Proceedings of the Second Middle East Turbomachinery Symposium
17 – 20 March 2013, Doha, Qatar

Modernization of Steam Turbine Diaphragms for the Saudi Aramco Gas Plants

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Mr. Lipski has a B.S. degree (Mechanical Engineering Technology) from Trenton State College.

ABSTRACT

The critical components of the Saudi Aramco Uthmaniyah and Shedgum gas plants are eight trains of turbomachinery, 4 trains per plant. Each train consists of a steam turbine and a centrifugal compressor. At the time of their design and manufacture, these steam turbines were at their chronological level of technology. They operated reliably and effectively for their original design. However, during their 35+ years of operation,

some of the diaphragms developed problems as the gas plants production gradually increased. With increased gas production, the mechanical strength of the heavily loaded diaphragms became marginal. As a result of the increased loading, these diaphragms deflected more than they did originally and caused axial rubs with serious diaphragm and rotor damage.

Over the years as the operating conditions changed, Saudi Aramco personnel made several requests for the manufacture of "spare" diaphragms. Even though the operating conditions changed, the basic Saudi Aramco requirements did not. Specifically:

- The diaphragms had to have adequate increased mechanical strength to withstand the increased loads.
- They had to be more efficient.
- They had to be a "drop-in" design which could be installed into the existing turbine without any rework of the case and rotor.

At each request, diaphragm design changes were made to produce diaphragms that met Saudi Aramco's requirements. Each design step exemplified the latest technology available at the times of their design and manufacture.

This paper describes the design and manufacturing evolution leading to the current diaphragm design. The advanced analytical tools (FEA for defining stresses and deflections) were used for evaluating the mechanical integrity of the various diaphragm designs. Design improvements (vane profile, tangential vane lean, tip and new shaft seals), and manufacturing improvements (electron beam welding, laser and water jet cutting, electric discharge machine wire cutting), which have led to substantial gains in mechanical strength, efficiency and reliability will be described in detail.

BRIEF HISTORY

Steam Turbine Design

Each impulse steam turbine located at Saudi Aramco's Uthmaniyah and Shedgum gas plants is, per design, a multi-stage and multi-valve, non-condensing unit with a power output of 32239 HP at a rotor speed of 5713 RPM. The turbine operates with inlet steam conditions of 4136 kPA, 371 °C, and a backpressure of 550 kPa, (600 PSIG, 700 °F and a backpressure of 80 PSIG) (Figure 1).

Fresh steam enters the steam chest through five valves, then expands along the steam path and leaves the turbine through an upper exhaust.

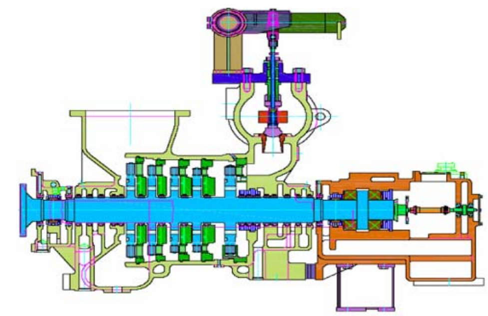

Figure 1 - Cross-Section of the Original Delaval KJ-MV-NC Steam Turbine

The turbine's steam path consists of five Rateau stages. The first (control) stage has a partial arc (approximately 50 percent) steam admission while the remaining four stages operate with full (100 percent) admission.

The nozzle ring and four diaphragms are of standard welded design. They were fabricated and machined using the manufacturer's proven procedures. The nozzle ring is attached to the case steam chest by special wedges on its outer diameter (OD), and by bolting on its inner diameter (ID). Four diaphragms are installed in the case. Their assembly into the case will be described later.

The entire rotor is made from a solid forging and, together with its rotating blades, is a rugged turbine component. Rotating blades are fixed on the rotor by axial fir-tree fasteners which mate with the broached axial grooves in each of the five discs. In each row, the tips of the blades are interconnected. In the first stage the blade tips are connected by a combination of integral and loose shrouds. In the remaining four stages the blade tips are connected by loose shrouds only.

The rotor is supported at each end by a tilt-pad journal bearing. A single, equalizing, tilt-pad thrust bearing locates the rotor axially.

Interstage shaft seals and both end seals, front and exhaust, are also of the manufacturer's standard design. Each seal ring is made from leaded-nickel bronze and consists of four spring-backed segments.

All things considered, in the mid 1970's these steam turbines were at the leading edge of turbomachinery design and manufacturing technology.

Operational History

Two Saudi Aramco gas plants, Uthmaniyah and Shedgum, form the largest integrated facility in the world. This facility went into commercial operation in 1980 and has gradually increased production (to NGL Fractionation plants and Sales gas to the power industry) over 30+ years. The subject eight turbomachinery trains are a critical component of the facility, since all LP sweet gas is compressed through them.

Shortly after the start of the production (mid 1980s) the nozzle rings in several turbines were found to have broken or damaged exit edges. While the exact cause of the damage was unclear until much later, the manufacturer addressed the problem as described in the "Optimal Edge-to-Edge Clearance" section, and the steam turbines operated without notable incidents for more than 10 years.

In the late 1990's inspections showed some damage and wear to the stage 3, 4, and 5 diaphragms. Therefore spare diaphragms of the original design were purchased, manufactured, and installed.

From 2001 to 2005 it became apparent that the original diaphragm design was no longer adequate for the plants increased output. Turbomachinery technology had also changed during the intervening years, and the manufacturer was able to offer modified diaphragms that improved the mechanical strength and efficiency. These modified diaphragms were purchased and some of them were put into operation.

In 2005 through 2009 Saudi Aramco experienced diaphragm and rotor failures in four steam turbines. It was clear the damage occurred due to an axial rotor rub against the diaphragms (Figure 2 to Figure 4). The damage at stages 4 and 5 was especially severe.

In another turbine with an undamaged rotor, the vanes at the horizontal joint of the stage 5 diaphragm were loose enough that they could be moved by hand (Figure 5). Although the damaged rotors were repaired and placed back into operation, it was established that both the original diaphragm design and the modified diaphragm design no longer had sufficient mechanical strength for the current plant operating conditions. This was confirmed with Finite Element Analysis (FEA) provided by the manufacturer.

Figure 2 - Severe damage of Row #4 Blades

Figure 3- Severe Disc #4 Damage: Rubbed Metal Filled Equalizing Holes

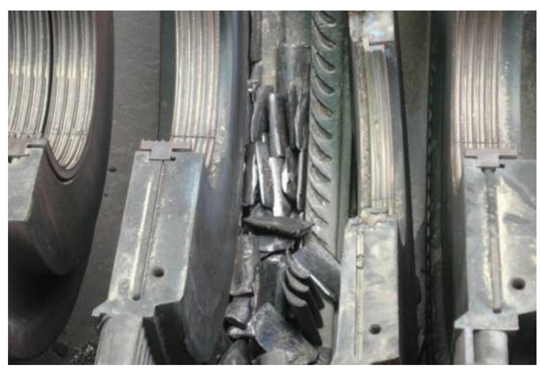

Figure 4- Complete Disintegration of Stage #4 Original Diaphragm

Working with the manufacturer, it was determined that it was possible to substantially improve the reliability and performance of the entire turbine steam path. However, optimizing the steam turbines did not make financial sense unless the entire turbine-compressor train was likewise upgraded and coordinated with the plans of the gas plant's development. Therefore Saudi Aramco postponed the entire turbine upgrade, and requested that the manufacturer create a new generation of diaphragms with maximum mechanical strength, reliability, and

Figure 5 - Unreliable Stage #5 Original Diaphragm: Vanes at Joint can be Shaken by Hand

efficiency for the existing steam turbines which would be installed in a “drop-in” manner and could be used with the existing rotors and cases. The requested generation of diaphragms would be designed for current operating conditions and incorporate the latest developments in diaphragm design and manufacturing technologies. These new diaphragms have been manufactured, installed, and are successfully operating in five steam turbines (Figure 6). Saudi Aramco plans to install these new diaphragms in their remaining steam turbines.

Figure 6 - New Current Diaphragms in the Turbine Case

Basically, the joint Aramco-Manufacturer experience in design, production, and operation of several diaphragm generations for the subject turbines, reflects the entire evolution of diaphragm design and manufacturing over the last 40 years. Below is a brief description of the major steps of this remarkable evolution.

DIAPHRAGMS: GENERAL FEATURES AND REQUIREMENTS

Main Requirements

Diaphragms are the major stationary component of a steam turbine. Diaphragms provide the following important functions (Bistritzkiy, 1956):

- Transform potential steam energy of the steam into kinetic energy of the powerful steam jet (Figure 7).
- Direct the steam jet into the rotating blades in the most effective manner.
- Establish the pressure drop per stage and withstand heavy pressure loads at high temperature, especially in the first few stages of the typical steam turbine.
- Ensure minimum deflections under heavy loads in order to prevent any rubs.
- Contain all the stage seals (tip, root, and shaft seals), which minimize steam leakages beyond the main steam path and thus improve stage efficiency (Figure 8).
- Divide the interior steam turbine space into separate compartments for each rotating wheel of the turbine rotor. A separate “room” for each rotating stage minimizes damage if a failure occurs (Figure 9).

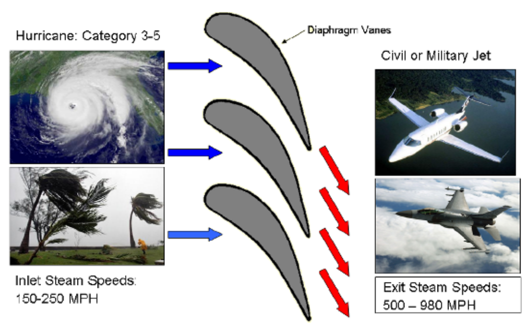

Figure 7 - Steam Velocities Through an Impulse Turbine

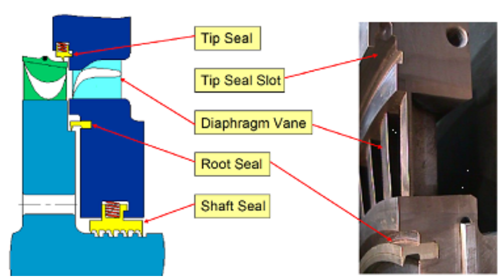

Figure 8 - Diaphragm: A Housing of the Stage Seals

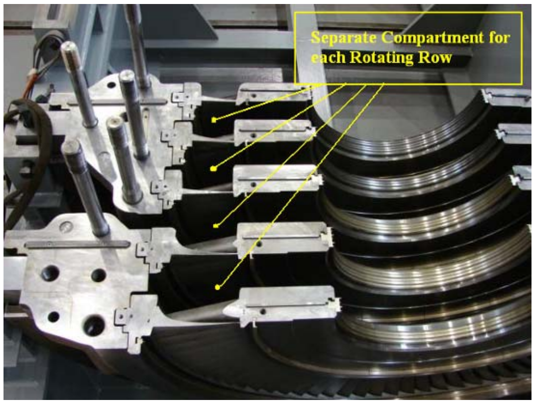

Figure 9 - Diaphragms Assembly in Case: Axial Positioning

General Design Features

Each turbine diaphragm is a complicated 360 degree plate, which is composed of three major components made from different materials. The outer and inner rings are reliably connected together with a set of vanes having complex geometry. While vanes, operating in the main steam flow, are made from special stainless steel, both rings can be made from such metals as: carbon steel, carbon-moly or chrome-moly alloys, and even cast iron - depending upon steam conditions (pressure and temperature) and loads (Bistritzkiy, 1956).

To accommodate installation of the rotor, the diaphragm is split into equal halves creating a horizontal joint. The outer ring supports the diaphragm in the case while the inner ring hangs free. The solid structure can be achieved by one of the following manufacturing methods:

- Welding
- Casting
- Pinning
- Machining from a solid plate

Welding is the most common, reliable, and cost effective method of diaphragm fabrication. Casting a diaphragm, consisting of cast iron rings and stainless steel vanes, is inaccurate and does not have sufficient strength. The two remaining methods provide accuracy and sufficient strength, but are very expensive. Therefore, these three fabrication methods are not used widely in industry and will not be discussed in this paper.

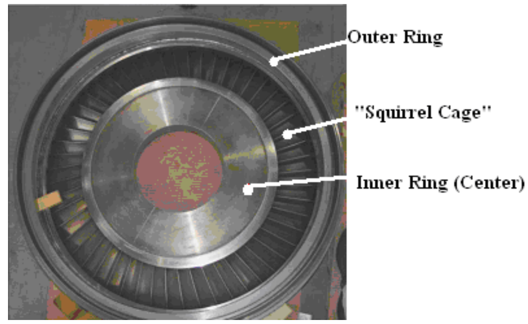

Figure 10 - Diaphragm: Overall View Of Assembled Diaphragm

A typical welded diaphragm assembly consists of three main parts (Figure 10): the outer ring, the “squirrel cage” (or cage), and the inner ring (or center). The “squirrel cage” is made of two concentric thin rings known as the outer and inner spacing strips, with the vanes between the rings. Each spacing strip contains through-holes. Their quantity is equal to the number of diaphragm vanes, and they are located evenly around the spacing strip circumference. Each vane is inserted through these profile holes in the outer and inner spacing strips and is affixed to them by welding. The completed “squirrel cage” is assembled with the outer and inner rings of the diaphragm and welded together by four significant circumferential welds. After welding is complete, each diaphragm must be stress-relieved and final machined.

The diaphragm assembly into the steam turbine case is arranged as follows: the tongue on the diaphragm’s outer diameter (OD) is inserted into a mating groove in the case; this fixes the diaphragm in the axial direction. Each diaphragm is secured in the horizontal and vertical direction by using a set of perpendicular special keys, also known as the “thermal key-cross” (Figure 11).

The lower half of a diaphragm is suspended in the case lower half at the horizontal joint by two special supports. These supports are located on the left and right side of the diaphragm and provide vertical alignment of the diaphragm. The horizontal alignment is accomplished by a centering key or pin located at 6 o’clock position. The “thermal key-cross” design provides the proper diaphragm position during operation, protecting from the inevitable thermal and mechanical case deformations.

