SOLID COUPLINGS WITH FLEXIBLE INTERMEDIATE SHAFTS FOR HIGH SPEED TURBOCOMPRESSOR TRAINS

by

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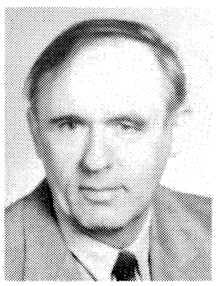
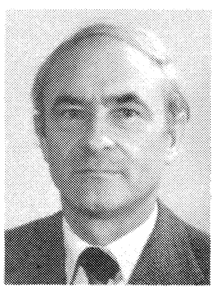


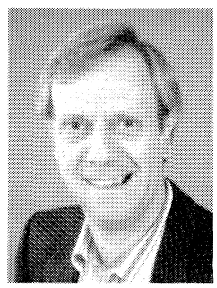
Heinrich Lorenzen is a Mechanical Engineer with more than 25 years of turbomachinery experience. He has been the Head of Design for Turbocompressors, Sulzer Escher Wyss, Zurich, Switzerland, since 1970. Mr. Lorenzen graduated in 1963 as a Dipl. Ing. in Mechanical Engineering from the German Institute of Technology in Kiel.

In 1963, Mr. Lorenzen joined Brown Boveri as a Design Engineer for turbocompressors. Transferring to Brown Boveri Sulzer Turbomachinery and finally to Sulzer Escher Wyss, he is now responsible for the mechanical design of all turbocompressors and tail gas expanders.



Ernst Niedermann is an Engineering Consultant for turbocompressors and expanders with 40 years experience. Until 1988, he was Engineering Manager of the Turbocompressor Department, Sulzer Escher Wyss in Zurich, Switzerland. Mr. Niedermann graduated in 1947 as Dipl. Ing. in Mechanical Engineering from the Swiss Federal Institute of Technology. He started his career with Escher Wyss in the Steam Turbine Department before taking up graduate work at the University of Kansas, USA, in 1949. After receiving his M.S.E. in 1951, he transferred to Westinghouse Steam Turbine Department in Lester, Pennsylvania, and later to ITE Circuit Breaker Company, Philadelphia, Pennsylvania.

In 1956, Mr. Niedermann joined Brown Boveri Switzerland to become manager of the Turbocompressor Development Group. Transferring to Brown Boveri Sulzer Turbomachinery and finally to Sulzer Escher Wyss in Zurich he was responsible for the design of turbocompressors and tail gas expanders, including development and plant layout. Mr. Niedermann retired at the end of 1988, and is now an Engineering Consultant.



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After ten years of practical experience as a mechanic, Mr. Wattinger graduated in 1974, as Dipl. Ing. in Mechanical Engineering from the Swiss Federal Institute of Technology.

Mr. Wattinger joined the Turbocompressor Division in 1975. He transferred from the Research and Development Department to the Design Department in 1979, when he became Head of the Mechanical Calculation Group. Today, Mr. Wattinger is responsible for the rotordynamic design of turbocompressors and tail gas expanders.

ABSTRACT

The design and application of solid couplings combined with flexible quill shafts for high speed turbocompressors are presented.

Solid couplings are rigid in the axial direction. They transmit any residual thrust from one compressor casing to another, and even into the gear. A single thrust bearing then positions the complete rotor train. Elimination of individual thrust bearings results in a smaller lube-oil system and a reduced power loss. Any additional thrust components from a prestressed flexible-disc coupling or a gear-type coupling (gear lock) are completely avoided. The use of a flexible intermediate shaft (quill shaft) gives the solid coupling the same lateral flexibility as for a gear-type or flexible-disc coupling. Criteria for allowable misalignment are given in this paper.

A solid coupling is neither exposed to any kind of wear nor endangered by corrosion or stress corrosion so that no maintenance is necessary.

Couplings have a decisive influence on rotordynamics. The conventional arrangement of a multicasing train with gear-type or flexible-disc couplings incorporates individual thrust bearings located outboard of the journal bearings (long shaft over-

hangs). Introducing solid couplings leads to a stiffer rotor by eliminating individual thrust bearings. A comparison of critical speeds of a high-speed/high-power compressor train is made alternatively equipped with solid couplings, flexible-disc or gear-type couplings. The response of the different rotors and couplings to specified unbalance weights is calculated and compared throughout the complete speed range.

Torsional and lateral critical frequencies can easily be influenced by modification of intermediate shafts, even after installation of a compressor train.

Operating experience on nearly 1500 compressors equipped with solid couplings have proven their excellent reliability especially on high speed, high power turbocompressor applications.

INTRODUCTION

For many decades, it has been quite common to apply solid couplings with bolted flanges to all large utility steam turbine generator sets positioning the complete rotor train on one single thrust bearing. No other types of couplings are able to transmit the extreme torques between the various turbine rotors and the generator. This simple concept was adopted by one compressor manufacturer over 40 years ago, whereby some details were modified to adapt the design to higher speeds, and for easy assembly and disassembly during overhauls.

Couplings are vital mechanical components in a multirotor train and must be designed to ensure continuous and trouble-free operation for years, without interruption for servicing. They may be exposed to the most adverse conditions, causing axial, angular or parallel misalignment of adjacent rotors. For example, there may be excessive thermal growth of rotors during a cold startup, extreme piping forces on casings, uneven settling of the foundation, and temperature gradients on baseplates. Couplings should take care of all such conditions without causing vibration problems, and without giving rise to excessive stresses or even failure in gear teeth, disc elements, shafts or thrust bearings.

The importance attributed to couplings for turbomachinery is expressed in the API Standard 671, "Special-Purpose Couplings For Refinery Services" [1], where recommendations for gear-type, flexible-disc and quill shaft couplings are given.

Historically, the oil-filled or continuously lubricated gear-type coupling was the only widely-used type in the turbomachinery world. Its lateral and axial flexibility is obtained by a rocking and sliding motion between mating gear teeth. Over many years of operation, fretting of the teeth may occur (the phenomenon of "gear lock") and the resulting high axial forces can cause overloading or even failure of thrust bearings of two adjacent rotors. This particularly holds true for high speed rotors where the danger of sludging gear teeth is inherent. Another problem attributed to gear-type couplings was the occasional excitation of double frequency rotor vibrations. Engineers have therefore been searching for solutions that would eliminate these difficulties.

To eliminate these problems, the authors' company adopted the quill shaft solution for high speed applications. The lateral and torsional flexibility is provided by means of an elastic intermediate shaft combined with the elasticity of the stub shafts of the adjacent rotors (Figure 1).

An alternative solution to the high speed deficiencies of gear-type couplings is the flexible-disc coupling or the diaphragm coupling. Both of these give lateral and axial flexibility, but each rotor must be axially positioned by an individual thrust bearing.

