Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45581

NEW LOCKING ARRANGEMENT FOR RADIAL ENTRY TURBINE BLADES

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ABSTRACT

Turbine blades with radial fasteners (T-shank, radial fir-tree, etc.) are commonly used in current steam turbomachinery, especially in power generation applications. However, this reliable and cost-effective design is limited by the strength of the axial pins which lock the closing part in the radial entry slot in the disc. In applications with high speed rotors or heavy blades, the centrifugal force of the blade exceeds the pin strength. In those applications, the airfoil portion of the closing blade is cut off leaving only the bottom portion which is located in the radial entry slot. Some original equipment manufactures (OEMs) also remove the airfoil of the blade 180 degrees opposite for better balancing. The absence of two airfoils is detrimental to efficiency and reliability of the entire row. Siemens Demag Delaval Turbomachinery, Inc. (SDDTI) developed a new locking arrangement which eliminates the above described shortcoming associated with standard radial entry blades. This paper presents the design of the new patented locking arrangement and mechanical stress calculations (FEA) of its major components. In order to verify the validity of the design and calculations, a full-scale row of modernized radial entry blades for an existing US Navy turbine with the new locking arrangement was tested. The testing was done over the full range of operating speeds in a vacuum bunker. The paper also describes the special test rotor, instrumentation used, and the test results which were compared with the stress calculations. The tests confirmed all the advantages of the new locking arrangement and showed acceptable correlation with the stress calculations. The patented design will expand applications for radial entry blades, modernize radial entry blades with missing blade airfoils, and provide a cost-effective method to repair localized cracks in the fastener area of the turbine discs.

INTRODUCTION

In a steam turbine, rotating blades transform the kinetic energy of steam flow into mechanical energy of the shaft rotation. These blades operate in extremely harsh conditions and are subjected to intensive static and dynamic loads.

The harsh operating conditions are as follows:

- High pressure / temperature steam flow with extremely high velocities
- Corrosion and pitting due to steam impurities
- Stress corrosion due to steam contaminations
- Water erosion

The main static and dynamic loads acting upon the rotating blades are:

- High tensile (centrifugal) and bending (aerodynamic) loads
- Significant dynamic loads (due to disturbances of steam flow from exit edges of stationary blades and numerous eddies, wakes, etc., stemming from design, machining, assembly and operation deviations)
- Turbine vibration

It is no surprise that damage / failures of the rotating blades are a leading cause of forced turbine outages; they are responsible for up to one-third of all turbine damages [1,2]. In the USA, blade failures cost large fossil fuel power plants over $200 million per year in maintenance and lost revenue [3]. Therefore, increasing rotating blade reliability (without sacrificing high efficiency) remains a major task for professionals in the turbomachinery field.

Combined experience gained from R&D and operation shows that the optimal design is a 1 x 360 degree blade-disc structure in which all the blades are reliably fastened to the disc and are interconnected to each other by their shrouds. The 360 degree bladed disc structure provides the most reliable solution for withstanding the steady and alternating loads. This is accomplished by significantly increasing the frequency of the structure's response, reducing the number of natural frequencies, and increasing rigidity and damping abilities.[4]

Reliable connection at the top of the blades can be achieved by integral shrouds, which at assembly, form an uninterrupted 360 degree ring. The integrity of this ring can be reinforced in different ways (overlapping loose shrouds, tie-wire, Z-lock engagement, etc.). In addition, the 360 degree uninterrupted shrouding provides better efficiency by reducing windage losses and steam leakages.

The main purpose of this paper is to present a reliable, efficient and cost effective engagement of all the blade fasteners with the rotor disc.

Currently, rotating blades are usually connected to the rotor disc by one of the following major mechanical engagements:

- With radial fasteners or "radial entry blades".
- With axial fasteners or "axial entry blades" which can be straight or curved.

Each of these fasteners has variations in geometry. For example, a radial fastener can be T-shank, external fir-tree, internal fir-tree, etc. (Figure 1) while an axial fastener can be ball-and-shank, axial fir-tree, dovetail, etc. However, the main principle of any mechanical engagement, regardless of the fastener geometry, is that the lugs of the blade fastener are reliably engaged with the mating lands in the rotor disc.

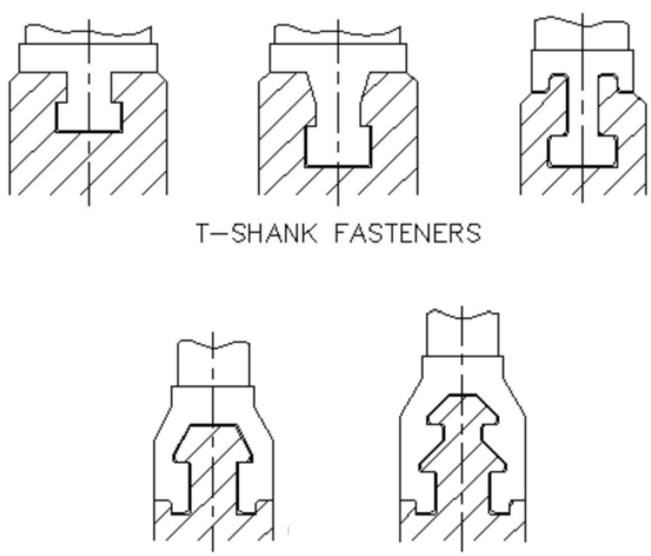

FIGURE 1 – DIFFERENT TYPES OF RADIAL FASTENERS

NOTE: There is another, finger or fork-type fastener, where the disc has one, or several, parallel circumferential grooves on its outer diameter (OD) which are engaged with the fingers on the blade fastener and pinned together with several axial pins. Basically, all the blades in a finger-type row actually operate as closing blades. Since this fastener is extremely cost prohibitive and time consuming to manufacture, it is not widely used nor will it be discussed in this paper.

The main distinguishing features of the blade-to-rotor engagements as described above are as follows:

1) <u>Radial Fastener Design</u>:

   a) All blades are assembled to the disc in a single circumferential groove (or on a circumferential rib), and kept in tight contact with each other by the platforms and fasteners.

   b) In order to install the blades into this groove (or on the rib), a radial entry slot is made in one location (or two opposite locations) of the disc rim. In the radial entry slot, all of the male lugs in the groove / rib are removed flush to the other remaining surfaces. The tangential width of the radial entry slot should slightly exceed the blade pitch at the root.

   c) After assembly of all the regular blades onto the disc, these blades are locked with the so called closing blade or closing piece.

      The closing blade has the same airfoil and platform as a regular blade, but differs by its fastener. The closing blade fastener is a solid block (without lugs / depressions) which fills all of the space of the radial entry slot with very small gaps under 0.001" (.025 mm). This fastener, together with a platform, must provide tight contact with the platforms and fasteners of the blades which are adjacent to the closing blade. The closing piece consists of only the closing blade platform and fastener (no airfoil).

