GUIDELINES FOR SPECIFYING AND EVALUATING THE RERATING AND REAPPLICATION OF STEAM TURBINES

by

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ABSTRACT
Steam turbines may be rerated, reapplied, or modified to meet several specific goals. This tutorial will review possible reasons for these changes, including optimizing performance, improving reliability, reducing maintenance requirements, solving operating problems, extending equipment life, and, finally, replacement of the turbine, either in whole or part, due to a catastrophic failure, normal wear, or problems found during an inspection. Whatever the project, it normally requires clear goals specified by the user and buyer. The vendor can then review these goals and determine what can realistically be accomplished within the constraints of the existing equipment, the project budget, and the time allowed. Most of the equipment limitations will be discussed in this tutorial as well as the presentation of a case study. For many of these projects, the vendor will not have access to the turbine since it will still be operating at the user’s facility. Therefore, closer than normal coordination between the user and the vendor is required to ensure smooth and timely completion of the project.

INTRODUCTION
The first part of this tutorial will define many of the reasons for rerating or reapplying a steam turbine, as well as what hardware or software may be available to accomplish the task. This is followed by a listing of the information needed from the user and an explanation of the limitations faced by the vendor—specifically those that are not usually encountered with new equipment. Finally, a case study of an uprate of a mechanical drive turbine is presented.

PERFORMANCE CHANGES
The basic power and/or speed rating of a steam turbine may change for many reasons. The most common one is an increase (or decrease) in the power required by the driven machine due to a plant expansion or debottlenecking. Other reasons include a reapplication of the turbine to drive a different machine, a search for increased efficiency, a change in the plant steam balance, or a change in steam pressure or temperature.

Power
An increase in power usually requires more steam flow area inside the turbine, which may or may not be possible within the physical limits of the existing casing. A decrease in power is almost always possible simply by blocking off some flow area, but maintaining efficiency usually requires a more sophisticated...
solution. Reapplication of a turbine often is the most difficult problem for the vendor since the power, speed, and steam conditions can all be considerably different from those for which the turbine was initially designed. This is discussed in more detail later in this paper.

Efficiency

More efficient blading may be available for older turbines. Considering the cost of energy today, it may make sense to invest in new blades (both rotating and stationary) if the gain in efficiency is large enough. However, higher efficiency blading often requires more axial spacing along the rotor, and there may not be enough room in the existing casing.

There are simpler ways to improve or maintain efficiency. Leakage through labyrinth seals can be reduced by up to 80 percent by integrating brush type seals with the usual stationary labyrinths as shown in Figure 1.

![Figure 1. Labyrinth-Brush Seal.](image1)

The brush seal consists of bristles that are angled slightly with shaft rotation. They can tolerate some deflection and still spring back to their original position. They can be incorporated between the labyrinth teeth as well as at the ends. For higher pressures, brush seals may require pressure balancing to avoid excessive downstream deflection. Note that due to the angled bristles, some brush seals may not tolerate reverse rotation. Figure 2 shows brush seals in a heavy metal retainer.

![Figure 2. Brush Seal in Metal Retainer.](image2)

Labyrinth teeth can be damaged by rubs, especially during startup or coastdown when the turbine rotor passes through a lateral critical speed. Rubs may also occur at startup due to different rates of thermal expansion between the seal and the rotor. The rub opens the clearance and reduces efficiency. Retractable packing is a possible solution if your turbine has this problem. The labyrinth ring is circumferentially divided into segments that are spring-loaded to hold them apart and therefore give a very generous clearance. Once the turbine starts and steam pressure builds up on the outside diameter of the seal, it overcomes the spring pressure and it closes the seal to normal clearance. When the turbine trips, steam pressure is reduced and the spring again opens the clearance for coastdown.

Almost all turbine stages have seals between each diaphragm and the shaft, but many turbines do not have tip seals between the rotor blade tips and the casing/diaphragm. If room permits, tip seals can be added to increase efficiency. Tip seals are more effective when used in higher pressure stages and stages with greater reaction. Tip seals are shown in Figure 3.

![Figure 3. Tip Seals.](image3)

Any of the seals mentioned above can combine the features of the brush seal and retractable seal.

RELIABILITY AND MAINTENANCE

The trend in most industries is for longer runs and shorter turnarounds. There is also a lot of pressure to eliminate unplanned shutdowns and reduce required maintenance.

Electronic Controls

Perhaps the biggest change in steam turbines in the past 20 years has been the conversion of the speed control and trip systems from totally mechanical to totally electronic. Many existing turbines can benefit from a change to electronic controls.