This presentation will serve to familiarize users of turbomachinery with the solid coupling (quill shaft coupling) by presenting basic design and by giving quantitative figures on allowable lateral misalignment for a specific example. Comparative figures

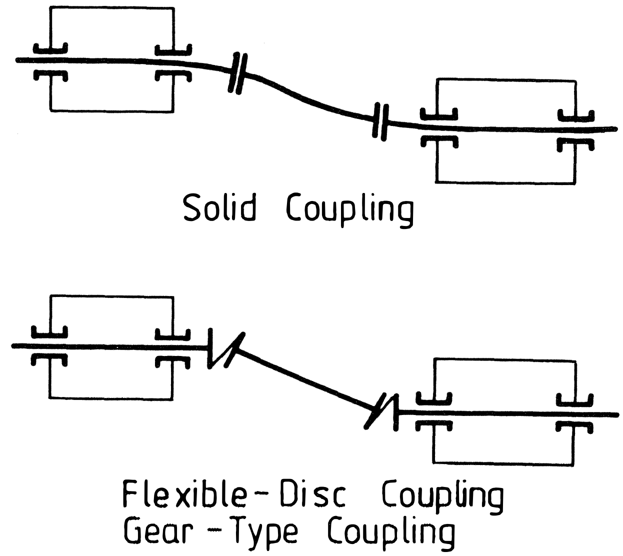

Figure 1. Lateral Misalignment of Different Types of Couplings.

for a conventional flexible-disc coupling and a gear-type coupling will be quoted. The influence on critical speeds and the response to unbalance weights for the three types of couplings are also presented.

DESIGN OF SOLID COUPLINGS

The design of the modern turbocompressor solid coupling has evolved from that used in large utility steam turbine sets. The original design was quite adequate when compressors operated at relatively moderate speeds. However, as speeds of modern turbocompressor sets steadily increased some problems started to arise. For instance integral coupling hubs for associated machine shafts with piloted face fits were occasionally the cause of rotor vibrations, especially after rotor overhauls. Piloted fits must by nature have a small positive clearance for repeated assembly and disassembly when rotors are serviced. A few hundredths of a millimeter of diametral clearance may be sufficient to upset the balance to an unacceptable level. In addition, the amount and the location of the unbalance may differ each time after reassembly. Because of the need for servicing of compressor seals, the original design with integral coupling flanges and piloted fits was improved. The modern design is equipped with removable hubs which enable the coupling to be balanced as a complete unit (hub/intermediate shaft/hub).

Principle Design Features

The principle design features of a modern turbocompressor solid coupling are listed below and shown in Figures 2 and 3:

- The design conforms to the API Standard 671, particularly to paragraph 2.4 [1].
- The coupling hubs are hydraulically fitted (tapered bore) and secured by a shaft end retaining nut.
- The three coupling components (the two hubs and the intermediate quill shaft) are centered by a number of tapered dowel pins. The corresponding bores for these dowels are reamed after fitting of the parts. They are designed to transmit the full torque.
- Separate tie bolts secure the coupling flanges to allow transmission of thrust forces.
- After removal of the tie bolts and of the dowel pins the intermediate shaft can easily be removed with the adjacent rotors remaining in place.

- The coupling is balanced as a complete assembled unit.
- There is no coupling lubrication.
- There are no wearing parts, and no maintenance is necessary over years of operation.
- The coupling is easy to inspect, to dismantle, and to refit (servicing of seals) without disturbing the balance of the coupling and adjacent rotors.
- Balancing holes are provided for possible field balancing of the rotor train.

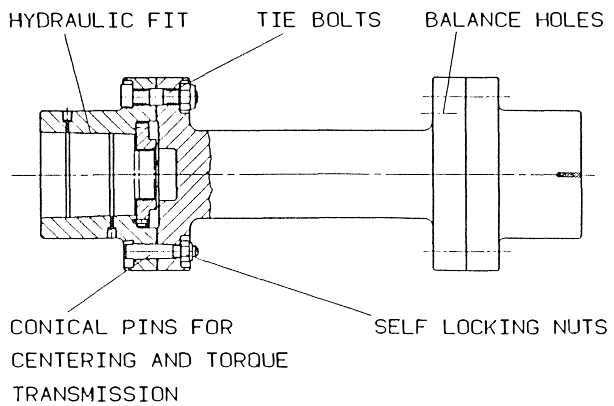

Figure 2. Basic Design of Solid Couplings.

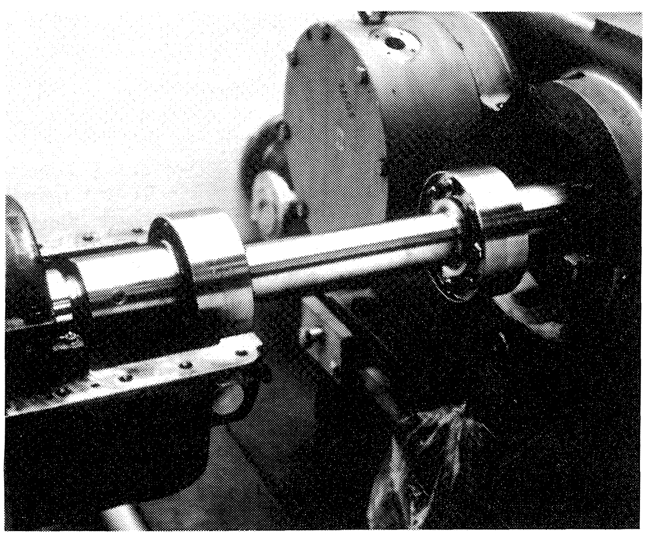

Figure 3. Solid Coupling Installed Between Compressor Shaft and Gear Pinion.

Maximum Torque Transmission

The solid couplings and coupling-to-shaft junctures are designed for continuous operation according to API 671, para. 2.1.1, [1] including requirements for transient conditions. The ratio of transmissible to nominal torque as a function of coupling diameter is given in Figure 4. For transient conditions, such as startup with a synchronous motor, the coupling must be able to handle several times the nominal torque without causing any damage. If a coupling, or particularly its shrink fit, is not adequate for transient conditions, the next bigger size has to be selected.

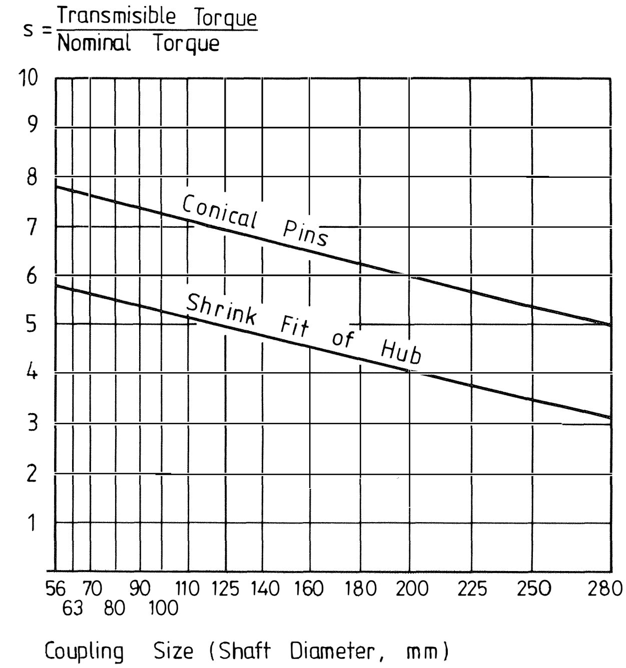

Figure 4. Transmissible Torque for Solid Couplings.

