PRACTICAL DESIGN SOLUTIONS FOR MECHANICAL DRIVE STEAM TURBINES

by

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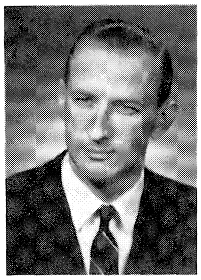

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ABSTRACT

Following a review of possible alternative solutions regarding the choice of a mechanical driver; i.e., electric motor, gas turbine or steam turbine, it is demonstrated that the steam turbine as applied in industry is ideally suited for integration in many processes and often provides an optimum heat/power balance. Although it involves a higher initial capital investment, the overall result is an economic optimum. The steam turbine is adaptable within a wide range as regards the specific requirements set by each application. It has a favorable output/speed characteristic and is an independent source of power, unaffected — given proper design — by unusual operating conditions.

Emphasizing the need for highly reliable machines, a design concept is presented, founded on a sound philosophy regarding mechanical design, safety and reliability. The system is an intelligent application of the modular design concept, supported by extensive research, testing and advanced computing technology. A description of requirements for applications in various widely differing industries demonstrates the suitability of this design concept. Attention is drawn to considerations leading to the actual choice of a turbine, such as efficiency, impulse or reaction design, and others. In view of this, an unbiased review is given of vital elements such as the rotor and its blading and the underlying design principles.

Aspects of blade erosion, axial thrust and blade fixing methods, including the control stage, follow. In the design stage of last stage blading, particular attention is required regarding flow patterns, strength and resonance characteristics, but it is demonstrated that wide range variable speed units with last stage blades larger than 500 mm (20") can be built reliably. A special section is devoted to the elimination of dynamic stresses by suitable blade design and methods such as damping or lacing wire.

Mechanical drivers can be coupled to the driven machines either by flexible or solid couplings, the latter being the only sensible solution for units of medium and large outputs, regardless of the number of cylinders. The paper concludes with some design aspects on rotor dynamics and shaft stability considerations.

INTRODUCTION

In many plants in the process industry, the petrochemical and other industries for the bulk production of chemicals, the compression of gases or transport of liquids is an essential part of the economically optimum process. For these purposes various types of compressors, blowers and pumps are employed. A special and frequent application of the latter are boiler feed pumps and condensate extraction pumps in power stations.

The above machines may be driven by:

Steam trubines

Gas turbines

Electric motors

CRITERIA FOR THE CHOICE OF A DRIVER

The choice of an optimum solution for the mechanical driver constitutes an important decision as to the economy and operating reliability of the plant. A brief comparison of the three possible solutions indicated before shows:

Steam Turbines

— For each application, a practically optimum design solution may be found.

— The speed of the turbine can, in nearly all cases, be adapted to the driven machine, which enables a direct drive.

— A high efficiency may be obtained over a large operating range.

— There is no limitation of power output.

— An optimum steam-power balance utilizing process steam is an economically optimum application of a mechanical drive steam turbine.

— On the other hand, the application of a steam turbine means a higher capital investment.

Gas Turbines

— Generally speaking, the installation of a gas turbine is easier and less capital is tied up.

— A gas turbine is much less adaptable to the requirements of the driven machine and has a lower efficiency than a steam turbine.

— A gas turbine cannot be integrated in a process steam cycle (unless a waste heat boiler suits the process).

Electric Motors

In principle the electric motor is the simplest solution for a mechanical drive. However, there are a few disadvantages:

— Often a gear between motor and driven machine is unavoidable.

TABLE 1.

BLOWERS AND COMPRESSORS				Pumps
Process	Chemicals	Petro-chemicals	LNG	
Iron smelting Air separation plants Evaporation plants Compressed air Gas works	Ammonia and methanol Nitric acid Uria synthesis Soda	Mineral oil Parafin Ethylene/Propylene Butylene/Butadiene	Various processes	Various industries Power stations

— The overall economy, particularly in the case of variable speed operation, is lower compared with a steam turbine.

— Integration in a process steam cycle is not possible.

— The reliability of the plant becomes dependent upon the continuity of the electric grid.

A general guide line for the application of drivers in the process industry seems to have evolved:

— There is a preference for the electric motor drive for small units in uncomplicated plants without process steam requirements.

— In all other cases, due to its obvious advantages, the mechanical steam turbine drive is preferred.

As far as pumps are concerned, the process industry in most cases seems to employ electric motors. In power stations with unit outputs of 200 - 300 MW and more, the boiler feed pump is almost exclusively driven by a steam turbine.

The advantages of the steam turbine indicated above are supported by many years of operational experience. The following various design aspects are discussed in detail, and it is demonstrated that, based on this experience and know-how, large units with variable speed and high initial steam conditions, specifically, may be employed without affecting plant reliability.

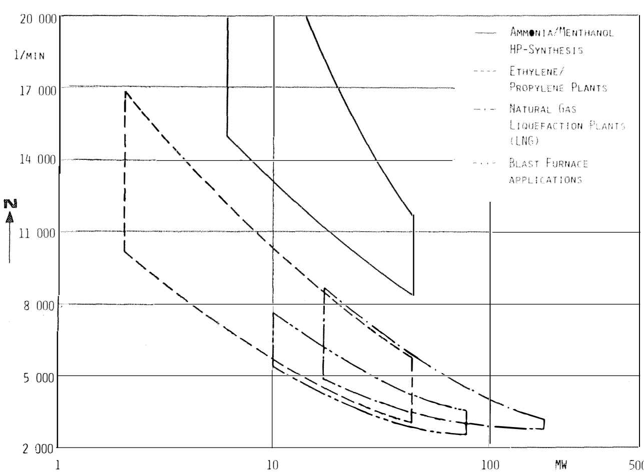

Figure 1. Areas of Application of Various Compressors.

N = Speed

P = Output

RANGE OF APPLICATION AND REQUIREMENTS

The field of application of steam turbines for driving compressors and pumps is extremely large, and consequently, the characteristics of the various cases differ widely. A summary of the main fields of application is given in Table 1. Examples of various applications, such as in the iron and steel industry and ammonia synthesis plants, are described.

Compressors and Blowers

The main characteristic of a blower driver; i.e., the relation speed/output (n/P) varies with the compressor designs and depends on the type of process and size of plant. Also, the initial steam data of the steam turbine depend mainly on the type of process.

Figure 1 shows the areas of application of the four main types of industrial processes:

— The highest speeds by far are required for machines for the ammonia and high pressure methanol synthesis plants. The characteristic for these processes is the fact that the initial steam data are given by the process, as the steam for the steam turbine is generated in the primary and secondary converter of the plant. The machinery of such a plant is shown in Figure 2. The speed of the turbines involved here is, due to the application of a gear, lower than the speed range indicated in Figure 1.

— In addition, the applications for the drive of blast furnace blowers, ethylene compressors and compressor units employed in the liquifaction of natural gas (LNG) are shown. The latter are currently the largest mechanical drivers in the world — understandably at the lower end of the speed scale. To simplify Figure 1, many other applications have been omitted.

Pumps

Pump drives which, due to the recent increase in unit sizes in the utility industry, have grown to outputs of 50,000 KW and more, as shown in Figure 3, operate at rather modest speeds.

Requirements for Design and Operation

Once the ratio n/P has been selected, other parameters such as initial steam data, efficiency, etc., are required for a particular application, as shown in Table 2. Also requirements as to the mode of operation for a particular application is needed. The initial steam conditions, as referred to previously, may be given either by the process steam data, or may be determined by optimizing the steam cycle which is usually the case, e.g., for blast furnace blower applications. Efficiency requirements are usually based on considerations resulting from the way of operation, the cost of primary energy, etc.

