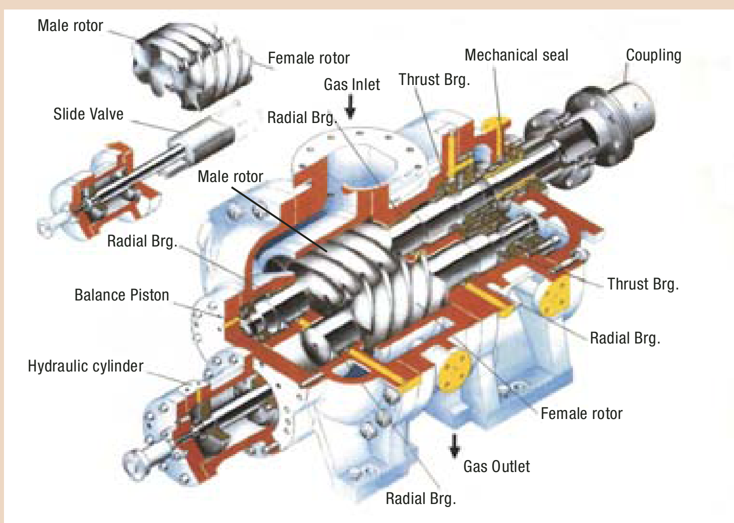
High Pressure Fuel Gas Boosting Compressors

Takao Koga, Kobelco EDTI Compressors Inc.

A fuel gas boosting compressor is the "blood pump" for gas turbine power plants because if the fuel gas compressor fails, the gas turbine power plant stops entirely.

As a result, compressors are critically important for reliable gas turbine power plant operation. At the same time, the demands on compressors have been increasing because gas turbines require higher fuel gas pressures to achieve increased performance efficiency and because pipeline pressures fluctuate due to increased overall and peak demand requirements. Therefore, selecting the correct type of fuel gas compressor is one of the most important factors in achieving successful plant operation.

Figure 1 CUTAWAY DRAWING OF OIL-INJECTED SCREW COMPRESSOR



Gas turbines require a certain pressure of fuel gas to combine the fuel gas and pressurized air before combustion. Historically, when main pipelines were built the gas pressure was adequate. Normally, the gas pressure was reduced from this high pipeline pressure and delivered at the gas turbine's required pressure. In the North American market, fuel gas compressors were not often required.

However, as higher efficiency gas turbines were developed, the required gas pressure kept increasing. The required gas pressure for conventional types of industrial gas turbines is around 250 to 300 pounds per square inch gauge (psig), or 17.5 to 21 barG on average. However, the latest generation of industrial gas turbine requires relatively high pressure gas, such as 500 to 600 psig (35 to 42 barG). In addition, aeroderivative-type gas turbines now require gas pressure as high as 700 to 1,000 psig (49 to 70 barG).

At the same time, natural gas demand has been increasing so that the new development and supply of natural gas is not sufficient for the growth of demand in many parts of the world. As a result, the gas pressure in the pipeline cannot be maintained as high as before. This is particularly true during peak demand such as summertime or daytime. It means that the gas pressure in the pipeline would gradually decrease and even fluctuate.