Comparison of the Misalignment Capability of a Solid Coupling With Other Types of Flexible Couplings

The solid coupling with flexible intermediate shaft allows a similar amount of radial misalignment as any other type of flexible coupling. The maximum allowable misalignment is never limited by the bending stress (fatigue) of the intermediate shaft, rather, it is determined by the loading of the associated bearings and their respective oil film temperatures. An example is shown in Figure 5 of a motor/gear driven compressor train, with two casings. Quantitative figures for the allowable misalignment have been calculated. The high speed shaft transmits 7,200 kW and rotates at 10,500 rpm. The string is equipped with solid couplings transmitting any residual aerodynamic thrust from both compressors through the gear pinion (equipped with thrust collars) to a low-speed thrust bearing. The calculated combined stress (torsion plus bending) at the critical location of the intermediate shaft I is shown in Figure 6 (Smith diagram), along with the changes in specific bearing loading and the bearing temperatures when the LP-compressor is raised or lowered (parallel offset). Both the intermediate shafts and the stub shafts of the adjacent rotors take part in the elastic lateral deformation, and

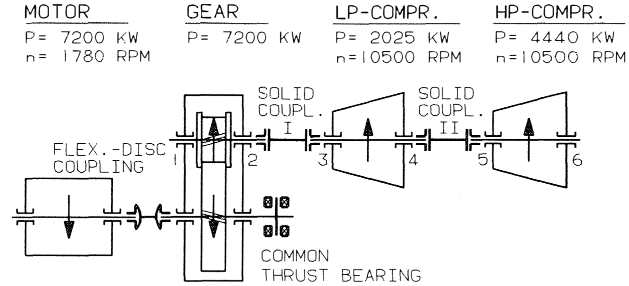

Figure 5. Basic Arrangement for a Motor/Gear Driven Compressor Train Equipped with Solid Couplings in the High-Speed Rotor String.

this helps to reduce all forces and loads. The stress equivalent to the nominal driving torque combined with the alternating bending stress in the intermediate shaft I leads to an allowable parallel offset of more than 10 mm. However, if a maximum bearing temperature of 100°C is considered acceptable, the possible parallel offset will be limited to 1.5 mm. Detailed figures for all bearings (bearing loads and temperatures) are given in Tables 1 and 2, along with the maximum alternating bending stresses along the intermediate shafts for a parallel offset of ± 1.5 mm. The fatigue limit for all shafts is at roughly 4,000 bar (58,000 lb/in^2). There is a wide safety margin with respect to the alternating stress, even considering a notch factor of 1.5. The length or the diameter of the intermediate shaft can be varied in order to influence the allowable parallel offset.

For the same example, the allowable parallel offset when applying other types of flexible couplings is shown in Figure 7. As a basis for an acceptable misalignment an angular deflection of

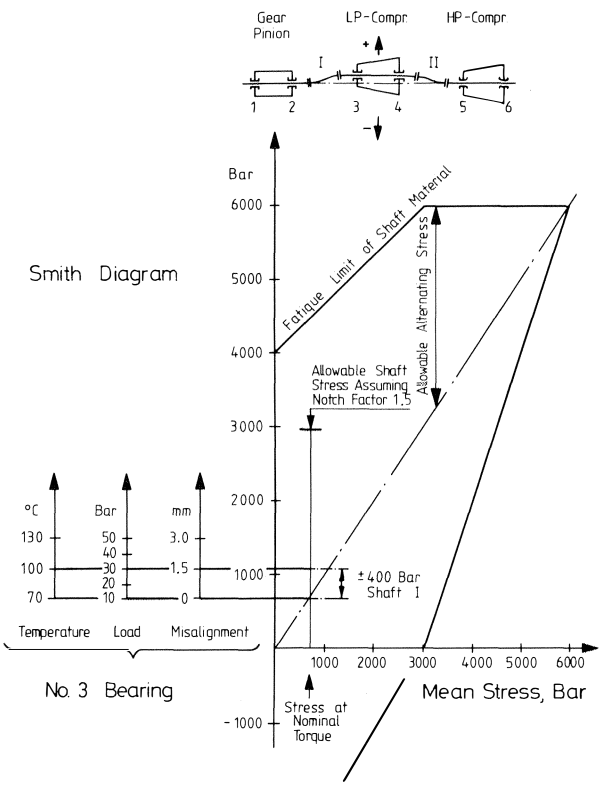

Figure 6. Allowable Misalignment of the LP-Compressor.

Table 1. Influence of +/−1.5 mm Parallel Offset of the LP-Compressor on Bearing Loads and Bearing Temperatures

		Dimen-sion	Bearing Number					
			1	2	3	4	5	6
perfect alignm.	bear.load	Bar	−22.1	−25.2	9.3	5.0	5.6	5.1
		lbf/in^2	−320	−365	135	72	81	74
	bear.temp.	°C	90	92	70	65	65	65
		°F	194	198	158	149	149	149
+1.5 mm offset	bear.load	Bar	−19	−35	29	17	−10	9
		lbf/in^2	−276	−508	421	247	−145	131
	bear.temp.	°C	87	100	100	80	70	69
		°F	189	212	212	176	158	156
−1.5 mm offset	bear.load	Bar	−25	−15	−8	−7	22	1
		lbf/in^2	−363	218	−116	−102	319	15
	bear. temp.	°C	92	85	68	67	88	60
		°F	198	185	154	153	190	140

Table 2. Influence of +/−1.5 mm Parallel Offset of the LP-Compressor on Alternating Bending Stresses

	Dimen-sion	Alternating Bending Stress	
		Intermediate Shaft I	Intermediate Shaft II
perfect alignment	Bar	0	0
	lbf/in^2	0	0
+/−1.5 mm offset	Bar	+/−400	+/−300
	lbf/in^2	+/−5800	+/−4350

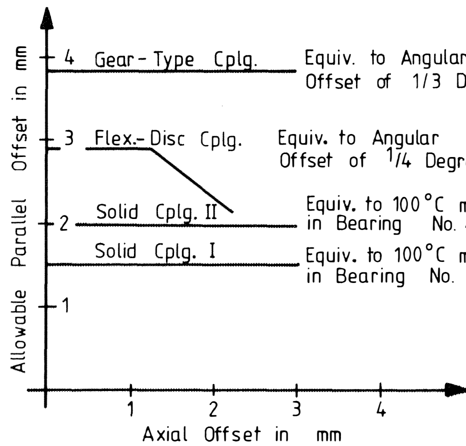

Figure 7. Comparison of Allowable Parallel Offset for: solid coupling, flexible-disc coupling, gear-type coupling.

the spacer of 1/3 degree for a gear-type coupling and ¼ degree for a flexible-disc coupling was taken.

It is easier to detect misalignment problems when using solid couplings with flexible intermediate shafts than when using other types of couplings. Possible causes for extreme misalignment such as excessive piping forces on casings, settling of foundation, or temperature gradients on baseplates, for instance hot days, cool nights in open air installations, will show up as a change of the bearing loads or bearing temperatures of adjacent journal bearings, so that corrective measures can be taken well before a failure may occur.

ROTOR TRAINS EQUIPPED WITH SOLID COUPLINGS

Application Examples

Two typical examples using solid couplings are presented in Figure 8. The first has a direct high-speed turbine drive and the second has the same compressor train with a motor/gear drive.

