TRANSFERRING CUSTODY OF IN-SERVICE TURBOMACHINERY— WHAT SELLERS AND BUYERS SHOULD CONSIDER

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ABSTRACT

Purchasing used machinery and custody transfer of in-service turbomachinery opened new vistas in the realm of process and utility industries. Turbomachinery engineers have to gear up to handle the challenges posed in executing such projects and they ought to know a good deal about control and instrumentation systems to eliminate machinery output limitations. This paper deals with how the custodianship of three gas turbines and two air compressors that were in continuous service catering to the utility needs of an oil refinery was accomplished. Details of eliminating the output limitations posed on gas turbines are discussed. Revisiting the basics solved the air compressor capacity discrepancy and intricacy with gas turbine load ramp rate testing. The integrated approach adapted to test the capacity performance and mechanical reliability including a safeguarding system and concepts used from applicable industry standards and codes with deviations taken during this project are explained. The experiences derived are combined to formulate generic checkpoints that can be used as a practical tool.

INTRODUCTION

Acquiring used turbomachinery for productive utilization has become a common approach in process and utility industries—significant underlying benefits being reduced project duration and cost. Used turbomachinery, by and large, falls into two main categories. The first group is machinery that has ceased to be operational, some even removed from its location, and kept mothballed. The other group, "machinery-in-use" and substantially different from the first group, is machinery that is still in service at the time of custody transfer. The main focus of this paper is on the "in-service" kind of machinery, and obviously the complete system of auxiliaries necessary for reliable operation of pertinent turbomachinery is considered to be part and parcel of the main equipment.

When the custodianship is transferred from one business entity (seller) to another (buyer), in order to take care of commercial interests, several technical aspects are to be considered right from the time the project is conceived. Like any project, the custodianship transfer from seller to buyer has few distinct phases, one of them being testing the turbomachinery for compliance with contractual obligation and subsequent acceptance. A significant part of this paper deals with the testing and acceptance phase, and the entire paper is organized into two portions. In the first part, the author enumerates his experience in testing the turbomachinery onsite including details of how the hurdles of noncompliance with the contractual obligation were dealt with. The remaining portion elaborates on how such experiences result in generalized checkpoints that will be useful to the turbomachinery personnel who might handle such projects.

BACKGROUND

The seller has been operating a refinery that included captive power and a utility plant. Sticking to the core business, the seller wanted to relinquish the operation and maintenance custody of the utility supply system to the buyer. The buyer will take fuel from the seller and in turn sell the utilities, which included boiler quality water, steam, electric power, and compressed air. The buyer augmented the capacity of the utility supply system by means of capital investment.

The ongoing refinery operation had to continue without break, and the custody of the utility supply system had to be transferred maintaining uninterrupted supply of utilities to the refinery. Thus the stage was set where operation and maintenance custody of the power generating units, compressed air generation systems, steam generators, and water desalination units had to be transferred from the seller to the buyer when these units were completely operational and in service.

Power generating units and compressed air generation systems have been taken up for this paper as central subjects. Among power generating units, the gas turbines are covered while the steam turbine driven turbo alternators are excluded since the transfer of this equipment is not completed yet. Units like steam generators and water desalination units are left behind as out of the scope for this paper.

MACHINERY

In total, three gas turbines and two air compressors, one steam turbine driven and the other electric motor driven, are discussed. The list of machinery taken up is provided in Table 1, which includes the technical specification of the machinery. The details provided are the original equipment specification data.

All three gas turbines are single shaft, dual firing capable engines that can use refinery fuel gas and/or distillate as fuel and

Table 1. List of Machinery.

	Capacity	Operating speed in rpm	Rated Power in MW
Gas Turbine 1	21.89 MW	5100	* 24.5 MW
Gas Turbine 2	21.5 MW	5070	* 24.5 MW
Gas Turbine 3	14.5 MW	5070	* 14.5 MW
Air Compressor 1	298.62Nm^3/min	19114	1.97 MW
Air Compressor 2	296 Nm^3/min	8400	2.27 MW

* Power rating at ISO conditions.

all of them are designed for "black" start capability. Startup and synchronizing capability of the gas turbine with a power grid using power supply from its own set of batteries without any resource external to the machinery is known as "black" start capability. Gas Turbine 1 (GT 1) and Gas Turbine 2 (GT 2) are equipped with waste heat boilers (WHB) for heat recovery. Each gas turbine is provided with a dedicated computerized control system that takes care of startup checks and sequence, surveillance of the gas turbine by continuous online monitoring, loading and load ramping controls, etc.

Air Compressor 1 (AC 1) is an integrally geared type compressor driven by a 6.6 kV electric motor running at 1500 rpm. It is a four-stage compressor having two rotors operating at 19,114 rpm; each rotor consists of two mixed flow wheels at each end of the rotor and a pinion gear in the middle. Capacity control is by means of an inlet guide vane at the first stage inlet.

Air Compressor 2 (AC 2) is a horizontally split twin casing centrifugal compressor having two sections, namely low pressure (LP) and high pressure (HP) sections, with an intercooler in between. The prime mover of this compressor is a steam turbine.

CONTRACTUAL PERFORMANCE CRITERIA

All the machinery pertinent to this paper had been in service prior to and during the contracting phase of the project. Therefore, the latest performances of the machinery trains were available along with the maintenance data. Besides the original equipment design specifications, such data as the day-to-day operating performance were utilized to derive the contractual performance values.

A summary of the contractual performance criteria of the machinery trains pertaining to the project and to the paper is provided below:

- *Gas Turbine 1*—To produce 21.5 MW of electricity with turbine exhaust to atmosphere and 19.5 MW with turbine exhaust routed to the waste heat boiler at a lagging power factor of 0.8 on refinery gas as well as gas oil firing mode.

- *Gas Turbine 2*—To produce 19.5 MW of electricity at a lagging power factor of 0.85 on two different firing modes: refinery gas and gas oil firing modes.

- *Gas Turbine 3*—To produce 14.5 MW of electricity at a lagging power factor of 0.85 on refinery gas and gas oil firing modes.

- *Air Compressor 1*—To produce and deliver 296 Nm^3/min (10,453 Nft^3/min dry air at 32°F and 14.7 psia) of compressed air at 7.0 barg (102 psig) pressure and at ambient temperature at a specified location, which is downstream of air-drying units.

- *Air Compressor 2*—To produce and deliver 298 Nm^3/min (10,524 Nft^3/min dry air at 32°F and 14.7 psia) of compressed air at 7.0 barg (102 psig) pressure and at ambient temperature at a specified location, which is downstream of air-drying units.

