REVAMPING THE AIR COMPRESSOR IN AN AMMONIA PLANT

by

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ABSTRACT

Revamping the compression equipment in an ammonia plant can help optimize plant output and improve overall efficiency. In this case, proper sizing of the revamp is important for optimum results.

The reformer air compressor is a primary revamp candidate as shown by the revamp of the air compressor train in an 1150 ton/day (TPD) ammonia plant in Sterlington, Louisana. This example illustrates the concerns which should be considered when planning such a revamp. The results demonstrate the success of the revamp approach.

INTRODUCTION

It is helpful to understand the background of the equipment in a Kellogg-type ammonia plant in order to appreciate the significance of the air compressor revamp. In the early 1960s, as single-stream ammonia plants grew in size to 600 TPD, centrifugal compressors began to replace reciprocating compressors in the three primary compression services: reformer air, synthesis gas, and ammonia refrigeration. By the mid-1960s, ammonia plants and their compression equipment were standardized in 1000 TPD, and later 1150 TPD sizes. During this period, emphasis was placed on the production of duplicate plants and, since fuel prices were generally cheap everywhere, efficiency was not a primary factor.

Machinery performance optimization was less important than reliability and shipment schedules, and more than 70 plants were built around the world with essentially duplicate compression equipment. During a three year period in the mid-1970s, for example, equipment for more than 10 plants per year was built (each plant requiring six compressors, four steam turbines and two gears for the three basic compression services).

In more recent years, increased energy costs and changing market demands have made optimization of production rates a prime factor. In new plants, significant process changes and more modern compression equipment have resulted in overall efficiency gains. In existing plants, the emphasis is often on increasing production by "debottlenecking" the process stream, and this usually results in a need for more air to the reformer.

Ammonia Process

A simplified process schematic diagram for a typical ammonia plant is presented in Figure 1. In this type of plant the amount of air required is stoichiometrically related to the amount of feed gas, and is thus directly proportional to the plant production rate. The air train is typically a two-body compressor with a total of four sections of compression, each separated by an intercooler, as shown in Figure 2. In the standard train arrangement, the low pressure (LP) compressor is driven by a condensing steam turbine, and the high pressure (HP) compressor is driven through a speed increasing gear coupled to the LP machine. A typical installation is shown in Figure 3.

Original Designs

The original 1960s vintage internals of a standard air train installed in lower half LP and HP casings during assembly are shown in Figure 4. The 1150 TPD LP aerodynamic design consisted of five impellers of approximately 32 in diameter. The two impellers in Section 1 were open wheels and the three in Section 2 were closed. All seven impellers in the HP body were conventional closed wheels (approximately 18 in diameter). All impellers were produced from precision castings using either 17-4 ph or 15-5 ph material, in order to ensure good

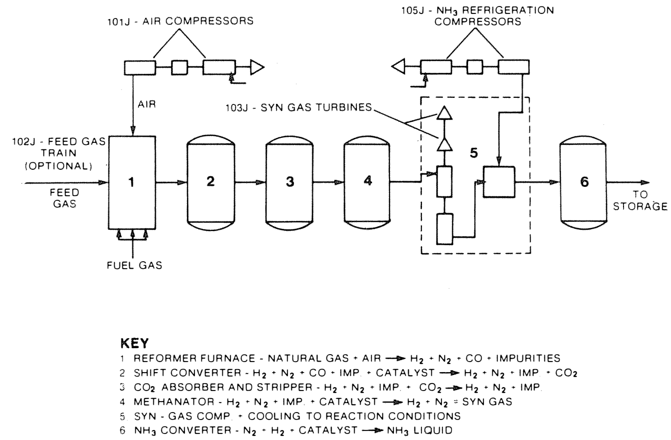

Figure 1. Typical Ammonia Plant.

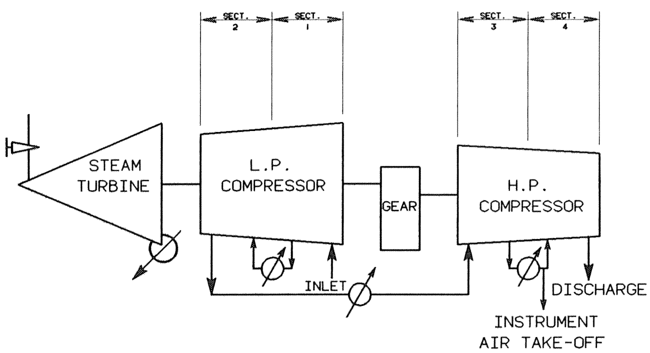

Figure 2. Air Train Arrangement.