Figure 4 shows a typical mechanical governor system for a straight-through turbine. The governor is powered off the turbine shaft by a worm and wheel drive. The governor itself has many internal moving parts including flyweights, springs, an oil pump, accumulator pistons, and several valves. These governors are quite reliable, but with all those moving, contacting parts, wear and maintenance are inevitable.
An electronic governor replaces the worm gear with a toothed wheel. Multiple noncontacting magnetic speed pickups read off the toothed wheel and provide the signal to the electronic governor. The power cylinder in Figure 4 is replaced with an electronic or pneumatic actuator. The pilot valve and restoring linkage remain intact. Many electronic governors are also more versatile in that they can be programmed to use parameters other than speed alone to control the turbine. Most electronic governors can also interface with the user’s distributed control system (DCS).

Things get much more complicated for mechanical controls when they are applied to an extraction turbine, as shown in Figure 5. Here the power cylinder controls the added extraction pressure regulator, which includes even more linkages and springs as well as a pneumatic signal to monitor extraction steam pressure. The extraction pressure regulator sends signals to either prepilot cylinder on either or both servos. This system can maintain a set turbine speed and an extraction pressure level. However, the linkages require maintenance, the linkages and springs wear, and if there is a change to extraction pressure, the linkages and springs need adjustment.

An electronic governor designed for extraction turbines works as described above for straight-through turbines, with the addition that it receives an electronic signal for extraction steam pressure and outputs signals to the electronic (or pneumatic) actuators that replace the prepilot cylinders on each servo motor. Again other signals (such as a compressor discharge pressure signal) can be fed to and processed by the electronic governor. Here a change to the extraction pressure is a simple set point change.

Electronic governors are not failproof, but they are available in redundant and triple modular redundant formats, so failure of electronic components will cause the governor to switch to backup components while the failed components are replaced. Most multivalve turbines still require hydraulic servo motors since electronic ones are not powerful enough to move the valve racks. We can go a step further, however, and eliminate the prepilot and pilot valves, in addition to the linkages, cams, and rollers shown as the restoring linkage in Figures 4 and 5. Figure 6 shows that the servo motor can be fed by a way valve. The way valve takes the oil directly from the control oil header and directs it to the proper side of the servo motor. The electronic governor sends a control signal to the actuator coil that is an integral part of the way valve. The restoring linkage is replaced by one or more linear variable differential transformers (LVDTs) that provide feedback to the control system. The way valve may have dual actuator coils for redundancy.

Turbines are still tripped by dumping the oil that holds the trip and throttle valve open. Figure 7 shows the arrangement formerly used on most turbines. The overspeed trip was initiated by a spring-loaded trip pin or a weighted Bellville spring that struck a mechanical lever that actuated a dump valve. The solenoid dump valve shown was for remote tripping purposes.

Most multivalve turbines still require hydraulic servo motors since
existing turbine to mount the necessary speed pickups, but beyond that it is fairly simple to add an electronic overspeed trip.

**Monitoring Systems**

There are many steam turbines that have been in operation for 30 or more years that do not have the level of instrumentation that is considered standard today. Radial and axial vibration probes, as well as bearing temperature instrumentation, may be fitted to existing turbines. Readouts from these instruments can be fed to sophisticated monitoring equipment that not only can display the current readings, but also can determine trends that may uncover a potential problem before it causes an unexpected outage. The instruments could indicate that a change to a different bearing (tilt shoe, spherical seat) or orientation, such as load between pad, would be beneficial.

The main stumbling block to retrofitting vibration probes is finding the space in the bearing housings to mount them with a corresponding free area on the shaft to read from. The shaft area may also have to be treated to reduce electrical runout to an acceptable level. Bearing temperature instrumentation will normally require some machining of the bearing retainers and housings to allow the wires to exit the turbine.

The user should be aware that it is fairly common for machines that have been operating satisfactorily for many years to show high vibration readings once probes are installed. The excessive vibration may have always been there, but it just has not been to the point where it has caused a problem. In such cases, the vibration level is normally still not a problem. Although it exceeds current standards, if the vibration has not caused any operating or maintenance problems in the past, there is no reason to believe that it will be a problem in the future. With the probes installed, however, it is now possible to monitor and trend the vibration levels and to avoid potential future problems.

