Use of High-speed Turbocompressors in Offshore Installations

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SYNOPSIS

A large number of high-speed turbocompressors are installed offshore for different applications. The compressor selections and operation methods are squeezed between the gas and/or oil field predictions and the process requirements. Well designed compressor packages allow for easy accessibility and maintainability and will result in high availability and reliability over a long period.

BACKGROUND EXPERIENCE

With the discovery and beginning of oil extraction in the North Sea in the late sixties, a number of turbocompressors for different duties in on- and off-shore applications were required. The growing market inspired further development of high-speed and high-pressure turbocompressors and their prime movers. The technological centre for the oil and gas extraction industries was previously the United States, but with gas extraction in Western Europe and oil production from the North Sea, the European turbocompressor manufacturers began to take an active part in turbomachinery developments.

The phenomenon of non-synchronous self-excited vibrations became more significant in the seventies when multi-stage turbocompressors were increasingly used in high-pressure (high-density) natural gas applications. The experience gained during this period led to research programmes, shared in part with research institutions, which were primarily initiated by European turbocompressor manufacturers and their customers.

OIL/GAS SEPARATION AND COMPRESSION

Oil wells deliver a mixture of gas, oil, water and sand, and possibly other components, to the platforms. In a first process these components are roughly separated (see Fig. 1). The separation vessels are sometimes of enormous size. On the Statfjord and Gullfax platforms, for example, they are approximately 15 m long and have a weight of 150 tons with a capacity of 70 000 litres of oil. Two further stages separate oil, water and gas from each other and reduce the pressure to approximately atmospheric. Turbocompressors are needed for the recompression of these separated gases:

1. To a level where the gases can be processed (gas gathering).
2. Back into wells (gas reinjection).
3. For increasing the oil production (gas lift).
4. For the transport of gases to onshore (gas export).

In all these cases, the gases have to be cooled in intercoolers during compression and the condensed liquid droplets are removed in separators. The flow composition, molecular weight and temperature of the gases change over short or long periods and the requirements on the rotating machinery are, therefore, quite varied.

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TYPICAL INSTALLATIONS

The South Cormorant Platform gathers oil from the Brent, Thistle and Murchison fields. Two high-speed skid-mounted compressor sets on the Cormorant platform compress the gas after the separation system from 0.55 to 21.4 bar absolute and have a running speed of 13 700 rev/min. The skids are supported at three points and include the electric motor driver, the two compressors in series and the integrated lube and seal oil systems (see Fig. 2).

Other typical examples are three skid-mounted compressor units for the Magnus Field platform. Here the three-point supported base plates contain only the compressor trains whereas the lube and seal oil systems are separately located. These compressors are used for gas gathering and gas compression to 54 bar absolute and have an operating speed of 12 000 rev/min (see Fig. 3).

A typical gas lift train is shown in Fig. 4. A 6 MW gas turbine drives the turbocompressor for recompression of the associated gas to the well pressure, in order to lift the gas and the oil to the surface. The structural base plate of this unit forms a part of the platform whereas the gas turbine and turbocompressor have their own smaller base plates (see Fig. 4). Five identical units are installed on an oil production platform in the Red Sea and are further described later.

THERMODYNAMIC PERFORMANCE

At high gas pressures, the inlet density is high and the actual volume flows are rather small and quite frequently close to the border between positive-displacement and turbo-type compressors. Small volumes require centrifugal compressors operating at high mechanical speeds.

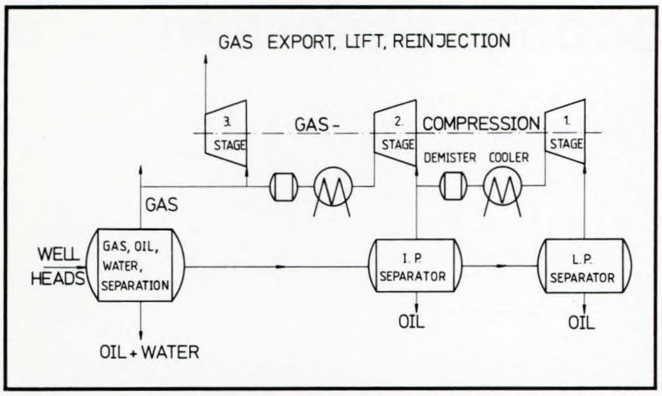

FIG. 1: Typical flow diagram of a compression system at an oil production platform

In turbocompressors, a good performance can be obtained only within a limited range of flow. The usual expression for flow is represented by the equation for the flow coefficient, ϕ

$$\phi = \frac{V}{U D^2}$$

where V is inlet flow volume (m^3/s), U is the impeller tip speed (m/s) and D is the impeller outlet diameter (m).

