20

PROCUREMENT, AUDIT, AND ASSET MANAGEMENT DECISIONS

By now the careful reader will have become acquainted with the basics of compressor technology. He or she will have little difficulty accepting the premise that considerable forethought has to go into compressor specification, procurement, manufacturing, inspection, testing, field erection, operation, and maintenance. All of these must retain a reliability focus, and that's where the concluding chapter topics will be of help.

20.1 INCENTIVES TO BUY FROM KNOWLEDGEABLE AND COOPERATIVE COMPRESSOR VENDORS

Reliability-focused companies generally do not find it too difficult to select from among several available compressor manufacturers. Their selection task is facilitated because as reliability-focused purchasers, they will invite only experienced manufacturers to submit their proposals. In other words, compressor preselection starts by inviting bids from prequalified, capable, and experienced manufacturers only.

Note that three principal characteristics identify capable, experienced equipment vendors [1]:

- They are in a position to provide extensive experience listings for equipment offered and will submit this information without much hesitation.
- Their compression equipment enjoys a reputation for sound design and infrequent maintenance requirements.
- Their marketing personnel are thoroughly supported by engineering departments. Also, both groups are willing to provide technical data beyond those that are customarily submitted with routine proposals.

A Practical Guide to Compressor Technology, Second Edition, By Heinz P. Bloch

Vendor competence and willingness to cooperate are shown in a number of ways, but data submittal is the first real test. When offering compression equipment required to comply with the standards of the API (the American Petroleum Institute, i.e., the latest applicable edition of API-617 for centrifugal, or API-618 for reciprocating compressors), a capable vendor will make diligent efforts to fill in all of the data requirements of the API specification sheet. However, the depth of technical know-how will show in the way in which a vendor-manufacturer explains exceptions taken to API standards or supplementary user's specifications.

20.2 INDUSTRY STANDARDS AND THEIR PURPOSE

Reliability specialists are encouraged to utilize industry standards whenever possible. Some of these standards comprise over 200 pages of narrative and illustrations, and Figs. 20.1 and 20.2 are only two of the many data pages of API-617 and API-618. These and related standards have been issued for compressor drivers and auxiliary systems; their use is considered mandatory by reliability-focused purchasers. However, those contemplating use of these specifications should realize that API standards contain a large number of clauses that require exercising certain purchase options. The pertinent paragraphs typically start with: "When specified …" or "With the purchaser's approval …" Obviously, these caveats should compel purchasers or their representatives to take a very close look at this or any other API specification for process machinery.

Still, decades of favorable experience support the contention that well-written standards improve equipment uniformity and quality. The foreword to the NEMA (National Electric Machinery Association) standard explains the scope and purpose and could serve as a model for what standards are all about.

20.2.1 Typical Scope of Standards

API, NEMA, and many other standards have been developed and approved for publication as industry guidelines, not laws. They are intended to assist users in the proper selection and application of machinery, and they serve that purpose exceedingly well. These standards are revised periodically to provide for changes in user needs, advances in technology, and changing economic trends. All persons having experience in the selection, use, or manufacture of the equipment at issue are encouraged to submit recommendations that will improve the usefulness of the standards.

The collective judgment of users and manufacturers on the performance and construction of machinery is represented in these standards. They are generally based on sound engineering principles, research, and records of test and field experience. Also involved is an appreciation of the problems of manufacture, installation, and use derived from consultation with and information obtained from manufacturers, users, inspection authorities, and others having specialized experience. For machines intended for general applications, anticipated user needs are often determined by the equipment manufacturer. The manufacturer perceives these needs through normal commercial contact with users.

For some machines intended for definite applications, the organizations that participated in the development of the standards are listed at the beginning of those definite-purpose equipment standards. Practical information concerning performance, safety, testing, construction, and manufacture of machinery within the product scopes are often defined in the applicable section or sections of a standard. Although some definite-purpose machines are occasionally included, the standards do not apply to all machines that are called, say, "compressors." For

CENTRIFUGAL COMPRESSOR VENDOR DRAWING AND DATA REQUIREMENTS

JOB NO. ______ ITEM NO. ______
PURCHASE ORDER NO. ______ DATE ______
REQUISITION NO. ______ DATE ______
INQUIRY NO. ______ DATE ______
PAGE 1 OF 2 BY ______

FOR ______
SITE ______
SERVICE ______

REVISION ______
UNIT ______
NO. REQUIRED ______

Proposal[a] — Bidder shall furnish ____ copies of data for all items indicated by an X.

Review[b] — Vendor shall furnish ____ copies and ____ transparencies of drawings and data indicated.

Final[b] — Vendor shall furnish ____ copies and ____ transparencies of drawings and data indicated.
Vendor shall furnish ____ operating and maintenance manuals.

DISTRIBUTION RECORD

Final—Received from vendor ______
Due from vendor[c] ______
Review—Returned to vendor ______
Review—Received from vendor ______
Review—Due from vendor[c] ______

Proposal[a]	Review[b]	Final[b]	DESCRIPTION	Review—Due from vendor[c]	Review—Received from vendor	Review—Returned to vendor	Due from vendor[c]	Final—Received from vendor
			1. Certified dimensional outline drawing and list of connections.					
			2. Cross-sectional drawing and bill of materials.					
			3. Rotor assembly drawing and bill of materials.					
			4. Thrust-bearing assembly drawing and bill of materials.					
			5. Journal-bearing assembly drawing and bill of materials.					
			6. Seal assembly drawing and bill of materials.					
			7. Coupling assembly drawing and bill of materials.					
			8. Seal-oil schematic and bill of materials.					
			9. Seal-oil assembly drawing and list of connections.					
			10. Seal-oil component drawings and data.					
			11. Lube-oil schematic and bill of materials.					
			12. Lube-oil assembly drawing and list of connections.					
			13. Lube-oil component drawings and data.					
			14. Electrical and instrumentation schematics and bill of materials.					
			15. Electrical and instrumentation arrangement drawing and list of connections.					
			16. Polytropic head and efficiency versus ICFM.					
			17. Discharge pressure and brake horsepower versus ICFM.					
			18. Balance line pressure versus thrust load.					
			19. Speed versus starting torque.					
			20. Vibration analysis data.					
			21. Lateral critical analysis.					
			22. Torsional critical analysis.					
			23. Transient torsional analysis.					
			24. Allowable flange loading.					
			25. Alignment diagram.					
			26. Weld procedures.					
			27. Hydrostatic test logs.					
			28. Mechanical run test logs.					

[a]Proposal drawings and data do not have to be certified or as-built.
[b]Purchaser will indicate in this column the time frame for submission of materials using the nomenclature given at the end of the form.
[c]Bidder shall complete these two columns to reflect his actual distribution schedule and include this form with his proposal.

FIGURE 20.1 Excerpts of vendor drawing and data requirements from API-617. (*American Petroleum Institute, Washington, D.C.*)

example, automotive air-conditioning compressors would not be included in API coverage. Nevertheless, in the preparation and revision of these standards, consideration is often given to the work of other organizations whose standards are in any way related to similar machinery. Moreover, credit is usually given to all those whose standards may have been helpful in the preparation of a particular issue or edition.

Standards such as those issued by API and NEMA continue to be developed through a voluntary consensus standards development process. This process brings together volunteers

CENTRIFUGAL COMPRESSOR DATA SHEET SI UNITS

PAGE 3 OF 6
JOB NO. ____ ITEM NO. ____
REVISION ____ DATE ____
BY ____

CONSTRUCTION FEATURES

☐ **SPEEDS:**
MAX. CONT.____RPM TRIP____RPM
MAX. TIP SPEEDS:____m/s @ RATED SPEED
____m/s @ MAX. CONT. SPEED
☐ **LATERAL CRITICAL SPEEDS (DAMPED)**
FIRST CRITICAL____ RPM____ MODE
SECOND CRITICAL____ RPM____ MODE
THIRD CRITICAL____ RPM____ MODE
FOURTH CRITICAL____ RPM____ MODE
○ TRAIN LATERAL ANALYSIS REQUIRED (2.9.2.3)
○ UNDAMPED STIFFNESS MAP REQUIRED (2.9.2.4e)
○ TRAIN TORSIONAL ANALYSIS REQUIRED (TURBINE DRIVEN TRAIN) (2.9.4.5)
☐ **TORSIONAL CRITICAL SPEEDS:**
FIRST CRITICAL____ RPM
SECOND CRITICAL____ RPM
THIRD CRITICAL____ RPM
FOURTH CRITICAL____ RPM
☐ **VIBRATION:**
ALLOWABLE TEST LEVEL____ mm (PEAK TO PEAK)
☐ **ROTATION, VIEWED FROM DRIVEN END**
○ **MATERIALS INSPECTION REQUIREMENTS (4.2.3)**
○ SPECIAL CHARPY TESTING (2.11.3)
○ RADIOGRAPHY REQUIRED FOR ____
○ MAGNETIC PARTICLE REQUIRED FOR____
○ LIQUID PENETRANT REQUIRED FOR____
☐ **CASING:**
MODEL____
CASING SPLIT ____
MATERIAL____
THICKNESS (mm)____CORR. ALLOW. (mm)____
MAX WORKING PRESS. ____ BARG
MAX DESIGN PRESS ____ BARG
TEST PRESS (BARG): HELIUM____ HYDRO____
MAX. OPER. TEMP.____°C MIN. OPER. TEMP.____°C
MAX. NO. OF IMPELLERS FOR CASING____
MAX. CASING CAPACITY (m ^{3}/H)____
RADIOGRAPH QUALITY ○ YES ○ NO
CASING SPLIT SEALING____
○ SYSTEM RELIEF VALVE SET PT. (2.2.3)____BARG
☐ **DIAPHRAGMS:**
MATERIAL____
☐ **IMPELLERS:**
NO.____ DIAMETERS____
NO. VANES EA. IMPELLER____

TYPE (OPEN, ENCLOSED, ETC.)____
TYPE FABRICATION ____
MATERIAL____
MAX. YIELD STRENGTH (N) ____
BRINNEL HARDNESS: MAX____ MIN.____
SMALLEST TIP INTERNAL WIDTH (mm) ____
MAX. MACH. NO. @ IMPELLER EYE____
MAX. IMPELLER HEAD @ RATED SPD (N-m/kg) ____
☐ **SHAFT:**
MATERIAL____
DIA @ IMPELLERS (mm)____ DIA @ COUPLING (mm)____
SHAFT END: TAPERED CYLINDRICAL
MAX. YIELD STRENGTH (BAR)____
SHAFT HARDNESS (BNH) (Rc) ____
STRESS AT COUPLING (BAR)____
☐ **BALANCE PISTON:**
MATERIAL____ AREA____(mm)
FIXATION METHOD ____
☐ **SHAFT SLEEVES (2.8.2):**
AT INTERSTG. CLOSE CLEARANCE POINTS MATL ____
AT SHAFT SEALS____ MATL ____
☐ **LABYRINTHS:**
INTERSTAGE
TYPE____ MATERIAL ____
BALANCE PISTON
TYPE____ MATERIAL ____
SHAFT SEALS:
○ SEAL TYPE (2.8.3) ____
○ SETTLING OUT PRESSURE (BARG)____
○ SPECIAL OPERATION (2.8.1) ____
○ SUPPLEMENTAL DEVICE REQUIRED FOR CONTACT SEALS (2.8.3.2) TYPE____
○ BUFFER GAS SYSTEM REQUIRED (2.8.7)
○ TYPE BUFFER GAS____
○ BUFFER GAS CONTROL SYSTEM SCHEMATIC BY VENDOR
○ PRESSURIZING GAS FOR SUBATMOSPHERIC SEALS (2.8.8)
☐ TYPE SEAL____
☐ INNER OIL LEAKAGE GUAR. (m ^{3}/DAY/SEAL) ____
BUFFER GAS REQUIRED FOR:
☐ AIR RUN-IN ☐ OTHER ____
☐ BUFFER GAS FLOW (PER SEAL):
NORM:____ kg/MIN @ ____ BAR Δ P____
MAX.:____ kg/MIN @ ____ BAR Δ P____
☐ **BEARING HOUSING CONSTRUCTION:**
TYPE (SEPARATE, INTEGRAL)____SPLIT____
MATERIAL ____

FIGURE 20.2 Typical (partial) data sheet from API-617. (*American Petroleum Institute, Washington, D.C.*)

and/or seeks out the views of persons who have an interest in the topic covered by a given publication. Although API, NEMA and others administer the process and establish rules to promote fairness in the development of consensus, they do not write the document and do not independently test, evaluate, or verify the accuracy or completeness of any information, or the soundness of any judgments contained in its standards and guideline publications.

