



CGA G-4.13—2023 CENTRIFUGAL COMPRESSORS FOR OXYGEN SERVICE

SECOND EDITION

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PREFACE

As part of a program of harmonization of industry standards, the Compressed Gas Association (CGA) has issued CGA G-4.13, *Centrifugal Compressors for Oxygen Service*, jointly produced by members of the International Harmonization Council and originally published by the European Industrial Gases Association (EIGA) as EIGA Doc 27, *Centrifugal Compressors for Oxygen Service*.

This publication is intended as an international harmonized standard for the worldwide use and application of all members of the Asia Industrial Gases Association (AIGA), Compressed Gas Association (CGA), European Industrial Gases Association (EIGA), and Japan Industrial and Medical Gases Association (JIMGA). Each association's technical content is identical, except for regional regulatory requirements and minor changes in formatting and spelling.

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1 Introduction

This publication has made a significant contribution to the safe compression of oxygen primarily because the manufacturers and users have shared their philosophies and experiences. It is recognized by the Working Group members that the feedback of operating experiences makes a powerful contribution to safe operation and design.

Oxygen compression represents a risk in that the compressor can burn violently. This publication defines design and operating parameters for centrifugal oxygen compressors. Compliance with this publication can reduce the likelihood of, and the hazards arising from, a fire in a compressor to a level equal to or less than those commonly accepted in the air separation industry.

The publication requires that all those who build and operate centrifugal oxygen compressors specified to comply with this publication should contribute towards it by fully reporting the circumstances surrounding oxygen fires. For the purpose of safe operation of the compressor and its auxiliaries, the user and the manufacturer shall establish full agreement on the possible and expected modes of compressor operation (for example, specified operating points, normal operating range, startup and shutdown, etc.)

2 Scope and purpose

This publication applies to single shaft horizontally split casing and vertically split casing (barrel) type compressors and integrally geared compressors. Operating experience exists in compressors up to 71 000 Nm³/hr (2 700 000 scfh) at pressures up to 1230 psi (8.5 MPa) for single shaft compressors and up to 650 psi (4.5 MPa) for integrally geared compressors with oxygen purity of 90% or greater and with maximum 10 ppm water (volume basis).^{1, 2}

The use of axial compressors in oxygen service is not covered by this publication.

The purpose of this publication is to provide guidance on the design, manufacture, installation, operation, and maintenance of centrifugal oxygen compressors, thereby safeguarding personnel and equipment. Fire in an oxygen compressor can be caused by a variety of reasons, which include mechanical deterioration resulting in excessive vibration and/or loss of running clearances within the compressor, ingress of oil (for example, through the seal system), foreign bodies passing through the machine, and surge.

An oxygen compressor shall be provided with a safety support system that shall minimize the development of a potentially dangerous operating condition. In the event of an incident on the compressor that results in combustion of the materials of construction, the safety systems shall be designed to minimize the effect of the fire.

The safe and reliable compression of oxygen using centrifugal compressors can only be achieved by the successful combination of many factors. This publication identifies and addresses the following factors.

2.1 Management of hazards

- identification of the hazard;
- hazard area, safety barriers, and enclosures;
- location of the compressor; and
- emergency procedures.

¹ psi, bar, and kPa/MPa shall indicate gauge pressure unless otherwise noted as (psia; bar, abs; and kPa/MPa, abs) for absolute pressure or (psid; bar, dif; and kPa/MPa dif) for differential pressure. All kPa values are rounded off per CGA P-11, *Guideline for Metric Practice in the Compressed Gas Industry* [1].

² References are shown by bracketed numbers and are listed in order of appearance in the reference section.

2.2 Design of the compressor system

- robust and well proven compressor design;
- stable rotor system;
- safe materials in critical areas;
- comprehensive instrumentation;
- safety shutdown system;
- auxiliary system; and
- control system including surge protection.

2.3 Cleaning, preservation, and inspection

- correct and properly enforced procedures carried out by trained personnel.

2.4 Erection, testing, and commissioning

- trained erection personnel; and
- comprehensive testing program to verify the design.

2.5 Operation

- trained personnel; and
- correct procedures.

2.6 Maintenance

- condition monitoring;
- planned preventive maintenance; and
- trained personnel.