TEST REQUIREMENTS AND OUTLINE

Capacity Performance

The power output in megawatts (MW) and air output in "normal" cubic meter per minute (Nm^3/min, Nft^3/min) were the parameters underlined in the requirements for transfer of the machinery. Due to the specific nature of the contract for fuel supply by the seller to the buyer, the efficiencies of the machinery were only of academic interest. Basically, there is no clause dedicated to the testing requirement per se; only the specification of machinery is mentioned in the contract.

The buyer requested performance tests according to the internationally followed standards like ASME Power Test Code (PTC). ASME PTC tests require elaborate set up and specific procedure be followed to accomplish such tests and it was not envisaged in the contract in the first place. Hence, on mutual consent, a tailor-made witnessed test procedure was agreed on to prove the capability of the machinery concerned.

Reliability Tests

Mechanical performance of the new machinery is generally tested through a "mechanical run test" as per API Standards. Through a mechanical run test, the machinery supplier shows the customer that the rotordynamics parameters are within certain limits as stipulated by API Standards. For instance, the amplification factor and separation margin, the proximity of the critical speed to the rated operating speed, etc., are checked to see if they conform to the guidelines recommended by API Standards. Such a mechanical run test comprises startup from idle position to the rated speed, operation of the machinery at maximum continuous rated speed of the machinery for four hours, and a coast down operation after the machinery is taken to the overspeed trip level.

The basic objective of the mechanical run test is to ascertain the soundness of the mechanical design and, hence, it is more apt for new machinery. For machinery that has been in operation for several years, it is not necessary to check the mechanical design unless design changes are executed. However, the buyer should be confident that the machinery operates reliably for a sustained period at rated capacity output before taking over the machinery. Hence it was mutually agreed that the machinery will be operated at the contract output for four hours without interruption during which period both performance parameters and reliability parameters like machinery vibration, bearing temperature, etc., will be recorded. Any interruption to the operation of the machinery during this four-hour run test due to factors either internal or external to the machinery system will render the test null and void. Another test will have to be scheduled when the entire system is stable.

Safeguarding Tests

Besides capacity performance tests, several witnessed tests were conducted on the instrumentation and safeguarding systems to ensure proper functioning of the machinery surveillance system. A 100 percent test of the safeguarding system was not practical. For instance the low lubricating oil pressure trip was tested by simulating low pressure at the pressure impulse line leading to the pressure switch. The simulation was accomplished using a block and bleed valve of the pressure impulse line without exposing the machinery to actual low lubricating oil pressure. However, in circumstances where there is no block and bleed valve in the impulse line leading to the pressure switch, the pressure switch was simulated and the trip was checked. Thus, exposure of the bearings to low oil pressures during the test was averted. A mutual consensus was arrived at on similar issues.

Formulation of Team

A dedicated team was formed from the seller's side and the buyer nominated counterparts from their organization. The composition of the team consisted of personnel from operation, maintenance (mechanical, instrumentation, and electrical), and technical services (plant engineering). A manager from the operations division was appointed to lead the team, consulting the legal section as and when required. Formulation of a coherent working team was vital for successful performance testing and custody transfer of these machinery trains.

PERFORMANCE TEST FOR MACHINERY OUTPUT

Prior to scheduling the final witnessed performance test, preliminary tests, "pretests," were conducted by the seller. These pretests brought several bottlenecks to the surface and it is very important to discuss these details, which laid the foundation for the checkpoints produced in the later part of this paper. The seller systematically solved the bottlenecks presented during the pretests, which will be elaborated on in the following sections. Final witnessed tests were conducted once the bottlenecks were eliminated. In the following discussions, only the power figures in MW are indicated and the power factor is deliberately not mentioned since it was always maintained at 0.85 lagging, which conformed to the contractual obligation. The buyer calibrated all of the performance instrumentations recently and the seller calibrated the other indicators both on the machinery and on the auxiliaries. Thus, calibration of the measuring instrument was acceptable for the seller and the buyer.

Gas Turbine 1

Load Limitation—Analysis and Solution

Gas Turbine 1 produced 18.8 MW during the pretest against the contractual quantity of 21.5 MW.

Cause and Effect Diagram—The load limitation of GT 1 to 18.8 MW was systematically analyzed using the cause and effect diagram depicted in Figure 1.

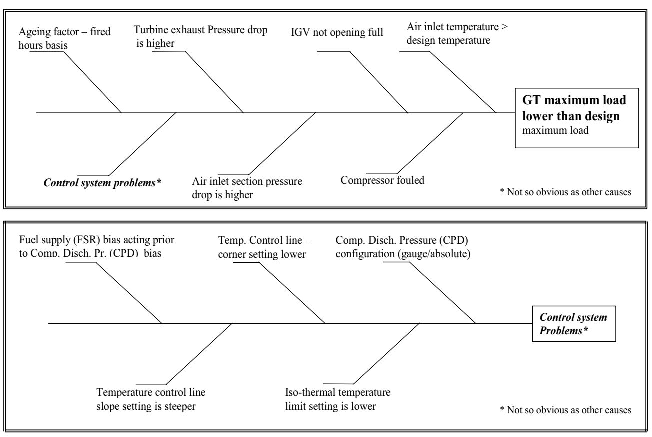


Figure 1. Cause and Effect Diagram—Gas Turbine Maximum Load is Short of Design Level.

The following are the three major areas deduced from Figure 1 that are to be investigated to solve the constraints of load limitation.

- Gas turbine inlet and exhaust system
- Condition and functioning of components of the gas turbine
- Control system configuration

Some constraints like the compressor fouling are obvious and can be tackled easily, whereas some other constraints, especially those related to the control systems, are difficult to identify and thorough understanding of the load limiting function of the control system is essential.

The influence of the prevailing ambient conditions like air inlet temperature and atmospheric pressure, pressure drop factors in the inlet, and exhaust sections of the gas turbine and the aging factor are dealt with in a later part of this paper under the heading *"Reconciliation with Contractual Obligation."* The not-so-obvious causes are discussed here in detail. Prior to indulging into such discussion, a good understanding of the load control system design for the gas turbine is in order.

Controlling and Limiting Parameter for Gas Turbine Output—The combustion process in the gas turbines has to be controlled and limited in order not to exceed the temperature limits that the hot gas path components of the gas turbine are designed to withstand. Typical hot gas path components for the gas turbine in discussion are shown in Figure 2.

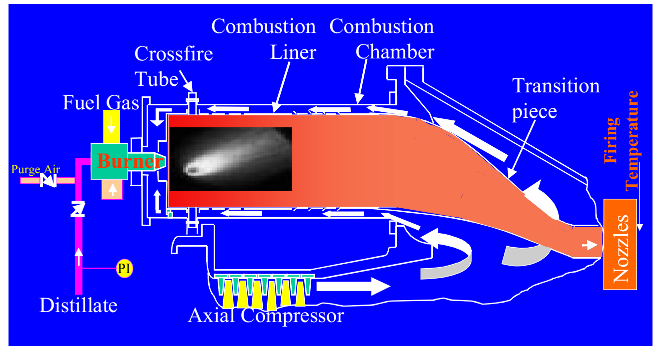


Figure 2 . Hot Gas Path Components—Gas Turbine.