      After final adjustments, the closing blade or closing piece is locked with the axial closing pins. (Figure 2)

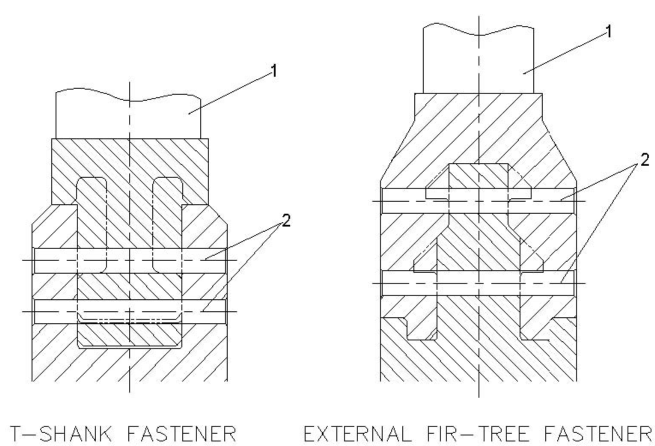

FIGURE 2 – EXISTING LOCKING ARRANGEMENT WITH CLOSING BLADE / CLOSING PIECE

2) <u>Axial Fastener Design</u>:

   a) Each blade is installed into its individual axial slot which is milled or broached in the disc rim of the rotor.

   b) There is no contact between the platforms and fasteners of any adjacent blade in the row.

   c) There is no need for any special locking device for preventing blade movement in the radial direction.

Advantages and Disadvantages of Blade Fasteners

Every type of blade fastener as described above has its own advantages and disadvantages.

Advantages of the radial fastener design:

1) Manufacturing of the blade fasteners and grooves / ribs in the rotor is very cost-effective because it does not require special cutting tools, fixtures and control devices.

2) In most applications, there are no limitations or special requirements for blade pitch, allowing use of the optimal quantity of blades in each row.

3) The radial fastener does not require machining tolerances as tight as the axial fastener. Therefore, the design is forgiving to deviations in the disc groove. This is a very favorable feature for operation and maintenance, since it is possible to clean up the radial groove / rib and to replace damaged blades with the standard spare blades.

Disadvantages of the radial fastener design:

The following major drawbacks of the radial fastener design are a result of the radial entry slot:

1) Limited applications of the rows with the closing blades: As described above, the closing blade is locked into the entering slot by the axial closing pins which are installed with a tight fit into drilled and reamed holes. These pins are the most stressed component of the entire row, since they withstand the centrifugal force of the whole blade (instead of the lugs / mating lands of the standard mechanical fastener). High stresses in the pins limit the application of the closing blades to rows with small blade heights and / or moderate rotor speeds.

2) Complexity of assembly and reblading: Replacing any blade in a row requires removal (as a minimum) of all the blades starting from the closing blade/piece up to the damaged blades.

3) Limited rebladings over the rotor lifetime: During long term operation, blades deteriorate due to different operating factors (direct blade failure, axial / radial rubs, FOD damage, pitting, erosion, corrosion, etc.). This deterioration often requires re-blading. Each re-blading necessitates re-drilling and re-reaming of the existing holes for the axial pins, resulting in a larger hole diameter and a shorter distance between the holes and the disc outer diameter. After several re-bladings, the enlarged hole can lead to cracking of the disc rim or the closing blade / piece.

4) In applications with taller blades and / or high rotor speeds, the closing pins cannot withstand the centrifugal force of the closing blade. Therefore, a closing piece must be used instead of a closing blade. Centrifugal force of the closing piece is a fraction of the closing blade force, allowing the use of closing pins. However, in order to balance the centrifugal forces within this type of row, a filling piece is installed 180 degrees apart from the closing piece. The filling piece consists only of the regular blade platform and fastener (no airfoil). Therefore, centrifugal forces of the closing and filling pieces are basically equal to each other. The use of a closing / filling piece combination decreases the number of blades in the row by two (2) and leaves two (2) openings in the steam path (i.e. airfoil) area. This negatively affects the row performance in the following ways (Figure 3):

FIGURE 3 – VINTAGE LP ROTOR FROM THE US NAVY AS-39 MAIN PROPULSION TURBINE

a) Due to the two openings, it is not possible to complete the most reliable blade structure (1 x 360 degree). In addition, these two (2) openings disrupt uniform steam flow through the stage and induce turbulence, wakes, and eddies. Blading reliability is significantly affected by these turbulent wakes.

b) There is a significant decrease in efficiency / output of the row, since the area of these two (2) openings is a minimum of 6 – 10% of the total steam passing area. However, steam leakage through these openings actually exceeds the above percentage because the flow coefficient through this area is significantly higher than the flow coefficient through the blade throat openings.

Advantages of the axial fastener design:

1) High load carrying capacity: Due to the blade fastener geometry, the load carrying area of the axial fastener is substantially larger than the area of the radial fastener, for the same size of blade airfoil. Therefore, the axial fastener can be used in all applications without any limitations in reliability.

2) A row with axial entry blades does not have a radial entry slot and therefore, avoids all of the associated shortcomings. Compared to radial entry blades, axial blades are simpler, less time consuming and more cost effective to assemble and replace.

3) Simplicity of blade replacement: Contrary to the radial entry blading, the scope of axial blade replacement is much simpler for rows with tie-wire or grouped loose shrouds. In rows with tie-wire, only the damaged blading and tie-wire needs to be replaced. In rows with grouped loose shrouds, only the group with the damaged blading needs to be replaced. In both cases, the undamaged blading (expect those in the same shroud group) remains installed.

Disadvantages of the axial fastener design:

1) Machining the axial slots and manufacturing of the blades is higher in cost and requires special cutting tools, measuring tools and fixtures. In addition, during machining of the slots, there is always a possibility to incur machining errors, which cause further costly efforts for correction.

2) Axial slots require significant space in the tangential direction, which limits the quantity of the blades on the row. In many rows, the blade quantity is less than the optimal number, which negatively affects efficiency.

3) Blading with axial fasteners is much less tolerant to deviations, or re-machining for slot cleaning during blade replacement, compared to blading with radial fasteners.

THE MAIN GOALS OF THE PROPOSED NEW LOCKING ARRANGEMENT

The main goal of the proposed design is to create a row of rotating blades which will possess the combined advantages of both major types of fasteners and, at the same time, be free from their disadvantages as described previously. The new design will achieve the following practical goals:

1) Expand the usage of the blading with radial fasteners, as being the most cost effective and tolerant design, in producing new turbines.

2) Upgrade the numerous fleets of existing vintage turbines with radial fastener blading, thereby, substantially improving reliability and efficiency.

3) Reduce the cost of repairing cracks in the disc rim. Radial cracks can initiate in the radial entry slot, between pin holes and / or between the pin holes and disc OD. Circumferential cracks can also initiate in the corners of the fastener with small radii, which are high stress concentration areas. If the depth of the radial cracks does not extend beyond the bottom of the fastener, and the circumferential crack is located within one blade pitch, successful repair can be provided by the new locking arrangement (instead of expensive welding). These cracks can be removed by machining the axial entry slot for the new locking arrangement in this affected location.

