IMPROVING PLANT ECONOMY THROUGH RETROFIT AND MAINTENANCE OF COMPRESSORS

by

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ABSTRACT

Retrofits on compressors can become attractive if the operating requirements of a plant change. Often, compressor modifications are imposed by process changes due to market demands or new process technologies. Sometimes, the rising costs of energy, equipment and labor have spurred the search for ways to reduce the cost of operation of a plant.

A number of possibilities for improving economy of operation through retrofits are pointed out. All of the stated possibilities and examples have been actually carried out in one way or another. The intent of this presentation is to provide awareness for operating engineers of the possibilities for improving operating economy of their own plants.

The methods of how investigations of retrofit projects are conducted and which aspects must be considered are also shown.

INTRODUCTION

Due to basic changes in the economy, the demands on older established plants in many cases have changed over the years. The major equipment items installed in these plants were, however, originally designed and built for long service life. Among this equipment are compressors and their drivers. While the process requirements of the plants may have changed, the mechanical condition of the compressors is usually found to be suitable for many more years of reliable service.

The objective in operating older plants, therefore, is to maintain plant economy and reliability. There are three major areas of concern in accomplishing this objective:

- Upgrading of compressors to adapt them to changing requirements.
- Upgrading of compressors to reduce operating costs.
- Maintenance of compressors with a view to improving reliability and availability.

REASONS FOR RETROFIT ENGINEERING

In most cases, modifications of compressors are required because of process changes or, to a lesser degree, because of environmental regulations. The choice is to install new equipment or to adapt existing equipment to the new conditions. New compressors usually offer the advantage of better efficiency. Conversions of existing compressors, on the other hand, generally result in appreciable savings of capital investment. The two alternatives must be weighed against each other to determine the most economical solution.

On a different level, compressor conversions can be aimed at improving profitability of a plant by lowering operating expenditures. Into this category fall modifications of compressors aimed at:

- improvement of efficiency in order to save power costs,
- improvement of availability and reliability to reduce compressor downtime and costs of turnarounds and repairs,
- reduction of cost for consumables.

A survey of conversions carried out over the last two years on Mannesmann Demag compressors worldwide indicates the reasons for the modifications. They are presented below on a percentage basis:

Process modifications	40%
Energy saving (improvement of efficiency)	10%
Improvement of availability and reliability	35%
Compliance with environmental requirements	3%
Automation	10%
Others	2%

The percentage figures do not necessarily represent the complete picture, since in the case of a compressor retrofit several objectives are often combined; e.g., if a compressor is modified because of a change in process requirements, energy saving features, etc., may also be incorporated in the modification.

The survey indicates, that the majority of retrofit engineering was done to adapt compressors to process changes. This shows, that conversions are usually carried out when dictated by external circumstances. Relatively little retrofit engineering effort was expended on improvement of compressor efficiency or overall operating economy. Possibly, this is an area that has not yet received due attention.

APPLICATIONS OF RETROFIT ENGINEERING

A number of examples of compressor modifications are presented to demonstrate the effectiveness of retrofit engineering. The purpose is to point out the various possibilities of upgrading compressor equipment. Often, the modification of a compressor to adapt it to new requirements is an attractive alternative to the purchase of new equipment. There are instances where upgrading of compressors does not necessarily imply major conversions. Minor improvements of a compressor can also pay off in added benefits. On the other hand, it is sometimes economical to replace the entire compressor with a new, more efficient one, e.g., replacing single shaft compressors with integral gear compressors because of their better efficiencies.

ADAPTATION OF COMPRESSOR PERFORMANCE TO PROCESS CHANGES

The aim in changing compressor performance to new conditions is to obtain the new operating conditions with the least amount of changes and achieve the best possible efficiency. In general, changes in the plant process will demand one or more of the following changes in compressor operating conditions:

- Changes of flow volume
- Changes of suction and discharge pressures and temperatures
- Changes of molecular weight

Bearing in mind the thermodynamic relationships of compressor design, the above changes may be achieved by:

- Increasing or decreasing rotor speed. For a single shaft compressor, this may require a speed range change of the drive turbine or a change of gear ratio of the speed increaser of the drive motor. For an integral gear compressor, it may require a change of gear sets.
- Reducing the exit diameter by trimming the impeller blades. Back and cover discs remain at the original outside diameter and form a vaneless diffuser.
- Retrofit of adjustable guide vane units, particularly on integral gear compressors.
- Changing of impellers and, if necessary, diaphragms with return channels or internal volutes.
- On compressor trains with several process stages, the rotors of certain process stages may be changed to adapt the entire train to new process conditions.

Even though major modification of compressors can be costly, the overall economy of a process plant may well warrant the expenditure.

The following example underlines the effectiveness of such a modification.

Example: Capacity Increase of a Feed Gas Compressor Train in a Large Ethylene Plant.

After a period of stagnation, worldwide market demand for ethylene products has been rising again, and chemical companies involved in the production of ethylene have been considering means to increase their production. Investigation of the two alternatives of either constructing new plants or expanding product output of existing plants indicated that in many cases, a considerable increase in production was possible by conversion of the existing plants. What made conversions especially attractive was the fact that costs were relatively low in relation to product gain as compared to new plant construction.

The feasibility of plant conversion usually hinges on the possibility of increasing the capacity of the major equipment items, in particular the compressors. They are also the dominating cost factors.

Typically, three different types of compressors are used in ethylene plants:

- Feed gas compressor train
- Ethylene compressor
- Propylene compressor

The following example deals with the conversion of the feed gas compressor train in a large ethylene plant in Brazil. At the time of conversion, the compressors had been in operation for seven years. Realization of the new production goal of the ethylene plant required an increase of volume flow of at least 20 percent. Suction pressure, discharge pressure and molecular weight were to remain unchanged. The compressor is driven by a steam turbine.

Compressor Description

The feed gas compressor train consists of a low pressure, medium pressure and high pressure unit. The cross sections of the three compressor casings are shown in Figures 1, 2, and 3. The gas is compressed in five process stages.

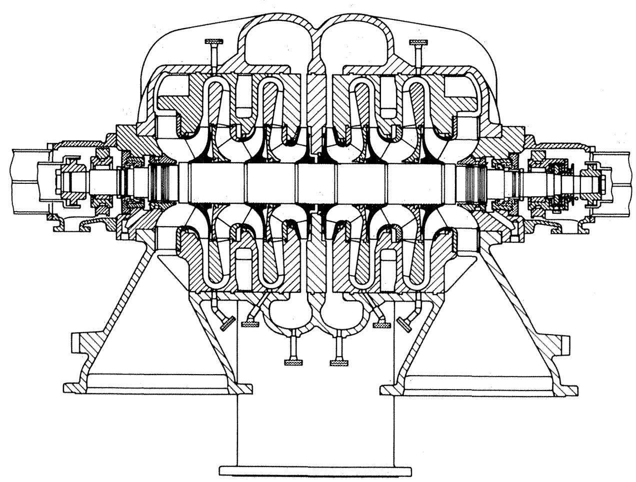

Figure 1. LP-Compressor (First Process Stage).

