

# CONVERSION OF A 19,000 HP PROPYLENE COMPRESSOR FROM STEAM TURBINE TO ELECTRIC MOTOR WITH GEARED VARIABLE SPEED TURBO COUPLING

by

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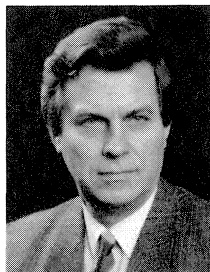
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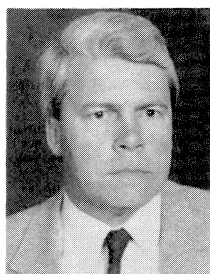
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## ABSTRACT

In the course of modernizing an ethylene plant in 1985, Union Rheinische Braunkohlen Kraftstoff AG converted a propylene compressor with 4 process stages from a steam turbine drive to an electric motor drive.

The economic objective of the conversion was to reduce the required high-pressure steam flowrate, e.g., initiated by converting from light to heavy feedstock.

Speed control of the compressor was to be maintained, since conversion to another control system with constant speed would

have been very difficult with the existing compressor. The decision between electrical variable speed motor and electric motor with geared variable speed turbo coupling was made in favor of the motor with geared variable speed turbo coupling type.

It was necessary to extend the foundation itself, due to the greater overall length of the drive unit (geared variable speed turbo coupling/electric motor) compared to the steam turbine. Additionally, the platform was extended and supported. The geared variable speed turbo coupling and the electric motor were mounted on a common frame designed specifically to be particularly rigid.

The variable speed fluid coupling operating on the hydrodynamic principle was incorporated in a common housing together with its gear stages.

*Available time:* three months during a scheduled shutdown.

No problems occurred during installation and commissioning. The process control system operated satisfactorily; all mechanical, electrical limit values and values relative to vibration were met with a safe margin.

## INTRODUCTION

Union Rheinische Braunkohlen Kraftstoff AG is operating an ethylene plant with two parallel lines of 220,000 metric tons/year ethylene capacity each. In 1985, one of the plants was converted from naphta feedstock to heavier feedstock. Normally the three large turbo compressors in ethylene plants, namely, charge gas, propylene, and ethylene are driven by steam turbines. The high pressure (HP) steam is generated in waste heat exchangers, i.e., so-called transfer line exchangers (TLX). In this case the design was first based on naphta feedstock. The resulting steam flow diagram is shown in Figure 1, in a simplified form.

Conversion to heavy feedstock results in a reduced high pressure steam generation of the plant, i.e., approximately 114 tons/hr, with the same ethylene production due to a higher average temperature of the waste heat exchanger (TLX). This results in a HP steam shortage of nearly 40 tons/hr, with the same steam consumption of the three large compressors. The 40 tons/hr, however, are the steam flowrate required to drive the turbine of the propylene compressor.

This time was opportune to replace this turbine with an electric motor drive, moreover, so the unit operates on a condensation principle. Thus, the changes to the steam system are

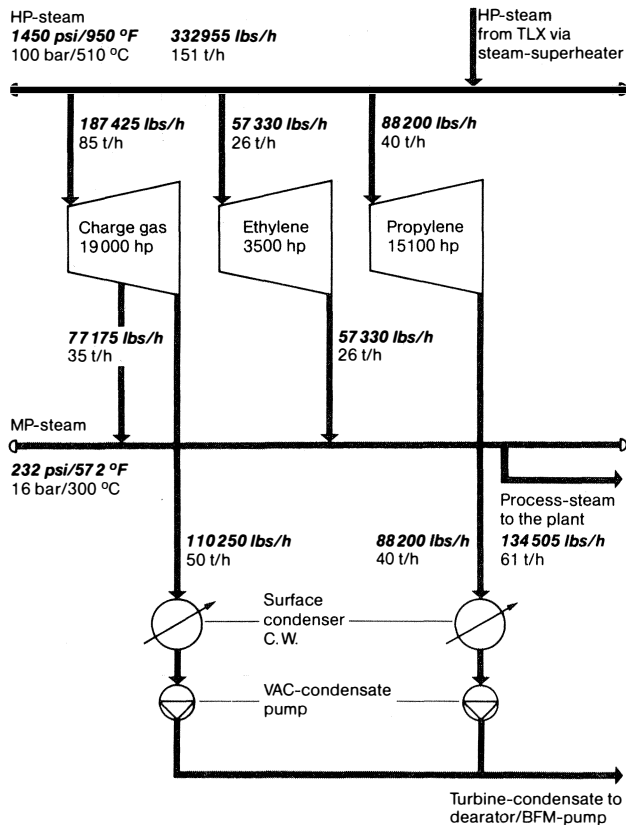


Figure 1. Simplified Steam Circuit Diagram Prior to Conversion.