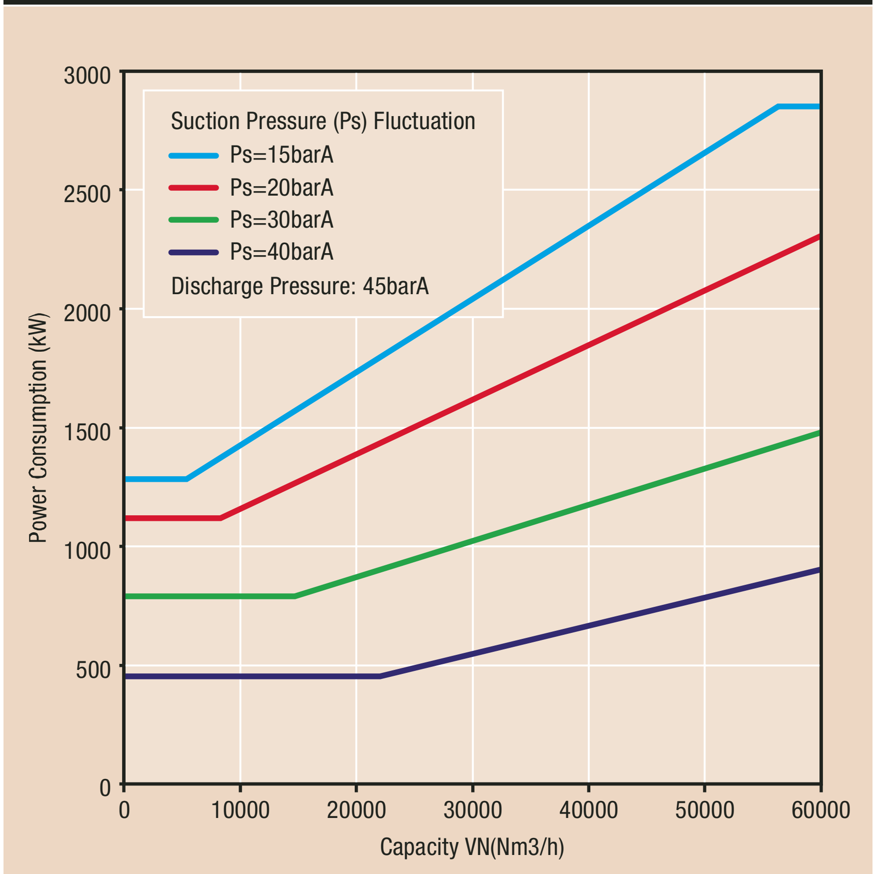
A fuel gas compressor should properly handle such fluctuating gas pressure from the pipeline while simultaneously meeting the gas turbine's required gas flow rate. Two changeable conditions are most relevant to fuel gas compressors: suction gas pressure fluctuation and the turbine's load changes.

In addition, reduced oil carryover and low pulsation/vibration are also important, because they can lead to mechanical problems. Pulsation in particular is a big concern to gas turbines with dry low NO_X combustors, since the combustor is more sensitive to pulsation. In addition, natural gas can include impurities or dirt and its precise composition can even be subject to change.

Three Designs

There are three basic gas compressor designs exist: centrifugal, reciprocating and screw. Each design has mechanically

Figure 2 SCREW COMPRESSOR PERFORMANCE CURVE



different features. To date, centrifugal and reciprocating types have been used mainly for fuel gas boosting services in the United States and Europe. Screw-type fuel gas boosting compressors have been used mainly in Japan and Asian markets. A cutaway diagram of a typical screw-type compressor is shown in Figure 1.

After development of high pressure screw gas compressors in late 1990's, this type of fuel gas boosting compressor has begun to be used for the most efficient gas turbines (for example the GE LM6000 and F7FA/FB, the MHI 501G, the Siemens/Westinghouse 501G, the Alstom GT26 and so on). Now, high pressure screw gas compressors can achieve up to 1,500 psig (equal to 100 barG), which covers any required fuel gas pressures of the most efficient and the most recently developed gas turbines.

Centrifugal compressors are suitable for large gas flow, low compression ratio and stable operational conditions; that is, basically for operations that present no change in pressure, gas flow and gas composition. As a result, this type of compressor has been used mainly for process gas services in petrochemical and oil refining fields and for natural gas boosting in pipelines. However, centrifugal compressors are equally well suited for changeable operating conditions because of its surging nature. This means that the compression ratio must remain constant and, if higher suction pressure than design pressure (minimum pressure) occur, it must be reduced to the design pressure (minimum pressure) and then boosted to the required discharge pressure to maintain the designed compression ratio. However, because pipeline pressures do fluctuate, the centrifugal compressor must furnish a suction throttle valve to regulate the suction pressure down to design pressure (minimum pressure). The drop of suction pressure is a waste of energy.

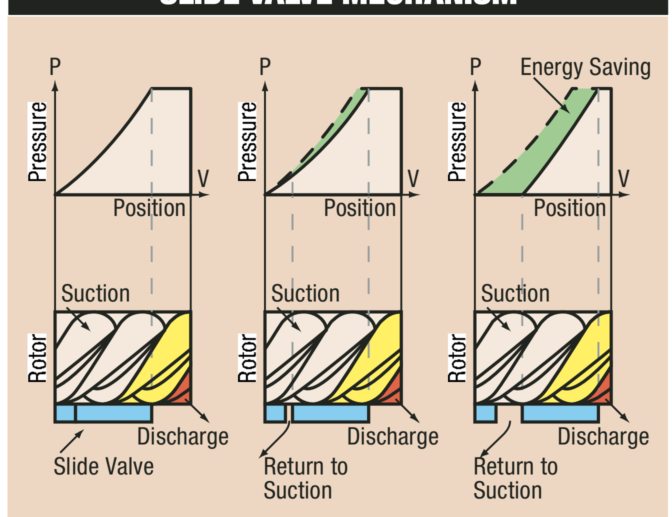
In addition, in some countries gas quality is not constant. For example, the gas composition of natural gas as it comes from the wells changes, which leads to molecular weight changes. The molecular weight change possibly could cause performance problems, especially for a centrifugal compressor due to the principle of the compression. Furthermore, although gas turbine load normally changes, centrifugal compressor turndown ranges are limited from 100 percent to about 70 percent, even with an inlet guide vane.

