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HYDROCARBON PROCESSING®
November 2000 issue, pgs 61–64
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Optimize gas turbine-driven centrifugal compressors

'Designed engineering' increases output by 100–500 kW

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At locations and plant sites where steam is not available, dual-shaft gas turbines are often used as drivers. These turbines have the ability to meet process variations by allowing the power turbine to operate at the speed dictated by system resistance curve. However, gas turbine limitations must be considered or output losses ranging from 100 kW (for a small compressor duty of 1 MW) to 500 kW (for larger duties of 15 MW and above) may result.

Gas turbines from 1,000 kW to 30,000 kW have been used as drivers for centrifugal compressors mainly in offshore and remote area applications. Two-shaft gas turbines have two separate rotors—the air compressor (also commonly referred to as gas producer) and power turbine. Both operate independently at separate speeds. The air compressor rotor operates at a speed dictated by the power required. The power turbine rotor operates at a speed required by the process compressor to develop the required discharge pressure.

During the design phase, various operating conditions are identified for the compressor. A common perception is that operators want an unrealistically flexible compressor that will meet all imaginable operating conditions. Wide operating ranges are nevertheless a necessity, and the design engineer must closely interact with both operators and vendors to resolve all the compressor design requirements for it to operate at maximum efficiency for extended periods.

If during the initial engineering stage proper consideration is not given to the various design factors, the operator will be saddled with an inefficient machine and high operating costs for many years. This cost can be significant: up to 100 kW (for a small machine) to 500 kW (for a large machine) loss per hour. Assuming $0.06 per kWh cost, a 100 kW loss is equivalent to $48,000 in energy cost alone (assuming 8,000 hr of operation per year). To this, one has to add production losses. From the foregoing, it is possible to determine life cycle losses.

To identify various losses, three factors need careful evaluation during the engineering stage:

1. Staging centrifugal compressor selection and matching it with the gas turbine that will result in the compressor operating at maximum efficiency most of the time.
2. Factors that affect gas turbine efficiency and power output.
3. Relevance and application of API 617 *(Centrifugal Compressors for Oil and Gas Industry)* requirements for 10% margin on rated power.

Table 1. Effects of molecular weight changes

Change in molecular weight	Process compressor speed	Process compressor efficiency	Effect on power turbine efficiency	Gas turbine power output
Higher than predicted	Decreases	Decreases as available power decreases	Decreases	Decreases
Lower than predicted	Increases	Generally decreases and machine operates nearer to surge line	Increases	Increases

Staging centrifugal compressor selection. Staging gas turbine-driven centrifugal compressor selection depends on:

- Optimum power turbine speed
- Number of impellers needed for developing the required head
- Intercooling requirements.

However, the following factors affect compressor operating efficiency and need to be considered during staging selection:

- ▶ Gas molecular weight
- ▶ Compressor startup case requirement
- ▶ Pre/intercooling requirements
- ▶ Condensate (heavy ends) recovery
- ▶ Suction valve throttling to meet process requirements.

Effects of change in molecular weight are shown in Table 1. As the operating speed changes, power turbine efficiency will change. This is due to the geometry of power turbine blades that are designed to receive hot gases at a particular angle, usually optimized for 100% power turbine speed. As power turbine speed changes, the hot gases impinge on the back side of the power turbine blades and energy conversion is reduced.

Little can be done about molecular weight change after the compressor is installed. However, the operator should verify the gas molecular weight after commissioning and consider monitoring actual molecular weight every six

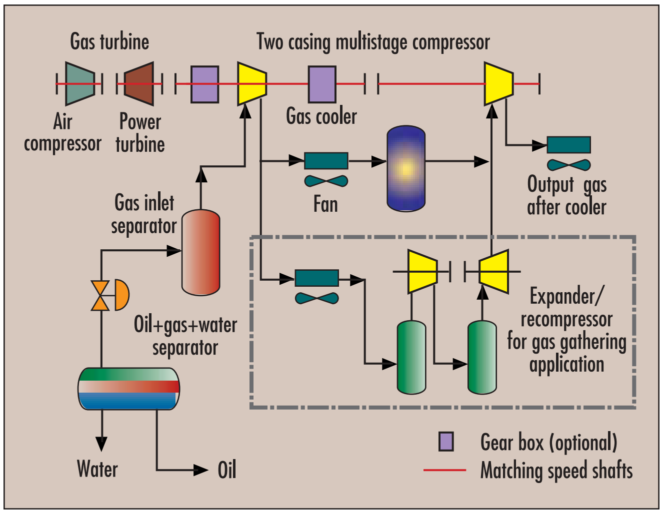

Fig. 1. Typical two-shaft gas turbine-driven compressor system.

months. If there are significant changes, restaging may be economically justifiable. The other benefit of gas analysis will be early identification of changes in the content of harmful gases such as hydrogen sulfide and carbon dioxide. Both of these gases, if not accounted for during engineering, warrant a review of construction materials used for the compressor and gas turbine hot-end section.