Direct measurement of the temperature at critical locations inside the gas turbine poses difficulties and so the temperatures are measured at the exhaust plenum, "exhaust temperatures," circumferentially at 18 locations. Further combustion process control is based on these measurements. The computer software programmed internally in the control system processes these 18 measurements and arrives at one value, "mean of exhaust temperatures" (TTXM), which will be used for control purposes. The actual process involves deleting the highest and lowest values from the 18 measurements referred to above and correlating to a median value. More details on this calculation are beyond the scope of this paper. Such an elaborate process is necessary to arrive at the exhaust temperature value for control purposes in order to avoid load restrictions imposed on the gas turbine based on one erroneous thermocouple while the machine is adequately protected against excessive firing temperature all the time.

A two-tier control system is programmed in the software to limit the gas turbine load.

- The first control is based on the compressor discharge pressure (CPD), which is a measure of total supply of air and system backpressure caused by fuel burning and is known as "CPD bias." A typical CPD bias curve for GT 1 is shown in Figure 3.

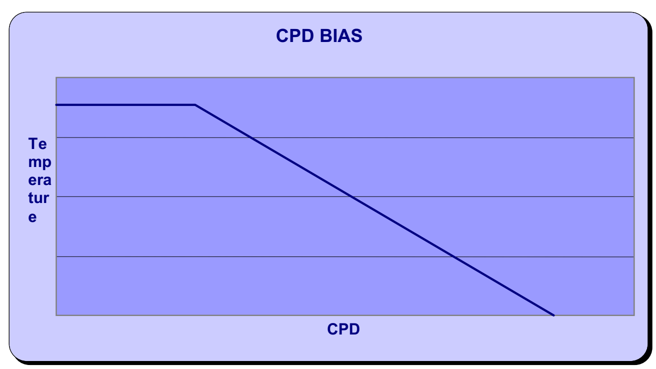


Figure 3 . CPD Bias.

- The other control, which is backup for the first one, is based on the fuel supply to the combustors. The measure used for this control is the position of the fuel supply valve, fuel stroke reference (FSR), which is calibrated for a certain range of fuel supply pressure. This control is known as "FSR bias," the typical curve of which is provided in Figure 4.

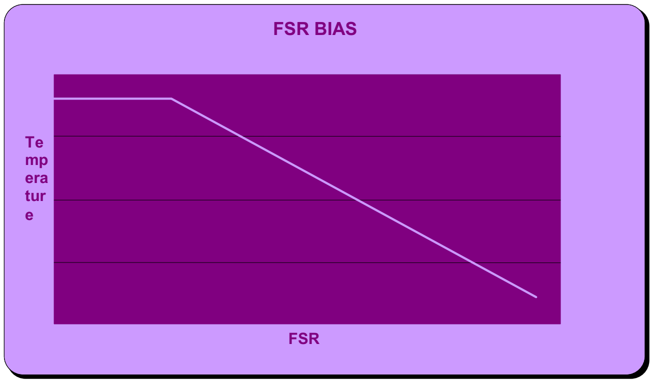


Figure 4 . FSR Bias.

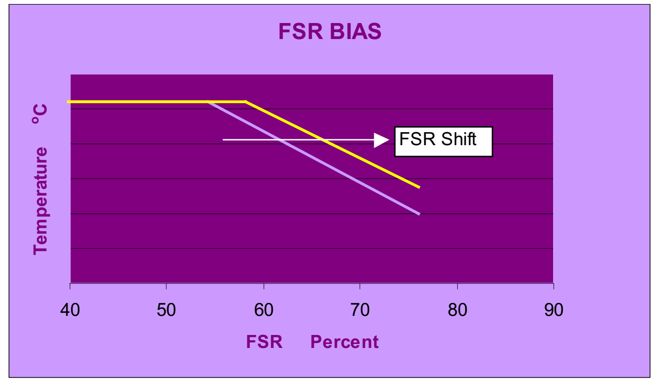


Figure 5 . FSR Bias Shift.

Both the CPD bias and the FSR bias act independently on a predetermined value of exhaust temperature, "isothermal temperature," configured in the control system. Typical value of isothermal temperature for these gas turbines is 520°C (968°F). The exhaust temperature control limit, TTRX, arrived at by the control system CPD bias curve is compared against the TTXM value as calculated from 18 exhaust temperature measurements, and the corresponding output signal is sent to limit the fuel supply to the gas turbine. The backup FSR bias system also engages simultaneously into a similar comparison of exhaust temperature values and arrives at a certain signal output. Under normal circumstances the temperature limit calculated by the FSR bias is marginally higher than that of the CPD bias, which starts limiting the load prior to the FSR bias.

CPD Configuration (Gauge or Absolute)—When the GT 1 load was limited to 18.8 MW instead of 21.5 MW, the exhaust temperature TTXM was 487°C (909°F). This temperature value is much lower compared to the usual value of 512°C to 514°C (954°F to 957°F). Investigation revealed that the compressor discharge pressure going to the control system was based on "absolute" basis, whereas the calibration of the field pressure transmitter that sent the signal to the control system was found to be based on "gauge" basis. Thus the CPD was mistaken by the control system to be higher by about 1 bar (15 psi) and triggered the load limitation at a lower exhaust temperature. After correcting the calibration of the pressure transmitter to "absolute" basis, GT 1 delivered a maximum of 19.5 MW and yet it is less compared to the design level. There exist more limitations.

FSR Bias Versus CPD Bias—The FSR bias constant set in the control system of GT 1 was 54.2 percent and when the GT load was limited to 19.5 MW, the actual FSR reached 54 percent. Thus, the FSR bias took control prior to CPD bias. The situation was corrected by changing the control system constants, as shown in Table 2. The directional shift of the FSR bias control line due to the changes in the control constants is shown schematically in Figure 5.

Table 2. Control System Constants.

	Before correction	After correction
TTKOK Intersection of isothermal and FSR bias line	54.2 %	58.2 %
TTKOM Slope of the FSR bias line	1.82 °C / %	1.67 °C / %
TTKOS Slope of the CPD bias line	15.2 °C / bar	13.5 °C / bar

Before taking a test run, a few additional corrective actions were executed.

Additional Corrective Actions—Industrial heavy-duty gas turbine axial compressors are provided with an interstage blow-off valve to avoid the onset of stage surging at the initial stages and overloading at later stages during startup. GT 1 has a 17-stage axial compressor with a blow-off provision at stage 10. The blow-off valve remains in the open position during startup and when the compressor reaches 80 percent of the operating speed the control system closes the blow-off valve completely. If the valve does not close at operating speed, substantial air will be bypassed through blow-off piping to the exhaust system. The interstage blow-off valve of GT 1 was not closing due to a damaged sealing ring in the piston of the valve. It was repaired. As a parallel action, compressor blade washing was also carried out prior to startup.