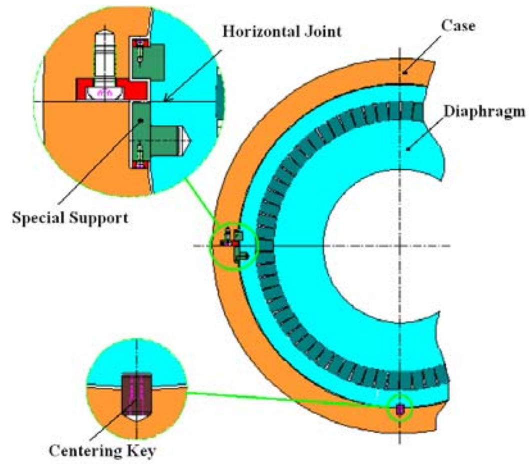

Figure 11 - Diaphragm Alignment in Case by Means of the "Thermal Key-Cross"

PROGRESS IN DIAPHRAGM DESIGN

Extensive R&D, using achievements in electronics, computation, and modeling during the last 20+ years, has yielded a new generation of advanced design tools such as Computational Fluid Dynamics, (CFD) (Deckers et al. 1997, Dawes 1988, Dawes 1990, Holmes and Tong 1985, Turner et al. 1993) and Finite Element Analysis, (FEA) (Moaveni 2008). Use of these calculation methods along with cascading test data and rotating telemetry testing, resulted in a greater fundamental understanding of the aerodynamic and thermodynamic processes inside a steam turbine. This new knowledge made it possible to generate significant design improvements in the major steam turbine components (Paul et al. 1989, Shcheglyaev 1993, Troyanovsky 1993, Cofer 1996).

In the area of diaphragm design, there are the following improvements, which resulted in a substantial increase in diaphragm reliability and efficiency:

- New vane airfoils
- Optimal tangential lean of the vanes
- Contoured endwalls in the HP diaphragms
- Optimal edge to edge clearance between vanes and rotating blades
- Advanced stage seals

Below is a brief description of these proven design achievements implemented in the UGP and SGP steam turbine diaphragms.

AIRFOIL DESIGN

Figure 12 shows a comparison of the original and the current airfoil design. The adjacent original airfoils form a channel that accelerates steam flow starting from the entrance; and only after the steam travels almost through the entire channel is it turned to the proper exit angle (or direction). Inevitably, turning the high velocity steam flow causes intensive steam separation resulting in substantial profile and secondary losses.

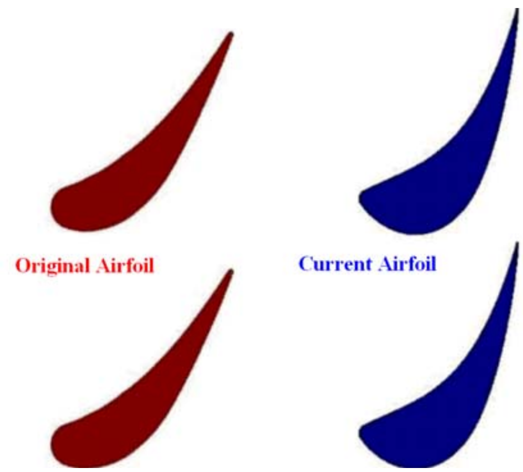

Figure 12 - Evolution in Vane Airfoil

The current airfoil design acts exactly opposite the original design. Adjacent current airfoils form a channel which first turns the entering steam flow to the proper direction and only then accelerates it. Since the steam flow velocity during its turn is relatively low, the overall losses are much smaller than those in the original design. Additionally, the current airfoil has the improved configuration (geometry) of the inlet edge, which allows it to accept steam flow at different angles without significant separation. This widens the range of effective operation. Also the new vane airfoil is significantly stronger than the original one which increases the mechanical strength and reliability of the entire diaphragm.

<u>References:</u>

Paul et al. 1989, Shcheglyaev 1993, Goel et al. 1993, Shelton et al. 1993, Dejch et al. 1994, Cofer 1996

OPTIMAL TANGENTIAL LEAN OF THE VANES

Maximum utilization of the available kinetic energy of the steam flow depends, among other factors, upon the smooth, unimpeded entrance of the steam jet into the rotating blades. This smooth entrance can only be achieved by directing the steam jet at a certain compound angle.

The axial angle component of this compound angle was defined long ago. The first ideas regarding the radial angle component and the importance of proper tangential vane orientation (lean), defining this radial angle, were published in the 1950's (Shcheglyaev 1993, Filippov and Zhong-Chi Wand 1964). However, the existing limitations in theoretical knowledge, tools, and technology did not allow these ideas to develop and be implemented into everyday practice.

Figure 13 shows the possible tangential leans of the vanes, the orientation of their exit edges and the schematic direction of the steam flow provided by the following leans:

a. Negative lean

b. Radial orientation (no lean)

c. Positive lean

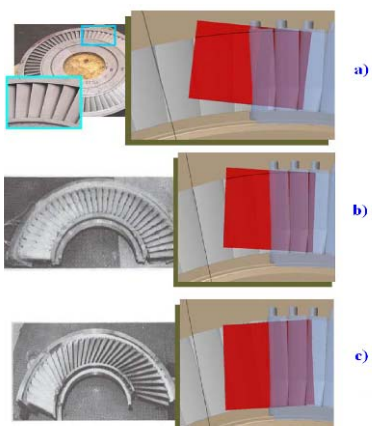

Figure 13 - Tangential Lean of the Vanes

Up until the 1970's, due to manufacturing limitations, the vanes were oriented with negative lean. As evident from Figure 13a, vanes with negative lean cannot effectively distribute the steam flow along the full height of the rotating blades. The peripheral area of the rotating blades is overloaded. Moreover, a significant portion of the steam flow is directed above the rotating blades, totally by-passing them. This is especially detrimental for designs without radial tip seals. Concurrent with the area of overload, some area at the root of the rotating blades experiences flow starvation. Such uneven steam flow distribution along the rotating blades seriously affects their efficiency and reliability.

In the period between the 1970's and 1990's the continuous design and technological progress, which is described later in this paper, allowed the improvement of the vane orientation from a negative lean to a radial orientation of the vane exit edges. As seen in Figure 13b, this change resulted in a better steam flow distribution along the rotating blades' height, which improved the efficiency and reliability. However, it did not completely resolve the problem.

Since the 1990's further design and manufacturing improvements made it possible to develop the optimal tangential vane orientation with positive lean of the exit edges. Such vane positioning almost provides an even steam flow distribution along the entire height of the rotating blades in the high and intermediate pressure stages (Figure 13c). However, it is not enough for low pressure stages with long, rotating blades. Only the latest 3-D design of the vane and blade airfoil areas, (not shown here), resolves the problem for these low pressure stages (Jansen and Ulm 1995, Oeynhausen et al. 1996). Figure 14 shows another view of the different tangential vane leans and their impact on stage efficiency, based upon the tests made in the former USSR and Japanese R&D (Kawagishi et al. 1991).

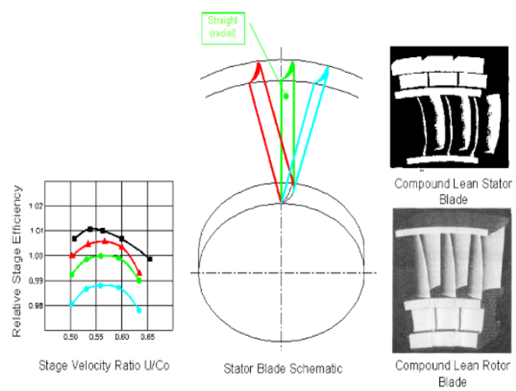

Figure 14 - Relative Stage Efficiency with Different Tangential Lean of the Vanes

Figure 15 shows actual diaphragms with negative and positive vane leans.

Figure 15 - Diaphragms with Different Tangential Lean of the Vanes

References:

Filippov et al. 1964, Troyanovsky 1991, Kawagishi et al. 1991, Shcheglyaev 1993, Troyanovsky 1993, Sakamoto et al. 1993, Tanuma et al. 1995, Singh et al. 1995, Jansen et al. 1995, Oeynhausen et al. 1996

CONTOURED END WALL

The first two to four stages of the steam turbine operate with high pressure, high temperature, and high density steam. The small volumetric steam flow in these stages requires partial arc steam admission and short vanes and rotating blades. The large pressure drops across the nozzle ring and diaphragms impose high loads, which dictate the use of special vane profiles with large nozzle ring/diaphragm axial widths. Therefore the stationary vanes and the rotor blading on the first few stages have a low aspect ratio, (i.e. small airfoil heights and large axial widths).

Steam flow at these stages moves through a long, narrow passage, which results in increased profile and secondary losses. The end wall boundary layer in these stages plays a much more negative role compared with the tall blading because it:

- Significantly decreases the steam passing area.
- It actively interacts with the main steam flow forming an additional radial flow, which causes disturbances and energy losses.

Similar to the tangential lean of the vanes, the idea about special profiling the outer wall of the nozzle ring and diaphragm with a low aspect ratio was published in the 1950's (Shcheglyaev 1993, Dejch et al. 1960). Again, this idea could not be materialized due to manufacturing limitations.

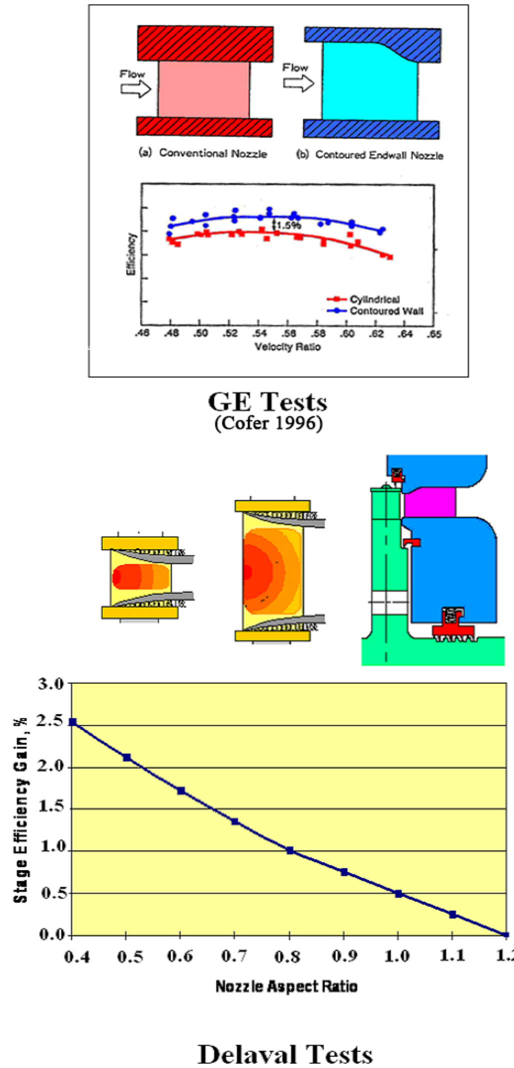

Figure 16 - Contoured End Wall Test Data

Starting in the 1980's, numerous studies on this issue resumed in different countries, and they continue to this day (McIntosh 2011). These studies confirmed that by special profiling of the nozzle ring and diaphragm outer wall (i.e. the inner diameter of the outer spacing strip), it is possible to significantly reduce the profile and secondary losses. The benefits of this profiling are:

- Substantial reduction of the outer boundary layer, especially near the exit edge of the airfoil area.
- Reduction of eddies and other disturbances in the main steam flow.
- Funneling of the main steam flow towards the inner cylindrical wall, (i.e. the outer diameter of the inner spacing strip).

So, the contouring of the outer end wall of the nozzle ring and diaphragms results in an increase in stage efficiency (Figure 16) and reliability of the stages with the low blading aspect ratio.

References:

Dejch et al. 1960, Sieverding 1975, Tran 1986, Atkins 1987, Denton 1987, Shcheglyaev 1993, Troyanovsky 1993, Cofer 1996, Leyzerovich 1997, McIntosh et al. 2011

OPTIMAL EDGE-TO-EDGE CLEARANCE (EEC)

EEC is the axial distance between the vane exit edge and rotating blade inlet edge of each stage (Figure 17).

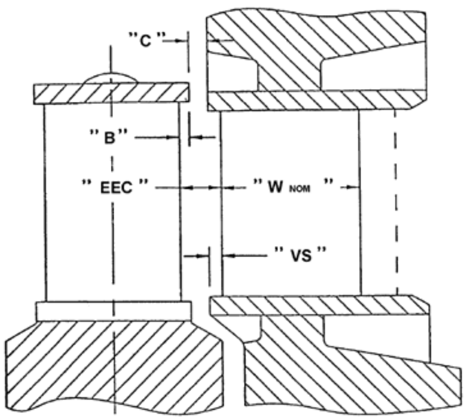

Figure 17 - Axial Clearance/ Vane Setback in a Turbine Stage

EEC is an important factor that affects stage performance (both efficiency and reliability). The required EEC for optimal performance is dependent on whether or not the stage has effective tip and root seals.

For a stage without effective seals, steam leakages beyond the rotating blades are a decisive factor of performance. Tests of a stage with 35 mm long blades showed that an increase in EEC from 0.8 mm to 2.5 mm resulted in an efficiency drop by more than 5 percent. For this reason, up to the 1970's, most turbine stages had small EEC. To achieve a small EEC, minimal vane setback ("vs" in Figure 17) was implemented (.254-.762 mm) or (.010-.030").

The main benefits of small EEC on a stage without effective tip and root seals are summarized as follows:

- Minimization of steam leakage beyond the designed steam flow path (above rotating blade shrouds and below platforms).
- Minimization of mixture between the main steam flow and leakage through the shaft seals.

The major drawbacks are as follows:

- Minimal vane setback leaves the vane exit edges susceptible to damage during manufacturing (sub-arc welding and final machining) and repair.
- The high velocity steam flow, in conjunction with limited axial spacing between the vanes and blades, can result in severe damage to both components (Figure 18), reducing reliability and efficiency.

As it was mentioned earlier, damage of the exit edges of the nozzle ring vanes in the first Aramco turbines occurred at the very beginning of their operation. It was evident that the vane failures occurred due to intensive excitation forces. Without knowing the exact nature of these forces, the problem was resolved by redesigning the nozzle ring. The strength of the vanes was substantially increased by changing the airfoil and by connecting all the vanes together by a circumferential rib located in the middle of their height. At the same time, similar nozzle ring damage in Russian high pressure turbines with power capacities 25-100MW was resolved by substantial increase (approximately 1.4 times) of the control stage EEC - without changing the nozzle ring design.

For a stage with effective tip and root seals, leakage from the main steam path is kept to a minimum, allowing the EEC to be increased. With a higher EEC, steam flow becomes more laminar along the pitches and blade length because irregularities in the steam flow (eddies, wakes, etc.) have more axial space to diminish. This produces the following benefits:

- Improved efficiency due to better steam path utilization.
- Improved reliability due to decreased excitation forces on the vanes and blades.
- Increased vane setback eliminates manufacturing damage of the vane exit edge and associated vane repair.