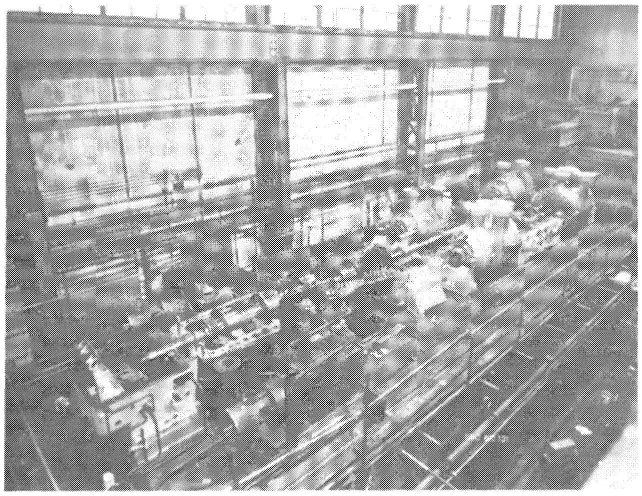

Figure 2. BASF Plant (Federal Republic of Germany). In operation since 1970.

Output at coupling	:	19 500	kW
Live steam pressure	:	98	bar
Live steam temperature	:	520	°C
Vacuum	:	0,127	bar
Speed	:	8 000	rpm
Extraction pressure	:	32,8	bar

The operational requirements are extremely important, since these are the basis for the reliability of the unit throughout its life. Some of the most important items are:

— speed variation and range,
— start-up under load,
— operation of condensing turbines against atmospheric back pressure (e.g., for blast furnace blower drivers),
— backward rotation (e.g., in the case of LNG compressor drivers).

PRACTICAL DESIGN APPROACH AND EXAMPLES

Design Approach

In order to meet the various requirements, suitable design solutions are necessary for mechanical steam turbine drives. In this regard, it is of vital importance to have a basically sound and uniform design concept:

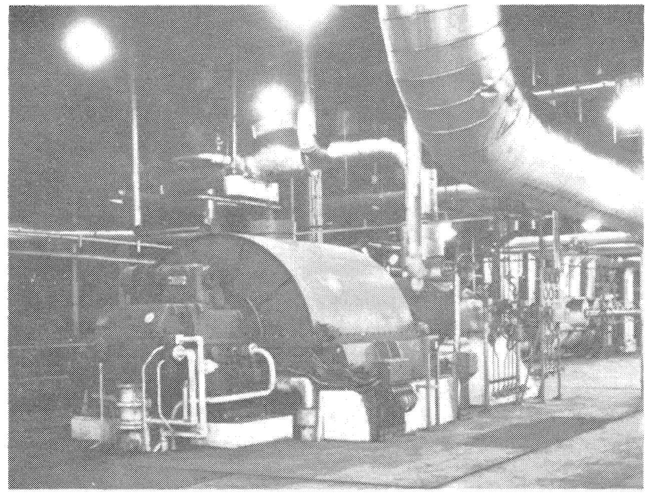

Figure 3. View of the World's Largest Feed Pump Turbine in Amos Power Station of the American Electrical Power Company (AEP). In operation since 1973.

Output at coupling	:	50 000	kW
Live steam pressure	:	7,5	bar
Live steam temperature	:	308	°C
Vacuum	:	0,1016	bar
Speed	:	4 025	rpm

— To establish and adhere to well-founded principles of mechanical design, a philosophy for safety and reliability, etc. Consequently, a uniform design approach for all steam turbines.
— Application of all modern aids for layout and design; i.e., the latest computer programs, up-to-date knowledge of material properties, etc.
— Research and testing in various areas of mechanical design and thermodynamics, in addition to, or to verify computer results, or to supplement calculations in areas where those can be insufficiently relied on or are impractical.

Some detail questions will be referred to later. Basically it can be said, however, that thermodynamic and design principles, supported by tests and research on the broadest possible basis, are a sound means to obtain machines complying with the requirements for mechanical steam drives for all applications. This systematic approach also allows the latest experience and know-how to be passed on continually to the end user.

2.

Plant	Initial Steam Data	Efficiency	Design Specification
Ammonia/Methanol	. . . 140 bar/540 °C	High	Large extraction flows at approx. 40 bar extraction pressure
Ethylene/Propylene	25 bar/300 °C . . . 100 bar/500 °C	Medium	
Iron smelting	30 bar/300 °C . . . 87 bar/510 °C	High	Optimum point usually required at part load
LNG	. . . 87 bar/510 °C	Medium-high	With or without feed heating

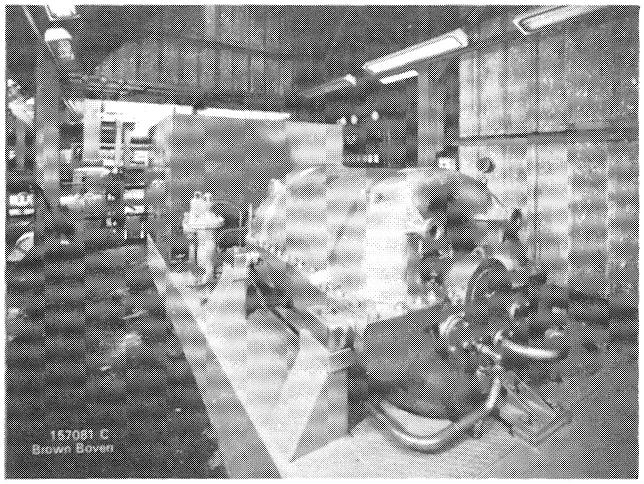

Figure 4. A Compressor Driven by a Back-Pressure Turbine Operating in a Chemical Work (Grangemouth, Great Britain). In operation since 1969.

Output at coupling	:	2 500	kW
Live steam pressure	:	39,0	bar
Live steam temperature	:	370	°C
Back-pressure	:	5,2	bar
Speed	:	7 600	rpm

Examples

As shown in Table 1, there are several areas of application for compressor and pump drivers, for which mainly back pressure, condensing and extraction condensing machines are employed, as shown in Figure 4. Some main areas of application are described in the following.

Iron and steel industry (blast furnace blowers)

Taking the steam turbine drivers of blast furnace blowers (Figure 5) as an example, it can be demonstrated how some design basics, although unchanged in principle over the years, have been improved in accordance with the advancing technology. In this manner it became possible, without any problems, to build the machines of output required today — the maximum output of the drivers for furnace blowers climbed from 20,000 KW to 50,000-70,000 KW over the years. In Figure 6, a machine built in 1962 is shown. It is clearly visible that the cylinder consists of two cast halves, each with integral

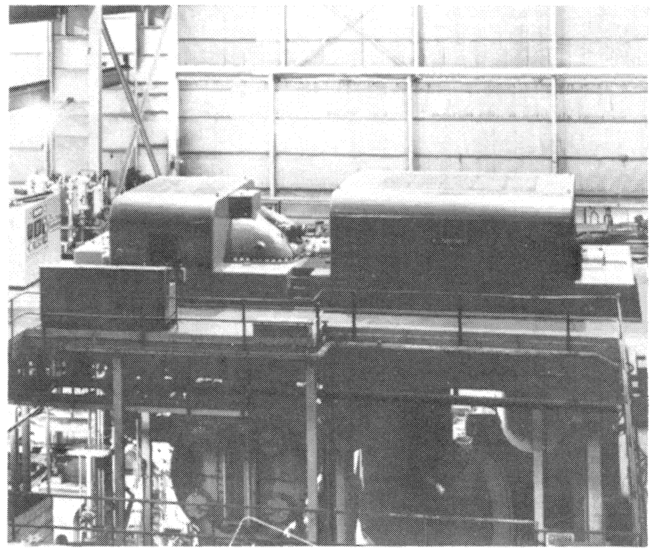

Figure 5. Blast Furnace Blower Driven by a Steam Turbine in a Steel Work (Australian Iron and Steel Pty. Ltd., Port Kembla, Australia). In operation since 1972.