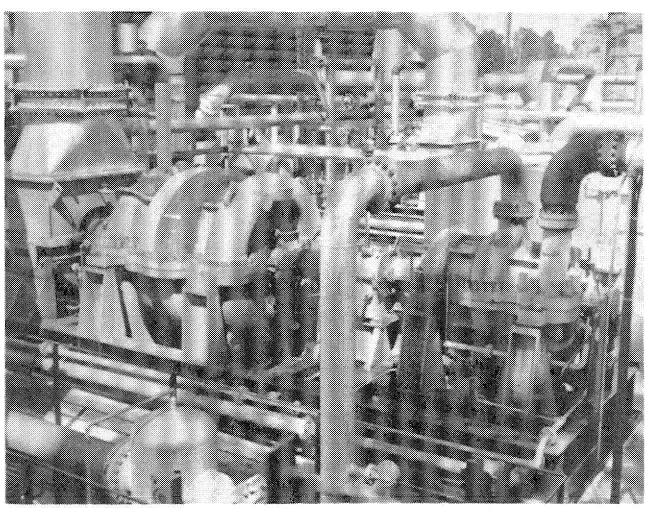

Figure 3. Air Train Installation.

quality castings and facilitate stocking of impellers. In the 1000 TPD configuration, four wheels, the first open, were used in the LP body. The HP body was the same for either 1000 or 1150 TPD.

When these machines were first designed, the open wheels used in Section 1 represented the highest flow capacity commonly available in a multistage compressor of that size. Wheel construction had not advanced so that these wheels would commonly be made enclosed, so some leakage loss had to be allowed at the shroud of the wheel, due to the clearance needed between rotating and stationary parts.

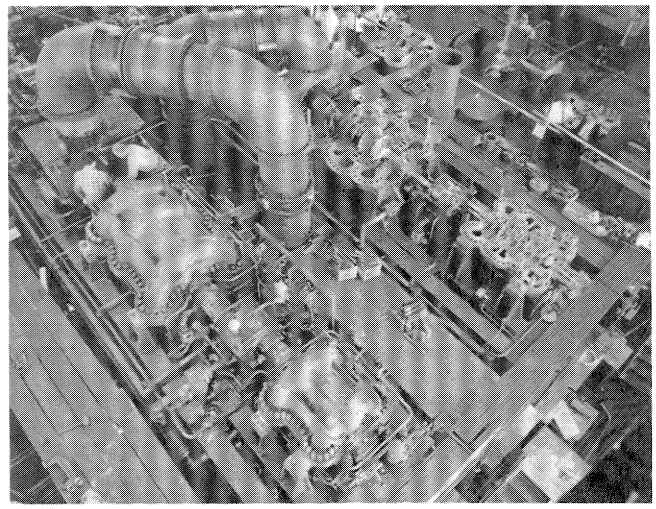

Figure 4. Original Air Compressor Internals.

Since all the wheels involved in these machines were castings, it was desirable to minimize the number of different wheel designs in order to reduce the number of patterns required. To get the widest application range from each, removable guide vanes were used ahead of the wheel in each stage. The guide vane shown in Figure 5 feeds into the second stage open wheel. These guide vanes were available in several standard vane angle settings from 20 degrees prerotation to 20 degrees counter rotation. By changing the angle of flow entering the wheel, the inlet velocity relationships were affected, causing a change to the stage characteristic curve (counter rotation shifts the curve towards higher flow and head, while prerotation has the opposite effect). By selecting the optimum vane wheel combination, a limited number of different wheels could be used (for example, the same wheel might be used in successive stages, with counter rotation ahead of the first wheel and prerotation ahead of the next wheel). However, there was a shortcoming to this approach—any vane setting other than neutral (i.e., either, prerotation or counter rotation) would result in a reduction in efficiency.