Performance of GT 1 after All Corrections—The output of GT 1 improved from 18.8 MW to 20.27 MW after all corrections were carried out. Yet the output was short by 1.23 MW as compared to the contractual obligation.

Reconciliation with Contractual Obligation

The basis of the original specification and verification of contractual capacity in relation to the original specification proved to be essential to resolve the discrepancy between the actual performance and contractual obligation. Complete specification of GT 1 is provided in Table 3.

Table 3. Complete Specifications—GT 1.

	Gas fuel		Distillate fuel	
Open Cycle ISO – Sea level altitude,15° C Site: 3 m and 31° C	15° C 25.28 MW	31° C 21.89 MW	15° C 24.8 MW	31° C 21.47 MW
Compressor and Turbine speed	5100 rpm			

The pressure and temperature conditions at the compressor inlet and turbine exhaust influence the power output of gas turbines. Similarly, the blade and bucket smoothness, internal clearances affected by distortion, etc., will have profound impact on the power generated. The original rating of the gas turbine implies that the blades and buckets are new and internal clearances are as designed by the original equipment manufacturer (OEM). Hence there should be a discount factor based on the fired operating hours of the gas turbine, and it is denoted as the aging factor and is developed empirically by the OEM. The aging factor curve deduced for GT 1 is shown in Figure 6.

Accounting for all these factors the rated power can be calculated as shown in Figure 7. Thus, after correcting for aging and pressure drop factors along with the agreed metering accuracy of ±2 percent, the difference was a maximum of 0.43 MW in distillate firing mode and it was acceptable by both the buyer and the seller.

Gas Turbine 2

A performance test was conducted for four hours similar to Gas Turbine 1 and the results are provided in Table 4. The gas turbine performance exceeded the contractual capacity. The maximum load provided in Table 4 is the load at which Gas Turbine 2 went under exhaust temperature control instead of speed droop control.

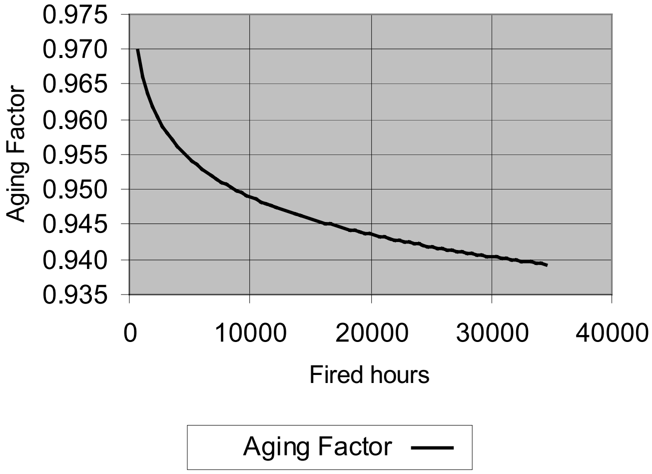


Figure 6 . Aging Factor for GT 1.

September 29,2003

GT 1 Load Correction - Pr. Drops and Aging Factor

Design specification

Air inlet temp.	31°C
Gas firing load in MW	21.89
Liquid firing load in MW	21.47
Contract capacity in MW	21.5

Ageing factor

10,000 hours	0.949
20,000 hours	0.9436

Running hours on March 8, 2001	106422.20
Running hours on Sep. 26, 2003	126901.50
Running hours since last overhaul	20,479.30

Pressure drops at Air inlet and turbine exhaust

Standard pressure drop at air inlet secition in mm Water colmn.	100
Standard pressure drop at turbine exhaust secition in mm Water colmn.	62
Actual pressure drop at air inlet section on 26.9.2003 in mm water column	177
Actual pressure drop at air inlet section on 26.9.2004 in mm water column.	104

Pressure drop factor at air inlet section *	0.98691
Pressure drop factor at turbine exhaust *	0.99706

* Calculated using OEM guideline for additional pressure drop over the standard pressure drop

Design load corrected to Aging factor (20,00hours) and for pr. Drops

Gas firing in MW	**20.32509**
Liquid firing in MW	**19.93512**

Contract capacity corrected to aging factor (20,000 hours) and for pr. Drops

Gas firing in MW	19.96297
Liquid firng in MW	19.96297

Actual load on 26.9.2003

Gas firing in MW	20.03	20.27
Liquid firng in MW	19.44	19.71

Figure 7 . Calculation of Power—GT 1.

Table 4. Performance Test Results.

	Contract Capacity in MW	Capacity during four hours Performance Test in MW	Maximum load tested in MW
Distillate firing	19.5	19.9 0– 20.39	21.46
Gas firing	19.5	19.95 – 20.60	22.03

Ramp Rate Discrepancy

During the performance test one of the performance indicators recorded was the speed with which the gas turbine picked up the load, the "ramp rate." The contract ramp rate was 15 MW/min and during the test maximum ramp rate achieved was 11.58 MW in three minutes and 30 seconds. It was reported by the buyer that the load on GT 2 had to be increased manually when the common power grid frequency reduced to 49.5 Hz from a normal frequency of 50 Hz during one of the upsets. The common grid has multiple suppliers and customers. Hence the complaint was that GT 2 did not pick up the load automatically and did not meet the required ramp rate.

Ramp Rate—Basics and Resolution

The control system of the individual provider to a common power grid comprising multiple providers and consumers can be configured in different ways.

- Isochronous control
- Speed droop control

In order to maintain a constant frequency of 50 Hz in the common grid, one of the providers is configured for isochronous control while others are on speed droop control. GT 2 is designated to be on speed droop control.

Speed Droop Control—The salient characteristics of the speed droop control system are listed here and shown schematically in Figure 8.

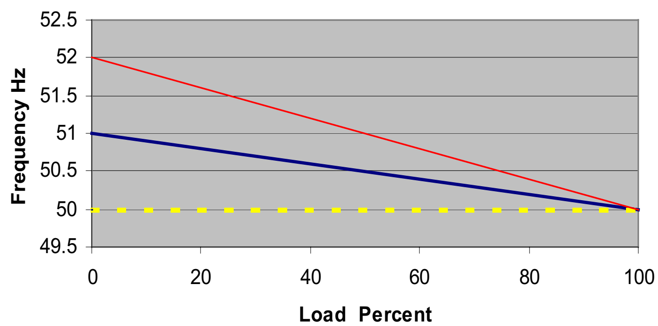


Figure 8 . Speed Droop Control.

