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Gentlemen:

SUBJECT: DE-AC08-79ET15426, R & D for Improved
Efficiency Small Steam Turbines,
Test Report - RIT Rotor Vibration Testing

Please find attached the subject Test Report for your
records. Do not hesitate to call if you have questions.

Very truly yours,

NORTHERN RESEARCH AND ENGINEERING
CORPORATION

Robert M. Sexton, Jr.
Program Manager

RMS/pjc

Encl.

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Project No. 1380
September 27, 1982

Northern Research and Engineering Corp.
39 Olympia Ave.
Woburn, MA 01801

TEST REPORT

RIT ROTOR VIBRATION TESTING

Summary - An RIT "B" rotor, including the impeller and shaft, was examined experimentally to determine vibratory characteristics. It was concluded that there are no specific speeds within the operating range with adequate resonance encroachment margins. It is recommended that performance tests be carried out with caution. Testing was performed by the Terry Steam Turbine Division for Northern Research and Engineering Corp. and the Department of Energy under Contract DE-AC03-79ET15426.

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TERRA

STEAM TURBINE DIVISION

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Copy No. 2**TECHNICAL MEMORANDUM**

PREPARED UNDER DEPARTMENT OF ENERGY
(DOE) CONTRACT DE-A003-79ET15426
FOR NORTHERN RESEARCH AND ENGINEERING
CORPORATION (NREC).

TM- 208

Title:

TEST REPORT - RADIAL INFLOW TURBINE
(R.I.T.) "B" ROTOR BENCH TESTING

Distribution:Control
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Written By: G. L. Chartier *GL Chartier* Date: September 7, 1982Approval By: J. G. Mosimann *JG Mosimann* Date: September 7, 1982Recommendations:

1. R.I.T. performance testing should be carried out with caution. There are no specific speeds in the operating range with adequate resonance encroachment margins.
2. For future use, impeller damping should be investigated.

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MASTER

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BACKGROUND:

To experimentally analyze the vibratory characteristics of a typical R.I.T. rotor, the reference contract specified both shake table and impulse vibration testing. The specific objectives were to:

1. Determine blade natural frequencies and display their mode shapes.
2. Correlate the results of the FEM (Finite Element Method) vibration analysis with those of the experimental analysis.
3. Determine the general level of damping.
4. Indicate likely high vibratory stress points on the impeller.

NREC provided a spare "B" rotor, including impeller and shaft, to serve as the test piece. Terry received the rotor and performed the testing in its Mechanical Engineering Laboratory. This TM presents the results of the testing.

CONCLUSIONS:

1. There were a plethora of natural frequencies found during test, most of which were associated with highly coupled blade, splitter, and disk modes. These frequencies fill the operating speed range with 1E and 2E vane passing excited resonances.
2. The first four blade-only natural frequencies and associated mode shapes, as determined by test, verify the results of the FEM analysis. Frequencies matched within 6%.
3. This solid impeller exhibits high sensitivity reflecting only low material damping, usually between .03 and .10% of critical damping (.20 to .60% log decrement critical damping), depending on the mode.
4. The test mode shapes indicate maximum vibratory bending stress along the blade to disk interface, primarily at blade leading and trailing edges.

RESULTS:

FIGURE NO.

1	Impulse excited "B" impeller test set-up.
1a	Close-Up of proximeter and impulse hammer.
2	Shake table excited "B" impeller test set-up.
3	Campbell diagram: Comparison of test vs. analysis.
4	Campbell diagram: Additional test frequencies.
5,6	Typical transfer functions.
7-10	Test mode shapes for blade frequencies.
11,12	Circle fit diagrams for damping evaluation.
13-16	F/E analysis mode shapes.
17	Strain gage locations determined by test.

