

RETROFITTING A TOPPING TURBINE FOR PERFORMANCE AND MECHANICAL IMPROVEMENT

by

Vincent S. Wagner, Jr.

Manager, Product Support Services

Transamerica Delaval, Incorporated

Trenton, New Jersey



Mr. Vincent Wagner, Jr., received his B.S. degree in Mechanical Engineering from Drexel University in Philadelphia, Pennsylvania.

He has been with Transamerica Delaval, Inc., Trenton, New Jersey, for the past thirteen years during which he has been assigned to a variety of duties in the Engineering, Manufacturing and Marketing areas. Mr. Wagner is currently the Manager of Product Support Services,

which includes the NH₃ plant activity for Transamerica Delaval. He is a member of ASME, current President of the Engineers Club of Trenton, founded in 1914, and a registered professional engineer in the State of New Jersey.

ABSTRACT

Current design technology applied to existing rotating equipment can produce significant energy and maintenance dollar savings.

The results achieved when a two-stage topping turbine was retrofitted to a three-stage configuration are presented. This unit is used to drive a synthesis gas compressor train in a 1000 ton/day Kellogg ammonia plant. The topping turbine retrofitting was conducted by Transamerica Delaval for Canadian Industries, Limited in their Sarnia, Ontario plant.

The presentation is broken into two parts: the first part examines the modifications made for performance improvements, and the second part examines the modifications made for improved mechanical performance. Data are presented comparing the rotor response predictions of both design and the actual field test results.

INTRODUCTION

The synthesis gas train in a typical 1000 ton/day Kellogg ammonia plant consists of a four-body train: a low and high pressure compressor driven by a two-body steam turbine tandem. The helper turbine, which is the outboard turbine, is a condensing unit rated variously from 4500 hp to 7500 hp, depending on the installation. Steam conditions are typically 545 psig and 610°F condensing to 4 in Hg. The topping turbine, which is the double-ended drive through turbine, is a back pressure unit rated variously from 15000 hp to 20000 hp, depending on the actual installation. Steam conditions are typically 1450 psig and 825°F let down to 555 psig. When coupled together, these units will produce up to a total of 27500 hp and run at a speed of 10850 cpm in the maximum configuration. The synthetic gas train arrangement is shown in Figure 1.

This combination of parameters presented some very unique design requirements back in the early 1960s, and over the past 20 to 25 years there have been several selected modifications proposed and implemented to provide a unit running as smooth-

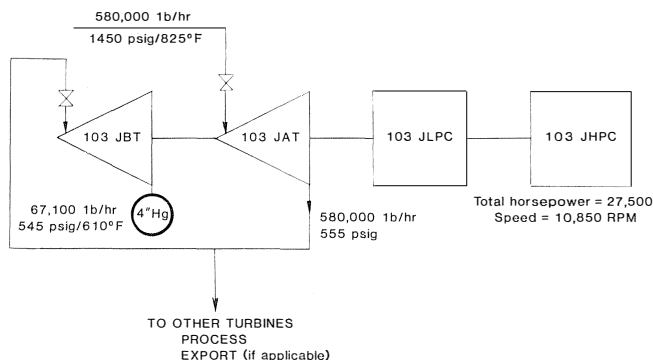


Figure 1. Typical Synthesis Gas Train Arrangement.

ly and trouble free as possible. Now, however, the designers were encouraged to start with an empty casing and incorporate all the lessons learned over those many years and many millions of operating hours.

The replacement of the present two-stage rotor and internals with a three-stage assembly is a result of that effort. A cross section view of the updated design is shown in Figure 2.

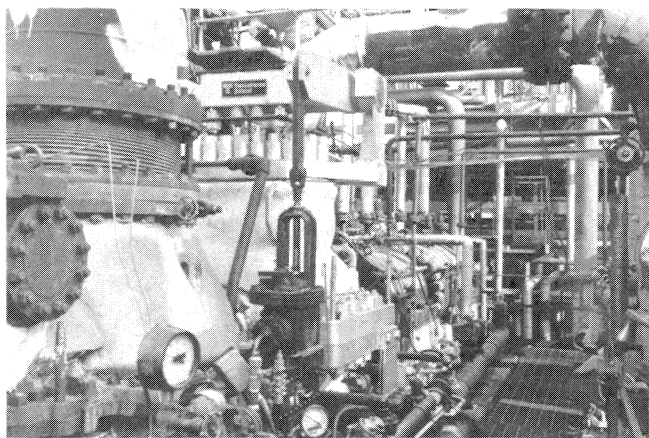
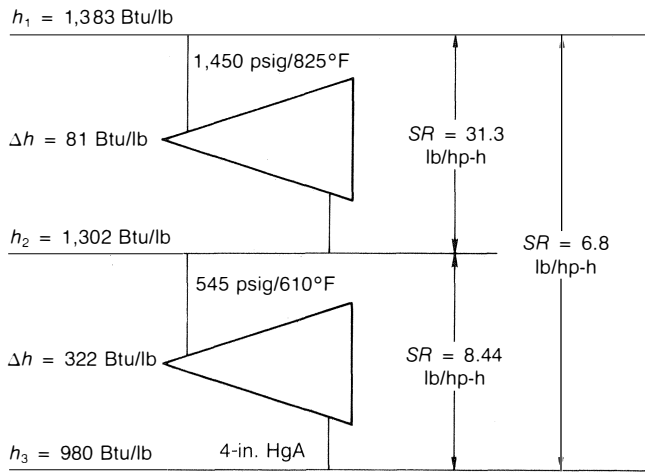


Figure 2. Photograph of the Synthetic Gas Train.