THE UNIQUE DESIGN FEATURES OF THE NEW LOCKING ARRANGEMENT

The goals mentioned above are achieved by using a row with the standard radial entry blades with a new locking arrangement which utilizes a proven standard axial entry fastener. This new innovative locking arrangement can be designed in different ways: using a special single closing blade only or a standard blade with special insert, etc. (as described in the patent).

The proposed new locking arrangement was patented in the USA [5], Mexico, Canada [6], and several European countries [7].

This paper presents one of the possible new locking arrangement designs that was developed and tested for upgrading an existing low pressure (LP) rotor (Figure 4). In the existing row of radial entry blades with a T-shank fastener, the existing locking arrangement consisting of a closing piece and filling piece combination was replaced with a special single closing blade and two (2) adjacent blades. This closing blade has the same top (tenon or integral shroud) and airfoil as the regular blade. However, its platform and fastener are very different. The platform of this closing blade extends from the blade base diameter down to the bottom of the radial groove. A standard Delaval ball-and-shank axial fastener is located below this platform (Figure 5). The contact faces of the platform with the adjacent blades are flat, while the inlet and exit sides completely match the geometry of the final inlet and exit faces of the assembled and machined disc rim.

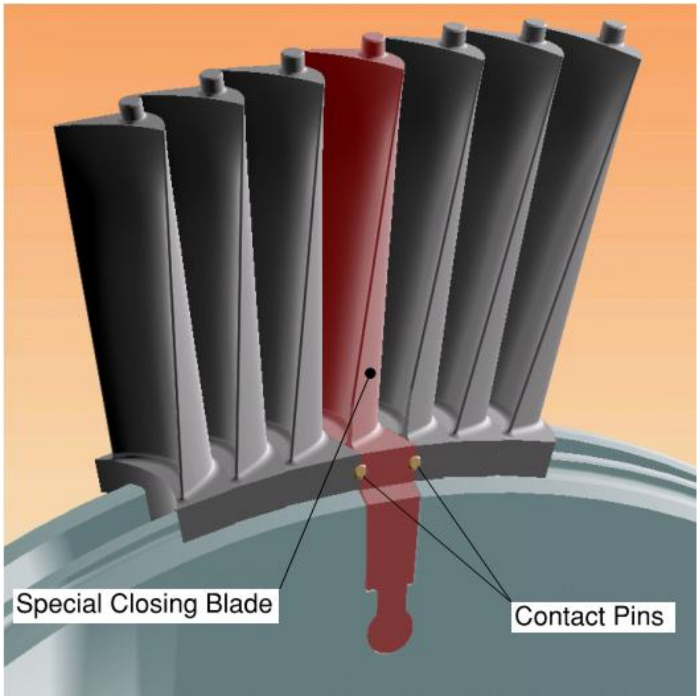

FIGURE 4 - NEW LOCKING ARRANGEMENT FOR RADIAL ENTRY BLADES WITH T-SHANK FASTENER

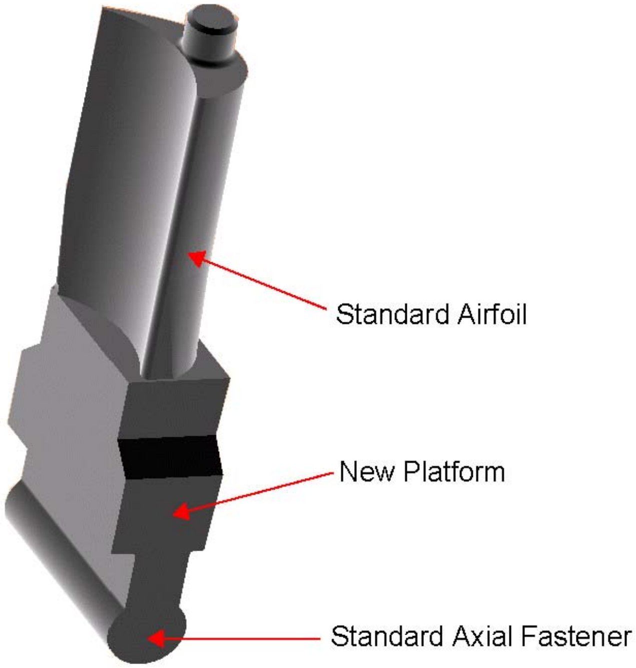

FIGURE 5 - NEW SPECIAL CLOSING BLADE

The width of the platform in the tangential direction (i.e. direction of rotation) is slightly larger than required per design, in order to provide stock for dressing this platform to the adjacent blades for zero or minimal clearance.

Adjacent blades provide transition from the new closing blade to the regular blades. Therefore, the geometry of each adjacent blade is different from that of a regular blade. One face of the adjacent blade's integral shroud, platform and T-shank fastener, which is in contact with the closing blade, is flat. The opposite face of each adjacent blade, which is in contact with the regular blade, repeats the regular blade curved configuration.

The new axial entry slot in the disc rim has a completely different geometry than the radial entry slot. The new axial entry slot spans across the entire thickness of the disc rim consisting of two portions, upper and lower. The upper portion of the new axial entry slot starts from the disc outer diameter and extends in the radial direction to the bottom of the circumferential groove in the rim. The tangential width of this portion is equal to the width of the existing radial entry slot. The lower portion, which is actually a standard ball-and-shank configuration, starts from the bottom of the upper portion and extends further into the disc. In general, the type, geometry and size of this fastener depends upon different variables: blade geometry, rotor speed, width of the upper portion of the slot, standards of the original equipment manufacturer (OEM) or repair facility, etc. (Figure 6).

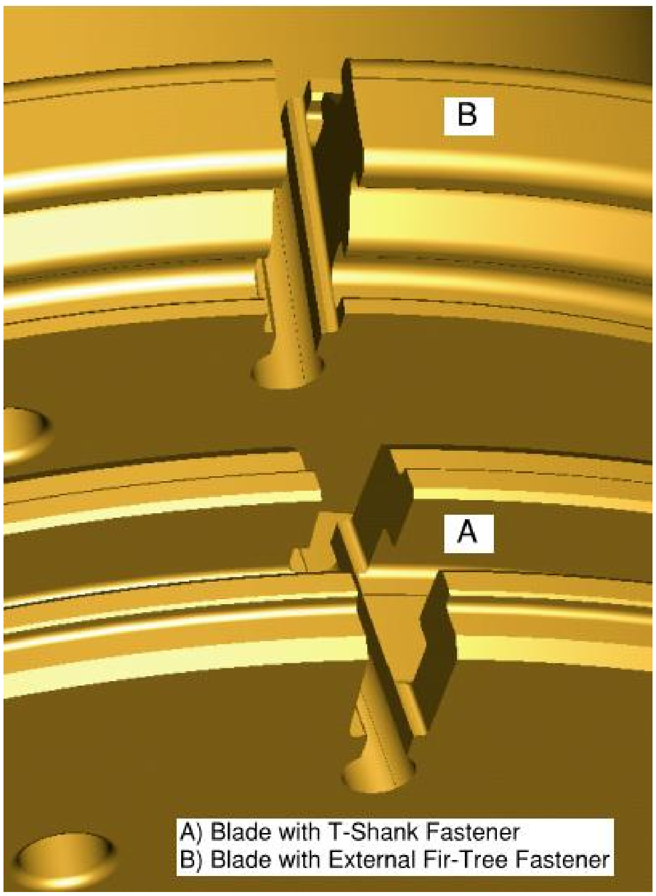

FIGURE 6 - NEW AXIAL ENTRY SLOTS IN THE DISC RIMS

The orientation of the axial entry slot relative to the disc faces can also be different, depending upon the configuration of the platform and fastener of the regular blade (which is defined to accommodate the airfoil profile at base diameter):

- Strictly perpendicular (which is the most favorable orientation)
- Deviated from perpendicularity (usually by up to 15 degrees)
- Repeats the curved configuration of the regular blade platform / fastener (this is an extremely complicated, expensive and time consuming design).