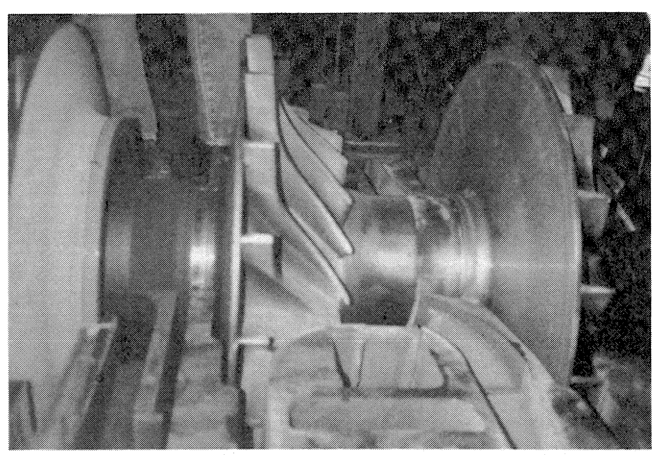

Figure 5. Removable Guide Vane-Original LP Internals.

Stationary aero designs consisted of vaneless diffusers and vaned return passages which fed into the inlet guide vanes of succeeding stages. Seals at the impeller shroud and behind the impeller were typically either aluminum or babbitt, and clearances were large to prevent damage from rubs. This was also true of the balance drum.

When first applied, these machines represented state-of-the-art compressor design. However, as time went on, various advances in compressor technology were developed that could be used to improve performance. At the same time, production rates were increased at many ammonia plants, causing the air train to be run off-design at higher flow rates, with corresponding increases in power and speed. These higher flows caused efficiency to suffer, and some plants became "air-limited." Compounded by increasing energy costs, it became apparent that improvements to the air compressors should be studied.

REVAMP CONCEPT

After studying the possibilities for retrofitting the air trains, two complimentary ideas became evident: first, that newer aero designs could offer significant efficiency improvements; and, second, that these revamp designs should be aimed at the higher flow rates desired by the ammonia plant operators.

Design Improvements

The new aerodynamic concepts had been developed over the preceding decade and are standard designs in modern compressors. The most basic improvement is in the impeller design, specifically the three dimensional (3-D) blade design illustrated in Figure 6. Primarily, the term "3-D" applies to the inlet or inducer portion of the impeller blade. From the inlet velocity triangle it is apparent that, moving along the blade entrance from hub to tip, the radius and, therefore, tangential velocity (U_1) is increasing. Also, as the flow exits from the previous stage and turns into the impeller eye, the velocity profile is not constant. The resultant relative inlet velocity vector (W_1) varies from hub to tip (shroud), and the inlet angle (β_1) increases. To avoid incidence losses, the physical 3-D blade angle is varied to match the flow angle for optimum performance. The discharge portion of the blade is backward-leaning to give good range and curve shape. The 3-D wheel design efficiently handles larger flows in an enclosed wheel than were previously feasible in an open wheel.

The use of 3-D impellers has been made practical largely through the advent of modern, tape-controlled three-dimensional milling machines, and the development of sophisticated welding methods. Prior to these manufacturing innovations, such complex blades could only be obtained from precision castings, which historically have been more difficult to produce from a QC standpoint than fabrications.

By using fabricated rather than cast wheels, it becomes more convenient to have a greater number of available wheels from which to choose. Consequently, the impeller line presently in use has over 30 flow contours available in each of several standard wheel diameters. This allows selection of every wheel to be within roughly five percent of its design point and eliminates the need for pre or counter rotation guide vanes upstream of each stage. The stationary aero path still includes vaneless diffusers and vaned return passages, which now feed into each succeeding stage at a neutral angle.