Figure 5 shows the important relationship between the flow coefficient and the efficiency of a multistage compressor. It can be seen that the overall efficiency of a multi-stage compressor can be improved substantially if the value of ϕ for the first impeller can be kept at about 0.08 and for the last impeller above 0.03. At high ϕ values the losses increase because of high gas velocities and at low ϕ values the efficiency drops rapidly because of the increasing influence of the unavoidable friction losses in the boundary layers on both sides of the discs, leakage losses between the impellers and the aerodynamic losses caused by narrow channels.

It rarely happens that the requirements for a gas compression application can be met by a stage operating at precisely the optimum flow coefficient. First, both the tip speed and the impeller diameter are limited by various practical constraints such as stress levels, speed of driver, size of machine, stability of rotor, etc. Secondly, in multi-stage compressors each successive stage delivers a lower flow volume so that the flow coefficient decreases through the machine.

The compression of high-density gases requires large power inputs. At these high powers the mechanical speed in revolutions per minute is often restricted from the driver side. For example, if the driver is a gas turbine, then the mechanical speed is often too low for direct coupling with high-pressure centrifugal compressors. The use of a gearbox to raise the speed is recommended if the additional mechanical losses (of the order of 2-3%) and the cost can be positively recovered with a smaller and more efficient compressor designed for optimum tip and mechanical speed. If a gearbox is used between compressor and driver then the admissible speed ratio is limited for design reasons: pitch line velocity, bending stress on the pinion, bearing speeds and loads.

Low speed means more stages, and more stages mean longer rotors. The rotor length is of influence on the rotor stability and consequently the compressor layout is squeezed between themodynamic optimum and mechanical limitations.

FLOW CONTROL AT OFF-DESIGN CONDITIONS

The characteristic curves of a radial compressor stage are limited at high flow coefficients by the choking of the stage and at low flow coefficients by the onset of surge.[1] For

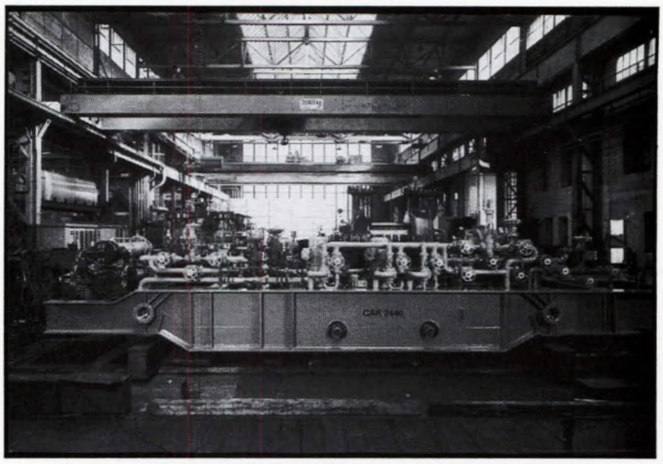

FIG. 2: Skid-mounted unit for Cormorant A platform in the North Sea. Typical three-pointed support baseplate including compressor, driver and lube/seal oil system.

FIG. 3: Skid-mounted unit for Magnus Field

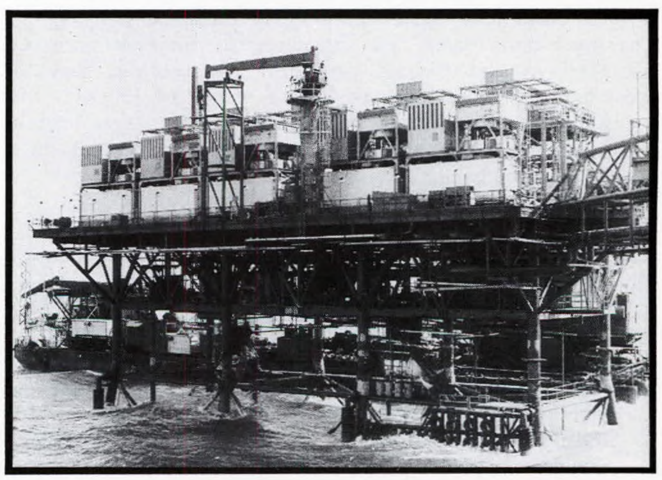

FIG. 4: Gas turbine compressor module installed at El Morgan field. Comprises five identical 6 MW Sulzer gas-turbine-driven tandem compressor trains for gas lift application.