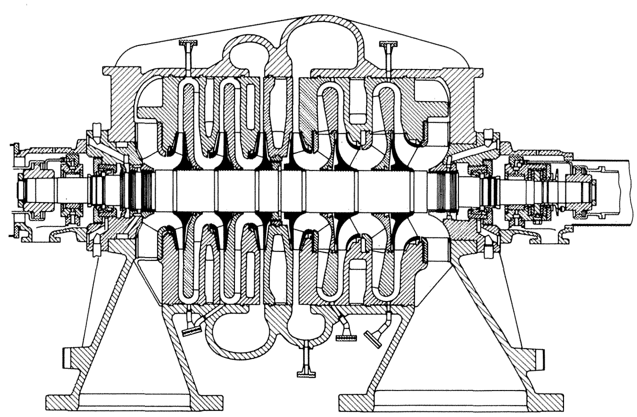

Figure 2. MP-compressor (Second and Third Process Stages).

The low pressure (LP) unit, which is the first process stage, is designed as a double flow compressor with two suction nozzles and one common discharge nozzle (Figure 1). Each flow side has three impellers which are arranged symmetrically on the shaft.

The moderate pressure (MP) unit consists of the second and third process stages, each process stage having three impellers (Figure 2).

The high pressure (HP) unit consists of the fourth and fifth process stages, each process stage having three impellers (Figure 3).

Investigation of Performance Changes

A step by step investigation of performance changes was conducted, beginning with the LP unit. The individual steps are presented in Figures 4, 5, 6, and 7. The performance curves

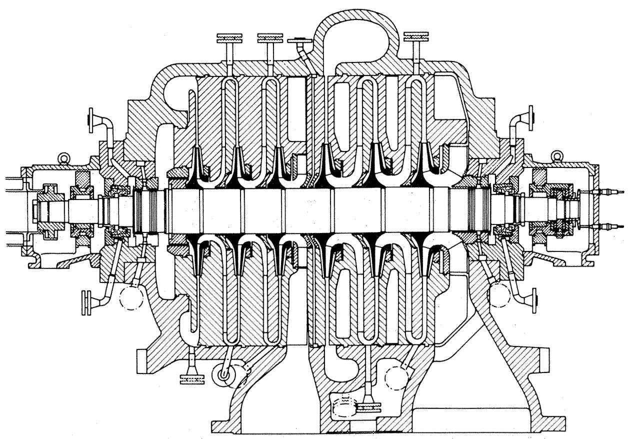

Figure 3. HP-compressor (Fourth and Fifth Process Stages).

show the original operating point and the new operating point for increased mass flow pertaining to a particular modification. The efficiency curves are also shown. Efficiency is presented as η *vs* η_{Ref} where η_{Ref} is the efficiency of the original design operating point.

Since the molecular weight, suction pressure and suction temperature remained nearly unchanged, the increase in actual suction volume is approximately equivalent to the increase in mass flow. The intermediate pressures and temperatures of all process stages are also approximately the same as original.

As a first step, the effect of an increase in operating speed without any modification of the compressor train was checked. The performance and efficiency curves for normal speed (100 percent) and increased speed (105 percent) of the first process stage are shown in Figure 4. It can be seen that the expected flow increase of 20 percent could not be obtained by a speed increase to 105 percent. Maximum volume increase would be 16 percent. The compressor would be operating very close to the choke limit and at a sharply reduced efficiency (about 90 percent of original design efficiency). The performance change due to

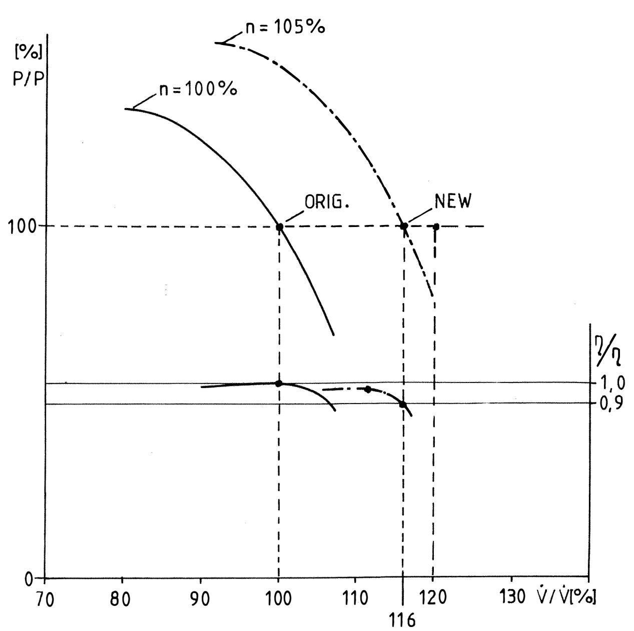

Figure 4. Performance Curve of First Process Stage (Speed Change).

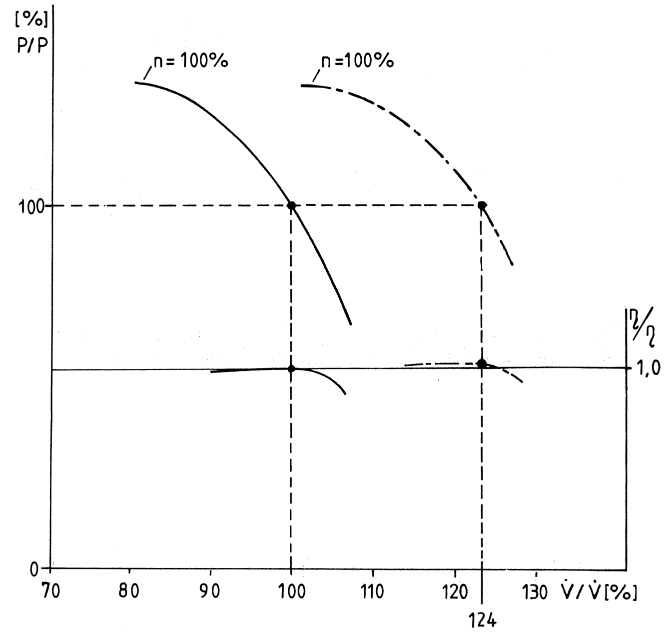

Figure 5. Performance Curve of First Process Stage after Modification.