20.2.2 Disclaimers in Standards

As is to be expected, the information in the standard publication was considered technically sound by the consensus of persons engaged in the development and approval of the document at the time it was developed. But consensus does not necessarily mean that there was unanimous agreement among every person participating in the development of the standards document.

The standards or guidelines presented in an industry standard are assumed technically sound at the time they are approved for publication. Yet they are not a substitute for a product seller's or user's own judgment with respect to the particular product referenced in the standard or guideline, and the issuing authority never guarantees the performance of any individual manufacturer's products by virtue of an industry standard or guide. In fact, entities such as API or NEMA expressly disavow responsibility for damages arising from the use, application, or reliance by others on the information contained in these standards or guidelines.

Understandably, the entity that issues the standard also disclaims liability for any personal injury, property, or other damages of any nature whatsoever, whether special, indirect, consequential, or compensatory, resulting directly or indirectly from the publication, use, application, or reliance on the standard. The issuing entity also disavows claims of guaranty or warranty, expressed or implied, as to the accuracy or completeness of any information published, and disclaims and makes no warranty that the information in the document will fulfill particular purposes or needs.

20.2.3 Going Beyond the Standards

Although logical and understandable in a societal environment where attorneys and legal experts abound, the discussion above should remind us that a good compressor specification must go beyond the various and sundry clauses we are likely to find in an API document. Moreover, the user/purchaser must follow a course of auditing and reviewing the design, manufacture, assembly, shoptesting, and field erection of important compressors.

For as long as official industry standards have been available, but especially since the mid-1960s, reliability-focused companies have seen fit to use supplementary standards and to engage in equipment audits. These user companies have come to recognize that mere compliance with API or other applicable standards is not always sufficient to ensure delivery of optimally configured, low life cycle cost compressors. Up-to-date user experience and special requirements must be spelled out in these supplementary specifications. Referring to Fig. 20.2, note as an arbitrary example how on line 44 a diaphragm material will have to be agreed on by the parties to this sale. A supplementary specification might state "fabricated steel" because this material is more easily repaired than cast iron, and perhaps repairability is of prime concern to a certain user or at a certain plant location. Similarly, elsewhere on this normally 6-page data sheet, a "requester" perhaps asks for variable-speed drivers. Again, the reviewer may know of a supplementary specification that might limit or even disallow their use at a certain plant for technical reasons. In essence, the review process described later is ensuring compliance with supplemental specifications and the special reliability needs of a particular purchaser.

Equipment layout and general assembly drawings are among the indispensable review and documentation requirements. Potential design weaknesses can be discovered in the course of reviewing dimensionally accurate cross-sectional drawings. Examination and review of suitable reference books (among them, Refs. 1 through 4) or other specialized texts will disclose dozens of areas to be questioned.

In general, there are two compelling reasons to conduct this drawing review during the bid evaluation phase of a project: First, some compressor manufacturers may not be able (or willing) to respond to user requests for accurate drawings after the order is placed; second, the design weakness could be significant enough to require extensive redesign. In the latter case, the purchaser may sometimes be better off to select a different compressor model [2].

20.3 DISADVANTAGES OF CHEAP PROCESS COMPRESSORS

It is intuitively evident that purchasing the least expensive machine will rarely be the wisest choice for users wishing to achieve long run times and low or moderate maintenance outlays. Some manufacturers have been accused of marketing "compressors for less" in the expectation of making up for the discrepancy by later selling spare parts with very high profit margins, because they are buying major components from poorly qualified third parties, or because they provide less than desirable inspection coverage at the point of origin.

Although there might be occasions when a company new to the compressor market is able to design and manufacture a better machine than that of an established manufacturer, it is simply not very likely that such newcomers will initially produce a superior product. It would thus be more reasonable, with rare exceptions, to choose from among the most respected *existing* manufacturers (i.e., manufacturers that *currently* enjoy a proven track record).

The first step should therefore to invite only those bidders that meet a number of predefined criteria. The decision as to who should be asked to bid on providing gas compressors for job situations demanding high reliability should take into account the following:

- Acceptable vendors must have experience with the size, pressure, temperature, flow, and service conditions specified.
- Vendors must have proven capability in manufacturing with the metallurgy and fabrication method chosen (e.g., sand casting, fabricated plate, steel with special weld overlay metallurgy).
- Vendor's "shop loading" must be able to accommodate an order within the required time frame (time to delivery of product).
- Vendors must have implemented satisfactory quality control and must be able to demonstrate a satisfactory on-time delivery history over the past several (usually, two) years.
- If unionized, the vendor must show that there is virtually no risk of labor strife (strikes or work stoppages) while manufacture of particular compressors is in progress.

Since many compressor manufacturers assume that first cost is of paramount importance to the purchaser, their first offer rarely includes all of the features and provisions that best serve a reliability-focused user. As mentioned earlier, it is thus advisable for the owner/purchaser to invoke supplemental specifications. These supplemental specifications are often applied in conjunction with an applicable API specification and amend, delete, or amplify certain API specification clauses.

Whenever such supplementary specifications are being compiled, it is good to keep in mind the following recommendations:

- Specify for low maintenance. Reliability-focused purchasers realize that selective upgrading of certain components will result in rapid payback. (*Note*: Components that are upgrade candidates have been identified in the references given in this book. Be sure to specify those.) Let's cite a simple example: Review failure statistics for principal failure causes. If bearings are prone to failure, realize that the failure cause may be incorrect lube application or lube contamination. Address these failure causes in the specifications.

- Evaluate the vendor response. Allow exceptions to the specification if they are both well explained and valid.
- Clearly document the equipment design; else, future failure analysis and troubleshooting efforts will be greatly impeded. For future repair and troubleshooting work, a plant will certainly require the various equipment cross-section views and other documents. The reviewer should not allow the vendor to claim that these documents are proprietary and that the purchaser is not entitled to them. Therefore, reliability-focused buyers place the vendor under contractual obligation to supply all agreed-upon documents in a predetermined time frame and make it clear that they will withhold 10 or 15% of the total purchase price until all contractual data transmittal requirements have been met.

On critical orders, reliability-focused buyers arrange *contractually* for access to a factory contact. Alternatively, these buyers insist on the nomination of a *management sponsor*, a vice president or director of manufacturing, or a person holding a similar job function at the manufacturer's facility or head offices. The reviewing professional will communicate with this person for redress on issues that could cause impaired quality or delayed delivery.

Following these guidelines will give the best assurance of meeting the expectations of reliability-focused owner–purchasers. But using a well-thought-out review or bid conditioning process, most owner–purchasers are willing to waive an occasional specification requirement if the vendor is able to offer sound engineering reasons. Suffice it to say, only the best-qualified compressor vendors can state their reasons convincingly.

20.4 AUDITS VS. REVIEWS

This overview defines a *machinery reliability audit* as any rigorous analysis of a vendor's overall design after issuance of the purchase order and before beginning equipment fabrication. A *reliability review* is defined as a less formal, ongoing assessment of component or subsystem selection, design, execution, or testing. Machinery reliability *audits* tend to utilize outside resources for brief, concentrated efforts beginning within two months of issuance of the purchase order. On the other hand, reliability *reviews* are typically assigned to one or more experienced machinery engineers who would start being involved in a project from the time specifications are written until the machinery leaves the vendor's shop for shipment to the plant site. In fact, a *review professional* stays with the project throughout the plant startup phase.

In summary, the primary purpose of the *audit effort* is to flush out deep-seated or fundamental design problems on major compressors and drivers. A secondary purpose is to determine which design parameters should be subjected to nonroutine computer analysis and to assist in defining whether follow-up reviews should employ other than routine approaches.

In contrast with the above, the machinery reliability *review effort* is aimed at ensuring compliance with all applicable specifications. These reviews will also judge the acceptability of certain deviations from applicable specifications. Moreover, an experienced reliability review engineer will provide guidance on a host of items, which either could not be, or simply had not been, specified in writing.

20.4.1 Staffing and Timing of Audits and Reviews

Machinery reliability audits as well as reviews can be a tremendously worthwhile investment. They must be performed by experienced engineers and in a well-structured manner. Of course, this presupposes that a perceptive project manager will see to it that the resulting recommendations are, in fact, implemented.

It has been estimated that a medium-sized grass-roots refinery valued at approximately \$1.5 billion (\$1,500,000,000) would optimally staff machinery reliability audits with four engineers for a four-month period and machinery reliability reviews with two engineers for a period of two to three years. The total cost of these efforts would be in the range \$1,800,000 to \$2,400,000. If this sounds like a lot of money, the reader may wish to contrast it with the typical value of a single startup-delay day: amounts in excess of \$600,000 per delay day, millions for two days of unplanned downtime, perhaps accompanied by the thunder of two tall flare stacks for the better portion of two days.

Of course, reliability assurance efforts made before delivery of the machinery are more cost-effective than postdelivery or poststartup endeavors aimed toward the same goals. However, the question remains how to conduct these efforts optimally, how to staff them, and which components or systems to subject to close scrutiny. This is where an analysis of available failure statistics will prove helpful. A review of the failure statistics of rotating machinery used in modern process plants will help determine where the company's money should be spent for the highest probable returns. Moreover, failure statistics [1, 2] can often be used to determine the value of and justification for these efforts.

20.4.2 Use of Equipment Downtime Statistics

Knowledge of failure causes and downtime statistics allows reliability professionals to determine which components merit closer prepurchase review. Also, properly kept records could alert the review engineer to equipment types or models that should be avoided. In some cases, failure statistics might provide key input to a definitive specification. In other words, "you learn from the mistakes of others." All of this presupposes that the others saw fit to record their experiences. If an engineer has these data, available he or she will no doubt use them before selecting machinery.

As of 2005, experienced petrochemical process plants using a conscientious program of mechanical and instrument condition surveillance could achieve eight-year runs on "clean gas" centrifugal process compressors and train availabilities often exceeding 99.5% per year. In best-of-class plants, unscheduled downtime events occur only perhaps once every five years for the average centrifugal compressor train. Recent statistics appear to track those shown in Ref. 1. These statistics show where detailed design reviews might prove profitable.

Rotor and shaft distress rank highest in downtime hours per year per train. Blade or impeller problems rank next, followed by motor failures. Obviously, turbomachinery reliability audits and follow-up reviews should concentrate on these areas first. In relevant texts (e.g., Ref. 2), reciprocating compressor failure events are listed by primary cause, extent of damage, typical repair cost, and average downtime. For important reciprocating compressors, a compelling case can be made for reliability audits and reviews. The user or machinery owner can justify spending many thousands of dollars for this work if the effort results in reduced failure risk.