- The reference speed setting determines the load of the gas turbine. For constant droop setting, the load of the gas turbine increases with an increasing reference speed setting.
- The load of the gas turbine adjusts itself to the point where the droop line and the common grid frequency cross each other.
- The droop of the governing system is the slope of the droop curve and once configured and calibrated it does not change. Only the relative position of the droop curve can be altered by means of reference speed.

Load Reserve Availability—Reserve load available in the gas turbine is twofold.

- Spinning reserve—automatic.
- Reserve available with manual intervention.

Figure 9 depicts the different level of reserve loads available depending on the extent of the load setting on the gas turbine. When the gas turbine is set at 25 percent of its full load and if the grid frequency drops to 49.5 Hz, the gas turbine load will walk down along the 25 percent load line and will reach equilibrium condition at 50 percent load where the grid frequency and 25 percent droop line intersect each other. This additional 25 percent load is "automatic spinning reserve," which is accomplished automatically and instantaneously by the gas turbine through the control system. However, if the droop line is shifted manually upwards to a 50 percent load curve position, then for the same 49.5 Hz grid frequency, the gas turbine will take 75 percent load, and this 25 percent load over and on top of the automatic spinning reserve is "reserve achieved through manual intervention."

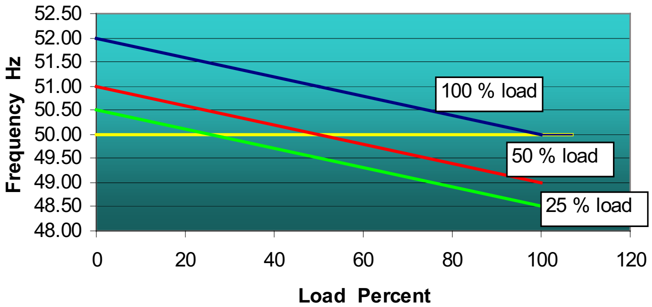


Figure 9 . Reserve Loads Available.

The fact that GT 2 did not pick up more load, despite having reserve capacity, when the common grid frequency dropped to 49.5 Hz is explained by the relative position of the droop line set by the operator at that time. GT 2 picked up the available automatic spinning reserve and further loading had to be accomplished manually.

Testing the Ramp Rate—Testing the high load ramp rate of one producer in a complex electrical network system poses difficulties and has inherent risks of major upset. Furthermore, the manual ramp rate using a push button for step changes is 30 seconds for zero to full load of 21.5 MW. Considering the system inertia, a live test of such a ramp rate was abandoned.

Gas Turbine 3

During the performance test, the gas turbine delivered up to 13.89 MW. However, the vibration of the generator at the exciter end, measured by seismic probe on the bearing housing, crept up to alarm value of 12.5 mm/sec (0.49 in/sec) in a time span of less than two hours. Hence the performance test was repeated at 10.42 MW load for four hours where the machine vibrations and other parameters were normal. The contractual obligation is to deliver 14.5 MW and after correcting to an aging factor based on fired hours of operation, the power to be delivered becomes 13.7 MW.

Gas Turbine GT 3 Behavior at Maximum Load

The thermodynamic and combustion parameters were normal whereas the mechanical parameter, specifically the vibration of the generator, increased gradually. When the vibration level reached the alarm value, the GT 3 load was reduced to 10 MW and the vibration also followed the reducing trend with a time lag and went to the normal operating level in due course. This behavior was very consistent over a substantial period. Detailed vibration analysis indicated the running frequency component of the vibration spectrum was predominant. However, pure unbalance could not be concluded since the phase analysis indicated variation over a time even though the speed was constant. Furthermore, the power factor had significant influence over the duration at which the vibration reached the alarm value.

Considering all these facts, it was concluded that the vibration was due to a temporary bow taken by the generator rotor at a certain temperature due to a possible restriction for free thermal elongation of the rotor copper. The generator rotor was inspected and the cooling air passage holes were checked for free passage and flushed to ensure good cooling. However, the vibration behavior remained the same.

The seller and the buyer are working together to resolve the issue with GT 3.

Air Compressor 1

Capacity Specification in the Contract Between the Seller and the Buyer

The specification is 296 Nm^3/min (10,453 Nft^3/min dry air at 32°F and 14.7 psia) at a pressure of 7 barg (102 psig) at the air interconnection point. The air interconnection point is the common air header downstream of the air dryer units. Therefore, the capacity specified above has to be delivered at a higher pressure than 7 barg (102 psig) since the discharge pressure has to account for the piping friction loss and the pressure drop across the air dryer units. During the performance pretest it was found out that this additional pressure drop was 1 bar (15 psi) and so the contract specification revised for site conditions is 296 Nm^3/min (10,453 Nft^3/min dry air at 32°F and 14.7 psia) at 8 barg (116.6 psig) compressor discharge pressure.

Compressor Design Capacity Versus Contractual Commitment

ICFM, SCFM, and "Normal" Capacity—Volumetric quantities of air, any gaseous substance, depend on the prevailing physical conditions like pressure and temperature besides the composition that includes the moisture content. Pressure, temperature, and relative humidity, a measure of moisture content, vary for different conditions specified like inlet, standard, and normal conditions. Accordingly the volumetric quantity varies for a given mass of air under different conditions mentioned above. Therefore it is imperative that when one specifies volumetric quantity, it has to be qualified clearly with the conditions referred. The only quantity that is unambiguous is the mass of dry air required for the process. Once this is specified, the volumetric quantity can be deduced to any prevailing or specified condition. Even though the common industrial practice is to specify either in terms of inlet cubic feet per minute (ICFM) or standard cubic feet per minute (SCFM), it is recommended to spell out clearly the conditions referred therein. The specification of Air Compressor 1 is calculated for different conditions and the results are shown in Table 5.

Table 5. Air Compressor 1—Capacity at Different Conditions.

Condition	Pressure	Temperature	Relative Humidity	Volumetric capacity
Specification as per the design book - Compressor Inlet conditions	1.013 bar a	31°C	70 %	298.62 m^3/min
"Normal" conditions	1.013 bar a	0° C	0 %	250 Nm^3/min
Contract conditions. It is implied to be prevailing ambient conditions	1.013 bar a	31°C	70 %	296 Nm^3/min

It is clear from Table 5 that the compressor is designed for 250 Nm^3/min (8829 Nft^3/min dry air at 32°F and 14.7 psia), which is equivalent to 298.62 inlet m^3/min (10,545.67 inlet ft^3/min air at 87.8°F, 14.7 psia, and 70 percent relative humidity). The contract capacity is overstated by 46 Nm^3/min (1625 Nft^3/min dry air at 32°F and 14.7 psia) compared to the design and that is 18.4 percent. Subsequent investigation revealed that the flow is measured at discharge of the compressor downstream of the aftercooler and converted into Nm^3/min (Nft^3/min) and indicated as just m^3/min (ft^3/min). Absolute value of this capacity figure was taken and designated in the contract as Nm^3/min (Nft^3/min).