Therefore, on occasion, centrifugal compressors have been used for fuel gas compressors for large industrial gas turbines and low compression ratio cases; however they are not so beneficial and reliable under changeable operating conditions as those which are present in fuel gas boosting services. Also, if the required compression ratio is high, the centrifugal compressor requires multiple stages, making it a less economical investment. The same applies to reciprocating compressors.

The reciprocating compressor is suitable for high pressure and middle gas flow services such as hydrogen make-up service in oil refining processes. The advantage of the reciprocating compressor is its high efficiency due to simple displacement compression. This means the reciprocating compressor has been historically the most traditional type. It also has been used for various gas compression services in the world.

However, the disadvantages of reciprocating compressors include reduced reliability and more frequent maintenance, which can mean that a

Figure 3 BASIC PRINCIPLE OF THE SLIDE VALVE MECHANISM



spare compressor is normally required in the case of continuous operation services. It also requires frequent maintenance within a short interval (approximately once a year) because of the many sliding and wearing parts inside. This makes the reciprocating compressor less suitable for continuous and long-term operation.

In case of the gas turbine's load change, a reciprocating compressor can save more power consumption than a centrifugal compressor, but it requires non-linear step controls (for example 100 percent, 75 percent, 50 percent and 25 percent). In addition, environmental issues can arise, such as noise, emissions, pulsation, vibration and so on.

To date, reciprocating compressors have been used in fuel gas boosting compressor applications for middle-sized gas turbines or high pressure aeroderivative gas turbines; however, they are growing less popular due to the disadvantages such as reduced reliability, increased maintenance and reduced environmental friendliness. Especially for dry low NO_x combustors, the pulsation caused by the piston movement of the reciprocating compressor is now a serious problem, meaning that their use for fuel gas compressors is declining.

Screw compressors were originally suitable for middle pressure and middle gas flow rate services. Their advantages include high reliability, reduced maintenance and power savings because of the built-in slide valve. However, the original applicable range was not wide because of pressure limitations.

TABLE 1 SCREW COMPRESSOR COST SAVINGS

Fuel Gas Compressor Specifications	
Gas Turbine Output (simple cycle)	250 MW
Fuel Gas	Natural Gas
Gas Capacity (Nm3/hr)	60,000; (37,331 SCFM)
Suction Pressure (bar A)	15 ~ 40; (214 ~ 571 PSIA)
Discharge Pressure (bar A)	45; 643 PSIA)
Power Consumption at 100% Load	2,850 kW
Effect of Power Saving-Cost Saving	
Avg. power difference between screw and centrifugal	1,250 kW
Cost per kWh (estimate)	5 cents
Operating Hours per Year	8,000 hours
Cost Savings per Year	$500,000
Estimated CO_2 Reduction	
Avg. power difference between screw and centrifugal	1,250 kW
Gas Turbine Efficiency	50%
Operating Hours per Year	8,000 hours
Reduction of CO_2 per Hour	435 kg

Source: KOBELCO

peak load operation. Normally, no spare compressor is required.

Centrifugal and reciprocating compressors have limited ability to adjust their performances to accommodate the changing operational conditions of (a) varying gas supply pressures, (b) gas turbine load changes and (c) gas composition changes. Screw-type compressors can accommodate these variables, and by the very nature of their design can contribute power savings to the owner/operator, should situation (a) or (b) develop. This means that screw-type fuel gas boosting compressors can contribute to a gas turbine power plant's total efficiency.

To illustrate, consider a 250 MW large gas turbine that operates 8,000 hours a year, has operating costs equal to 5 cents a kWh, a required gas turbine pressure of 643 psia (45 barA), gas pipeline pressure fluctuations that range from 214 psia to 571 psia (15 barA to 40 barA) and an average gas pipeline pressure of 430 psia (30 barA).