Again, the reliability review topic is truly vast, and volumes have been written to deal with it. The purpose of including the topic in our book is to give the reader a feel for major machinery reliability audits and reviews. Since centrifugal and reciprocating compressors are representative of dynamic and positive displacement machinery systems, we chose to explain the review concept by highlighting these machines.

20.5 AUDITING AND REVIEWING COMPRESSORS

As mentioned earlier, machinery reliability review engineers should start their assignment at the time the specifications are written, should be deeply involved in the bid review process, and should take an active part in *bid conditioning*. Bid conditioning is a process of making value judgments and of assigning credits or debits to certain design features, performance parameters, service capabilities, deviations from the specifications, and so on. It is during this conditioning process that vendor qualifications and possible extrapolations from past experience should receive close scrutiny.

Deviations or extrapolations from past experience may be the result of the purchaser's specifying certain service conditions, which in turn cause the machinery manufacturer to offer equipment outside prior parameters:

- Pressure rating
- Molecular weight
- Volumetric capacity
- Power rating
- Speed
- Temperature

This, of course, may lead to deviations or extrapolations in the mechanical design area:

- Sealing systems, including packing in reciprocating compressors
- Bearing design and loading
- Number of stages and staging arrangement
- Casing or cylinder size and design
- Casing joint design, reciprocating compressor distance piece configuration, and compartment venting
- Rotor and/or balancing dynamics
- Impeller or piston structural design and performance
- Material selection
- Power transmission component design and arrangement
- Valve materials, valve lift, and gas velocity
- Rotor speed or piston velocity
- Others, according to industry experience and statistics

Note that although these considerations should have influenced the equipment selection, they also merit review after the purchase order has been issued.

There is little difference in how experienced engineers approach and review tasks for various compressors as opposed to turbines, gears, and other machinery. In each case they must obtain drawings and other technical data from equipment vendors. They must then review all pertinent documentation for consistency, safety, compliance with specifications, and so on, and document all areas requiring follow-up.

Compressor documentation requirements probably exceed those of most other machinery with the possible exception of large mechanical-drive steam turbines. Lists of relevant documentation are presented in the appendixes of the various API standards. Using the API tabulation (see Fig. 20.1 for an abbreviated version) facilitates outlining the items recommended for review. The review engineer can use these tabulations to keep track of this work.

The review includes the following but is, of course, not limited to the items listed here:

1. Certified dimensional outline drawing, including:
   a. Journal bearing clearances
   b. Rotor float
   c. Labyrinth, packing, and seal clearances
   d. Axial position of impellers relative to guide vanes
   e. List of connections
   - Journal bearing clearances may be required for rotor sensitivity studies. Bearing dimensions allow rapid calculation of bearing loading and serve to screen the tendency for oil whirl to occur.
   - Labyrinth, packing, and seal clearances may be too tight for normal process operation. The vendor may attempt to show good efficiency (abnormally low recirculation) during shop performance tests.
   - Axial position of impellers relative to guide vanes needs to be reviewed in conjunction with rotor float dimension. Is rubbing contact likely to occur?
   - The list of connections may uncover dimensional mismatching with purchaser's lines, excessive flow velocities, omission of specified injection points, and so on.
2. Cross-sectional drawing and bill of materials
   - These documents are used primarily for verification of impeller dimensions, internal porting, visualization of maintenance access, materials selection, assessment of number of spare parts needed, and so on. A copy of this drawing and the bill of materials should also be forwarded to responsible maintenance personnel.
3. Rotor or cylinder assembly drawing, including:
   a. Axial position from active thrust-collar face to each impeller
   b. Each radial probe
   c. Each journal-bearing centerline
   d. High-pressure side of balance drum
   e. Thrust-collar assembly details, including:
      — Collar-shaft fit with tolerance
      — Concentricity (or runout) tolerance
      — Required torque for locknut
      — Surface finish requirements for collar faces
      — Preheat method and temperature requirements for "shrunk-on" collar installation
   f. Running gear (crankshaft and crosshead) assemblies
   g. Attachment and securing methods for piston rods

h. Balance drum (or tailrod, in reciprocating compressors) details including:
   - Length of drum
   - Diameter of drum
   - Labyrinth details
i. Dimensioned shaft end(s) for coupling mounting(s)
j. Bill of materials
   - Axial position data are required for rotor dynamics analyses and maintenance records. Accurate rotor dynamics studies would further require the submission of weight or mass moment of inertia data for impellers and balance drum.
   - Thrust collar assembly details are to be analyzed for nonfretting engagement and feasibility of field maintenance. Hydraulic fit is preferred.
   - Balance drum details are needed for rotor dynamics analyses and maintenance reviews.
   - Dimensioned shaft ends for coupling mountings allow calculation of stress levels, margins of safety, uprateability, and coupling maintenance.
   - The bill of materials is again used for comparison of component designs and materials being released for fabrication. Again, the bill of materials will allow definition of spare parts requirements.

4. Thrust-bearing assembly drawing and bill of materials
   - These are used to verify thrust bearing size and capacity. They are important if directed oil spray lubrication has been specified. They contain essential maintenance information.
5. Journal-bearing assembly drawing and bill of materials
   - Bearing dimensions are required for calculation of bearing loading, rotor dynamic behavior, and maintenance records.
6. Seal assembly drawing and bill of materials
   - These are required to compare seal dimensions, clearances, and tolerances with similar data from seals operating properly under essentially identical operating conditions.
7. Coupling assembly drawing and bill of materials
   - These are used for calculations verifying load-carrying capacity, mass moment of inertia, overhung weight, shaft-fit criteria, dimensional compatibility between driver and driven equipment, material selection, match marking, assembly and disassembly provisions, and review of spare parts availability.
8. Seal oil (or cylinder lubrication, in reciprocating compressors) schematic, including:
   a. Steady-state and transient oil flows and pressures
   b. Control, alarm, and trip settings
   c. Heat loads
   d. Utility requirements, including electrical, water, and air
   e. Pipe and valve sizes
   f. Bill of materials

It should also be noted that:

- Oil flows and pressures must change as a function of gas pressure and compressor speed changes. The review must verify that the seal oil supply can accommodate all requirements anticipated for a given compressor. This would include operation during run-in on air.

- Control, alarm, and trip settings are required for operating and maintenance manuals as well as for initial field implementation by the contractor.
- Heat loads are required for capacity checks on oil coolers.
- Utilities requirements are required for proper sizing of switchgear, steam lines, and so on.
- Pipe and valve size are employed in calculations, verifying that maximum acceptable flow velocities are not exceeded.

9. Seal oil assembly drawing and list of connections
   - These are required for the contractor's (purchaser's) connecting design.
10. Seal oil component drawings and data, including:
    a. Pumps and drivers
       (1) Certified dimensional outline drawing
       (2) Cross section and bill of materials
       (3) Mechanical seal drawing and bill of materials
       (4) Priced spare parts list and recommendations
       (5) Instruction and operating manuals
       (6) Completed data forms for pumps and drivers
    b. Overhead tank (or cylinder lubricator in reciprocating compressors), reservoir, and drain tanks
       (1) Fabrication drawings
       (2) Maximum, minimum, and normal liquid levels
       (3) Design calculations and capacity and retention data
    c. Coolers and filters
       (1) Fabrication drawings
       (2) Priced spare parts list and recommendations
       (3) Completed data form for cooler(s)
    d. Instrumentation
       (1) Controllers
       (2) Switches
       (3) Control valves
       (4) Gauges
    - Pumps and drivers are reviewed for accessibility, coupling arrangements, baseplace mounting method, proximity of discharge and suction pipe, and so on.
    - Overhead tank, main reservoir, and drain tanks (e.g., degassing tank, sour seal oil reservoir) must comply with specifications. Should overhead tanks be given thermal insulation?
    - Coolers must be suitable for *heating* the seal oil during oil flushing operations. Are they sized to cool the oil flow resulting from more than one pump operation? Can filters be fully drained? Do they have vent provisions? What is their collapsing pressure? What types of cartridges do they accept? Specification compliance must be ascertained.

- Is instrumentation accessible? Can it be checked, calibrated, or replaced without causing a shutdown? Is it identified properly? Are controllers and transmitters located at optimum locations for rapid sensing and control? Are switches of sound design, and are they manufactured by a reputable company? Are control valves sized properly? Are gauges made of acceptable metallurgy? Are the ranges correct?

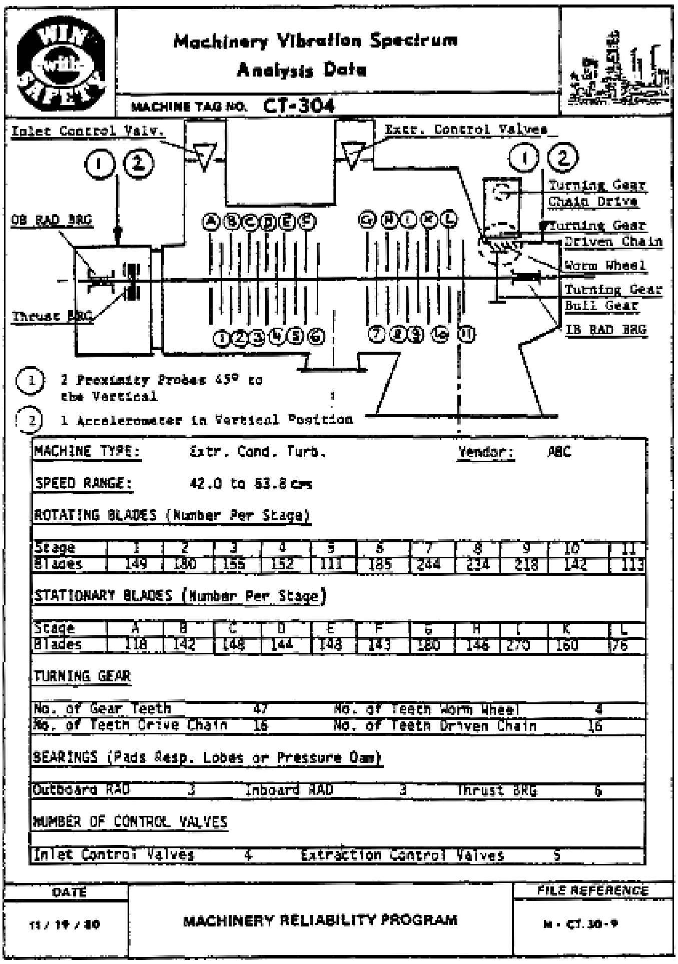
11. Lube-oil schematic, including:
    a. Steady-state and transient oil flows and pressures
    b. Control, alarm, and trip settings
    c. Heat loads
    d. Utility requirements, including electrical, water, air, steam, and nitrogen
    e. Pipe and valve sizes
    f. Bill of materials
    - Are steady-state and transient flows within the capability of the pumps and accumulator? Will the pumps and accumulators satisfy the driver hydraulic transients? Is the accumulator maintainable?
    - Have the control, alarm, and trip settings been tabulated?
    - Must the heat loads be accommodated by fouled coolers?
    - The utility requirements are needed to allow plant design to proceed in such areas as electrical protective devices, water supply lines, and nitrogen supply for blanketing of the reservoir. Identify the steam requirements for turbine-driven pumps.
    - The pipe and valve sizes need to be checked to determine an acceptable flow velocity.
    - The bill of materials should be reviewed to identify inexpensive or difficult-to-components. It should also be reviewed by maintenance personnel. Are O-rings, rolling element bearings, and so on, identified so as to allow purchase from the *actual* manufacturers of these components?
12. Lube oil assembly drawing and list of connections
    - These are required for contractor's (purchaser's) connecting design.
13. Lube-oil component drawings and data, including:
    a. Pumps and drivers
        (1) Certified dimensional outline drawing
        (2) Cross section and bill of materials
        (3) Mechanical seal drawing and bill of materials
        (4) Performance curves for centrifugal pumps
        (5) Priced spare parts list and recommendations
        (6) Instruction and operating manuals
        (7) Completed data forms for pumps and drivers
    b. Coolers, filters, and reservoir
        (1) Fabrication drawings
        (2) Maximum, minimum, and normal liquid levels in reservoir
        (3) Completed data form for cooler(s)
        (4) Priced spare parts list and recommendations

c. Instrumentation
   (1) Controllers
   (2) Switches
   (3) Control valves
   (4) Gauges

- Refer to item 10. The same reviews are necessary here. Note that performance curves are required whenever pumps are involved, regardless of whether they are of the centrifugal or positive displacement (screw) type. Positive displacement pumps undergo “slippage,” which varies with the viscosity of the pumped fluid.
- Instruction and operating manuals are intended for future incorporation in owners’ mechanical procedures and conventional plant operating manuals.