Output at coupling	:	29 500	kW
Live steam pressure	:	43	bar
Live steam temperature	:	454	°C
Vacuum	:	0,1	bar
Speed	:	3 500	rpm

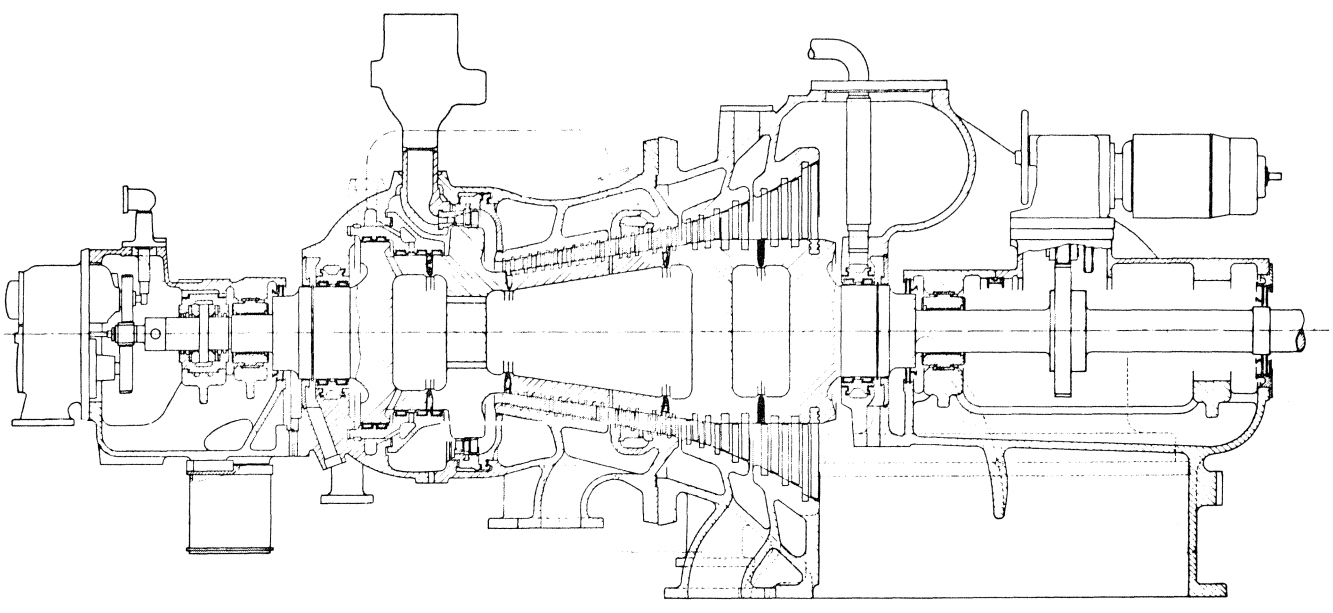

Figure 6. Cross Section Through a Mechanical Drive Turbine of 15 500 kW in a Steel Works, ISCOR, South Africa. In operation since 1964.

Output at coupling	:	15 500	kW
Live steam pressure	:	31,5	bar
Live steam temperature	:	371	°C
Vacuum	:	0,08	bar
Speed	:	4 200	rpm

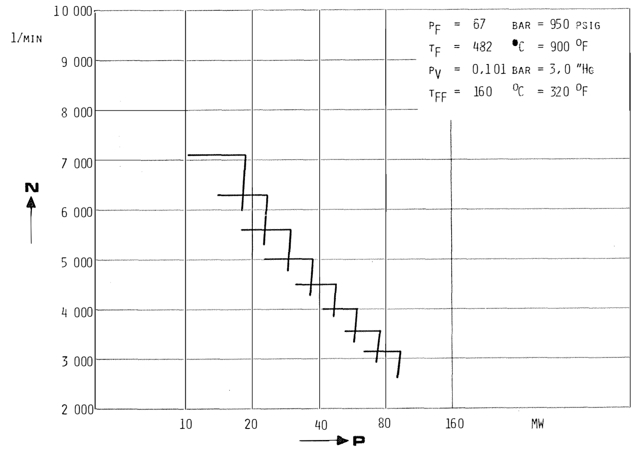

Figure 7. Maximum Coupling Output of Condensing Turbines to Drive Blast Furnace Blowers. Valid for the given layout data.

p_F = Live steam pressure
t_F = Live steam temperature
p_v = Vacuum
t_{FF} = Final feed temperature
N = Speed
P = Output

stationary blade carriers. Nowadays, as a rule, only the outer shell of the cylinder consists of two cast halves, and the blade carriers are manufactured separately to give an optimum design both thermodynamically and from the point of view of operational reliability — the latter in particular as a result of their thermal elasticity and individual supports.

Due to the wide range of the speed/output characteristic of mechanical steam turbine drivers, a number of frame sizes depending on speed are required. Since the last stage blades of condensing steam turbines used in this area are expensive in terms of design and manufacture, they have been standardized. Using laws of analogy, the amount of research work and design for the development of such blades is reduced and the experience of one blade can be transferred to the next.

Figure 7 shows the areas of steam turbine drivers for blast furnace blowers, and a cross-section of such a machine is shown in Figure 8. It may be noted that the blading consists of a 'Curtis' control stage and a number of reaction stages. Due to the required efficiency characteristic, this design is often employed in order to obtain an optimum efficiency in the operating points most frequently run, and at the same time have an adequate output margin to operate at full load.

A particular but common requirement for the design of a blast furnace blower driver is the possibility of operating against atmospheric back pressure when vacuum is lost. Under these conditions it may be observed:

— The exhaust steam temperature rises to a value of 130-160° C.
— This is not critical for the blading, since the heat drop across the last stages is appreciably smaller than in normal operation. The permissible blade stresses under the higher temperature are only fractionally lower.
— Whether the rotor bearing support is influenced by this temperature depends on the design in this region.
— Experience gained with this mode of operation has been very satisfactory with all units, in none of which any difficulty occurred.

Ammonia/Methanol synthesis

Characteristically, turbines for these applications, as shown in Figure 9, have:

— high speeds
— high extraction pressures
— large extraction flows

Consequently, machines like this have a high power/weight ratio. Due to the design conditions, usually a two-cylinder design is employed. The high pressure cylinder is an extraction back pressure turbine and the low pressure cylinder has either a single or a double flow exhaust. A generally valid diagram showing areas of application cannot be given due to the great variation in extraction conditions. It is primarily the LP section which is crucial in meeting the required speed demand. Figure 10 shows the maximum massflows of single and double flow LP exhaust bladings.

These turbines nowadays can be built employing pre-engineered modular components. The high pressure cylinder consists of a normal extraction back pressure turbine and the low pressure cylinder is of double exhaust flow design. Standard exhaust casings are used for small turbines whereas for large machines welded exhaust casings, such as used for boiler feed pump drivers, are employed, as can be seen in Figure 11. Due to the unique character of these turbines and the particular importance of the blading for this application, a special section is devoted to this subject.

