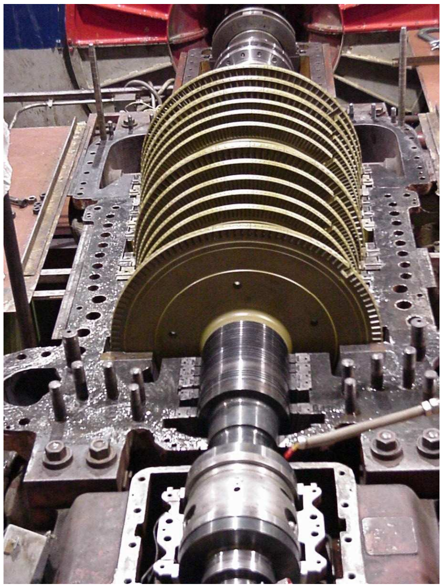
Engineering Technical Review

New Page Paper

Technical review of replacement steam path

for

Four stage Dresser-Rand Non-Condensing steam turbine



TurboCare Inc

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2. ENGINEERING STUDY

2.1. Overview

The subject unit is a four-stage, non-condensing turbine that drives an 1800 RPM generator through a reduction gear. The turbine was manufactured by Dresser-Rand, as a non-condensing unit with an uncontrolled extraction after the Stage #1 two-row Curtis wheel. The unit was originally rated at approx 8 MW @ an operating speed of 4521 RPM with steam conditions of 700 F/ 800 Psig /60 Psig exhaust. The unit was originally installed in 1985. Although the unit generally provided successful service during the initial 20 years of service, it has been reported to have experienced issues related to fluctuations in thrust loading and bearing wear. The unit was rebuilt by Dresser Rand in 2009 with the dual objectives of supplying exhaust steam to process, increasing turbine efficiency and maximizing electrical generation. The rebuild work scope included the design of a replacement rotor and steam path that would provide reliable long term service and accommodate the customer specified operating requirements for output and process steam. The previously noted issues with reverse thrust were apparently resolved with the turbine rerate, which included the supply of a new rotor, improved tilting pad thrust bearing and the elimination of the uncontrolled extraction. The uprated unit ran successfully until the stage #4 blade failure.

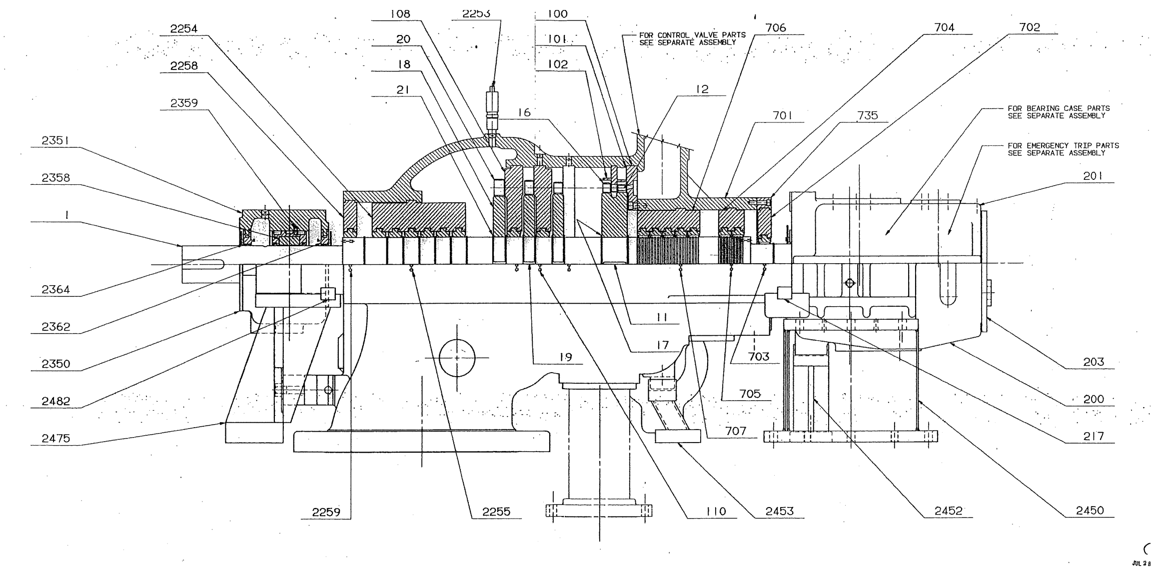


Figure #1

Representative view of the original turbine X-Section.

Background:

This existing steam path operated successfully for approximately 2 months following the D-R uprate prior to suffering a highly unusual and catastrophic failure of the L-0 wheel and blades. Subsequent examination of the rotor and blades implicated stress corrosion cracking as a potential contributing factor in the failure.