After installation of the final dressed closing blade into the slot, its reliable contact with the adjacent blades is secured with the axial contact pins which are located above the disc outer diameter in the platform near the base diameter. These pins should be reliably staked around their circumference. This staking secures the pins in their holes as well as the new closing blade in the proper axial position. Contrary to standard locking arrangements, the pins do not carry centrifugal load (axial fasteners carry load) but only provide reliable contact between blades and secure the closing blade in axial position.

MECHANICAL CALCULATION

The calculation portion of the analysis is accomplished by the use of two finite element programs. The first is a steam turbine blade specific program. This generates the model for a regular blade in the row. The model is composed of a single sector of the wheel containing the blade, disc and shrouding (Figure 7). The second finite element program is a general-purpose code. This is used to model the fully bladed wheel. It is also used to run the required analyses (steady stress, modal and forced response). For the purposes of this project, the mechanical analysis only included the steady stress analysis of centrifugal loading. Blade steam loading, modal analysis and forced response are not part of the mechanical calculation or running test.

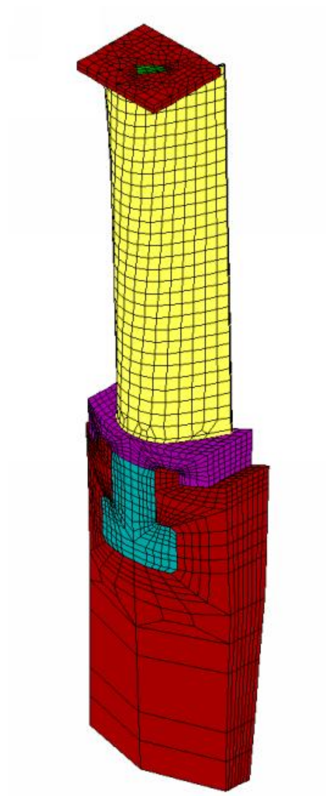

FIGURE 7 - FEA SINGLE SECTOR BLADED WHEEL MODEL

The fully bladed wheel model (Figure 8) is composed of a five-blade group (Figure 9) and the remaining blades. The five-blade group contains the closing blade, two special adjacent blades and two flanking regular blades. The remaining blades are standard and are included by using a modeling technique called superelements. Superelements reduce the size of the analysis by replacing the thousands of elements in the typical single sector blade model (Figure 7) with a single element.

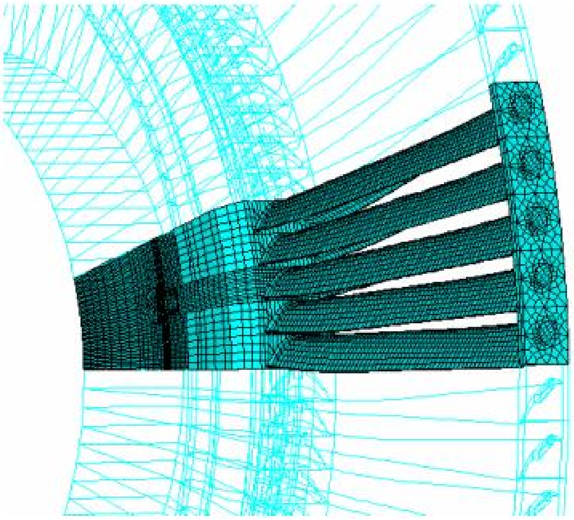

FIGURE 8 – FEA FULLY BLADED WHEEL MODEL

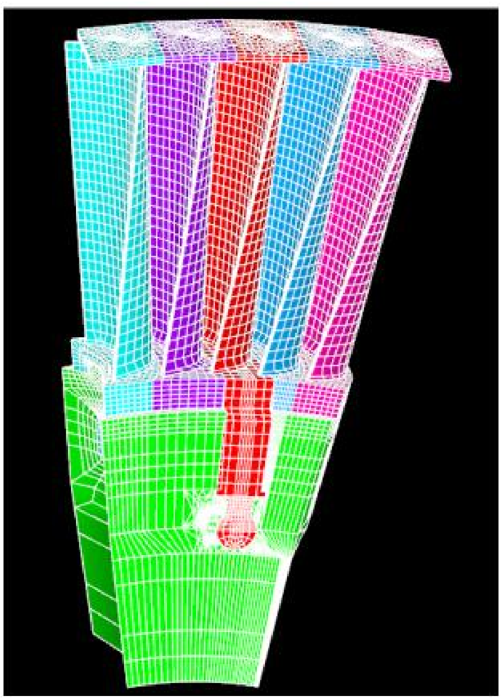

FIGURE 9 - FEA 5-BLADE GROUP BLADED WHEEL SECTOR MODEL

One additional model was created to complete the analysis. This dealt with the possible interaction between the closing blade slot and the disc steam balance holes (Figure 10). A separate model was necessary because the steam balance hole pattern is different than the blade pattern. This steady stress analysis showed that the local stress pattern around the closing blade slot does not interact with the local stress pattern around the balance hole (Figure 11).