Liquefaction of natural gas

LNG plants have been of increasing importance and interest since the end of the 1960's, and some plants have been

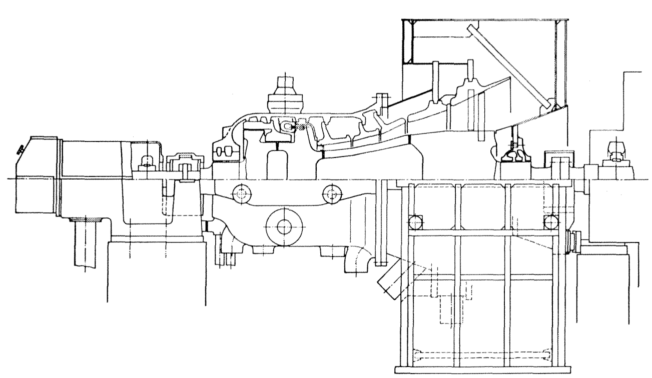

Figure 8. Typical Design of a Modern Blast Furnace Blower Driver for a Power Output Range of 30,000 — 100,000 kW.

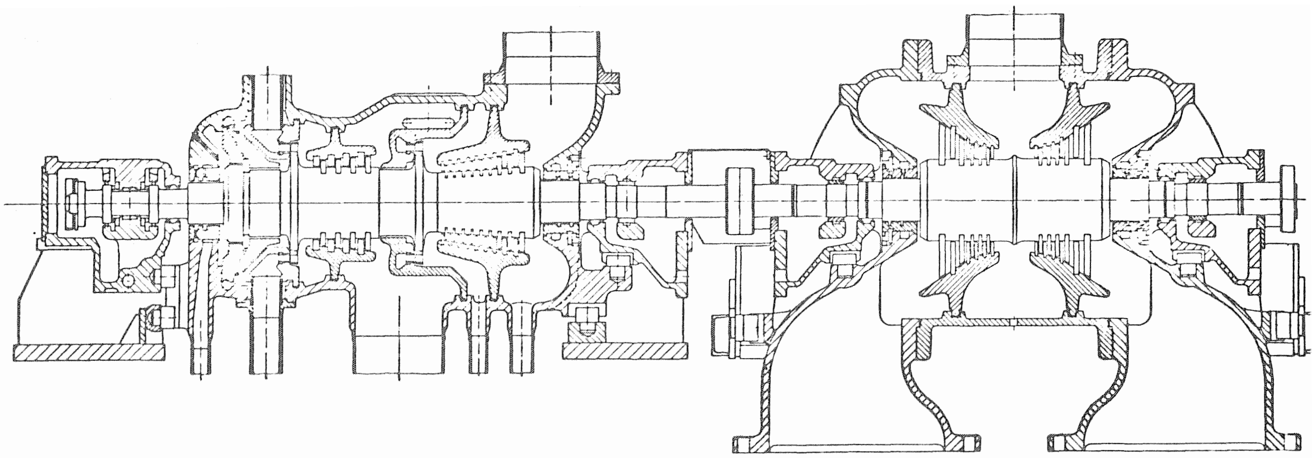

Figure 9. Cross Section Through a Two-Cylinder Mechanical Drive Turbine for an Ammonia Synthesis Plant.

Output at coupling	:	22 000	kW	Live steam temperature	:	520	°C	Speed	:	14 200	rpm
Live steam pressure	:	108	bar	Vacuum	:	0,21	bar	Extraction pressure	:	42,2	bar

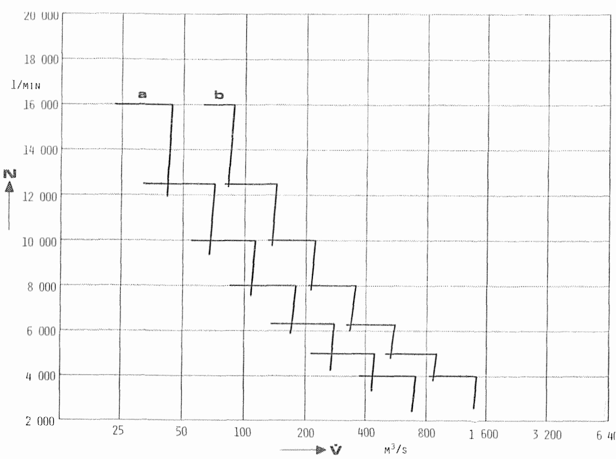

Figure 10. Maximum Volumetric Exhaust Flow for Single Flow (a) and Double Flow (b) High Speed Condensing Turbine Last Stage Blading.

N = Speed
V = Volumetric flow

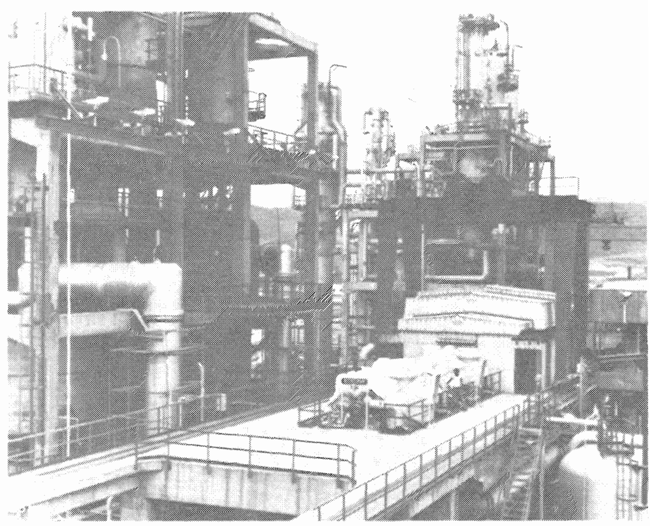

Figure 12. View of One of the Compressor Groups in the SKIKDA LNG Plant, Algeria. In operation since 1972.

Output at coupling	:	80 000	kW
Live steam pressure	:	67	bar
Live steam temperature	:	480	°C
Back-pressure	:	69,4	bar
Speed	:	3 490	rpm

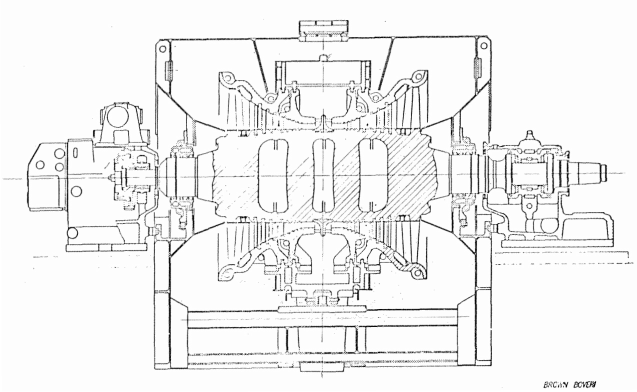

Figure 11. Typical Cross Section Through a Double Flow Feed Pump Turbine Drive with an Output Range of 10,000-60,000 kW.

in continuous operation for some years (see Figure 12). The size and number of the compressor sets involved depends on the process chosen for the liquefaction of the natural gas. Employing very large axial compressors and mechanical drivers which were at that time available, liquefaction plants like those in Skikda/Algeria use only a few machines with high unit outputs. Figure 13 indicates the area of application of steam turbine drivers up to the highest output for future projects being studied now.