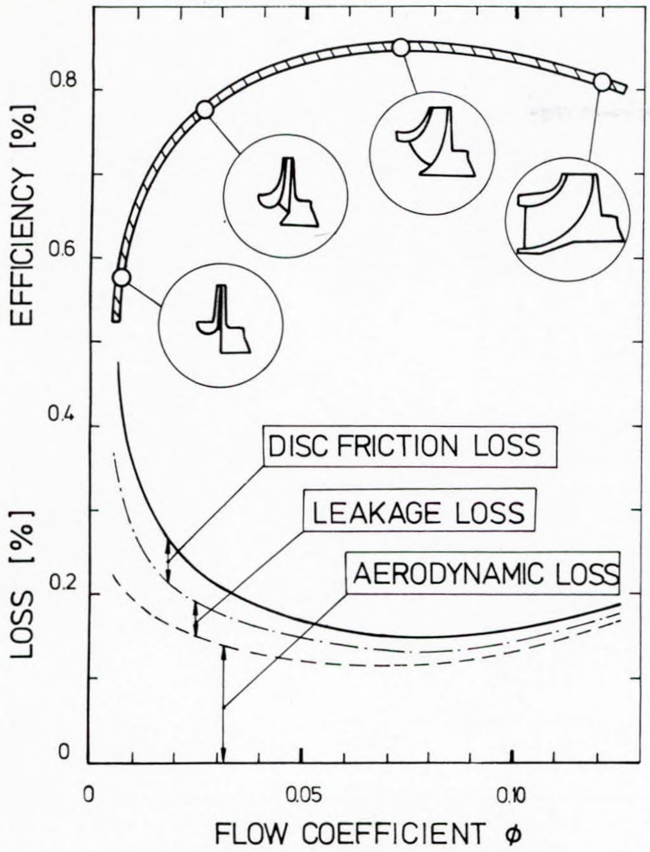

FIG. 5: Plot of flow coefficient φ against impeller efficiencies, loss and impeller shape.

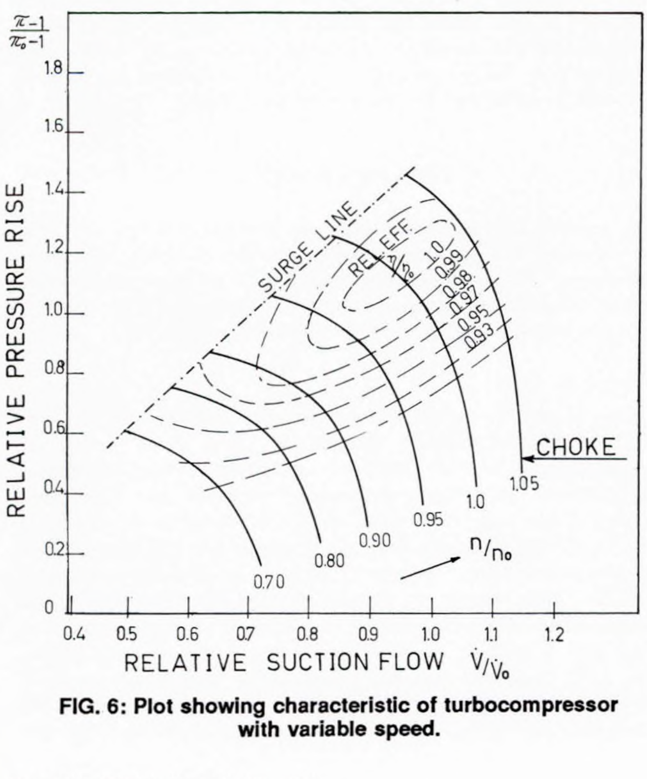

FIG. 6: Plot showing characteristic of turbocompressor with variable speed.

practical reasons a wide margin is required between choking and surge.

If the mass flow through a stage is steadily increased then the flow velocity across the whole of the flow passage will at some point reach the speed of sound. No further increase in mass flow is then possible and the stage is said to be choked. Choking can occur in the inlet passages, in the rotating impeller passages or in the diffuser vane passages and is of no danger to the compressor. If on the other hand the mass flow is steadily reduced, two different types of instability can be observed. The most serious type of unstable flow is known as surge and is characterised by violent changes in the mass flow through the stage and increased noise and vibration. The second type of instability is known as rotating stall and is characterized by small pressure fluctuations and a steady mean mass flow through the machine.

Several methods of adjusting a compressor to operate at off-design conditions are available: by-pass or blow-off valve, suction throttle, variable speed, adjustable inlet guide vanes and adjustable diffuser vanes or combinations of these. In offshore applications speed control or inlet throttling are the most common means of regulation (see Fig. 6). Inlet guide vanes are sometimes used in refrigeration compressors.

HIGH-SPEED ELEMENTS

The compressors used offshore can be divided into two different fields of applications:

1. Standard applications for which the industry has developed well known and proven equipment. The compressors are highly standardized and manufactured in large quantities on stock programmes. They are, however, usable for specific gases only and under preselected conditions of flow and head. Typical examples are air machines and compressors for refrigerant packages.

2. Engineered applications for which the industry can offer (a) proven equipment 'as built', preferably applicable to the complete unit but often to individual components only of the proposed equipment, and (b) comparable experience in design, manufacture, testing, operating and environmental conditions.

The borderline between these two broad fields of compressor applications is shifting constantly. It may often depend on the viewpoint whether a compressor project is classified as a 'standard' or as an 'engineered' application. There would, however, be convincing statistical evidence that compressors which turned out to be a 'critical' piece of equipment (short of predicted performance, excessive wear, unexpected mechanical problems) were or should have been classified under engineered applications and not under standard applications. Certain compressor requirements may even be more appropriately ranged in a third field: special compressors for special applications such as new processes or different environments.