Another significant aerodynamic improvement in the revamp design is the use of abradable labyrinth seals, as shown in Figures 7 and 8. In this design, the labyrinth teeth are on the rotating element and are allowed to cut into the stationary abradable seal material, resulting effectively in a staggered labyrinth seal, which reduces leakage losses and controls leak-

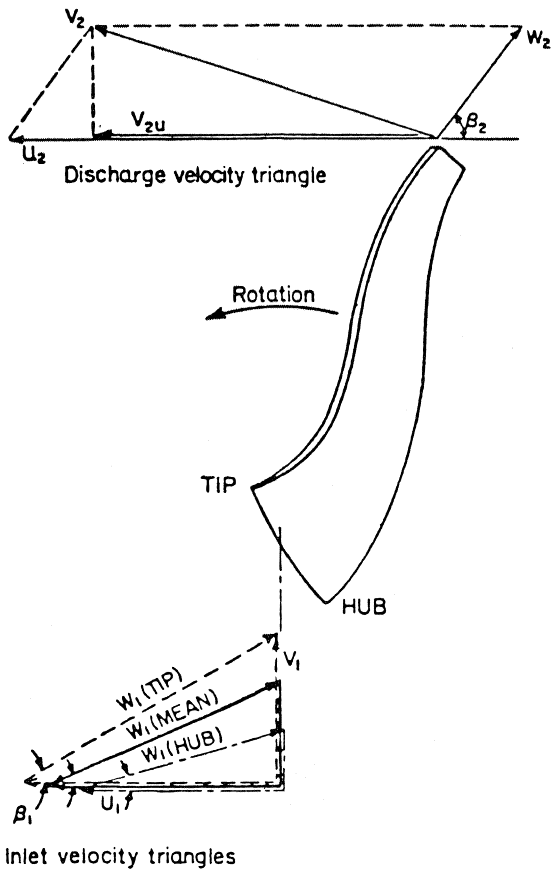

Figure 6. 3-D Impeller Blade.

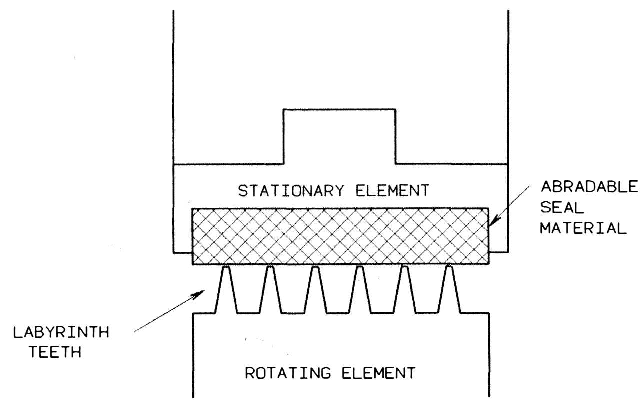

Figure 7. Abradable Labyrinth Seal.

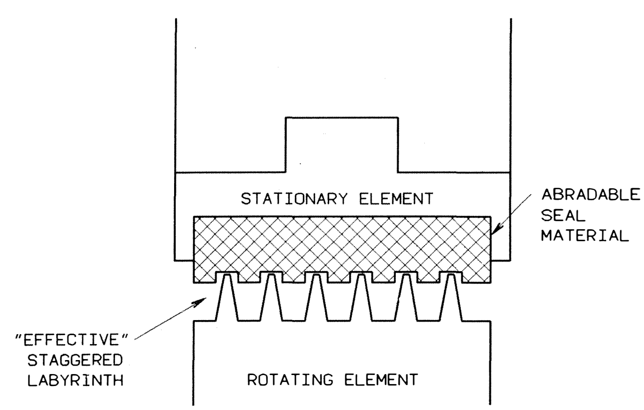

Figure 8. Operating Configuration of Abradable Seal.

age rates for extended periods. This seal design is used for stage seals and balance drum seals in all applications, and is often used for end seals where labyrinth seals are applicable. The abradable seal material consists of a synthetic mica-filled TFE fluorocarbon. The addition of mica improves its performance characteristics over pure TFE. The Fluorosint® is held in place by a patented design steel ring which allows the assembly to be installed and removed like any conventional metal labyrinth.

The resulting revamp internal designs developed for the existing air compressor casings are shown in Figures 9 (LP) and 10 (HP). The LP design consists of two 32.5 in wheels in Section 1, and three 27.25 in wheels in Section 2. The Section 2 wheels are reduced in diameter to increase their specific speed and thereby improve performance. The HP body utilizes eight 16.25 in wheels, four in each section. All wheels in both bodies are the 3-D design and are fabricated (welded). The first wheel in each section is made of a material such as 17-4 ph to offer increased erosion protection, since the air is often moisture laden at the machine inlet and after leaving the intercoolers. The remaining wheels are made of a quenched and tempered alloy steel (similar to ASTM-A543).