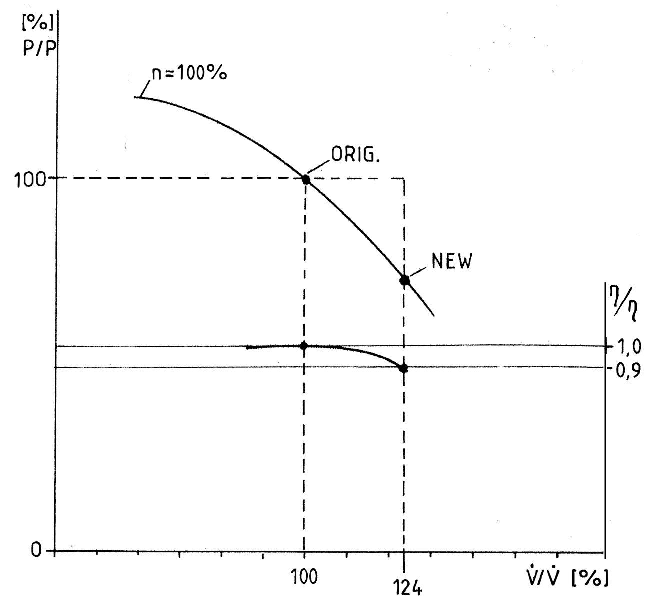

Figure 6. Performance Curve of Second Process Stage (Unmodified).

speed increase indicated for the first process stage would apply similarly to the subsequent process stages. This step of the investigation showed that increased speed alone would not produce the desired results.

The second step of performance change was based on modification of the LP unit and leaving MP and HP unit unchanged. Modification consisted of a new rotor with impellers designed for higher flow and new diaphragms with larger return passages to accommodate the higher flow.

The performance change for the modified LP compressor is reflected in Figure 5. The new operating point on the performance curve of the modified LP compressor is now located near maximum efficiency and safely away from the choke limit. The

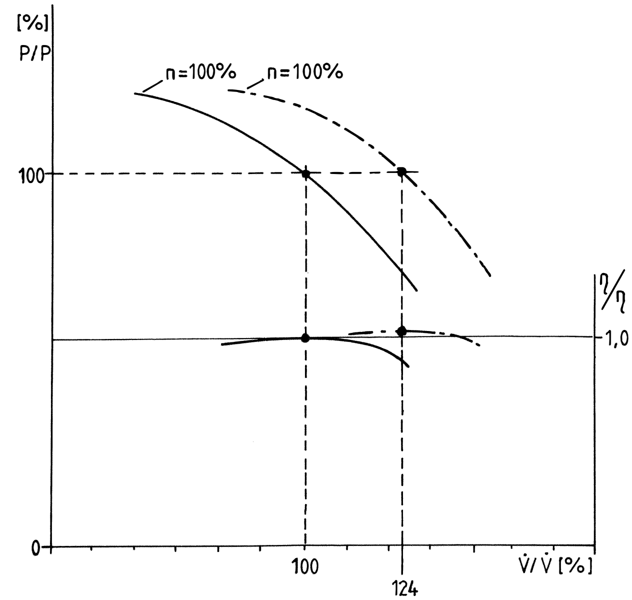

Figure 7. Performance Curve of Second Process Stage after Modification.

efficiency is slightly higher now than originally because of improved impeller design.

The resulting change of performance of the unmodified second process stage is shown in Figure 6. The new operating point would move too close to the choke limit and the efficiency dropoff would be unacceptable. The decrease in head would have to be compensated by a speed increase.

This led to a third step performance change investigation based on modification of the second process stage section of the MP compressor. By installing new impellers designed for higher flow, the new operating point on the performance curve of the modified second process stage shifted into the range of maximum efficiency and away from the choke limit (Figure 7).

The same considerations basically apply also for the third, fourth, and fifth process stages. However, the adverse effect of the performance changes described diminishes toward the higher process stages, i.e., the new operating points are further away from the choke limit and stay within the range of best efficiency. For this particular compressor train, the performances of the last three process stages were acceptable without modification.

The turbine did not have to be modified, since it had sufficient power reserve for driving the compressor at the higher load requirement. The original couplings could also be retained, even though the power input to the compressor had increased by 24.8 percent.

The complete conversion of the compressor train consisted of changing impellers and return channel diaphragms of the LP unit (first process stage) and changing impellers and return channel diaphragms of the low pressure section of the MP unit (second process stage). The new impellers were of an advanced design. Blade width and impeller inlet diameter were increased because of the larger flow. Impeller heads remained approximately the same. Efficiencies were slightly higher, due to the improved design.

The overall performance curves for the converted compressor train (solid lines) are shown in Figure 8. The performance curves based on modification of only the LP unit are also shown as dotted lines. The advantage of changing also the second process stage section of the MP unit can be clearly recognized. The actually achieved increase of volume flow was 24 percent.

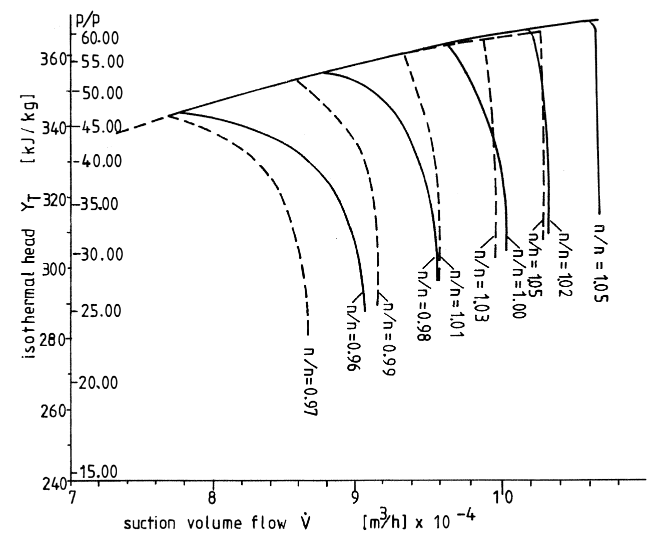

Figure 8. Overall Performance Curves of Compressor Train for Modified First Process Stage and for Modified First and Second Process Stages.

A comparison is presented in Table 1 of the original and the new operating data achieved by modification of the LP unit only and modification of LP and MP unit. For Case 2, the required coupling power is 2.5 percent lower than for case 1, which means that the speed reserve for the compressor train is 7 percent. This speed reserve is important, since the volume flow can be maintained by raising the speed, in case of fouling of the impellers, which is to be expected in time, due to the dirty feed gas. The cost of the conversion as compared to the cost of a new compressor train, based on a percentage basis, amounted to:

- 30 percent for conversion of the LP unit only (first process stage).
- 53 percent for conversion of the LP and MP unit (first and second process stages).