2.7 Other specifications

For more information, see CGA P-8, *Safe Practices Guide for Cryogenic Air Separation Plants* [2].

In case of conflict between this publication and the user's specification, the information included in the order shall be the more stringent. The supply shall be in conformity with the rules of the country of the user and/or of the manufacturer.

See also Section 17, References.

2.8 Speed

Speeds shall be according to 1.5 of API Standard 617, *Axial and Centrifugal Compressors and Expander-compressors*, for example, rated speed, normal speed, maximum continuous speed as well as trip speed [3].

3 Definitions

For the purpose of this publication, the following definitions apply.

3.1 Publication terminology

3.1.1 Shall

Indicates that the procedure is mandatory. It is used wherever the criterion for conformance to specific recommendations allows no deviation.

3.1.2 Should

Indicates that a procedure is recommended.

3.1.3 May

Indicates that the procedure is optional.

3.1.4 Will

Is used only to indicate the future, not a degree of requirement.

3.1.5 Can

Indicates a possibility or ability.

3.2 Technical definitions

3.2.1 Dry, oil-free, air or nitrogen

Air or nitrogen with a dew point of -40°F (-40°C) or less and an oil content of 0.5 mg/m^3 or less.

3.2.2 Hazard area

Area where an incident is most likely to occur and as a consequence is capable of causing danger and/or injury to personnel.

NOTE—The term hazard area is not to be confused with electrical hazardous area classification.

3.2.3 Maximum operating temperature

Highest temperature that can be measured anywhere in the main gas stream under the most severe operating conditions.

3.2.4 Seal gas

Ambient temperature dry (dew point of less than -40°F [-40°C]), oil-free, particle-free, and carbon dioxide-free (less than 3 ppm) nitrogen, argon, or air.

NOTE—Also known as buffer gas.

4 Compressor design

4.1 Design criteria

4.1.1 Possible causes of an oxygen compressor fire

It is normally very difficult to precisely ascertain the cause of a fire in an oxygen compressor because the material at and around the ignition site is completely burnt up. Therefore, during the design, manufacture, and maintenance of centrifugal oxygen compressors both active and passive safety measures shall be taken to guard against all of the causes of ignition.

Causes of ignition, sources of friction, or foreign material can include:

- mechanical rub—improper design, clearances, vibration, operating pressure, assembly or maintenance errors, bearing failure, thrust, alignment, improper intercooling, transient instability such as startup, shutdown, or process upset (can include shock and surge);

- large debris impact—screen/filter failure or improper mesh size, weld debris or slag, friction/shock, maintenance debris, shot, sand;
- debris—screens, weld debris or slag, oxides such as rust, maintenance debris, shot, sand;
- contamination—oil, assembly grease, improper design of bearings/seals and/or associated vents and drains, deposits from cleaning agents;
- resonance—debris in dead areas;
- nonmetallic materials with a low autoignition temperature; and
- long-term deterioration of components, key tolerances, etc.

4.1.2 Machine configuration

Compared with single shaft compressors with closed wheels, integral gear compressors can have open and/or closed wheels and have the following design differences:

- Greater number of seals letting down to atmospheric pressure and in close proximity to the gearbox;
- More complex assembly;
- Stable rotor dynamic design of integrally geared compressors is more difficult to achieve than in single shaft compressors, in general. Therefore, additional consideration in the rotor dynamic design shall be taken to provide acceptable rotor stability in oxygen service; and
- Greater sensitivity to upset condition such as surge.

4.2 General materials

4.2.1 Construction materials

When selecting materials of construction for an oxygen compressor, components that come into contact with oxygen shall have oxygen compatibility. Materials that fulfil these criteria usually have the following properties:

- high ignition temperature;
- high thermal conductivity;
- high specific heat; and
- low heat of combustion.

4.2.2 Use of aluminum

Because of its high heat of combustion, the use of aluminum or alloys containing aluminum shall be limited for oxygen wetted or potentially oxygen wetted parts. Aluminum will not sustain combustion at less than certain pressures and purities. The use of aluminum shall be limited to a pressure less than or equal to 0.2 MPa (29 psi) for the oxygen purity range covered by this publication.

In addition, the maximum permitted aluminum content in a copper alloy is 2.5%.