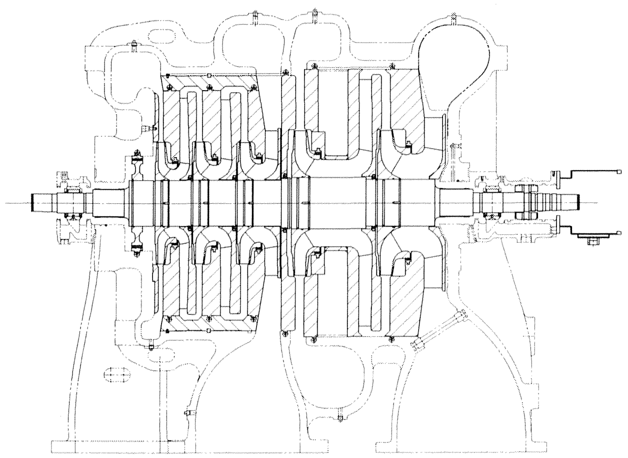

Figure 9. LP Compressor Revamp Design.

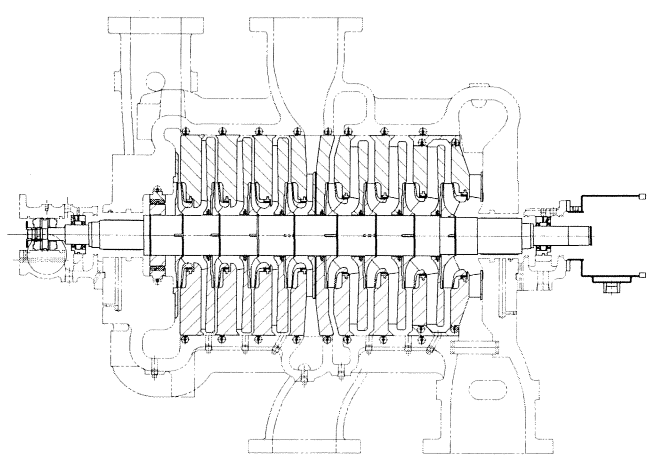

Figure 10. HP Compressor Revamp Design.

New parts consist of new rotors and all stationary flow parts in both bodies. The new stationary parts in the second section of the LP compressor are shown in Figure 11. Both rotors have slightly larger shaft diameters under the impellers with the revamped design than with the original machines, giving nominally improved rotordynamics.

The revamp is designed to be accomplished in the field, with no case machining required for 1150 TPD size trains. Field machining of a diaphragm groove is necessary in the 1000 TPD size LP case. The conversion is reversible, in that the original internals can be reinstalled in the cases if necessary (as spares, for example).