Table 1. Comparison of Operating Data before and after Conversion.

Operating conditions		Orig. cond.	Modification of I.Process stage (case 1)	Modification of I+II. Proc. stage (case 2)
Volume flow	[m³/h]	76558	94853	94853
Suction press.	[bar]	1,21	1,23	1,23
Suction temp.	[°C]	38	38	38
Disch. press.	[bar]	38,28	41,03	41,03
Coupling power	[kW]	13090	16744	16335
Speed	[%]	96,9	99,6	97,8

In addition, erection costs were considerably lower than for the installation of a complete new compressor train (approximately 45 percent).

Of interest is also the increase in ethylene production against the total conversion cost of the ethylene plant. The production increase was 25 percent, while the cost of conversion of the ethylene plant was only 3.4 percent of the original cost.

REDUCTION OF PLANT OPERATING COST

Plant economy requires that the cost of operating the equipment is kept to a minimum. To achieve this, the basic approaches are:

- Reducing maintenance cost by increasing service intervals
- Saving cost of energy
- Saving cost of consumables.

In order to keep compressors in good working order, proper maintenance is a necessity. However, maintenance in itself is a cost factor. Also, major maintenance requires shutdown of the compressor. This reduces the availability of the compressor, which in turn has a direct effect on plant economy. The aim, therefore, is to increase the time intervals between turnarounds.

Monitoring of Compressor Operations to Extend Service Intervals

Longer service intervals necessitate better monitoring of major functions of compressors. This can be accomplished by retrofitting older compressors with remote monitoring instrumentation. Into this category of retrofits fall:

- installation of thermocouples or RTDs in compressor bearings for temperature monitoring. Thermocouples and RTDs measure actual Babbitt temperature of the bearing pads. On older compressors, usually only the drain oil temperature of the bearings is measured. Measurement of the babbitt temperature yields more conclusive information on the condition of the bearings.
- installation of shaft vibration and shaft position monitors. Shaft vibration gives an indication of the condition of the rotor and thus, assists the operating personnel in making decisions on the necessity of an overhaul.

Shaft position monitoring indicates the axial position of the rotor with respect to the thrust bearing. Changes in rotor position are indicative of thrust bearing wear.

Liquid Injection into Impellers to Prevent Fouling

On compressors handling dirty gases or air, the major problem is often the formation of deposits in the flow passages of impellers, in return channels of diaphragms and in volute casings. Buildup of deposits in flow passages causes reduction of volume flow and a decline in efficiency. In addition, deposits in impellers can also cause severe unbalance. As a result, it can become necessary to shut the compressor down for maintenance. Severe unbalance may even cause forced shut downs which are particularly disruptive to the plant process.

The aforementioned problem has been observed mostly on feed gas compressors in chemical plants and on air compressors located in areas of heavy air pollution.

A relatively simple but effective means to eliminate or at least reduce the problem is liquid injection into the impellers. On air compressors, the injection medium is water (treated water or condensate). On feed gas compressors light hydrocarbon liquids are used. The injection systems consist of permanently installed nozzles in the suction pipes ahead of the impellers.

Washing of the flow passages by liquid injection into the impellers offers two advantages:

- Extension of service intervals.
- Maintaining compressor efficiency and flow capacity through prevention of impeller fouling.

Actually, the principle of water injection into impellers is quite old. For over 35 years, water injection has been used on nitrous gas compressors to wash off nitrates forming on the impellers and in the flow channels of the casing.

To show the effectiveness and the economy of injection systems, an actually installed system is presented.

Example: Water Injection System for a Four-Stage Integral Gear Compressor

One of the first air compressors to be equipped with a condensate injection system was a four stage integral gear compressor in a large chemical plant in Germany, designed for a capacity of 125,000 m^3/h and a discharge pressure of 7.0 bar.

In spite of good air inlet filtration, dirt deposits were forming rather rapidly on the impellers, resulting in loss of flow capacity and efficiency and, at times, high vibration due to impeller unbalance. Before the water injection was installed, the compressor had to be opened every 12 to 18 months for cleaning and rebalancing of the rotors and cleaning of the volutes.

Since installation of the water injection system, turnaround intervals have been extended to periods of four to five years. After this time, the turnarounds became necessary for reasons other than fouling of the impellers. Impellers and volutes were still found to be clean.

It is interesting to note that the cost of retrofitting the compressor with a water injection system was less than the cost of a turnaround for cleaning and balancing of the rotors.

Description of the Water Injection System

A schematic arrangement drawing of the water injection system is depicted in Figure 9. It shows the integral gear compressor, the condensate reservoir, the pump and the piping connections to the spray nozzles in the compressor suction pipes. The spray nozzle arrangement for the first stage is shown in Figure 10(a), the spray nozzle arrangement for the second, third and fourth stage suction pipes between coolers and impeller suctions is shown in Figure 10(b). The spray nozzle and nozzle holder are shown in Figure 11. The nozzle holder is inserted into the suction pipe through a flanged connection.

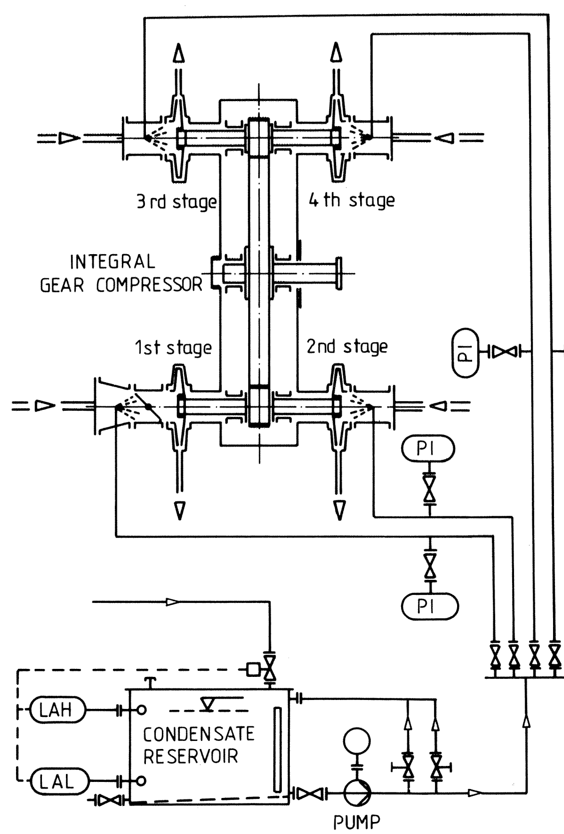

Figure 9. Schematic Arrangement of Water Injection System.