A look at the overall steam flowchart in a typical Kellogg NH₃ plant shows the crucial fact that all of the high pressure steam is "let down" through the topping turbine. Additionally, the topping turbine converts just about 20 percent of the available steam energy into mechanical power in these plants. Thus, its importance from an energy consumption standpoint cannot be overlooked, and certainly encourages the performance improvements undertaken in this program (Figure 3).

The other area which was given very serious review was that of mechanical design and rotor stability. In rotating equipment of this size and class, based on experience, the empirical balancing "break point" is in the area of 7500 cpm to 8000 cpm. Above that speed, it is good practice to utilize an operating speed balance whenever possible. It is particularly important to



From ASME Steam Tables: theoretical steam rate (TSR) = quantity of steam required to produce a unit amount of work in an ideal engine or turbine, and it may be expressed in pounds per kilowatt hour.

- From 1,450 psig/825°F to 545 psig: TSR = 31.457 lb/kw-h; assuming an isentropic efficiency of 75% and converting to lb/hp-h yields an actual steam rate (SR) of 31.3 lb/hp-h
- From 545 psig/610°F to 4-in. HgA: TSR = 8.484 lb/kw-h; again assuming an isentropic efficiency of 75% and converting yields a SR = 8.44 lb/hp-h
- From 1,450 psig/825°F to 4-in. HgA: TSR = 6.835 lb/kw-h; again assuming 75% isentropic efficiency and converting yields a SR = 6.8 lb/hp-h

Δh or h_1-h_2 , h_2-h_3 , h_1-h_3 all represent the Rankine cycle work, or the isentropic expansion energy across the engine or turbine.

Figure 3. Energy Savings Evaluation.

review the rotor response and critical speed analytics, and it is well to keep in mind that a smaller margin of forgiveness exists. Generally higher rotating speeds go hand in hand with lighter rotors and greater sensitivity in the area of rotor dynamics.

The topping turbine, which, as originally configured, has a two-stage rotor, is an example of a very light rotor operating at quite high rotative speeds, some as high as 11000 cpm. Some of the mechanical improvements which have been incorporated over the years include a reduced overhung coupling on the exhaust end, a change from a 4 in to 5 in journal bearing on the inlet end, and other various modifications.

IMPROVEMENTS MADE FOR PERFORMANCE IMPROVEMENTS

Explanation

There are several areas of improvement which were addressed in the attempt to maximize the overall efficiency of the topping turbine. These include:

- The addition of one stage, for a total of three. The extra stage, with a slight increase in reaction and velocity ratio, improves the efficiency and increases the reheat effect.
- The addition of radial seals to control leakage losses in the steam path.
- The use of a full arc of blading and axial entry blading. Rather than using the existing radial entry blading, axial entry blades along with a full row of blades are being used, thus eliminating the losses for missing blades now found in the steam turbine.
- Reduce the valve pressure losses by the removal of the center pivoted valve in the valve gear.

It is worth spending a few minutes discussing some of the pitfalls encountered in maximizing any turbine performance, including, most importantly, the topping turbine retrofit sizing.

There is a tendency to select a much larger steam flow margin than will ever be required during operation. Especially in these mature plants, there should be a very good idea of the maximum power and steam flow necessary in actual operation.

The following rules of thumb should be recognized:

- The unit should be sized with a modest margin above the normal operating point.
- There is about a three percent loss of efficiency for each ten percent excess horsepower capability.
- Remember that additional horsepower is available when required.

A cross sectional drawing appears in Figure 4.

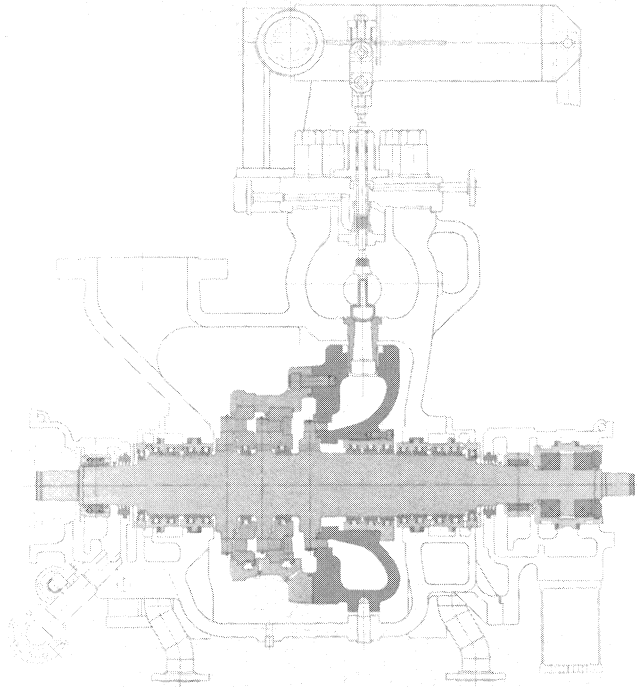


Figure 4. Cross Section Three-Stage Topping Turbine.