Referring to the first example, a steam or a gas turbine drive in general has its own thrust bearing. An axially and radially flexible coupling between the drive and the driven machines is then mandatory. The two-casing compressor string is positioned by only one single thrust bearing located on the intermediate shaft between the compressors. It is easily accessible for a quick checkup. This location also allows better access to the seals without disassembly of the thrust bearing. In addition, the thrust collar can now be integral with the intermediate shaft, as is preferred by API 617 [2]. Axial expansion of the associated compressor rotors in either direction is no problem if an appropriate flexible-disc coupling is selected on the driving end. The compressor thrust bearing must be capable of handling the combined residual aerodynamical thrust of both rotors, plus the axial force exerted by the flexible-disc coupling on the driving side. It

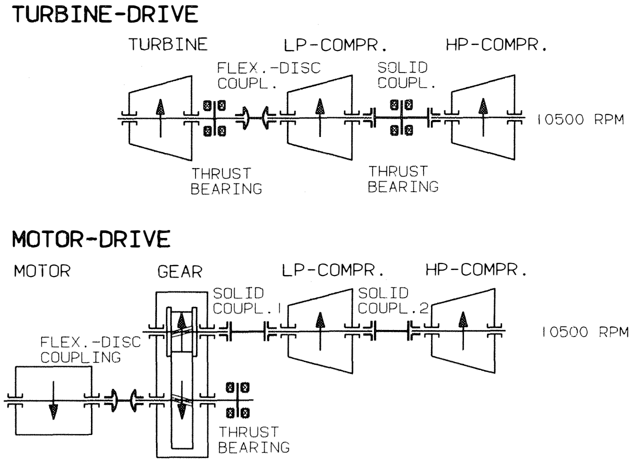

Figure 8. Basic Arrangement of a High-Speed Compressor Train with Turbine Drive or a Motor/Gear Drive.

would be possible to add a third compressor casing to the existing train by means of another solid coupling if consideration is given to the additional thermal growth towards the tail end.

Introducing a motor drive combined with a speed increasing gear (Figure 8), the same two-casing compressor train as previously described is now lined up with a gear pinion by installing another solid coupling. The pinion of the single helical gear is equipped with thrust collars. These compensate the helix thrust in situ, and allow the gear to transmit considerable additional residual thrust from the compressor rotors to a high capacity thrust bearing on the low-speed shaft of the gear. The driving motor coupled through a flexible-disc coupling can also be positioned by the same thrust bearing, if consideration to possible axial vibration is given.

Illustrations of compressor trains executed with solid couplings are shown in Figures 9 and 10.

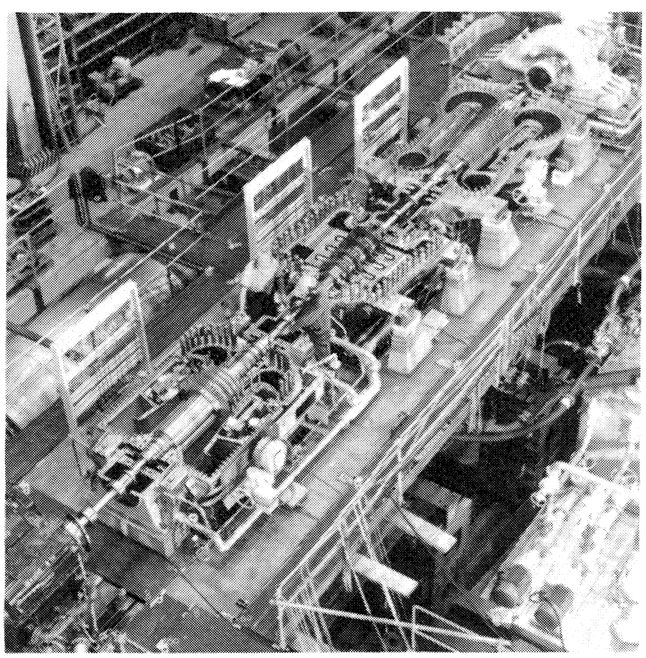

Figure 9. Turbine Driven Compressor Train: steam turbine/flexible-disc coupling/axial compressor/thrust bearing/solid coupling/centrifugal compressor/solid coupling/tail gas expander/solid coupling/gear/solid coupling/generator.

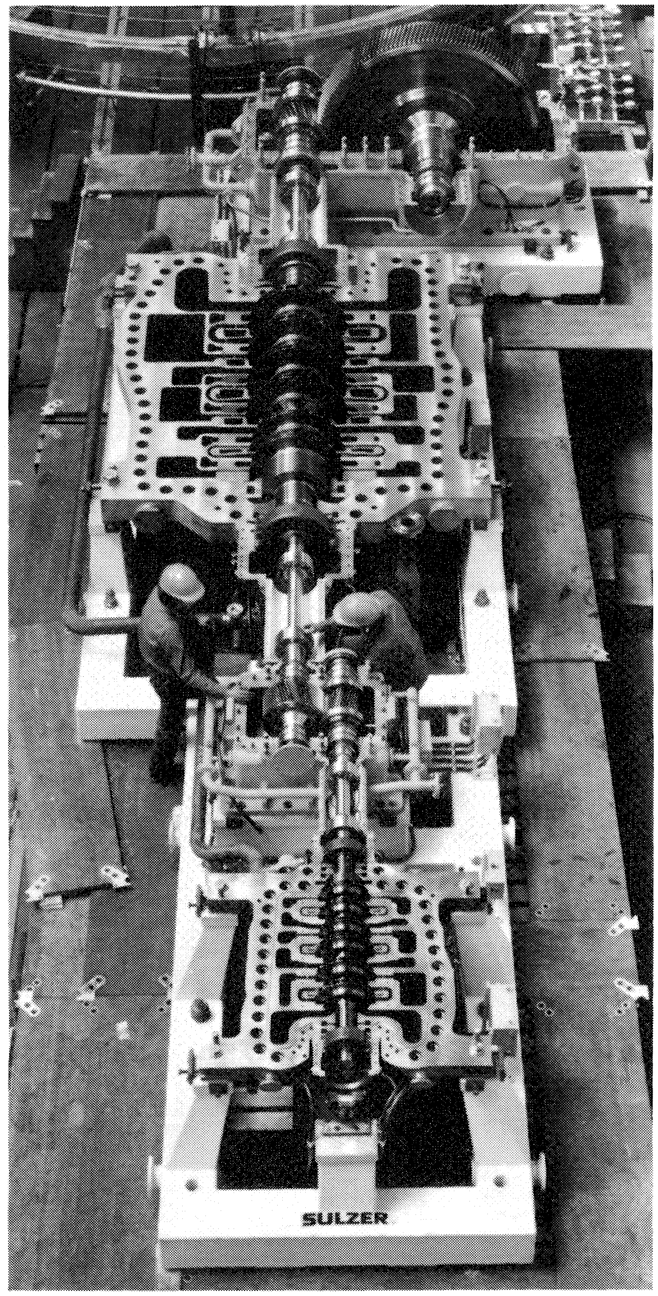

Figure 10. Motor/Gear-Driven Compressor Train: motor/gear coupling/gear with low speed thrust bearing/pinion/solid coupling/LP-compressor/solid coupling/gear/solid coupling/HP compressor.

Basic Alignment of Rotors During Erection

For the erection of a multicasing train in its cold condition, the rotor axes have to be set to allow for any uneven temperature expansion of the bearing supports during startup. This procedure is standard for any kind of coupling. For solid couplings, however, the rules given for large steam turbine-generator sets apply. Instead of leveling each casing individually, the rotors are lined up to have parallel faces between two adjacent coupling-flanges (Figure 11). This method minimizes the additional bending moments originating from the natural sag of the rotors. The rotational axis of the combined rotor string will therefore be a curved line.

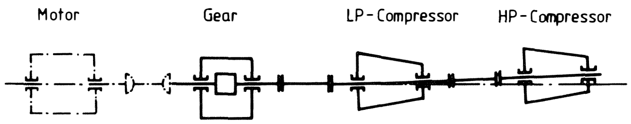

Figure 11. Vertical Alignment of Compressor Train Equipped with Solid Couplings. Coupling flange faces are set parallel (allowance for rotor sag).