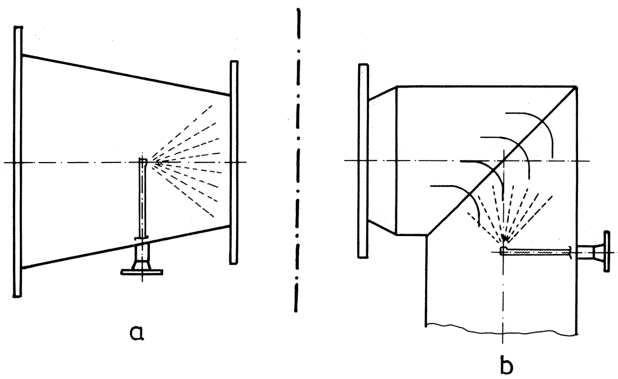

Figure 10. Spray Nozzle Arrangement in Suction Pipes.

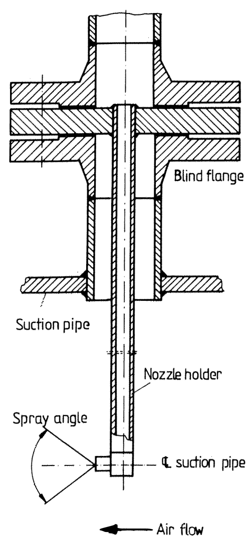

Figure 11. Spray Nozzle Mounting Arrangement.

The spray nozzles are of a type that assures a fine spray and even distribution over the entire cross section of the suction pipes to prevent erosion of the impellers.

Operation of the Water Injection System

The injection medium is condensate from a steam boiler plant. The amount of condensate injected during the washing operation is less than two percent of the air flow weight. Injection pressure is 7.0 bar above the stage suction pressure.

Only one of the spray nozzles is operated at a time. Washing of the compressor stages is conducted in a systematic sequence starting with the last stage. If the initial stages were washed first, dirt deposits on the subsequent impeller stages would come off partially, causing temporary unbalance and, as a result, high vibration. The deposits washed off the impellers are drained with the condensate from the water separator of the subsequent cooler.

During water injection, the cooler drain bypass valve is open to prevent carryover of water into the downstream impeller stages.

The duration of water injection for each stage depends on the amount of accumulated dirt deposits. Washing of the stage is continued until the condensate drained from the cooler separator stays clean. The cleaning procedure is then set forth for each preceding stage. The first stage is the last one to be washed.

Frequency and duration of water injection must be established on the basis of actual operating experience. In the case of this particular compressor, the washing operation is conducted once per shift.

As indicated previously, one of the main advantages of water injection in addition to the extension of turnaround intervals is, that the compressor will always operate at design efficiency and design flow. Both aspects contribute in this case to improved economy of plant operation.

Improvement of Thermodynamic Efficiency

Over the years, considerable research and development has been conducted on aerodynamic design of impellers to increase stage efficiencies. However, the improvements are not sufficient to justify the cost of a rotor and casing conversion to achieve better overall plant economy. If, on the other hand, a compressor must be modified to meet new process conditions, utilization of more efficient impellers becomes attractive.

For economic justification of a retrofit, the increase in efficiency must be high enough to offset the cost of conversion within a limited time. This has been actually achieved by replacing old single shaft compressors with integral gear compressors of new design. For a given compressor size, the efficiency of an integral gear compressor is inherently better than for a single shaft compressor, because of the overhung impeller design (no shaft obstructions), more than one speed, and intercooling after each stage. Differences in efficiency can be up to 20 percent. The integral gear compressor was originally designed for air service. However, it is increasingly employed for process applications, e.g., nitrogen and carbon monoxyde.

Based on today's cost of equipment *vs* cost of energy, replacement can become attractive at power savings of five percent for large compressors of about 10,000 kW, to 12 percent for small compressors of about 2,000 kW. The evaluation of energy cost is determined by the user for each particular case and depends on local electric power cost, writeoff time for equipment cost, etc.. The more existing equipment like coolers and auxiliary units can be used, the lower will be break even point for a conversion.

Curves roughly indicating the break even points for replacement of single shaft compressors with integral gear compressors based on evaluation of energy costs are presented in Figure 12. For example, based on an energy evaluation of DM 5,000– per kW, the break-even point for conversion of a compressor with a suction flow of 15,000 m^3/hr (discharge pressure 30 bar, pressure ratio 5.0 to 6.0) would be at an efficiency improvement of 9.5!

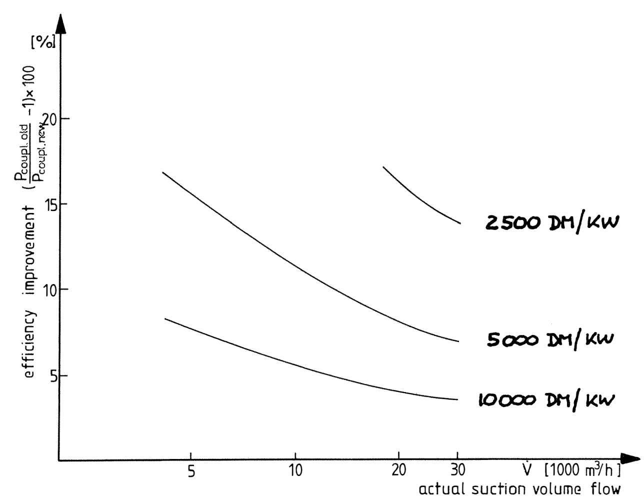

Figure 12. Retrofit of Single Shaft Compressors with Integral Gear Compressors (Evaluation of Energy Cost).

Change of Oil Seals to Eliminate Oil Consumption

Gas face seals offer an economic advantage over liquid film seals for two reasons. First, no sealing medium like seal oil is required. On floating ring seals, the oil leaking through the inner seal rings comes in contact with the gas and, depending on the type of gas, can be heavily contaminated, causing deterioration to the point where it becomes unsuitable for sealing (sour oil). This oil, which must be either discarded or regenerated, represents a considerable cost factor in the operation of a plant. Another problem with floating ring seals is contamination of the process gas with oil, leading to spoilage of the catalyst.

Second, liquid film seals have relatively high mechanical losses. In addition, the electrical power for driving the seal oil pumps adds to the mechanical losses.

For above reasons, gas face seals are more often replacing liquid film seals, particularly floating ring seals.