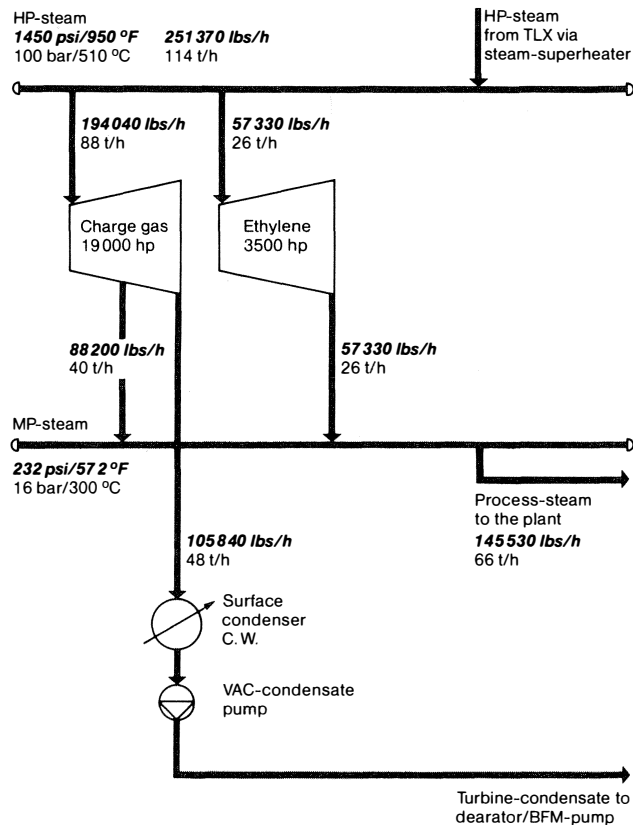


Figure 2. Simplified Steam Circuit Diagram after Conversion.

relatively uncomplicated, resulting in the modified flow diagram shown in Figure 2.

Another alternative would be to procure the shortage of hp steam from the plant's power station. This would, however, exceed the firing capacity of 300 MW of the power station, necessitating the installation of a flue gas desulfurization system. At the time of planning, government regulations required that fuels containing sulfur could not be utilized in stations of more than 300 MW without a desulfurization system. This made the selection of an electric motor drive the logical guideline for planning.

The propylene compressor to be converted is a refrigeration unit with four process stages (Figure 3) and a required power of 19,000 hp. Together with the ethylene refrigerating compressor, it is the coolant cascade for gas separation in an ethylene plant.

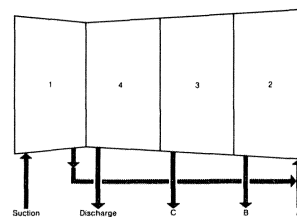
Therefore, the two most important criteria apply for the propylene refrigerating compressor are:

- High availability
- Quick restart capability after trips (shutdowns)

**DEVELOPMENT OF THE CONVERSION CONCEPT**

After it was certain that the only solution to balance the steam rate of the whole plant would be the conversion of the propylene compressor to an electric motor drive, the available solutions were investigated.

The control concept of the compressor was based on a stabilization of suction pressure of the first process stage by controlling the compressor speed.



Section		1	2	3	4
Suction pressure,	psi	19	43	101	158
Suction pressure,	ata	1.34	3.0	7.1	11.1
Suction temperature,	°F	-40	51	90	135
Suction temperature,	°C	-40	10.6	32.4	57.1
Discharge pressure,	psi	43	101	158	223
Discharge pressure,	ata	3.0	7.1	11.1	15.7
Discharge temperature,	°F	-27.5	90	135	169
Discharge temperature,	°C	-2.5	32.4	57.1	75.9
Weight flow,	lbs/h	346671	653505	630886	534127
Weight flow,	kg/h	157250	296430	286170	242280
Inlet vol flow,	FP <sup>3</sup> /Min	30840	28545	13125	7475
Inlet vol flow,	actm <sup>3</sup> /h	52400	48500	22300	12700
<b>Sidestreams</b>					
Pressure,	psi	44	101	158	
Pressure,	ata	3.1	7.1	11.1	
Temperature,	°F	-4			
Temperature,	°C	-20			
Weight flow,	lbs/h	306835	22619	96759	
Weight flow	kg/h	139180	10260	43890	
Compressor speed	rpm	4100			
Compressor total	hp	15100			

Figure 3. Process Flow Diagram-Compressor.