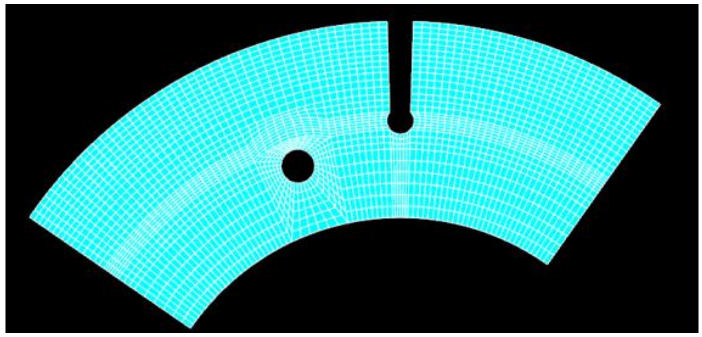

FIGURE 10 - FEA DISC AXIAL ENTRY SLOT AND BALANCE HOLE

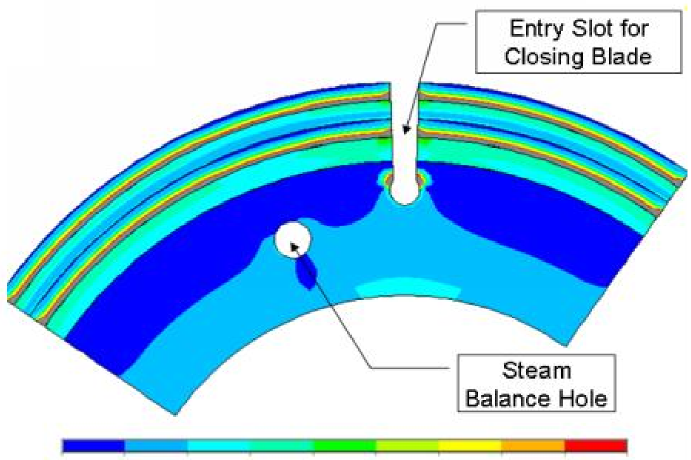

FIGURE 11 - FEA STRESSES IN THE DISC AREA WITH AXIAL ENTRY SLOT AND BALANCE HOLE

Results of Calculations

- Stresses are within design allowables
- There was no interaction between steam balance hole and axial entry slot (Figure 10)
- The 360 degree shrouding arrangement experienced significantly lower dynamic stresses as illustrated by a typical Goodman diagram (Figure 12)
- Centrifugal stresses in the blade airfoil for the three types of blades in the test row (regular, adjacent, and closing) are different despite having the same airfoil section. This can be explained by the differences in geometry of the platforms and fasteners, and by the transfer of loading from one blade to another via the 360 degree interconnected shrouding.

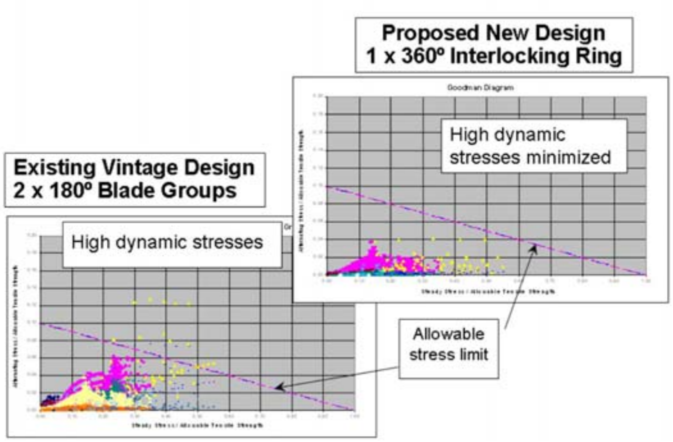

FIGURE 12 – TYPICAL GOODMAN DIAGRAM

PREPARATION FOR TESTING

Considering the innovative design of the new locking arrangement, complicated mechanical strength calculations (especially for the possible interference between equalizing holes and a new axial entry slot in the disc), and a wide range of possible implementation of this design into practice, it was decided to conduct extensive testing of the new locking arrangement in an actual row of blades with operation rotor speeds in a vacuum bunker.

Row # 5 (L-1), of the existing main propulsion turbine for the AS-39 US Navy ship, was selected for testing. Listed below are the major design features:

1) The row consists of eighty-eight (88) blades with a T-shank fastener airfoil height of 6.500" (16.51 cm) on a 27.8" (70.61 cm) base diameter, used for Delaval radial entry blades.

2) The locking arrangement in this row is a standard closing piece / filling piece combination.

3) The blades contact (nest) to each other by a curved platform / fastener configuration.

4) The US Navy considered this row a limited life operation component due to previous failures during 35+ years of operation.

5) The blades are connected on the top by loose shrouds in five (5) blade groups. Best case scenario for this row with a closing and filling piece is a 2 x 180 degree grouping only.

In order to improve reliability, efficiency and extend the operational life-time, we proposed to upgrade this row by creating a rigid disc-blading structure by:

- Replacing the existing closing piece and filling piece locking arrangement with a new design.
- Implementing an integral shroud (instead of loose shrouds)
- Significantly improving blade connection at their tops with the 1 x 360 degree uninterrupted shrouding, which is reinforced by a combination of 3 overlapping laminated shrouds and loose shroud.

As mentioned earlier, the finite element analysis (FEA) confirmed the dramatic increase in reliability and longevity of the upgraded row.

The main purposes of the testing were to:

1) Confirm the feasibility of manufacturing and assembly of the upgraded row of blades with the new locking arrangement.

2) Measure the actual strain / centrifugal stress levels at critical locations in the blades and disc.

3) Compare the measured centrifugal stresses in critical blades and disc locations with the results of the mechanical calculations (FEA).

Availability of the vacuum bunker for testing was severely limited. Therefore, in order to accumulate the maximum possible test data, it was decided to install two (2) separate closing arrangements on the same row (located 180 degrees apart) and test them simultaneously.

EXPERIMENTAL ROW OF BLADES

Each closing arrangement was fit into a "testing blade group", which consisted of five (5) blades:

- One (1) special closing blade – in the middle of the group
- Two (2) adjacent blades – on each side of the special closing blade
- Two (2) regular blades – each contacting with the adjacent blade.

The blades in each testing group were instrumented, as shown on Figure 13.

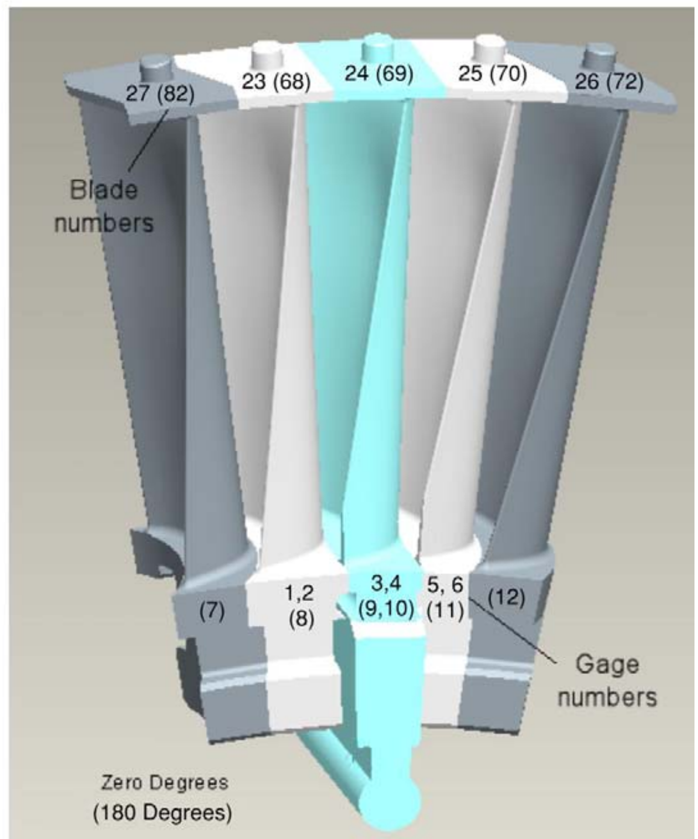

FIGURE 13 - TESTING BLADE GROUPS

The entire testing process was a highly complicated technological endeavor, which involved: creating a test shaft, assembling an experimental row of blades, instrumenting them with multiple strain gages, tracing the wires from the gages to the data acquiring system through the rotor and vacuum bunker and, finally, conducting the test with rotor speeds up to 5,000 rpm.

