ROTATING MACHINERY SERVICES,INC.

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THE FINISH LINE

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EXCITING NEWS AT RMS

What's Inside

Letter from the Vice President

Rotating Machinery Services is excited to announce the groundbreaking of our new facility, to be completed in the Summer of 2007. The 15,000 sq. ft. building is located on 3-1/2 acres in a corporate / industrial setting that is conveniently accessed by major highways and nearby airports. Not far from our current location, we will be situated in the Lehigh Valley, Pennsylvania area, less than two hours away from both Philadelphia and New York City.

Expanding into this new space allows us to better serve our customers by providing room for continued company growth, permitting RMS to increase our variety and level of services. Our new facility will include visiting customer office space, expanded meeting areas and in-house training facilities. Besides our new Pennsylvania location, Rotating Machinery Services will continue to maintain our sales office in the Houston area. We will continue to provide updates as construction progresses.

We at RMS wish our customers and suppliers the very best in 2007.

Regards,

Jerry Hallman

2006 Tradeshows

RMS thanks all who stopped by to visit us at the following conferences.

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Industrial Gas Turbine Symposium
Kinder Morgan Supplier Show
35th Turbomachinery Conference
GMC 2006 Conference

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Wishing you and yours a very happy and safe Holiday Season

RULE OF THUMB - GENERAL

By Neal Wikert

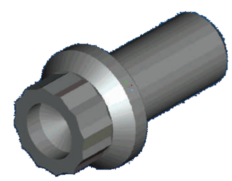
Size	Torque value * (Ft. lbs.)
1/8	
¼ -20	5
5/16 –18	10
3/8 –16	17
7/16 –14	25
½ - 13	40
9/16	63
5/8 –11	85
¾ -10	205
7/8- 9	330
1 –8	495
1 1/8 –7	700
1 ¼ -8	1025
1 3/8	1385
1 ½ - 9	1850
1 5/8	2220
1 ¾ - 8	3020
1 7/8	4020
2 – 8	4600
2 ¼	6560
2 ½ - 8	9260

Size	Keyed Hub	Keyless Hub
1.5	.018”	.048”
2	.024”	.064”
2.5	.030”	.080”
3	.036”	.096”
3.5	.042”	.112”
4	.048”	.128”
4.5	.054”	.144”
5	.060”	.160”
5.5	.066”	.176”
6	.072”	.192”
6..5	.078”	.208”
7	.084”	.224”
7.5	.090”	.240”
8	.096”	.256”
9	.108”	.288”
10	.120”	.320”

Clearances, internal — A good rule to follow for **axial** clearance is 1/4” for hard to hard parts and 1/8” for hard to soft parts.

Fastener – nut removal — To remove hex nuts, heat one of the hex flats until red and break nut loose. The heat effectively loosens the nut and gives room for corrodents to move.

Socket head cap-screw removal: — Heat the head of the cap-screw until cherry red. Allow cap-screw to cool momentarily so that heat can soak down the threads. Turn to remove. Cap-screw will loosen.



STEAM TURBINES - HOW MANY STAGES?

By Sydney Gross

In the last issue, we calculated turbine efficiency based on inlet and exhaust conditions of a hypothetical turbine. You may recall that the inlet was 650 psi and 700°F and the exhaust was 150 psi. Our turbine worked out to be 67.8% efficient when compared to an ideal process.

Now, suppose this turbine is driving a wet gas centrifugal compressor and operates at a design speed of 7,000 rpm. We also know that this turbine has four Rateau stages, each of which has an approximate pitch diameter of 21 inches. The pitch diameter is the diameter at mid-height of the rotor blade. We want to calculate the ideal number of stages for this turbine application and see how it compares to what’s in the field.