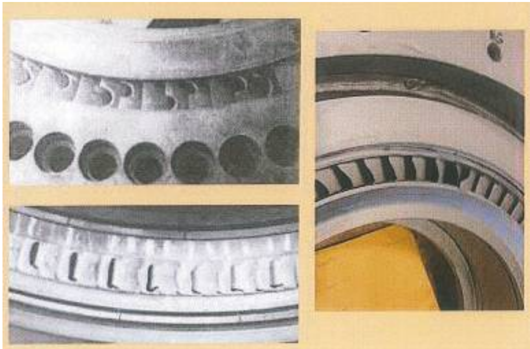

Figure 18 - Damaged Vanes Due to Insufficient Edge-to-Edge Clearances

In summary, the magnitude of EEC required for optimal performance is dependent on other design features of the stage, namely, the use of tip and root seals. For stages with no effective tip and root seals (most vintage turbines), it is beneficial to maintain a smaller EEC for the best performance, at the cost of reliability. For stages with effective tip and root seals (used on most modern turbines), EEC does not have to be used as a method to minimize steam leakage. The designer is able to increase the EEC, thereby reducing turbulent flow and excitation forces (Figure 19).

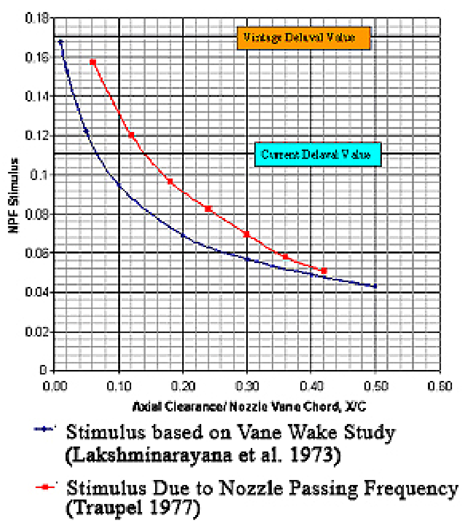

Figure 19 - Dependence of the NPF Steam Stimulus upon Relative Edge-to-Edge Clearance

References:

Lakshminarayana et al. 1973, Traupel 1977, Langston 1980, Shcheglyaev 1993, Lazzeri et al. 1996

INTERSTAGE SEALS

In a steam turbine, inter-stage steam leakage bypasses or invades the main steam flow through the steam path (i.e.: vanes and rotating blades). Steam leakage is a significant source of performance loss due to:

- The missing energy of the steam which leaves or bypasses the main flow.
- The flow path disturbances (wakes, eddies, etc.) caused when leaked steam enters and collides with the main steam flow, affecting both efficiency and reliability.

Figure 20 shows the major causes of deterioration, or reduction, of the turbine stage performance based upon the research performed in the former USSR and in the USA ("TurboCare Retractable Packing" 1999).

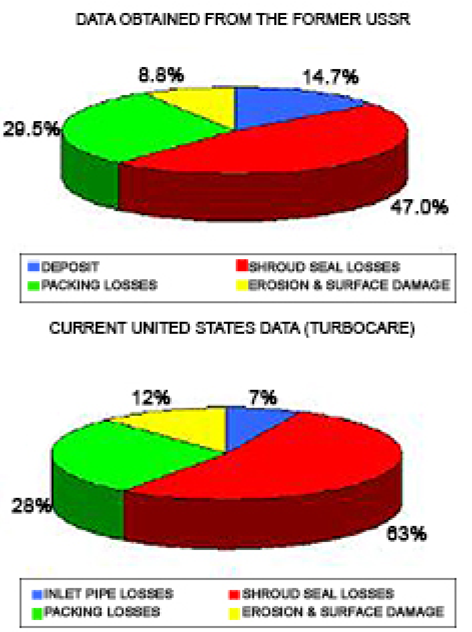

Figure 20 - Major Causes of Reduction in Steam Turbine Efficiency & Performance

Despite the different timing in data acquisition, differences in turbine design, power output and operational modes, both pie charts illustrate the same conclusions: steam leakage beyond the blade shrouds and roots, and between the shaft and the diaphragms is the main source (76.5 percent to 81.0 percent) of turbine performance deterioration.

Minimizing steam leakages and keeping them stable during long- term operation is the most effective way to achieve and maintain higher steam turbine efficiency and reliability.

In a typical impulse stage, there are three routes for steam leaks through the gaps between stator and rotor components:

- Between the diaphragm ID and shaft OD – a so called, "shaft steam leak".
- Over the outer diameter of the rotating wheel, above the blades – a "tip steam leak".
- Near the root of the rotating blades through axial space between diaphragm and rotating wheel – a "root steam leak".

The amount of steam leaks and the possible direction of their flows can be quite different. The specific leak flow pattern depends upon multiple factors, including the characteristics of the main flow, the seal design(s), the configuration of the axial space between the diaphragm and the rotating wheel, the number of pressure-balance (equalizing) holes in the disk, their diameter and location, etc.

Accordingly, in order to minimize and contain these leaks, the modern turbine stage incorporates three types of seals: shaft seals, tip seals and root seals. (Figure 21)

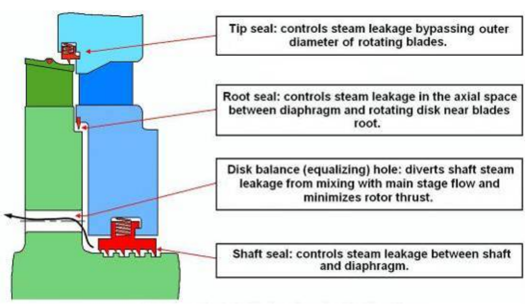

Figure 21 – Interstage Seals in Modern Impulse Turbine Stage

This goal can be realized by improving the interstage seals.

TIP SEALS

The primary function of tip seals is to prevent steam leak from the main steam flow into the space above the rotating blades. Among the stage seals, the leak through the tip seals is the largest source of efficiency loss due to:

- Largest leak area.
- Highest reaction in this area.
- Deviation of steam flow because of centrifugal effect of rotary blades.

Similarly to all other steam turbine components, tip seals went through a long evolution in design, from the first primitive axial rigid seals to current complicated multi-fin or honeycomb designs.

Below is a description of the major known tip seal designs: axial rigid seal, radial rigid seals, radial flexible (spring-backed) seal, and multi-fin seals. The special anti-vibration tip seal design is described in a separate section titled "Interstage Seals and Low Frequency Vibrations".

Axial Rigid Seal

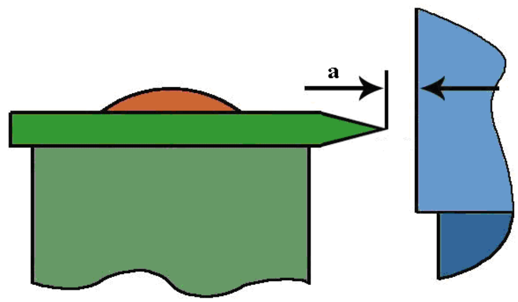

Figure 22 - Axial Rigid Seal

This primitive seal design is a fin, formed on the inlet edge of the rotating blades shroud which works against a mating flat face of the diaphragm. Figure 22 shows the blading and diaphragm which are both made from hard metal.

The axial clearance "a" between the shroud fin and the flat mating face controls steam leakage to some degree, especially in the new condition. However, the axial clearance in this type of seal has to be large in order to prevent rubbing which results in damage to the blading. During turbine operation this clearance increases due to rotor thermal expansion, thus significantly decreasing its sealing ability. Moreover, as the rotor thermal expansion changes, depending upon the operational regime, the axial clearance also changes making the performance of the entire tip seal unstable.

Therefore, today this design is used by some turbomachinery manufacturers only as an additional component to the other more advanced tip seal designs.

Radial Rigid Seal

This seal is formed by a thin (approximately 2.54mm or 0.100" thick) segmented plate (Figure 23) or by a very thin (approximately 1.0mm or 0.040" thick) sheet (Figure 24) installed into a radial groove.

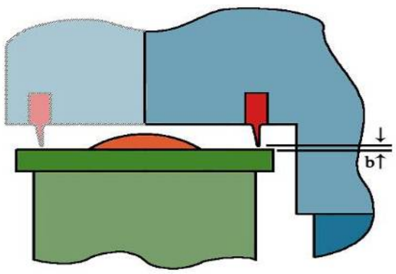

Figure 23 - Radial Rigid Seal (Segmented Plate)

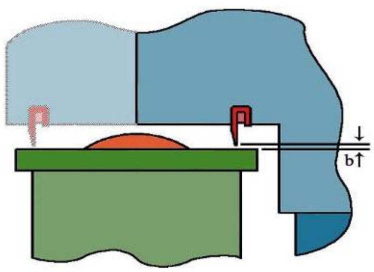

Figure 24 - Radial Rigid Seal (Thin Sheet)

This radial groove is located either in the diaphragm outer ring (if one fin is required), or in the special tip ring, attached to the outer ring by bolting or welding (if several fins are required). Each radial groove is oriented perpendicular to the shaft axis. Segments of the plate are staked in the groove, while the sheet metal is fixed in the groove by a caulking wire. The geometry of the plate or sheet metal at the inner diameter provides a thin fin which is engaged with a cylindrical surface of the rotating blades shrouding. The radial clearance "b" between the thin fin and blades shroud controls the steam leakage.

Since the radial clearance remains relatively unchanged* during different operational regimes, this seal is much more stable compared to the axial rigid seal. Another positive feature of the segmented plate seal is that this plate can be made from different materials (special anti-friction bronze, Ni-resist, or stainless steel – depending mainly upon steam temperature). However, the thin sheet is usually made from stainless steel. Also, radial rigid seals can be easily replaced during turbine overhauls.

In spite of the previously described positive features, these seals have the following substantial drawbacks:

- They (especially the sheet metal design) are vulnerable to mechanical damage that may occur during turbine operation (e.g. foreign object damage due to steam impurity) and from handling during maintenance/repairs, overhauls and transportation.
- Being rigidly fixed in the diaphragm, these seals are subjected to severe radial rubs when rotor radial displacements are larger than the radial clearance. This situation takes place:
    - During start-ups and shut-downs (when rotor passes through first critical speed).
    - During sharp dynamic regimes – "cycling", (changes in power output steam conditions, etc.) leading to sudden significant misalignments in the steam path.

During rubbing, the seal surfaces (the thin fin and blades shrouds) wear down (thus increasing the radial clearance). In some cases, the rigid seal may cut a groove in the shroud which requires replacement of the entire row of rotating blades (Figure 25).

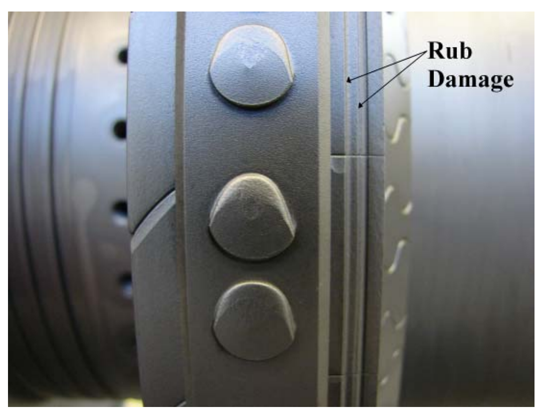

Figure 25 – Shroud Damage Due to Heavy Tip Seal Rub

* In reality, during operation the radial clearances also change their value from the original (cold) condition, because of inevitable misalignments between rotor and stator, as it is described below in the "Interstage Seals and Low-Frequency Vibration" section. However, these changes are much smaller compared to the axial clearance changes in the axial rigid seals.

Radial Flexible (Spring-backed) Seal

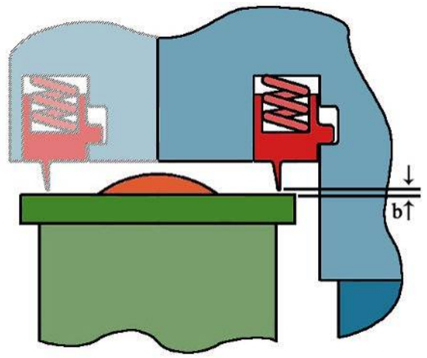

Figure 26 - Radial Flexible (Spring-backed) Seal

This seal is formed by a segmented ring, installed into a diaphragm radial groove with complex geometry (Figure 26). The location of this groove and its orientation is the same as the radial rigid seal. The outer portion of this ring has a special geometry which allows its reliable positioning in the groove. The ring ID has one or two shaped thin fins engaged with the cylindrical surface of the rotating blades shroud.

Similarly to the previous design, the radial clearance "b" between the thin fin(s) and the shroud controls the steam leakage. Since this clearance remains relatively unchanged (see previous note *) during different regimes, this seal also remains stable during turbine operation.

Each seal ring consists of several independent segments (usually between 8 and 12-depending upon outer diameter of the rotating wheel). Each segment is spring backed by flat or coiled springs, located on its OD. These springs fix each segment in the proper position in the groove providing the designed radial clearance between the fins during turbine assembly. During operation, steam pressure over the seal OD, reinforces the spring action, reliably keeping all the segments in the proper position. However, during some operational regimes when the rotor would otherwise contact the fin(s), the rotor overcomes the joint forces of spring and steam pressure and pushes the segments outwards, thus minimizing all the negative effects from rubbing as described above. When turbine vibration and/or rotor misalignments die down, the springs and steam pressure will push segments back into their original position, thus restoring the radial clearance. An additional advantage of this design is that the segmented seal ring is much stronger than the rigid seal and therefore is not vulnerable to foreign object damage or mishandling. It is also maintenance friendly and can be easily replaced during brief maintenance or overhaul. However, this type of tip seal is more expensive and requires more space in axial and radial directions compared to radial rigid seals.

Because of the positive features described above, this seal can provide the effective leak control while increasing longevity of the seal ring and preventing rotor damage. These tip seals are implemented in the current Saudi Aramco diaphragms.

Multi-Fin Tip Seals

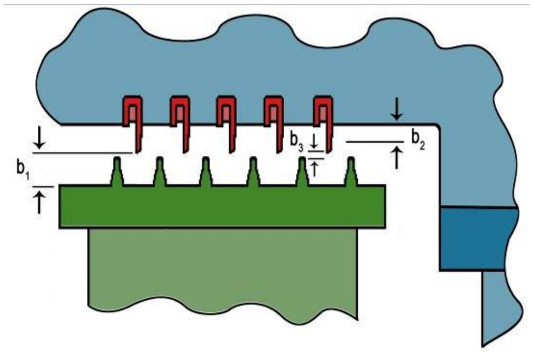

Figure 27 - Radial Multi-fin Tip Seal (Solid and Staked Design)

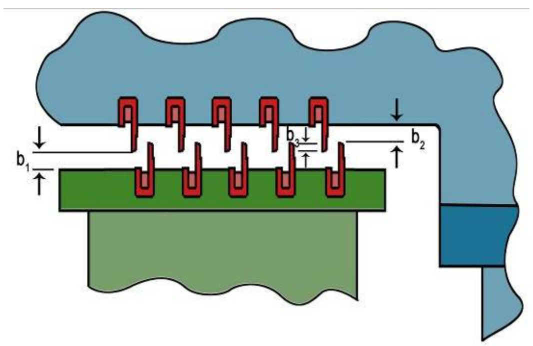

Figure 28 - Radial Multi-fin Tip Seal (Staked Design)

This seal is formed by two sets of multiple radial rigid fins, one set of fins in the stator (diaphragm) and another set in the shroud OD of rotating blades. Fins in the diaphragm are of rigid radial design described above. Fins in the shroud can be machined either directly from the shroud (Figure 27) or using the same design as in the diaphragm (Figure 28). The number of fins (and pitch) in each set is different. Radial clearances between fins and cylindrical surfaces of the diaphragm and rotor "b_1" and b_2" are much larger compared to radial seals, while radial clearances between fins in both sets "b_3" are close to clearances in rigid seals. Effective sealing ability of this design is provided by a combination of different clearances and resistance of chambers with different geometry.

