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BRUSH SEALS FOR THE NO. 3 BEARING OF A MODEL 7EA GAS TURBINE

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ABSTRACT

Gas Turbines in power generation are frequently of the single rotor type. The rotor is directly connected to the electrical generator. The rotor may be supported by two journal bearings or in some cases there is an additional journal bearing situated between the axial compressor discharge and the gas turbine intake. This third bearing serves to provide the rotor with additional support required to reduce rotor dynamic instabilities. The third bearing is, therefore, inside the machine housing and a significant amount of maintenance work is necessary to inspect it. The third bearing is also exposed to elevated temperatures by, essentially, being surrounded by compressor discharge air. A certain amount of compressor discharge air leaks through the seals into the cylindrical space around the third bearing housing and from there, due to significant pressure gradients, into the third bearing. Labyrinth seals are provided to impede air leakage from the pressurized cylindrical space into the bearing cavity. The air that leaks into the bearing housing mixes with a buffer air stream. This buffer air stream serves to cool the bearing cavity and to prevent leakage of hot, high-pressure air into the bearing cavity. Two dry air streams are then routed into the atmosphere via the coaxial space formed by two cylindrical surfaces. The portion of the buffer air stream contacting the bearing lubricating oil is de-misted in a special de-mister vessel. The de-misted air is exhausted into the atmosphere and the separated oil is returned to the gas turbine lubricating oil reservoir. This Paper discusses the introduction of brush seals into the No. 3 bearing housing as an additional element in retarding the high pressure, high temperature air infiltration into the No. 3 bearing housing.

INTRODUCTION

Watson Cogeneration Plant, located in the Los Angeles Refinery, California, operates four GE-made, model 7EA, heavy industrial gas turbines.The turbine rotor assembly is supported by three bearings. The No. 3 bearing is located inside the machine between the compressor and the turbine rotor sections. The No. 3 bearing housing is completely surrounded by high-pressure hot air. The buffer air stream is extracted from the fifth stage of the compressor and then routed into the No. 3 bearing housing. A fixed orifice installed in the extraction line controls the buffer airflow. The buffer air serves to prevent eventual mixing of the lubricating oil stream with the air surrounding the bearing housing. It also serves to cool the No. 3 bearing housing. However, a certain amount of the buffer air stream contacts the lubricating oil stream. The oily air stream is separately routed via coaxial pipe into the de-misting vessel. Inside the vessel the oil is separated from the air. The separated oil is collected at the bottom of de-misting vessel and, from there, drained back into the turbine lubricating oil reservoir. Clean, de-misted air is then exhausted into the atmosphere. The de-misted air, however, still contains a small amount of oil fumes that are invisible to the naked eye. Evaluation of the irreversible air losses must incorporate the following air streams escaping into the surrounding atmosphere:

- oil de-misted air
- mixture of buffer air and high pressure air

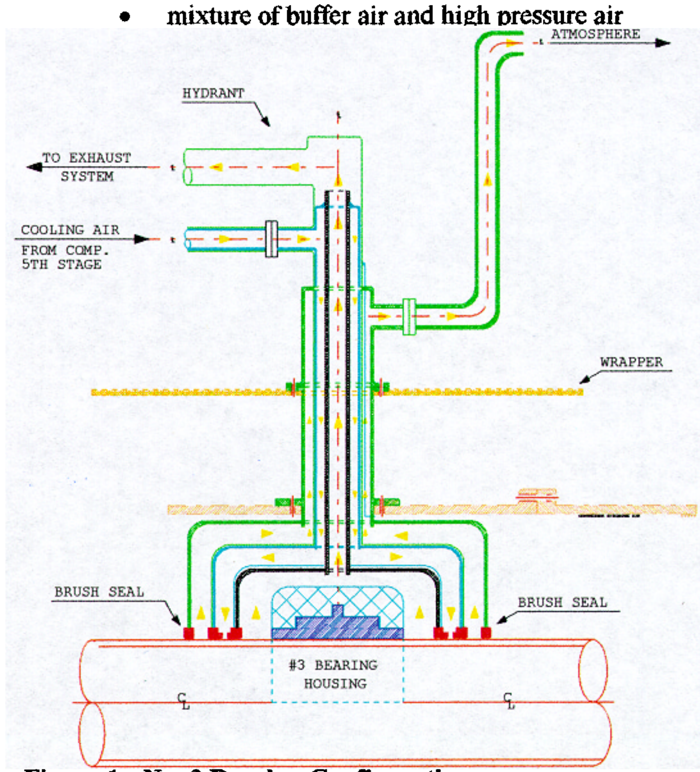

Figure 1 - No. 3 Bearing Configuration

Figure 1 shows the complex routing of various air streams that represent irreversible losses. However, a certain portion of the high pressure, high temperature air stream does not leak into the bearing No. 3 housing. It leaks, instead, internally, just behind the first stage turbine nozzles. Here it joins the hotter gas stream being expanded through the turbine. Figure 1 also shows three sets of labyrinth seals:

- Outer set
- Middle set
- Inner set

The outer set of labyrinth seals serves to control high pressure, high temperature air leakage into the bearing housing.
The middle set of labyrinth seals serves to contain the buffer gas and to further reduce high pressure, high temperature air leakage.
The inner set of labyrinth seals serve to reduce the leakage of the buffer air into the No. 3 bearing and to prevent eventual lubricating oil leakage into the cylindrical cavity surrounding the No. 3 bearing housing.
The airflow via the labyrinth seals even when they are in a new, damage-free condition is significant. Special calibrated Pitot tubes were used to measure parasitic air losses into the surrounding atmosphere. The amount of air measured in the silencer end of the pipe represents the combined portions of the air streams coming from the high pressure, high temperature air and from fifth stage extraction of the axial compressor. Table I shows the calculated air losses in the outer set of labyrinth seals for several design options:

Table I – Outer Labyrinth Performance Comparison

Seal Type	Calculated Leakage Flow (LB/HR)	Measured Leakage	Description
Labyrinth, OEM	4,200 per side	NA	9 teeth, slanted (1)
Labyrinth by TurboCare	3,440per side	NA	9 teeth straight (2)
Labyrinth by TurboCare	2,700 per side	NA	7 teeth straight (3)

(1) original radial clearance
(2) original radial clearance
(3) reduced radial clearance

The total leakage at the stack is the sum of the calculated leakage of high pressure, high temperature air through the outer labyrinths and the portion of the buffer air leakage through the middle labyrinths into the stack. From the stacks the mixture of the two air streams escapes into the surrounding atmosphere. The mixture of the two air streams is still hot. The mixture temperature may exceed 550 degrees F by the time it is released into the atmosphere. The field measurements were taken at the top of the clean, hot air exhaust stack. A Pitot tube calibrated instrument was used to determine the airflow at the top of the stack. Four airflow measurements were recorded. The measurements indicated a total air loss into the surrounding atmosphere equal to 10,400 pounds per hour. These measurements may not be accurate enough because at the top of the stack there are significant air exit losses. The operating time each turbine unit accumulates per year is approximately 8,000 hours. Hot air leakage into the surrounding atmosphere is, therefore, significant. Furthermore, the air that leaks into the surrounding atmosphere originates from the intake air house where it is filtered and cooled. Filtered and cooled intake air is then, in turn, drawn into the compressor intake. In other words, considerable efforts are required to deliver clean, cooled air to the compressor. The filtered and cooled air is then compressed stage-by-stage inside the compressor until the air reaches the desired delivery pressure. Any internal air leaks and especially irreversible losses into the surrounding atmosphere are costly. A significant amount of compressor discharge air leaks into the cylindrical cavity that is part of axial compressor discharge housing. The pressure of air in the cavity is about 50% of the compressor discharge pressure. The temperature of the air inside the cylindrical cavity may approach 700 degrees F. The air that leaks from the compressor discharge into the cylindrical cavity still has high internal energy. The axial compressor intake air must be properly filtered and in this particular case cooled. This means that extra expenses are accrued to deliver clean, cooled, compressed air to the combustion liners. Figure 1 shows the loss of compressor discharge air that infiltrated the cylindrical space around bearing No. 3 and, in turn, leaked through the bearing No. 3 outer labyrinths into the space where the buffer air is present. The mixture of these two streams is still hot and the amount of leakage is significant. To reduce the original source of the compressor discharge air leaking into the cylindrical cavity a brush seal was installed in front of the compressor discharge labyrinths. The field results showed reduced air leakage from the compressor discharge into the cylindrical space surrounding the No. 3 bearing housing. Subsequent examination of the compressor discharge seal revealed a moderate loss of bristles in the region of the compressor casing horizontal split. A decision was made to further reduce air parasitic losses and to install a set of the brush seals in the No. 3 bearing. The study of parasitic losses indicated that there was a good opportunity to reduce the air leakage. The reduction of the air leakage could be obtained by:

- reducing labyrinth radial clearances
- increasing the number of labyrinth teeth
- modifying the labyrinth teeth
- using rub-tolerant material
- installing brush seals

Figure 2 shows the modified bronze labyrinth seal with a brush seal situated at the midpoint of the labyrinth teeth.

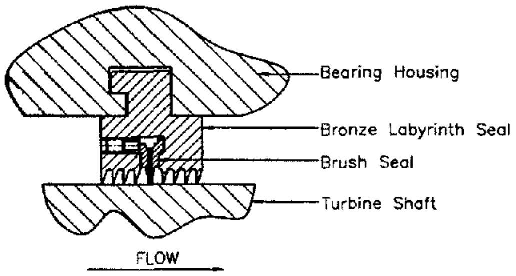

Figure 2 - New Labyrinth Seal and Brush Seal

Figure 3 compares the air leakage flows for:

- OEM labyrinth seal
- labyrinth seal furnished with a brush seal

Note that the new labyrinth seal in addition to tighter radial clearances is provided with an extra labyrinth tooth. The new form of the labyrinth tooth was selected for the ease of machining and for efficient sealing. Examination of Figure 3 shows that for a 60 psi total pressure drop across the seals the "flow factor" for the new labyrinth seal furnished with the brush seal is slightly less than half of the OEM-designed labyrinth seal. In other words, the parasitic air leakage (from the cylindrical cavity surrounding the No. 3 bearing housing into the No. 3 bearing housing) is cut virtually in half.

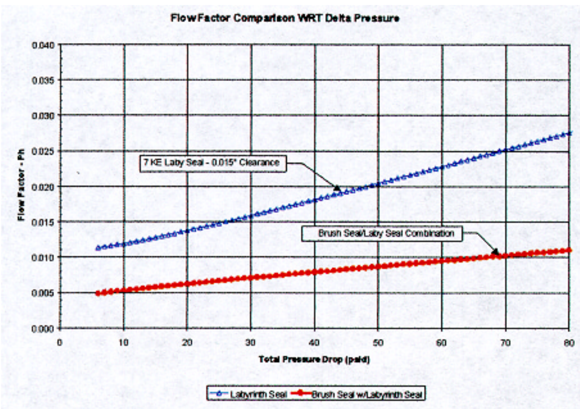

Figure 3 - Leakage Flow Comparison

MANUFACTURE and FIELD RESULTS

TurboCare and WCC started to design the outer set of labyrinth seals with the provision to install additional brush seals. The operating conditions and proximity of the bearing seals to the No. 3 bearing indicated that the original labyrinth clearances could be safely reduced. The operating conditions during start up, shut down, and steady state base load operation were evaluated to ensure that the labyrinth seal teeth would not contact the rotor journal and that the brush seals were not working for a prolonged period of time under a rub condition. Therefore, the selection of the brush seal location along the labyrinth seal is very important. The pressure gradients across the labyrinth seal teeth, relative displacements of the labyrinth seal housing and the rotor displacements had to be taken into consideration. Figure 2 shows the brush seal installed behind the 4th labyrinth tooth. The brush seal holder " hook fit "was not modified because the bearing housing shell had to be re-used. The new labyrinth seal was made from the rub-tolerant nickel-leaded bronze suitable to operate at the particular operating conditions. The form of the new labyrinth seal teeth was significantly modified to ensure better air sealing and to simplify the manufacture process. The original radial clearance and calculated air leakage were both reduced by 33%. This reduction in air parasitic losses does not include the benefit from the proposed brush seal modification.