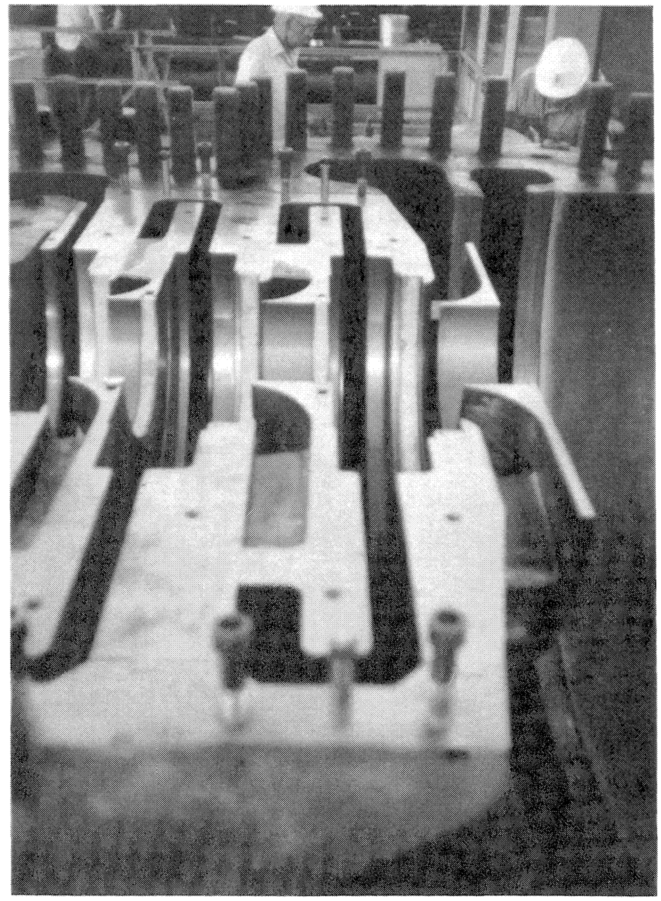

Figure 11. Revamp Stationary Internals—LP Compressor.

Revamp Sizing

In addition to improving the aerodynamics, the second part of the revamp concept for the air trains is to design the new internals for the flows actually needed for the desired production rate. This approach is taken to optimize the energy savings gained with the revamp, as illustrated in Figure 12, by eliminating the efficiency penalty incurred by operating the original machinery "off-design." In most instances, this approach results in a revamp sized for a significant increase in flow above the original design point. Taking advantage of the 3-D wheel design, the revamped compressors can handle an increase in inlet flow of more than 40 percent, which corresponds approximately with the casing (nozzle) flow limitations. The actual air flow requirement chosen as the new design point governs the impeller selection for the revamp.

To show the importance of optimizing the revamp design, power comparisons are given in Table 1 for three typical air flow requirements. At 120,000 lb/hr air flow, the optimum revamp design is essentially equal in capacity to the original 1150 TPD design. At 132,000 lb/hr there is only a 0.4 percent extra saving by optimizing the design rather than using the 1150 TPD revamp. However, at 145,000 lb/hr, the optimum revamp yields an additional savings of 2.7 percent over the savings obtained with an 1150 TPD revamp design. This 2.7

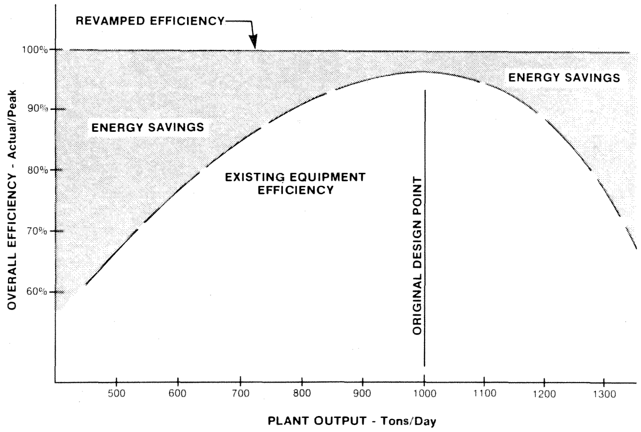

Figure 12. Potential Energy Savings.

percent represents the penalty which would result from running an 1150 TPD revamp design at a flow rate significantly above its design point.

Table 1. Power Savings Comparison (typical conditions).

Air Flow (lb/hr)	120000	132000	145000
Original 1000 TPD			
Power Required (hp)	10800	12100	—
Original 1150 TPD			
Power Required (hp)	10650	11800	13320
Savings over Original 1000 TPD (%)	1.4	2.5	—
Revamp 1150 TPD			
Power Required (hp)	10200	11300	12750
Savings over Original 1000 TPD (%)	5.6	6.6	—
Savings over Original 1150 TPD (%)	4.2	4.2	4.2
Revamp Optimum			
Power Required (hp)	10200	11250	12400
Savings over Original 1000 TPD (%)	5.6	7.0	—
Savings over Original 1150 TPD (%)	4.2	4.6	6.9
Savings over Revamp 1150 TPD (%)	—	0.4	2.7

Turbine Considerations

Once the compressor flow and power requirements have been established for a revamp, the driver must be considered. The air train turbine will often have the capability to handle the extra power requirements of the revamped compressors. If not, a turbine uprate consisting of a new nozzle ring, diaphragms and some new blades can usually raise the capability of the turbine sufficiently. In any case, the condition of the turbine should be evaluated and the turbine refurbished, if required, to obtain optimum performance.