In spite of the high efficiency of such a design, it has the following deficiencies:

- It is expensive.
- It is extremely vulnerable to damage by foreign objects in steam or by mishandling.
- Replacement of damaged fins in diaphragm is costly and complicated.
- The longevity (service life) is limited because restoration/replacement of damaged fins in the shroud is almost impossible without replacement of the entire row of blades.

Another configuration of multi-fin design is described below in the "Interstage Seals and Low-Frequency Vibration" section.

All in all, any "healthy" radial tip seal improves the stage efficiency by approximately 2.5 percent compared to axial rigid seal. It is possible to achieve even better results using different combinations of radial tip seals in HP and LP stages. For example, implementation of the contoured end wall and effective tip seals in the control stage of a large MHI turbine resulted in increased stage efficiency by about 7% which was obtained by MHI bench tests (Troyanovski 1993).

References:

Paul et al. 1989, Shcheglyav 1993, Troyanovsky 1993, Cofer 1996, Leyzerovich 1997

ROOT SEALS

Root seals perform two functions:

- They should prevent the shaft leakage from re-entering in the main flow at the diaphragm exit and mixing with it. Such a collision forms additional wakes (eddies) and thus affects:
  - Efficiency, because of diminishing high tangential momentum of the main flow.
  - Reliability, since wakes and eddies in the main flow increase stimulus acting on the rotating blades.
- They also should prevent leakage, (energy loss) from the main steam flow into the shaft area, through equalizing holes in the disc.

While less common than tip and shaft seals, nevertheless, root seals up to this day remain a less investigated component in turbomachinery design due to the very complicated conditions of their operation. As it was stated earlier, every wheel rotates in a separate compartment formed by two neighboring diaphragms, and creates two cavities which are interconnected with the equalizing holes. Steam flow in this area is complicated, as steam moves, swirling in all (axial, radial and circumferential) directions, depending upon numerous variables, such as: geometry (shape and dimensions) of both cavities, equalizing holes (location, diameter, inlet and exit contour), steam conditions (pressure, stage reaction), regime of operation, etc. Therefore, steam flow in this area is unstable and can be dramatically changed depending upon balance of those variables.

The first attempts to describe the steam flow behavior in this area are dated to the late 1930's (Lomakin, 1940) and continued since then with periods of higher and lower intensity, as it can be seen in the works of Samoylovich and Morozov 1957, Shvetz et al. 1960, Kapinos et al. 1983, Wilson et al. 1997. It still remains a point of considerable interest (Cao et al. 2003, Moroz and Tarasov, 2003 and 2004). Theoretical research was combined with testing of various root seal designs. These theoretical and experimental investigations resulted in better understanding of the processes inside the described areas but did not define general criteria for root seal design. Therefore, each turbomachinery facility develops its own approach in resolving this problem. Below is given a brief description of the following major known root seal designs: axial rigid seal, radial rigid seals, aerodynamic "curtain" seal and cantilevered semi-flexible seal.

Axial Rigid Seal

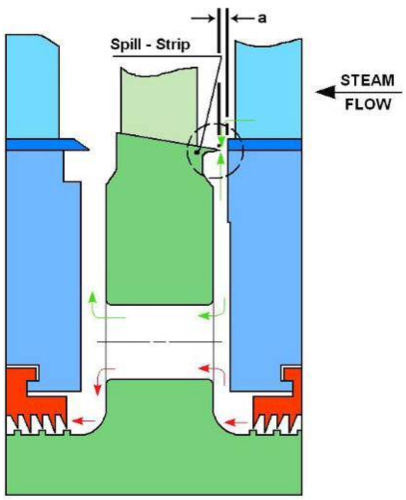

Figure 29 - Axial Rigid Flat Design Root Seal

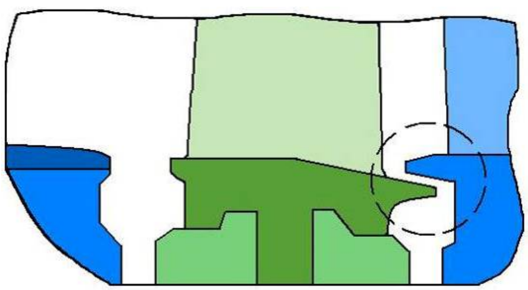

Figure 30 - Axial Rigid Staggered Design Root Seal

This seal is formed by one or two so called spill-strips (thin fins) working against a mating flat face or towards each other. Spill-strips can be located either on the rotating blade platforms or on the diaphragm face (Figure 29) or on both components (Figure 30). Being comprised of blades and diaphragm, both mating components of this seal are made from hard metal.

The seal shown on Figure 30 appears to be more efficient since it does not allow direct collision of the shaft leak steam with the main flow and also provides more resistance to steam leaking from the main flow down. However it is more costly and provides higher shaft leak entrance into the rotating blades due to the suction effect of the main flow.

Since the design of this seal is almost identical to the axial rigid tip seals, its operation and drawbacks are the same as described in the previous section.

Radial Rigid Seal

Similarly to tip seals, this seal is formed by either a thin (approximately 2.54 mm or 0.10” thick) segmented plate (Figure 31.1) or by a very thin (approximately 1.0 mm or 0.040” thick) sheet (Figure 31.2) installed into the diaphragm radial groove. The radial groove is located in the shoulder of the diaphragm inner ring, near the rotating blade fasteners and is oriented perpendicular to the shaft axis. Segments of the plate are staked in the groove, while the sheet metal is fixed in the groove by a caulking wire. The geometry of the plate or sheet metal at the inner diameter also provides a thin fin, which is engaged with a cylindrical surface on the shoulder of the disk. The radial clearance “b” between the thin fin and disk shoulder controls the steam leakage.

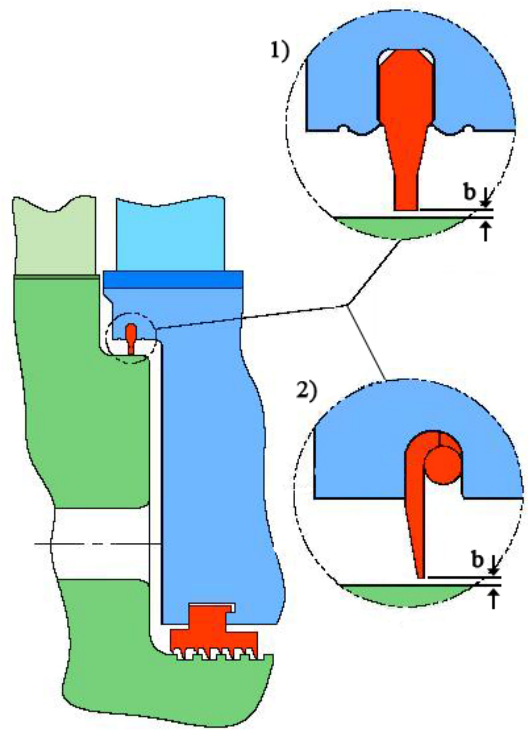

Figure 31 – 1) Radial Rigid Root Seal Made From Thin Segmented Plate; 2) Radial Rigid Root Seal Made From Very Thin Sheet Metal

Since the design of this seal is almost identical to the radial rigid tip seals, their operation and drawbacks are the same and are described in the previous section.

Aerodynamic Curtain

By design, during operation, a small amount of steam is extracted from the main flow and is directed axially from the bottom portion of the diaphragm vanes (near base diameter) into the rotating wheel passages "p" between the disk OD and blades platforms (Figure 32). This additional steam flow forms an aerodynamic barrier ("curtain") which prevents collision of the steam leakage from the shaft seals with the main steam flow. The shaft seal leakage mixes together with the "curtain steam" and moves through passages at the disk OD. In order to eliminate any additional routes for steam leaks and to minimize amount of the extracted "curtain steam", the balance (equalizing) holes in the rotating disks are eliminated.

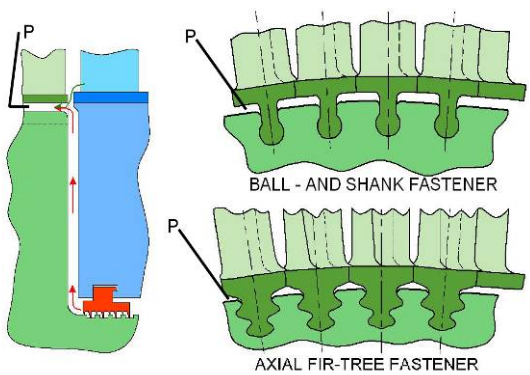

Figure 32 - "Aerodynamic Curtain" Root Seals for Axial Entry Rows of Blades

According to testing (Troyanovsky 1993), properly designed "steam curtains" can improve stage efficiency by up to 0.5 percent. While effective at base load regimes, this design has the following drawbacks:

- Limited application, since this design can be used only with rotating blades having axial (not radial) fasteners.
- Limited effectiveness at partial and dynamic regimes.
- Elimination of the balance holes in the disks results in significant increase of the thrust load, which requires implementation of a special balance ("dummy") piston on the turbine front seal and requires a larger thrust bearing. The substantially increased leakages of high potential steam through this balance piston together with increased friction losses in the larger thrust bearing diminishes the positive effect of the "aerodynamic curtain" seals.

Cantilevered Semi-Flexible Seal

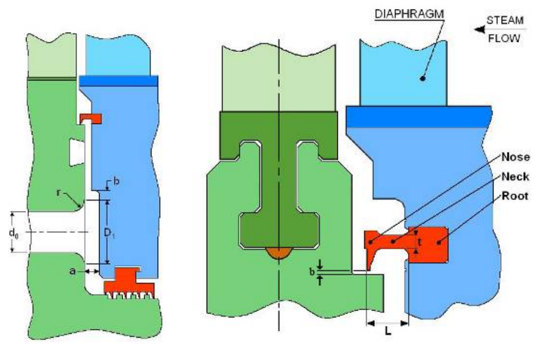

Figure 33 - Cantilevered Semi-Flexible Root Seal

This design consists of a cantilevered semi-flexible segmented seal ring and specially shaped both (front and exit) cavities which are interconnected by carefully shaped equalizing holes. The main design features of the entire root seal area are as follows (Figure 33):

- The cantilever style segmented seal ring consists of three major elements: a relatively thick short "root", an elongated "neck" and a "nose" with a thin fin at the end.
- The "root" of this ring is installed into an axial groove of the diaphragm and is reliably fixed there by staking.
- The thin fin of the ring "nose" is engaged with the cylindrical surface of the rotating disk shoulder, forming a small radial clearance "b" which controls the steam leakages.
- The proper geometry of the "neck" (dimensions "L", "t", etc.) and entire segment provides the necessary flexibility to this seal.
- The inlet cavity is divided by a cantilevered seal ring in two areas. The lower portion protrudes radially from the shaft up to the seal ring and is shaped to provide the favorable conditions for steam leaking through shaft seals to be evacuated through equalizing holes to the next turbine stage. The upper portion of this cavity protrudes radially from the seal ring up to the rotating blades platform and is formed to minimize steam leak in radial direction.

- The equalizing holes have proper diameter, quantity and profiled inlet and exit entrance to provide the steam flow passage with minimum resistance.

The cantilevered semi-flexible seal design has the following major advantages:

- It effectively prevents shaft steam leaks from entering and colliding with the main steam flow by creating:
  - Minimal resistance for shaft steam leakage to pass through the balance holes.
  - Maximum resistance against moving radially upward – into the main steam flow.

This improves reliability.

- At the same time, it minimizes steam leakage from the main steam flow (through the axial space between the rotating blade roots and diaphragm) and provides stable leak control during operation (radial clearance in the root seal does not change with rotor thermal expansion).
- This is a very robust design, since the cantilever seal ring can be individually sized and made from different materials. Thus choosing a material combination for each stage gives optimal flexibility depending upon particular conditions. Therefore, when the turbine experiences vibrations and there is a chance of rubs, each segment can absorb these shaft movements and minimize the wear/damage.
- This design does not require additional axial space compared to radial rigid seals and can be incorporated in each stage.
- In addition to its longevity, this design is maintenance-friendly because it is very easy to replace the cantilevered seal ring.

Figure 34 and Figure 35 present combined results of testing some root seals by MHI (Hirota et al. 1985) and by a Russian R&D facility. A proper root seal design can result in 1.5 – 2.0 percent efficiency increase per stage.

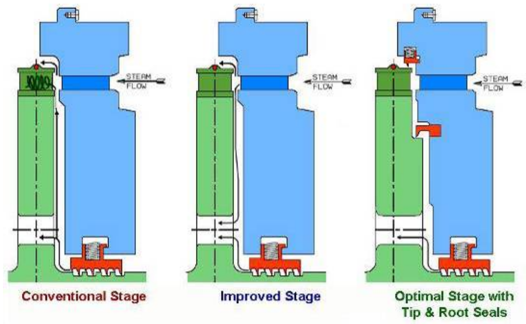

Figure 34 - Different Turbine Stages

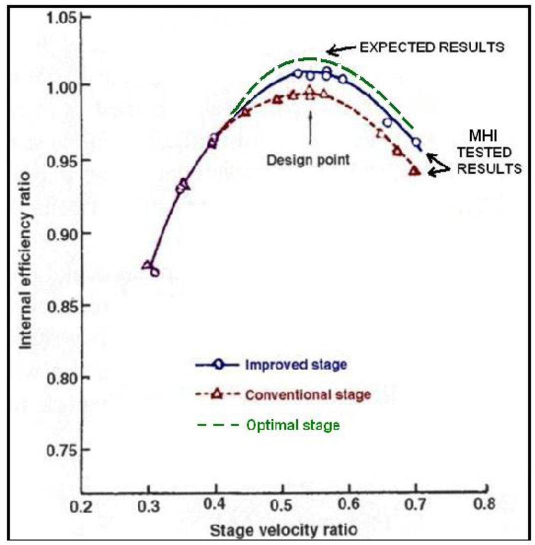

Figure 35 - Influence of the Root Seal upon the Internal Efficiency of an Impulse Turbine Stage

References:

Lomakin 1940, Samoylovich et al. 1957, Shvetz et al. 1960, Nedzvetzkiy et al. 1974, Kapinos et al. 1983, Hirota et al. 1985, Wilson et al. 1997, Pigott et al. 1986, Shcheglyaev 1993, Troyanovsky 1993, Cao et al. 2003, Moroz et al. 2004

SHAFT SEALS

Meanwhile, extensive research in seal technology has also resulted in significant advances of the shaft seal design. The main R&D efforts were to create a labyrinth seal ring able to operate with minimal clearances between its fins and the rotor, and, at the same time, do not rub against the rotor and wear which takes place during long-term operation (Figure 36).