A more economical conversion alternative might have been the installation of an electric motor with constant speed spur

gear. With regard to the process control system, the installation of additional inlet guide vanes may have been necessary on the suction side of the first, second and possibly the third compressor process stage. This additional installation of inlet guide vanes in the existing compressor housing (Figure 4) could only have been achieved under great difficulties. This would possibly have resulted in the need to modify the construction of the housing. The necessary consequences would have been that a common spare parts stock could no longer be kept for the two parallel compressor units of the two systems. As the project was under great time pressure, further investigation of this alternative was waived.

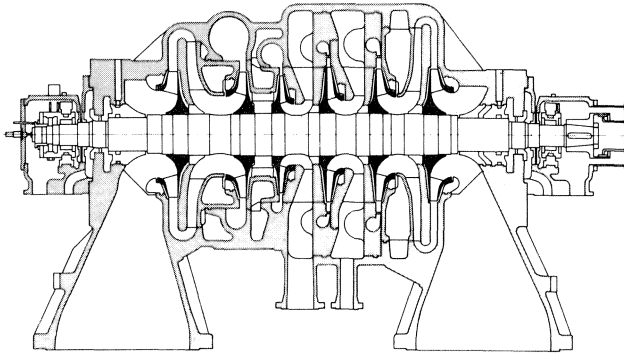


Figure 4. Sectional View of Compressor.

#### Keeping the Speed Control System

Thus only two alternatives were available that would allow keeping the existing speed control system:

- Installation of an electric variable speed motor
  - With gear unit and electric motor, rated speed  $n = 1500$  rpm
  - With direct drive without gear and rated speed  $n = 4100$  rpm = rated speed of the compressor.
- Installation of an electric motor with geared variable speed turbo coupling controlling the compressor speed within the range of 80 to 100 percent of rated speed  $n = 4100$  rpm, at constant speed of  $n = 1500$  rpm.

The electric motor with geared variable speed turbo coupling was chosen over the electric variable speed motor drive for the following reasons:

- The cost for the electric variable speed motor drive would have been twice the cost for the drive by electric motor with geared variable speed turbo coupling. Also, references were not available for such electric motor drives with the required power of 19,000 hp at speeds of  $n = 4100$  rpm.
- By selecting the electric motor with geared variable speed turbo coupling, it was possible to place all process trips of the compressor onto the coupling. This offered the distinct advantage that after a process trip the compressor could run up to operating speed immediately after elimination of the shutdown criterion (e.g., high level in the suction drum). This is possible because the drive motor keeps running, and thus, no release for switching the motor needs to be given from the electric control room.

Another advantage of this system is that the load on the power buss with a no-load startup of the drive motor (runup time approximately six seconds) is much lower, compared to a startup using an electric motor with the compressor directly connected. Controlled startup of the compressor, while always maintaining motor current under the rated current, further reduces the load on the electrical system.

- The whole concept of the plant is laid out so that the compressor operates at design speed or near design speed during normal operation. Thus, the hydraulic losses of coupling have no decisive influence.

#### CONVERSION PROCEDURE

##### *Retrofit Using the Existing Foundation*

Due to limited space within the plant area, only a conversion on the existing foundation was ever taken into consideration.

By dismantling the turbine and the surface condenser with the auxiliary units, sufficient space was made available for installation of the geared variable speed turbo coupling in the foundation recess of the old steam turbine. There were two alternatives for arrangement of the electric motors:

- Arranged on its own new foundation and coupled to the geared variable speed turbo coupling via a long input shaft,
- Arranged with its gear side motor feet on the existing foundation and extension of the existing foundation to support the motor feet at the motor end.

Due to the expected uneven settling of a new motor foundation compared with the existing compressor foundation and the necessary additional repositioning of compressor oil system, the engineers proceeded with the planning based on an "extension of existing foundation" for additional accommodation of the electric motor.

##### *Selection of Geared Variable Speed Turbo Coupling*

The geared variable speed turbo couplings type R 110 K 630-API (API American Petroleum Institute, Standards 613, 614, 615, and 670) and type R 111 KGS were the available alternatives.

These geared variable speed turbo couplings combine (in one closed housing) mechanical gear stages and hydrodynamic variable speed coupling. The variable speed coupling is a fluid coupling, transmitting the energy input by means of the kinetic energy of a fluid flow between pump impeller and turbine wheel into a closed working chamber. By positioning a scoop tube, which is arranged in radial position within the scoop chamber, accordingly, the degree of oil filling in the coupling can be varied at random during operation. Thus, the speed of the driven machine is infinitely variable over a wide range when operating over different load characteristics. The above-mentioned type R 110 K 630-API corresponds to the basic concept of the geared variable speed turbo coupling consisting of a step up gear followed by a variable speed coupling. (R-K Scheme) (Figure 5). The center distance between shafts of this size is 630 mm.