Figure 14 shows the steam turbine for the fourth unit of the Skikda plant during workshop assembly. Some problems in the initial phase of operation seemed to support the opinion of

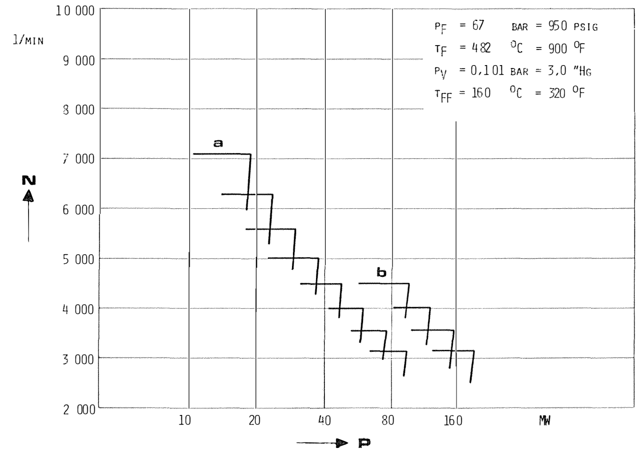

Figure 13. Maximum Coupling Output for Single Flow (a) and Double Flow (b) Condensing Turbines for LNG Compressor Drive. Valid for the given layout data.

p_F = Live steam pressure
t_F = Live steam temperature
p_v = Vacuum
t_{FF} = Final feed temperature
N = Speed
P = Output

the critics of compressor plants of such size; however, the very successful subsequent operation of these machines proves that they may be regarded to be of proven design (1).

For most users of turbine compressor sets, the unit size of the Skikda machines in particular is unusual. However, for a manufacturer engaged in the production of utility steam turbines, such machines are part of his standard range and well within his scope of experience. Such a manufacturer can refer to proven design elements, such as the welded turbine rotor. Some interesting points of this particular design feature are:

— The individual pieces are relatively small and can be easily forged, heat treated and tested, enabling a close check on the areas with the highest stresses.
— Due to the random arrangement of the individual sections, a symmetrical metallurgical structure is impossible and subsequent thermal instability cannot occur. A heater box run is not necessary.
— A lower thermal mass enables shorter start-up times and faster load changes.
— Testing of the individual rotor sections is easily conducted.
— A forging reject is more quickly replaced than an entire rotor.
— Machining of the remaining sections can continue unhindered.

The welding of steam turbine rotors was first employed in 1930 and over the years has been developed to a very high degree of perfection.

Boiler feed pumps

From medium output upwards, boiler feed pumps in power stations are driven almost exclusively by steam turbines. The main reason for this is the improvement in the overall plant efficiency due to the adaptability of the steam turbine to output and speed as compared with an electric motor. Over a

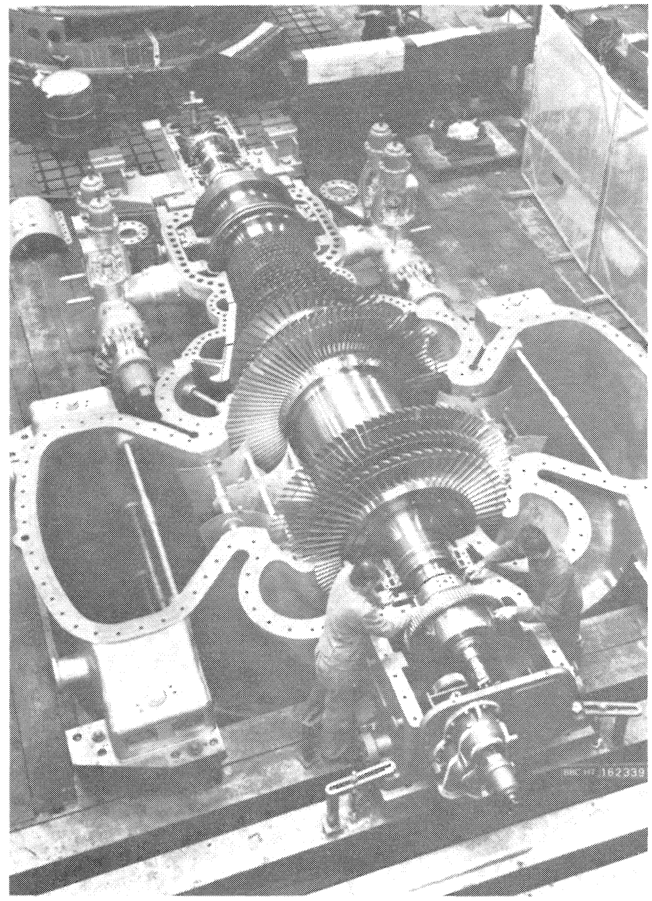

Figure 14. Shop Erection of the World's Largest Compressor Drive Turbine at Present, Data see Figure 12.

large range of operations, these turbines are operated with steam extracted from the generating unit, at a relatively low pressure. Depending on size and exhaust pressure, single or double flow exhaust turbines may be used (see Figure 11). More extensive information may be obtained from (2).

POINTS OF SPECIAL INTEREST

Choice of Turbine

Since their invention, steam turbines have been built according to either the impulse or the reaction principle. For historical reasons, the idea became common that the impulse turbine would be more suitable as a mechancial driver (3). On the other hand, however, the largest mechanical drivers for pumps (2) and compressors (4) built today employ reaction blading. The main difference between the two types may be grouped in two sections:

— thermodynamics
— mechanical design.

For the reliability of the turbine the mechanical design differences are of particular importance. The main differences between the two types are listed in Table 3.

Efficiency

It is common knowledge that a great variety of factors affect efficiency. For a comparison between an impulse and a reaction type turbine, it is useful to introduce a dimensionless

TABLE 3.

Characteristic	Impulse Turbine	Reaction Turbine
Efficiency	Better at small volume flows, poorer at medium and high volume flows	Better at medium and higher volume flows
Rotor	Shaft-and-disc design	Drum type
Blading	Few stages, wide axial space required	More stages, little axial space required
	Stationary blades mounted in diaphragms	Stationary blades mounted in blade carrier or in casing
	Moving blades on discs of rotor	Moving blades on drum

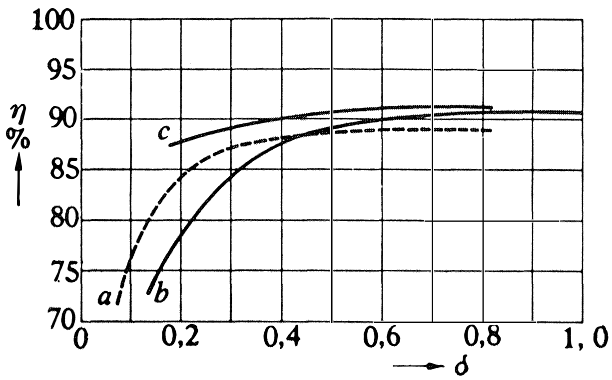

Figure 15. Comparison of the Efficiencies η of Impulse and Reaction Blading.

a = Impulse blading
b = Reaction blading with tip sealing
c = Reaction blading with shrouding
δ = Volume coefficient = $\dot{V}/r^2 \cdot u$
$\dot{V}$ = Volumetric flow
r = Radius (centre of blade passage)
u = Speed (formed with r)

characteristic (see Figure 15). The comparison listed in Table 3 is in line with the efficiency curves for impulse blading (curve a) and the reaction blading with tip sealing (curve b). However, Figure 15 also clearly shows that an appreciable improvement in efficiency of reaction turbines, particularly at low mass flows, can be obtained by shrouds (curve c). For the overall efficiency of a reaction type turbine, the losses of the balance piston must be taken into account. Summing up, it can be said that at low mass flows the impulse turbine has its advantages, but at medium and high outputs the reaction type turbine unmistakably has its merits.