4.2.3 Oxygen compatibility of nonmetallic materials

Nonmetallic materials shall be tested for their suitability for oxygen service. Information can be found in M 034-1 List of non-metallic materials compatible with oxygen by BAM and ASTM G63, Standard Guide for Evaluating Nonmetallic Materials for Oxygen Service [4, 5]. This does not preclude other methods of determining compatibility such as by other independent bodies, customers, and suppliers.

4.3 Compressor parts

4.3.1 Casings

4.3.1.1 Casing allowable working pressure

Calculations shall be carried out to determine the maximum pressure that the casing can experience during operation. It shall be the highest pressure of the following options that can be reached in the casing (or subdivision of casings into chambers) multiplied by an agreed safety factor between the user and the manufacturer.

- The maximum operating pressure at the surge limit resulting from the maximum specified suction pressure at the maximum continuous operating speed. Agreed deviations from gas properties and suction temperature are to be considered;

NOTE—In some instances a rotor stability test at greater than the maximum design operating pressure is specified. If this is the case, it should be taken into account when specifying the casing allowable working pressure;

- The maximum pressure that results from the maximum specified suction pressure and the greatest pressure rise possible with the given maximum drive power at the maximum continuous operating speed. Agreed deviations from gas properties and suction temperature are to be considered;
- The maximum equilibrium pressure reached in the compressor system under certain running or shutdown conditions; and
- If the casing pressure is limited by a safety device set to a pressure agreed between the user and manufacturer, this pressure can be used as the casing allowable working pressure. The casing may also be subdivided into chambers for calculation and testing. In this case, the maximum possible pressure in these chambers is to be used as a basis, taking into consideration the aforementioned aspects.

4.3.1.2 Pressure tests

The compressor main casing or volutes shall be hydrostatically tested in the manufacturing facility with potable water at a minimum test pressure of 1.3 times the allowable working pressure of each portion of the casing. The casing allowable working pressure is defined in 4.3.1.1.

The test pressure shall be held for at least 30 minutes to permit complete examination of the casing under pressure. Casings and volutes that leak under hydrostatic test shall not be acceptable.

4.3.1.3 Casing material

The following materials have proved satisfactory with regard to the criteria listed in 4.2:

- grey cast iron;
- nodular cast iron;
- high alloy steel—cast or fabricated; and
- welding of cast steel and fabricated steel casings is permitted if the execution and heat treatment are properly conducted.

4.3.1.4 Casing repairs

All internal spaces of the casing should be easily accessible for cleaning and inspection. Hard soldering or metal locking repairs to cast iron casings are not permitted unless agreed between the manufacturer and the user. Minor defects in cast casings may be repaired with screwed plugs if agreed between the manufacturer and the user. This publication requires that these plugs be positively prevented from falling into the compressor. The preferred way is to use positively locked, taper plugs from the outside only. Welding repairs to grey cast iron is forbidden; however, with the permission of the user, it may be carried out on the other materials listed previously. The use of nonmetallic materials for repair work is forbidden.

4.3.1.5 Casing sealing material

If nonmetallic materials are employed for sealing the casing, they shall be oxygen compatible and agreed to by the manufacturer and the user. If used, liquid sealant shall be applied so that it does not creep or project into the inside of the machine. If required, threads shall also be sealed by materials that are compatible with oxygen.

4.3.1.6 Anti-galling compound

If an anti-galling compound is to be applied to centering fits, bolts, studs, etc., only compounds compatible with oxygen service shall be used. Compounds shall be mutually agreed between the manufacturer and the user.

4.3.1.7 External forces and moments

The compressor manufacturer shall specify the nozzle displacements due to thermal movements of the compressor. It is preferred that the permissible forces and moments on each flange/nozzle to which the user has to connect are 1.85 times the values calculated in accordance with NEMA SM 23, *Steam Turbines for Mechanical Drive Service* [6]. If this is not possible, they shall be mutually agreed between the manufacturer and the user.

4.3.2 Diaphragms and diffusers

4.3.2.1 Materials of inter-stage diaphragms and diffusers associated with closed impellers

The diaphragms shall be designed to withstand the maximum possible differential pressures. Grey cast iron and nodular cast iron have been widely used up to 725 psi (5 MPa). High alloy steel, copper alloys, nickel alloys, and copper nickel alloys are recommended greater than 725 psi (5 MPa).