Figure 36 - Wear Pattern of Standard Shaft Seals

The long-term factors which severely affect reliable operation of the labyrinth seals are as follows:

- Case thermal deformation due to temperature differences across its wall thickness (Figure 37), and top-to-bottom halves of the case (Figure 38).
- Start-up and shutdown regimes when turbine rotor runs through 1st critical speed which is linked to with high vibration (Figure 39).

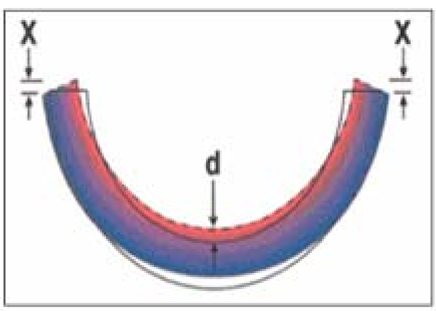

Figure 37 - Case Thermal Deformation Due to Temperature Difference Across the Wall

The effective solution in overcoming the case thermal deformation is to align all the diaphragms and packing boxes in the case with the "thermal key-cross" (as it was described previously), and provide a sufficient radial clearance between the case bore and diaphragm/labyrinth box OD along the circumference. Also, suspension of the diaphragms and packing boxes at the case joint allows for different radial clearances between the fins and the shaft along the circumference (larger ones in the lower half and smaller in the upper half). This design solution makes seals less vulnerable to case thermal and mechanical deformations and decreases their wear, thus, increasing turbine efficiency and reliability.

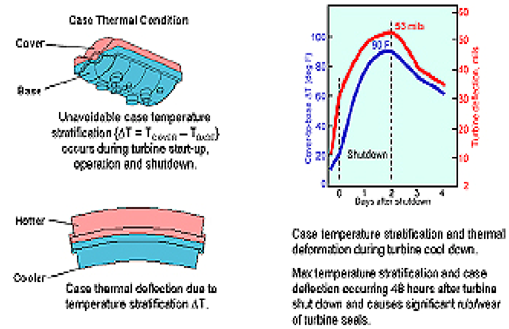

Figure 38 - Case Thermal Deformation Due to Cover-to-Base Temperature Difference

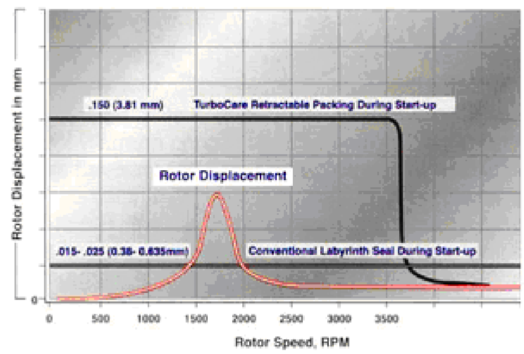

Figure 39 - Rotor Displacements during Typical Turbine Startup and Shutdown

References:

Traupel 1997, Shcheglyaev 1993, Troyanovsky1993, Cofer IV 1996

RETRACTABLE/BRUSH SEALS

The next significant step in seal technology was the development of a retractable seal design.

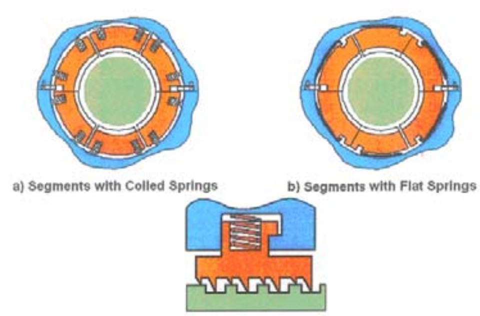

Figure 40 - Standard Spring-Back Shaft Seal Ring

In the standard labyrinth seal (Figure 40), each segment has on its OD springs that force it to be in the design position during cold start-up and shut-down turbine conditions. During the operation, in addition to springs, steam pressure acting over OD segments keeps them in design position.

Contrary to the standard design, in retractable seals springs are located in the sides of a segment (Figure 41). They push segments away from the shaft, creating large clearances between seal fins and the shaft during turbine start-ups or shut-downs. As these clearances are much larger than rotor displacements during the 1st critical speed, the retractable labyrinth seal does not suffer damage or wear during start-up and shut-down regimes. During turbine operation at loads exceeding approximately 5 percent of the rated capacity, steam pressure overcomes the spring resistance and pushes the segments back into working position closing the radial clearances to design values. Using retractable seals improves efficiency by approximately 2-3 percent and increases reliability (Little et al. 2001).

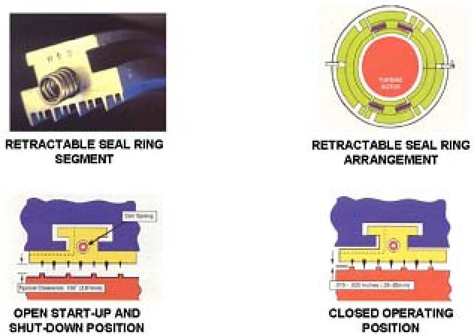

Figure 41 - Retractable Shaft Seal Ring

The latest seal technology achievement is a retractable brush seal design. In this seal, one standard fin seal in a retractable seal ring is replaced by a brush seal insert which consists of a pack of brush bristles secured by two plates (Figure 42).

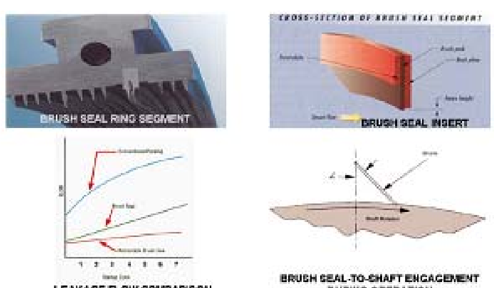

Figure 42 - Retractable Brush Seal Ring

Brush insert bristles are oriented not perpendicular to the shaft surface (as the standard fin seals are) but at a certain angle in the direction of shaft rotation. Such orientation, together with bristles flexibility, allows minimizing radial clearance down to 0.000-0.005" (0.00-0.13mm). Using retractable brush seals will result in additional efficiency gain of 1-2 percent without compromising reliability (Little et al. 2001, Sulda 1999, Foley 2000, Neef et al. 2006).

Retractable brush seals are not only more efficient than standard spring-backed seals; they are much more stable during long-term operation. The leakage through a new standard seal is 5-8 times higher than through a retractable brush seal. During turbine operation, standard seals wear out and the steam leakage though them increases by 2-3 times more (compared to new condition), while leakage through a brush seal remains mostly unchanged. Figure 43 is a plot of tested leakage through different seal ring designs versus pressure ratio across the seals, showing substantial advantages of retractable and brush seals, especially for long term operation.

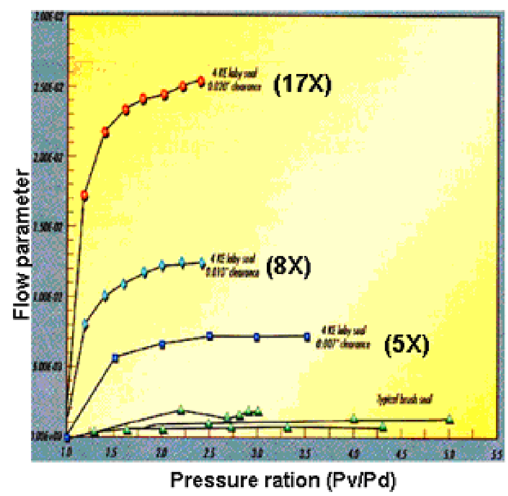

Figure 43 - Tested Effectiveness of Different Shaft Seals

References:

Schofield 1981, TurboCare Retractable Packing 1999, Sulda 1999, Foley 2000, Little et al. 2001, Neef et al. 2006

INTERSTAGE SEALS AND LOW-FREQUENCY VIBRATION

It is evident from the above description of the interstage seals, that the best are the seals with the minimum technically allowable radial clearances, because they can minimize the steam leaks by-passing the main steam flow or colliding with it. In reality this is not always the case. Extremely small clearances in the interstage seals can cause serious sub-synchronous (low-frequency) rotor vibration. This situation usually occurs in steam turbines operating at high pressures and high power outputs in which rotors tend to be relatively light weight. These rotors are typically found in high pressure turbine-generator sets and in high pressure / high speed steam turbines used as mechanical drives in petrochemical and other industries.

Low-frequency vibrations are self-excited rotor oscillations in the journal bearings oil film. They occur at operating speed with a frequency equal to the first natural rotor frequency in the entire train rotor system, which incidentally, is close to one-half of the steam turbine rotor frequency. This vibration is called "self-excited" because a turbine rotor which is operating smoothly will reach a load, called the "threshold" load, where it suddenly loses its stability and produces intense vibrations - without any visible outside causes.

There are two main sources of the low-frequency vibration:

- "Oil swirl" (rotor oscillations) caused by hydrodynamic forces in oil film of the journal bearings at low static load.
- "Steam swirl" (rotor oscillations) caused by unbalanced aerodynamic forces in the steam path which can act separately or in combination with the additional hydrodynamic excitation in journal bearings.

The theory of "steam swirl" anti-vibrational stability of the interconnected rotors system was developed by A.G. Kostyuk in 1972 and V.I. Olimpiyev in 1976, and was described in detail by A.D. Trukhny in 1990.

“Steam swirl”, according to the theory, can be generated by three types of aerodynamic forces that occur within a steam path:

- Circulating forces which are produced by the rotating row of blades due to irregularities in the rotating force.
- Shroud forces which are the result of non-uniform pressure distribution along circumference of the area beyond the rotating blades shrouds, i.e. in tip seals.
- Labyrinth forces which are caused by non-uniform pressure along circumference of the multiple chambers of shaft seals.

This theoretical analysis, supported by numerous tests and experience gained while resolving the sub-synchronous vibration in actual steam turbines, revealed that the aerodynamic excitation forces in the high pressure tip seals are the dominant cause of the low-frequency vibration.

The aerodynamic excitation forces are generated by the tip seals due to inevitable misalignments between rotor and stator. During turbine operation, it is impossible to provide perfect concentricity between rotor and stator because of the following reasons:

- Rotor precession in journal bearings (i.e. its constant movement within the oil film limits).
- Thermal misalignment due to different rotor and stator temperatures in axial and radial directions.
- Mechanical deformations due to steam pressures, vacuum in condenser, piping influence, etc.

These misalignments cause different clearances in the interstage seals along the circumference of the rotor. The uneven clearances cause uneven steam leaks. Figure 44 explains the mechanism of steam low-frequency vibration.

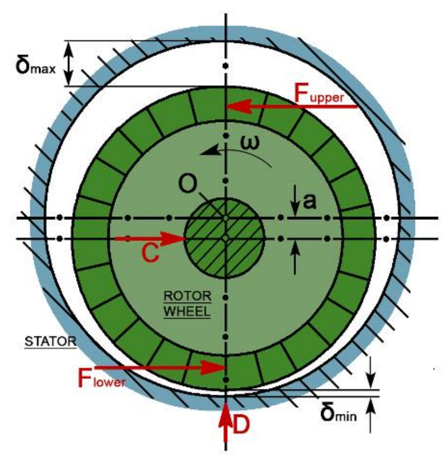

Figure 44 – Interstage Seals and Low Frequency Vibration

In this figure the rotor is shown displaced in vertical direction from its original (cold assembly) position “O” down by a value "a". The clearances in lower half are smaller than in the upper half. The minimal clearance is at the 6 o’clock position, while the maximum clearance in upper half is at 12 o’clock position.

So, steam leaks increase through the upper half and decrease through the lower half. As a result of uneven leakages, the main steam flow through the stage will also be uneven: it will be decreased in the upper half and increased in the lower half, generating uneven circulating force ($F_{\mathbf{upper}}$ < $F_{\mathbf{lower}}$). The resulting unbalanced aerodynamic radial force "C" is applied to the center of the rotor and rotates together with the rotor but 90 degree ahead of the dynamic rotor displacement. Concurrently, uneven clearances in the interstage seals cause another radial force "D" in the area of reduced clearances, due to the increased steam pressure there. This force acts as a "lifting" element trying to return the rotor from the misplaced position back to the center of the stator. These two unbalanced radial forces depend upon numerous variables, such as: steam path (including interstage seals) design, steam pressure, stage reaction, value of rotor radial displacements, etc.

The turbomachinery industry has developed effective anti-vibration seal designs and recommendations for their implementation.

For tip seals it is recommended to use a single-fin seal with the minimal possible radial clearances or to use two (inlet and exit) single-fin seals with different clearances. The seal on the inlet side of the flow should have the smaller clearance and the seal on the exit side of the flow should have the larger radial clearance. Also, the chamfer between the seals in a two seal design should have a certain volume to compensate for the "shroud" excitation forces. This is the exact design implemented in the current Aramco diaphragms.

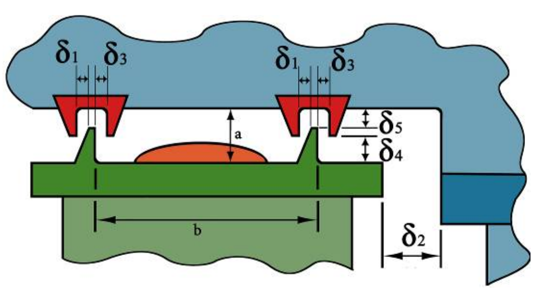

Figure 45 – Anti-Vibration Tip Seal

One of the best anti-vibration tip seal designs is shown on Figure 45. The main feature of this design is large radial clearances δ_4 and δ_5 in all the fins. The effectiveness of this design is achieved by the proper combination of axial (δ_1 and δ_3) and radial clearances (δ_4 and δ_5). Large radial clearances are insensitive to radial rotor displacements: they do not cause substantial change in steam leakages, maintaining the balance between main steam flow and leakages around circumference during turbine operation.