Compressor Performance and Capacity Reconciliation

The compressor capacity control is by means of adjustable inlet guide vanes. During the original shop test, the compressor delivered a guaranteed capacity of 298 Nm^3/min (10,524 Nft^3/min dry air at 32°F and 14.7 psia) at 75 percent guide vane open position. This design margin was utilized during the witnessed performance test and the guide vane setting was set at a 100 percent open position. Table 6 provides the data.

Table 6. Compressor Performance Summary.

	Units	Design Data		Performance Test	
Discharge Pressure	Bar a	10.0	9.5	10.0	9.4
Volume flow at inlet conditions	Inlet m^3/min	298.62	301.10	296.2 to 298.0	309.2 to 312.7
Normal Flow	N m^3/min	250.0	266.7	262.39 to 264.0	274.1 to 277.0

The compressor delivered the capacity of 296.2 to 298 inlet m^3/min (10,460.2 to 10,524 inlet ft^3/min air at 87.8°F, 14.7 psia, and 70 percent relative humidity) at the discharge pressure of 10.0 bara (145 psia) and so it performed as per the design.

On "normalizing" the flow, the compressor delivered 274.1 to 277.0 Nm^3/min (9679.8 to 9782.2 Nft^3/min dry air at 32°F and 14.7 psia) at discharge pressure of 9.4 bara (136.3 psia) against the contract capacity of 296 Nm^3/min (10,453 Nft^3/min dry air at 32°F and 14.7 psia). On mutual consent by the seller and the buyer, the discrepancy was accommodated by suitably altering related clauses of the contract.

Air Compressor 2

Table 7 shows the performance test data along with the design values. The salient aspects of the test and data evaluation are:

Table 7. Air Compressor 2 Performance Test Data.

	Units	Design data		Performance Test
Discharge pressure	Bar a	10.6	9.6	9.6 to 9.7
Speed	rpm	8400	8400	8180 to 8272
Weight flow (inlet)	Kg/hr	21585	22800	22030 to 23233
Volume flow at inlet conditions	Inlet m^3/min	325	353	316 to 333
				324 to 338 (Corrected to 8400 rpm)
Inlet flow at "normal" conditions	N m^3/min	278	294	284 to 299 (Test Speed)
				291 to 303 (Corrected to 8400 rpm)

- The guarantee conditions of the original specifications were taken as reference for comparison.
- During the performance test run the compressor could not be set at discharge pressure of 10.6 bara (153.7 psia) as per the original design. The pressure that could be reached was 9.6 bara (139.2 psia) and the limitation was the steam conditions available at that time. It was mutually agreed that the performance figures will be extracted from the tested compressor curve for the pressure of 9.6 bara (139.2 psia) and the tested flow and pressure for lower speed will be calculated for design speed of 8400 rpm using fan laws.

These highlighted data from Table 7 are the final result of the performance data evaluation. The compressor performed as per the design and delivered the contract capacity of 298 Nm^3/min (10,524 Nft^3/min dry air at 32°F and 14.7 psia).

RELIABILITY AND SAFEGUARDING TESTS

Performance tests for capacity compliance are important. Equally important, if not more, are the tests to ensure reliability and safeguarding of the installation. Such tests are also as elaborate as performance tests. Safeguarding items provided for the specific machinery were tested, separately from the performance tests, whereas the reliability parameters were checked during the four-hour performance tests and thus it was integrated with the performance test.

Reliability Parameters and Tests

Key reliability parameters are vibration, bearing temperatures, axial position measurements, lubricating oil pressures and temperatures, and any other abnormality like sound, leakage, etc. Most of these parameters pertain to a typical "mechanical run test" in the realm of turbomachinery tests. One vital aspect with these parameters is the stabilization of the condition. Therefore, either snapshots or very large intervals between measurements, say more than 15 minutes, are not recommended. All of the machinery included in this paper have most of these parameters recorded in a distributed control system (DCS). Hence even though snapshot measurements were included in the report, it was ensured and witnessed that the reported values reflect the right trend of these parameters.

Safeguarding Tests

Gas Turbines

The safeguarding system provided for all three gas turbines was conceptually the same. Table 8 shows the list of safeguarding items that were tested either live or by simulation as detailed in the table.

Table 8. Safeguarding Tests—Gas Turbines.

Safeguarding Item	Testing method	Comments
1. Reverse power trip of the Generator	Live	When the machine was unloaded, the Gas Turbine (GT) trips out of the grid and goes to Full Speed No Load (FSNL).
2. Emergency manual trip	Live/Simulation	Emergency trips from the local cabin, from the control room (remote) and via the trip lever at the machine.
3. Flame detector eye alarm and trip	Partial Live	The flame eye passage was blocked when the GT was at FSNL.
4. Lubricating oil pressure low alarm and trip	Simulation	The pressure switch was simulated and the auxiliary pump cut in function and later trip function were tested.
5. Fire detector	Simulation	Smoke detector switch was simulated to check activation of CO_2 system and later trip function.
6. Exhaust over temperature trip	Simulation	The control system trip set point was lowered and the GT tripped from FSNL.
7. Overspeed trip	Simulation and Live	Electronic overspeed trip setting was lowered and the G T speed was increased to reach the set value to trip the GT.
8. High vibration alarm and trip	Live and Simulation	High vibration trip set value was lowered in the control system to normal operating level vibration and the GT tripped.
9. Gas fuel knockout vessel level high alarm and trip	Simulation	The level was simulated and the GT transferred from gas firing to Distillate firing mode. When the level reached the trip setting in gas firing mode the GT tripped.
10. Stack damper position	Live and Simulation	In the Gas Turbine 1 and 2, where the Waste Heat Boiler is part of the machinery, functioning of the Boiler and Stack dampers were tested.

Mutual agreement on the testing method is vital prior to conducting these tests. The control system constants were recorded prior to any test in order to preserve the design settings. Wherever necessary, the control system constants had to be altered to facilitate testing, which was normalized after the test. Both changing the constants as well as restoration of the constants were executed by the seller and witnessed by the buyer.

Air Compressors

The safeguarding systems of air compressors are not as complex as gas turbines. However, for the sake of completeness, Table 9 was produced.

ADDITIONAL TESTS SPECIFIC TO GAS TURBINES

Contractual obligation is to hand over the gas turbines in good working order with specified capabilities, which include the following:

- Cold start, hot start, black start.
- Online and automatic firing mode transfer from gas firing mode to distillate firing mode and vice versa.
- Online and automatic exhaust mode transfer from ambient mode to WHB mode and vice versa.