**OPERATING PROBLEMS**

Many operating problems may be solved by some of the items that have already been discussed. Monitoring systems can uncover problems such as a bowed rotor or fouling. There are also much better tools to determine rotordynamic characteristics than there were 30 years ago. The seals that were discussed can eliminate water contamination of the oil and excessive gland leakage. The electronic controls can cure some process control problems or speed fluctuations. Speed fluctuation can also be caused by improperly sized governor valves, or valve and seat wear. Performance problems have to be analyzed on an individual basis. Again, there are more sophisticated tools available now than there were just a few years ago, such as computational fluid dynamics (CFD).

**LIFE EXTENSION**

There are several options available to extend the life of an existing turbine.

Most types of rotor damage can now be repaired by machining off the damaged area, rebuilding it with weld, and remachining the rotor. Some casing damage may be repaired in a like manner.

Better materials are often available to solve erosion problems. Blades also can be protected by a stellite overlay in critical areas. Some coatings are also available that may help with erosion, corrosion, or fouling problems.

A bearing upgrade or something as simple as an at-speed balance could also extend the life of a turbine by eliminating a vibration problem.

**REPLACEMENT**

If the user knows in advance that they will be replacing their turbine, they have many options. A new turbine is certainly an option. Or there may be used turbines on the market that can be refurbished to meet the requirements. The user may even have another turbine that they would like to reapply to the existing service.

If the turbine is damaged in a wreck, it is an entirely different matter. This is an emergency and a quick solution is essential. A new turbine is usually out of the question because the lead time is too long.

Depending upon the extent of the damage, most original equipment manufacturers (OEMs) and shops that specialize in turbines can accomplish fairly major repairs in a short period of time by utilizing welding, plating, and other proven repair techniques. Blades and other specialized parts might be available in a reasonable amount of time.

Catastrophic damage will require a search for a suitable replacement. Again, most OEMs and turbine specialty shops keep track of equipment that is available on the used equipment market. The vendor may even have used turbines of his own to reapply. A suitable turbine may be overhauled with minor changes and be back in service in a fraction of the time required for a new unit.

If no suitable turbine can be found on the used market, it may even be possible to reapply another of the user’s turbines (possibly in a less critical service) to replace the damaged turbine. It may then be feasible to replace that turbine with either a new or used unit.

**STARTING THE PROCESS**

**Define the Objective**

Although not considered a vital part of the process, possibly the most critical part of the project is to define the objective of the project. Many times projects are initiated without a clear objective. This lack of a defined goal often leads to failure of the entire project.

The process starts by an analysis of the driven machine to determine the actual power requirement and speed and evaluation of steam conditions at all the expected operational points, including excess power margin. Steam pressure drops in the supply, exhaust, and extraction piping must be accurately calculated. On extraction units, the extraction pressure and flow must also be defined.

In the initial purchase, many turbine standards were considered, including corporate, American Petroleum Institute (API), National Electrical Manufacturers Association (NEMA), industrial, and local governmental standards. It is up to the engineering staff directing the changes to determine which, if any, of these specifications are required to be applied. Some of the original specifications or specifications that have been enacted since the unit’s original commissioning may need to be applied. One note of caution: older turbines must be carefully analyzed, as many older units cannot meet current standards or may only do so at considerable added lead time and expense.

**Construction Rating**

Turbines undergoing a change in inlet pressure, temperature, and/or exhaust pressure require that maximum casing pressure and temperature rating be checked. This includes a review of the casing and flange ratings to ensure the casing is suitable for the new conditions. Multistage turbines with changes in pressure and flow require a verification of the first stage maximum allowable pressure. In the case of a multistage extraction turbine, a change in flow or extraction pressure will also require that intermediate casing and extraction flange be reviewed.

A change in inlet temperature will require a verification of the casing material. Typical limits are given in Table 1.

<table>
<thead>
<tr>
<th>Casing Material</th>
<th>Maximum Temperature (Deg F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ASTM A278 Class 40</td>
<td>600</td>
</tr>
<tr>
<td>ASTM A216 Grade WCB</td>
<td>775</td>
</tr>
<tr>
<td>ASTM A217 Grade WC1</td>
<td>825</td>
</tr>
<tr>
<td>ASTM A217 Grade WC6</td>
<td>900</td>
</tr>
<tr>
<td>ASTM A217 Grade WC9</td>
<td>1000</td>
</tr>
</tbody>
</table>
There are many other materials that have been used over the years, but these are the major casing materials currently in use. Before exceeding these limits, a thorough review of the metallurgy of the existing casing must be done.