Only a few manufacturers are today in a position to produce reliable turbocompression equipment for critical and demanding applications, be it with equipment 'as built' or with specially designed turbomachines.

Compressor rotors

It is recognized today that a fair amount of problems, known as subsynchron vibration, encountered with turbomachines built in the past were often solved without knowing the true causes. The solutions were always aimed toward rotors and increased damping. Since they cured the difficulties, the solutions must have been correct. The lack of knowledge of the real causes, however, prevented the development of more advanced or more 'critical' turbocompressor rotors.

In the recent years many investments have been made to improve rotor stability.[2,3] Investigations about labyrinth excitations as the cause of rotor instability and the theoretical approval and practical introduction of swirl brakes result in stable rotors. New experimental data for labyrinth damping will allow classical rotors to be designed with seven to eight impellers which are stable for extremely high discharge pressures and at high speeds.

The materials of construction of turbocompressor casings, diffusors, shafts and impellers are well known and the requirements of API 617 or NACE MR-01-75 can normally be observed without restrictions with current designs. The admissible stress levels can be met without difficulty for the static and dynamic parts up to the highest pressures and speeds.

The rotating elements are subject to centrifugal and aerodynamic forces. At low densities the former are pre-eminent to such an extent that the latter can be neglected. With increased densities the aerodynamic forces become important, representing a large portion of the total stress field applied to the rotor. Similarly, a change in design must be made when centrifugal impellers are used for high-density mediums. The rotor blade passage may not only induce vibrations of the diffusor blades, it may also excite the entire disc and shroud of the impeller itself.

The problems arising from pressure fluctuations caused by unsuitable numbers of diffusor and impeller blades have also been studied as well as those related to disc vibrations. The thickness of the blades and the discs of the impellers must be able to withstand the static bending and the fluctuating stresses induced by fluids of high density in order to keep the stress level within admissible limits. Blade and disc resonance conditions must also be avoided over the entire working range.

Couplings and gearboxes

The development of high-speed turbomachines is closely connected with the development of gearboxes. The most suitable gearboxes for high-speed turbomachines are equipped with a thrust collar system.

The introduction of the thrust collar for single helical gears, patented in 1935, was the basis for the solid coupling technique (see Fig. 7). It was an excellent match of a proven design advantage, developed for single helical gearboxes, with the most elementary and simple design of any coupling, substituting a variety of more complex and less reliable coupling designs, in particular toothed type couplings. It is not always realised that the solid coupling is identical to those used almost exclusively for utility type steam turbine generator units up to the largest ratings. It is not surprising, therefore, that the first applications of solid type couplings were in directly coupled steam turbine driven compressors such as blast furnace blowers. Their combination with thrust collar type gearboxes appeared almost four decades later.

As shown in Fig. 7, the meshing of single helical gears produces an axial thrust $P_{A1} = P_{A2}$ acting in opposite directions on the driving and the driven shafts. In order to balance these axial forces thrust collars on the pinion act against corresponding surfaces of the gear wheel. Collar and wheel shoulder have slightly tapered, hardened and ground surfaces. Because of the low sliding velocities Δu and full hydrodynamic lubrication, the oil film has a considerable carrying capacity with low losses and virtually no wear. As a matter of fact, the grinding marks on the surfaces of the collar and wheel remain visible after 50 000 to 100 000 hours of operation.

In gear boxes with thrust collars the pinion is axially linked to the low-speed gear wheel, the position of the latter being fixed by a large low-speed thrust bearing. A considerable axial thrust from the high-speed compressor shaft can thus be transmitted through the pinion to the low-speed shaft and its thrust bearing.

This thrust collar design enables one to take up the thrust of the turbomachines in line with the pinion and these are then coupled solidly together. The solid coupling is combined