Another point is that the components of a seal oil system, e.g., pumps, coolers, filters, accumulators, drain traps and controllers, are subject to failure over the years, resulting in unscheduled shutdowns and increased maintenance. The costs of auxiliary power, increased maintenance and loss of oil often justify a change to gas face seals.

In cases where gas face seals are not suitable for the gas, e.g., due to tar deposits, etc., being formed on the seal parts by the leakage gas, high performance liquid face seals can be the used to replace floating ring seals. Liquid face seals will drastically reduce the inner seal oil leakage.

There has been some reluctance on the part of customers to operate liquid face seals or gas face seals since failure of a seal would cause a forced shut down of the compressor, resulting in severe production losses. Floating rings, on the other hand, usually allow continued operation after an inner seal ring failure until time is found for a planned shut down, although the seal oil leakage rates become very high.

Today's liquid face seals and gas face seals have attained an extremely good safety record, so that the possibility of a forced shutdown due to failure is very remote.

Labyrinth Seals with Abradable Lining

"Abradable seals" are a special type of labyrinth seals which are used to reduce seal clearance without risk of damage in case of contact between labyrinth teeth and abradable lining. They actually offer two advantages:

- Improved stage efficiency because of reduction of recirculation losses, in particular for low specific flow and small diameter impellers. The seals are designed for minimum operating clearance. In case of contact the labyrinth teeth will cut into the abradable lining and generate their own required clearance.
- In case of operational upsets causing temporary high vibration, the risk of rotor damage is reduced considerably for the same reasons stated above. Because of the low friction forces generated on contact between labyrinth teeth and abradable lining and the low heat flux into the shaft, thermal bending of the shaft with resultant unbalance is prevented.

Most older compressors are equipped with stationary labyrinth seals. The labyrinth teeth seal against the rotor shaft, the impeller inlet rings and the balance piston. For conversion to abradable seals, the labyrinth teeth must be machined into the sealing sections of the rotor. The stationary labyrinth seals must be replaced by sleeve type seals with abradable lining.

Example: Conversion of a 3700 kW, Eight-Stage Gas Compressor to Abradable Seals

The compressor is operating on an offshore production platform in the North Sea. The rotor had sustained damage twice during startup of the compressor after routine shutdowns. Apparently, high momentary vibration occured when liquid slugs entered the impellers, causing the rotor to rub severely against the seals. The friction heat caused a slight bow of the shaft. The stationary labyrinth seals were damaged beyond repair. All impellers were in good condition.

In spite of improvements to the inlet scrubbers, the possibility of liquid slugs entering the compressor remained. It was decided to convert the compressor to abradable seals. A stress calculation indicated that the thickness of the impeller inlet rings could be reduced for machining of labyrinth teeth. Labyrinth teeth were also machined into the sealing sections of the shaft between the impellers and into the balance piston. The original stationary labyrinth seals were replaced by new sleeve type seals with sprayed on abradable linings. The changes made to the impeller inlet rings are shown in Figure 13.

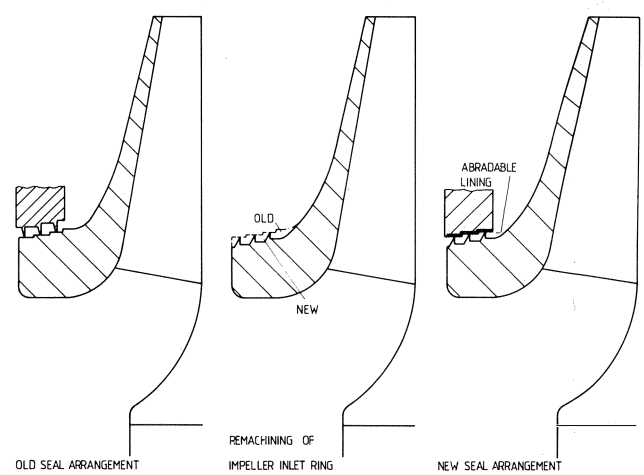

Figure 13. Abradable Seal Conversion.

The main reason for conversion of this compressor to abradable seals was to improve operational safety even under extreme operating conditions, to prevent unscheduled shutdowns which had proven to be the dominating cost factor for this offshore production facility. At the same time, the power consumption was reduced by 60 kW.

Retrofit of Inlet Guide Vanes

Many radial compressors operating at fixed speed, in particular integral gear compressors, are fitted with inlet guide vanes. The purpose of inlet guide vanes is to increase the operating range of a compressor and the efficiency at operation off the design point.

While retrofitting of single shaft compressors with adjustable inlet guide vanes is usually not possible, integral gear driven compressors lend themselves more readily to installation of guide vanes. Not only can they be retrofitted with guide vanes on the first stage, but also on all subsequent stages, since each impeller has an open suction and is mounted on the outside end of a pinion shaft. The adjustable guide vanes are accommodated between impeller suction and the suction pipe (Figure 14). The suction pipes are usually of fabricated design and can be modified at moderate cost. If several stages are equipped with guide vane units, they are operated together either by a mechanical linkage system driven by a common actuator, or by an electric control system linking the individual guide vane actuators. Due to the combined adjustment of the guide vane units, the single stage performances are also matched, allowing all impellers to operate at equivalent points on their characteristic curves (Figure 15).

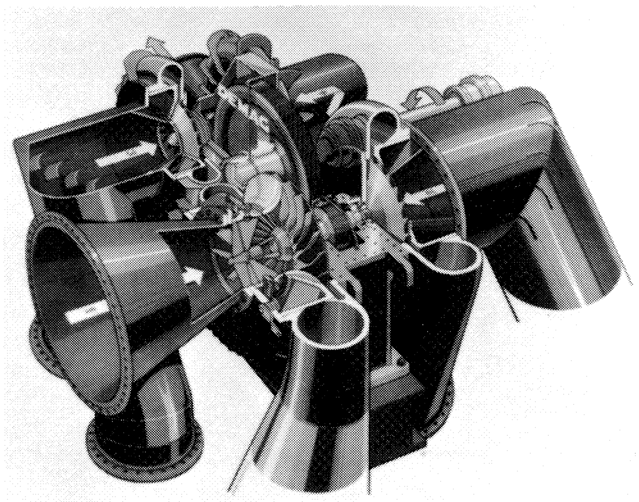

Figure 14. Four-Stage Integral Gear Compressor with Guide Vanes on First Stage.

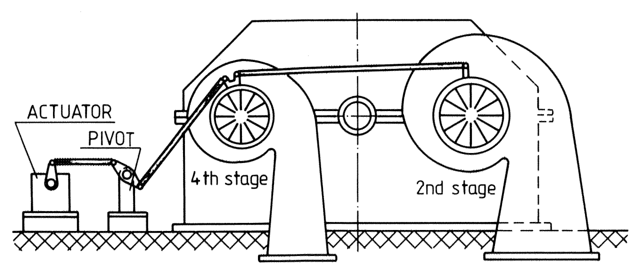

Figure 15. Mechanical Guide Vane Actuator Linkage System.