Even though no test to verify these capabilities was stipulated in the contractual agreement, it is imperative that the seller demonstrates the capabilities to the buyer by means of witnessed tests. Some salient aspects of such tests are listed below:

Table 9. Safeguarding Tests—Air Compressors.

Safeguarding Item	Testing method	Comments
1. Lubricating oil alarm and trip.	Simulation	The pressure signal lines were blocked and pressure was reduced by bleeding. Thus auxiliary pump cut in function and trip were checked.
2. Anti surge blow off valve	Simulation	The relevant pressure and flow switches were simulated and the blow off action was checked. The compressor was at low load.
3. High vibration alarm and trip	Partial Live	The alarm and trip/2nd alarm settings were reduced in the vibration monitoring system.
4. Rotor axial position alarm and trip	Partial Live	The alarm and trip alarm settings were reduced in the axial position monitoring system.
5. High discharge air temperature alarm and trip	Simulation	The temperature element was put in a heated oil bath and the tip function was tested.
6. Liquid level alarm and trip in knockout vessels.	Simulation	The level instrument was simulated to check the functioning.
7. Push button trip from local and remote panels.	Live	The remote push button trip had to be configured by the Buyer.
8. Overspeed trip for the Steam Turbine.	Live/Simulation	The overspeed trip setting in electronic surveillance system was lowered and test was conducted.

- The purpose of the cold start and hot start tests are to demonstrate that the control system governing the startup sequence, fuel supply system, and the combustion process function are normal besides the mechanical behavior like vibration, bearing temperature, etc., is within acceptable levels.
- A cold start warrants a cooling down period of 24 hours. Temperature indication anywhere along the hot gas path including the exhaust plenum should not be more than 25°C (77°F) over the ambient temperature.
- The exhaust temperature spread and absolute temperatures are important from an excessive combustion perspective.
- The load on the gas turbine will fluctuate if the firing mode is transferred when the gas turbine is coupled to the grid. Such fluctuation depends, among several parameters, on the prevailing load at the time of transfer, FSR calibration, and calorific value of the fuels, etc. Load variation during the transfer test was between 1 MW and 3.3. MW.
- The exhaust mode transfer test ensures that the gas turbine and WHB can go on stand-alone mode whenever one of the two suffers an upset and trips.

TEST CODES AND ACTUAL TESTS

Performance tests according to the codes followed by the industry are intended for new machinery and consume significant resources, time, and money. Insistence on such tests at the conceptual stage itself might have adverse impact on even the total project, since not many owners of used or in-service machinery will be willing to undertake such liability. At the same juncture, the buyer ought to ensure that the machinery delivers what was agreed on with the seller, before taking custodianship. Therefore, a certain kind of "improvised test" is in order. For these improvised tests to be dependable, many of the underlying concepts from the industry acknowledged standards and codes should be followed. The standards and codes referred to with relevance to the machinery discussed in this paper are:

- ASME PTC 22, 1997, Performance Test Code on Gas Turbines.
- ASME PTC 10, 1997, Performance Test Code on Compressors and Exhausters.
- API Standard 616, Gas Turbines for the Petroleum, Chemical and Gas Industry Services.
- API Standard 672, Packaged Integrally Geared Centrifugal Air Compressors for Petroleum, Chemical and Gas Industry Services.
- API Standard 617, Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services.

Beyond this section, the standards and codes listed above are referred to as PTC 22, PTC 10, API 616, API 672, and API 617, respectively.

Similarities and Contrasts

Highlights of the similarities and deviations between the actual improvised tests conducted in the field and relevant clauses of the standards and codes are summarized as follows.

Gas Turbines

- PTC 22 does not stipulate how the test results can be correlated to the contractual commitment. Furthermore, it recommends that the parties engaged in the test agree on the method of correlation. The load indicated in the contract between the buyer and the seller was consensually agreed to be the load at the ambient conditions provided as per the original specification. Rating for the prevailing ambient conditions was deduced as necessary. Influence of the ambient conditions, pressure, temperature, and humidity, cannot be overemphasized.
- PTC 22 aims to ascertain the output power along with the thermal efficiency of gas turbines. The objective of the test was to determine only the load. Thermal efficiency evaluation was not contemplated.
- Even though several points contained in PTC 22 regarding the agreement before the tests were taken care of, the all important point of the timing of the test could not be achieved. Timing of the test is specified by PTC 22 to be the soonest possible time following commissioning of the gas turbine. It can be construed, in case of used or in-service machinery, as immediately after major overhaul where the turbine blades and buckets are renewed and the internals of the gas turbine are in as-new condition. However due to practical reasons and due to the fact that the gas turbines had to supply power to the operating refinery on an uninterrupted basis, such timing of the test could not be realized. As an alternate, the seller revised the contract capacity based on the empirical guideline for evaluation of the actual test results. The buyer agreed with this approach after the evaluation even though it was not discussed prior to the test.
- The electrical power output variation during the four-hour run test was confined within ±2 percent for most of the test duration. There were instances where the variation shot to ±3.3 percent. PTC 22 limits the variation to ±2 percent. However, the test duration of four hours far exceeds the PTC 22 recommendation of a maximum of 30 minutes.
- The time interval between the measurements was also longer, 15 to 20 minutes, in the actual tests compared to the PTC 22 guideline of a maximum of 10 minutes.
- The mechanical run test as per API 616 intends to test only the gas turbine on no load condition. Even the "complete unit test," which is optional as per the same standard, is to test the gas turbine on no load. The objective of mechanical reliability testing was met by the actual integrated performance and mechanical run tests.

Air Compressors

- PTC 10 does not stipulate how the test results can be correlated to the contractual commitment. Furthermore, it recommends that the parties engaged in the test agree on the method of correlation. Capacity is mentioned in terms of Nm^3/min (Nft^3/min) in the contract and the test procedures agreed to by the parties indicate that the evaluation will be on the basis of actual flow converted to "normal" flow.
- API 672 incorporates a combined mechanical and performance test for integrally geared air compressors and mandates zero percent negative tolerance at the rated discharge pressure and at rated speed. PTC 10 allows a capacity deviation of up to 4 percent

for a Type 1 test, which is comparable to our test. In view of the fact that the tested compressors in our case were online with live refinery and utility plants, the capacity deviation allowable during the test was agreed to be 3 percent. However, an additional clause was included that deviation of even up to 5 percent will be considered mutually by both the parties for possible acceptance based on overall performance and prevailing condition at the time of such deviation.

- Configurations of the instrumentation arrangement as per PTC 10 were not followed. The as-built instrumentation system according to the original process and instruments diagram (P&ID) and piping diagram was used for the test. Therefore, it was acknowledged by the buyer and the seller that there could be differences between the performance during the test onsite and the performances measured as per PTC 10 at the time the seller bought the equipment from the manufacturer. The objective of the test was to show the capability of the compressors regarding the delivery capacity and not to verify if the compressor performance matches the original tested curve exactly.