Once the casing rating has been determined, the original hydrotest pressures must be determined. This can be done by contacting the original OEM, reviewing the original construction documentation (i.e., hydrotest certificates, data sheets, correspondence), or reviewing the turbine casing for markings. The hydrotest pressure must meet the new conditions based on the latest industry standards. The API guidelines refer to ASME Boiler & Pressure Vessel Code - Section VIII (2004).

In some cases, the casing will need to be rehydrotested to the new conditions. This may need to be done to one or several sections or the entire turbine casing.

A decision must be made as to whether NEMA limits need to be taken into account. For new construction, most OEMs verify the casing design will ensure the casing will meet the maximum design pressure and temperature plus 120 percent of the rated pressure combined with a temperature increase of 50ºF. This verification is also included in the evaluation of the flange rating.

**Flange Sizing**

If during the modification of the turbine, the flow increases or the specific volume decreases, the size of the inlet flange must be reviewed. Typical maximum value for the inlet velocity is 175 ft/sec. Equation (1) determines the inlet velocity.

\[
V = 0.509 M \sqrt{\frac{d^2}{V}}
\]  

(1)

where:
- \(V\) = Velocity in flange (ft/sec)
- \(M\) = Mass flow (lb/hr)
- \(d\) = Diameter of the inlet (inch)
- \(V\) = Specific volume of steam (cu ft/lb)

In smaller turbines, the inlet flange with the steam chest and control valving can easily be changed. On larger turbines, increasing the inlet size may be very difficult. Some turbines have the ability to add an additional inlet on the existing steam chest by welding on a flange followed by a local stress relief. Figure 8 shows a flange undergoing modification from an 8 inch inlet to a 10 inch inlet.

Some turbines may have a blank flange that can be easily removed and piped for additional inlet area. Refer to Figure 9 for a typical steam end with dual inlet capability.

**Figure 8. Increasing a Flange in Size.**

The trip and throttle (T&T) valve must also be reviewed, but this will be discussed in a later section. As a worst case option, the steam velocity can be allowed to exceed the 175 ft/sec limit as long as the correct pressure drop is taken into account and additional acoustic protection is provided as the noise level will increase.

The extraction and exhaust line sizes are also areas that need to be considered. The maximum value for an extraction line is 250 ft/sec. The maximum for a noncondensing exhaust is 250 ft/sec and for a condensing exhaust it is 450 ft/sec. Typically there are not many options to upgrade an extraction or exhaust connection. The most practical solution is to increase the exhaust header size as close to the turbine casing as soon as possible. Again the appropriate pressure drop calculations must be done to determine the pressure at the flange. It will be an iterative procedure with good communication between the engineering staff involved in the rerate and the engineering staff with site responsibility.

**Nozzle Ring Capacity**

After the external issues have been decided, the first internal component requiring review is the nozzle ring or nozzle block. This is the inlet to the control stage that ultimately controls the amount of steam a given turbine will be able to pass. If during the original manufacture of the turbine, the nozzle ring had additional area available, the turbine may be upgraded by adding additional nozzles. Depending on the design, this can be as easy as installing a new nozzle ring. Some small turbines have nozzles machined into the steam end and modifying these turbines is difficult, if not impossible. It may be cost effective to purchase a replacement turbine if this is the case.

If the change in flow is significant, the nozzle may have to be replaced with a nozzle ring that has an increased nozzle height. This will normally require increased blade height on the turbine rotor. The maximum number of nozzles in each bank or segment of the steam end is dictated by the original design parameters. Many times, the maximum nozzle area in a steam end is matched to the volumetric area of the inlet of the turbine.

The same issues affect the extraction nozzle ring on an extraction turbine although the extraction diaphragm in some instances can be changed or modified to allow for additional nozzle area.

**Steam Path Analysis**

Once the inlet nozzling has been reviewed, the remainder of the steam path must be reviewed. An increase in flow may require an increase in diaphragm nozzle height and rotor blade height in some or all stages. As stage flow increases, the heat drop across the stage will increase with the associated decrease in velocity ratio. The velocity ratio is defined as the ratio of the blade at the pitch diameter to the steam jet velocity, as defined in Equation (2).

\[
V_o = \frac{DN \pi}{224 \sqrt{\Delta h}}
\]  

(2)
where:
\[ V_o = \text{Velocity ratio} \]
\[ D = \text{Pitch diameter of rotating blades} \]
\[ N = \text{Shaft rotational speed} \]
\[ \Delta h = \text{Enthalpy drop across the stage} \]

In order to increase the efficiency, the diaphragm and blade path areas must be carefully matched. Each stage design has an optimum velocity ratio with the equivalent maximum efficiency. During the modification, some of the stages will require changes for mechanical or other reasons. Some of the stages will be acceptable with less than optimal efficiency. Decisions must compare the loss in efficiency versus the cost of modifying each stage.