The testing was conducted by Sensing Systems Corporation (SSC). SSC is renowned for stress measurements in different power plant components. The testing program consisted of measuring strain at twenty-four (24) critical locations of the row, sixteen (16) locations on the blades and eight (8) locations on the disc.

EXPERIMENTAL ROTOR MACHINING

A shaft from a decommissioned turbine was refurbished for the testing in the following way (Figure 14):

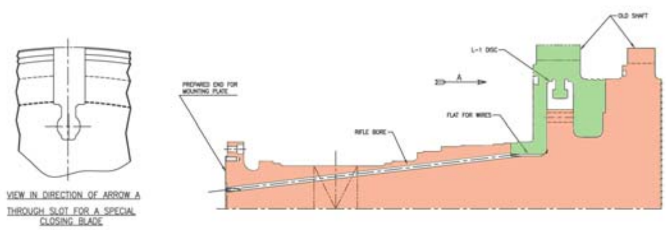

FIGURE 14 - EXPERIMENTAL ROTOR - PRE-MACHINING FOR BLADES ASSEMBLY AND INSTRUMENTATION

- The shaft (from the last disc to the coupling) was re-machined to repeat the same geometry as the original US Navy L-1 disc for the tested row of blades.
- Two (2) diametrically opposite (180 degrees apart) axial entry slots were milled in the disc rim, for the special closing blades of the subject new closing arrangements.
- Two (2) diametrically opposite "rifle bores" were water-jet drilled in the same planes at the axial entry slots. These 0.500" ± .001 / .000 (12.7 mm ± .025 / .000) diameter bores started from the end face of the shaft, oriented at 7 degrees to the shaft center line and extended approximately 35.0" (88.9 cm) to the face of the test disc.
- Two (2) rectangular flat surfaces 3.50" x 3.75" (88.9 mm x 95.25 mm) each were milled in the shaft at the base of the disc inlet face, at the same locations as the rifle bores. Their depth was such to expose the rifle bores at that shaft location.
- Numerous grooves .093"(2.36mm) x 60 degrees each were ground in the inlet disc face. These grooves were traced radially from each strain gage down to the flats.
- The shaft coupling end was re-machined for accommodation of the special mounting plate which contained all of the necessary rotating instrumentation equipment.
- The opposite (thrust) end of the shaft was re-machined for the "drive end detail" which served as an adaptor for the drive shaft of the vacuum bunker.

EXPERIMENTAL ROTOR ASSEMBLY

The assembly of the experimental row (containing the 2 "test blade groups") onto the rotor was accomplished in the following manner:

- All the radial entry blades were installed as typical blades with T-shank fasteners in 2 x 180 degree groups, with the adjacent blades as the first and last blade of each group.
- Each adjacent blade extended into the axial entry slot by approximately 0.020 – 0.030" (0.50 mm - 0.75 mm).
- The remaining space between the adjacent blades was carefully measured in the axial slot and between the integral shrouds.
- Each closing blade was accurately machined to these dimensions, in order to provide free passage into the axial slot with zero or minimal clearances at the contact areas of the blade platforms and shrouds.
- It was necessary to slightly trim a small portion of the inlet edge on either the closing blade or adjacent blades, which is a normal procedure for axial entry blade assembly.
- Two axial holes were drilled and reamed at the closing blade-to-adjacent blade joint, near the top of the platform. Axial pins were inserted into these holes to provide reliable contact between the closing blade and adjacent blades and to fix the closing blade in the proper axial position.
- A set of three laminated overlapping and loose shrouds were installed onto the integral shroud and taumel peened to provide a reliable 360 degree interconnection of all the blades in this row.

INSTRUMENTATION

The instrumentation equipment consisted of the following items:

- Strain gages bonded to the blades and the disc
- Slip rings
- Amplifiers and completion resistors
- Mounting plate
- Wiring between: strain gages and amplifiers; amplifiers and slip rings; slip rings and data acquisition system

Strain Gages

Twenty-four (24) single uni-axial strain gages model EA-06-062AQ-350/e (from Vishay) were selected for use in each critical location. Since the signals generated by these gages are low level (in the millivolt range), amplifiers were used to condition and increase the signal level to a range of ± 5 volt direct current (VDC). A total of eight (8) amplifiers with three (3) channels each was enough for twenty-four (24) strain gages. Amplifier model AMP-SG3-U2 was designed and manufactured by Michigan Scientific Corporation for use in high G-force environments. The high level signals from the amplifiers were sent through the slip rings to the data acquisition system.

Precision Resistors

Also, since the single uni-axial strain gages (¼ bridge) were used, three (3) completion resistors were required for each strain gage channel to complete the circuit. So, a total of ninety-six (96) 350 Ω precision resistors were used to complete the twenty-four (24) Wheatstone bridges.

Special Mounting Plate

Completion resistors, amplifiers, and slip rings were assembled in the special mounting plate. This mounting plate with limited dimensions, was populated with eight (8) amplifiers of three (3) channels each, ninety-six (96) completion resistors, three (3) slip rings and all the interconnecting wires.

Wiring

The wires between the strain gages and amplifier were subjected to the highest loads (G-forces) during the test and therefore, were very vulnerable to mechanical damage. Their proper routing and protection were critical for success of the entire testing. The strongest (34 gage) magnet wire leads were used for cabling. They were encapsulated throughout their entire length. Each wire from the strain gages was located in the special grooves ground in the face of disc, and then routed down one (1) of two (2) flats in the shaft near the disc face. On each flat thirty (30) wires were assembled into one (1) tight bundle which was encapsulated and placed inside of a tube located in the "rifle bore". From the "rifle bore" each wire was attached to the proper amplifier located in the special mounting plate bolted on the rotor coupling. All the exposed magnetic wire leads, the grooves, terminals, amplifier posts and slip ring posts were encapsulated with GA-2 adhesive (by Vishay). The encapsulation locked the wiring in place and protected it during rotor operation. The electrical connections between the amplifiers and slip rings were also made from (34 gage) magnet wire. Three (3) individual cables were connected to the stator of each slip ring. The stators of all the slip rings were secured to a bunker structure by C-clamps. The three (3) cables from the slip rings were routed inside the bunker to a penetration point in the wall. The cables were epoxied inside the removable penetration point to ensure tight fit in order to hold vacuum inside the bunker during tests.

DATA ACQUISITION SYSTEM

The data acquisition system consisted of hardware equipment and software programs.