Effects of a Baseplate Deformation on the Rotor String

A baseplate of a compressor train may be exposed to changing forces of the piping, additional weights installed, to periodic temperature gradients (day/night effect, sun radiation), or even wind forces on a platform. Each of these can cause a deformation of the baseplate and consequently of the rotor string.

In order to quantify such effects, a model calculation has been performed for the compressor train shown in Figure 5. The compressor train is arranged on a common baseplate together with the complete lube-oil and seal-oil system (Figure 12). A linear temperature gradient of 50°C from the bottom to the top of the main supporting beams was assumed (sun radiation during daytime) as typical of an outdoor installation. This temperature gradient will cause the baseplate to rise by 7.5 mm in the middle between the two supports. The axis of the rotor string will lift correspondingly causing bending mainly in the intermediate quill shafts and also influencing bearing loads. The results of this calculation are listed in Tables 3 and 4. In comparison to a local offset of the LP-compressor (Tables 1 and 2), a smooth variation of a long rotor string has a negligible effect. Another model calculation introducing additional weights or forces (equivalent to an addition of 20 percent of the weight of the total structure) alters the natural sag of 4.5 mm of the baseplate by only 1.0 mm, showing it to have even less effect than the temperature gradient. The minute influence of a baseplate deformation on the rotor string can be explained by the highly elastic nature of the rotor string in comparison to the baseplate.

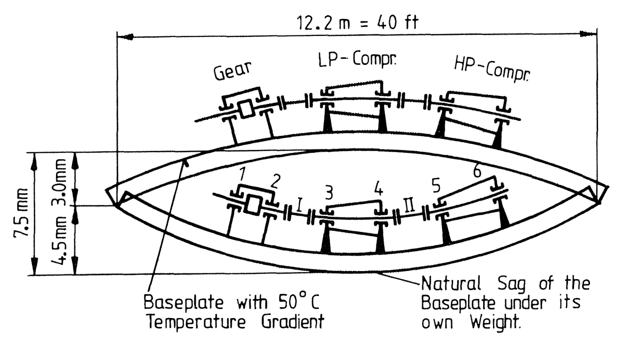

Figure 12. Effect of a 50°C Temperature Gradient in the Baseplate (from the bottom to the top) on the Rotor String.

Table 3. Effect of Baseplate Deformation on Bearing Loading

Bearing	Dimension	Specific Bearing Loading 1	2	3	4	5	6
Natural Sag of Baseplate 4.5 mm = 0.18 in.	bar	−22.1	−25.2	9.3	5.0	5.6	5.1
	lbf/in²	−320	−365	135	72	81	74
Temperature Gradient 50°C 7.5 mm lift = 0.30 in.	bar	−22.6	−24.8	10.0	4.6	6.5	4.4
	lbf/in²	−328	−360	145	67	94	64

Table 4. Effect of Baseplate Deformation on Bending Stresses of Intermediate Shafts

	Dimension	Alternating Bending Stresses: Intermediate Shaft I	Intermediate Shaft 2
Natural Sag of Baseplate 4.5 mm = 0.18 in.	bar	0	0
	lbf/in²	0	0
Temperature Gradient 50°C 7.5 mm lift = 0.30 in.	bar	80	90
	lbf/in²	1160	1300

TRAIN ARRANGEMENTS WITH THREE DIFFERENT KINDS OF COUPLINGS

Misalignment problems of the rotor string can basically be solved by any type of coupling. This section gives some consideration to the train arrangements as affected by the choice of coupling, and the next section discusses the effect of the coupling on rotordynamics.

Taking the example with a motor/gear drive of the compressor train, as shown in Figure 5, the effect of solid, flexible-disc or gear-type couplings on the high speed string is now considered (Figure 13). For the solid couplings (case 1), there is but one common thrust bearing for the complete train (including motor) located on the low speed shaft of the gear. Any residual thrust from the compressors is transmitted through the thrust collars

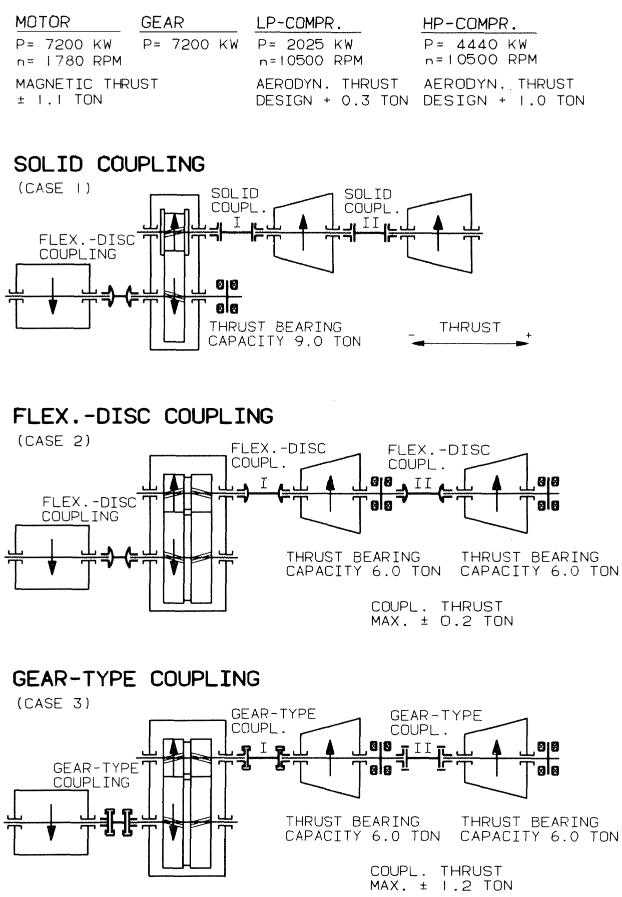

Figure 13. Train Arrangements for Solid Coupling, Flexible-Disc Coupling and Gear-Type Coupling.

of the pinion to this bearing. The motor magnetic thrust will be transmitted through a flexible-disc coupling. A rated thrust capacity of nine tons (equal to 50 percent of ultimate load) was selected for this particular case. However, this capacity could be doubled with no restriction, a feature which could be particularly important for compressors working at extremely high pressure levels.

When using a flexible-disc coupling (case 2), or a gear-type coupling (case 3), individual high speed thrust bearings must be installed in each compressor. If the same kind of thrust collar gear or alternatively a double helical gear is chosen, the drive motor can be axially positioned by the thrust bearing of the LP-compressor, thus avoiding an additional thrust bearing.

It is well known that the thrust of turbocompressors, especially for high pressure applications, may change a great deal in the course of time. Fouling, corrosion and increased clearances of internal labyrinth seals due to a rub may cause changes in thrust. A compressor manufacturer must consider these effects by estimating their influence. Precautions can then be taken where necessary by providing extra capacity for thrust bearings which would enable an operator to maintain production without the danger of a shutdown.

To demonstrate this, a calculation of the variation of thrust for both compressors has been performed assuming increased clearances on all impeller labyrinth seals (2.0 mm instead of 0.3 mm). The results for each coupling type are given in Table 5. The maximum component of the magnetic thrust of the motor, plus that exerted by the flexible-disc coupling, or alternatively gear-type coupling, must be added to the aerodynamic thrust. It is well known that high speed thrust bearings are very limited in their thrust carrying capacity when compared to low speed bearings.