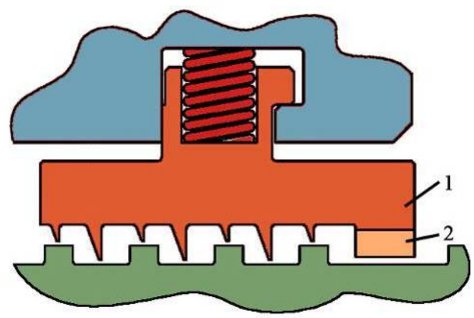

Figure 46 – Standard Shaft Seal with Anti-Vibration Addition

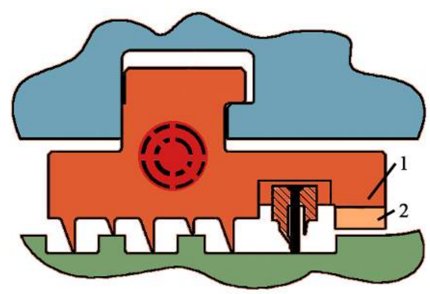

Figure 47 – Retractable Brush Seal with Anti-Vibration

The effective designs of anti-vibration shaft seals are shown on Figure 46 and Figure 47. A special "nose" (1), with a set of axial channels (2), (located along circumference of its bore) is arranged in front of seal fins. These axial channels damp irregularities in the leakage steam, minimizing labyrinth forces. Those seals were not used in Aramco turbines since calculated labyrinth forces in the subject steam path are insignificant.

References:

Kostyuk 1972, Olimpiyev et al. 1975, Olimpiyev 1976, Runov 1982, Trukhny 1990, Leyzerovich 1997

PROGRESS IN MECHANICAL STRENGTH CALCULAIONS

A turbine diaphragm, due to its complex structure and harsh operating conditions, is a very difficult subject for strength calculations.

As previously stated, structurally, a diaphragm is a complicated 360 degree plate, which is composed of several major components made from different materials. The outer and inner rings are connected together with a set of beams/vanes of complex geometry. The diaphragm is split in two halves at the horizontal joint. The outer ring is supported by its OD in the case while the inner ring is "free." The diaphragm is loaded in two main ways. First, the entire diaphragm is subjected to a uniformly distributed steam pressure load across its inlet face. Second, the vanes are subjected to a bending moment due to the impact of high velocity steam flow. In addition to these loading conditions, other factors of concern are thermal gradients in the radial direction and between diaphragm halves in the range of 25-65°F (15-35°C), and vibration forces caused by the steam flow.

The combination of two diaphragm halves (instead of a solid continuous plate), and the plurality of the vanes with complex geometry present the main problems for stress and deflection calculations. Because of this structural complexity, strength calculations were possible in the past only by simplifications of the mathematical model.

However, mechanical strength calculations for diaphragms have also been progressing along with the evolution in their design.

Up to the 1950's, OEM's calculated diaphragms' strength using classical methods developed by A.M. Wahl (1930, 1932), and D.M. Smith (1938). A.M. Wahl analyzed diaphragms as a solid annular half ring plate (without vanes) which was rigidly supported by its outer contour and was subjected to a uniformly distributed load. D.M. Smith considered diaphragms as a plurality of rigid beams, each beam being supported rigidly in the case and rigidly connected with a massive inner ring. This structure was also considered to be subjected to a uniformly distributed load.

Naturally, stresses and deflections produced by these calculations significantly deviated from the actual values. Therefore the turbomachinery industry continued intensive R&D, both theoretical research (Muster and and Sadowsky 1956, Naumov 1960, Ingultsov 1958) and experimental testing (Taylor 1951, Sentsov 1958, Kulagina 1960, Mellerovich and Bliznyukova 1961) – in order to improve the existing calculation methods.

The tests revealed that, contrary to theoretical assumptions, all the connections between the vanes and both (outer and inner) rings, and between the outer ring and the case, are elastic (not rigid). Elasticity in those connections turned out to be critical for strength calculations.

All in all, this R&D allowed to:

- Modify the Smith Method by introducing new semi-empirical factors which more accurately reflect real loading in the diaphragm components.
- Together with the gained experience of turbines operation, establish semi-empirical "safe" allowable stresses and deflections in the diaphragms calculated by the Smith Method.

Despite the fact that measured stresses in the vanes located at the horizontal joint were still significantly higher than calculated values, the "Modified Smith Method" suited well for designing diaphragms of small and medium size turbines.

However, this method showed its limitations when applied to larger turbines operating with higher steam conditions and/or higher loads. Several catastrophic failures occurred: due to vanes overloading they were broken completely resulting in disintegration of the entire diaphragm with severe damage to the rotor and case (Dodd et al. 1975).

Such failures stimulated turbine manufacturers to develop new, more accurate calculation tools for large steam turbine diaphragms. One of these methods was introduced in the mid 1980's (Kostyuk 1982) based upon the mathematical model proposed by Pakhomov in 1934, and the results of numerous theoretical and experimental works done in the former Soviet Union. However, being quite simple and handy, this method still did not reflect the actual stresses, deformations, and deflections; especially in the vanes.

The real break-through in diaphragm strength calculations occurred with the advent of the finite element analysis (FEA) method; a numerical procedure in which the solution domain (i.e. diaphragm) is discretized into nodes and elements, producing a large set of algebraic equations representing the entire system. After the application of boundary conditions and loads, these equations are solved simultaneously to obtain information at the location of the nodes such as displacement or stress. While the finite element method can be traced back to the early 1900's, its true value was not realized until combined with computers in the 1970's. With the help of computers, a component can be modeled with thousands of nodes and elements, allowing for highly accurate solutions (Moaveni 2008). With this valuable tool, a diaphragm can be quickly modeled and analyzed, allowing the designer to retrieve data (accurate stresses and deformations) at any point in the component.

FEA of Saudi Aramco Diaphragms

During the production of all Saudi Aramco diaphragms (except for the "current design"), the mechanical strength was calculated using the "Modified Smith Method." Mechanical strength of all the "current design" diaphragms was calculated by FEA.

Since the majority of Saudi Aramco steam turbine rotor damage occurred in stage #4 area, the following stage #4 diaphragms were analyzed at the current operating conditions:

- The original diaphragm with welding kerfs using 2 x 180 degree sub-arc welding technology.
- An "imaginary" diaphragm: the original diaphragm design geometry but fabricated using 1 x 360 degree electron beam (EB) welding with 100 percent penetration.
- The current diaphragm design (2010 – present)

FEA results are presented on Figure 48, Figure 49, Figure 50 and Table 1. Figure 48 and Figure 49 show calculated deflections and equivalent stresses in the entire original (A) and current (C) diaphragm halves. Figure 50 demonstrates the distribution of vane stresses (as percentage above yield strength) in one diaphragm half for these diaphragms and for all remaining known diaphragms that failed during 40 plus years of operation. The "0" and "180" degree locations are representative of the vanes at the horizontal joint of the diaphragm. The acceptable allowance has been set at below 2 percent of vane nodes above yield strength. It is clearly shown that no matter the diaphragm, stresses are always highest at these two locations in the diaphragm arc. The diaphragms are represented by the blue, tan, purple, orange and teal lines are all remaining known diaphragms that have experienced failures during 40 plus years of operation due to greatly exceeding the yield limits of the vane material at the joint. The red, yellow, and pink lines represent the subject Aramco diaphragms (all at the current pressure differential). Clearly, the vanes of the original design exceed yield strength limits (above 2 percent of nodes). However, with the introduction of EB welding and 360 degree construction, the stresses are significantly reduced (as shown by the yellow line) to less than 1 percent of nodes exceeding yield strength. The current diaphragm represented by the pink line, has even smaller stresses with essentially zero nodes exceeding the yield strength limit.

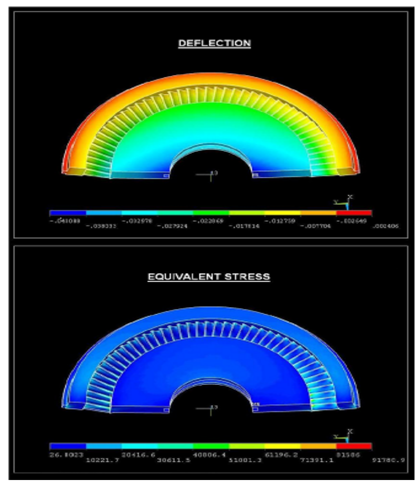

Figure 48 - FEA Calculations: Deflections and Equivalent Stresses in Original Design

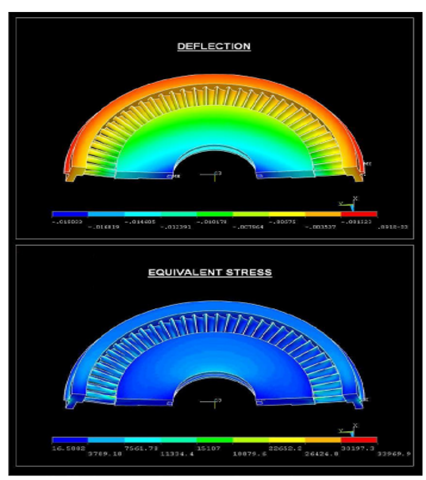

Figure 49 - FEA Calculations: Deflections and Equivalent Stresses in Current Design

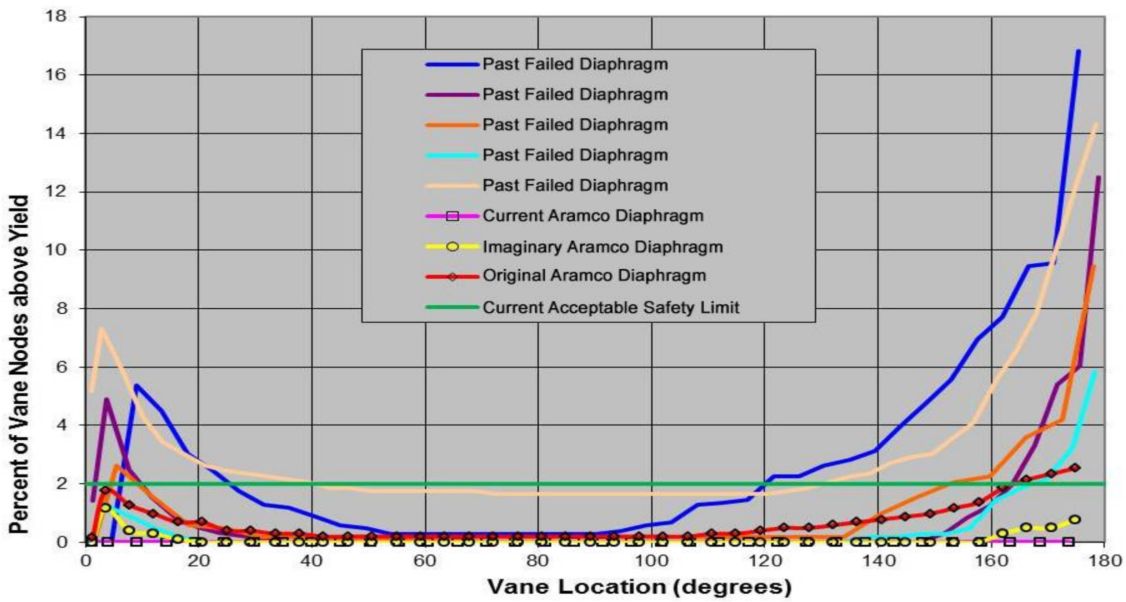

Figure 50 - Historical Data: Vanes Stress Evaluation

Table 1 - FEA Results for the Stage #4 Diaphragm Scenarios

Definition	Original Diaphragm	"Imaginary" Diaphragm	Current Diaphragm
Maximum Von Mises Equivalent Stresses (Vanes at Joint) **kPa (PSI)**	**632800 (91780)**	**433997 (62946)**	**234208 (33969)**
Maximum Deflection (Center Inner Diameter at Joint) **mm (inches)**	**1.092 (0.043")**	**0.991 (0.039")**	**0.483 (0.019")**

Table 1 presents maximal stresses and deflections for all three chosen diaphragm designs. It is evident from Table 1 that:

- Even with ideal kerfs and welding processes (i.e. no deviations from the nominal design dimensions and procedures), the original diaphragm is unsuitable for the current operating conditions due to the high deflections and the very high calculated stresses in the vanes at the horizontal joint, which exceed the material yield stresses even at room temperature. In actual original diaphragms having typical deviations from the design made during pre-machining, assembly and welding, the stresses and deflections may be significantly higher than the values given in Table 1.
- The original vanes were not strong enough. Even fabricated by EB welding (as demonstrated by the "imaginary" diaphragm) with 100 percent penetration, they still have high stresses and produce nearly the same deflections as the original sub-arc welded diaphragm. Therefore, the diaphragms failures occurred not due to operation and maintenance (O&M) mistakes, but rather to the mismatch of the original diaphragms design specifications and the new operating conditions that were imposed by the plant requirements.
- The mechanical strength of the current diaphragm is substantially higher compared to the original diaphragm due to the implemented advances in design, described above, and manufacturing, described in the following section. Calculated stresses and deflections for current diaphragms are well within the design limits for current operating conditions.

<u>References:</u>

Wahl 1930 and 1932, Pakhomov 1934, Smith 1938, Taylor 1951, Muster et al. 1956, Ingultsov 1958, Sentsov 1958, Kulagina 1960, Mellerovich et al. 1961, Dodd et al. 1975, Kostyuk 1982, Moaveni 2008

EVOLUTION IN DIAPHRAGM MANUFACTURING

Generally, a steam turbine diaphragm is a very difficult component to manufacture due the following conflicting requirements:

- It consists of different parts made of different materials.
- It must have exceptional mechanical strength in harsh environments.
- The diaphragm vanes must be precisely oriented, and have uniform pitches, heights, throats, and leans, to form and direct the steam flow (jet), into the rotating blades with maximum efficiency and minimal stimulus.

Combination of heavy welding with high accuracy presents approximately the same dilemma as "making a Swiss watch with a sledge hammer". Therefore diaphragm manufacturing includes numerous intermediate check/inspection points, including charting of the steam passing area, and a significant amount of hand-dressing prior to shipment.

Progress in electronics, computerization and modeling resulted not only in the break-through design improvements described above. It also accelerated significant progress throughout many industries. New tools and manufacturing processes (advances in metallurgy, numerically controlled machinery, electron beam welding, shot peening, laser cutting, water jet cutting and electro-discharge machining (EDM) wire cutting, etc.) made it possible to produce steam turbine diaphragms and other steam turbine components with a higher degree of accuracy and a better surface finish in a more cost-effective manner.

Below is a brief description of the evolution in diaphragm manufacturing. This evolution is also shown in Figure 51, which summarizes the progress in design and manufacturing of turbine diaphragms achieved during the last 40 years.