**Rotor Blade Loading**

With the change of the power, flow, or speed of a turbine, each rotating row of blading requires review of its mechanical properties. This includes a review of the speed limits and analysis of both Goodman and Campbell diagrams. The blade mechanical speed limits are related to the disk and blade geometry, stress values for material, and shroud design.

The Campbell diagram graphically compares the blade natural frequency to the turbine speed range. The natural frequencies of the blading can be obtained from the OEM or determined experimentally. If the speed range is not changed in the modification, the Campbell diagrams will not change except for any blading that has been changed. Figure 10 illustrates a typical Campbell diagram.

The Goodman diagram will evaluate the combined effects of alternating stresses and steady-state stresses in the turbine root and base of vane locations. This diagram contains a Goodman line, which in theory represents the minimum combination of steady-state stress and alternating stress, above which a blade fatigue failure could occur. Figure 11 is a typical Goodman diagram. The OEM will normally have minimum allowable values for a Goodman number. There may be several values depending on the location of the blade within the steam path; typically partial arc admission and the moisture transition stages will have higher limits. A significant change in the stress levels of the blading may require replacement or redesigning of the blading. In some rare cases, no blading design can be found and low Goodman numbers may need to be accepted.

**Thrust Bearing Loading**

In many cases, an increase in flow and speed may increase the thrust values. Depending on the age of the turbine in question, a replacement thrust bearing may be required. There are higher capacity thrust bearings available that will fit in the existing cavity. In the most extreme cases, a bearing housing with a larger thrust bearing may be required.

**Governor Valve Capacity**

During the design cycle, the capacity of the governor valve must be checked to ensure the correct flow area is present. The valve area must be carefully matched to the area of the nozzle ring. In most cases the nozzle area should be the limiting point of the flow in a valve bank. Setting the valve as the limit will not provide the most efficient conversion of pressure into velocity.

If the valve(s) and seat(s) require an increase in size, most casings can accept a larger size valve and seat. Some single valve turbines are able to have the steam chest size increased to the next larger size.

**Shaft End Torque Limit**

With an increase in power or a decrease in speed, the shaft end stress will need to be reviewed. The basic equation required to calculate shaft stress is given in Equation (3).

\[
\tau = \frac{321000P}{Nd^3}
\]
where:
\[ \tau = \text{Shaft shear stress} \]
\[ P = \text{Power (hp)} \]
\[ N = \text{Shaft speed (rpm)} \]
\[ d = \text{Shaft diameter (inch)} \]

The limit of the shaft shear stress is a function of the material and the heat treatment as well as the shaft end design. Typical turbine shaft ends are single or double keyed, NEMA taper, hydraulically fit, or integral. Each of these has different allowable stresses depending on the stress risers within the design. Typical turbine shaft materials are given in Table 2.

### Table 2. Turbine Shaft Materials.

<table>
<thead>
<tr>
<th>Commercial Designation</th>
<th>Material</th>
<th>Typical Maximum shear stress (PSI)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AISI C-1040</td>
<td>Medium Carbon Steel</td>
<td>5000</td>
</tr>
<tr>
<td>AISI 4140</td>
<td>Chrome Moly Alloy Steel</td>
<td>11500</td>
</tr>
<tr>
<td>AISI 4340</td>
<td>Nickel Chrome Moly Alloy Steel</td>
<td>12500</td>
</tr>
<tr>
<td>ASTM A470 Class 4, 7, 8</td>
<td>Nickel Chrome Moly Vanadium Alloy Steel Forging</td>
<td>11500, 12500, 12000</td>
</tr>
</tbody>
</table>

The loading of the driven machine may also have an effect on these values. For example, a generator drive may reduce the allowable stress further to account for short circuit loading.

### Speed Range Changes

Although this has been briefly discussed in other sections, care must be taken when making significant changes to the speed range. The most important is the blading speed limits (refer to the previous section, Rotor Blade Loading), but other areas require analysis such as lateral critical speeds, torsional critical speeds, coupling speed limits, etc.