A listing of the parts which must be replaced includes:

- turbine rotor
- inner steam chest
- inner case
- five sets of piston rings
- five nuts for the piston rings
- both journal assemblies
- thrust bearing housing
- second and third stage diaphragms
- inner chest box
- two inner case boxes (innermost)
- two inner case boxes (intermediate)
- two inner case boxes (outermost)
- one thrust bearing complete
- center valve and all five valve seats.

Additionally, there are several other areas which must be reviewed. These include the casing and valve gear cover studs, which should be converted to heated studs, the valve gear ratio,

the servo-motor travel, and perhaps other remedial repairs which are necessary due to long term wear and tear.

The overall objective of this type of rebuild is to return the turbine as closely as possible to the original condition. Proper bracket and casing alignments must be tended to prior to reassembly. Joint facing flatness and proper clearances and concentricities must be measured, charted and brought within specifications.

Attention to all of these details helps ensure that the proper design clearances are realized. The fact that improved mechanical operation also results is an important side benefit. The point to be emphasized is that by simply installing parts of a more efficient design without restoring the casing and other major parts to original design specifications, will very likely result in a shortfall over the expected performance improvement.

ASME PTC 6 Field Test Agenda and Instrumentation

In order to accurately measure the results of the retrofit, it was absolutely essential that a field performance test be accomplished with the very minimum of test uncertainty. The following test agenda was developed with that end in mind.

PROPOSED INSTRUMENTATION FOR FIELD TEST

Note: Refer to ASME PTC 6 Guidelines (Figure 5).

- Based on a ten inch pipe to the trip and throttle valve steam flow orifice, the beta ratio shall be 0.7 and the pressure taps for the differential pressures (Δp) shall be in accordance with ASME installation guidelines. Two separate measurement orifice taps shall be provided and measured. A sufficient straight length of pipe containing flow straighteners shall be provided to meet code requirements. Differential pressure measurements shall be made using two calibrated and traceable differential pressure (ΔP) cells of the appropriate range (Figure 6).

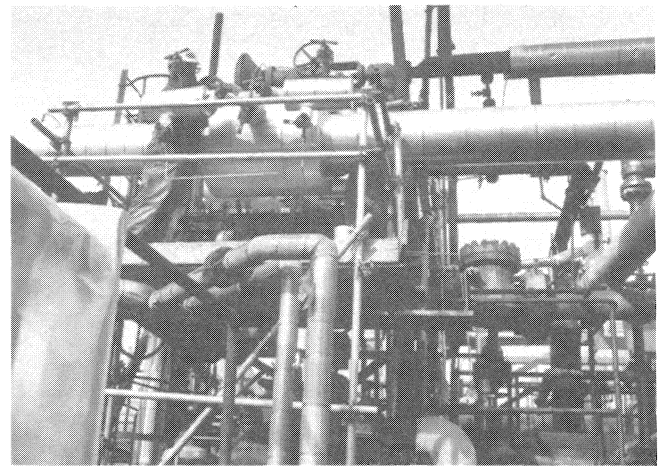


Figure 6. Installation of the Flow Measuring Orifice.

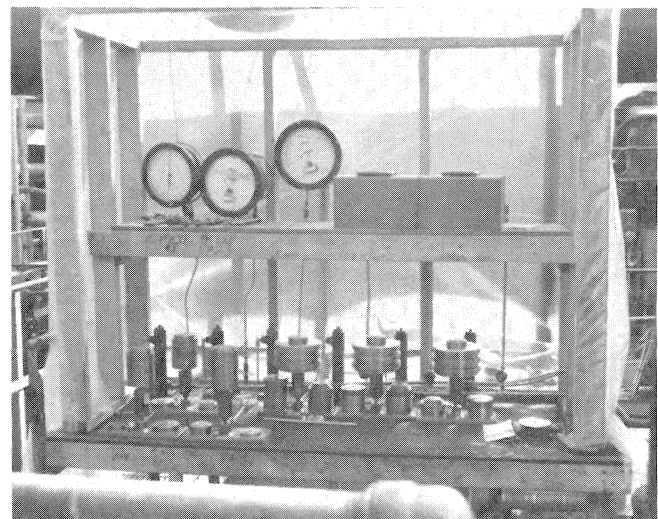


Figure 7. Installation of the Dead Weight Gages for Pressure Measurement.

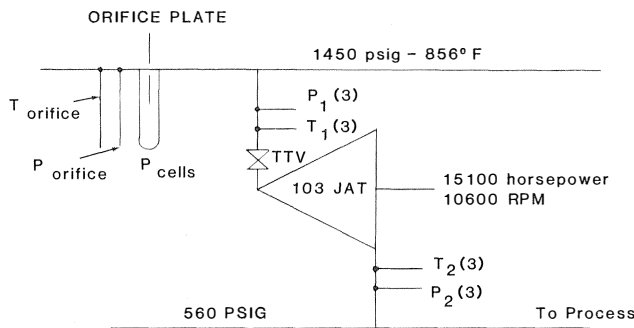


Figure 5. Proposed Field Performance Test Set Up.