DISCUSSION:

The Campbell Diagram of Figure No. 4 displays the frequencies found during testing. Included are those associated with modes involving highly coupled blade, splitter, and disk motions. Figures No. 5 and 6 show these frequencies (up to 12 KHZ) identified by typical transfer functions developed using the impulse technique. Some transfer functions, such as Figure No. 5, were developed using proximity probe output while others, such as Figure No. 6, were based on strain gages required to detect the very small distortions of the higher vibratory modes. There were many other frequencies above 12 KHZ and, in truth, the entire turbine operating speed range will be filled with resonances. Whether these resonances will be highly excited during operation remains to be seen. The 9E and 18E excitation lines are on the Campbell Diagrams to represent the first and second harmonics of vane passing frequency; there are nine stationary vanes.

The Campbell Diagram in Figure No. 3 shows good agreement between analysis and test results for blades only. The range from blade to blade in first mode natural frequency of 3600 to 3700 Hz., as found by test, is due to machining tolerances. The fact that the FEM analysis provided frequencies for all four modes lower than those found in test shows the consistency of the results.

The level of damping for the frequency peaks found was between .03 and .10% of critical. In terms of log decrement the values increase by a factor of 2π to become .20 to .60% log decrement damping. These values imply material damping only. Mechanical damping is virtually non-existent.

The worst stress locations were derived from the test mode shapes. A criterion of maximum concentrated stress due to blade bending was adopted. This lead immediately to locations in the blade fillet area. Since the blade root is cantilevered off the disk surface everywhere along its length, the maximum bending stress locations were easily extracted from the mode shapes by determining points of maximum transverse motion of the set of nodes in the Zonic blade model adjacent to the points of fixity to the disk.

The Appendix contains a description of the processing techniques, schematics of the set-up, and a detailed list of the equipment used.

REFERENCE:

Test Plan - Research and Development for Improved Efficiency, Revision A, Small Steam Turbine Project No. 1380, phase II, August, 1981, pp. 7-9, Northern Research and Engineering Corporation, Department of Energy Contract No. DE-AC03-79ET15426.

FIGURE 1: IMPULSE EXCITED "B" IMPELLER TEST SETUP

FIGURE 1A: CLOSE-UP OF PROXIMETER AND IMPULSE HAMMER

FIGURE 2: SHAKE-TABLE EXCITED "B" IMPELLER TEST SETUP

CAMPBELL DIAGRAM

CORRELATION OF R.I.T. SPARE
"B" ROTOR TEST DATA WITH
ANALYTICAL RESULTS FOR
LARGE BLADES

FIGURE NO. 3

46 1323

CAMPBELL DIAGRAM

COMPLETE SET OF
FREQUENCIES FOUND
(UP TO 12 KHz) DURING
R.I.T. SPARE "B" ROTOR
TESTING. DASHED LINES
DENOTE PURE BLADE MODES.

FIGURE NO. 4

TYPICAL TRANSFER FUNCTION FOR "B" ROTOR RESONANCE IDENTIFICATION (0-6 KHZ) FIGURE NO. 5

2/1

6080 MULTICHANNEL FFT ANALYZER

#AUGS=15

FNC# = 26

TRANSFER FUNCTION - PROXIMITY
PROBE DIVIDED BY FORCE - (INCHES / LBF)

TYPICAL TRANSFER FUNCTION FOR "B" ROTOR RESONANCE IDENTIFICATION (6-12 KHZ) FIGURE NO. 6

2/1

6080 MULTICHANNEL FFT ANALYZER

#AVGS=20

FNC#=26

TRANSFER FUNCTION - STRAIN GAGE
OUTPUT (AMPLIFIED) DIVIDED BY FORCE ($\mu\text{IN./IN./LBF.}$)

ANIMATED PLOT

VIEWPOINT:

X= -1.00

Y= 1.00

Z= 1.00

Scale of
Defl.= 1

Mode # 1

Freq. (Hz)= 3605

FIGURE 7 - RADIAL INFLOW TURBINE "B" IMPELLER

EXPERIMENTALLY DETERMINED MODE SHAPE FOR MODE 1

ANIMATED PLOT

VIEWPOINT:

X= -1.00

Y= 1.00

Z= 1.00

Scale of

Defl.= 1

Mode # 2

Freq. (Hz)= 9382.5

FIGURE 8 - RADIAL INFLOW TURBINE "B" IMPELLER

EXPERIMENTALLY DETERMINED MODE SHAPE FOR MODE 2

ANIMATED PLOT

VIEWPOINT:

X= 1.00

Y= 1.00

Z= 0.00

Scale of
Defl.= 1

Mode # 3

Freq. (Hz)= 11,783

FIGURE 9 - RADIAL INFLOW TURBINE "B" IMPELLER
EXPERIMENTALLY DETERMINED MODE SHAPE FOR MODE 3

ANIMATED PLOT

VIEWPOINT:

X= 1.50

Y= 0.50

Z= 0.00

Scale of

Defl.= 1

Mode # 4

Freq.(Hz)= 13,089

FIGURE 10 - RADIAL INFLOW TURBINE "B" IMPELLER

EXPERIMENTALLY DETERMINED MODE SHAPE FOR MODE 4

R. I. T. "B" ROTOR IMPELLER BLADE

CIRCLE FIT
 Mode #1
 Freq. (Hz)
 = 3.660E+003
 Pt. #7
 Resp. Dir = -Z
 Coord. of Ctr.:
 X = -3.036E-007
 Y = -5.052E-007
 Radius
 = 2.151E-007
 Ave. Err. (%)
 = 10.84
 Fit range:
 3630.000
 3690.000

Modal
 Parameters:
 N. Freq. (Hz)
 3.659E+003
 M = 4.816E+004
 K = 2.545E+013
 C = 2.325E+006
 Damp. Ratio
 1.050E-003

% Cr. Damping
 0.10%

DAMPING VALUE

R.I.T. "B" ROTOR IMPELLER DISK

CIRCLE FIT
 Mode #1
 Freq. (Hz)
 = 3.930E+003
 Pt. #2
 Resp. Dir = -2
 Coord. of Ctr.:
 X = -6.742E-006
 Y = -1.469E-006
 Radius
 = 5.100E-006
 Ave. Err. (\$) = 0.00
 Fit range:
 3902.500
 3957.500

Modal
 Parameters:
 N. Freq. (Hz)
 3.943E+003
 M = 6.671E+003
 K = 4.094E+012
 C = 9.804E+004
 Damp. Ratio
 2.966E-004
 % Cr. Damping
 0.03%

DAMPING VALUE

FIGURE NO. 12
 COUPLED BLADE - SPLITTER-
 DISK MODE DAMPING
 EVALUATION (TYPICAL RESULT
 FOR LOW LEVEL IMPULSE
 EXCITATION)

FIGURE NO. 13 - FEM ANALYSIS
MODE SHAPE FOR "B" ROTOR
BLADE VIBRATION (MODE 1)

FIGURE NO.14 - FEM ANALYSIS
MODE SHAPE FOR "B" ROTOR
BLADE VIBRATION (MODE 2)

FIGURE NO. 15 - FEM ANALYSIS
MODE SHAPE FOR "B" ROTOR
BLADE VIBRATION (MODE 3)

FIGURE NO. 16 - FEM ANALYSIS
MODE SHAPE FOR "B" ROTOR
BLADE VIBRATION (MODE 4)

FIGURE NO. 17 - MAXIMUM VIBRATORY BENDING
STRESS LOCATIONS

NOTE: ALL GAGES ARE TO BE
ON THE BLADE SURFACE NEAR
THE FILLET. GAGE AXES
SHOULD BE PERPENDICULAR TO
THE LOCAL DISK SURFACE.

STRAIN GAGE	$\bar{z}^*$ θ	**HIGHLY STRESSED MODE
(A)	1.89" 24.33°	1, 3, 4
(B)	1.21" 2.70°	2
(C)	.03" 0°	3, 4

* SEE NREG DWG. NO. 1380DI0001
** JUDGED FROM MODE SHAPES
IN FIGURES NO. 8-II

APPENDIX

- Data Processing Techniques
- Schematic Diagrams of Set-Up
 1. Impulse Hammer with Proximity Probe
 2. Shake Table with Strain Gaging
- Equipment List

DATA PROCESSING TECHNIQUES:

The R.I.T. rotor impulse testing technique used to evaluate natural frequencies and mode shapes employs an instrumented hammer, a proximity probe, and strain gages. The test impeller is struck with a hammer thereby exciting it at all frequencies simultaneously. This occurs because, as can be demonstrated by Fourier analysis, a hammer impulse signal is a transient phenomenon comprised of a continuum of frequencies. The impeller responds strongly at its particular natural frequencies and the associated motions are detected by the proximity probe or, when the motion is very small, the strain gage.