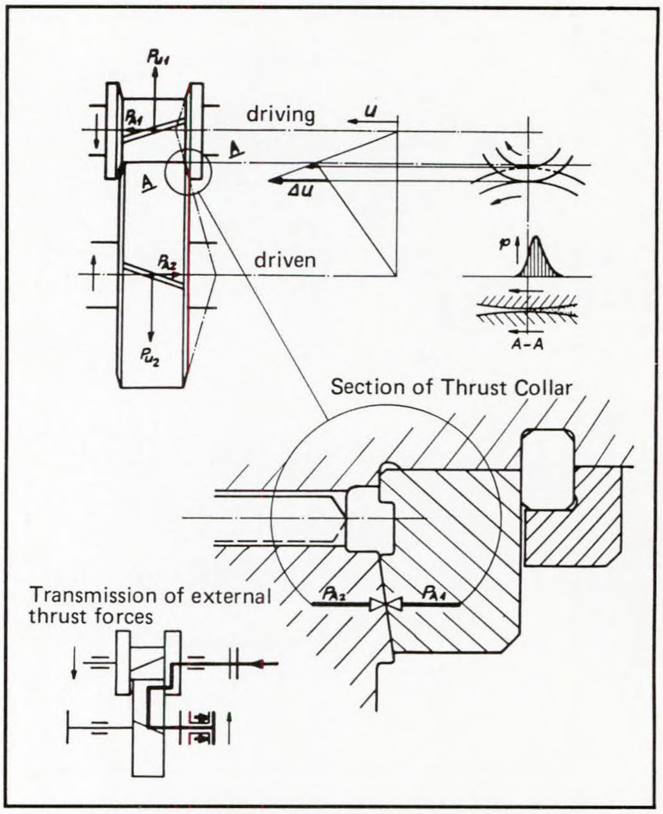

FIG. 7: Single helical gear with thrust collar and schematic arrangement of compressor thrust transmission

with an intermediate flexible shaft portion (quill shaft) with end-flanges bolted tightly to the corresponding flanges of the rotors.

In this manner, the residual thrust of a high-speed shaft combination is taken up by the thrust collars and transmitted to the low speed thrust bearing of high and non-restricted carrying capacity. High-speed thrust bearings and high-speed tooth couplings can be avoided, which improves the reliability of the plant. An added advantage is the resultant lower mechanical losses (of the order of 0.5 up to 1.0%). The corresponding energy savings are also considerable, besides the reduced capital cost for the smaller oil system.

PACKAGING

A compressor unit must meet its intended design not only for volume and pressure but also for a host of other conditions. This is especially true under offshore and severe weather conditions where a number of design criteria have to be met to create a well engineered single-lift module.

Base plate

One of the key elements of the package unit is the base plate. Besides the cases where the base plate forms an integrated part of the platform construction, the compressor and/or driver underbase are usually arranged for three-point support. Since this base plate must also contain as much as possible of the auxiliary systems required for the operation of the compressor unit, the optimum form is usually a box-type assembly from beams or square profils. In many cases the compressor is manufactured at a different location from the driver and a split of the base plate is therefore advantageous. After both machines are fully assembled and tested in their respective manufacturers' works, the module can then be assembled at any other location and bolted together with tension grip bolts. A complete unit test and/or a string test can thus be performed easily at the site or module yard.

The base plate also has to absorb piping forces and moments, distortion of the module structure caused by emplacement offshore and must isolate vibrations from and to the platform construction. The latter is mostly done by isolating supports. In the design phase the base plate itself is calculated using the finite element method, and with computer simulation programmes the base plate structure can be optimized (see Fig. 8)

On-skid equipment

A large diversity of auxiliary equipment has to be arranged within the confines of the base plate: piping, valves, impulse lines, cables, instrument boards and in many cases the lube and/or seal oil system, drain traps and degaser tanks. For maintenance and operation service it is of utmost importance that a proper lay-out gives adequate access to the skid-mounted equipment. An example of a well engineered unit with an integrated seal oil system but without the driver is shown in Fig. 9. A further example of a more compact unit is shown in Fig. 8.

Compressor lube and seal oil systems for offshore application mostly follow the API 614 'Lube and Seal Oil System' recommendations. Space and weight limitations in refineries are not a problem. However, if these specifications are also required for offshore installations, a number of recommendations are questionable. Many non-API installations and even some API units are running with this simplified equipment: one single oil cooler instead of twin coolers, shaft driven lube and/or seal oil pump instead of motor driven main pump avoiding large overhead run down tanks and oil tank capacities with five times retention time instead of eight times, thereby saving weight and space.

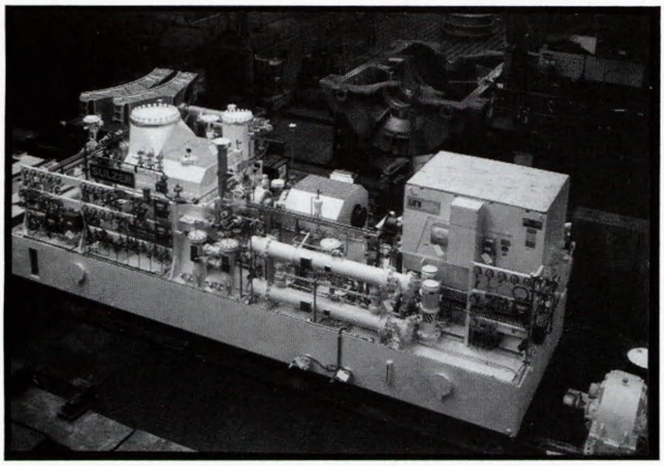

FIG. 8: Box-type base plate, refrigeration compressor unit for Brae B platform during works assembly.