Hardware Equipment

Hardware equipment included the following items:

- Terminal Panel – used to connect cables to the data acquisition equipment and power supply.
- Power Supply – Model 72-8142 (from Tenma) operated on 11 VAC and supplied a regulated 13.8 VDC with 10 AMP maximum service.
- Amplifier Control Unit – Model PS-DC (from Michigan Scientific Corporation) powered the amplifiers with ± 15 VDC and controlled their operation. It also provided shunt calibration to check each strain gage channel prior to testing.
- Data Logger – A CR5000 Model (from Campbell Scientific) acquired voltage signals from amplifiers from all the strain gage channels for a period of 30 seconds at a rate of 50 samples/sec. for each of strain gage channel, totaling 1,500 samples of data for each channel at each test condition. The data logger was controlled with a laptop computer through its own communications protocol.
- Laptop Computer – A laptop computer running Windows XP was used for a controlled acquisition of all the test data from the data logger. The laptop was also used for down loading of data.

Software Programs

The following three (3) software programs were used to obtain, decipher, store and present test data:

- CR5000 Data Acquisition Program - written by Sensing Systems Corporation. This program acquired the test data, specified the parameters of the laptop computer to control test data, and specified the data format (ASCII) and storage destination to be used by the CR5000 Data Logger.
- RTDAQ – This windows based software package was bought together with the CR5000 and was used to store the test data and to transfer it to the laptop computer.
- Microsoft Excel – Using two (2) templates in Microsoft Excel, the raw electrical test data (in millivolts) was normalized into the mechanical strain engineering units (microinch/inch).

TESTING

The completely assembled and instrumented rotor is presented in Figure 15, Figure 16, and Figure 17.

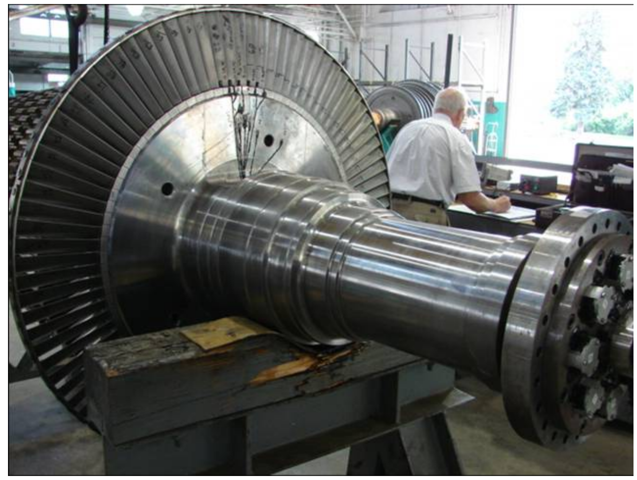

FIGURE 15 - COMPLETELY ASSEMBLED AND INSTRUMENTED ROTOR

FIGURE 16 - TEST WHEEL INSTRUMENTATION (DETAIL)

FIGURE 17 - MOUNTING PLATE WITH INSTRUMENTATION ON THE ROTOR END (DETAIL)

Testing was performed at Siemens Demag Delaval Turbomachinery, Inc. (Hamilton, NJ) facility over the course of two (2) days. Each test was conducted at progressively increasing rotor speeds within the range of 500 – 5,000 rpm in accordance with the test procedure. Following the finish of a complete sequence, the tests were performed two (2) more times allowing repeatability to be confirmed.

The test environment was recognized as very challenging, since high acceleration and deep vacuum subjected all the instrumentation components to inhospitable conditions in which they could fail. All strain gages, except for interconnecting wiring (magnet wire leads), were functioning properly before and after testing. Some wiring failures occurred at locations where relative movement of components took place at very high rotational speeds. Also, some deviations from the test procedure were made to prevent overheating of the slip rings at very high rotor speeds.

Acquired data was reviewed for repeatability, consistency and magnitude. The review was performed immediately after acquiring the data and again, after the completion of all testing. This review showed that trends and magnitudes were consistent and repeatable within ± 5% from one run to the next for each channel. A comparison of data received from the testing blade groups also showed consistency of the data between similar components.

Strain levels of different gages were slightly different at rotor speeds up to 3,353 rpm (40% of maximum acceleration). However, starting from this speed and higher, all strain levels became very similar (regardless of test sequence) in magnitude with very good repeatability as rotational speed increased.

TEST RESULTS

Test results were obtained as steady centrifugal stresses in the chosen blades and disc locations (where each strain gage was attached) at different rotor speeds. Stress data for each strain gage is given in pounds per square inch (psi). All gage locations were assumed to undergo uni-axial stresses. The measured strain values were converted to stresses by multiplying the measured strain by the modulus of elasticity E = 29.7 x 10^6 psi (204.77 GPa).

Graphically, all these test results are presented on Figure 18 through Figure 23 as a function $\bar{S} = f(\bar{n})$; where: $\bar{S} = \frac{s}{[\sigma]}$ and $\bar{n} = \frac{n}{n_r}$; s is measured (or calculated) steady stresses; $[\sigma]$ is allowable steady stresses at this location; n is rotor test speed; n_r is rated rotor speed.

The steady stresses obtained by testing are presented together with the calculated stresses in the same locations. As shown in Figure 24, there is satisfactory correlation (within 20%) between measured and calculated steady stresses. Test results confirmed FEA findings that centrifugal stresses in the blade airfoil depend upon geometry (rigidity) of the platform and fastener design. The higher airfoil stresses are in the blades with the more rigid platform and fastener. Therefore, closing blades have the highest airfoil stresses among all the blades in the row. It is worth noting that deviations in closing blade position due to assembly variables (which can't be calculated) also contribute to higher stresses in the closing blade airfoil.