- The pressure measurement at the orifice inlet shall be made utilizing dead weight pressure gages with traceable certification. Two separate measurement taps shall be provided and measured (Figure 7).

- The temperature measurement at the orifice inlet shall be made utilizing thermocouples and precision potentiometric readout devices. Two separate measurement taps, with wells, shall be provided and measured. Location of these thermocouples shall be in accordance with ASME guidelines. These shall not be installed in the same plane as the two pressure taps previously mentioned (Figure 6).

- Inlet pressure measurement at the trip and throttle valve shall be made utilizing dead weight pressure gages with traceable certification. Three separate measurement taps shall be provided, spaced at 120 degree intervals in the pipe circumfer-

ence. The axial location in the piping shall be in accordance with ASME installation guidelines (Figure 8).

- Inlet temperature measurements at the trip and throttle valve shall be made utilizing thermocouples and precision potentiometric readout devices. Three separate measurement taps, with wells, shall be provided at 120 degree intervals in the pipe circumference and out of plane with the pressure taps. The

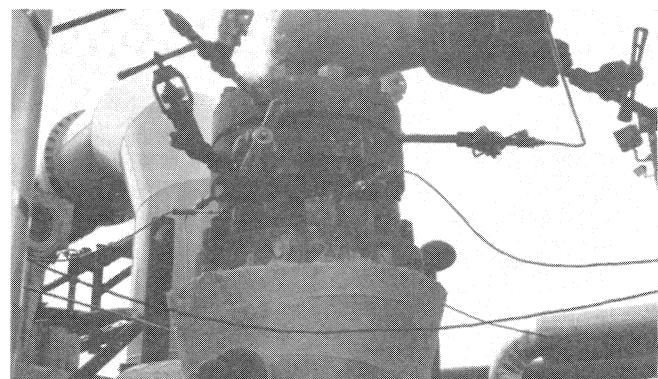


Figure 8. Installation for Inlet Pressure and Temperature Measurement.

axial location in the piping shall be in accordance with ASME installation guidelines (Figure 8).

- Exhaust pressure measurement shall be made at the turbine exhaust flange utilizing dead weight gages with traceable certification. Three separate measurement taps shall be provided at 120 degree increments in the flange circumference (Figure 9).

- Exhaust temperature measurements at the turbine exhaust shall be made utilizing thermocouples and precision potentiometric readout devices. Three separate taps, with wells, shall be provided in the pipe circumference. The axial location shall be in accordance with ASME installation guidelines (Figure 9).

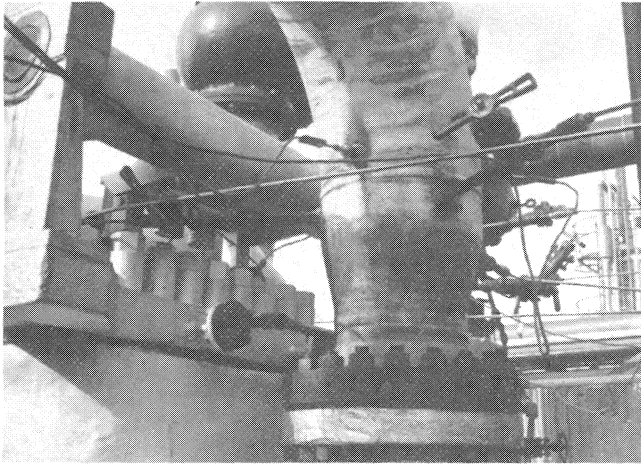


Figure 9. Installation for Exhaust Pressure and Temperature Measurement.

- Speed shall be determined by two separate speed indicating devices. These shall measure the frequency of an electrical output signal analogous to that of a rotating 60-tooth gear.

- Thermocouples and lead wires shall be continuous and be terminated in a switch box. The switch box shall route the temperature analog signals to a precision potentiometric device. Three inlet, three exhaust, three differential and two orifice temperatures will be read and switched. All thermocouple outputs shall be referenced to an ice bath (Figure 10).

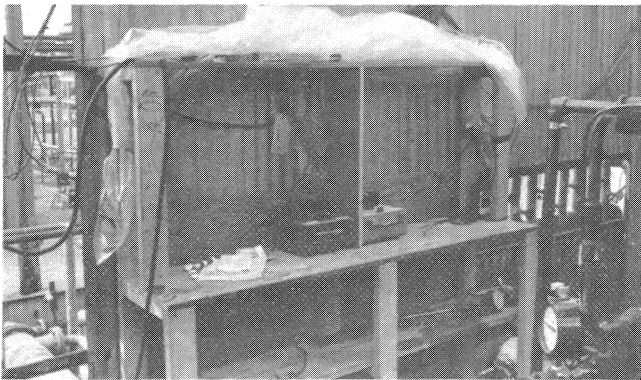


Figure 10. Installation of Switch Box for Thermocouple Leads.

