar at 18 mol wt)

(b) and (c) and (d) above are also a function of the tangential velocity of the gas in the close clearances of labyrinths and impeller shrouds. The effects can be significantly reduced in labyrinth seals by destroying the tangential components of velocity at entry to the clearance space, a practice commonly adopted by some vendors.

The journal bearing geometry has influence on the log dec, 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:-

Type of Journal	Cross Coupling Suppression Ranking	Damping Ranking
Cylindrical	5	1
Lower bore	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.

Second order effects on stability limits are:-

- (a) Bearing clearance, increase to increase stability
- (b) Oil viscosity, increase to increase stability
- (c) Pre-load, decrease to increase stability
- (d) Bearing length, 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.

Liquid film shaft seals can generate significant cross coupling forces if the floating ring locks under its axial loading and the bore is plain cylindrical. Floating bushings should therefore be of a balanced design reducing the axial loads and consequently the radial shaft support force possible to a minimum. The bore should incorporate



features to minimise cross coupling such as, circumferential grooves, profiled bores (lobing) etc.

It is known that shrink fits introduce hysteresis, leading to cross coupling forces. These forces have not been quantified, but their effects can be minimised by relieving of long shrink fits, and ensuring no slipping occurs.

A rough assessment of a design can be made by plotting the ratio first flexural critical speed/max continuous speed against gas density for existing successful designs with the same design features from the same vendor. For cases close to or beyond this boundary a rotor stability analysis should be performed.

Stability analysis results should preferably be presented graphically to show:-

- (a) The change in logarithmic decrement with speed up to trip speed.
- (b) The change in logarithmic decrement with cross coupling forces at trip speed.

The vendor should as far as possible individually quantify the cross coupling forces at trip speed originating from all effects.

C5 Torsional Excitation

This Commentary relates to clause 2.9.4.5.

Clause 2.6.18 of this Specification refers to high torques that can be generated by a.c. motors.

Of particular concern is the variable frequency excitation at $2 \times$ slip frequency during the run up of synchronous motors. It is common for shaft systems to have at least one torsional critical speed below $2 \times$ line frequency which would therefore be excited transiently during acceleration. The transient torsional analysis is essential to determine the build up of torsional oscillations and to ensure that damaging stresses do not develop. It is a mandatory requirement of API clause 2.9.4.6

Induction motors also generate large excitation on starting, but at constant (line) frequency which should not coincide with a torsional critical speed. The amplitude of the excitation will decay during run up and the amplitude of the response is therefore best determined by a transient analysis. However, a steady state forced response analysis would suffice provided it assumed the maximum value of excitation.

Similarly, the torques arising from short circuits and, if applicable, reswitching, are of constant frequency and reducing magnitude, and the effect of them on shaft amplitudes and stress is best determined by a transient forced damped response analysis.



Variable speed a.c. motor drives are subject to torsional excitations at integer multiples of the converter frequency.

The magnitude of the torsional oscillation is a function of the pulse frequency and the harmonic of the converter frequency. Typical values are (%):-

Harmonic	6 Pulse	12 Pulse
5	25	1
6	20	-
7	9	1
11	8	8
12	-	4
13	5	5
17	4	1
19	3	1
23	3	3
25	2	2

C6 High and Low Speed Balancing

This Commentary relates to clause 2.9.5.4.

Low speed incremental balancing can result in consistently acceptable vibration levels at operating speeds if carried out thoroughly, accurately and with many steps. High speed balancing has the potential of achieving very low vibration levels at operating speeds and, depending on the particular situation, may take less time and effort than the low speed incremental method.

High speed balancing has potential difficulties if the operating speed range is wide and a critical speed must be traversed and the number of balancing planes is limited.

In general, to be able to achieve a theoretically perfect state of balance at high speeds:-

$J > KL$ where

J = number of correction planes
 K = number of speeds to be balanced
 L = number of measurement planes.

For example, if the vibration is to be minimised at three speeds (perhaps trip speed, minimum operating speed and the first critical speed) and there are two vibration measurement points, then seven planes are required theoretically to achieve a perfect balance at each speed.



In practice there are often significant restraints on the number of balance planes and they may not be ideally located with respect to the mode shape. A compromise in the vibration achieved at some or all of the speeds is usually necessary.

If the distribution of correction planes is inappropriate then relatively large mass corrections may be required at planes some distance from the anti-nodes to balance successfully at high speeds, leading to a degradation in the low speed residual unbalance results. The success of high speed balancing can not be assessed using low speed residual unbalance measurements.

The high speed acceptance criteria is difficult to establish and little guidance is available from published standards. The high speed flexible rotor residual unbalance (that is the distance between the mass and geometric centres) can not be determined and the balance must be assessed by a vibration measurement. This measurement should be either a rotor displacement or bearing housing velocity. Experienced personnel may provide advice on achievable tolerances, eg. a figure of 1 mm/s maximum bearing housing velocity from the figure critical to the operating speed might be mutually agreed as an acceptance criteria.