- The mechanical run test as per API 672 and API 617 calls for the vibration measurement sweep to cover 0.25 times to eight times the rated operating speed. Such specific measurements were not included in the actual test. One of the conditions for successful completion of the combined performance and mechanical run test was that the machine should operate without generating any alarm on any of the parameters. The parameters for performance and surveillance monitoring as designed originally by the manufacturer are retained. All this machinery was built and tested as per one or more of the standards referred to above.

GENERALIZED CHECKPOINTS

Projects have several phases and the author identifies three distinct phases, from the perspective of the turbomachinery engineer, in projects involving used turbomachinery, including the in-service group of machinery.

- Stage 1, Conceptual Phase—Definition of requirement or availability

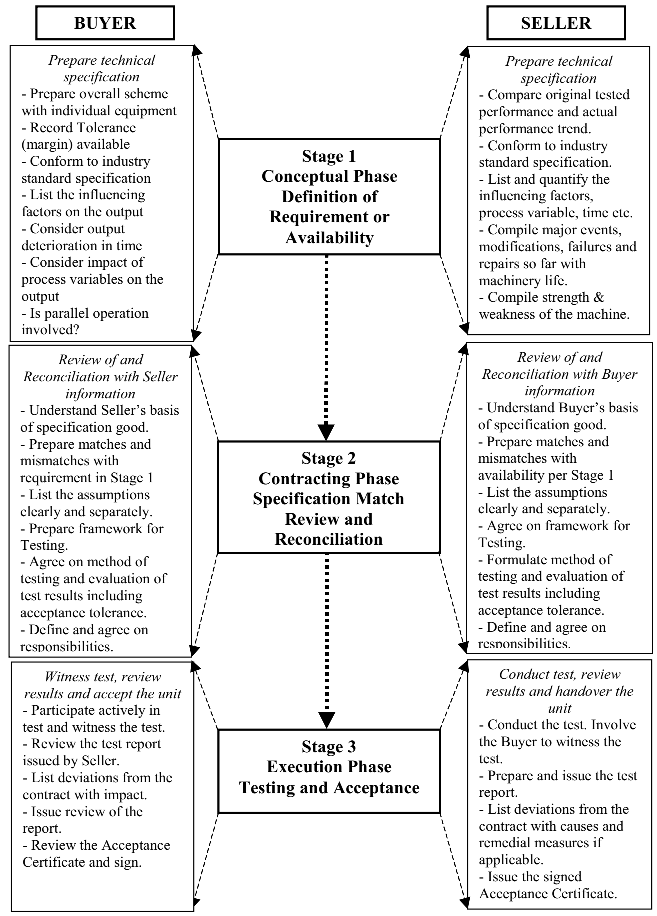
- Stage 2, Contracting Phase—Matching specification, review, and reconciliation

- Stage 3, Execution Phase—Testing and acceptance

Figure 10 shows these three stages schematically and includes the activities encompassed in each of the stages. The schematic also includes the generic checkpoints to be examined in each stage. The list of checkpoints provided is not exhaustive in any sense but is intended to be indicative. The author forges a framework out of his experiences and encourages the readers to add checkpoints relevant to each type of machinery and to each project.

Commercial and Legal Aspects of Project

In projects such as discussed in this paper, no technical activity can take place in isolation, without commercial interests. Once commercial contracts and activities are contemplated, involvement of legal experts is essential. Therefore, active participation of commercial and legal experts, to a varying degree, in all the three stages becomes vital. The success of commercial and legal experts in Stage 1, and the first half of Stage 2, largely depends on how unambiguous the technical specification is besides the extent of technical background details available as per the checkpoints of Figure 10. Potential involvement of survey engineers from financial organizations who provide a financial guarantee for the project cannot be ignored. Such involvement will lead to possible rigorous and stringent testing methods making final agreements more laborious and complicated. The contract-drafting phase should consider all such processes in advance.



Note: The generic list is indicative and from technical perspective only

Figure 10. Generalized Checkpoints.

MISCELLANEOUS POINTS FROM PLANT NOTEBOOK

The author shares the following aspects extracted from the plant notebook used during this project.

- Spare parts to be included during the transfer of equipment should be clearly agreed on during the contracting phase. Neither the OEM recommended list of spares nor the list proposed unilaterally either by the seller or by the buyer might be optimum and agreeable for both parties. Such lists can only be references. Detailed assessment of spares is impractical at this juncture. Therefore, a clear and well-defined concept on spares will avoid prolonged discussions in the later stage of the project.

- Time limits for document review should be made clear in the protocol.

- When an adverse situation presents itself in the middle of the project, which might delay the handover process, seller or buyer might be forced to agree on one-sided amendments. To avoid such inconveniences, a contingency planning clause to phase out the handing over process should be included in the contract.

- If parallel operations of centrifugal compressors are involved, review the characteristic curves and the control system configuration to ensure compatibility for parallel operation between machinery designed separately and independently.

- Avoid, to the extent possible, situations of shared responsibility. Especially the situation where one party is responsible for operation and the other party for maintenance of the same machinery. This should be a last resort solution. If such a situation becomes inevitable, take time to prepare another agreement to effect the transfer of machinery taking care of the interests of buyer and seller.

- If certain readings cannot be practically taken during the testing, it should be evaluated if there is substitute measurement possible in order to continue the test. Later on the original provision can be restored. The gearbox bearing temperature resistance transducer (RTD) was defective in Gas Turbine 2. The gearbox had to be opened to install a new RTD. The entire test was not jeopardized for want of this RTD. Instead the bearing return-oil temperature was substituted as a corroborative measure for bearing temperature and the test was continued.
- The instrumentation system test should include checking the set value at the time of action and not just action like on or off.

CONCLUSION

Three gas turbines and two air compressors were tested onsite successfully and handed over to the buyer. Several obstacles in achieving the specified output in the gas turbines like shortfall in output due to control system configuration, aging factor, and additional pressure drops in the inlet and exhaust systems were all solved and accounted for, to facilitate the transfer of custody. An air compressor capacity discrepancy was cleared by revisiting the basis of specification and an addendum to the original covenant had to be drawn out prior to handing over the machine. The industry standards and codes were used to understand, formulate, and execute various testing methods. Besides the best use of the codes, suitable and informed deviations were taken on a mutual agreement basis, which facilitated progress of the project without additional resources. Finally, the essence of the experiences is distilled into useful learning points and shared through this paper. A useful tool is formulated in the form of framework and checkpoints are indicated. Additional points typical to the situation on hand can be added to this list to make this more complete and more useful to both buyer and seller, to intermediaries, and to turbomachinery survey personnel from financial organizations.