In order to properly analyze the potential gains with an air compressor revamp, it is important to understand the effect of the revamp on the overall plant steam balance, as shown in Figure 13. A change to any of the compression equipment is ultimately reflected in a change to the high-pressure steam flow, and often additional optimization is possible by applying the revamp principles of improved aerodynamics and proper design point selection to the synthesis gas turbine (this is particularly true where syngas compression requirements have changed due to modifications to the synthesis process).

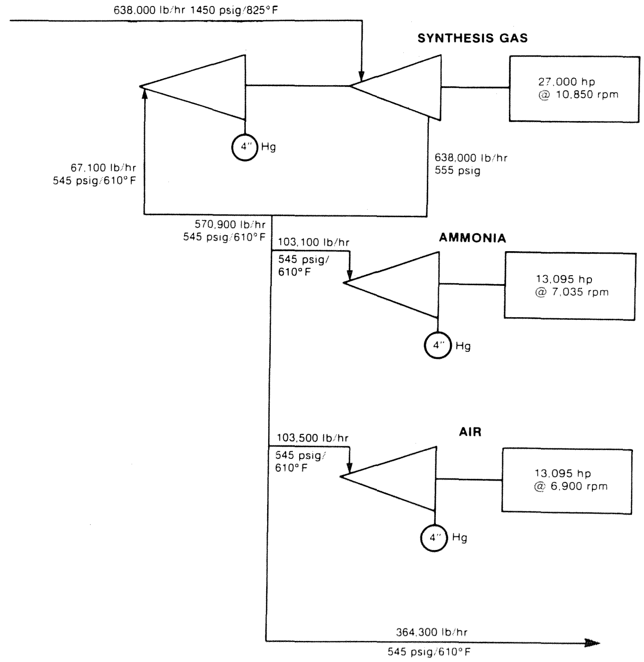

Figure 13. Typical Ammonia Plant Steam Balance.

IMC REVAMP

A prime example of the air compressor revamp concept is the revamp done in 1985 to the air train in IMC's 1150 TPD plant at Sterlington, Louisiana, to alleviate an "air-limited" production situation. A comparison of original and revamp design conditions is shown in Table 2. The 160,000 lb/hr revamp design flow represents a 35 percent increase over the original design flow. The revamped internals are as shown in Figures 9 and 10.

Table 2. Design Conditions.

		Original	*Revamp*
Plant Rate	(TPD)	1150	1590
Flow	(lb/hr)	118000	160000
Power	(hp)	10350	13350
Inlet Pressure	(psia)	14.3	
Inlet Temperature	(°F)	95	
Relative Humidity	(%)	90	
Intercooler Temp.	(°F)	105	
Intercooler ΔP's	psi	1,3,5	
Discharge Pressure	(psia)	520	

In addition to the aerodynamic changes made to the compressor internals, several other items should be considered when making a capacity increase of this magnitude. These items include all components of the equipment train (driver, couplings, etc.), along with items in the process flow path (air piping, intercoolers, etc.). This revamp gives several illustrations of this point.

Air Piping

A 35 percent air flow increase results in significantly higher velocities in the air piping, which in turn causes higher pressure drops and ultimately can add to the consumed power. After studying their air piping, the company made two changes. The 30 in diameter pipe between the inlet filter and the LP compressor inlet was replaced with a rectangular duct equivalent to a 42 in diameter pipe. The duct was used instead of round pipe, to avoid several potential interference areas. The inlet filter was already adequately sized and had a 42" outlet. The ductwork was designed to fasten horizontally under the compressor deck in an arrangement that featured mitered bends in a manner less restrictive than the original inlet pipe.

Intercoolers

Typically, the 1150 TPD size intercoolers should be adequate for the increased air flow of a compressor revamp, and, in this case, this proved true. However, care should be taken to ensure that the coolers are in good condition and functioning properly. During operation prior to the revamp, the company discovered that the trap from the second intercooler was plugged. When maintenance was performed on the cooler, it was found that the demisters were coming apart, resulting in small pieces of stainless steel wire plugging the cooler. New demisters were installed during the revamp. The intercoolers were and are cleaned and hydro-tested during each annual turn around, and continue to perform well.