Original Diaphragms

The original diaphragms of the subject turbines were built using 1950's design and technology (Sentsov 1956, German and Kulakova 1957, Zemzin and Frenkel 1960, Houldcroft 1977). The cage and the entire diaphragm were made in two 180 degree halves. The profile holes in the spacing strips were punched while the strips were in the flat condition. Only after punching the profile holes, were the spacing strips rolled to form two 180 degree semicircles of the proper diameter. The profile holes and their pitching were inaccurate. The plunging process limited

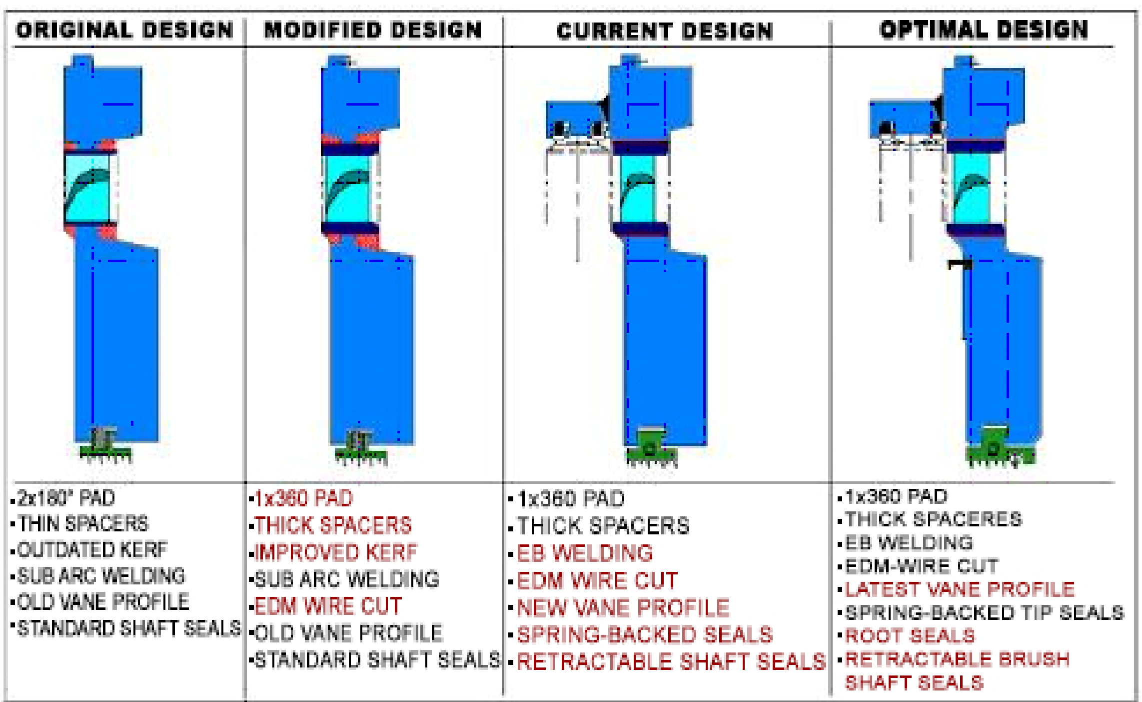

Figure 51 - Diaphragm Design and Fabrication Evolution

the spacing strip thickness to 0.140” (3.6mm) maximum and required larger holes and radial orientation of the plunger tool axis. Therefore the exit edge of the profile was parallel to the radial line which resulted in the negative tangential lean of the vanes with all the deficiencies described above.

The welding kerfs in the outer ring and center had non-optimal geometry which could not provide the maximum possible mechanical strength. Four massive sub-arc welds generated an enormous amount of heat, which caused significant thermal deformation. Also during welding, the thin spacing strips were often burned through, allowing melted metal to penetrate into the steam path and damage the vanes. Figure 52 and Figure 53 show the major steps in such diaphragm fabrication.

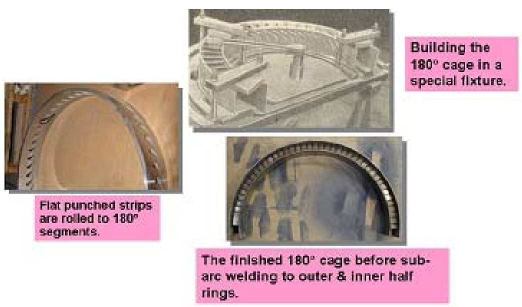

Figure 52 - Obsolete Building of a 180 Degree Cage

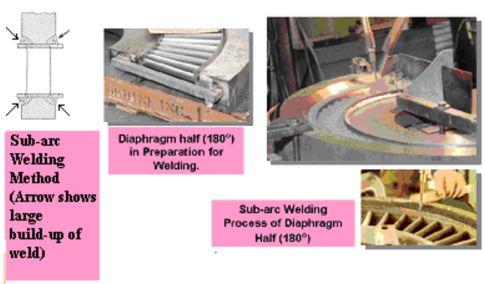

Figure 53 - Obsolete Fabrication of 2 x180 Degree Diaphragms by Sub-Arc Welding

During the machining of the diaphragm horizontal joint, the vanes in both diaphragm halves were unavoidably split and partially cut. The cut vanes at the horizontal joint of one half of the diaphragm did not accurately match with their mating parts in the opposite half. Also due to the minimal vane set back, all the vanes were vulnerable to mechanical damage during the welding process and by the loose metal chips produced during final machining.

So the described manufacturing method could not provide desired accuracy and required significant hand-dressing of each set of vanes.

This resulted in reduced efficiency (from poor channel aerodynamics), reliability, (because of the large stimulus spectrum acting on the rotating blades), and the reduced diaphragm strength.

Improved Diaphragms

Manufacturing progress in the 1970’s and 1980’s led to significant improvements in cage construction. Industry started laser or water-jet cutting profile holes while the spacing strips were in the rolled condition (Matsui et al. 1987). This allowed for a substantial increase of the spacing strip thickness (1.4 times and more) compared to the punching method, making the spacing strips sturdier and preventing burn-through damage. Better accuracy of the profile-hole geometry and pitching resulted in improved efficiency and decreased stimulus on the rotating blades.

Although cage construction improved, the original diaphragm fabrication method of producing two separate 180 degree halves using the original vane welding kerfs and sub-arc welding remained. Therefore, this manufacturing method had the same major original flaws as described above.

So, the improved diaphragm also required significant hand-dressing of the vanes and had limitations in efficiency, reliability and strength.

Modified Diaphragms

The limitations of the improved diaphragms were still troublesome. The next step in diaphragm evolution was the “modified diaphragm”. The modified diaphragm utilized a new machining method: Electrical Discharge Machining, (EDM), which was developed in the late 1980’s to the early 1990’s (Fuller 1989).

While the sub-arc welding method and the vane airfoils remained the same, the modified diaphragm utilized substantially improved cage and diaphragm construction. The diaphragm was built as a single, 360 degree ring with significantly improved welding kerfs, (instead of two 180 degree halves with the original kerfs). Improved welding kerfs provided better bonding between the outer ring, "squirrel cage", and the center ring, increasing the diaphragm mechanical strength by approximately 15 percent and decreasing thermal deformation. EDM wire cutting was used to split the diaphragm in halves using a wire diameter of 0.010 – 0.015 inch (0.25-0.38 mm). The subject cutting method did not remove much metal and therefore, allowed accurate mating of the split vanes at the horizontal joint. This modified diaphragm had increased accuracy, efficiency and reliability (due to decreased stimulus) while the amount of hand-dressing of the diaphragm vanes was reduced. But their mechanical strength soon became inadequate to the changed operation conditions.

Current Diaphragms

The current diaphragms utilize further improvements which occurred in the 1990's: the development of new analytical tools and manufacturing technologies combined with continuous design improvements.

Diaphragm stresses and deflections in all major components were obtained using FEA which was calibrated against in-house test results.

It became possible to use Electron-Beam (EB) welding technology instead of traditional sub-arc welding (Akutsu 1980, Schiller et al. 1982, Matsui 1987). EB welding, as the name implies, uses a high energy electron beam in deep vacuum, directed at the joint of the tightly assembled parts. The high temperature of the beam melts the metal at the joint where the assembled parts meet, welding them together in a very short period of time, which minimizes their heating. Welding in deep vacuum produces higher quality welds which have no porosity or inclusions. The possible axial depth (penetration) of EB welds is significantly higher than the depth of sub-arc welds. EB welds in many diaphragms penetrate 100 percent of the axial vane width. This deep weld penetration, combined with the high quality of the EB weld and new stronger airfoil, substantially increases the diaphragm strength when compared to a similar diaphragm fabricated by sub-arc welding. Also the EB welding method produces 8-10 times less heat compared to sub-arc welding methods. Therefore, EB welding causes insignificant thermal deformation of the diaphragm, which in turn results in higher efficiency and reliability, (due to reduced stimulus on the rotating blades) and minimizes amount of hand-dressing required. Figure 54 and Figure 55 show the major steps in diaphragm EB welding, while Figure 56 and Figure 57 show the differences in diaphragm quality produced by sub-arc and EB welding.

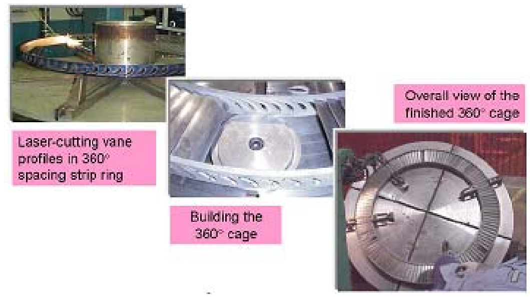

Figure 54 - Current Building of a 1 x 360 Degree Cage

Figure 55 - Advance Fabrication of a 1 x 360 Degree Diaphragm by EB Welding

So, utilization of the advances in stress analysis, (FEA), design achievements, (vane profiles, vane lean, tip and retractable shaft seals, etc.), and progressive manufacturing technologies, (CNC machining, EB welding and EDM wire cutting) allowed creation of the new, current generation of diaphragms having much higher efficiency, reliability, and mechanical strength without reworking the steam turbine cases or shafts. Saudi Aramco has already replaced existing diaphragms with new ones in 5 of their turbines and plan to complete this replacement on the remaining 3 units by the end of 2013.

Figure 56 - Deficiencies of Obsolete 2 x 180 Degree Diaphragm Fabricated by Sub-Arc Welding

Figure 57 - Advantages of Current 1 x 360 Degree Diaphragm Fabricated by EB Welding with EDM-Wire Cutting

Optimal Diaphragms

Current diaphragms, being superior to previous designs, nevertheless cannot provide the highest possible benefits (efficiency and reliability) to the steam turbine since they do not utilize the following design improvements which require changes to the existing rotors:

- New rotating blades with advanced airfoils and integral shrouds;
- Root seals;
- Retractable brush seals;
- Optimal smooth 50 percent to 100 percent steam admission.

New vanes with advanced airfoils in the current diaphragms cannot be installed in the optimal position, (set-up angle), because they have to operate with the original vintage rotating blades. Therefore, they could not realize their full potential efficiency. Only a combination of new vane with new rotating blade can provide the highest possible efficiency.

As it was described before, root seals and retractable brush seals also improve the steam turbine efficiency. However, root seals require a special shoulder in the inlet face of each rotor wheel to interact with the fin of the seal ring that would be installed into the exit face of the diaphragm. Retractable brush seals require additional axial space on the shaft in order to accommodate the brush which imposes change the standard rotor tongue-and-groove configuration.

The existing steam path is characterized by an abrupt 50 to 100 percent steam admission due to non-optimal steam path geometry of both the diaphragms and the rotor which does not accommodate the specifics of natural steam expansion. High velocity steam flow leaving the first (control) stage with a partial (approximate 50 percent) arc admission does not have enough room to spread 100 percent circumferentially. Therefore it can't pass through the second stage and beyond (designed for full arc admission) without suffering significant energy losses that seriously affect the overall steam turbine efficiency.

New diaphragms paired with a new/modified rotor allow a steam path design with a smooth, gradual transition from 50 to 100 percent steam admission designed in accordance with natural steam expansion for a given turbine geometry. This will minimize/eliminate the energy losses associated with abrupt steam admission and thus substantially increase the overall steam turbine efficiency.

Implementation of these advanced features in developing the optimal steam path, (optimal diaphragms and new/modified rotors combinations within the steam turbine), will further improve efficiency by 8 to 10 percent due to minimizing energy losses in the main steam flow and decreased leakages while also improving reliability due to decreased wakes and smoother main steam flow.

However, this design is much more expensive compared to all previous diaphragm designs. Therefore Saudi Aramco has decided to postpone its implementation pending the further development of the entire UGP and SGP plants.

<u>References:</u>

Sentsov 1956, Bistritzkiy 1956, German et al 1957, Zemzin et al. 1960, Houldcroft 1977, Akutsu 1980, Schiller et al. 1982, Matsui et al. 1987, Fuller 1989, Sun et al. 1996, Dragunov et al. 2005

CONCLUSIONS

- The entire story of manufacturing several generations of diaphragms for Saudi Aramco UGP and SGP steam turbines presents an evolution in diaphragm design, calculations, and manufacturing in the steam turbomachinery industry for almost forty years.
- Evolution in steam turbine diaphragm design, strength calculations, and manufacturing resulted in substantial increase of their efficiency, reliability, mechanical strength, and maintainability.
- Each diaphragm generation, for UGP and SGP, starting from the original parts, was designed and manufactured at the edge of existing technology. While being reliable for the provided design operating conditions, the margin of their mechanical strength decreased at aggravated loads imposed by increased plant output. Therefore, some of them deteriorated and failed causing serious rotor damage.
- The current generation of diaphragms, being in the process of installation in the UGP and SGP turbines with existing rotors, has substantially increased mechanical strength, efficiency, reliability, and maintainability. These diaphragms are completely adequate for current and planned future increases in plant output.
- There is potential for implementation of optimal diaphragm design, which, combined with the modernization of existing rotors and other components, will further increase turbine efficiency, reliability, and maintainability.

REFERENCES

Atkins, M.J., 1987, “Secondary Losses and End-Wall Profiling in a Turbine Cascade”, @ *Imeche Paper,* C 255/87.

Akutsu, Y., May 1980, “Application of Electron Beam Welding to Steam Turbine Diaphragms”, *Proc. of the International Beam Technology Conference*, Essen, Germany, pp. 63-69

Bistritzkiy, N.D., 1956, “Steam Turbine Diaphragms”, *Mashgiz* (In Russian)

Cao, C., Chew, J.W., Millington, P.R., Hogg, S.I., 2003, “Interaction of Rim Seal and Annulus Flows in an Axial Flow Turbine”, Proc. of ASME Turbo Expo 2003, Power for Land, Sea, and Air, June 16-19, 2003, Atlanta, Georgia, USA, GT2003-38368

Cofer IV, J.I., April 1996, “Advances in Steam Path Technology”, *ASME Journal of Engineering For Gas Turbines and Power* Vol., 118, pp. 337-352

Dawes, W.N., 1988, “Development of a 3D Navier Stokes Solver For Application to All Types of Turbomachinery”, @ *ASME Paper* 88-GT-70.

Dawes, W.N. , 1990, “Towards Improved Through Flow Capability: The Use of 3D Viscous Flow Solvers a MultiStage Environment”, @ *ASME Paper* 90-GT-18.