4.3.2.2 Materials of shrouds (diaphragms) and diffusers associated with open impellers

It is not permitted to use open impellers with shrouds made of materials less compatible with oxygen than copper alloys or nickel alloys.

4.3.2.3 Diffusers—Design features

4.3.2.3.1 Vaneless diffuser with spiral collector

Pressure variation around the circumference of the diffuser can be powerful enough to excite the covers and backplates of the impellers. This phenomenon becomes more pronounced at surge and at stonewall conditions and more powerful at high pressures. If the diffuser is long enough, this phenomenon is avoided. Therefore, the diffuser diameter shall be greater than 1.4 times the impeller diameter.

4.3.2.3.2 Fixed diffuser vanes

- The use of fixed diffuser vanes as an integral part of the diffuser assembly in oxygen service has been proven over several years of operation. Fixed diffuser vanes can be used in oxygen service;
- The vanes are subject to strong excitation forces being close to the impeller and in an area of high velocity and changing density. Therefore, the vanes shall be subject to careful analysis to ensure that resonant modes are not excited;
- Diffuser vanes shall have no high energy excitation frequencies corresponding to multiples of the number of impeller blades and the rotating speed. The number of diffuser blades and the number of impeller blades shall have no common denominator and should preferably be prime numbers; and
- The use of vaned diffusers is not permitted unless resonance calculations have been carried out. These calculations shall be based upon test data.

4.3.2.3.3 Variable diffuser vanes

Variable diffuser vanes shall not be used. Variable diffuser vanes involve very small angular movements, tight side clearances, blades with long unsupported lengths, and complex operating mechanisms. When these are combined with high excitation forces and their physical position in the compressor, they represent a considerable additional risk.

4.3.3 Variable inlet guide vanes

The use of variable inlet guide vanes is permitted. In normal operation, the inlet guide vane (IGV) opening or closing time should be limited to maximum 1% per second of the controller output to avoid abrupt flow change and surge.

The design of the variable inlet guide vanes shall take into account:

- Excitation due to the flow disturbances caused by the stage inlet piping;
- Excitation of the impeller;
- The design shall be so either it is physically impossible for the vanes to go to the fully shut position or, if the vanes are permitted to go to the fully shut position, there shall be sufficient flow area to prevent the vanes being overloaded and to dissipate the heat caused by windage;
- They shall be of a nonlubricated design;
- The design shall avoid the risk of oxygen leakage to the atmosphere. The use of a seal gas system is recommended; and
- Suitable materials for oxygen service that shall be resistant to impingement and high velocity of gas.

4.3.4 Rotating assembly

4.3.4.1 Impellers

4.3.4.1.1 Materials

High alloy steels (not austenitic) are the materials normally used for impellers.

4.3.4.1.2 Manufacture

Impellers may be cast, forged, spark-eroded, milled, brazed, or welded. Riveted impellers shall not be used. The impellers shall be subjected to an over-speed test for 3 minutes at the speeds stated in 2.8. Following this test, the impellers shall be crack-tested and checked for dimensional changes. Two diameters on the impeller should be marked and the dimensional change for each diameter is to be found by comparing the length before and after the overspeed test. Impellers shall also be dye penetrant or magnetic particle tested. All dye and other penetrant shall be carefully removed after the test. Acceptance criteria shall be agreed between the manufacturer and the user.

4.3.4.2 Open impellers for geared compressors

- Open impellers have a smaller clearance between the rotating and stationary part of the stage, which leads to an increased risk of a high speed rub. This small clearance has to be set with the compressor cold. During transient operation, particularly during startup, different parts of the compressor heat up at varying rates and there is a danger that this will cause the impeller to touch. The risk associated with this can be reduced by ensuring that the compressor is always started up using dry, oil-free, air or nitrogen and brought close to operating temperature before changing over to oxygen;
- Open impellers are normally used in an overhung configuration. The seal system required for oxygen is quite long. The result is that the amplitude of vibration of the impeller can be large when a resonant mode of the rotor is excited. The smaller impeller to stator clearances used with open impellers results in a greater risk of a high speed rub;

- Impeller stress levels permit open impellers to be run at higher tip speeds with a resultant higher rubbing velocity; and
- The blades on an open impeller are only supported at their base; therefore, there is a greater likelihood of them being excited at a resonant frequency and failing in consequence. A frequency analysis of the individual impellers is of the utmost importance. The manufacturer shall demonstrate the location of the first three natural frequencies of the impeller and that these do not correspond to known existing frequencies due to, for example, diffuser vanes.