Startup case: Compressor startup may impose special head requirements. Startup gas from a different source is often lighter than the normal operating gas, and the machine has to operate at maximum speed to develop the required pressure. At times, availability of only low molecular weight gas during startup drives the compressor staging selection such that it has an adverse impact on machine efficiency during normal operation (Table 2), with power turbines typically operating between 75% and 85% of optimum power turbine speed. This results in power loss due to reduced power turbine efficiency. If a power turbine is operating at 70% of its optimum speed, it can lose up to 10% of available power or between 3% and 4% thermal efficiency at ISO rating conditions. Most vendors provide information about power loss that results during operation away from optimum speed.

To prevent this loss during normal operation, consider the following alternatives alone or in combination for difficult startup gas conditions:

- Increase suction pressure during startup to reduce the required differential head.
- Compressor wheels are overspeed-tested for 115% of the maximum speed, and it is possible to operate power turbines at up to 110% of their maximum speed. If required, consider operating the compressor at 105% of rated speed during startup. Usually there is overspeed protection (trip at 110% speed) to safeguard the compressor and power turbine.
- Check if it is possible to operate at a lower discharge pressure during startup.

During normal operation, the power turbine should run as close as possible to its optimum speed, and other operating conditions should not influence compressor design to the detriment of efficiency. Most vendors can easily predict actual operating speed within ±3%. Some vendors recommend normal operating speeds up to 97% of optimum since power loss due to reduced efficiency at 97% speed is negligible. Moreover, API 616 and 617 require that compressor and turbine should be capable of operating continuously at 105% of the rated speed.

Pre/intercooling: Process gas may be pre- or intercooled to reduce power consumption and cope with the occasional temperature limitation of the compressor labyrinths. Any hydrocarbon mix pre- or intercooling will reduce:

- Suction temperature
- Mass throughput for a given volume (compressor being a volume machine)
- Gas molecular weight.

The effect of compressor pre- or intercooling should be evaluated, and the desired cooling levels must be identified and provided to the operator. Less than desired cooling will:

- ▶ Increase power consumption
- ▶ Reduce operating speed, resulting in decreased power turbine efficiency.

Excessive cooling will have the following effects:

- Increase head required to achieve the required pressure
- Increase operating speed, moving the operating point nearer to the surge line
- Compressor may operate in an aerodynamically inefficient zone.

At times, users want the compressor train to be capable of operating under an upset condition of the Joule-Thomson valve. When this becomes the dominant factor in deciding downstream compressor staging, the drive turbine may operate away from its optimum speed, resulting in a large efficiency loss. Consider installing a standby expander-compressor or two expander compressors capable of 60–75% duty. This will increase compressor train availability and, consequently, the gas supply. But more importantly, it reduces the system design losses.

Controlling the compressor by suction valve throttling will have two undesired effects:

- Pressure in the separator upstream of the throttle valve increases. This will result in condensing heavies (generally C_{6+}), reducing the gas molecular weight.
- Pressure drop across the suction valve experiences the Joule-Thomson effect, resulting in condensation in the compressor inlet separator (Fig. 1).

Reduced suction pressure could mean a large energy loss. As a rule-of-thumb for a compressor train having a pressure ratio of 10 and compressing a 30 molecular weight hydrocarbon gas mixture, suction pressure loss of 1 kPa is equal to increasing the discharge pressure by 12 kPa (a factor of 12). If possible, suction pressure throttling should be avoided as a means of flow control. In some applications, the machines are discharge pressure controlled, with suction pressure dependent on power available to pull the pressure down to the extent possible.

Factors affecting gas turbine efficiency and output power. During operation, gas turbine output can be adversely affected by:

- Increased ambient temperature
- Air inlet filter and air compressor blade fouling
- Operating at part loads.