14. Electrical and instrumentation schematics and bill of materials
    - Machinery review engineers should be given responsibility for obtaining these data and forwarding them to the engineer’s electrical/instrument engineering counterparts for review and comment.
15. Electrical and instrumentation arrangement drawing and list of connections
    - Same as above. At the completion of reviews by electrical/instrument engineering personnel, the final arrangement will be implemented by the contractor.
16. Polytropic head and polytropic efficiency vs. icfm curves for each section or casing on multiple section or casing units in addition to composite curves at 80, 90, 100, and 105% of rated speed
    - On reciprocating compressors, request detailed information on compressor unloading (i.e., operation with partial load).
    - For dynamic compressors, request information on probable location of surge lines for various molecular-weight gases, as required. These are important data for future uprate and general performance verification studies. These can be used for the purchaser’s check on the vendor’s predicted performance.
17. Discharge pressure and brake horsepower vs. icfm curves at rated conditions for each section or casing on multiple-section or multiple-casing units in addition to composite curves at 80, 90, 100, and 105% of rated speed
    - For variable molecular-weight (MW) gases, curves must also be furnished at maximum and minimum MW. For air compressors, curves must also be furnished at three additional specified inlet temperatures.
18. “Pressure above suction pressure behind the balance drum” vs. “unit loading of the thrust shoes,” both in psi (bar), using rated conditions as the curve basis
    - The curve extends from a pressure equal to suction pressure behind the drum to one corresponding to at least 500 psi (~35 bar) unit loading on the thrust shoes. Balance drum OD, effective balance drum area, and expected and maximum recommended allowable pressure behind the balance drum are shown on the curve sheet.
    - Will balance drum labyrinth wear cause overloading of the thrust bearing? What happens when fouling (polymerization) occurs in the balance line? Is the design safe for a wide range of suction pressures?

19. Speed vs. starting torque curve
    - Will the motor be designed to start the compressor safely? (This is even more important for gas turbine drivers!)
20. Vibration analysis data, including:
    a. Number of vanes (each impeller)
    b. Number of vanes (each guide vane)
    c. Number of teeth (gear-type couplings)
    - These data, required for machine signature real-time online diagnostic or spectrum analysis, will allow identification of relevant frequencies, possibly useful in determining which component has undergone deterioration. Refer also to the illustrative example in Fig. 20.3. The review engineer should ensure that pertinent data are provided in this combination tabular and pictorial form.



MACHINE TYPE: Extr. Cond. Turb. Vendor: ABC

SPEED RANGE: 42.0 to 53.8 cps

ROTATING BLADES (Number Per Stage)

Stage	1	2	3	4	5	6	7	8	9	10	11
Blades	149	180	155	152	111	185	244	234	218	142	113

STATIONARY BLADES (Number Per Stage)

Stage	A	B	C	D	E	F	G	H	I	K	L
Blades	118	142	148	144	148	143	180	146	270	160	76

TURNING GEAR

No. of Gear Teeth	47	No. of Teeth Worm Wheel	4
No. of Teeth Drive Chain	16	No. of Teeth Driven Chain	16

BEARINGS (Pads Resp. Lobes or Pressure Dam)

Outboard RAD	3	Inboard RAD	3	Thrust BRG	6

NUMBER OF CONTROL VALVES

Inlet Control Valves	4	Extraction Control Valves	5

DATE		FILE REFERENCE
11 / 19 / 80	MACHINERY RELIABILITY PROGRAM	M - CT.30-9

FIGURE 20.3 A simple sketch showing that the number of blades in a given stage, or teeth per gear, may allow linking vibration frequencies to a specific blade row, or a particular gear. (Source: Bloch, Heinz P., *Improving Machinery Reliability*, 3rd Edition, 1998, Gulf Publishing Company, Houston, TX.)

21. Rotor dynamic analyses, including:
    a. Method used
    b. Graphic display of bearing and support stiffness and its effect on critical speeds
    c. Graphic display of rotor response to unbalance
    d. Graphic display of overhung moment and its effect on critical speed
    e. Graphic display of rotor stability
    - Reviews will identify if there is a risk of operating too close to critical speed or if the rotor is likely to vibrate excessively even at slight unbalance. If gear couplings are used, the effective (instantaneous) overhung moment may change as a function of tooth loading or tooth friction. The probability of encountering critical speed problems as a function of gear coupling deterioration can be investigated by examining graphic displays of effective overhung moment vs. critical speed.
    - Turbomachinery vendors must provide an accurate mathematical model of the rotor system before they will be able to proceed with their calculations of lateral critical speeds. This model must consider each significant shaft section length and diameter. Also, all concentrated masses and their inertias must be included and the effective stiffness and damping at the bearings must be represented as accurately as possible. The effective stiffness is influenced by oil film characteristics, bearing housing and pedestal configuration, and foundation features.
    - Rotor instabilities can occur when rotors operate above the first critical speed in support systems with low effective damping. The resulting vibration often shows up at subsynchronous frequencies and can cause serious damage without adequate warning. A good design should indicate stable well-damped operation at speeds and gas loads representative of some future, uprated design or operating condition.
22. Torsional critical speed analysis for all motor and gear units, including:
    a. Method used
    b. Graphic display of mass-elastic system
    c. Tabulation identifying the mass moment torsional stiffness for each component in the mass elastic system
    d. Graphic display of exciting sources (e.g., revolutions per minute of any gear in the train)
    e. Graphic display of torsional critical speeds and deflections (mode shape diagrams)
    - Torsional critical speeds coinciding with the running speeds of rotating elements in a turbine–gear–compressor or motor–gear–compressor train can cause oscillatory forces of such magnitude as to shorten component life drastically. The data listed are required to determine the probability of speed coincidence, and should coincidence exist, will allow calculation of resulting stresses. Purchasers may opt to duplicate the manufacturer's torsional analysis with in-house or outside resources. Alternatively, they may arrange for a field test of actual torsional stresses.
23. Transient torsional analysis for all synchronous motor-driven units
    - Transient, momentary torsional stresses on synchronous motors can be extremely severe and have been responsible for a number of catastrophic failures. Vendors should submit their analysis for review by the purchasers or their consultants.

24. Allowable flange loading is not to be exceeded by piping forces and moments. These forces and moments can be calculated readily by computers, and virtually all contractors now employ this analysis tool. Correctly used, it will ensure that equipment flange loadings remain within acceptable limits not only under all foreseeable operating conditions, but also while spare equipment connected to the same piping system is temporarily removed for maintenance.
25. An alignment diagram, including recommended limits during operation
    - Cold alignment offset calculations are to be reviewed for accuracy and appropriateness of manufacturers' assumptions. These data are then used for initial cold alignments (via reverse indicator readings).
26. Weld procedure
    - These procedures are commonly reviewed by purchasers' metallurgy specialists. Improper procedures have been responsible for commissioning delays and serious failures. A review of weld procedures can encompass piping, vessels, machinery casings, and even fan blade spares.
27. Hydrostatic test logs
    - Together with weld procedures, hydrostatic test logs should become part of the inspection record system of modern process plants.
28. Mechanical run test logs, including:
    a. Oil flows and temperatures
    b. Vibration
    c. Bearing metal temperatures
    d. Actual critical speeds
    - These test logs should provide verification for all predicted values. If audits and reviews have been conducted properly, the mechanical run tests will, at best, uncover vendor quality control errors. Deep-seated design errors should not surface at this stage in the job execution.
    - The mechanical run test can provide typical target values for comparison with initial field operation of major machinery. These logs should be retained for future reference.
29. Rotor balance logs
    - Rotor balance target values given by manufacturers can be compared with typical values quoted in the literature. Figure 3 shows a typical comparison chart. Rotor balance logs should also be retained in purchasers' equipment records.
30. Rotor mechanical and electrical runout
    - Maximum acceptable mechanical runout values are specified in the API standards.
31. As-built data sheets
    - As-built data sheets or schematics indicating critical dimensions are key ingredients of a machinery turnaround records system. The merits of cataloging these essential data are self-evident. Observation and determination of wear is important for failure analysis, and as-built data sheets provide a record of materials used in equipment fabrication. Furthermore, these sheets allow both determination and restoration of amounts worn off.

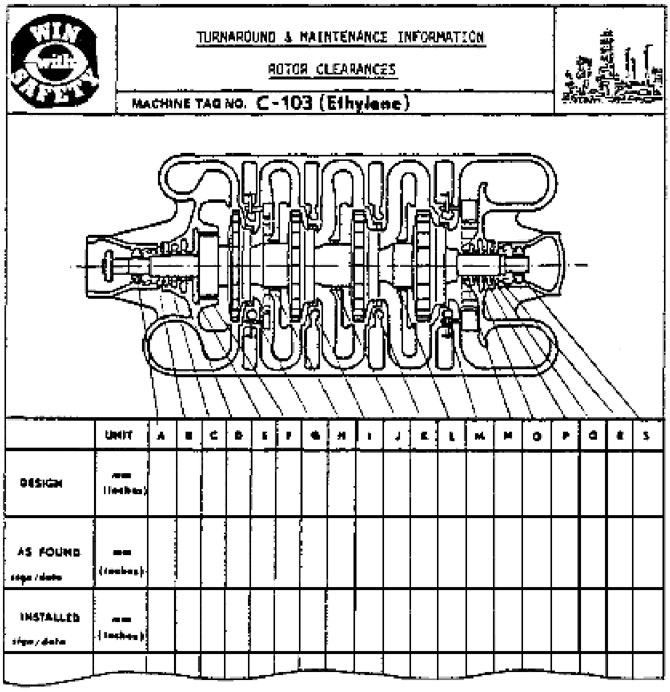


FIGURE 20.4 Using a "machine sketch" and recording clearance data prove helpful in monitoring component wear. (Source: Bloch, Heinz P., *Improving Machinery Reliability*, 3rd Edition, 1998, Gulf Publishing Company, Houston, TX.)