It is desirable to use a balancing facility which mimics the design rotor-bearing assembly but this is not generally practical. The contract bearings may be used to retain some of the design features, however, the response of the rotor in a balancing facility will also be affected by bearing housing support stiffness (which will usually be lower than the design) and coupling characteristics.

The application of a computerised influence coefficient technique is usually available when a high speed balancing facility is employed. Even so, the success of balancing depends largely on the skill and competence of personnel in taking measurements and in applying trial and correction masses just as in any manual balancing, particularly when correction plans are limited in number and not ideally located.

Vendor experience with rotors of similar design should be assessed. Particular attention should be paid to the axial distribution of mass (potential unbalance), location of correction planes, and rotor mode shapes at the speed of interest.



COMPRESSOR PERFORMANCE ESTIMATION

Flow Coefficient vs. Mach Number Ratio

60 Degree Blade Angle @ Eye

Mach Number Ratio (Tip /Eye) m_t/m_e

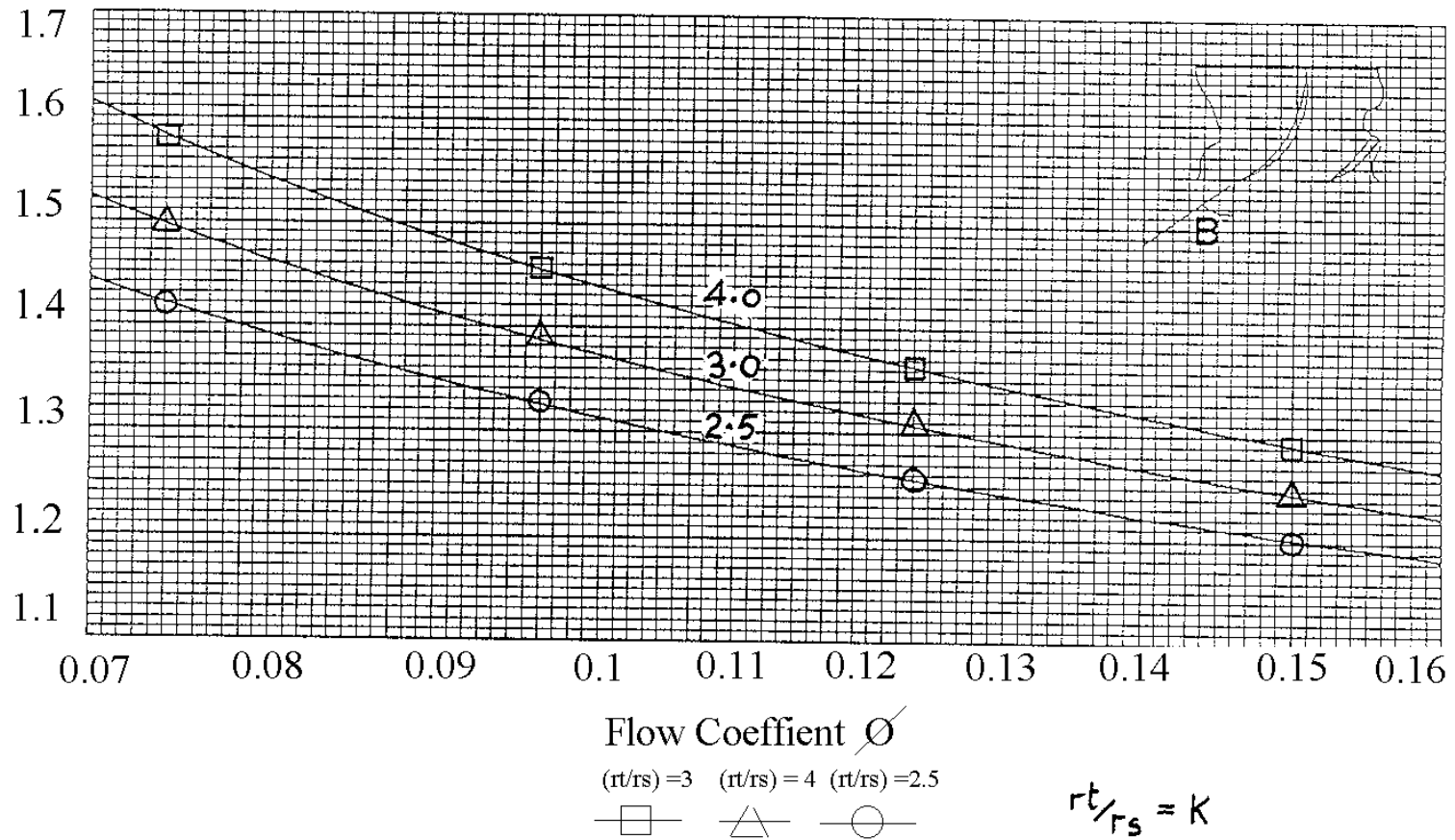


FIGURE C1
RELATIONSHIP BETWEEN M_t , M_e , f and K