Design

a) Rotor. For troublefree operation of a steam turbine, design and behavior of the rotor is of supreme importance. An attempt is made to compare the characteristics of both the impulse and reaction turbine in the following.

Impulse turbines:
— The shaft-and-discs design usually permits machining from a solid forging.
— The discs must be considered individually from the point of view of vibrations. Disc vibrations may add to rotor vibrations during operation and involve also the blading, leading to a complex vibration behavior of the system.
— Material faults may cause the entire rotor to be rejected and replaced.
— Rapid temperature changes; e.g., during start up, more readily lead to shaft bending due to the relatively small diameter of the impulse turbine rotor.
— The main mass of the shaft is in the center of the rotor. As the blades are fixed at the rim of relatively thin discs, such a rotor is more sensitive to a uniform distribution of blades than a drum type rotor of a reaction turbine.

Reaction type turbine:
— This type of machine has a drum type rotor either machined out of a solid forging or built up of sections.
— The disadvantages of a disc type rotor indicated above do not exist.
— When a drum type rotor machine from a solid forging is employed, the situation is the same as for a disc and shaft type rotor regarding rejects due to material faults.

b) *Blading.* Apart from the rotor the most marked differences between the impulse and reaction designs are in the blading. Generally speaking, the reaction turbine employs approximately 75 to 85 per cent more stages for a given heat drop than the impulse turbine. The length of the cylinder, however, is for both types roughly the same, since the axial space required for a single reaction stage is appreciably smaller than that for an impulse stage where the diaphragm and the disc rim require their share of the length of the machine.

The fact that an impulse turbine has fewer stages often leads to the mistaken view that for stress reasons or for operational safety the reaction stage can not absorb as high a loading per stage and therefore requires more stages. This is not the case at all. The number of stages is determined by thermodynamics.

Blade stresses

The dynamic loading of blades is determined by means of the following equation (5):

in which
S = stimulus
V = resonance rise
H = vibration form factor
α = bonding factor
σ_b = static bending stress due to steam flow forces

Based on the well-known fact that resonance due to excitation by the preceding nozzles (nozzle excitation) at the funda-

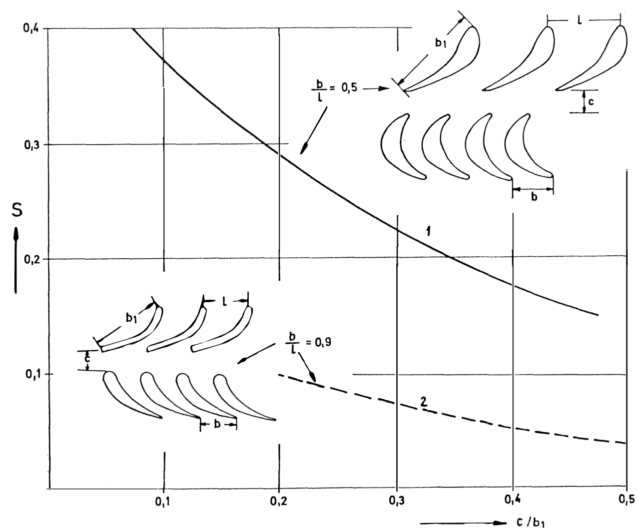

Figure 16. Stimulus S Related to Corresponding Axial Gap C.
1 = Impulse
2 = Reaction

mental bending frequency of the blades must be avoided, a comparison of the vibration behavior of impulse and reaction blades can be summarized as follows:

— The degree of excitation of a vibration is best illustrated by the stimulus S. Earlier (6) and later (7, 8, 9) studies reveal values of S = 0.25 for impulse and S = 0.1 for reaction blading.

Taking into account that the stimulus decreases with increasing axial distance (Figure 16), this factor is higher for impulse blading than for reaction blading. Given equal dynamic loading; i.e., equal safety factor, the reaction blading allows a higher specific loading.

— A considerable advantage of reaction blading is the fact that a resonance between the fundamental vibration and excitation by the preceding nozzles is practically impossible (3), whereas, on the other hand, the impulse blading must be designed with allowance for this influence of the axial distance between nozzle and blade.

Summing up, from the point of view of dynamic loading, the reaction blading clearly has its advantages.

Blade failures

The previously mentioned fact, that the blades of an impulse turbine are subject to greater vibration excitation than the blades of a reaction turbine seems to be born out by statistics concerning blade failures (10). According to statistics with impulse turbines one failure occurred after 1 million hours of operation, whereas with reaction turbines a failure would only occur after 4.8 million operating hours of the same element. Additionally, taking into account the number of elements employed (number of stages) it appears that for each impulse type turbine as an average one blade failure may occur after 25,000 operating hours, and for each reaction type turbine one such failure after 56,000 hours of operation.

Erosion behavior

With condensing turbines it is generally accepted that attention must be paid to the behavior of the blades in the region of increasing wetness. Under unfavorable conditions of water content, peripheral velocity and geometry, the water droplets that form can erode the moving blades; i.e., material is worn away from the leading edge. The measures to prevent erosion damage are known (11).

The opinion is often expressed that reaction turbines are more susceptible to erosion. It should be noted that, with the application of advanced technology, as far as the final stages of condensing turbines are concerned, the impulse and reaction designs have become very similar (12). In stages where there is a danger of erosion occurring, the degree of reaction increases along the blade. Modern last stages operate on the inside more or less as impulse stages with slight or moderate reaction, but very definitely as reaction stages towards the tip.

There is therefore no reason to prefer one type of blade over another from the point of view of erosion behavior. In this respect, earlier ideas must be corrected.

Axial thrust

Under normal circumstances, the axial thrust in an impulse turbine is small, as none or little of the pressure drop takes place in the moving blades. Should the pressure distribution in the turbine change, for example, due to mineral deposits on the blades, (an occurrence which is not rare in industrial turbines), considerable thrust forces can arise. This constitutes more of a danger for the impulse than the reaction machines because the increased pressure in front of the moving blade row acts on the whole disc area.

Thrust variations present no problems for reaction turbines when each cylinder is provided with its own balance piston — which is always the case with single-cylinder turbines. The balance piston is usually arranged with a number of stepped diameters so that the abnormal thrust conditions, caused either by the deposits mentioned before or by changes in pass-out or extraction flow, have no effect on the reliability of the turbine.

Blade design

The blades of mechanical drive turbines are often subject to adverse operating conditions and a correspondingly robust design is called for.

Control stage

The reliability of the control stage blading is assured by observing the following rules:

— low stress levels
— high standards of production of the blade fixation
— joining the blades together

Three of the possible blade fixing methods are briefly compared and the corresponding operating experience given. It is important to know that dimensioning of the control stage must be carried out for 2 load points. The highest bending stress and therefore, according to equation (1), the largest possible dynamic loading occurs at the first valve point. The highest operating temperature occurs at full load and it is at this point that blade design is made for static stress due to the centrifugal force. By employing a sufficiently robust blade, the bending stress can be kept as low as is considered necessary; on the other hand however, a very definite limit exists in the second case as a result of the blade geometry.

The inverted T root is eminently suitable for low steam conditions and peripheral velocities. The fir-tree root can absorb a considerably higher loading compared with the inverted T root since the load carrying areas relate in the ratio of 1.6:1.