The effect of retrofitting a four-stage integral gear compressor with guidevanes on the second, third, and fourth stages is shown by the performance maps in Figures 16 and 17.

The performance map of the compressor with the original guide vanes on the first stage only is shown in Figure 17. The curves are plotted as pressure ratio vs. volume ratio. The lines of constant efficiency are plotted as efficiency ratio (actual efficiency *vs* design point efficiency). The compressor has an operating range from 60 percent to 105 percent volume at design operating pressure. The drop in efficiency as compared to design point efficiency is approximately eight percent at the surge limit and two percent at the choke limit, based on design operating pressure ratio.

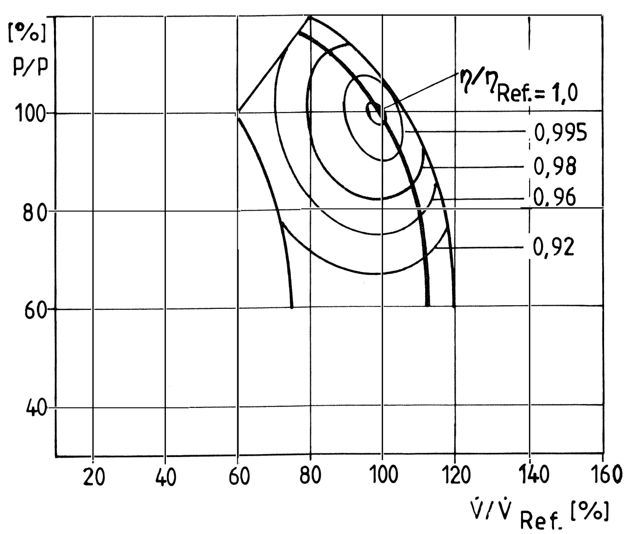

Figure 16. Performance Curves of Four-Stage Integral Gear Compressor with Inlet Guide Vanes before Stage One.

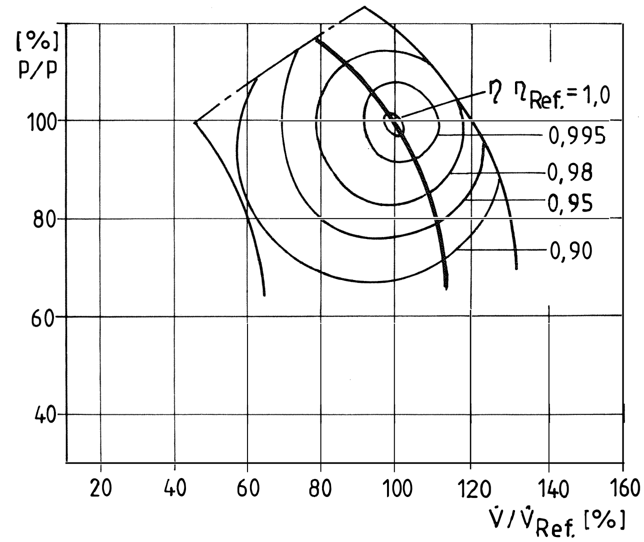

Figure 17. Performance Curves of Four-Stage Integral Gear Compressor with Inlet Guide Vanes before Each Stage.

By fitting the same compressor with additional inlet guide vanes on the second, third, and fourth stages, the performance range is considerably increased as shown in Figure 17. The operating range now extends from 45 percent to 120 percent volume at design operating pressure ratio. The drop in efficiency as compared to design point efficiency is about the same as before, but for a much wider operating range.

This type of retrofit is attractive where a wide range of operating points has to be covered during normal operation. A further advantage is the fact that efficiency does not drop off sharply at operation off design point.

Change of Main Driver

Many chemical plants have their own steam generating facilities and compressors are driven by steam turbines. Steam is often generated in waste heat exchangers. Due to changes in process and feedstock, the steam generation of the plant may be reduced. This has happened in several large ethylene plants. Because of the reduction in steam output, the choice was to either install additional steam generating equipment or to change from steam turbine drive to electric motor drive for the compressors. Especially where the capacity of the compressor was to be increased at the same time, a shortage of steam resulted.

Investigation of the alternatives showed that a change to electric motor drive was the more economical solution. Since turbine driven compressors have variable speed control, it is usually desirable to retain the speed control system. This necessitates either a variable speed motor or a motor with a variable speed gear, e.g., a fluid drive turbo coupling.

An example of conversion from steam turbine drive to electric motor drive with a variable speed turbo coupling was reported during the Sixteenth Turbomachinery Symposium [1]. It concerned the conversion of a feed gas compressor train with a 19,000 hp steam turbine to electric motor drive with a variable speed gear in a large ethylene plant in Germany. Here, too, the alternatives were to either increase the steam generation capacity of the power plant or to change to electric motor drive. In this case, because of the increased steam generating capacity requirement, more severe environmental regulations would have applied to the power plant, adding further to the cost of conversion. Evaluation of the overall cost favored a change to electric motor drive.

Waste Heat Recovery

In most plants, the thermal energy generated by compressors is wasted in the cooling water systems. Especially in small and medium sized plants, more and more attention is given to energy saving by recovery of the hot water energy from the coolers for process or heating purposes. Heat energy from compressor coolers can be reclaimed in different ways.

If the temperature difference between cooling water inlet and outlet is sufficiently high, the cooling water of the coolers may be used directly for heat recovery. However, high cooling water exit temperatures are contrary to the demands of a compressor for high efficiency. High recooling temperatures increase the power requirement at the compressor coupling.

A better solution for heat recovery is to add waste heat exchangers to the existing coolers on the hot gas side. This means that waste heat exchangers and coolers work in series. The waste heat exchangers utilize only the upper range of the compressor discharge temperature, while the rest of the cooling is accomplished by the original coolers. By adding waste heat exchangers to the existing coolers, the recooling temperatures can be lowered, resulting in improvement of compressor efficiency.

A diagram of a waste heat exchanger system for a packaged air compressor with a coupling power of 2,100 kW and a flow of 21,500 m^3/h is shown in Figure 18. The heat recovery from the three intercoolers and the aftercooler amounted to more than 1,300 kW and was used for heating purposes.