If operating data indicates either excessive cooler pressure drop or inadequate heat transfer, maintenance or replacement is suggested, since these deficiencies have a significant impact on overall performance. For the company revamp design condition a 5°F increase on all three reentry temperatures adds roughly 100 hp to the total power requirement, while one psi extra pressure drop on all three coolers costs about 200 hp.

Couplings

For a range of modest capacity increase, the existing couplings were probably adequate. However, for a 35 percent increase, some of the couplings appeared marginal. Consideration was also given to the company's wish to change couplings for maintenance-related reasons. A decision was made, therefore, to replace the original lubricated, gear type couplings with dry, diaphragm couplings. This necessitated replacing the original coupling guards also. The existing gear was used without modification.

Installation & Operation

The revamp internals were installed in mid-1985. Because some of the internal parts could not be checked in a final assembled condition until they were installed in the machine, some fit problems were encountered. To alleviate this on future revamps, inspection fixtures were made (Figure 14) to simulate the compressor cases and allow for assembly checks in the factory.

Since initial startup, the revamped air train has run continuously for approximately two years, with one annual maintenance turnaround and only brief outages. One of the most important factors contributing to the high reliability of the equipment is the sophisticated monitoring system which the company uses to continuously evaluate the operational "health" of the rotating machinery.

Results

Overall performance curves are shown in Figure 15 for both the original and revamped compressor train. The maximum flow previously obtained with the original machinery is plotted along with a typical actual operating point for the revamp. This data is also compared in Table 3. The overall result is clear—a substantial improvement in both delivered air flow and compression efficiency. The ultimate proof lies in the ability to meet desired production rates.

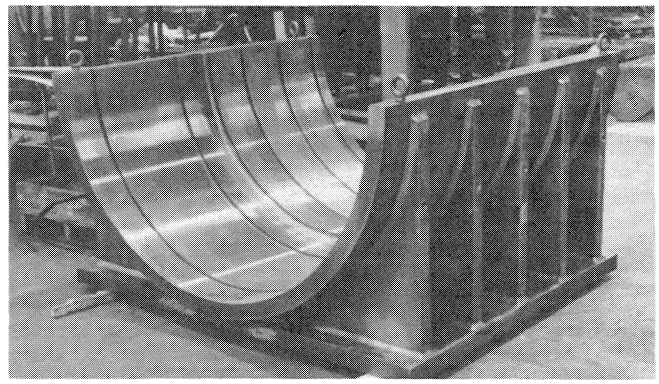

Figure 14. LP Case Inspection Fixture.

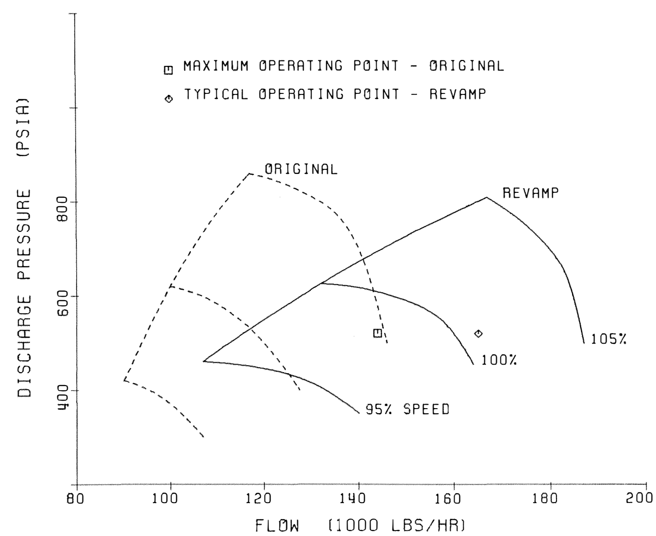

Figure 15. Overall Performance Curves.

Table 3. Actual Operation (typical operating conditions).

	Original	*Revamp*
Plant Rate (TPD)	1490	1600
Flow (lb/hr)	144000	165000
Average Sectional Polytropic Efficiency (%)	74	78

CONCLUSION

By using modern compressor aerodynamic designs applied to the desired flow rates, the air compressor in an ammonia plant can be revamped to help optimize plant output and improve overall efficiency. The revamp of the 1150 TPD air train provides a fine example of this concept. The key to the success of this type of revamp is the selection of the proper revamp design point, through the determination of the actual air flow needed to match the desired production rate.