By directing the output of both the force gage of the instrumented hammer and the motion transducer to the Zonic multichannel FFT analyzer it is possible to perform a Fourier decomposition of both signals. The ratio of response motion or strain amplitude to excitation force amplitude is then plotted as a function of frequency. The resulting spectrum then peaks at frequencies of maximum ratio of response per unit excitation, i.e. frequencies of maximum magnification factor - in other words, the natural frequencies of the test impeller.

Further employment of this technique allows the description of mode shapes. The response per unit excitation at each location of the structure can be measured and the resulting values assigned to corresponding geometric grid points in a computer terminal model of the test impeller. The computer terminal then allows animated display of the model in the deformed state of each mode for viewing and obtaining hard copies.

Damping is evaluated with software by examining the form of the spectrum peaks. The technique is based on the fact that the equation of one degree of freedom system, $M\ddot{x} + \dot{c}\dot{x} + kx = F(\sin \omega t)$, can be solved for the complex quantity called mobility, $\dot{x}/F$, i.e. velocity divided by exciting force, and the imaginary part plotted against the real part to obtain the graph of a perfect circle. The Zonic system software resolves the real and imaginary parts of the measured spectrum at discrete frequencies near the resonance peak and then fits this "data" to a circle equation using a least squares routine. The damping is then back calculated using the equations and the circle diameter. The circle fit also acts to check the reliability of the damping evaluation. Deviation of individual data points from a true circle provides a measure of error. One caution to be observed when applying the measured values is the dependence of material damping on stress levels. The low stresses generated during test cause damping to be underestimated.

DATA PROCESSING TECHNIQUES:

The same test method was used with shake table testing except for two differences. First, the shake table excites at only one frequency at a time so it is necessary to "sweep" the spectrum from low to high frequencies, covering the range of interest. Secondly, where the impulse setup uses a vibration pickup fixed in one location while the hammer excites the structure sequentially at each grid point, the shake table is permanently connected to and excites only one point on the structure and the vibration pickup is moved from grid point to grid point. The ability to test using a fixed (moving) excitation with a moving (fixed) vibration pickup derives from certain reciprocity laws of mechanics and is programmed into the software of the Zonic system.

IMPULSE TEST SET-UP

SHAKE TABLE TEST SET-UP

EQUIPMENT LIST

ITEM	BRAND	MODEL NO.	SERIAL NO.
FFT ANALYZER	ZONIC	6080-4-R-BM	230
TERMINAL	TEKTRONIX	4052	B023882
HARD COPIER	TEKTRONIX	4631	B183629
OSCILLOSCOPE	TEKTRONIX	5111A	B010219
SHAKE TABLE	LING DYNAMIC SYSTEMS, LTD.	405	116
COOLING FAN	AUTO WEST	556765R	N/A
POWER AMPLIFIER	TAPCO	CP-500	0200022
SIGNAL GENERATOR	HEWLETT-PACKARD	3325A	1748A05643
BRIDGE AMPLIFIER	HONEYWELL	218-2	4501MB79
POWER UNIT	PCB	480D06	2214
PROBE DRIVER	KAMAN	KD-2300-.5SUM	023212
POWER UNIT	KAMAN	P-3000	PS-1309
IMPULSE HAMMER	PCB	105M41	1393
PROXIMITY PROBE	KAMAN	KD-2300-.5SUM	24692
FORCE GAGE	PCB	208A02	2427
STRAIN GAGES	BLH	FAE-03S-12-S6ES	3-DE-UK
BRITTLE COATING	TENS-LAC	N/A	N/A