32. As-built dimensions and data
    - Review engineers have the responsibility of retrieving these data in a form that is useful to plant maintenance and technical departments. It is most helpful to let manufacturers provide the data both tabular and pictorially, as shown in Fig. 20.4.

    a. Shaft or sleeve diameters at:
       (1) Thrust collar
       (2) Each seal component
       (3) Each impeller
       (4) Each interstage labyrinth
       (5) Each journal bearing

    b. Each impeller bore

    c. Each labyrinth bore

    d. Each bushing seal component

    e. Each journal-bearing ID

    f. Thrust-bearing concentricity

    g. Metallurgy and heat treatment for:
       (1) Shafts
       (2) Impellers

(3) Thrust collars

(4) Hardness readings when H_2S is present

- Typical wear parts, or parts typically destroyed by massive equipment failures, are listed as items a through f. Metallurgy and heat treatment of highly stressed parts coming in contact with H_2S are important if failure due to stress corrosion cracking is to be avoided (g).

33. Operating and maintenance manuals

- Manuals must be furnished describing installation, operation, and maintenance procedures. Each manual includes the following sections:

Section 1—Installation

a. Storage instructions
b. Foundation
c. Grouting
d. Setting equipment, rigging procedures, and component weights
e. Alignment
f. Piping recommendations
g. Composite outline drawing for compressor train, including anchor-bolt locations

- Although used primarily for installation guidance, Section 1 contains information that should go into purchasers' construction record systems.

Section 2—Operation

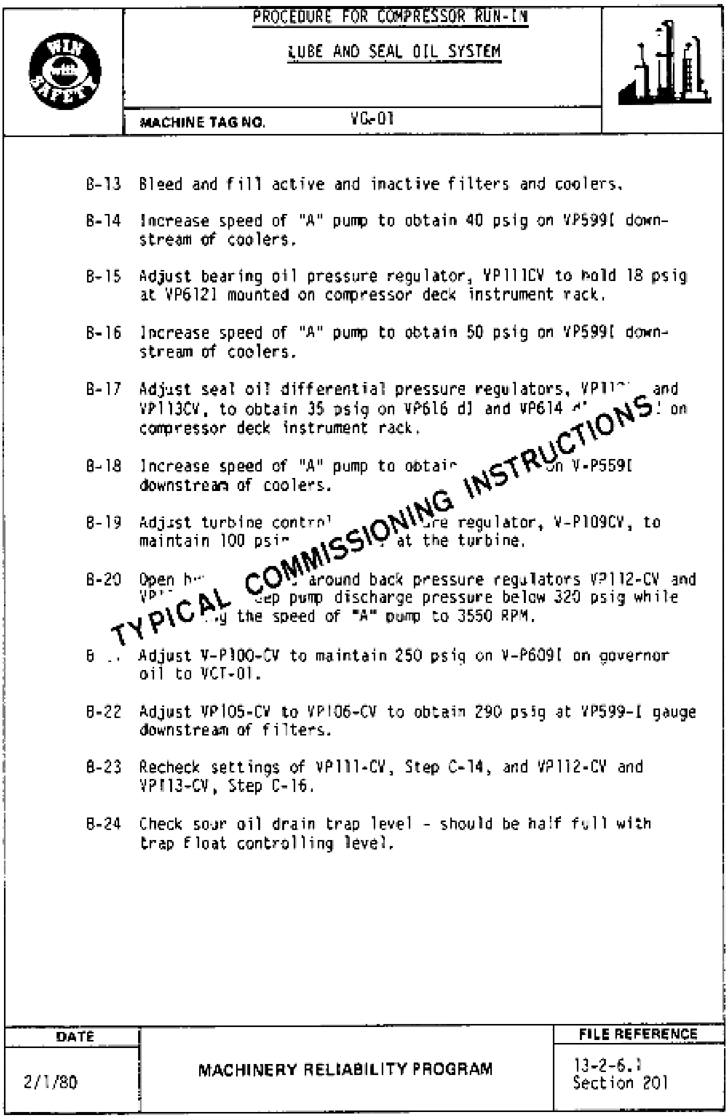
a. Startup
b. Normal shutdown
c. Emergency shutdown
d. Operating limits
e. Routine operational procedures
f. Lube and seal oil recommendations

- Section 2 contains information that purchasers' machinery engineers should utilize in developing such comprehensive machinery instructions as lube and seal oil flushing and checkout procedures, compressor air, helium or vacuum run-in instructions, compressor process runs, and compressor field performance runs. A typical page from this instruction section is shown in Fig. 20.5.

Section 3—Disassembly and Reassembly Instructions

a. Rotor in casing
b. Rotor unstacking and restacking procedures
c. Journal bearings
d. Thrust bearings
e. Seals
f. Thrust collars

- These instructions are indispensable for field maintenance. They should preferably go into a mechanical procedures or turnaround manual. Close screening of these instructions may reveal special tooling or shop facilities requirements.



PROCEDURE FOR COMPRESSOR RUN-IN

LUBE AND SEAL OIL SYSTEM

MACHINE TAG NO. VG-01

B-13 Bleed and fill active and inactive filters and coolers.

B-14 Increase speed of "A" pump to obtain 40 psig on VP599I down-stream of coolers.

B-15 Adjust bearing oil pressure regulator, VP111CV to hold 18 psig at VP612I mounted on compressor deck instrument rack.

B-16 Increase speed of "A" pump to obtain 50 psig on VP599I down-stream of coolers.

B-17 Adjust seal oil differential pressure regulators, VP11[illegible] and VP113CV, to obtain 35 psig on VP616 dI and VP614 [illegible] on compressor deck instrument rack.

B-18 Increase speed of "A" pump to obtai[illegible] on V-P559I downstream of coolers.

B-19 Adjust turbine contr[illegible]re regulator, V-P109CV, to maintain 100 psi[illegible] at the turbine.

B-20 Open b[illegible] around back pressure regulators VP112-CV and VP[illegible]ep pump discharge pressure below 320 psig while [illegible]g the speed of "A" pump to 3550 RPM.

B[illegible] Adjust V-P100-CV to maintain 250 psig on V-P609I on governor oil to VCT-01.

B-22 Adjust VP105-CV to VP106-CV to obtain 290 psig at VP599-I gauge downstream of filters.

B-23 Recheck settings of VP111-CV, Step C-14, and VP112-CV and VP113-CV, Step C-16.

B-24 Check sour oil drain trap level - should be half full with trap float controlling level.

TYPICAL COMMISSIONING INSTRUCTIONS

DATE		FILE REFERENCE
2/1/80	MACHINERY RELIABILITY PROGRAM	13-2-6.1 Section 201

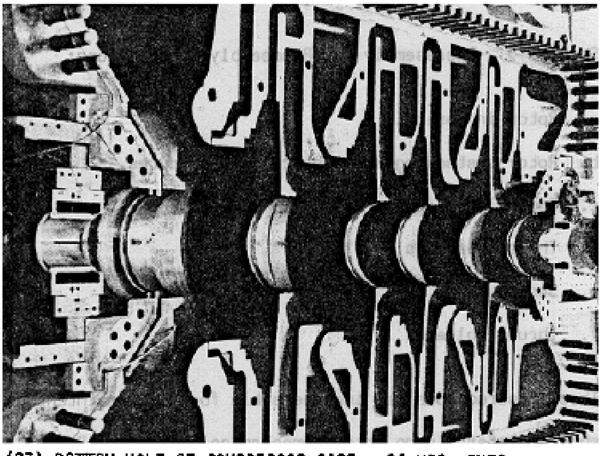
FIGURE 20.5 Detailed commissioning instructions and written operating procedures are valuable aids in reducing human error. (Source: Bloch, Heinz P., *Improving Machinery Reliability*, 3rd Edition, 1998, Gulf Publishing Company, Houston, TX.)

These instructions should make liberal use of photographs and sketches and should, if possible, give the number of labor-hours needed to accomplish a task. Refer to Figs. 20.6 and 20.7 for typical pages.

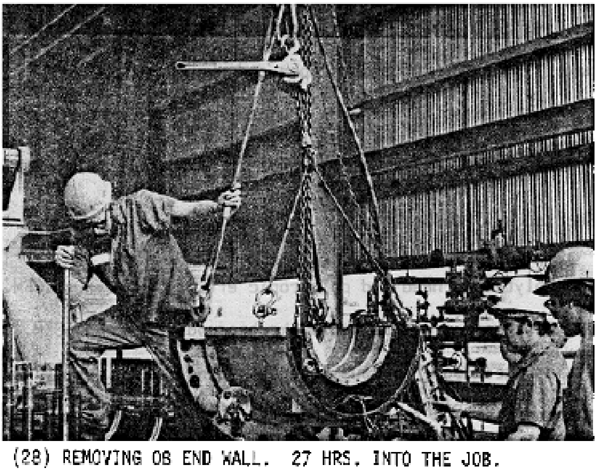
Section 4—Performance Curves

a. Polytropic head and polytropic efficiency vs. icfm
b. Discharge pressure and brake horsepower vs. icfm
c. Balance drum pressure vs. thrust loading
d. Speed vs. starting torque

ZER	MAINTENANCE AND TURNAROUND INFORMATION	ZER
	MACHINE TAG NO. VI-01/02	



(27) BOTTOM HALF OF COMPRESSOR CASE. 26 HRS. INTO THE JOB.

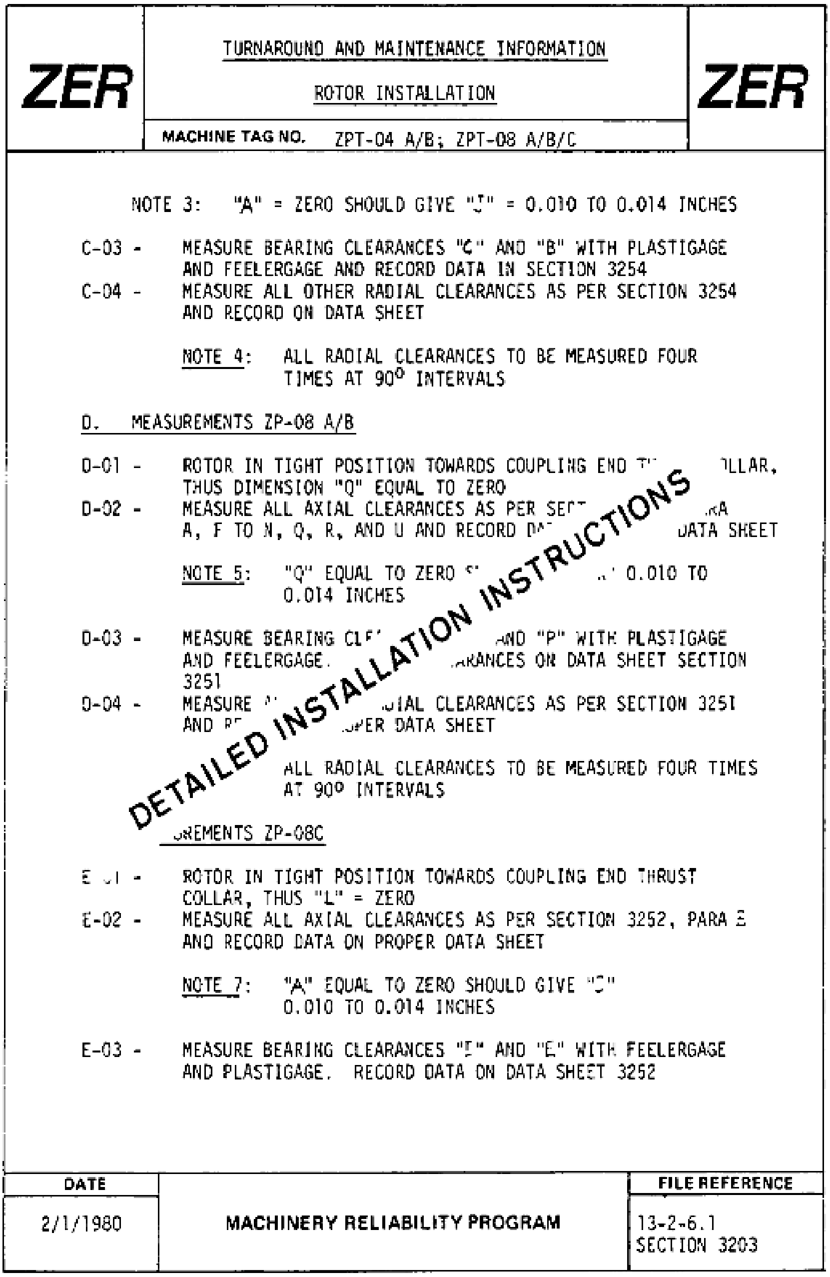


(28) REMOVING OB END WALL. 27 HRS. INTO THE JOB.