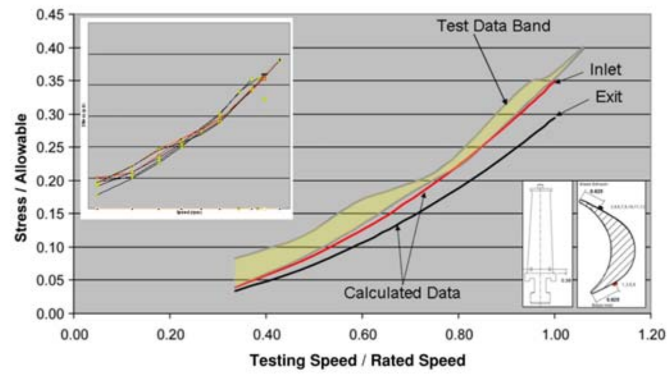

FIGURE 18 - STEADY STRESSES IN NEW CLOSING BLADE AIRFOIL

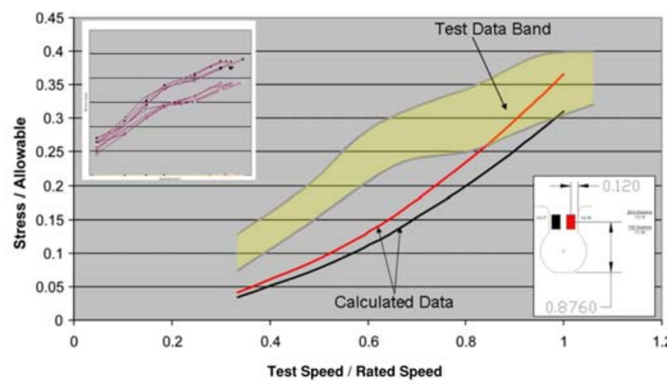

FIGURE 19 - STEADY STRESSES IN NEW CLOSING BLADE FASTENER

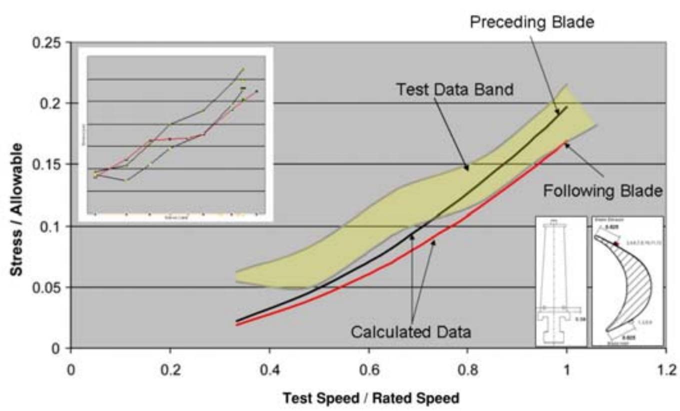

FIGURE 20 - STEADY STRESSES IN ADJACENT BLADE AIRFOIL

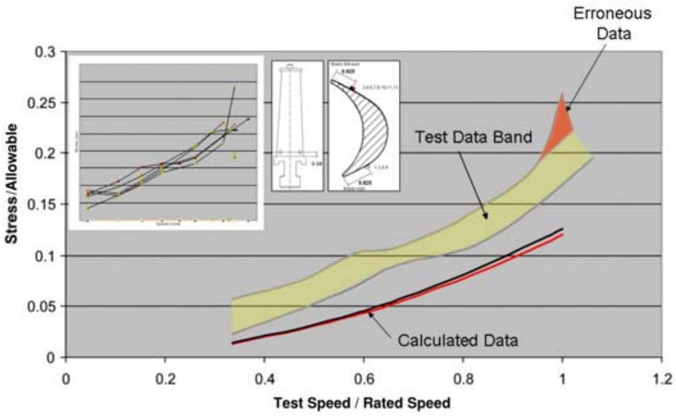

FIGURE 21 - STEADY STRESSES IN REGULAR BLADE AIRFOIL

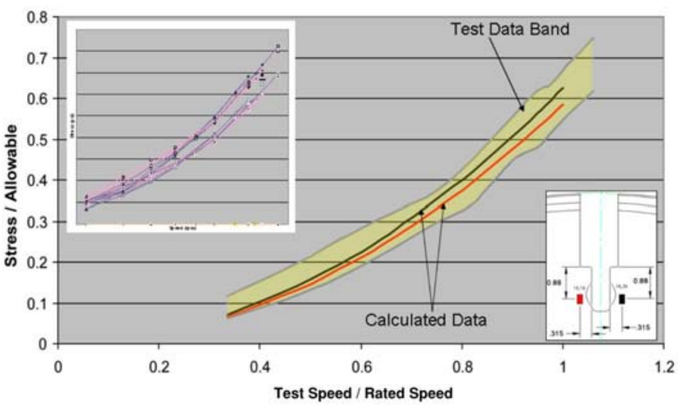

FIGURE 22 - STRESSES IN THE DISC AXIAL ENTRY SLOT

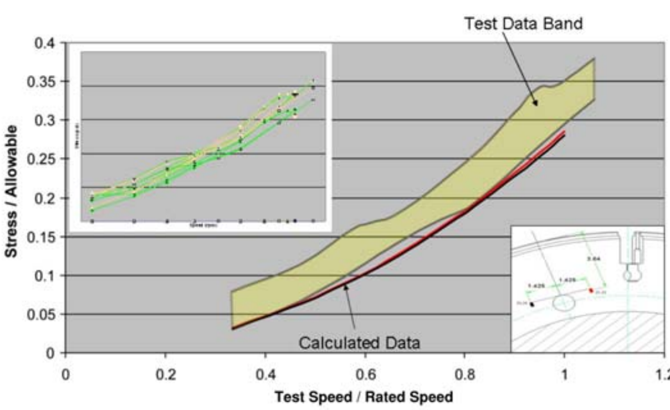

FIGURE 23 - STRESSES IN THE DISC BALANCE HOLE AREA

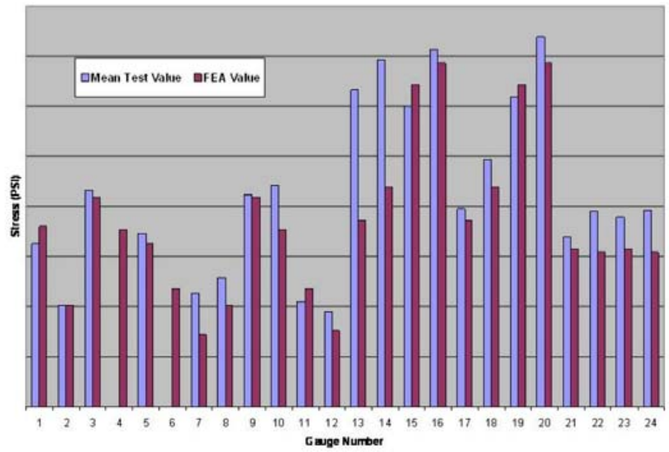

FIGURE 24 - COMPARISON BETWEEN CALCULATED (FEA) AND TEST STRESSES AT 5000 RPM ROTOR SPEED

The manufacturing of the experimental row of blades with the new locking arrangement, its assembly onto the disc and successful testing has proven its technological feasibility and reliability.

CONCLUSIONS

1) The row of radial entry blades with patented closing arrangement utilizes advantages of both radial and axial fasteners, and results in significant improvements in efficiency and reliability.

2) The new closing arrangement can also be used as a cost effective and time saving method to repair rotors with radial cracks in the disc rim (if the crack length does not exceed the blade pitch and fastener depth).

3) The testing of the row of full scale blades with the proposed arrangement at the operating speed range 0 – 5,000 rpm proved the feasibility of the patented design, i.e.:

   a) It is relatively easy to manufacture and assemble.

   b) Steady centrifugal stresses in all the critical blading and disc areas are well within the design limits.

   c) There is no negative effect on the steady centrifugal stress in the disc due to the position of the equalizing holes relative to the closing arrangement entry slots.

   d) The steady centrifugal stresses in the closing and adjacent blade airfoil are higher than in the regular blade due to differences in geometry of the platform and fastener as well as assembly variables.

   e) There is satisfactory correlation (within 20%) between the FEA calculated and measured steady stresses.

4) The patented locking arrangement is recommended for implementation into new and modernized units.

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ACKNOWLEDGEMENTS

The authors wish to express their gratitude to the following co-workers for their valuable assistance in preparing and organizing this paper: James E. Luker, Carolyn B. Smith and Marcelle R. Seals.