For the alternatives type R 111 KGS-API, the variable speed coupling is followed by a second gear stage for high output speeds (Figure 6). The center distance of this machine is 170 mm.

Both units originate from a series shown in the selection diagram (Figure 7). This diagram indicates the high pressure capacity as a function of the respective input or output speeds for the different coupling sizes.

The price of the R 110 K 630-API geared variable speed turbo coupling would have been about 20 percent lower. In addition, the mechanical power losses of the single-stage design were lower. However, the selection of a coupling runner with a larger working diameter, and a possible reduction of runner speed by the second gear stage, resulted in a reduction of hydraulic losses, and, thus, the total losses remained the same as those of the R 110 K 630-API.

Due to the greater center distance of the R 110 K 630-API and an offset arrangement of the electric motor with a weight of approximately 25 tons, this would have resulted in an in-

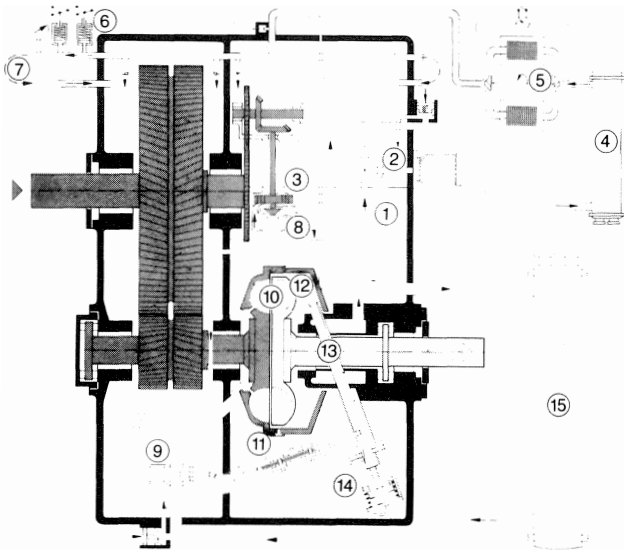


Figure 5. Oil Circuit Diagram of Geared Variable Speed Turbo Coupling Type R.K. 1) oil collecting tank; 2) auxiliary lube pump; 3) main lube pump; 4) lube oil cooler; 5) reversible double filter; 6) instruments for lube oil monitoring; 7) lube oil to other machines; 8) working oil pump; 9) oil circulation control valve; 10) working circuit; 11) fusible pump; 12) scoop tube chamber; 13) scoop tube; 14) scoop tube control; 15) working oil cooler.

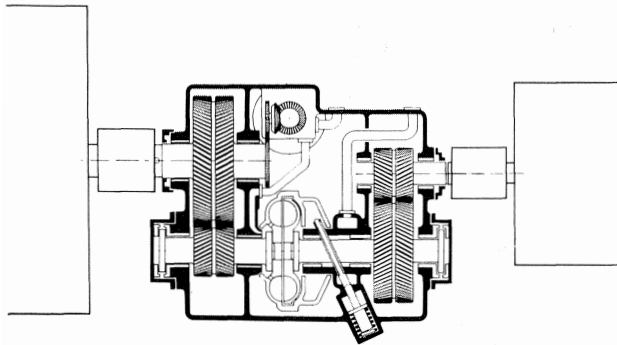


Figure 6. Simplified Sectional View of Geared Variable Speed Turbo Coupling.

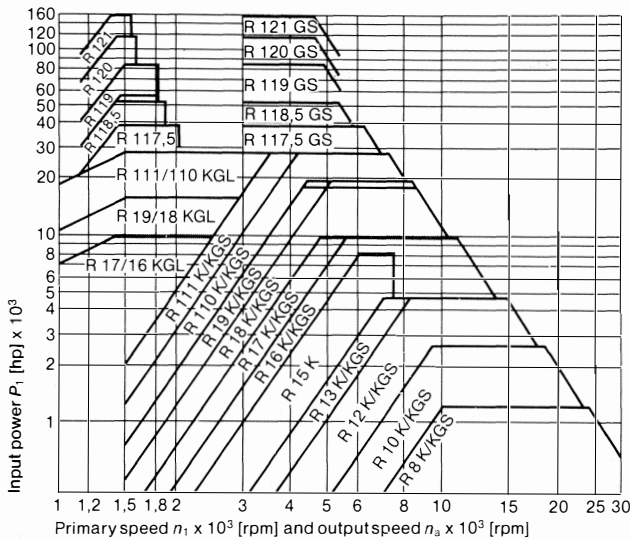


Figure 7. Performance Characteristics.

creased load for one of the two foundation supports, and, consequently, also a one-sided increased end pressure of the foundation plate. For this reason, the type R 111 KGS-API with a smaller center distance of 170 mm between input shaft and output shaft was selected.