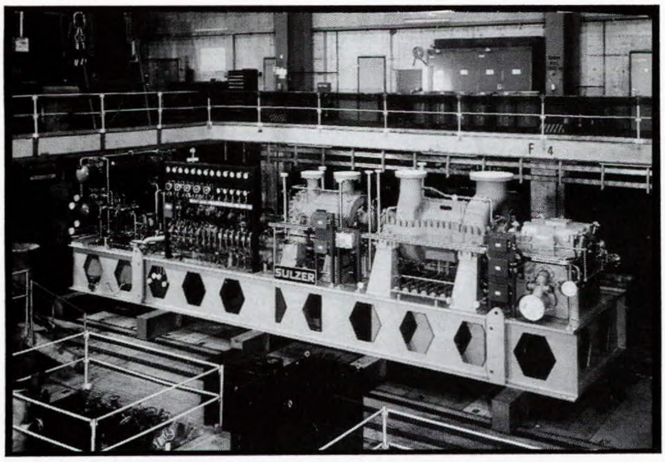

FIG. 9: Skid-mounted compressor set with low-pressure and high-pressure compressor, gearbox and integrated seal oil system. This is a compact but accessible lightweight skid.

OPERATING EXPERIENCE

Operating experience for each compressor unit is different. Mostly based on former experience with compressor units, the engineer or user specifies certain auxiliary systems based on that experience. Also the space requirement, arrangement and selection of adequate components and materials are of importance for comparing operating experience.

In normal cases compressor units are operated by the user and the manufacturer has little influence on their operation and maintenance. My company has a number of operation and maintenance contracts which show that very high availability figures can be reached if operation and maintenance are done properly. For example, the oil production platform El Morgan in the Red Sea is equipped with five gas turbines, each 6 MW, driving via a gearbox two compressors arranged in tandem.[4]

In 1979 the company entered into a five years operating and maintenance contract covering all five units. The equipment was, therefore, not only maintained but also operated by the manufacturer's personnel. A unique feature of this operation and maintenance contract was that the average availability of the modules had to be guaranteed. The following items were included in the contract: job and site management, onshore camp facilities for the operating personnel, onshore workshop for minor repair and spare parts store, offshore service shop and storage area, all spare and replacement parts, all operating and maintenance personnel, catering services and local transport for contract personnel, and training of customer personnel.

After about six years of industrial operation the experience can be summarized as:

1. In spite of two technical problems in the first period of operation, the gas requirements of the customer were fully met at all times.

2. The arrangements made in the planning phase of this job with regard to lay-out of the unit and personnel requirements at site proved to be adequate.

3. The personnel selection and the extensive training sequences throughout the manufacturing stages, work-tests, site erection and commissioning proved to be a major benefit to the performance of the plant.

In order to define the operating experience it is important to know the plant extent (in this case five complete gas turbine/turbocompressor trains including all auxiliary equipment) and definitions of the Logbook figures:

(a) installed hours
(b) operating hours
(c) scheduled outage hours
(d) unscheduled outage hours
(e) stand-by hours

$$\text{Utilisation} = \frac{\text{operating hours}}{\text{installed hours}} = \frac{b}{a}$$

$$\text{Availability} = \frac{\text{available hours}}{\text{installed hours}} = \frac{b + e}{a}$$

$$\text{Reliability} = \frac{\text{operating hours}}{\text{operating + unscheduled outage hours}} = \frac{b}{b + d}$$

The operating hours, utilitsation, availability and reliability figures for the described El Morgan unit for the period October 1979 - July 1986 are shown in Table I. In a

Table I: Reliability Data for operating period October 1979 - July 1986 of five Sulzer gas turbine driven compressor modules for gas lift duty on El Morgan platform

	Module					
	1	2	3	4	5	Total
(a) Installed (hours)	59971	59 830	59779	59821	59700	299101
(b) Operating (hours)	54128	54268	55604	55134	54419	273553
(c) Scheduled outage (hours)	2591	1368	1467	1269	2550	9245
(d) Unscheduled outage (hours)	273	931	188	490	97	1980
(e) Stand-by (hours)	2979	3263	2519	2928	2634	14323
% Utilisation b/a	90.3	90.7	93.0	92.2	91.2	91.5
% Availability (b+e)/a	95.2	96.2	97.2	97.1	95.6	96.3
% Reliability b/(b+d)	99.5	98.3	99.7	99.1	99.8	99.3

six year period reliability figures of more than 99% were reached.

The advantages of such a contract is more than just the high reliability. The equipment supplier receives comprehensive and competent feedback of field experience, which is of great importance for early detection and remedy of technical problems, and also increases his know-how in such fields of operations, and the customer improves the availability of his turbomachinery because of the professional supervision of operation and maintenance provided by the equipment supplier.

CONCLUSIONS

Industrial turbocompressor design is a most active field of applied mechanical engineering. The development of turbocompressors is never at a standstill. The progess and insight gained during the last ten years is, in retrospect, impressive. Looking forward it is the necessary foundation for the majority of today's compressor projects. There are good reasons to assume that the designs in another ten years will be different from those of today, but in what respects can only be guessed. Very likely speeds, efficiency, power and density will again move upwards. The methods of design and manufacture will rely increasingly on automation and non-destructive quality control methods will be more efficient and more easily reproducible.