Lateral critical speeds are a function of the rotor design, bearing design, and bearing support design. Typically the original critical speed is denoted on the nameplate for the turbine and also in the vendor literature. If no changes are being done to the rotor and operational data agree with the noted critical speed, this value normally will not change. Should changes be made to the rotor or bearing system, a new lateral analysis will need to be done to confirm the speed of the lateral critical speed. A typical report may cost between $3000 and $30,000 depending on the complexity of the rotor and what existing data can be reused.

Torsional critical speeds will be required. Each body in the string must be modeled as well as each coupling and gear set. Direct drive units are not normally an issue, but strings with gears require the torsional to be checked. These may not be as readily available as the lateral critical speed. Normally this documentation is supplied by the vendor in the form of a report with engineering documentation. The cost to complete a report will vary from $3000 to $10,000 per body.

The turbine governor, which will need to be modified or reprogrammed to meet the new speeds as well as the trip mechanism, could either be mechanical or electronic. Other driven systems will require review such as a gear (if used) or any shaft driven oil pumps.

### Auxiliary Equipment Review

Each turbine has auxiliary equipment that will require some review during any modification. These may include the T&T valve, other steam block valves, leakoff and sealing steam system, surface condenser system, relief valve sizing, lubrication and control oil systems, valve actuation system, and supervisory instrumentation.

The T&T valve may be acceptable as is for the change in the flow, but may be upgraded for reliability or the addition of a manual exerciser. Some T&T valve OEMs are no longer actively pursuing this market, so an upgrade may be more expensive than a complete new valve of a current design. For extraction turbines, the nonreturn valve should also be looked at to determine acceptability of long-term continued operation. Other steam valves in the system should be reviewed for proper sizing and good mechanical operation.

The leakoff and sealing steam system must be reviewed to determine if additional leakoff flow will be experienced with the new conditions. API 612 (1995) and 611 (1997) both require 300 percent capacity in the leakoff system. If a change has been made to the first stage pressure or the exhaust pressure, this additional capacity may be difficult to achieve without major rework. The gland condenser will require a review along with the ejector or vacuum pump to ensure these are of adequate capacity. From a piping point of view, it may be a good decision to replace some or all of the leakoff and drain piping if significant corrosion or buildup is noted on the inner diameter (ID) of the piping from the turbine.

On a condensing turbine, the surface condenser system must be reviewed to ensure the proper capacity is available with the current cooling water temperature and available cooling water flow. The hotwell capacity and pump sizing must be verified to ensure the correct hotwell level can be maintained. Instrumentation connected with the condenser system should be reviewed and upgraded at this time.

When modifying a backpressure turbine, the relief valve sizing must be reviewed once the final flow capacity is determined to ensure the exhaust casing is protected from overpressurization. With a condensing turbine, the relief valve or rupture disc sizing must be reviewed to ensure the exhaust casing is prevented from going over the maximum pressure rating, which is normally 5 psig.

The lubrication and control oil system must be carefully checked to ensure the new required oil flow and the proper cooling capacity are available. If the control oil system is being modified, check to ensure the proper oil pressure and flow are available.

### CASE STUDY

This example illustrates the rerate procedure for a large mechanical-drive steam turbine in a process plant. The turbine of interest drives a compressor train in ethylene charge gas service. The compressors were to be rerated as part of a plant expansion, and the as-built turbine power rating was insufficient for the planned compressor rerate.

Necessary turbine performance for the driven equipment modifications is shown in Table 3.

### Table 3. Summary of Performance Changes.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Change (Rerate vs. Original)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated Power</td>
<td>+25%</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>-1%</td>
</tr>
<tr>
<td>Steam Inlet Pressure</td>
<td>No Change</td>
</tr>
<tr>
<td>Steam Inlet Temperature</td>
<td>No Change</td>
</tr>
<tr>
<td>Exhaust Pressure</td>
<td>As Required</td>
</tr>
</tbody>
</table>

There is no extraction steam requirement in this application. The scope of hardware change to affect this magnitude of power increase was beyond that of an efficiency upgrade alone. Steam flow could roughly be expected to increase by the same 25 percent as rated power. The flow limit of each steam path section was evaluated in order to use the existing casing.

Using Equation (4), inlet steam flow, G, is:

\[ S.R. \times P = G \]  
(4)
where S.R. is the steam rate in lb/hp-hr and the rated power, P, is in horsepower. For this rerate, power is 25 percent higher than the original design. Estimated inlet steam inlet flow is:

\[ 7.50 \times 37,180 \text{ hp} \times 1.25 = 348,563 \text{ lb/hr} \]  

(5)

For this particular turbine, the original design specifications and turbine layout were available for reference. As a first step, the engineer prepared a turbine summary shown in Table 4.