DATE		FILE REFERENCE
2/1/80	MACHINERY RELIABILITY PROGRAM	13-2-6.1 SECTION 2501

FIGURE 20.6 Best-in-class companies rely on picture sequences when performing maintenance and turnaround activities on large or critically important machinery. (Source: Bloch, Heinz P., *Improving Machinery Reliability*, 3rd Edition, 1998, Gulf Publishing Company, Houston, TX.)

- Performance curves allow us to review probable vs. actual surge limits, probable uprate capabilities of major machinery, and maximum permissible balance line fouling before the onset of thrust bearing distress. The speed vs. starting torque characteristic curves are important for driver sizing.



ZER | TURNAROUND AND MAINTENANCE INFORMATION | ZER

ROTOR INSTALLATION

MACHINE TAG NO. ZPT-04 A/B; ZPT-08 A/B/C

NOTE 3: "A" = ZERO SHOULD GIVE "J" = 0.010 TO 0.014 INCHES

C-03 - MEASURE BEARING CLEARANCES "C" AND "B" WITH PLASTIGAGE AND FEELERGAGE AND RECORD DATA IN SECTION 3254

C-04 - MEASURE ALL OTHER RADIAL CLEARANCES AS PER SECTION 3254 AND RECORD ON DATA SHEET

NOTE 4: ALL RADIAL CLEARANCES TO BE MEASURED FOUR TIMES AT 90° INTERVALS

D. MEASUREMENTS ZP-08 A/B

D-01 - ROTOR IN TIGHT POSITION TOWARDS COUPLING END T... ...LLAR, THUS DIMENSION "Q" EQUAL TO ZERO

D-02 - MEASURE ALL AXIAL CLEARANCES AS PER SE... ...A A, F TO N, Q, R, AND U AND RECORD D... ...DATA SHEET

NOTE 5: "Q" EQUAL TO ZERO ... 0.010 TO 0.014 INCHES

D-03 - MEASURE BEARING CL... ...AND "P" WITH PLASTIGAGE AND FEELERGAGE. ... ...ARANCES ON DATA SHEET SECTION 3251

D-04 - MEASURE ... ...DIAL CLEARANCES AS PER SECTION 3251 AND R... ...PER DATA SHEET

... ALL RADIAL CLEARANCES TO BE MEASURED FOUR TIMES AT 90° INTERVALS

...SUREMENTS ZP-08C

E-01 - ROTOR IN TIGHT POSITION TOWARDS COUPLING END THRUST COLLAR, THUS "L" = ZERO

E-02 - MEASURE ALL AXIAL CLEARANCES AS PER SECTION 3252, PARA E AND RECORD DATA ON PROPER DATA SHEET

NOTE 7: "A" EQUAL TO ZERO SHOULD GIVE "J" 0.010 TO 0.014 INCHES

E-03 - MEASURE BEARING CLEARANCES "I" AND "E" WITH FEELERGAGE AND PLASTIGAGE. RECORD DATA ON DATA SHEET 3252

DETAILED INSTALLATION INSTRUCTIONS

DATE		FILE REFERENCE
2/1/1980	MACHINERY RELIABILITY PROGRAM	13-2-6.1 SECTION 3203

FIGURE 20.7 Detailed installation instructions tend to decrease failure risk and increase machinery uptime. (Source: Bloch, Heinz P., *Improving Machinery Reliability*, 3rd Edition, 1998, Gulf Publishing Company, Houston, TX.)

Section 5—Vibration Data

a. Vibration analysis data
b. Lateral critical speed analysis
c. Torsional critical speed analysis
d. Transient torsional analysis

- Vibration data submitted previously by the vendor are updated, supplemented, or simply included in their original form for the purpose of having official data in a single manual.

Section 6—As-Built Data

a. As-built data sheets
b. As-built dimensions and data
c. Hydrostatic test logs
d. Mechanical run test logs
e. Rotor balance logs
f. Rotor mechanical and electrical runout

- As above, a comprehensive, final updated issue.

Section 7—Drawing and Data Requirements

a. Certified dimensional outline drawing and list of connections
b. Cross-sectional drawing and bill of materials
c. Rotor drawing and bill of materials
d. Thrust-bearing assembly drawing and bill of materials
e. Journal-bearing assembly drawing and bill of materials
f. Seal assembly drawing and bill of materials
g. Seal oil schematic and bill of materials
h. Seal oil arrangement drawing and list of connections
i. Seal oil component drawings and data
j. Lube oil schematic and bill of materials
k. Lube oil arrangement drawing and list of connections
l. Lube oil component drawings and data
m. Electrical and instrumentation schematics and bill of materials
n. Electrical and instrumentation arrangement drawing and list of connections

- Again, a final wrap-up of certified drawings for purchasers' permanent records.

Vendors usually issue official operating and maintenance manuals around the time completed machines leave their factories. Although this may be an acceptable timetable in view of the fact that all of these data had previously been made available to purchasers or their contractors, any delay in the *originally* scheduled transmission of essential documents may have a serious impact on the completion of construction and therefore on machinery startup targets. To forestall any such delays, project managers would be well advised to link progress payments to the timely and diligent execution of all data transmittal requirements. Once full payment has been made while certain review documents are outstanding, purchasers lose virtually all their leverage and may not be able to proceed with critically important review or audit efforts.

20.6 COMPRESSOR INSPECTION: EXTENSION OF THE AUDIT EFFORT

In the late 1990s and early 2000s, most plants underwent workforce reductions. They have downsized and "right-sized," delegated work to contractors and subcontractors, and gone offshore for labor and materials. Quality control has often suffered in the process; indeed, one of the first job functions to be curtailed or deleted is often the quality control or inspection department.

Of course, delivering a defect-free compressor is acknowledged to be the responsibility of the manufacturer. If the product is delivered with flaws and assuming that these flaws are detected within the warranty period, the manufacturer will implement and pay for repairs. However, the compressor user–owner facility is not being compensated for downtime expenses unless it happens to have (very unusual) zero-deductible insurance coverage for business interruptions. Yet in that particular and certainly rare case, insurance premiums would be exceedingly high.

It follows that arranging for inspection coverage by a firm's own inspectors or by a contract inspection agency is often to the advantage of the purchaser. Unfortunately, neither scope nor required thoroughness of inspection coverage are always appreciated by parties representing the owner. Experience shows that both inspection scope and thoroughness must be predefined and sometimes, negotiated. Although it is certainly not our purpose in this book to give detailed guidance on *all* the elements to be inspected, examining the inspection coverage of a welded impeller and preexisting rotor will serve as an example and will be of help.

The owner's *inspector* function may be outsourced to an experienced inspection company. However, the owner's *engineer* should be a competent person whose involvement in the project started with equipment specification, bid review, and selection. From there, his or her involvement progresses to lead roles in the audit and review process, moving on to field erection, plant startup, and a year or so with the owner's plant. One cannot overlook certain inspection requirements if one claims to be truly reliability focused.

The following example is presented simply for the purpose of illustrating the great degree of detail with which the inspection of compressor components and subassemblies is to be conducted. Our example assumes that an existing machine is being revamped, or uprated. These guidelines can be used to question, challenge, or understand when a service provider or manufacturer deviates from the methods or work procedures. Of course, this questioning process may actually lead to valid explanations that allow you to accept a deviation from past ways of doing things. Similarly, it may lead to insistence that "our way is the less risky way," in which case you will be pleased to have this guidance at your fingertips.

20.6.1 Inspection of a Welded Impeller (Wheel) and the Entire Rotor

Suppose that we were dealing with a compressor designed to operate in a refinery process gas environment. In that case, the owner's inspector (or agency inspector reporting to the owner's machinery engineer) should inspect the following:

- Disassembly (unstacking) of the (in this case, already existing) rotors
- Reconditioning and certification of the shaft
- Reconditioning and certification of (insert number, but assume several in this illustrative example) previously used impellers
- Fabrication and certification of (insert number, but again assume several in this illustrative example) new impellers
- Restacking and certification of the modified rotor

Specific guidelines and areas of concern typically include the activities described below.

Unstacking This entails the removal of impellers from the shaft. If prior experience shows that careless unstacking procedures at the manufacturer's facility had, on other occasions, caused extreme deformation of impeller bores, point this out[3]. After all, we wish to avoid repetition of this problem. Therefore:

1. Determine if the centrifugal rotor assembly is made with uniform shrink fits (typically, 0.00075 to 0.0015 in./in. of shaft diameter). This requires impeller heating or, in extreme cases, a combination process of heating the impeller and cooling the shaft
2. The shrink fits are generally calculated to be released when the impeller is heated to 600°F maximum. To exceed this figure on materials other than AISI 410 or AISI 4140 could result in metallurgical changes in the wheel. For a reliability-focused owner's impellers (AISI 4140), allow a maximum temperature of 400°F. Tempil sticks should be used to ensure that this is not exceeded. The entire diameter of an impeller must be heated uniformly using "rosebud" tips—two or more at the same time.
3. The important thing to remember when removing impellers is that the heat must be applied quickly to the rim section first and after it has been heated, to the hub section, starting at the outside. Never apply heat toward the bore with the remainder of the impeller cool.
4. To disassemble rotors, the parts should be marked carefully as taken apart so that identical parts can be replaced in the proper position. A sketch of rotor component position should be made using the thrust collar or shoulder to adjacent impeller hub exit area. Measure and record the distance between all impellers. Each impeller should be stenciled. From the thrust end, the first impeller should be stenciled T-1; the second wheel, T-2; and so on. If working from the coupling end, stencil the first wheel C-1; the second wheel, C-2; and so on. This requirement would not be significant in a re-rate job, where only some of the impellers are being reused. However, there *is* a possibility that after unstacking, the owner might experience an emergency and would ask for the rotor to quickly be restacked and shipped back to the plant site.

   Following the prescribed marking procedure would be of extreme importance if such a need should develop unexpectedly. A large rotor should preferably be suspended vertically above a sandbox to soften the impact of the impeller as it falls from the shaft. Alternatively, the rotors may be suspended over wooden scaffolding or similar rigging as long as the drop distance from the impeller edge to the wooden structure does not exceed 2 in. A smaller rotor may be mounted horizontally but must then be rotated continuously while the heating and impeller removal procedure is in progress

   It may be necessary to tap the heated impeller with a soft (lead) hammer in order to get it moving. The weight of the impeller should causes it to slide when it is hot enough. (*Note*: We insist on using a soft hammer.)