The brush seal design must incorporate the size of the wire, density of the bristle pack, material of the wire and the bristle pack stiffness. The stiffness of the brush seal must be optimized to ensure proper sealing and minimal wear of the brush wires.

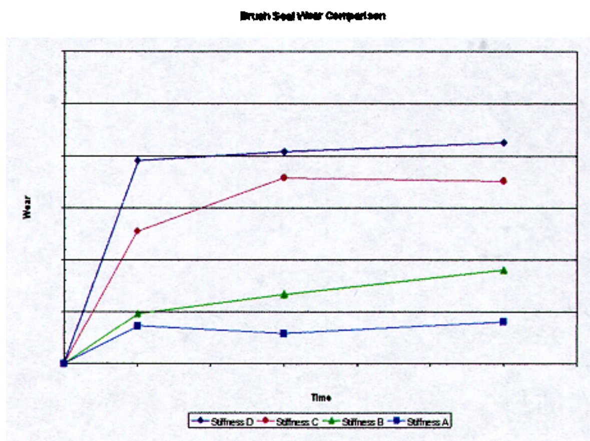

Figure 4 - Brush Seal Wear Comparison

Figure 4 shows brush seal wear as a function of the operating hours. The wear of four different brush seal stiffness values is compared. Without proper consideration of the brush seal stiffness, a groove in the shaft could be formed during prolonged interference with the bristle pack. This must be avoided because any significant surface groove is in actuality a stress riser. In reality, the possibility of this scenario is remote because the brush seals are in close proximity to the No. 3 bearing. For the specific application the brush wire diameter and material was selected based on 0.010 inch of shaft radial displacement and the bristle wear of 0.002 inch during a 60 minute continuous rub. The bristles are firmly wedged between the front and back plate. The back plate supports the bristle pack and prevents it from deflecting downstream due to significant pressure gradients across the teeth. Figure 5 shows the elements of the brush seal and a description of the various parts.

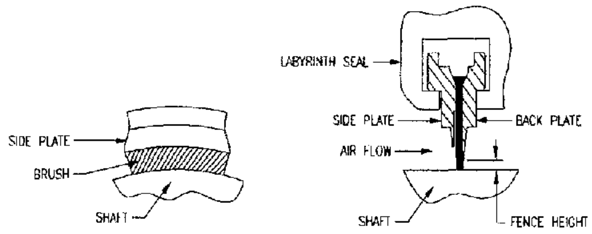

Figure 5 - Brush Seal Elements and Nomenclature

The back plate radial clearance is sometimes slightly larger than the unsupported bristle length. The unsupported bristle length is shown as the " fence height ". The difference in the radial direction of the back plate clearance and the "fence height" represents the cold brush seal clearance. The selection of the bristle lay angle is also important. The shaft circumferential velocity, shaft material and the material and diameter of the bristle wires dictate the magnitude of the bristle lay angle. Haynes 25 alloy was selected for the wire material. The brush seal holder was made from 400 series stainless steel. The brush seal holder is comprised of the side plate and bristle pack back plate. The bristles are, therefore, sandwiched between these two plates. The back plate radial clearance is selected to avoid rotor-to-holder contact and also to ensure proper axial support to the bristle pack.

The brush seal assembly was fabricated in two halves with an anti-rotational device. Special setscrews were provided to firmly hold the brush seal assembly against the labyrinth seal. The new set of outer labyrinth seals was installed during the winter of 1999. The gas turbine unit was restarted in March of 1999. The same Pitot tube impact type flow meter was used to measure the air leakage at the top of the stack. Table II shows the air leakage into the atmosphere before and after the modification of the outer set of labyrinth seals.