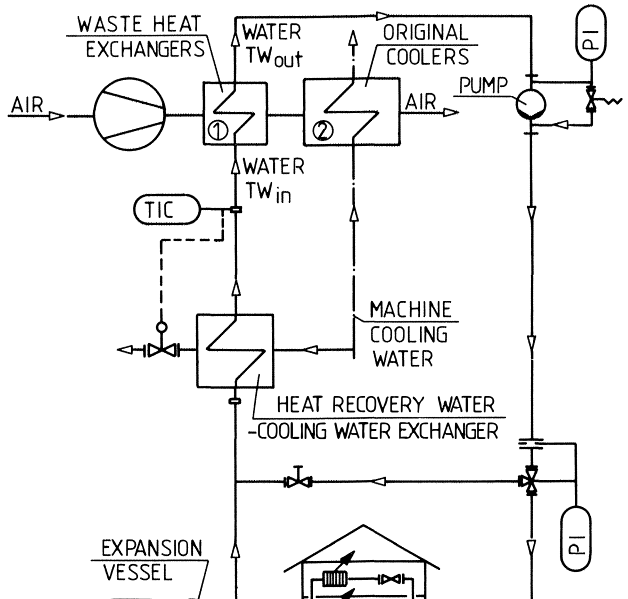

Figure 18. Waste Heat Recovery System.

TECHNICAL AND COMMERCIAL ASPECTS OF RETROFIT ENGINEERING

Standard retrofits and modifications on a smaller scale require little engineering investigation. The technical feasibility of the conversion is determined and a quotation for the necessary hardware is submitted.

Retrofit projects on a larger scale must be investigated on their own merits. Rarely can it be decided from the outset that a major modification of a compressor is possible and will yield the expected results. In these cases, a detailed engineering study is necessary to determine the technical feasibility of the project and the extent of the required modifications. The study also establishes the extent of required parts, shop work and field work and serves as a basis for developing the total cost of the proposed project.

By weighing the total cost of the conversion against the expected benefits, the customer can then make a decision on the profitability of the contemplated modification.

Indepth engineering studies can become quite involved. To avoid unnecessary costs, extensive investigations are conducted only where chances for a successful conversion are promising. Normal procedure, therefore, is to conduct a preliminary study to estimate the feasibility of a particular retrofit project.

If the proposed retrofit project appears feasible, a detailed engineering study is performed against an engineering charge. The actual engineering and drawing work is conducted in conjunction with the respective specialized engineering departments, e.g., thermodynamics, mechanical engineering, etc. Project management remains with the retrofit engineering department throughout the entire engineering, manufacturing and field erection process.

If at any time during the course of the engineering study it should turn out that the proposed retrofit project cannot be realized, only the cost of engineering up to this point is charged.

Obviously, rerating and upgrading of compressor equipment requires an indepth understanding of compressor technology, e.g., aerodynamics, thermodynamics, rotordynamics, etc., and the metallurgy of the materials used. Furthermore, when making changes to a compressor to adapt it to new conditions, the original engineering records and drawings are indispensable. Since this knowledge is an essential prerequisite for successful retrofitting and upgrading of compressors, modifications should be carried out in consultation with or under the responsibility of the compressor manufacturer.

IMPROVEMENT OF PLANT ECONOMY THROUGH MAINTENANCE

An important factor affecting plant economy is maintenance of compressors. Maintenance should not just consist of keeping the equipment operating and doing the necessary repairs if something breaks down. Increasing weight is placed today on preventive maintenance. The aim is to eliminate conditions which could adversly affect compressor operation. Preventive maintenance also means to recognize any beginning malfunctioning of a compressor at an early stage, and to take remedial action before it deteriorates to the point where major repairs and unscheduled shutdowns become necessary.

Advanced planning in maintenance requires good data collection of compressor functions, because no preventive maintenance action can be taken if the condition of the compressor is not known. Data collection and proper interpretation are the main tools for planning of inspections and turnarounds.

Maintenance directly affects the reliability of equipment and, as a result, its operational availability. High operational availability of compressors improves overall plant economy and more than offsets the higher costs of good maintenance programs, the necessary diagnostic tools and equipment modifications designed for preventive maintenance.

Several examples of compressor modifications aimed at improvement of preventive maintenance were presented herein.

For instance, to allow better prediction of the condition of a compressor, it is necessary to install the required monitoring equipment. Reference was made to retrofit of RTD's or thermocouples in bearings to monitor actual babbitt temperatures. This temperature data is more accurate than measurement of bearing oil drain temperature and yields continuous and accurate information on actual bearing condition.

It was also shown that many older compressors are operating without shaft vibration and shaft position monitoring systems. In some instances, casing vibration is measured periodically with portable equipment, but the information gained on the vibrational behavior of the compressors is not sufficient to guide maintenance decisions. Beginning problems are not always noticed in time to prevent major damage.

Older compressors can readily be retrofitted with shaft vibration and shaft position monitors at moderate cost. Continuous monitoring of shaft vibration and shaft position improves safety of operation, aids in maintenance planning and saves on cost of repairs and downtime.

Another problem affecting compressor operation is fouling of the impellers due to dirty gas or air. The problem can sometimes be alleviated by improvement of the filter or scrubber systems ahead of the compressor. Often, however, nothing can be done on the process side to eliminate or reduce the problem. In cases where adverse operating conditions exist, the possibility of water injection into the impellers could be investigated.

From the examples presented, it can be seen that improvement of plant maintenance often goes hand in hand with retrofit engineering.

Another area of interest for maintenance engineers is the continued application of new technologies to existing compressors. Repairs and replacement parts are normally regarded simply as means to bring the compressor back to its original condition. However, compressor manufacturers are constantly working on improving the design and the materials of their compressor products. The resulting improvements are not limited to new compressors. One of the benefits in the development of new compressor designs is to apply the resulting advanced technologies also to existing compressors when it comes to repairs or the supply of spare parts. This refers to design as well as materials.

As a result of this policy, the mechanical reliability and efficiency of a compressor does not need to decline as time goes on, but it can even improve with time under a farsighted maintenance program.

CONCLUSIONS

The foregoing examples of retrofit applications and maintenance considerations demonstrate that older compressors do not necessarily become uneconomical or unreliable with time. In many instances, retrofitting of older compressors is the preferred choice over purchasing of new equipment, because the high capital cost for new compressors often offsets the advantage of higher efficiencies.

Efficient plant economy is not only a function of compressor efficiency, but also of operational reliability and availability. Therefore, continued efforts should be made to maintain older compressors in good condition and to improve their reliability and availability by applying latest technologies whenever possible and reasonable.

REFERENCE

1. Henschel, F. K., and Rappold, W., "Conversion of a 19,000 HP Propylene Compressor Drive from Steam Turbine to Electric Motor with Geared Variable Speed Turbo Coupling," Proceedings of the Sixteenth Turbomachinery Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1987).