For even higher initial steam conditions welding of the blades to the rotor is an excellent solution, see Figure 17. The

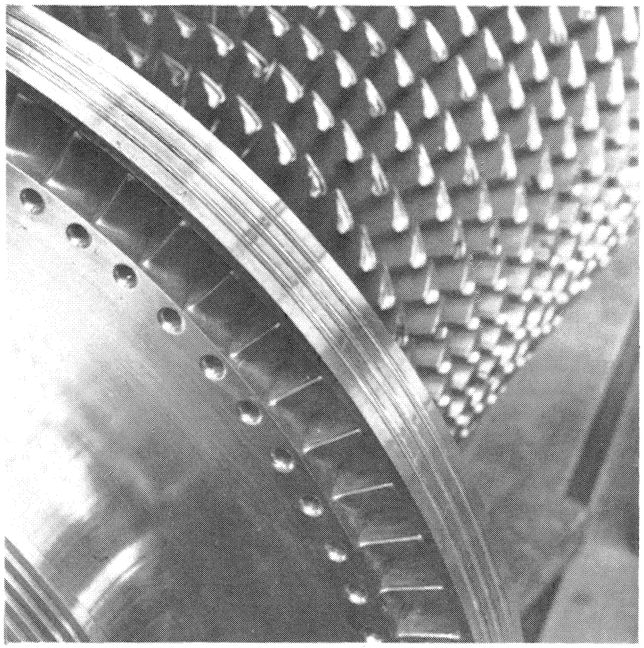

Figure 17. Welded Blades of a Control Stage.

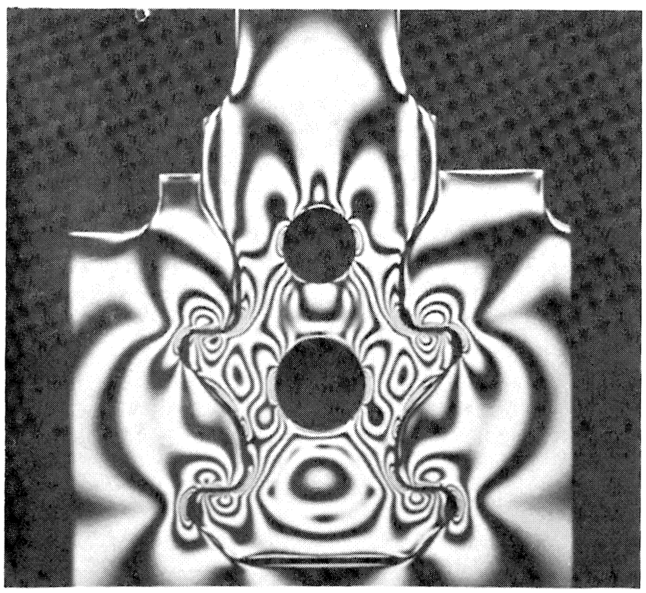

Figure 18. Optical Stress Test on a Reaction Blade.

manufacturing procedure has been well known for over 40 years and this method of fixing has performed well with high steam conditions and units of the largest outputs.

Reaction stages

Geometrically similar standard profiles are used for the reaction stages which can be made either with or without shrouding. Highly stressed blade fixings are designed with a two or four shoulder integral root. The calculation of such elements is backed up by means of two test methods; the shape of the shoulders can be optimally arranged using the optical stress test shown in Figure 18 and the rupturing stress can be checked using the rupture test shown in Figure 19.

Exact knowledge of the blade fixing behavior, particularly at high temperatures, makes it possible to use the available materials as economically as possible.

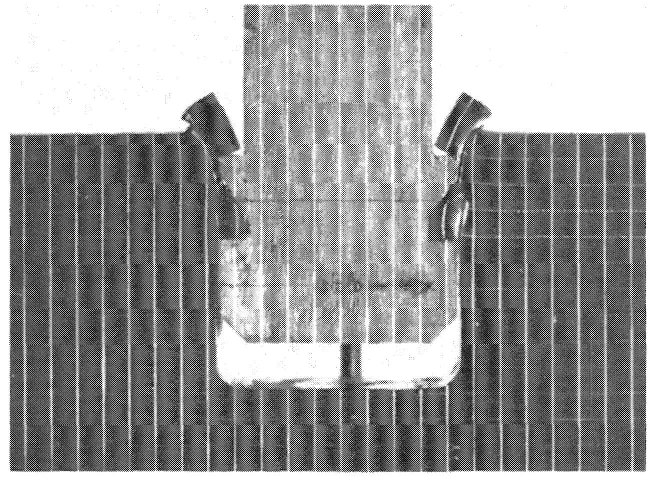

Figure 19. Rupture test on a Blade Root.

Last stage blading

The last stage blades of condensing turbines receive particular attention. Extreme care must be taken to design these blades from the point of view of flow patterns, strength and vibration considerations.

Space does not allow for full details here — suffice it to mention that nowadays low pressure blading is fully standardized and always manufactured to identical designs.

Based on the operating experience gained with compressor drive turbines and know-how resulting from the manufacture of long final stages for utility turbines, it is possible, without entering into any risks, to build final stages for variable speed mechanical drives of over 500 mm in length. However, in this respect the following point should be taken into account.

Equation (1) shows that dynamic stresses that might occur are proportional to the static bending stress σ_b. Special considerations show that these static stresses must not exceed certain limits. On the other hand, due to ever increasing exhaust pressures, continually larger exhaust mass flows for given axial annulus areas are encountered. It is quite possible today to design last stage blading to meet these conditions. Generally, they are characterized by considerably stronger profiles than blades previously used for lower exhaust pressures.

Blade vibration control

The partial admission control stage has already been referred to. By tying the blades together at the shrouds they can be designed in such a way that dynamic stresses are entirely eliminated.

A careful vibration analysis is carried out on the reaction stages. Whether and to what extent dynamic stresses occur depends on the magnitude of the factors contained in equation (1):

— The stimulus S has already been discussed and can be established according to (3).

— The resonance rise V depends on the resonance frequency of the blades.

— According to (5) the vibration form factor H is approximately 0.89 at the fundamental frequency and 0.08 at the first harmonic frequency; i.e., the dynamic stresses due to resonance at the first harmonic frequency is only one tenth of that due to the fundamental resonance frequency.

— The bonding factor α equals 1 for unconnected blades.

It may be seen from these values that it is particularly important to avoid resonance at the fundamental frequency. This condition, however, practically does not exist with reaction blading (3).

The remaining question is that of the low speed frequency harmonics of the last stages of a turbine. Theoretically it would be possible to design all stages of a constant speed turbine to operate completely without resonance. At variable speed this is naturally not possible and a method must be found to suppress vibration. As is apparent from the influence of the individual factors, this is only possible by means of factor α; i.e., the blades have to be connected with each other. This is practically always possible by using damping wires that cancel out excitation forces on the blades of a stage. The damping wires can either be inserted loosely or brazed to the blades. However, it is important that the wire is correctly dimensioned.

Specifications which require that second harmonic resonances or resonances up to eight times the normal speed resonance must be taken into consideration are, according to the theory discussed above, superfluous and are not supported by practical experience with reaction turbines.

Vibration measurement

Some specifications for compressor drive turbines require measurement of the harmonic frequency values of the moving blades on the rotor before shipment of the turbine.

This point calls for the following comments:

— No direct conclusions can be made as to the expected reliability of the blading as a consequence of measurements on the stationary rotor. This would require knowledge of the harmonic frequency values at operating speed.
— These relations may only be understood by means of calculation and testing. Once this is achieved, the blades can be accurately and correctly dimensioned in the design stage.

COMPRESSOR-TURBINE ASSEMBLY

The way of assembly of the compressor with its driver, in particular the arrangement and the type of coupling used, is undoubtedly important for the reliability of the group.

Sometimes the opinion is expressed that the simplest solution is an arrangement with the least possible mutual influence; i.e., to use some sort of flexible coupling. In practice, however, the issue is more complex and the correct design should be determined from case to case. A comparison of the main characteristics of the gear coupling and the solid friction coupling are shown in Table 4.