Impeller Inspection and Overspeed Testing Impeller inspection and overspeed testing are required for new impellers and for any impellers presently in a reliability-focused owner's spare rotor storage. Such testing would also be required on impellers slated for reuse in replacement rotors being assembled by a compressor manufacturer.

Impeller inspection is divided into two logical segments: *before* and *after* overspeed tests. Before overspeed testing, the owner's inspector should be present when the compressor manufacturer conducts important work, such as:

- Ultrasonic testing of forgings prior to machining. If ultrasonic testing has been or will be performed at the foundry, the owner's inspector should review all applicable certificates and forward them to the designated owner's engineer or project team. Impellers should be balanced individually before overspeed testing. Grinding to achieve proper balance shall in no case reduce the remaining material thickness below the drawing-specified dimension. If necessary, the vendor should machine an entire quadrant or similar portion of the impeller.

- Liquid penetrant testing or, alternatively, magnetic particle examination. Liquid penetrant testing or magnetic particle examination should be performed after each weld operation and after each heat cycle. The owner's inspector should visually examine the cover, disks and vanes for surface flaws. This should be done at the same time that preliminary liquid penetrant or magnetic particle examination is made.
- Measurement and recording of critical dimensions.

Fabrication Inspection A reliability-focused owner will actually check the vendor's fabrication, inspection procedures, and workmanship standards for welded impellers as discussed below:

Materials Impeller materials such as 4140 used in typical compressor rotors fall within the requirements for grade B of ASTM A294, a Ni–Cr–Mo alloy steel. This steel can, theoretically, be heat treated to moderately high yield strenghts of 80 to 100,000 psi and ultimate strengths of 110 to 130,000 psi. Conservative, reliability-focused owner companies insist that rotors in H_2S-containing process gas service and have *yield* strength and hardness limitations of 90,000 psi and RC-22, respectively.

1. Hardness limits imposed by H_2S service are an indirect limitation on *yield* strength because of the correlation that exists between tensile strength and hardness. These issues are addressed in such standards as ASTM A-370. However, the hardness limit may be exceeded in the weld region of impellers whose critical dimensions have been fully established.
2. The compressor manufacturer may start with annealed material and heat-treat the completed wheel to obtain the desired physical properties. Alternatively, the manufacturer may begin with quenched and tempered material and post-weld-treat the assembly.
3. Determine if the impeller material requires that the parts be preheated and kept heated during welding. Establish if the weldment must receive postweld heat treatment. Failure to keep some materials hot for welding will cause cracking underneath the bead.

Wheel (Impeller) Assembly Compressor manufacturers, of course, have several wheel designs. The design controls the sequence of assembly, the weld joint configuration, the welding process used, and so on.

1. The owner's inspector should determine the compressor manufacturer's methods in building a wheel, including its workmanship standards. If workmanship is not considered acceptable, this must be resolved through discussion and agreement with the manufacturer and the owner's machinery engineer. Resolution should start at the preinspection meeting. It must be kept in mind that if standards need to be improved, the requests must be made in such a manner that extra charges are avoided. For example, there has been difficulty convincing a few manufacturers that they should be

concerned about undercutting in welds on impellers. Reliability-focused users do not feel that the amount of undercut permitted should exceed the following values:

a. Maximum depth 0.030 in., up to 1 in. long.
b. Maximum depth 0.010 in., up to 6 in. long.
c. Individual linear indications shall not exceed $\frac{3}{16}$ in.
d. Concavity beyond drawing-specified crown must be ground down.
e. *Fillets:* the specified leg length must be maintained. This is an indirect control on maximum throat thickness. The weld bead must be fused at the root and toes.
f. Root porosities are not to exceed $\frac{3}{32}$ in. in diameter. Occasionally, an excessive gap at the vane-disk or cover interface causes root cracking.
g. If full penetration tee welds are not required at the wheel edge, the cross section of the weld on the machined edge of the wheel should be checked visually for root cracks and repaired if any are found.
h. *Pinholes (piping):* maximum diameter $\frac{1}{16}$ in. (1.5 mm); not more than one in each 4 in. ($\sim$100 mm) of weld length. Deviations are allowed for certain materials and should be discussed with the owner's machinery engineer.
i. *Cracks:* transverse—none permitted. Ask if the manufacturer has a procedure!
j. *Notches, slag pockets, and arc craters:* on unfinished impellers, remove by grinding unless the remaining weld metal is under a specified thickness, in which case the area should be filled with clean weld metal.
k. *Spatter:* all spatter must be removed.
l. *Lack of fusion:* none permitted in transverse direction.

2. Unless approved by the owner's machinery engineer, the inspector cannot incur extra charges to obtain improved workmanship.
3. The compressor manufacturer is usually able to provide maps showing indications requiring repair on preused (facility-owned) impellers. The owner's inspector must verify these indications and request the compressor manufacturer's formal advice regarding the cost of required repairs.
4. Base metal indications and their removal on new and/or used impellers are generally governed by the procedures of competent compressor manufacturers.

Impeller (Wheel) Inspection The inspector should spot-check the weld shop periodically to see that the compressor manufacturer's own procedures are being followed. These checks should be performed at agreed-upon times, if necessary with the manufacturer's escort in proprietary areas of the plant. Checks typically cover the following:

1. Joint preparation. A good many designs call for the vanes to be double-fillet-welded to the disk and cover, although butt welds and a slot weld have been used. At points of high stress, as on the eye end of the vane and possibly at the outer end, complete root penetration may be specified. This requires some type of back-grinding, gouging, and so on, after one side of the fillet is made.
2. Measurement of the amount of preheat and interpass temperature being maintained. Low temperature can cause "underbead" cracking.
3. Whether correct electrodes are being used and if they are being cared for properly. Wheel welds may be made in one or two passes. As an example, AWS class E7018

electrodes are commonly used for the root pass of two-pass welds or for one-pass deposits. This electrode has a low hydrogen coating and good resistance to cracking. For the second and final layer of weld metal, an E6027 electrode has been used. This electrode produces flat or slightly concave fillets with fine ripples which minimize the amount of cleaning and finishing required. The E7018 electrode deposit is not quite as good in this respect. Some undercut may be found along the edges where it is difficult to get the electrode in the right position, as in the gas passages. More spatter can also be expected from the 7018 electrode. Note that E7018 electrodes must be kept in an oven at 225°F until actual use. Both electrodes have lower strength than the parent metal. It is estimated that the weld deposit of an E7018 wire, through alloy pickup from the base metal and final heat treatment, might end up with a tensile strength of 75,000 psi. The tensile strength of the outer layer of weld metal E6027 might be between 60,000 and 65,000 psi upon completion. This approach has proven satisfactory for compressor wheels. However, some experts feel that the strength of the weld deposit should match or slightly exceed the strength of the base metal.

4. Because the finished impeller has a Rockwell C hardness limitation due to H_2 service, the base metal in each wheel should be checked with a portable instrument. Accurate determination of weld metal and heat-affected zone hardness on a finished impeller is most difficult without destroying the impeller. The hardness requirement for the weld can be satisfied by having the manufacturer make a mock-up joint using the welding procedure employing the maximum thickness of impeller material to be used, and duplicating the same joint, electrodes, heat treatment, and so on, that each impeller receives. The mock-up should be cut so that the cross section of the weld is exposed and a Rockwell C hardness traverse can be taken across the face. The traverse should be made parallel to and not over 2 mm below the surface. If high hardness is verified, the mock-up must be heat-treated, resectioned, rechecked, and so on, until satisfactory hardness is obtained. The welding procedure, heat treatment, and so on, that produced acceptable hardness levels must be used on the wheels.

This qualification test need not be repeated as long as none of the essential variables are changed. The compressor manufacturer should keep the results of this test on file for at least five years. In other words, if it can be established that the qualification procedure has been followed previously for impellers of identical material, none of the above might have to be applied to the owner's impellers.

Weld Examination

Radiography Method Radiography has not been widely used for checking weld quality in welded impellers, due primarily to the type of welds used and because of wheel configuration. If special application is made of radiography on welded impellers, the acceptance level for weld flaws must be determined at the preinspection meeting. Moreover, it is necessary to agree how often radiography will be used and what follow-up is required when defective welding is found. The inspector must obtain the owner's machinery engineer's approval of the applicable manufacturer's standards. The quality of the radiographs in terms of density, sensitivity, and so on, should correspond to ASME Section VIII, Par. UW-51 standards, bearing in mind that weld configuration and impeller construction may prevent strict compliance with code requirements.

Liquid Penetrant Method This technique only discloses flaws open to the surface. The fluorescent penetrants are more sensitive than the visible dyes because of the viewing conditions. If the vendor opts to use liquid penetrant, the standards should always be reviewed by the inspector. Cracks and cracklike indications are unacceptable. Scattered porosity can be accepted provided that there are fewer than four rounded pores in a line, separated by more than $\frac{1}{16}$ in. edge to edge, axially oriented with respect to the weld. Gross surface porosity density should not exceed that indicated by the medium-porosity chart for $\frac{1}{2}$ in. thick welds in Appendix IV of Section VIII, Div. 1 of the ASME code. More relaxed standards must be approved by the owner's machinery engineer. The weld surface flaw standards in the AWS structural welding code are often considered to be too lenient by reliability-focused owner–users.

Magnetic Particle Method This method is preferred for linear flaws on or within $\frac{1}{8}$ in. of the surface in materials that can be magnetized. To be effective, the magnetic field must be oriented so that it crosses the flaw at an angle of roughly 45°. Fortunately, most flaws in new impellers are longitudinally oriented with respect to the weld. Fatigue cracks in an impeller that has been in service might have random orientation, so that the magnetic field should be applied in two directions roughly 90° apart. It is possible to detect an open gap under a vane where the fillets do not have full penetration. This is especially so if the throat of the *weld* is undersize or the gap is excessively wide. If these indications are strong (heavy), the inspector must be satisfied that the weld is acceptable. In such a case, it might be necessary to weld a mock-up with a known flaw of the type suspected. A cracklike flaw will give a sharper-edged indication. The magnetizing force should meet or exceed ASME Code, Sec. VIII requirements.

All cracks and cracklike flaws are to be addressed as stated earlier. Any porosity indications should be judged the same as those disclosed by liquid penetrant examination.

Ultrasonic Examination If the vendor opts to use this inspection method, the following should apply:

1. The shear wave of the weld will determine the degree of weld penetration and detect flaws per ASME Code, Section VIII. A straight beam can be used on fillet welds. Ultrasonic examination is not used routinely on welded impellers. It has been used for special applications such as checking for underbead cracks and on plug welds, for lack of root penetration. Examination of the fillet welds joining the vane to either the disk or cover presents practical problems. These become more acute when the fillets do not have complete penetration, as is usually the case. The difficulties are:
   a. Flaw orientation. Tight subsurface throat cracks and lack of penetration at the root of the weld may not be found by ultrasonic testing.
   b. Small clearances in the gas passages. Usually, they do not permit the use of crystals inside the passages. This requires that any examination be done through the disk or cover.
   c. Varying material thickness. The disks and covers usually taper in thickness from the hub toward the periphery. Crystal movement must be adjusted to compensate for this.
   d. Ultrasonic testing response from the open root of the tee joint makes interpretation difficult and confusing. With this type of joint only the area underneath the toe of fillets in the disks and covers can be tested confidently. More response is obtained if the weld has complete penetration through the vane.