A major limitation of gas turbines is that output power varies with ambient temperature. Other factors, like inlet air filter or air compressor blade fouling, reduce power developed by the gas turbine. As output power from the turbine is reduced, the compressor operating point shifts toward the surge line, reducing compressor efficiency. Therefore, when the gas turbine output power decreases due to inherent limitations, it can compound economic losses.

Operating at part load reduces gas turbine thermal efficiency. Selecting a drive turbine for a centrifugal compressor is based, therefore, on:

- Power required by the compressor at rated conditions (rated conditions are usually the conditions seen by the compressor during normal operation) multiplied by 1.1 (adding 10% margin as specified by API 617)
- Maximum ambient temperature recorded at the installation in the past 20–100 years is taken as the worst condition. A suitable gas turbine is selected that, after derating for maximum ambient temperature, meets the API criteria.

For a site with a maximum ambient temperature of 40°C, this means that the selected gas turbine at ISO rating develops 30–40% more power than the compressor rated power requirement. With turbines often operating at ambient temperatures below 40°C, they would be operating at part load. As mentioned earlier, part load operation means higher operating costs over the entire equipment life cycle. A turbine operating at 75% load will consume about 5% more fuel on a per kilowatt basis than a turbine at full load.

For what ambient temperature should the turbine be derated? It is evident that derating the turbine for the maximum anticipated ambient temperature is not an optimum solution. The typical daily temperature pattern of many sites is given in Table 3.

Maximum ambient temperature occurs for perhaps 1–2 hr/day and more than approximately 30 days in a year. In an 8,000 hr/yr operation, a turbine has to run for about 60 hr or less than 1% of operating time at maximum ambient temperature.

To derate a turbine, consider obtaining ambient temperatures for the past five years and at four suggested times of the day. Work out the site's daily average temperature. Use this temperature for derating/uprating the turbine ISO rating, and decide on power available for the maximum period. Where contractual requirements must be met, the equipment owner must consider the benefit of sizing for lower ambient temperatures or for low demand periods.

Applying the API 617 requirement of 10% margin over rated power. Since manufacturing technology has improved, most can estimate power consumption and other centrifugal compressor operating parameters to within ±2%. Fixed power loss in turbine inlet and exhaust can be estimated with reasonable accuracy. During operation, any incremental loss due to air inlet fouling can be assessed by monitoring turbine operating parameters with torque meters.

Compressor power requirements must be matched as closely as possible to the available turbine output power. To arrive at the available turbine output power for a given site condition, derating the turbine for every known loss and applying a further 10% margin requirement is inappropriate. Excessively conservative derating will only result in higher operating cost over the life cycle of the machine.

Other design considerations include:

- The air inlet suction and exhaust ducting should be of minimum length possible. Roughly, 100 mm of water column pressure drop in suction or discharge ducting means a 1% power loss.

Table 2. Effects of operating away from optimum power turbine speed

	Normal operation	Startup operation
Gas suction pressure	10 bar	10 bar
Gas discharge pressure	100 bar	100 bar
Molecular weight	MW	0.6–0.8 × MW
Power turbine optimum speed	*A*	*A*
Compressor operating speed	0.70–0.8 × *A*	*A*
Turbine optimum output power	*B*	*B*
Power available	0.9–0.95 *B*	*B*
Total power loss	5–10%	—

Table 3. Temperature variations

No.	Time	Ambient temperature
1	5 a.m.	Minimum
2	10 a.m.	Between maximum and minimum
3	2 to 3 p.m.	Maximum
4	9 p.m.	Between maximum and minimum

- Air inlet filtration can have a significant impact on turbine output power. Cheap filters may need frequent replacement and are likely to reduce air compressor discharge pressure and efficiency, resulting in reduced power output.
- Consider using dry gas seals for large compressors above 15 MW. Oil seals can consume as much as 100 kW of compression power, and a liquid seal oil system will add to the installed cost.
- Most gas turbine-driven compressors have the main oil pump driven by the turbine shaft. Consider having both main and standby pumps driven by electric motors. This will generally increase gas turbine availability.

Since turbine and compressor technology have matured, it is possible to accurately determine design requirements. Operating conditions should be carefully evaluated to arrive at optimum sizing and most efficient operating speed of both the compressor and its drive turbine.

Proper design can result in the best return on investment, least operating cost and an environmentally responsible approach. For gas turbine-driven compressors, a conservative engineering approach of building in margins for every possible upset condition is not necessarily an economic or desirable solution. The design engineering recommendations of this article, if properly applied, could reduce operating cost and increase profit margins. ■

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