With the information above, we can calculate the blade speed, V_b, in feet per second.

$$V_b = \frac{7000\ \frac{\text{revolutions}}{\text{minute}}}{60\ \frac{\text{seconds}}{\text{minute}}} \times \frac{2\pi\ \frac{\text{radians}}{\text{revolution}}\ \frac{21}{2}\ \text{inches}}{12\ \frac{\text{inches}}{\text{foot}}} \qquad V_b = 641\ \frac{\text{feet}}{\text{second}}$$

We know from previous discussions that the ideal velocity ratio, V_b/V_j, is .5 for a Rateau stage. That would make the ideal steam jet velocity from the nozzle for each stage 1,282 feet/second. We also know that the steam jet velocity from the nozzle is proportional to the energy drop across the stage. The equation for calculating the jet velocity from the energy drop is as follows:

$$V_j\ (\text{feet/second}) = 223.7 \times \sqrt{\Delta H_{is}}$$

where ΔH_{is} is the Isentropic Enthalpy drop across the turbine stage in BTU/lbm. If we solve the stage ΔH_{is} using our ideal steam jet velocity of 1,282 feet/second calculated above, we get 32.8 BTU/lbm. You will recall we introduced the Isentropic Enthalpy concept in the last issue and calculated ΔH_{is} for the entire turbine as:

$$\Delta H_{\Delta\text{-Bis}} = 1347.9\ \text{BTU/lbm} - 1200.1\ \text{BTU/lbm} = 147.8\ \text{BTU/lbm.}$$

Dividing the turbine Isentropic Enthalpy drop by the ideal stage Isentropic Enthalpy drop gives us the ideal number of stages for the turbine operating under the specified conditions.

$$147.8/32.8 = 4.5\ \text{stages}$$

We know that we can’t have half a stage so we will settle on 4 stages being less expensive to manufacture and maintain than 5 stages.

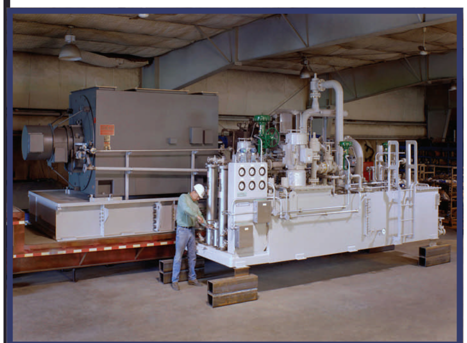
We have been able to match the number of stages that the original manufacturer used in the design of this turbine by using similar methods. The original manufacturer would have started with a speed and power requirement as well as steam inlet and exhaust conditions. Based on the power and

STEAM TURBINES - HOW MANY STAGES? Continued

an approximate efficiency, the necessary steam flow would have been determined. Due to stress limitations, the wheel diameter for the stages would have been limited based on speed. This information would have allowed the manufacturer to select a turbine frame size. The number of stages would have followed as we calculated above. In later issues, we will look at rerate possibilities for this turbine.

Up to this point, we have limited our discussion to the Rateau stage design. However, there is another impulse stage design known as the velocity compounded or Curtis stage which allows us to reduce the total number of stages in the machine at the expense of efficiency. We will look at this design and its advantages in the next issue.

LARGE MACHINERY PROJECTS IN 2006



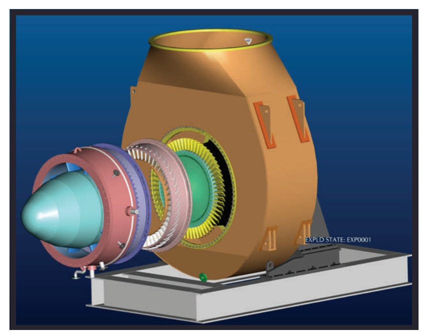
PACKAGED MOTOR, GEAR AND LUBE SYSTEM FOR COMPRESSOR DRIVER RETROFIT

A large Natural Gas Pipeline company currently operates a compressor station with a 12,000 HP gas turbine. With natural gas prices on the rise, our customer decided to replace the gas turbine with a 15,000 HP horsepower variable frequency motor to drive their rerated compressor. RMS manufactured a common baseplate for the motor and gearbox, provided a new common lube and seal system and packaged the train for supply to the jobsite. At the jobsite, a custom designed motor ventilation system will be installed to cool the motor.