2. Ordinarily, the compressor manufacturer's standards for flaw acceptance and instrument calibration can be used. As a guide, when difficulty was being experienced with underbead cracking, all flaws with an indicated depth greater than $\frac{1}{8}$ in. in length were rejected.
3. Repairs
   a. If the examinations show defective welds, or the like, the impeller must be repaired, reexamined as before, centrifugated again, and followed by any final NDT required.
   b. If the material air-hardens in response to the heat input from welding and will require preheat, maintenance of interpass temperature, and PWHT, these requirements must be met when repairs are made. The inspector should not accept a repair on an impeller that was not made in accordance with the welding procedure used when the impeller was built, unless it is specifically approved by the owner's machinery engineer.

Dimensional Checking Dimensional checking is required for impeller hub bore, outside diameter, eye diameter, vane width, and disk and cover thickness. Be especially careful not to allow unacceptable tapering or out-of- roundness of impeller bores on impellers that have been removed from a preexisting rotor. Dimensions must be within drawing tolerances or in accordance with engineering instructions superseding these drawings. They should be recorded for comparison with measurement made after overspeed testing. The owner's inspector should always witness these checks.

Overspeed Testing Overspeed testing of each impeller should be witnessed at 115% of compressor maximum continuous speed for proven designs, or 120% of maximum continuous speed for new impeller designs. The overspeed test speeds should be specified in the correct order and may sometimes differ from those presently used in the impellers and rotors operating at the owner's facility. After the overspeed test, each impeller should be visually reexamined and any required NDT examinations witnessed. The inspector should note that the points of highest stress are in the cover close to the eye of the impeller near where the vanes terminate. These are points where indications of possible failure should first show.

1. API 617 assumes that compressor manufacturers have established their own acceptance standards for flaw indications. For casting and forging flaws, a compressor manufacturer's standards can be compared with those given in the ASME Code, Section VIII, Div. I. A good inspector will be familiar with ASME NDT flaw acceptance criteria and will consult all applicable references. If the compressor manufacturer's standards are more lenient than those listed in the applicable ASME documents, the inspector should request instructions from the owner's machinery engineer unless this specification has already covered the deviation.
2. If repairs are necessary, the repaired area must be reexamined by the specified NDT method and the wheel overspeed retested.
3. Impeller diameters, including hub bore, should be rechecked. If the growth exceeds the compressor manufacturer's tolerances, the wheel must be rejected by the owner's inspector. Following that, the compressor manufacturer's proposed action should be referred to the owner's machinery engineer for approval.

Rotor Inspection

1. The major components of the rotor assembly are the shaft, shaft spacers, impellers, balancing drum, and thrust collar. If an order utilizes shafts that have originally run in the owner's compressors, ultrasonic testing of the shaft is not required.
2. The critical shaft dimensions are the diameters over which shrink fits will be made, where keys will be placed, and at the journals. These dimensions must be checked carefully and recorded. The finish of the journals and probe surfaces can be examined again when runout of the rotor is checked.
3. Any potential proposal to correct an undersized journal or shaft area by chrome plating cannot be approved by the inspector; this must be done by the owner's machinery engineer. A reliability-focused owner's basic policy is not to accept plating as a repair for increasing shaft diameters, but there have been cases where 10 to 15 mils (0.010 to 0.015 in. or 0.25 to 0.37 mm) were added to the diameter, and approved.
4. When the impellers are assembled on the shaft with a shrink fit, the inspector should verify that manufacturers' responsible personnel control the bore diameters of the hubs and the temperature to which the impellers are being heated.
5. Before witnessing the final balance, the inspector should review shop assembly records and review the interference fits of wheels on shafts against the manufacturer's standards. Normal interference is 0.001 in. per inch of shaft diameter.
6. High-speed dynamic balancing of the compressor rotors is required and the final balance check must be witnessed. The inspector must know if an incremental balancing procedure is required. Responsible engineering personnel should be identified, and it must be ascertained that such a procedure is actually followed by these personnel.
   a. Runout checks of the assembled rotor should be witnessed. The runout check made after rotor assembly is particularly important, since the measurements will indicate if the rotor has been assembled properly or has bowed due to stresses introduced during assembly or by mishandling.
   b. For a runout check, the rotor can be supported on level knife edges or the check can be made while the rotor is still in the balancing machine. A dial indicator is set up on the diameter to be checked, and the rotor is rotated. The total reading is the runout. Runout checks should be made on the bearing journal surface, the radial vibration probe surface, impeller eyes, and thrust collar surfaces.

These readings are to be compared to those on shop assembly drawings. Any measurements outside of tolerance must be questioned, as there may be bowing of the shaft or assembly errors. The owner's inspector must be certain that the mechanical runout at the radial vibration probe surfaces does not exceed 0.2 mil. The inspector must also check that the shaft surface finish at radial probe locations is equal to the finish on the journals. Axial probe sensing surfaces must be perpendicular to the shaft axis within 0.2 mil.

Next, the electrical runout in the eddy-current probe areas of the shaft must be checked and recorded. If the total (mechanical plus electrical) runout exceeds 0.25 mil on new shafts or 0.5 mil on reused shafts, the surface must be burnished, or fitted with a sleeve. Final compliance must be verified by the owner's inspector. At this stage in the manufacturing cycle, it must be verified that the residual magnetism in shafts and impellers does not exceed 3 G. This is an important requirement that is often overlooked.

Safety The inspection work described above is typical and must be amended or structured for a particular job. It obviously involves close visual examinations, witnessing of tests, and at times, the use of gauges to check the compressor manufacturer's quality control effort. But safety is part of the inspection job. The minimum eye protection required while engaged in this type of work includes safety glasses with side shields. Hearing protection may be required in certain areas of the compressor manufacturer's plant. As a guide, if conversation is difficult due to noise level, use hearing protection. Beware of damage to the fingers while inspecting impellers. One should never examine equipment while suspended from a crane. Also, the inspector must not wear ties, dangling decorations, loose-fitting clothing, or loose long hair while working around rotating machinery.

Often, this work is done on a test stand that presents its own hazards to the inspector. Beware of slippery surfaces; temporary connections of steam, oil, and water; electrical and instrument lines; temporary platforms and access; exposed couplings; and inadequate lighting. The inspector should be careful of tripping hazards, ungrounded electrical test equipment, and temporary manhole or pit covers.

Odd-hour visits require special precautions, and inspectors should plan for their departure by parking their cars in a secure area. One should realize that safety procedures or regulations may be relaxed, exit doors locked, and fire protection unavailable outside normal working hours. Competent inspectors will thus plan their odd-hour visits to include safety considerations. They will plan ahead and will be careful of overhead cranes. Rational people will not work under cranes or around forklift trucks. Competent inspectors stay clear of aisles, check housekeeping around the work area, make a safety appraisal of the activity, and practice exposure control.

20.7 COMPRESSOR INSTALLATION SPECIFICATIONS

Installation specifications are designed to guide a project team in the installation of a centrifugal compressor train. Such information is among the dozens of sets of documents that must be on hand and must either be accepted by, or negotiated with, the various contractors and suppliers involved in a project. Unless the machinery engineer develops, collects, reviews, understands, and verifies the actual implementation of procedures and specifications such as these, satisfactory long-term compressor performance will be difficult to achieve.

It is obvious, then, that compressor train installation and startup require planning and forethought. There are definite routines and structured activities that must be executed by the parties involved. Virtually all of the decisions made at this stage have long-term impact on equipment reliability and plant profitability. It is thus essential to place detailed information about the compressor and its driver in the hands of the project team during the planning stage. At the inception of a project involving compressors, a competent machinery engineer must be assigned to advise process designers on the best equipment choices. Reliable compressors will cost more to purchase, yet will pay back the incremental outlay many times over the life of the plant. There have been many instances where the incremental cost of superior compression machinery was recovered within weeks after a successful startup [4].

In any event, competent machinery engineers should assist the cost estimators in determining accurate project cost. Many projects start out in trouble because of unrealistically low-cost estimates being used to obtain project approval. Once the money is allocated, project costs are often considered firm and escalations are frowned upon. Efforts at coming in under budget generally affect the rotating equipment and maintenance reduction features. The ultimate effect is easy to predict: There will be an increased number of downtime events, and costly repeat maintenance work will reduce plant profits for years to come.

The development of complete compressor design, installation, and commissioning specifications is thus a key mandate for the machinery engineer. He or she must review each specification to see the "up-front" planning that was done. With detailed information presented up front, the engineer is equipped with instructions understood to be binding and will become the "yardstick" used to measure the vendor's compliance.

20.7.1 Field Erection and Installation Specifications for Special-Purpose Machinery

It is neither the purpose nor is it within the scope of this book to give detailed field erection and installation specifications for the many compressor models found in modern process industry. However, these specifications are definitely needed by reliability-focused plants. Moreover, they must be reviewed, understood, and approved by a competent machinery engineer. All of these specifications have a few things in common:

- The scope of a standard must be explained. For example, a field erection and installation standard would cover mandatory requirements governing installation and erection for compressors and drivers mounted on baseplates or soleplates.
- Additional information is almost always superimposed on existing industry standards. An asterisk (*) might be used to indicate that additional information is required. Here, the contractor may have to specify, and the owner's machinery engineer may have to approve information.
- A summary of additional requirements is provided. A separate tabulation of applicable cross-references usually lists documents that have to be used with this standard.
- Design requirements are explained. Concrete foundation must be properly sized and proportioned for adequate machinery support and prevailing piping forces. The complete compressor train (compressor, gear, and motor or other drivers) must have a common foundation.
- Foundations must rest on natural rock or entirely on solid earth or good compacted and stabilized soil. They must be supported on pilings that have a rigid continuous cap or slab cover.
- Foundation must be isolated from all other structures, such as walls, other foundations, or operating platforms. They have to be designed to avoid resonant vibration frequencies at operating speeds, 40 to 50% of operating speeds, rotor critical speeds, gear meshing frequencies, two times operating speeds, and known, specified background vibration frequencies.
- The temperature surrounding a foundation must be analyzed to verify uniformity so as to prevent any distortion and misalignment. Concrete foundations must also be cured properly (approximately 28 days) before loading.
- Foundation arrangements are described. Anchor bolts must be designed by specialty firms and must be sleeved. In most instances, a civil engineer will provide and/or certify a foundation drawing or separate foundation specification.
- Around the perimeter a W-8 or larger I-beam must be properly anchored to the foundation for supporting small piping, conduit, and instruments. Auxiliary structures, including piping, merit special and separate design.
- Typical compressor, gear, and motor foundation arrangements and baseplates must be completely filled with epoxy grout. Soleplates must be completely supported with epoxy grout.

- Reinforcing rods, ties, or any steel members must be a minimum of 2 in. (~50 mm) below concrete surface to permit chipping away 1 in. of concrete without interference.
- A minimum space of 1 in. (~25 mm) must be provided between the foundation and a chock block for proper grout flow. The maximum distance between foundation and baseplate should not exceed 4 in. (~100 mm). The minimum distance between the foundation and the baseplate should not be less than $2\frac{1}{4}$ in. (approximately 55 mm).
- For epoxy chock applications, the distance between the baseplate or soleplate and the top of the grout should be 1 in. (~25 mm) unless otherwise approved by the owner's machinery engineer.
- The chock block arrangement and installation are described. Chock blocks must be sized properly to distribute anchor bolt and machine loads so as not to exceed 10% of the weakest compressive strength material in the foundation structure. (The customary design is 300 psi for concrete.)
- Instructions and appropriate illustrations of field erection and assembly tools must be provided. For instance, a special hydraulic coupling hub-to-output-shaft installation tool will probably be used in most modern plants.

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