Table II – Seal Performance Summary

Description	Before	After
Seal type	labyrinth	laby+brush
Laby material	Bronze alloy	Nickel-lead-bronze
Number of teeth	7	8
Type of teeth	inclined	straight
Radial clearance %	100	65
Total loss @ stack	10.400 lbs./hour	5,720 lbs./hour
Temp at stack	525 F	465 F

The temperature reduction at the top of the stack is due to a significant reduction of high pressure, high temperature airflow from the inner cylindrical enclosure into the bearing No. 3 housing. The heat and mass balance calculations (out of the scope of this paper) can determine the reduction in flow of each air stream. As indicated before the air stream leaving the stack and exhausting into the atmosphere is comprised of a portion of the buffer air stream and the air stream coming from the inner cylindrical surface. The buffer air stream temperature could not be measured. However, the pressures were available and the temperature was estimated at 210 degrees F. The loss of high pressure, high temperature air is even more expensive. Referring to Figure 1, a portion of the buffer air contacts the lubricating oil and the oil mist-laden air is then routed, via the de-mister, into the atmosphere. The new outer set of labyrinth seals (furnished with the brush seal) reduce the amount of high pressure, high temperature airflow. The reduced amount of this air stream contacts a portion of the buffer air prior to being exhausted, together, via the stack into the atmosphere. This reduced air jet will pull less buffer air into the atmosphere. The de-mister system may see less loading of oil mist-laden air though this was not verified in the field.

In addition, since the leakage of hot air into No. 3 bearing is significantly reduced, the bearing housing temperatures are reduced and the life expectancy of the No. 3 bearing may be extended.

The remainder of the gas turbine units will be retrofitted with the bearing No. 3 brush seals in the year 2000. During a major inspection is an opportune time for the installation of the new labyrinth seals and new brush seals. A particular plant may justify earlier installation. For example, a location where the fuel is very expensive may result in an economic study that justifies additional expenditures associated with the installation of sealing elements in the bearing No. 3 prior to the next scheduled overhaul.

CONCLUSIONS

The installation of the brush seals into the modified set of labyrinth seals proved to be beneficial in reducing the non-return air parasitic losses.

The goal to reduce the expensive air leaks into the surrounding atmosphere was achieved and exceeded.

The irreversible air losses were reduced approximately 45%.

The clean, dry, hot air losses are comprised of the following air streams:

- 5^{th} stage of extraction to the No. 3 bearing
- compressor discharge leakage into the No. 3 bearing

The leakage loss of compressor discharge air into the surrounding atmosphere is irreversible and expensive. OEM data indicates that installation of the brush seals may save up to 340kw at the particular location. In conclusion, installation of the brush seals in the No. 3 bearing labyrinth seals is beneficial from a reliability and economical standpoint.

REFERENCES

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APPENDIX

The air leakage flow can be calculated with an acceptable degree of accuracy knowing the labyrinth seal geometry, number and shape of labyrinth teeth, labyrinth material and air differential pressure across the labyrinth teeth. Frequently, the last labyrinth tooth may be choking for certain air differential pressure gradient. The air leakage flow can be calculated as follow:

$$W = 25KA\sqrt{\frac{1-(P_2/P_1)^2}{N-\log_e(P_2/P_1)}}\sqrt{\frac{P_1}{\vartheta_1}} \qquad (1)$$

Where W=labyrinth air flow, lb./hr

K=factor for labyrinth type

A=leakage area, sq. in

N=number of labyrinth teeth

$$A = \frac{\pi}{4}\left(D_l^{\,2} - D_j^{\,2}\right) \qquad (2)$$

Where D_l = labyrinth bore diameter, in and

D_j =shaft journal diameter, in

Since the radial gap δ is a small value in comparison to shaft diameter, equation 2 can be simplified:

$$A \cong \pi\delta D_l \qquad (3)$$

P_1 = initial pressure, psia

P_2 =final pressure, psia

$\vartheta_{1=}$ =initial, air specific volume, cu ft/lb.

Values for the K factor can be found in Table III.

Table III – Values for K

Radial clearance in mills	Factor K
10	75
20	60
30	50
40	45
50	40

The K factors are empirical, and based on test data. Equation 1 (1) is identified as Martin's Formula.