The new VFD with its associated switchgear and controls will be prepackaged into a new climate controlled building.

LOW-EROSION RERATE AND REDESIGN OF FCCU EXPANDER

A large U.S. refinery operates an Ex48 FCC expander that was previously rerated in 2001. Because of a change in FCC operating plans, the expander was significantly oversized. In 2005, RMS received an order to rerate this expander for current operating conditions and to redesign the flowpath to minimize erosion.

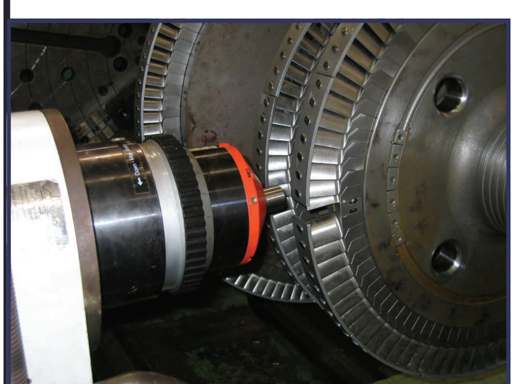


To minimize rotor blade and stator erosion, RMS provided new-technology stator and rotor blade airfoil designs. The RMS designs were optimized for erosion using extensive computational fluid dynamics (CFD) flow and erosion modeling. Our CFD modeling predicts the industry's lowest rotor blade erosion rate for this expander frame size. RMS also incorporated platform seals on the rotor blade to increase stress margins, and to reduce corrosion potential at the critical blade attachment area. To insure adequate cooling, RMS added wheelspace temperature monitoring.

The full upgrade and rerate included the supply of the following major components by RMS:

Rotor Blades	Exhaust Casing
Intake Casing	Integral Stator Shroud
Intermediate Casing	Exhaust Diffuser
Cooling & Sealing Console	

RMS has installed low-erosion technology in Ex38 and Ex48 FCC expanders. This technology can be applied to the full range of installed expanders.



MAJOR UPRATE OF A GENERAL ELECTRIC STEAM TURBINE

RMS has performed a major uprating of a General Electric steam turbine for a mid-West petroleum refiner. The turbine horsepower capability has been increased from 5170 to 7000 as well as an increase speed capability. The uprate was accomplished through a complete redesign of the flow path to allow the necessary higher steam flow to pass through the turbine. Valves, nozzles, diaphragms and rotor blades are replaced while retaining the original steam chest, casing and integral rotor. The rotor shaft end has been redesigned to convert it from keyed to hydraulic and has been welded and remachined through a qualified procedure. Analytical services include flow path aerodynamic redesign, structural and frequency analysis of the new nozzles and blades and rotor dynamic lateral critical analysis. The new components are being installed during a Fall outage.

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"Quality Service from Start to *Finish*"

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We're on the Web!

www.rotatingmachinery.com

If you would like to receive our newsletter via email, please contact Kathy Ehasz at 908-859-8440 or kehasz@rotatingmachinery.com.

35th Turbomachinery Symposium

By Kathy Ehasz

A great time was had by all this year at the 35th Turbomachinery Symposium in Houston Texas. RMS extends our appreciation and gratitude to Martha Brown and all involved in planning, coordinating and preparing this conference. Once again, you all did an outstanding job.

The conference was not only enjoyable, but very educational as well. All areas in the Turbomachinery Industry had such great exposure. We also thank those company's who hosted Hospitality Suites. We had a wonderful time and established many great relationships.

RMS is looking forward to attending the 36th Turbomachinery in September 2007 and also visiting with all our friends.

RMS has some great and exciting surprises in store for the New Year!

Best wishes to all in 2007!