ACKNOWLEDGEMENTS

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Discussion

J. CLIFFE (Offshore Consultant): I would question Mr Howe's contention that the success of areo-derivative gas turbines offshore has been due to their light weight, as such. In fact the weight of an overall skid package differs little from that of the so-called heavyweight gas turbine, the weight of either type being in the range of 150-200 tons. Space is often also greater on areo-devivatives due to the use of a plenum chamber for the intake.

I would suggest that the aero-derivative success is primarily due to their ease of maintenance and complete gas generator removal, as opposed to the in situ strip downs on heavy duty gas turbines. The tendency is now for the two types to merge with the development of lightweight industrial machines with, for example, electron beam welded compressor rotors, eg the Sulzer type 10.

No mention is made in the paper of high-velocity bag intake filters which are now being widely adopted in the North Sea. What is Rolls-Royce's experience and views on this type. The three-stage marine filter has virtually been abandoned for new construction since it has not proved satisfactory against dry salt particles and solid pollutants, cement dust etc.

Mr Howe's suggestion for high-frequency generators using single-shaft versions of aero-derivative turbines is unlikely to gain ready acceptance in the offshore industry, where only proven equipment is required. The stress levels reached on compact high-speed rotating electrical elements are likely to be a problem.

Regarding Mr Beckers interesting paper, I do not fully understand his point about avoiding the use of speed increaser gears on the grounds of mechanical losses. Also pitch line velocity limitations are rarely reached on offshore applications. With gas turbine mechanical speeds typically around 6000 rev/min for high powers, there is almost always an advantage in using a higher-speed compressor and gear drive. Mr Beckers is in any case later advocates the use of thrust collar single helical gears for balancing compressor thrust.

Where power losses and high oil flow and capacity are a problem, double helical gears have some advantages since they are more efficient with only one low-speed thrust bearing and lower oil consumption. The oil flow on a single helical gear usually exceeds that of the entire compressor turbine driveline, and considerable economies in tank capacity are possible with double helicals.

Simplification of auxiliary equipment is highly desirable and it is worth pointing out that gas turbines almost always have a shaft-driven oil pump, to API 617 as opposed to the API 614 requirement for seperately driven oil pumps, based mainly on refinery practice. I would agree there is a strong case for non-API systems in offshore applications.

Can Mr Beckers say if Sulzer has any experience with dry running gas seals eliminating the seal oil system.

E. D. COOPER (British Petroleum): It is interesting to note the comparison between aero-derivative and the so called 'heavyweight' types of gas turbine. The reference to high availablity/reliability as a significant factor in the success of the areo-derivative is in need of some elaboration. Recent assessments I have made indicate little difference over the complete maintenance cycle, and the main factor possibly relates to the quick turnaround on a gas generator replacement. However, I would like to see some detailed analysis of the causes of unscheduled outages to establish the relative importance of gas generator replacement. There is the distinct likelihood that problems requiring gas generator replacement are in fact less significant than failures elsewhere in the package.

It is somewhat misleading to refer for power generation to a large contingency reserve with 2 x 100% compared with 4 x 33%. The four smaller units undoubtedly in many (but not all) cases offer improved overall hydrocarbon production rates, by their improved statistical probability of always providing some power. The two large machines give a higher probability of more frequent complete process shutdowns, with resultant operational difficulties, production losses, and delays which may take days to restore to full production rate. However, a large number of smaller machines may well be more expensive for both machinery and topsides space and weight. As usual you pay your money and take your choice. Recent studies have indicated there is some merit in going as far as 3 x 33% where non-critical loads, such as water injection, are both very large and capable of automatic shedding on a gas turbine trip.

The use of high-frequency alternators to obtain great reductions in size and weight sounds laudable but I question the hidden development costs on the distribution side. There is perhaps more incentive to displace gas turbines by variable speed motors for large compressor drive and go for larger central power generation.

S. OHARA (Toyo Engineering Corporation): Mr Howe mentioned that in order to avoid misalignment which could result in vibration or bearing problems, the three-point mounting method can be applied to minimize the influence on the machinery caused by wave effect and so on.

Can this idea also be also applied for more severe wave conditions which would be expected on floating production platforms such as tanker-based floating production (FPSO) systems or is special consideration necessary to counter the more severe wave motions likely when rotating machinery is installed on FPSO systems?

Authors' replies

B. N. Howe

To answer Mr Cliffe, I would maintain that the term 'lightweight', as used to describe an aero-derivative gas turbine, is fundamentally a reference to the 'thermal block', or gas generator, which is, at 1.5-3.0 t, significantly lighter than a 'heavyweight' thermal block of comparable output. This would generally hold true even if one took the term 'thermal block' to mean the complete driver unit, including the power turbine, on its baseplate.