**Table 4. Turbine Summary.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frame size</td>
<td>2NV-11</td>
</tr>
<tr>
<td>Rated Power</td>
<td>37,180 hp</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>4200 rpm</td>
</tr>
<tr>
<td>Maximum Continuous Speed</td>
<td>4200 rpm</td>
</tr>
<tr>
<td>Inlet Steam Pressure</td>
<td>600 psig</td>
</tr>
<tr>
<td>Inlet Steam Temperature</td>
<td>750°F</td>
</tr>
<tr>
<td>Exhaust Pressure</td>
<td>2 in. Hg Abs.</td>
</tr>
<tr>
<td>Steam Rate</td>
<td>7.50 lb/hp-hr</td>
</tr>
<tr>
<td>Inlet Flange Size</td>
<td>10” CL900FF</td>
</tr>
<tr>
<td>Exhaust Flange Size</td>
<td>87 inches × 110 inches rect. (100 inches diameter equivalent)</td>
</tr>
<tr>
<td>Exhaust Quality</td>
<td>92%</td>
</tr>
<tr>
<td>Staging Description</td>
<td>Curtis control stage</td>
</tr>
<tr>
<td></td>
<td>11 Rateau stages, all impulse type, 32 inch to 54 inch pitch diameter</td>
</tr>
<tr>
<td>Rotating Blade Heights</td>
<td>2 inch average on the Curtis stage increasing to 16 inches</td>
</tr>
<tr>
<td>Shaft Diameter Between Rotor Disks</td>
<td>12.5 inches</td>
</tr>
<tr>
<td>Nominal Shaft End Diameter</td>
<td>7.0 inches</td>
</tr>
<tr>
<td>Journal Bearings</td>
<td>7 inch diameter, tilt pad, at inlet end 8 inch diameter, tilt pad, at exhaust end</td>
</tr>
<tr>
<td>Thrust Bearing</td>
<td>112.5 square inches</td>
</tr>
<tr>
<td>Main Casing Seals</td>
<td>Labyrinth</td>
</tr>
<tr>
<td>Governor type</td>
<td>Oil relay</td>
</tr>
</tbody>
</table>

The engineer next qualified the turbine casing for this rerate by calculating velocities at the inlet and exhaust flanges. Exhaust pressure for this rerate is permitted to vary, so the engineer assumed a pressure of 4 in HgA. Table 5 shows a comparison of flange velocities with NEMA maximum velocity.

**Table 5. Flange Velocity Estimate.**

<table>
<thead>
<tr>
<th>Flange</th>
<th>Calculated Velocity</th>
<th>NEMA Table 8-1 Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>10” Main Inlet</td>
<td>196</td>
<td>175</td>
</tr>
<tr>
<td>87” × 110” Exhaust</td>
<td>289</td>
<td>450</td>
</tr>
</tbody>
</table>

Inlet velocity exceeding NEMA maximum did not disqualify the turbine casing at this phase of the rerate. However, this area of the turbine would have to be carefully checked after actual performance was calculated.

The engineer proceeded to the next step of selecting a preliminary steam path for the rerate. However, axial space available on the rotor and casing dimensions allowed larger geometry nozzle and buckets only on the Curtis control stage. The same high strength 900 lb nozzle and bucket profiles were retained. Existing staging downstream of the Curtis stage would be retained or replaced as indicated by shop inspection.

Calculations showed that the existing 12-stage steam path could not pass the required additional flow. As flow in the model increases, stage pressure at each successive nozzle row also increases. This takes place in an actual turbine because nozzle area is fixed. In this case, pressure at the first stage nozzle exit rose to the point at which even new larger nozzles became choked.

In general, flow can increase through a certain size nozzle until the resulting pressure drop declines to a critical ratio of approximately 0.6. For ratios higher than 0.6, nozzle flow cannot increase and the nozzle is described as choked.

The only available means of reducing the first stage nozzle exit pressure was to remove nozzles and buckets, starting with the second stage. The engineer determined that the required power could only be achieved by removing the existing second and third stages.

Pressures at each remaining downstream nozzle row were higher than in the original power rating. The diaphragm of the new second stage (original fourth stage) was replaced with a reinforced weldment. Removal of the second and third stages relieved the first stage nozzle exit pressure to an acceptable ratio with inlet pressure. Diaphragms and rotor discs of these stages were replaced with a flow tunnel to smooth the flow in the empty area of the casing. Performance with the new 10-stage steam path is shown in Table 6.