Table 5. Thrust Forces, Thrust Bearing Capacities for 3 Types of Couplings

Coupling Type	Dimension	Case 1 solid	Case 2 flex.-disc	Case 3 gear-type
LP-Compressor				
aerodynamic thrust, design	ton	+0.3	+0.3	+0.3
max.	ton	+1.2	+1.2	+1.2
max. thrust from coupling	ton	-	+0.2	+1.2
max. thrust from motor	ton	+/−1.1	+/−1.1	+/−1.1
max. net thrust	ton	+2.3	+2.5	+3.5
thrust bear. cap. (50% ult. load)	ton	-	+/−4.0	+/−4.0
thrust bearing margin	ton	-	1.5	0.5
thrust bearing power loss	KW	-	37	37
thrust bear. lube oil consump.	ltr/min	-	90	90
lube oil cons. gear-type coupl.	ltr/min	-	-	10
HP-Compressor				
aerodynamic thrust, design	ton	+1.0	+1.0	+1.0
max.	ton	+4.2	+4.2	+4.2
max. thrust from coupling	ton	-	+0.2	+1.2
max. net thrust	ton	+4.2	+4.4	+5.4
thrust bear. cap. (50% ult. load)	ton	-	+/−6.0	+/−6.0
thrust bearing margin	ton	-	1.6	0.6
thrust bearing power loss	KW	-	27	27
thrust bear. lube oil consump.	ltr/min	-	60	60
lube oil cons. gear-type coupl.	ltr/min	-	-	10
Gear				
max. net thrust	ton	+6.5	-	-
thrust bear. cap. (50% ult. load)	ton	+/−9.0	-	-
thrust bearing margin	ton	2.5	-	-
thrust bearing power loss	KW	6	-	-
thrust bear. lube oil consump.	ltr/min	13	-	-
Tot. thrust bear. power loss	KW	6	64	64
Tot. thrust bear. lube oil cons.	ltr/min	13	150	170
Tot. lub. oil cons. (mot/gear/compr)	ltr/min	306	443	463
Tot. lub. oil cons. (mot/gear/compr) (including 25% margin)	ltr/min	383	554	579
Tot. lub. oil tank capacity	ltr	3065	4432	4632

Low speed thrust bearings have many tons of extra margin and only a small power loss. In the example given, the thrust capacity could, if necessary, be raised from 9 tons to 13 tons with a corresponding power loss of only ten kilowatts instead of six kilowatts.

Because of the high power loss of the high speed thrust bearings for cases 2 and 3, the corresponding lube oil system has to be 45 percent larger (oil pump, coolers, filters, oil tank, etc.). The additional capital cost for the oil system is accompanied by the high energy cost for the extra power loss (64 kW instead of 6 kW).

ROTORDYNAMICS

Critical Speed Maps for the LP-Compressor with Different Kinds of Couplings

Based on the train arrangements shown in Figure 13 (case 1: solid coupling, case 2: flexible-disc coupling and case 3: gear-type coupling), a study of the rotordynamics of all three cases has been performed. The compressors for all three versions are identical except for the thrust bearings installed on the non-driven sides of both compressors (cases 2 and 3). The calculation shows that the type of coupling has only a marginal influence on the first bending modes and the conical modes. However, when the overhang critical speeds are compared for the flexible couplings with those for the solid coupling (intermediate shaft criticals), essential differences do appear. This becomes clear from the critical speed maps of the LP-compressor equipped with the various types of couplings (Figures 14, 15, 16, 17, 18). For cases 2 and 3, a thrust bearing is situated between the journal bearing 4 and the coupling II, resulting in a long and heavy overhang. Distinct overhang criticals at relatively low frequencies are apparent for the cases with flexible couplings.

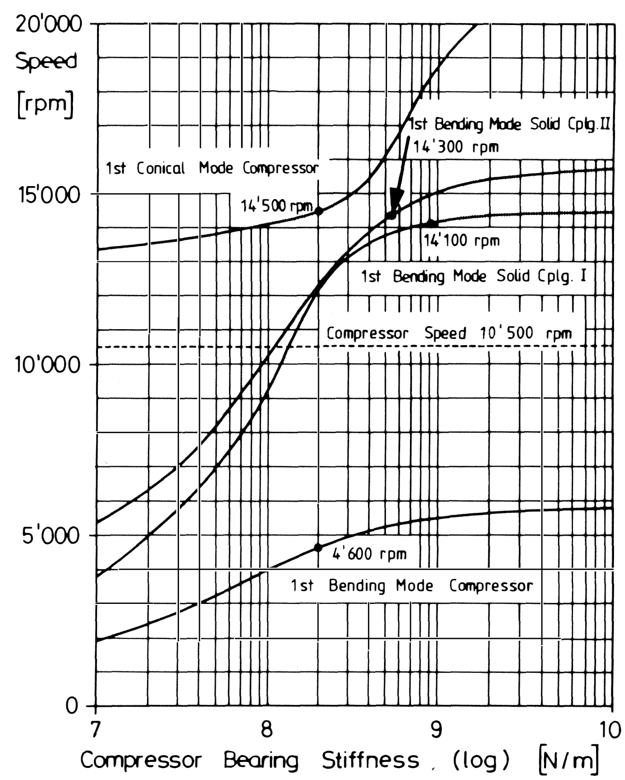

Figure 14. Lateral Critical Speed Map for the LP-Compressor. Case 1. solid coupling (steel), normal shaft.

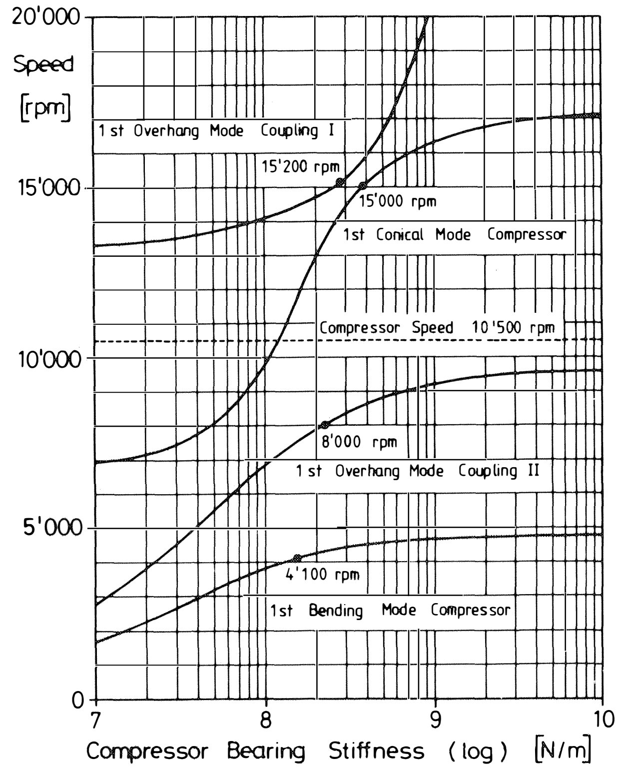

Figure 15. Lateral Critical Speed Map for the LP-Compressor. Case 2a. flexible-disc coupling (steel), normal shaft.