Proposed Instrumentation

- Dead weight gages—range 2000 psig, Quantity: 8 (certified and traceable)
- Thermocouples—Chromel-Alumel (calibrated), Quantity: 8
- Temperature readout devices—precision micro voltmeter, Quantity: 2 (calibrated)
- Counters (speed)—Quantity: 2 (calibrated)

- Differential pressure cells—precision units of appropriate range, Quantity: 2 (calibrated)

The following ASME guidelines will be observed:

- A preview of the customer's piping and instrumentation diagrams will be required.

- Agreement on the details of the installation of the instrumentation will be reached.

- ASME PTC-6 steam turbines shall be used as a guide.

- PTC-6 report "Guideline for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines" (1969) shall be used as a guide to obtain instrument uncertainties.

The proposed test setup is shown in Figure 4.

Item	Assumptions	Uncertainty	Effect on Efficiency
Flow	Uncalibrated Orifice Inspected Before Test $1.1K + 0.95$ $K_1 + K_2 = 1$ $K_3 = 2.9$ for $\beta = 0.7$	8.54%	± 2.04
P_1	Dead weight Gages—Chandler Model 15-001-B-T Range 2000 PSIG	± 0.1 PSIG	Neg.
T_1	Continuous Leads Calibrated +0.03% Potentiometer Chromel-Alumel (calibrated) TC	$\pm 1^\circ\text{F}$	± 0.12
P_{exh}	Same as P_1 Range 1000 PSIG	± 0.1 PSIG	Neg.
T_{exh}	Same as T_1	$\pm 1^\circ\text{F}$	$\pm 0.7\%$

Total Effect of Uncertainty on Turbine Efficiency

$$\sqrt{(2.04)^2 + (0.12)^2 + (0.7)^2} = \pm 2.16\%$$

This was amended due to the removal of flow straighteners.

Field Test Results

The results achieved when the field test was completed are presented in Table 1. There are several additional points which should be mentioned. First, there was very close agreement seen between all of the sets of corresponding instruments which read each of the parameters. Second, the plant flow meter was found to read approximately ten percent greater flow than the ASME orifice, thus the amount of steam apparently required by the topping turbine was considerably greater than the actual steam flow. This shortfall directly manifests itself in a higher steam rate or lower efficiency and explained a very low apparent initial performance level.

This highlights one of the conflicts any plant operator faces, namely, when instrumenting to achieve accurate tests with high precision flow measurement, very often unacceptably high pressure drops must be introduced into the steam lines. The desire in these situations is to minimize the length of time the plant is run with such line drops in place, but normal plant turnarounds occur only every year or two.

The comparison of the original two-stage design and the three-stage retrofitted design is graphically illustrated in Figure 11.

IMPROVEMENTS MADE FOR MECHANICAL IMPROVEMENT

Explanation

While the major thrust of this retrofit was initially directed in the area of thermodynamic improvement, there was also an

Table 1. Field Test Results.

	Original (2 Stage)		Retrofit (3 Stage)	
	Data Sheet	Observed	Guaranteed	Test (ASME PTC6)
Inlet pressure (psig)	1,450	1,450	1,450	1,450
Inlet temperature (F)	825	856	856	856
Exhaust pressure (psig)	555	555	560	560
Exhaust temperature (F)	618		646	641
Throttle flow (lb/hr)	494,800	566,250*	467,200	459,120
Horsepower	15,100	15,100**	15,100	15,100
Speed (rpm)	10,411	10,411	10,600	10,600
S.R.	32.7	37.5	30.93	30.4
T.S.R.	23.86	23.26	23.26	23.26
Efficiency (%)	72.95	62.0***	75.2	76.5

*Measured with existing flowmeter
 **Best estimate
 ***Actually thought to be about 65%

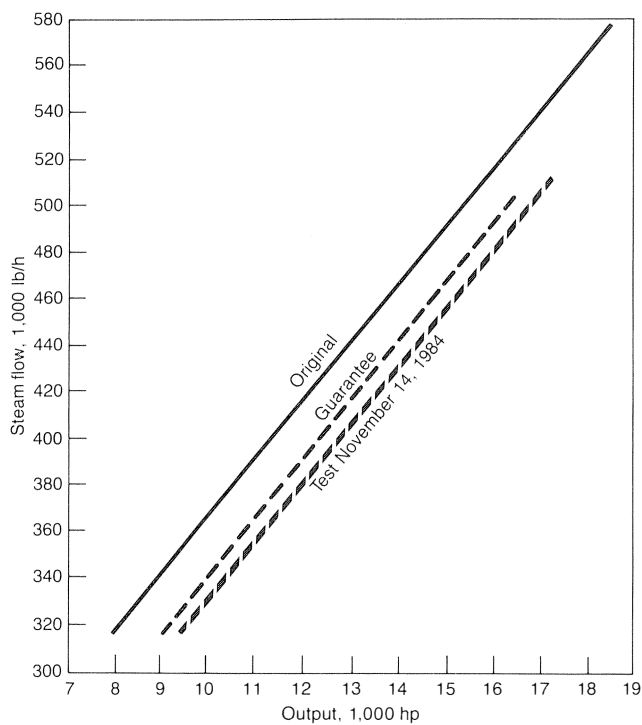


Figure 11. Comparison of Original and Retrofit and Test Willans Line.

opportunity at hand to remedy the various mechanical problems experienced over the past 22 years. While many plants have not experienced any of the symptoms addressed here, the overall aim was to optimize the design in every regard. Thus, all aspects were incorporated into this retrofit (Figures 12 and 13).