4.3.5 Shafts

The shafts of centrifugal compressors shall be forged from one piece and checked for defects using ultrasonic tests. The electrical and mechanical run-outs in the planes of the vibration probes shall be reduced to 6 micron peak to peak or less during the course of the manufacturing program.

4.3.6 Rotor assembly

Shaft sleeves are permissible. Components shrunk onto or fitted to the shaft shall be degreased before fitting.

For single shaft compressors, assembled rotors with shrunk on components shall be submitted to an overspeed run prior to the final rotor balance in order to release all unequal settings of components on the shaft. Whenever a rotor is rebuilt, overspeed shall be considered prior to balancing (see 4.5.1).

Thrust collars shall be machined as part of the solid shaft or positively retained using a locknut, shear ring, or grip enhancement method. The use of a simple interference fit shall not be used.

4.3.7 Seals

4.3.7.1 Internal rotor seals

Depending on the type of compressor (single shaft or integrally geared), the internal rotor sealing has the function of keeping the amount of gas leaking between impeller outlet and impeller inlet and between adjacent stages as low as possible. Adequate clearances shall be provided between sealing tips and sealing faces, so that contact is limited to an amount agreed between the manufacturer and the user under all operating conditions. The internal seals of an oxygen compressor shall only be of the labyrinth type. The design and the choice of materials for the tips and sealing faces shall be so in the event of contact the least possible amount of heat is developed and the resulting heat is readily dissipated.

The seal assembly is usually replaceable and typically held in place using a sliding "rail" fit. In such designs, consideration should be given to avoid fretting, for example, apply hard coating to the seal locating surface.

4.3.7.1.1 Rotating tips

The following materials shall be used:

- rotating tip—copper alloy or nickel alloy; and
- stationary face—silver layer bonded to a copper alloy or nickel alloy backing.

As a minimum, the thickness of the silver layer shall take into account the shaft movement that can occur in the event of:

- total bearing failure; and/or
- rotor being excited in resonance.

The silver shall be of such a thickness that the rotating tip cannot cut through the silver layer and touch the copper alloy or nickel alloy backing.

These criteria apply to both radial and axial labyrinths. It is important with this type of seal that the tips and the silver are designed in a way that ensures that the tips cut satisfactorily into the silver face.

Silver has shown itself to be a very safe material for use in seals. Experience has shown that it is safe to permit the rotating tips to cut into the stationary silver face during rotor excursions that occur during startup and surge. The amount of cut in shall be agreed between the manufacturer and the user.

For overhung impellers, for example, gear-type compressor, the benefits of a silver counter face are well established in seal design. However, a rotating labyrinth running against a silver stationary counter face can lead to violent rotor excursions in the event of rub.

4.3.7.1.2 Stationary tips

The following materials shall be used:

- stationary tip—silver mounted on copper alloy or nickel alloy base; and
- rotating face—high alloy steel, copper alloy, or nickel alloy.

The stationary tip shall be of sufficient width to provide adequate strength and of sufficient height to prevent contact between the rotating shaft and the stationary copper or nickel alloy base in the event of a rotor excursion due either to a bearing failure or rotor instability. These criteria apply to both radial and axial labyrinths.

4.3.7.2 Atmospheric rotor seals

4.3.7.2.1 Function

The function of the atmospheric portion of the gas seal is to preclude the escape of oxygen out of the compressor along the shaft as well as the possibility of the introduction of air or oil via the seal.

The seal shall be effective during all operating conditions including standstill, startup, and rundown (see Section 10).

4.3.7.2.2 Compressor atmospheric rotor seals—Labyrinth type

The atmospheric rotor seals shall be of the labyrinth type, which is the only type of seal permitted by this publication except under exceptional circumstances (see 4.3.7.2.3). With respect to design, materials, and clearances, this type of seal shall comply with 4.3.7.1.

At least three sealing chambers shall be provided. The inner chambers are connected to the suction in order to reduce the differential pressure across the seal to a minimum. The center chamber is for venting or exhausting. The outer chamber is for the supply of seal gas.