*Requirement of a New Oil System for the Geared Variable Speed Turbo Coupling*

For the previous compressor/turbine unit, a common lube oil and seal oil system was installed which provided both the bearing oil supply for the compressor and turbine along with the control oil supply to the turbine and seal oil supply to the compressor (Figure 8). To avoid any interference in the propylene refrigerating system (lowest process temperature approximately  $-40^{\circ}\text{C}$ ) in case of loss of seal oil into the process system, a special refrigerator oil with a solidification point of  $-45^{\circ}\text{C}$  was used for the oil system. However, this refrigerator oil is unsuitable for lubrication of gears. It was therefore necessary to reinstall a separate oil system for the geared variable speed turbo coupling (Figure 9).

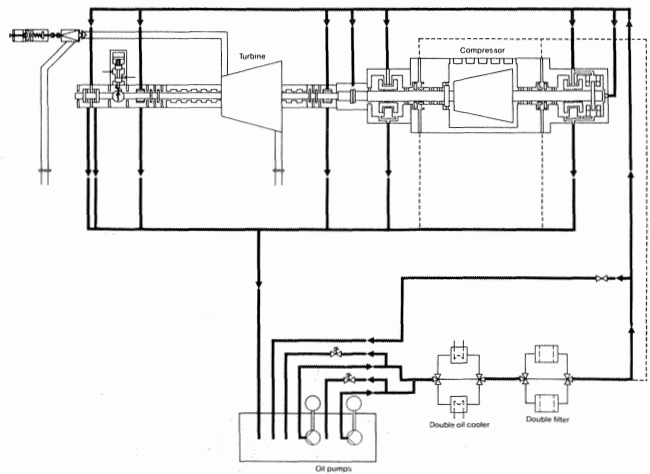


Figure 8. Lube Oil Circuit Diagram Prior to Conversion.

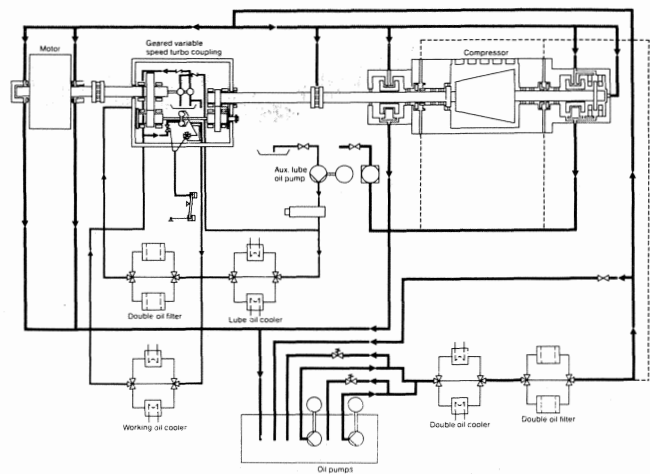


Figure 9. Lube Oil Circuit Diagram after Conversion.

The space under the geared variable speed turbo coupling was so small that it was impossible to install a package unit consisting of oil tank, 1 + 1 working oil cooler, 1 + 1 lube oil cooler, double oil filter, 1 + 1 working oil pump and 1 + 1 lube oil pump. Therefore, the selected unit was the compact design for the geared variable speed turbo coupling, with the oil tank arranged directly under the turbo coupling in the foundation recess and the working oil pump and main lube oil pump directly driven and arranged within the turbo coupling. Thus, the package unit for the oil system only consisted of 1 + 1 working oil cooler, 1 + 1 lube oil cooler, double oil filter for the lube oil and the auxiliary lube oil pump. This unit could, therefore, be accommodated under the geared variable speed turbo coupling, where previously the surface condenser of the turbine was located (Figure 10).