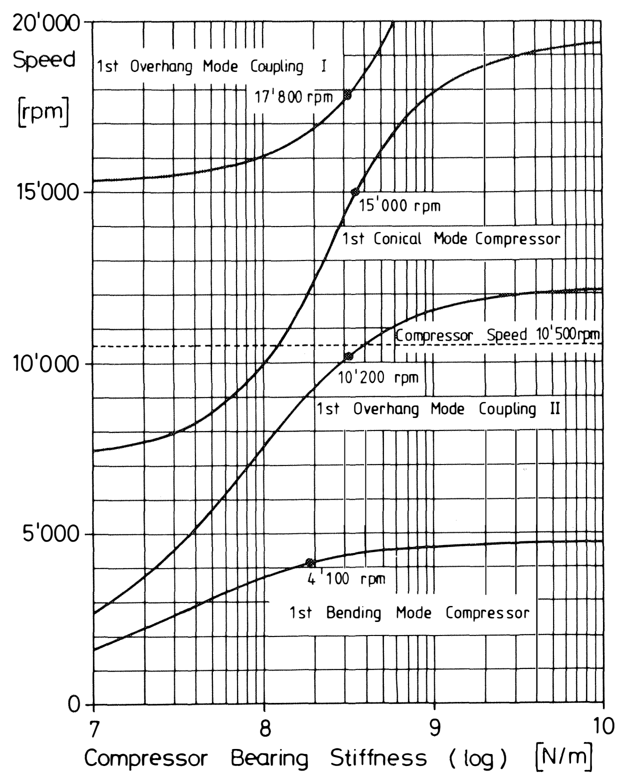

Figure 16. Lateral Critical Speed Map for the LP-Compressor. Case 2b. flexible-disc coupling (steel), reinforced shafts.

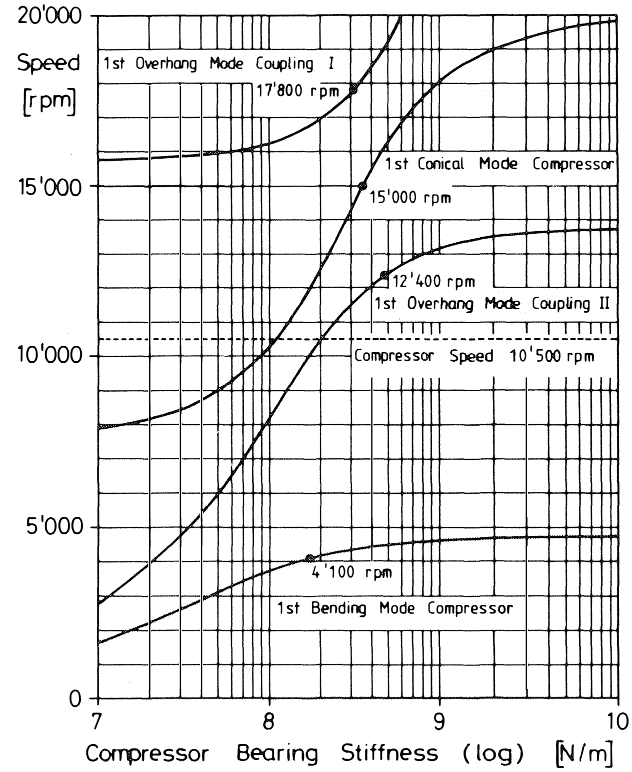

Figure 17. Lateral Critical Speed Map for the LP-Compressor. Case 2c. flexible-disc coupling (titanium), reinforced shafts.

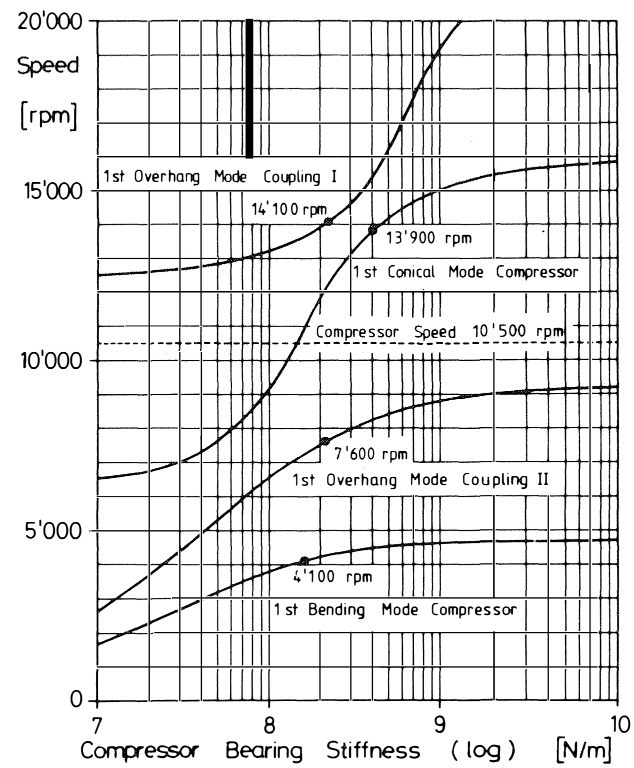

Figure 18. Lateral Critical Speed Map for the LP-Compressor. Case 3. gear-type coupling (steel), normal shaft ends.

Five different calculations with different coupling configurations have been investigated, and the results are presented in Figures 14 to 18. The cases are:

- Case 1. *Solid coupling:* steel, normal shaft ends (Figure 14).
- Case 2a. *Flexible-Disc Coupling:* steel, normal shaft ends (Figure 15).
- Case 2b. *Flexible-Disc Coupling:* steel, reinforced shaft ends (Figure 16).
- Case 2c. *Flexible-Disc Coupling:* Titanium for coupling II, reinforced shaft ends (Figure 17).
- Case 3. *Gear-Type Coupling:* steel, normal shaft ends (Figure 18).

The following comments may help in the interpretation of Figures 14 to 18:

Case 1. Solid couplings: This version has the widest margin between the operating speed (10500 rpm) and any of the criticals. The closest one (14,100 rpm) is the first bending mode of the intermediate shaft I. This gives a margin of 34 percent. In other words, from the point of view of critical speeds, this rotor could easily run as high as about 12,000 rpm, and even higher if shaft ends were reinforced. From the point of view of impeller tip speed, this would also be possible.

Case 2a. Flexible-Disc Couplings: With identical shaft ends as for the solid coupling, an overhang critical speed of 8,000 rpm at coupling II appears. During each startup or shutdown, the compressor train must run through that critical which, in the course of time, may pose a vibration problem.

Case 2b. Flexible-Disc Coupling with Reinforced Shaft Ends: The overhang critical speed is nearly coincident with the operating speed. Additional reinforcement of the shaft ends does not improve the situation much and, in any case, is also not feasible as far as the seals and the bearings are concerned. This solution must, therefore, be abandoned.

Case 2c. Flexible-Disc Coupling with Titanium Hub: Because of the light weight coupling, the overhang critical has moved up to 12,400 rpm, which is acceptable with the amplification factor below 3.55 [2]. However, there is no additional margin for a higher speed as there is for the solid coupling solution. Applications requiring higher speeds could not be handled any more. In other words, additional stages would have to be introduced or the next bigger compressor frame size must be selected.

Case 3. Gear-Type Couplings with Normal Shaft Ends: The overhang critical speed has dropped to 7,600 rpm. So the comment is the same as for the flexible-disc coupling with normal shaft ends. Reinforcement of the shaft ends does not improve the situation very much.

Response Analysis for Different Kinds of Couplings

Coupling II was found to be the determining factor in terms of rotordynamics (long overhang), and an unbalance response calculation was made for this coupling. For each case quoted in the previous section, unbalance weights of 16 g(cm) at both flanges of coupling II were introduced. This value corresponds to seven times the residual unbalance weight at this location. The respective results are given in Figure 19. The amplitudes (peak to peak) refer to the journal bearing 4. For the case of the solid coupling, the amplitude at the operating speed is low. In addition, it should be noted that with the solid coupling, one does not need to run through a critical during a startup or a shutdown where an unbalance may cause a problem.