These improvements include:

- larger exhaust end journal bearing
- heavier shaft
- install a double acting Kingsbury thrust bearing
- larger exhaust end oil drain
- refurbished casing bores and joints.

It is interesting to note that the topping turbine is even today, almost 25 years later, still one of the highest speed/highest horsepower steam turbines found anywhere in the world. At the

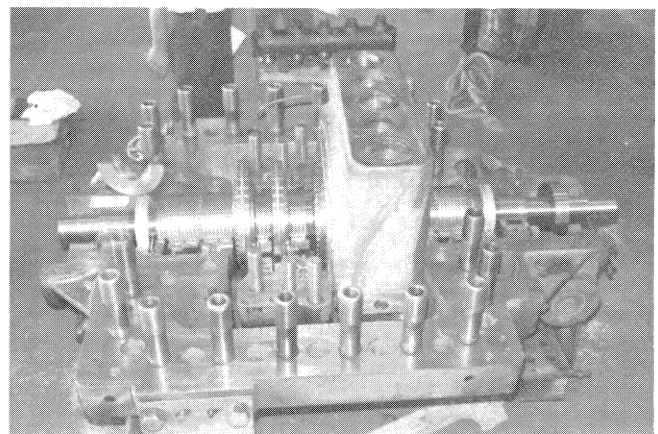


Figure 12. Photograph of the Bottom Half Casing with Rotor and Nozzle Block in Place.

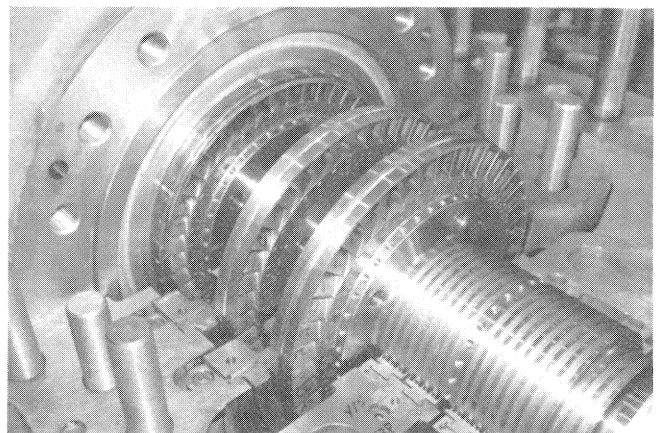


Figure 13. Close-up of the Three-Stage Rotor in the Bottom Half Casing.

time of its original design back in the early 1960s, there was no other unit in its class.

Additionally, its location in the synthetic gas train, as a drive through unit, makes it vulnerable to misalignment effects and coupling overhang effects. Over the years there have been a number of modifications aimed at minimizing these problems, but the results of this retrofit were certainly more beneficial than

expected. Mechanical improvements fall into the area which is the most difficult to quantify economically, but the easiest to appreciate in actual operation.

Rotor Response Data Comparing the Two-Stage and the Three-stage Rotor Designs

In this section the rotor response data for a two-stage and three-stage synthetic gas topping turbine will be compared. The rated speed is 10850 cpm with an output of 20000 hp for both. The steam conditions are 1450 psig, 856°F, and the exhaust pressure is 560 psig. The bearing sizes are 4 in at the inlet end and 5 in at the exhaust end for the two-stage, while the bearings are 5 in at both ends for the three-stage version. The bearing span is identical for both the two-stage and three-stage versions. The two-stage rotor is known as the "short shaft" version, and uses a light weight, reduced 3 in coupling on the exhaust end.

The rotor response map plot (Figure 14) gives rotor response speeds as a function of variable bearing stiffness. The bearings used have a stiffness of approximately 5×10^5 lbs/in. The stiffness is low, due to the light loading of the bearings.