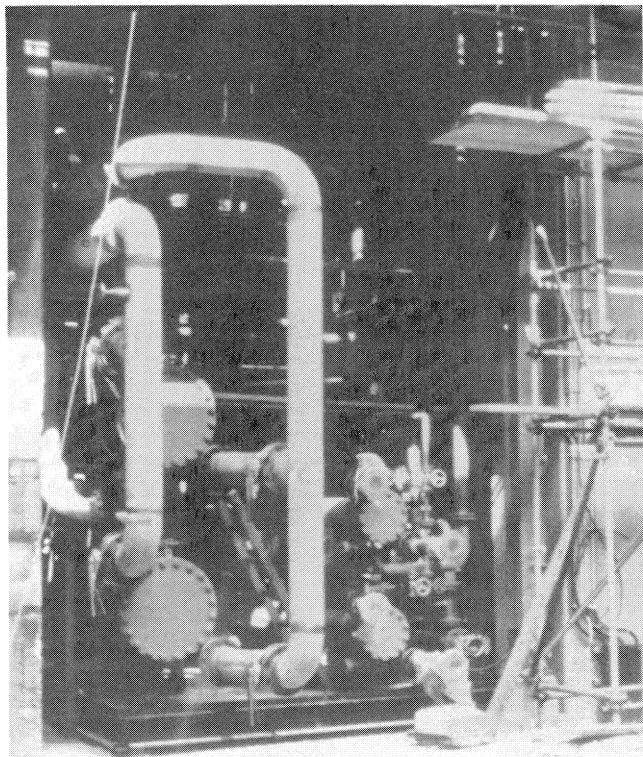


Figure 10. Cooler and Pump Unit.

*Modifications of the Compressor Oil System*

Due to elimination of the steam turbine with its relatively high lube oil requirement for bearing oil and control oil, the existing oil system also had to be modified. Since the rate of lube oil required to lubricate the bearings of the electric motor connected to the existing system was relatively small, the oil pump drive motors with a speed of  $n = 1500$  rpm were replaced by motors of the same type of construction with a speed of  $n = 1000$  rpm. These measures and replacement of the valve insert in a control valve were sufficient to have the lube oil/seal oil system for the compressor and drive motor operate stable again.

*Extension of Foundation*

Because of the longer overall length of the drive unit geared variable speed turbo coupling/electric motor compared to the steam turbine, the foundation platform had to be extended by 2050 mm (Figure 11). To put the dead weight of rear motor feet onto the foundation plate, the platform was additionally supported by two diagonal struts combined in an enclosure of the two outer foundation supports.

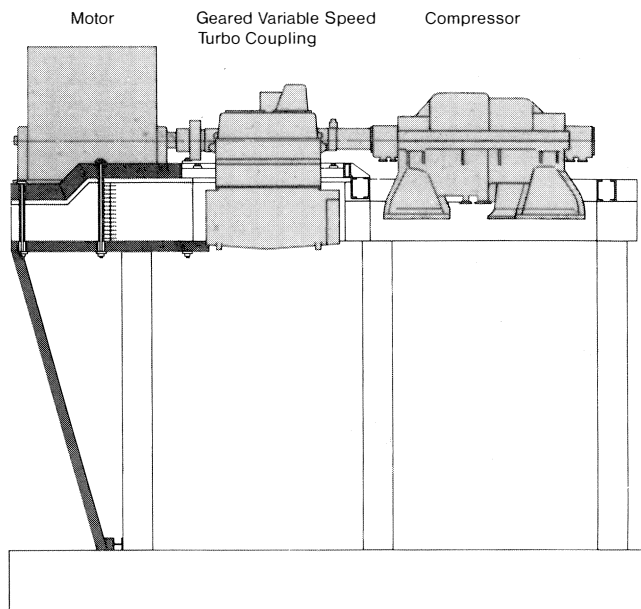


Figure 11. Design of Foundation after Conversion.

By selecting an electric motor with a center height of 800 mm, corresponding to the center height of the compressor, no modifications in the arrangement of the compressor were required.

A new and common (extra rigid) base frame of fabricated steel was made for the geared variable speed turbo coupling and the electric motor (Figure 12).

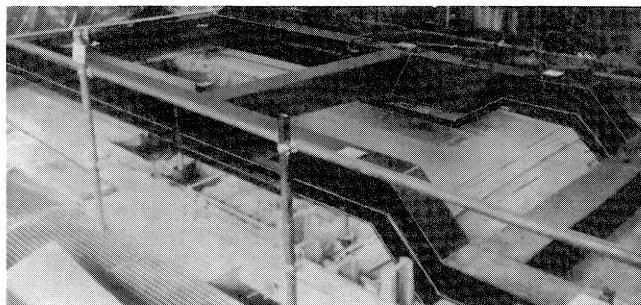


Figure 12. Base Frame of Fabricated Steel.

The existing foundation bolts of the turbine were used again for gear side anchoring on the foundation. The motor side part of the base frame was cast into and additionally connected with the extended foundation platform by using foundation bolts.