Deckers, M., Simon, V. and Scheuerer, G., 1997, “The Application of CFD to Advanced Steam Turbine Design”, *International Journal of Computer Applications Technology*

Dejch, M.E. and Dejler, S., 1994, “Experimental Investigations of a Stator Cascade Generating Reduced Secondary Losses”, *Teploenergetika,* No. 10 (in Russian)

Dejch, M.E., Zariankin, A.E. and Filippov, G.A., 1960, “ The Method of the Efficiency Improvement in Turbine Stages with Short Blades”, *Teploenergetika* No. 2, pp. 18-24 (in Russian)

Denton, J.D., “Loss Mechanisms in Turbomachines,” *ASME Journal of Turbomachinery*, Vol. 115, pp. 621-656

Dodd, V.R., Dubner, R.M. and Caruso, W.J., September 1975, “Steam Turbine Diaphragm Failure”, *Proc. of the ASME Petroleum Mechanical Engineering Conference*, Tulsa, Oklahoma, USA.

Dragunov, V.K., Goncharov, A.L. and Nemytov , D.S., 2005, "Formation of Diffusion Interlayers in Electron Beam Welded Joints in Diaphragms of Steam Turbines", *Welding International*, Vol. 19, No. 9, pp. 717-723

Filippov, G.A. and Zhong-Chi Wand., 1964, "The Effect of Flow Twisting on the Characteristics of Guide Rows", *Teploenergetika,* pp. 54-57 (in Russian)

Foley, M., July 2000, "Retractable Brush Seals Allow Sweeping Improvements in Steam Turbine Efficiency", *Modern Power Systems*, pp. 37-39

Fuller, J.E. *,* 1989, "Electrical Discharge Machining", *ASM Handbook,* Vol. 16 (machining)*,* ASM International, pp. 557-564.

German, S.I. and Kulakova, G.N., 1957, "Investigation of the Welded Joints Produced by Semi-Automatic Sub-Arc Welding Shielded by Carbon-Dioxide", *Svarochnoe Proizvodstvo*, No. 10 (in Russian).

Goel, S., Singh, H. and Cofer IV, J.I., 1993, "Turbine Airfoil Design Optimization", @ *ASME Paper* 93-GT-161.

Hirota, Y., Mizutori, T. Watanabe, E, et al., 1985, "Recent Technology on Turbine Performance Improvement", *Technical Review-Mitsubishi Heavy Industries*, Vol. 22, No. 3, pp. 234-240

Holmes, D.G. and Tong, S.S., 1985, "A Three-Dimensional Euler Solver For Turbomachinery Blade Rows", *ASME Journal of Engineering For Gas Turbines And Power*, Vol. 107, pp. 258-264

Houldcroft, P.T., 1977, "Submerged-Arc Welding", *Woodhead Publishing Ltd.*, Cambridge, England.

Ingultsov, V.L., 1958, "Stress Calculation of a Diaphragm as a Semicircular Ring Supported by an Elastic Contour", *Energomashinostroenie,* No. 1 (in Russian)

Jansen, M. and Ulm, W., 1995, "Modern Blade Design for Improving Steam Turbine Efficiency", *Proc. 1st European Conference on Turbomachinery-Fluid Dynamic and Thermodynamic Aspects*, Erlangen.

Kapinos, V.M., Matveev, Yu.Ya., Pustovalov, V.N., 1983, "Natural Convection of Unventilated Cavities of Steam Turbine Rotors", Moscow, *Heat-and-Power Engineering,* No. 8, pp. 36-39 (in Russian)

Kawagishi, H. and Kawasaki, S., 1991, "The Effect of Nozzle Lean on Turbine Efficiency", *PWR,* Vol. 13*, Design, Repair and Refurbishment of Steam Turbines,* ASME.

Kostyuk, A.G. , 1982, "Dynamics and Strength of Turbomachinery", *Mashinostroenie*, Moscow (in Russian).

Kostyuk, A.G., 1972, "Theoretical Analysis of Aerodynamic Forces in a Turbomachine Labrything Seals", *Teploenergetika,* No. 11, pp. 29-33 (in Russian)

Kulagina, G.F. ,1960, "Experimental Investigation of the Diaphragms Stresses and Deflections", *LMZ Collection "Investigations of the Components for Steam and Gas Turbines, and Axial Compressors"*, LMZ, Vol. 6, Mashgiz (in Russian)

Lakshminarayana, B., et al., 1973, "Characteristics of the Wake Behind a Cascade of Airfoils," *Journal of Fluid Mechanics*, Vol. 61, Issue 04, pp. 707-730.

Langston, L.S., October 1980, "Crossflows in a Turbine Cascade Package", *ASME Journal of Engineering for Power,* Vol. 102, No. 4, pp. 866-874

Lazzeri, L., Cecchi, S., Maretto, A. and Torre, A. , 1996, "Steam Turbine Design Development for High Sufficiency Power Plants with Supercritical Inlet Steam Conditions", *Proc. Power-Gen Europe '96*, Budapest Vol. 2, pp. 567-574

Leyzerovich, A. Sh., 1997, "Large Rower Steam Turbines: Design and Operation", Vol. 2, *PennWell Publishing Company*, Tulsa, Oklahoma

Little, N., Sulda, E. and Terezakis, T., 2001, "Efficiency Improvement of Large Mechanical Drive Steam Turbines", *Proc. of the Texas A&M 30th Turbomachinery Symposium,* Houston, TX, USA

Lomakin, A.A., 1940, "Axial Pressure in the Centrifugal Pumps With Take Into Account a Clearance", Moscow, *Soviet Boiler Fabrication*, No. 12, pp. 431-437 (in Russian)

Matsui, S., Matsumura, H., Yasuda, K. et. al., 1987, "The Application of Laser Cutting and Electron Beam Welding to the Manufacturing of Turbine Nozzle Diaphragms", *Welding in the World*, Vol. 25, No. 1/2, pp. 16-24

McIntosh, J., MacPherson, R., Ingram, G. and Hogg, S., 2011, "Profiled Endwall Design Using Generic Algorithms with Different Objective Functions", @ Paper No. GT 2011-45836, *Proc. of ASME Turbo Expo 2011*, Vancouver.

Mellerovich, G.M. and Bliznyukova, N.M., 1961, "Tenzometric Investigations of the Steam Turbines Components at the KhTGZ", *Energomashinostroenie,* No. 2 (in Russian)

Moaveni, Saeed., 2008, "Finite Element Analysis: Theory and Application with ANSYS", 3rd edition, *Pearson Prentice Hall*.

Moroz, L., Tarasov, A., 2004, "Flow Phenomenon in Steam Turbine Disk-Stator Cavities Channeled By Balance Holes", Proc. of ASME Turbo Expo 2004, Power for Land, Sea, and Air, June 14-17, 2004, Vienna, Austria

Muster, D.F. and Sadowsky, M.A., 1956, "Bending of a Uniformly Loaded Semicircular Plate Simply Supported Around the Curved Edge and Free Along the Diameter", *ASME Journal of National Applied Mechanics,* Vol. 23, No. 3, pp. 329-335

Naumov, V.K., 1960, "Stress Calculation of the Steam and Gas Turbines Diaphragms", *LMZ Collection "Investigations of the Components for Steam and Gas Turbines, and Axial Compressors"*, LMZ, Vol. 6, Mashgiz (in Russian)

Nedzvetzkiy, V.G. and Lagerev, V.V., 1974, "Some Specifics of Gas Flow Through Equalizing Holes in Turbine Disks", *Energomachinostroyenie*, No. 5, pp. 38-41 (in Russian)

Neef, M., Sürken, N., Sulda, E. and Walkenhorst, J., May 2006, "Design Features and Performance Details of Brush Seals for Turbine Applications", *Proc. of ASME Turbo Expo 2006-Power for Land, Sea, and Air*, Barcelona, Spain

Oeynhausen, H., Drosdziok, A. and Deckers, M., 1996, "Dampfturbinen für die Neue Kraftwerksgeneration", *VGB Kraftwerkstechnik,* Vol. 76, pp. 974-979 (In German)

Olimpiyev, V.I., 1976, "Resolving Problem of the Low-Frequency Vibration in Large Steam Turbine Rotor System", *Teploenergetika,* No. 9, p. 8-13 (in Russian)

Olimpiyev, V.I., Orlov, I.I. and Kosarev, V.V., 1975, "Problems Regarding Elimination of Low-Frequency Vibration in Large Steam Turbines", *Energetik,* No. 5, pp. 10-12 (in Russian)

Pakhomov, G.I., 1934, "Deflections of the Steam Turbine Diaphragms", *Sovetskoe Kotloturbostroenie,* No. 6 (in Russian).

Paul de, M.V., Wallon, M. and Annis, A., 1989, "Twenty Years Progress In Steam Turbine Aerodynamics", *Proc. of American Power Conference*, Vol. 51, Chicago, pp. 166-172

Piatt, B.J., 1998, "Appropriate Turbine Repairs", *1998 International Joint Power Generation Conference* Vol. 2, pp. 263-266

Pigott, E. and Warner, R.E., October 1986, "Steam Turbine Blade Developments", *IEE/ASME Joint Power Generation Conference*, Portland, Oregon.

Runov, B.T., 1982, "Research and Prevention of Steam Turbine Vibration*", Energoatomizdat*, Moscow (in Russian)

Sakamoto, T., Kawagishi, H. and Kawasaki S., 1993, "Three-Dimensional Viscous Flow Analysis of Turbine Stage with Leaned Nozzle", *Proc. JSME-ASME Int. Conference on Power Engineering-'93*, Vol. 2, pp. 27-32

Samoylovich, G.S. and Morozov, B.I., 1957, "Flow Coefficients Through Equalizing Holes in the Steam Turbine Disks", *Teploenergetika*, No. 8, pp. 18-28 (in Russian)

Schiller, S., Heisig, U. and Panzer, S., 1982, "Electron Beam Technology", *John Wiley and Sons,* Berlin, Germany

Schofield, P., 1981, "Maintaining Optimum Steam Turbine-Generator Thermal Performance", *Missouri Valley Electric Association 1981 Engineering Conference,* Kansas City, Mo.

Sentsov, N.D., 1958, "Some Investigation Results of Stresses and Deflections in Steam Turbine Welded Diaphragms", *Energomashinostroenie,* No. 8 (in Russian)

Sentsov, N.D., 1956, "Welded Diaphragms of the LMZ Steam Turbines", LMZ Collection "Developments of the Machine Design and Technology in Large Steam-Hydromachinery", Mashgiz (in Russian).

Shcheglyaev, A.V., 1993, "Steam Turbines" (In 2 volumes), 6th edition, Revised and expended by B.M. Troyanovsky, *Energoatomizdat*, Moscow (in Russian)

Shelton, M.L., Gregory, B.A. and Lamson, S.H., 1993, "Optimization of a Transonic Turbine Airfoil Using Artificial Intelligence, CFD and Cascade Testing", @ *ASME Paper* 93-GT-161.

Shvetz, I.T. and Dyban, E.P., et al., 1960, "Defining Flow Coefficients of the Rotating Channels", *Proc. of Ukranian Academy of Science Thermal Engineering Institute*, Kiev, No. 18, pp. 16-27 (in Russian)

Sieverding, C.H., January 1975, "Secondary Flows in Turbomachines", @ *VKI Lecture Series 72*.

Singh, G. Walker, P.J. and Haller, B.R., 1995, "Development of Three-Dimensional Stage Viscous Time Marching Method for Optimization of Short Height Stages", *Proc. 1st European Conference on Turbomachinery-Fluid Dynamic and Thermodynamic Aspects*, Erlangen, pp. 1-9

Smith, D.M., 1938, "Deflections and Stress in Built-Up Half Diaphragms", *Proc.of the Fifth International Congress for Applied Mechanics,* pp. 62-65

Sulda, E, 1999, "Retractable Brush Seal Optimizes Efficiency and Availability for Cycling and Baseload Steam Turbines", *Power Engineering*, Vol. 103, No. 11, pp. 96-101.

Sun, Z. and Karppi, R., 1996, "The Application of Electron Beam Welding for the Joining of Dissimilar Metals: An Overview", *Journal of Materials Processing Technology*, Vol. 59, pp. 257-267

Tanuma, T., Nagao, S., Sakamoto, T., Kawasaki, S., Matsuda, M. and Imai, K. , 1995, "Aerodynamic Development of Advanced Steam Turbine Blades", *PWR*, Vol. 28, *Proc. 1995 joint Power Generation Conference ASME,* pp. 367-374

Taylor, V.C., 1951, "Stress and Deflection Tests of Steam Turbine Diaphragms", *Trans. ASME*, Vol. 73, pp. 877-890

Tran, M., October 1986, “Recent Developments in Blading to Improve Turbine Efficiency”, @ *ASME Paper* 86-JPGC-PWR-35

Traupel, W., 1977, *“Thermal Turbomachinery”,* 3rd ed., Springer, Heidelberg, (in German).

Troyanovsky, B.M., 1991, “Steam Turbines for Power Plants (New and Modernized Units)”, *Thermal Engineering*, Vol. 38, No. 11, pp. 637-646

Troyanovsky, B.M., 1993, “Ways for Improving the Efficiency of Steam Turbines”, *Thermal Engineering*, Vol. 40, No. 5, pp. 386-393 and No. 7, pp. 528-536

Trukhny, A.D., 1990, “Stationary Steam Turbines, *2nd ed., Energoatomizdat*, Moscow (in Russian)

“TurboCare Retractable Packing”, 1999, LGGETRR $^{8}/_{99}$, *Demag Delaval Turbomachinery Corp. Reference Library*.

Turner, M.G., Liang, T., Beauchamp, P.P. and Jennions, I.K., 1993, “ The Use of Orthogonal Grids in Turbine CFD Computations”, @ *ASME Paper* 93-GT-38.

Wahl, A.M., 1930, “Stess and Deflections in Flat Circular Plates with Central Holes”, *Trans. ASME*, Vol. 52, pp. 29-43

Wahl, A.M., 1932, “Strength of Semicircular Plates and Rings Under Uniform External Pressure”, *Trans. ASME*, Vol. 54, pp. 311-320

Wilson, M., Pilbrow, R., Owen, J.M., 1997, “Flow and Heat Transfer in a Preswirl Rotor-Stator System”, *ASME Journal of Turbomachinery*, Vol. 119, pp. 364-373

Zemzin, V.N. and Frenkel, L.D., 1960, “Welded Components of Steam and Gas Turbines”, *Mashgiz*, (in Russian).

AKNOWLEDGEMENTS

The authors wish to express their gratitude to the following people for their active participation in the entire project, and valuable assistance in preparing and organizing this paper: Carolyn B. Smith, Kenneth E. Bruce, Timothy Ewer, Russel V. Caggiano, Tyler H. Aaron, Shannon Pedersen, and Tariq Al-Shaikh.