I must take issue with Mr. Cliffe's point that the aero-derivative takes up more platform space. The length dimension from gas generator air intake to power turbine drive shaft is generally shorter than for the comparable 'heavyweight' and, where space is important, the lighter end of the set, including the inlet filtration package and plenum chamber, can be cantilevered out over the sea using the single basepoint concept. This can often account for the front 40% of the set length.

High-velocity bag filters have indeed been widely adopted in the North Sea, where there has been a considerable amount of retrofit of such systems by operators onto Rolls-Royce gas turbine installations. Many thousands of hours of satisfactory operation of these systems have now been achieved with the overall result that filter lives have been considerably extended. For new installations, of course, the added benefits are lower frontal area, lower weight and lower cost.

My comment on high-frequency generators was designed to provoke a reaction from the oil companies who, through the recent period of low activity due to the oil price level and instability, have been seeking ways of making platforms smaller and cheaper. Of course, there are problems but I do not believe these are insurmountable. It will require oil company leadership for the topside equipment manufacturers to go down such a route, with its resultant impact on development spend.

In reply to Mr Cooper, it is certainly true that the great proportion of set outages are caused by 'rest of set' equipment, and most particularly the controls. Overall, therefore, one would expect the reliability factors of heavyweight and aero-derivatives to be comparable. The difference in availability factors, however, can be significant and is almost entirely due to the rapid change-out capability of the aero-derivative.

Comparative information is difficult to come by, but perhaps the most comprehensive study was carried out by the Environmental Protection Reseach Institute (EPRI) in the USA, which in summary showed aero-derivatives to be on average 4.5% more reliable and 5% more available than their heavyweight competitors.

It was not my intention to mislead in discussing redundancy philosophy, but merely to report past history and predict a change in the future. I agree with Mr Cooper's remaining points on this subject.

Certainly the development costs associated with the high-frequency generation concept will be significant. Topsides equipment manufacturers are unlikely to take the plunge without strong leadership from their customers, the oil companies. I put it forward as a means of assisting the oil companies in their efforts to reduce the costs of hydrocarbon production and thereby perhaps make marginal fields more economically viable.

To answer Mr Ohara, in principle the three-point mounting system concept would be suitable for FPSO systems, but obviously the specific of each particular installation (wave effects, ship displacement, deck stiffness etc.) would have to be investigated before the most appropriate design could be selected.

Rolls-Royce's marine propulsion gas turbine packages are generally bolted to the ship machinery deck at more than three points and through shock absorbers designed to allow the rotating machinery to remain operational when subjected to shock loadings equivalent to static loadings as high as 40g.

FPSO systems do not need to be designed to such limits. The three-point mounting system as a means of ensuring machinery alignment is, therefore, just as valid a solution for FPSO systems as it is for the large offshore production platforms which stand firmly on the sea bed.

J. Beckers

As Mr Cliffe correctly states the tendency is now to merge industrial and aero-derivative gas turbines. The Sulzer type 10 is a heavy-duty industrial gas turbine which, however, makes selective use of advanced technologies, some of which were originally developed for aircraft applications. Such selections were only made where improvements could be gained without compromising the traditional advantages of the industrial unit, ie long life, reliability etc.

The main design objectives were a highly efficient, compact and reliable engine. Low weight was not a primary design objective as can be seen by the use of normal cast iron and steels for outer casings and blade carriers. A substantial weight reduction was, however, achieved by reducing the engine size.

The maintenance concept is also important in as much as that while the unit can be maintained completely on site, modular changeout or complete gas generator removal is possible.

The gas turbine speeds range from 3600 to 7700 rev/min for high powers. In most injection and high-pressure applications a speed increasing gearbox between 10 000 and 14 000 rev/min is selected in order to have a highly efficient compressor.

For large-flow and low-pressure applications a gearbox is also justified if the compressor efficiency increases by at least the gearbox losses of the order of maginude of 2-3%. For offshore installations a smaller compressor frame size and a gearbox may also be selected for reasons of space or weight.

Single or double helical gears may be used in modest power and speed ranges but for high powers combined with high pitch line velocities there is a preference to use single helical gearboxes. In general the double helical gearboxes have slightly better efficiency and a smaller oil comsumption but in the higher speed and power ranges the manufacturing of the continuous tooth of the single helical gear is easier to control and possible axial vibration can be avoided. The advantages of both gear types are integrated in the single helical gear with thrust collar as described in my paper.

There is certainly a trend to use more and more dry gas seals. Sulzer's US manufacturer STI in Latham has supplied six units with this type of seals. The operating experience is so far satisfactory.

In reply to Mr Ohara, the three-point support system for turbocompressors normally used in offshore installations can also be used for floating production platforms. The fixing of the baseplate to the platform and possible vibrations produced by other equipment on the platform has to be taken into account when designing the turbocompressor group. If this is done we do not expect any movement problems resulting from severe wave motions.