*Design of Geared Variable Speed Turbo Coupling and Working Oil/Lube Oil System*

The geared variable speed turbo coupling was designed as follows:

Power requirement of compressor	$P_a = 16\ 200$ HP
Motor speed	$n_e = 1\ 493$ RPM
Gear ratio	$= 79/31$
	$U \frac{Z_e}{Z_1}$
	$= 61/55$
	$\frac{Z_2}{Z_a}$
Full load slip	$s = 2\%$
Starting range	$= 0 - 70\%$
Operating range	$= 70-100\%$

A complete spare runner was made as a safety measure. For the design calculation and manufacture of the units, the following measures were taken:

The working oil circuit inclusive of the double heat exchanger integrated in the direct circulation system was designed according to the manufacturer's standard, since this oil system meets, specifically, the requirement of hydrodynamic power transmission without lubrication.

The rotating parts of the variable speed turbo coupling primary wheel, secondary wheel, and shell were made of special steels.

The housing split horizontally at the shaft center height was designed so that after removing the top, all bearings, coupling and gear parts were accessible.

Some details concerning runners and gear units include:

- final machining of tooth flanks took place in assembled condition.
- the pinion is designed as pinion shaft.
- both single helical as well as continuous double helical teeth are permissible per API, however, collar end bearings are not permitted. Double helical gearing was selected.
- the radial bearings are four-lobe bearings with babbit metal lining, axial forces are supported by tilting-pad bearings.
- vibration pickups with transducers are fitted on the various bearings/shafts for vibration monitoring.

A special constructional detail needs to be mentioned. During startup of the compressor, critical speed ranges are to be passed through quickly, apart from maintaining various speeds to establish steady-state conditions for the process; on the other hand a wide spread input signal range should be available for the actual control range.

This is achieved by the installation of two pneumatic power cylinders coupled in series. Both are operating with independent input signals of 0.2 to 1.0 bar (equal to 4 - 20 mA) for the range up to approximately 75 percent (compressor limiting speed) or between 75 percent and 100 percent of rated speed.

The cylinders are acting on an adjustable cam disc. This design is ideal to carry out adjustments on the running machine during trial run and continuous operation. This means that linearization between input signal and output signal was very easy as a function of the actual characteristic curve of the unit.

#### *Project Coordination*

The compressor manufacturer was responsible for the coordination of the project which involved complete engineering, including the coordination of all concerned parties.

#### *Torsional Analysis*

An analysis of torsional vibration was prepared for the complete drive train, with the result that the coupling between electric motor and geared variable speed turbo coupling had to be designed as a resilient coupling.

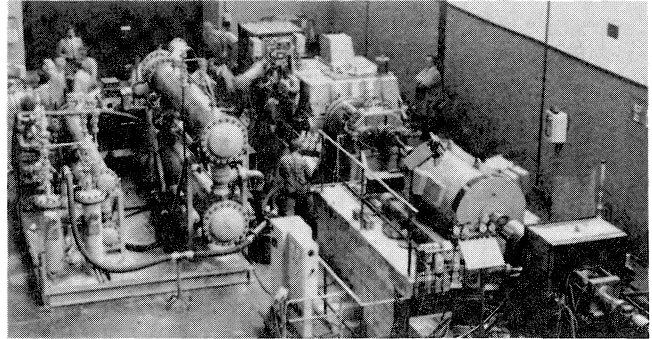
The torsional analysis showed that for the compressor/geared variable speed turbo coupling unit, the previous connecting coupling could no longer be used and had to be replaced by a high-speed gear coupling with a torque shaft.

An additional vibration analysis of the complete machine foundation system was carried out by the expert under a research contract. The result of this analysis showed that during compressor startup from zero percent to 80 percent speed, there was no danger of resonance within the complete system and that the intended range from 80 percent to 100 percent speed could be operated free of resonance.

#### *Factory Test Run*

The factory test run was performed with both the geared variable speed turbo coupling and the spare runner.

The test run was arranged as pictured in Figure 13. The test run itself was performed with the phases startup/run in, adjustment, and load operation—within a total duration of six to seven hours. The engineers measured and recorded pressures, temperatures, housing and shaft vibration, noise and power losses. All measured test results for the main and spare runner were within the guaranteed values and tolerances confirmed during the project state.