Stability

The logarithmic decrement (log dec) for all five coupling arrangements is plotted in Figure 20. Because of the stiffening effect of the quill shaft the arrangement with solid couplings has by far the highest value of 39 percent, whereby for all the other versions only 26 percent is obtained. This may express itself in lower vibration amplitudes when, for instance, a compressor goes into surge. The main advantage appears, however, in high pressure applications where higher log. decrements allow a given compressor set to reach higher pressure levels before running into subsynchronous vibration problems.

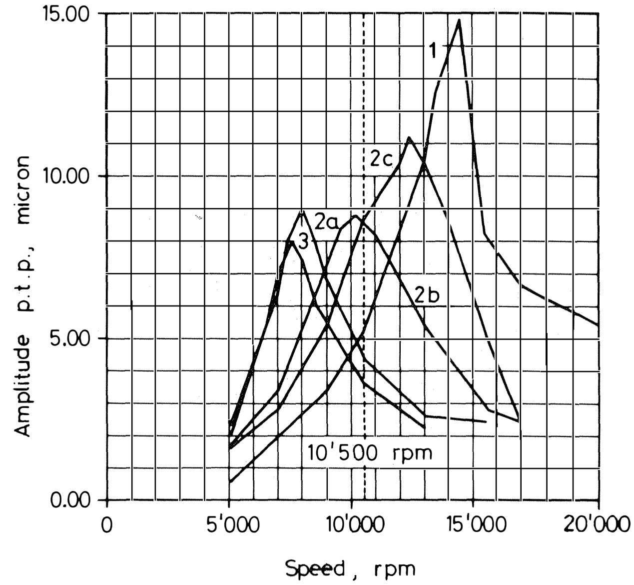

Figure 19. Unbalance Response Curves for Coupling II at Bearing No. 4. Case 1. solid coupling (steel) normal shaft. Case 2a. flexible-disc coupling (steel), normal shaft. Case 2b. flexible-disc coupling (steel), reinforced shaft. Case 2c. flexible-disc coupling (titanium), reinforced shaft. Case 3. gear-type coupling (steel), normal shaft.

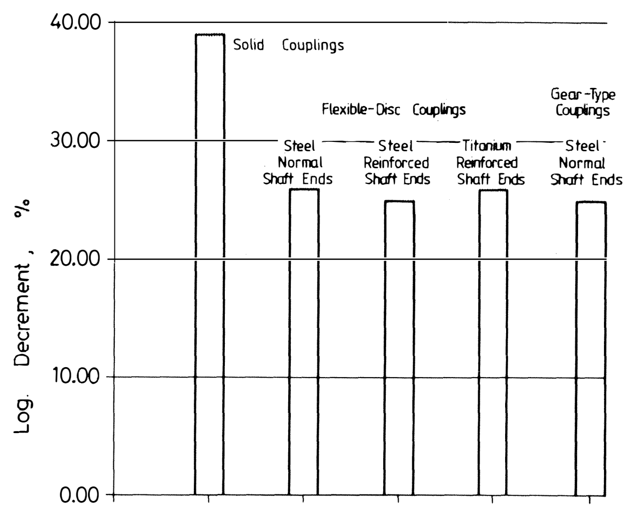

Figure 20. Stability Diagram for the LP-Compressor. Log Dec. for the five coupling alternatives.

Influence of Misalignment on Critical Speeds When Solid Couplings Are Installed

Misalignment of adjacent rotors has only a small influence on the damped natural frequencies of the combined rotor string. Applying the same model as shown in Figure 5, the LP compressor was moved up and down by ± 2.0 mm, and the corresponding influence on the frequencies is plotted in Figure 21. The frequency for the first bending modes of the two compressors lies within a band of 20 percent each. However, this is of no importance, because these frequencies are way below the operating speed. The frequencies for the first bending modes of the two intermediate shafts move within a band of only six percent, and this small variation is of no concern.

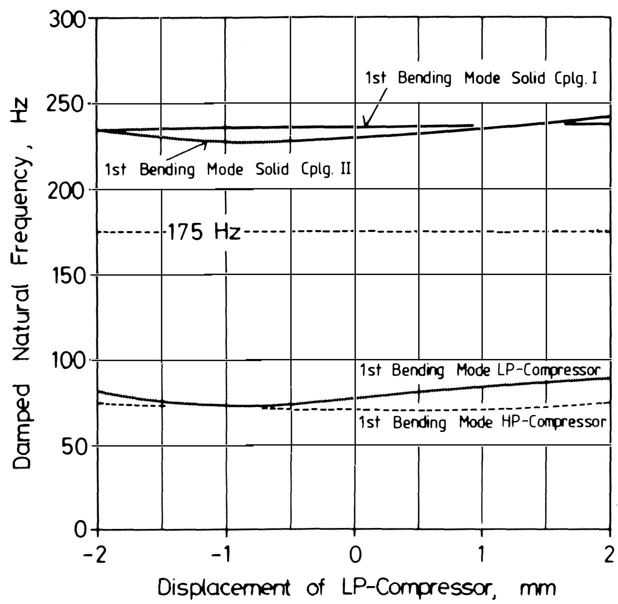

Figure 21. Influence of Misalignment of the LP-Compressor on the Damped Natural Frequencies.

REFERENCES AND OPERATING EXPERIENCE

Nearly 1,500 solid couplings combined with flexible intermediate shafts have been produced and installed by the authors' company. They range in speed from 3,000 to 18,000 rpm, and in power, from a few hundred kW up to 90,000 kW. The approximate numbers of solid couplings are shown in Table 6 for the various power ranges. Extensive operating experience has been gained over the last 40 years. The original design with piloted face fits occasionally caused minor rotor vibrations on high speed compressor sets, especially after overhauls of rotors. Excellent experience has been gained with the improved design over the last ten years when the piloted face fits of the coupling flanges were abandoned and replaced by the tapered dowel pins and flat faces. With these pins centering of rotors became an easy task and unbalances have no longer arisen.

Table 6. Power Ranges of Installed Solid Couplings

Power Range	Number of Solid Couplings
0- 5000 KW	1020
5000-10000	260
10000-20000	110
20000-40000	55
40000-60000	6
60000-90000	6

CONCLUSIONS

A solid coupling, combined with a flexible intermediate shaft, is an excellent means for the transmission of power between adjacent rotors. This holds especially true when high torques at high speeds are involved. Because of its simple design requiring no maintenance, it is suitable for all kinds of applications, especially for those where extreme environmental conditions exist, for instance, highly corrosive and dirty atmosphere. Installing solid couplings on high speed strings eliminates or reduces the number of high speed thrust bearings and, thus, reduces power losses. By the same token the capacity of the lube-oil system may be cut by as much as one-third.

In terms of rotordynamics, the solid coupling eliminates overhang criticals which always occur at relatively low frequencies. Solid couplings, therefore, enable compressors to operate at higher speeds (stiffer rotors). They also lead to higher damping coefficients, which causes the rotors to be more stable, especially for high pressure applications where the danger of subsynchronous vibrations may be involved.

In order to secure satisfactory performance of complete rotor strings, solid couplings, like any other mechanical component, must be properly engineered for each application. They will then give excellent and continuous service over many decades.

REFERENCES

1. American Petroleum Institute, API Standard 671, First Edition, "Special-Purpose Couplings For Refinery Services" (1979).
2. American Petroleum Institute, API Standard 617, Fifth Edition, "Centrifugal Compressors for General Refinery Service" (1988).