The conclusion which can be drawn from the known facts is that the gear coupling is the sensible solution to the problem for smaller units with low to medium speeds. In fact, the advantages of the gear coupling often tend to be overrated and there are many indications of somewhat problematic operation (13).

The amount of positive operating experience with the solid coupling for medium and large units guarantees that they can be employed without reservation. The assembly arrangement of both machines, often consisting of several cylinders, generally presents few problems.

The vital criteria for the choice of the assembly arrangement are:

— Number of cylinders.
— The amount of differential expansion between the rotating and stationary parts.
— The ability, if any, of the compressor to take up the turbine differential expansion.
— The situation of the fixed points.

ROTOR DYNAMICS

In comparing the types of turbines, particularly in the case of the high output ammonia synthesis compressors, the effects of the self excited vibrations of a turbine rotor should be borne in mind.

Over the last few years, considerable efforts have been made by means of theoretical investigations, combined with the evaluation of practical results, to extend the knowledge about the phenomenon of rotor vibrations. Such self excited vibration effects have occasionally occurred on high speed compressors and industrial steam turbines. This instability is primarily assigned to a behavior of the journal bearing lubricating film known as "oil whip"; above a certain speed the shaft vibrations

TABLE 4.

	Gear Type Coupling	Solid Coupling
Power transmission	Line contact	Friction
Installation	Meticulous care	Normal care
Lubrication	Very clean oil required, problems above certain peripheral speeds	Not required
Wear	Possible if not assembled correctly	None
Number of thrust bearings	Separate thrust bearings for turbine and compressor	Only one thrust bearing (turbine)
Thrust bearing loading	Influenced by the friction coefficient	Clearly
Thrust balance	Not possible	Possible
Transmission of differential expansions	No	Yes
Mutual influence on critical speed	Slight	Normal

suddenly increase. The speed at which this occurs usually lies above the first critical speed, whereas the frequency of the vibrations tends to correspond to the first critical speed of the shaft.

Output related shaft instability has also been noticed in other cases at constant operating speeds. H.P. shafts of large turbines or rotors of industrial turbines of high specific output may be affected by this phenomenon. This output related instability has a characteristic sudden increase of vibration level which prevents further loading of the unit.

Influences on the stability

Every rotating shaft together with its bearings and bearing supports forms a system capable of vibration. Excitation forces stimulate the system into vibration, whereas damping forces tend to reduce existing vibrations. As long as the damping forces are in the majority the system will remain stable. As soon as the excitation forces increase the vibration level rises and the system becomes unstable. Nowadays the following are regarded as the main sources of excitation forces:

a) stimulation from the bearing lube oil film
b) stimulation resulting from the steam flowing through the turbine (gap excitation).

Stimulation from the bearing oil film results from the exchange of energy which takes place between the vibrating rotor and the oil film, and here the spring and damping characteristics of the bearing lube oil film play a large part. These characteristics are defined for every type of bearing using the spring and damping factor which has been established by means of special tests (14).

When a turbine shaft is no longer running in the true center of the casing, the blade and sealing clearances vary and transverse forces, proportional to the change in the clearances, arise due to the unsymmetrical pressure distribution. In turbines with a high output/mass (weight) ratio, these forces may cause shaft vibration to the extent that further operation becomes impossible. The output which may be obtained before these instabilities occur is proportional to the first critical speed of the rotor. Consequently, the stiffer drum type shaft has a higher output limit in this respect than the more flexible shaft-and-disc type rotor.

By using today's large computers it is possible to calculate the stability of large output shafts in advance. The result can then be calibrated on plants already in operation. Should a possibility of "oil whip" be apparent, then this can be reduced by using dampened bearings. If, on the other hand, the danger of gap excitation prevails, it can be eliminated by suitable design of the blading.

Such stability calculations have been carried out on the turbine shown in Figure 9 with the result that no stability problems are to be expected in the required operating range. Using suitable bearings, in this case segment bearings, the operating point is sufficiently remote from the stability limit, as can be seen from Figure 20.

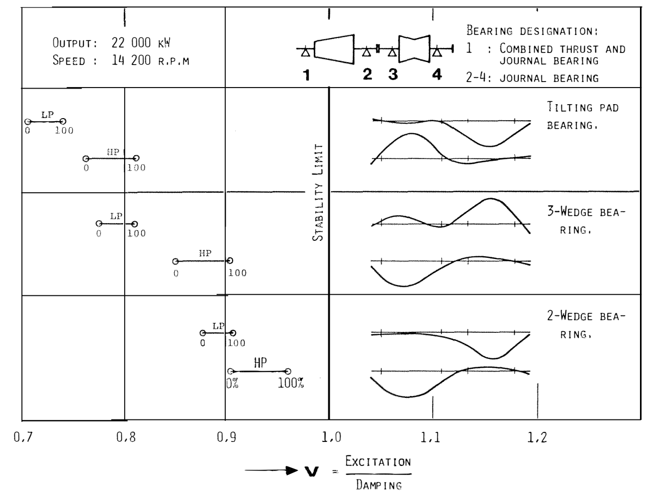

Figure 20. Result of a Stability Calculation on the Rotor of an Ammonia Synthesis Compressor Driver.

REFERENCES

1. R. N. DiNapoli: Trends in base load LNG Plants. Hydrocarbon Processing, December 1975, 87-90.
2. H. Mühlhäuser: Modern feedpump turbines. Brown Boveri Rev. 58 1971 (10), 436-451.
3. A. M. Spechtenhauser: Fundamentals of the Suitability of the Reaction-type Steam Turbine for Compressor and Pump Drive. Paper for the 10th Intersociety Energy Conversion Engineering Conference (759 206). Newark, Delaware, August 1975.
4. G. A. Burkard: Steam turbines for industry. Brown Boveri Rev. 61 1974 (1), 4-8.
5. W. Traupel: Thermische Turbomaschinen, Vol. I & II, 2nd edition, Springer, Berlin/Heidelberg/New York 1968.
6. W. Hort: Schwingungen der Räder und Schaufeln in Dampfturbinen. VDI-Z. 70 1926 (42) 1375-1381 and (43), 1419-1424.
7. M. Lotz: Erregung von Schaufelschwingungen in axialen Turbomaschinen durch die benachbarten Schaufelgitter. Wärme 72 1965 (2), 59-68.
8. J. J. Kirillov, A. S. Laskin: Untersuchung der wechselnden aerodynamischen Kräfte in dem von einem nichtstationären Strom umflossenen Turbinengitter. (Russian.) Energomashinostrojenie 12 1966 (12), 29-32.
9. A. A. Moissejev: Einfluss des Axialspiels auf die Vibrationsfestigkeit der Ueberdruck-Laufschaufeln einer Axialturbine. (Russian.) Energomashinostrojenie 17 1971 (5), 21-23.
10. H. Huppmann: Häufigkeit und Ursachen von Schäden an Bauteilen grosser Dampfturbinen. Maschinenschaden 43 1970 (1), 1-6.
11. E. Somm: A means of estimating the erosion hazard in low-pressure steam turbines. Brown Boveri Rev. 58 1971 (10), 458-472.
12. A. Renaudin, E. Somm: Quasi-three-dimensional flow in a multistage turbine. Flow research on blading. Elsevier, Amsterdam/London/New York 1970.
13. K. Kraemer: Are couplings the weak link in rotating machinery systems. Hydrocarbon Processing, February 1975, 97-100.
14. A. Hohn: Bearings for Steam Turbosets. Brown Boveri 62 1975 (3), 72-83.