*Figure 13. Test Stand Arrangement.*

#### *Soundproofing Measures*

Soundproofing measures were planned from the beginning because of the acoustic power level of 108 dB(A) (measuring surface sound pressure level 98 dB(A)). First, the intention was to provide these in the form of an inspectable force-cooled cover. However, the sound test methods used by the expert during the factory test run allowed the change from the direct enclosure of the geared variable speed turbo coupling and the performing of soundproofing measures on the compressor housing only. The costs for the soundproofing measures taken on the compressor housing were the same as for the inspectable acoustic cover. The advantage of this solution, however, was that the geared variable speed turbo coupling was easily accessible during operation for the inspection by the supervising staff.

#### *Assembly and Commissioning*

Total retrofit including dismantling of the steam turbine with accessories had to be carried out within a scheduled shutdown period of three months. Therefore, it was necessary to prepare the equipment parts for delivery to the site, so that the time required for assembly could be reduced to a minimum. It was necessary to supply the large parts of equipment, e.g., the new oil system for the geared variable speed turbo coupling, practically ready for installation. Also the complete piping had to be prefabricated for removal of the steam turbine and incorporation of the geared variable speed turbo coupling and electric motor prior to assembly, so that adaptation and welding operations were reduced to a minimum for the conversion during shutdown.

A very extensive production planning and intensive followup was necessary in this case. Because the excellent preparations and close cooperation of all participants, delivery could be improved by one week. This time saved could be used to perform test runs with the machine and, above all, the staff could be familiarized with the new drive.

Prior to actual commissioning, several scheduled tests were performed. Thus, all parts of the system were tested one by one. First, the engineers made startup tests, with just the electric motor which confirmed the runup time of six seconds, as promised by the motor supplier. Then the engineers coupled the geared variable speed turbo coupling with the motor and

carried out no-load tests which also proved that the oil systems (working oil and lube oil) were operable.

Finally, tests were performed with the compressor connected to prove that the drive unit was sufficiently dimensioned for quick startup of the compressor, and that the power reserves of the electric motor were sufficient for the cold operating phase of the complete refrigerating system (Figure 14). All these points were proved successfully. The engineers also proved, during switching-off tests of the compressor, that the idle speed of the compressor, after switching off the coupling at  $n = 600$  to 800 rpm, was on the safe side, i.e., much below the first critical bending speed of compressor  $n = 1700$  rpm.

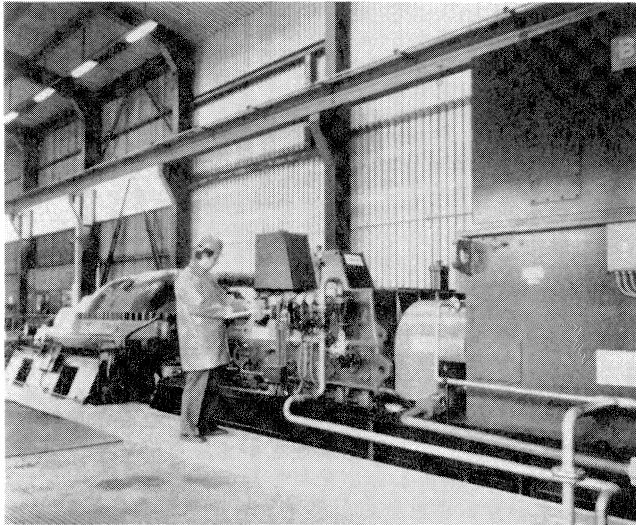


Figure 14. General View after Conversion.

Thus the band brake, installed as a precaution on the secondary side of the geared variable speed turbo coupling, could be dismantled, to avoid possible faulty connections with the machine running (the only purpose of the band brake would have been to brake the compressor, if it got too close to the first critical bending speed with its idle speed, after coupling switchoff).

Further commissioning of the compressor set under normal process conditions also proved perfect functioning of the process-oriented control system and the whole remaining electric, measuring and control system which also had to be modified for the new drive.

The vibration behavior, both during startup, and also within the working range, was absolutely perfect. All mechanical and electrical limit values were being adhered to with a safe margin.

## CONCLUSION

With a retrospective view on the handling of the project and successful recommissioning after the conversion shows that a machine of this size can be retrofitted successfully, if required due to economical or environmental conditions. It was absolutely necessary to solve all problems step by step, in cooperation with the compressor manufacturer, who was responsible for engineering and coordination, the manufacturer of the geared variable speed turbo coupling, and the operator who was in charge of structural engineering, electrical, measuring and control engineering equipment along with assembly and connection of piping. It has been found that problems which appeared to be unsolvable could be solved successfully in joint discussion.

The converted machine has been in operation since October 1985, with 12,000 operating hours as of March 1987.

During this time, the compressor unit was switched off once, due to an instrument error in the lube oil system and twice scheduled for other reasons.