The internal pressure of outer and center seal chambers should be measured by providing separate measuring connections close to the seal chambers to ensure that the pressure measurement is affected as little as possible by the gas flow in the seal system. If the design of the compressor makes it impossible to fit separate measuring connections, when it is agreed between the manufacturer and the user, it is acceptable to measure the pressure away from the seal chambers provided that the pressure drop due to flow between the seal chamber and the measuring point is insignificant compared to the pressure being measured. The manufacturer shall provide pressure drop calculations at seal clearances that are four times design.

4.3.7.2.3 Compressor atmospheric rotor seals—Alternative types

There are certain applications such as pipeline compressors where the use of labyrinth seals presents operating difficulties (for example, where availability of seal gas is limited). Other types of seals may be considered as agreed to between the manufacturer and the user.

4.3.7.3 Bearing housing seal

The function of this seal is to prevent oxygen from getting into the oil system and to prevent oil vapor escaping from the oil system. There are no oxygen compatibility or seal material requirements.

4.3.7.4 Separation of rotor process gas seal and oil seals

Precautions shall be taken to avoid contamination of the process gas seals by oil and/or oil mist leakage or oxygen migration into the lubricated parts, which can lead to major safety hazards.

4.3.7.4.1 Single shaft oxygen compressors

An air gap open to the atmosphere between the compressor casing and bearing housing shall be provided. This shall have an arc width at least equal to the shaft diameter and large enough to guarantee atmospheric pressure in the gap and enable the shaft to be clearly viewed. Weather protection may be necessary in outdoor installations. No restriction or pipe shall be fitted to this opening. A continuously falling drain should be led from the bottom of the chamber in order to remove oil and detect leaks. The size of the drain shall be large enough to avoid the risk of blockage.

4.3.7.4.2 Integrally geared oxygen compressors with an air gap

Some integral gear compressor designs have an air gap open to the atmosphere. If they have an air gap open to the atmosphere, they shall be in accordance with 4.3.7.4.1.

4.3.7.4.3 Integrally geared oxygen compressors without an air gap

If there is no air gap, additional instrumentation shall be installed according to Section 10. The interspace between gas seal and gearbox seal shall be vented and drained through drillings in the gearbox and/or volute. Vent sizing shall limit the velocity to 100 ft/sec (30 m/sec).

4.3.8 Bearings and bearing housings

4.3.8.1 Bearing type

Radial and thrust bearings should be of the hydrodynamic type, designed to damp out self-excited or externally excited vibration, and designed to accept backward rotation. Pinion radial bearings for integrally geared compressors shall be of the tilting pad design. Use of any bearings other than the hydrodynamic type shall be agreed between the manufacturer and the user.

4.3.8.2 Thrust bearing size

The thrust bearings shall be sized for continuous operation under the most adverse specified operating conditions. Calculation of the thrust force shall include but not be limited to the following factors:

- Seal minimum design internal clearances and twice the maximum design internal clearances;
- Pressurized rotor diameter step changes;
- Stage maximum differential pressures;
- Specified extreme variations in inlet, interstage, and discharge pressures;
- External thrust forces transmitted through the couplings;
- Maximum thrust force from the sleeve-bearing-type drive motor, if the motor is directly connected;
- Thrust forces for diaphragm-type couplings shall be calculated on the basis of the maximum allowable deflection permitted by the coupling manufacturer; and
- If two or more rotor thrust forces are to be carried by one thrust bearing (such as in a gearbox), the resultant of the forces shall be used, provided the directions of the forces make them numerically additive; otherwise, the largest of the forces shall be used.

4.3.8.3 Provision for vibration probes

Bearing housings shall be designed to incorporate the following vibration measuring instruments:

- two noncontacting shaft vibration probes at right angles to one another on or near each high speed bearing; and
- one keyphaser probe per high speed shaft.

4.3.8.4 Bearing failure—Resultant rubs

During normal operational procedures, an agreed amount of limited contact is permitted in the seal (see 4.3.7). The manufacturer shall carry out an analysis to determine what parts of the compressor will rub in the event of a catastrophic rotor excursion, which can be caused by an axial or radial bearing failure. The manufacturer shall make every effort to ensure that the resulting rubs that occur during the compressor rundown shall meet the following criteria:

- The rubbing components shall be any combination of silver, copper alloy, nickel alloy, high alloy steel, for example, a cast iron to low alloy steel rub is not permitted; and
- At the rub site, there is high heat capacity and good heat transfer.