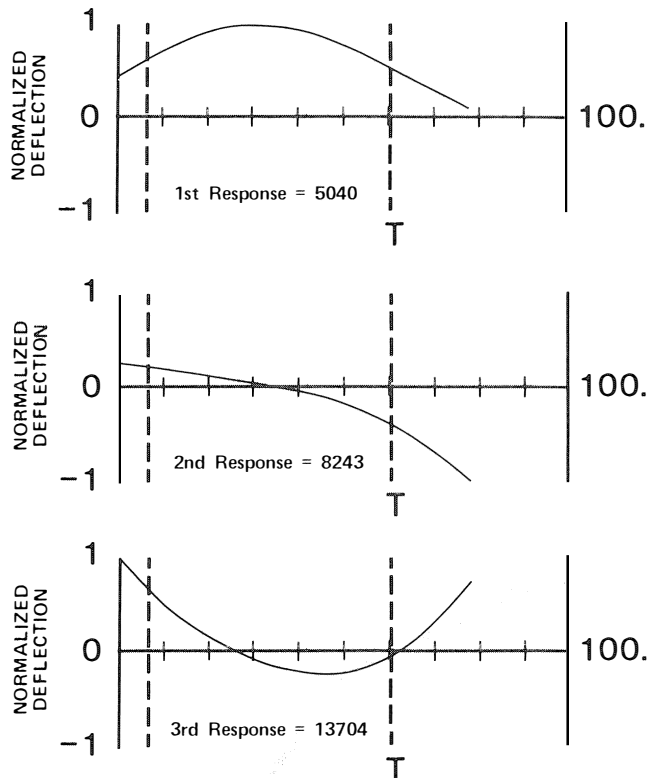


Figure 14. Rotor Response Map—Original Two-Stage Design Bearing Stiffness = 500,000 lb/in.

The first, second, and third mode shapes are shown for a stiffness of 5×10^5 lbs/in. The first mode response has little bending and large deflections at the bearings, typical of a well damped system. The second mode response is a typical "rigid" shaft response, commonly called the "see saw" or "conical" mode. The shaft exhibits little bending and large deflections at the bearing, typical of a well damped system. Since this response is "invisible" in actual operation, it is not considered or listed as a critical speed. The third mode response, which is above the operating range, has a shape similar to the classical "first critical" response. There is significant bending in the rotor and the nodal points tend to migrate toward the bearing. Note

that the deflections are normalized to a unity maximum deflection in these plots.

Three-Stage Rotor Response Data

The rotor response data for the three-stage rotor is shown in Figure 15. The operating data is identical to the two-stage rotor with the exception of the journal bearings previously mentioned. The comments which were made previously concerning the rotor response maps and rotor mode shapes apply for both the three-stage rotor and the two-stage rotor. It is worth re-emphasizing that a radically different rotor design is not being discussed here, but rather an incremental improvement over an already mature design.

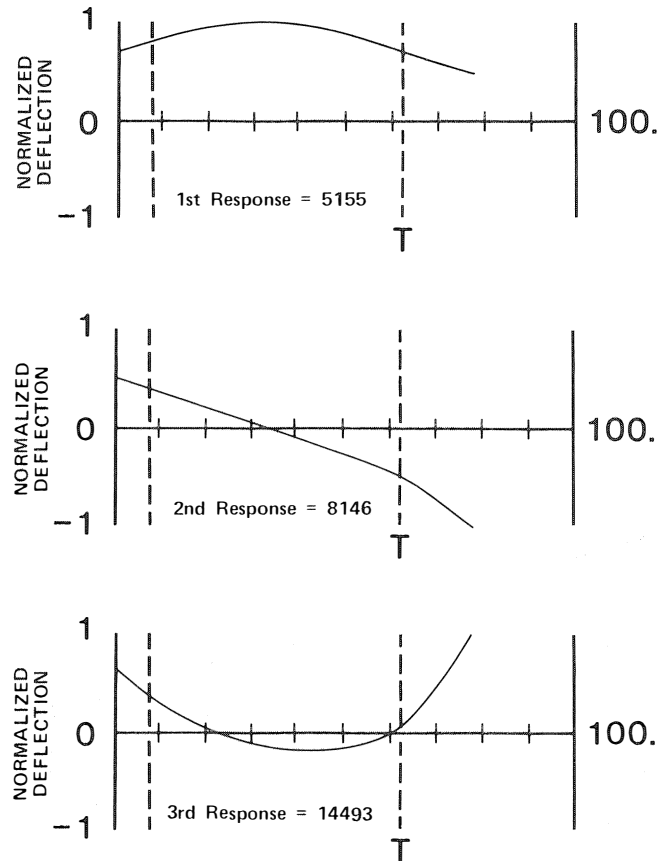


Figure 15. Rotor Response Map—Up-rated Three-Stage Design Bearing Stiffness = 500,000 lb/in.

Three-Stage Topping Turbine Field Test Data

The results of the field test conducted at the plant are included in Figure 16. The data shows relatively high levels of vibration on both ends of the topping turbine's couplings, whereas the levels on the topping turbine rotor itself are very small. This indicates that the unit runs very smoothly on its own, and maintains that stability in the coupled configuration with significant excitations immediately adjacent to it (Figures 17 and 18).

It should be mentioned that operation under field conditions is the true test of any design. While it is always important to utilize all of the modern analytics available today to fine tune a given design, it is only when that unit is put into service that it can be determined whether all intentions have been realized. This turbine certainly passed that test.