**Table 6. Calculated Rerate Performance.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Calculated</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated Power</td>
<td>46,350 hp</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>4150 rpm</td>
</tr>
<tr>
<td>Rated Steam Flow</td>
<td>336,210 lb/hr</td>
</tr>
<tr>
<td>Inlet Steam Pressure</td>
<td>600 psig</td>
</tr>
<tr>
<td>Inlet Steam Temperature</td>
<td>750°F</td>
</tr>
<tr>
<td>Exhaust Pressure</td>
<td>4 in. Hg Abs.</td>
</tr>
<tr>
<td>Exhaust Quality</td>
<td>91%</td>
</tr>
</tbody>
</table>

The rerate stage selection is compared with the original arrangement in Table 7. Note that stages downstream of the original third stage are unchanged.

**Table 7. Rerate Staging Comparison with Original Staging.**

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.125 in</td>
<td>1.250 in</td>
</tr>
<tr>
<td>2</td>
<td>0.813 in</td>
<td>- - -</td>
</tr>
<tr>
<td>3</td>
<td>0.938 in</td>
<td>- - -</td>
</tr>
<tr>
<td>4</td>
<td>1.125 in</td>
<td>1.125 in</td>
</tr>
<tr>
<td>5</td>
<td>1.500 in</td>
<td>1.500 in</td>
</tr>
<tr>
<td>6</td>
<td>1.875 in</td>
<td>1.875 in</td>
</tr>
<tr>
<td>7</td>
<td>2.375 in</td>
<td>2.375 in</td>
</tr>
<tr>
<td>8</td>
<td>2.450 in</td>
<td>2.450 in</td>
</tr>
<tr>
<td>9</td>
<td>3.600 in</td>
<td>3.600 in</td>
</tr>
<tr>
<td>10</td>
<td>4.170 in</td>
<td>4.170 in</td>
</tr>
<tr>
<td>11</td>
<td>6.500 in</td>
<td>6.500 in</td>
</tr>
<tr>
<td>12</td>
<td>14.76 in</td>
<td>14.76 in</td>
</tr>
</tbody>
</table>
To confirm suitability of this turbine steam path, the engineer conducted a series of limit checks, starting with flange velocities, shown in Table 8.

**Table 8. Rerate Flange Velocity.**

<table>
<thead>
<tr>
<th>Flange</th>
<th>Calculated Velocity</th>
<th>NEMA Table 8-1 Maximum</th>
</tr>
</thead>
<tbody>
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<td>189</td>
<td>175</td>
</tr>
<tr>
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<td>279</td>
<td>450</td>
</tr>
</tbody>
</table>

NEMA SM 23 (1991) maximum inlet velocity may be regarded as a suggested limit. Exceeding this velocity did not disqualify the turbine casing. To provide sufficient flow area in the steam chest and reduce losses, all governor valves were replaced.

Blade stress was acceptable for all rows of buckets. Governor speed range did not change, so blade frequency was of concern only on the new Curtis bucket rows. Evaluation showed that there was no interference with natural modes for these buckets within the operating speed range.

The engineer evaluated the turbine shaft end diameter at the rerate power of 46,350 hp. From Equation (6), we have minimum shaft end diameter, \( d \), equal to:

\[
\frac{321,000}{(P)(S)(N)}^{1/3}
\]

Maximum allowable torsional shear stress for the existing shaft material was 11,500 psi. The equation for diameter then became:

\[
\frac{321,000(46,350)}{(11,500)(4150)}^{1/3} = 3.08 \text{ inches}
\]

The new conditions required a minimum shaft diameter of 6.8 inches. Since the existing shaft end diameter was 7.0 inches, no new shaft or subarc weld buildup of the existing shaft end was required.

**CONCLUSION**

Rerating or reapplying steam turbines can save considerable time and money compared to buying new units. It is usually possible to get more power and/or speed out of existing units by changing items in the steam path. Efficiency can often be improved at the same time.

Older steam turbines can be brought up to present day turbine standards by retrofitting items such as more efficient seals, modern bearings, electronic controls and, monitoring/trending systems. Turbine life can be extended by changing to better materials, adding coatings, and repairing existing damage. These courses of action, however, have more physical constraints and normally require closer coordination between the buyer, user, and vendor than in the purchase of a new machine. Attention to detail in these matters will result in a successful project.

**REFERENCES**