At the design stage, the manufacturer shall supply a table of clearances and materials that demonstrate that these previous requirements have been complied with.

4.4 Rotor dynamic analysis, verification tests, and data to be provided

4.4.1 Summary

An important contributor to the safe compression of oxygen is a well-designed compressor and an important aspect of the compressor design is a stable and well damped rotor system. An unstable rotor results in high vibrations and large rotor deflections, which in turn cause high speed rubs that are a primary cause of oxygen fires. It is for this reason that this publication emphasizes the need for detailed modelling of the rotor system over the whole range of expected operating parameters followed by tests in the workshop or field to verify that the rotor system is satisfactory.

4.4.2 References

This publication basically follows internationally recognized standards and practices. API Standard 617 shall be used as the basis including testing and acceptance criteria, with the exceptions and clarifications given as follows [3].

- For compressors with rigid coupling, train analysis shall always be performed [3];
- An internal rub on an oxygen compressor is of much greater importance than on, for example, an air compressor since it represents a possible source of ignition. For this reason, the maximum amplitude of any component within the oxygen envelope shall not exceed 75% of the internal clearance when the displacement at the probe location is at the trip level according to 4.5.2 [3];
- Since stable rotor dynamics are essential for an oxygen compressor, any failure to meet the design requirements shall be rectified to meet the requirements of 4.4. Retesting after modification should be conducted; and
- Verification tests shall be performed using dry, oil-free, air or nitrogen. Blended dry, oil-free gases may be used to reach the correct molecular weight (for example, mixture of carbon dioxide and nitrogen). Oxygen shall not be used for verification tests.

Due to their physical nature, any responding shaft vibrations that occur can always be related to forced, to self-excited, or to parameter-excited vibrations. The sources of these vibrations and their effects on the rotor system shall be analyzed by calculations if they are expected to occur in the actual design.

4.5 Balancing and vibration

4.5.1 Balancing

High speed balancing shall be considered on all rotors running greater than their first bending critical speed. The first bending critical is the mode in the real rotor system corresponding to the first critical of the same rotor in rigid bearings. When performed, the assembled rotors shall be balanced at their maximum operating speed. Balance corrections shall be done according to the mode shapes without any unallowable influence of the low speed balanced quality according to ISO 1940-1, *Mechanical Vibration — Balance quality requirements for rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances* [7].

The acceptance criteria to be met for the high speed balancing shall be as follows.

The bearing pedestal vibrations shall be in accordance with ISO 10816, *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 1: General guidelines* and shall not exceed the following limits [8]:

- At critical speeds = 4.5 mm/s (root mean square [RMS]);
- Within the operating speed range (from minimum operating speed to maximum continuous speed) = 1.8 mm/s (RMS);
- Up to and including trip speed = 4.5 mm/s (RMS); and
- The relative shaft vibrations in normal operating conditions shall be in accordance with API Standard 617 [3].

4.5.2 Vibration alarms and trips

There is no recognized rule for setting alarm and trip levels. Many operators base the setting upon the actual running levels achieved in operation. Unless otherwise specified by the manufacturer of the compressor, the following values set out are based on API Standard 617 and should be regarded as maximum levels provided that these levels are less than 75% of the nominal clearance [3].

- Maximum permissible alarm setting:

$$A = 2.0 \times (25.4 \times (12\,000/N_{\max})^{1/2})$$
- Maximum permissible trip setting:

$$A = 3.0 \times (25.4 \times (12\,000/N_{\max})^{1/2})$$

Where:

$N_{\max}$ = max continuous speed (rpm)

A = Amplitude of unfiltered vibration in micron peak to peak

4.6 Insulation and grounding

Care shall be taken to insulate and ground the electric drive motor correctly to prevent currents circulating through the compressor that, experience has shown, can damage the bearings, couplings, and gear teeth. This phenomenon can occur in all types of compressor but additional care is required in the case of oxygen compressors because the consequence of bearing damage could be a fire. Compressor shafts may be grounded. See Figure 1.