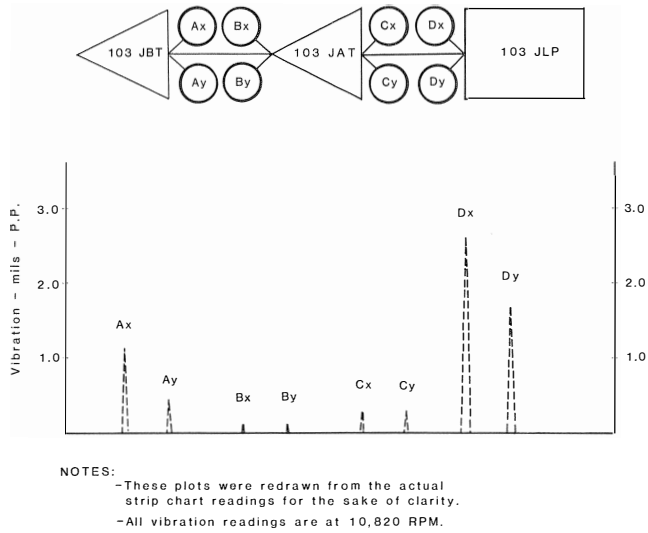


Figure 16. Comparison of Outboard End Vibration Levels at Operating Speed.

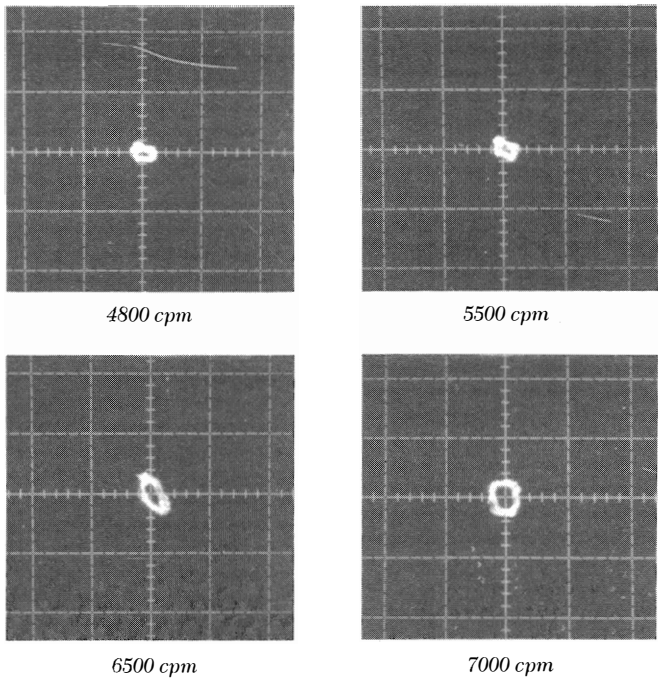


Figure 18. Orbital Data of the 103 JAT Turbine (LP Compressor End) as the Unit Comes Up Through First Critical Speed. (Probe 7 and Probe 8 output end horsepower turbine 1 mil/din peak-to-peak.)

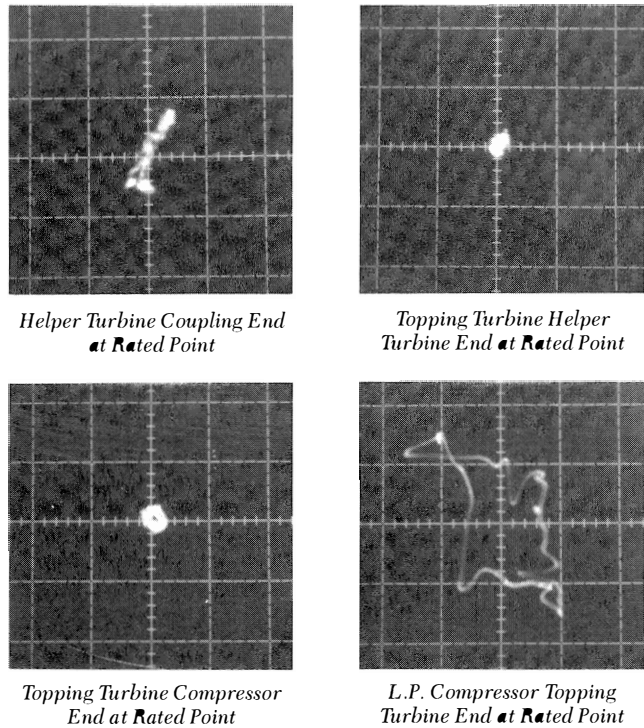


Figure 17. Orbital Data Showing Both Ends of the 103 JAT and the Ends of the Units Immediately Adjacent to It. (1 mil/din peak-to-peak.)

CONCLUSION

Two aspects of an upgrade or retrofit conversion have been presented. The thermodynamic improvements were made to improve performance over existing levels. Often, similar types of modifications are made when steam conditions change during the evolution of a process plant's life.

Mechanical modifications are very often implemented as design improvements are made available. This retrofit, which involved wholesale changes to an already mature design steam turbine, was somewhat different. The reason is that the impetus for the retrofit was based on economic savings. The steam turbine owner had a choice when considering this program, whereas in most other retrofits, acceptable future operation demands that modifications be made to the machinery.

The final point to be made is that when modernizing many of today's plants, very often the best solution is the simplest solution.

REFERENCE

1. ASME PTC-6, "Guideline for Evaluation Uncertainty in Performance Test of Steam Turbines" (1963 - Reaffirmed 1973).