Screw compressors can save power consumption by 1,250 kW/hr on average, with an operational cost savings of over $10 million for 20 years of operation (see Table 1). Should the gas turbine's load also change during this time frame, additional cost savings may be realized. This power savings means reduced CO_2 emissions as an additional benefit.

In case the gas composition changes, screw compressors do not have any problem because the compression principle is a positive displacement type. Even if the composition of natural gas would change, screw compressor performance is not affected. Also, the screw compressor has few internal parts, so less maintenance is required than with reciprocating compressors. Five to six years of continuous operation may be possible without any overhauling.

The screw compressor is also an environmentally friendly machine. For example, the noise level is low and its frequency is high so that it is easy to reduce the level to 85 dbA at 3 ft (1m) with minimal sound attenuation. Also, there is no pulsation and vibration issue because it has no piston movement, only a rotating movement.

A cutaway drawing of a typical oil-injected type screw compressor is shown in Figure 3. There are two rotors inside the casing and they contact each other at lobe surface via an oil film.

Screw compressors can save power consumption by 1,250 kW/hr on average with an operational cost savings of over $10 million for 20 years of operation.

Screw compressors have been used for small- or middle-sized gas turbines. Following development of high-pressure screw compressors in late 1990's, screw compressors are now being used for fuel gas boosting services for large industrial and high efficiency aeroderivative gas turbines as well.

Several reasons exist to demonstrate why screw compressors are chosen for fuel gas boosting services.

First, screw compressor reliability is quite high so that it is suitable for long-term continuous operation and even

Oil is supplied not only to the bearing and seal, but also to the rotor chamber directly and oil will act as lubricant, coolant and sealant in the rotor chamber. Typically, the male rotor is driven by a directly coupled 2-pole or 4-pole electric motor and it drives the female rotor. An external gear unit is typically not used since the tip speed of the oil injected screw compressor is within the proper design limits when driven at motor speed.

Since oil is injected into the rotor chamber, the seal area between the rotor lobe and bearing is no longer necessary. There is a single mechanical seal located at the drive shaft end. There are typically sleeve-type journal bearings on both ends of the rotor lobes. Thrust bearings are typically tilting pad type and are located on the outside of the journal bearings.

The oil and gas mixture is discharged through the compressor discharge nozzle into an oil separation system located downstream of the compressor. The oil that is separated in the oil separation system is circulated in the compressor lube system.

An unloaded slide valve is located in the compressor beneath the twin rotors and is used to adjust the inlet volume. The inlet volume of the compressed gas can be adjusted by moving the slide valve, which is actuated by a hydraulic cylinder.

Compressor lubricant oil is present in the process side, which means the lube oil selection is very different from other types of machines. The bulk of the oil is separated in the primary oil separator, but a secondary coalescing oil separator may be used as an additional separator. Oil separation is one of the important factors for oil injected screw compressors. Typically, a combination of demister mesh pad and coalescing elements are used. For example, 0.1ppm wt level can be achieved by combining a demister mesh pad and two stages of coalescing elements. In addition, since almost pure synthetic oil such as PAO (polyalphaolefin oil)or vacuum oil are used, there is no influence to the gas turbine by the oil.

Normally, the latest generation gas turbines accept less than 0.5ppm wt oil carryover from the fuel gas compressors so that screw compressor has no problem as far as such conservative design for oil separation system is done. Borocilicate micro fiber is a typical material used in coalescing elements and sub-micronic particles of oil can be separated from the compressed gas. Unlike reciprocating compressors, oil from the compressor experiences no deterioration by piston rubbing so oil can be re-circulated in the system as lubricant for longer life. The lube oil circulation system consists of compressor lube lines, oil cooler, oil filters and oil pump. The oil pump may be double or single configuration. The design of a single pump system may be used when the pump is required only during startup. In some cases, after the compressor starts and discharge pressure is established, oil can circulate in the system by utilizing gas differential pressure between suction and discharge.

The slide valve is used to load and unload the compressor to maintain suction pressure or discharge pressure. There is a spool valve actuated by air with solenoid valves to switch over the oil lines to pressurize the slide valve cylinders to load side or unload side. Typical control range by slide valve is from 100 percent to 15 percent and is stepless by inlet volume.

Screw compressors for fuel gas boosting services in gas turbine power plants contribute to increase the total efficiency in the power industry. Screw compressors minimize any excessive initial and maintenance costs, resulting in real cost savings for the users and emissions are reduced as well. pe

Takao Koga is general manager of sales for Kobelco EDTI Compressors Inc. and is based in Houston.

For more features on gas turbines visit www.power-eng.com