Photo #1

Failed 4th stage showing internal dovetail and wheel dovetail failure

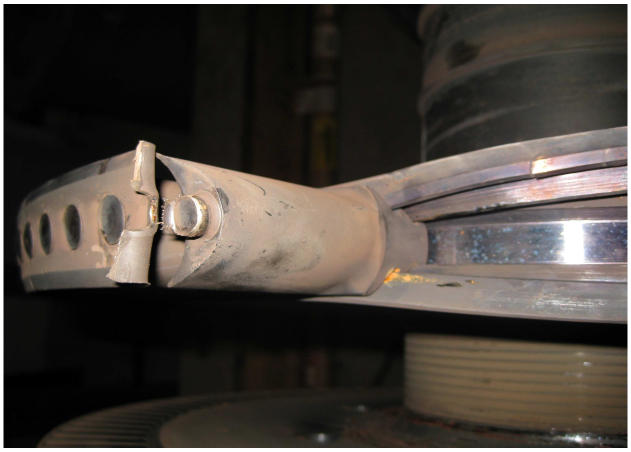


Photo #2

Photo of existing (failed) stage # 4 blade design with 2 hook internal dovetail and section of wheel

The L-0 blade, as applied by D-R on the uprated and redesigned rotor, used a conventional 2 hook slotted / double "T" root internal dovetail. Although this design has been successfully applied by many manufacturers, TurboCare prefers to use a more robust external dovetail configuration as shown in fig #2. It has been our experience that the external dovetail design represents a more robust and reliable configuration.

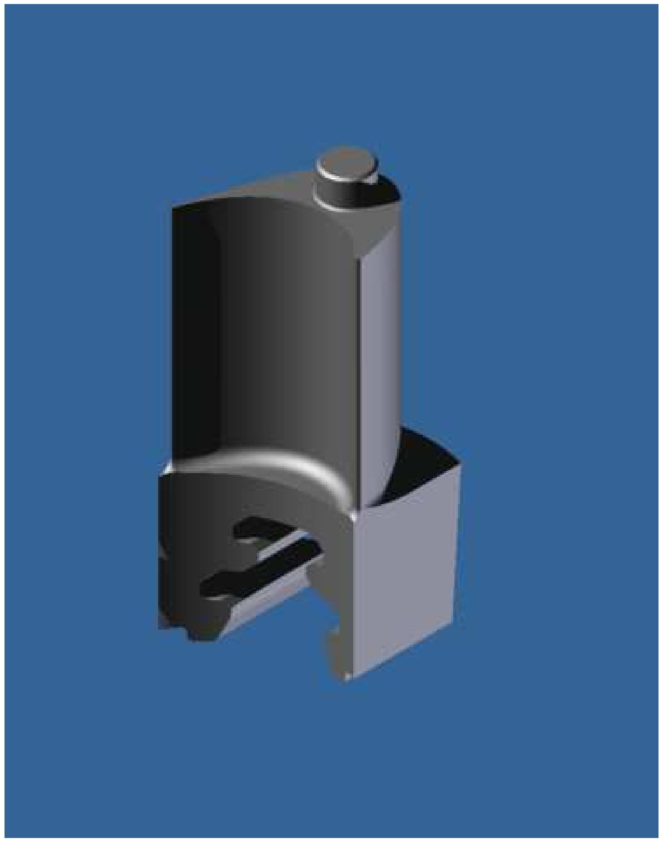


Fig # 2

Proposed TurboCare Stage # 4 blade design showing robust two hook external dovetail for maximum reliability and stress reduction

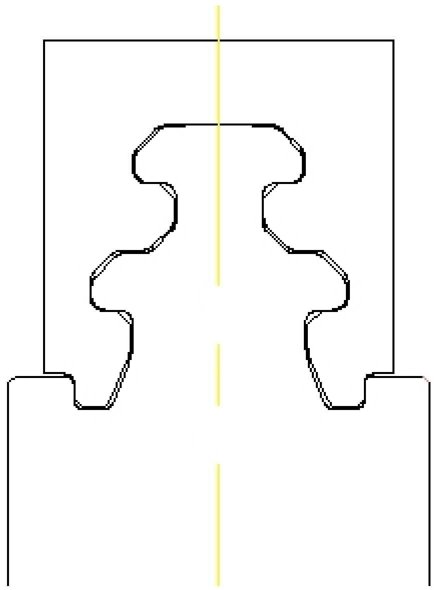


Figure # 3

Detail of blade and wheel dovetail w/ retention grooves in wheel rim.

Thermodynamic Design:

The design requirements as specified by New Page Paper included the design of a replacement rotor and steam path that would provide long term service and maintain a maximum level of exhaust superheat over the full range of operating conditions. Maintaining adequate superheat in the exhaust steam was required to support mill process requirements. The steam path will use modern aerodynamic nozzle and rotating blade designs to maximize aerodynamic efficiency and reliability. All of the stages will use a low stress multi-hook, external dovetail configuration for minimum stress and maximum reliability. The blades will be matched with modern aerodynamic nozzle profiles for maximum efficiency and energy recovery.

The thermodynamic performance of the proposed redesigned steam path over the full range of customer specified operating conditions is outlined in table # 1 below:

Table #1
Thermodynamic Performance

Load Point	INLET PRESS (PSIG)	INLET TEMP (℉)	EXHAUST PRESS (PSIG)	THROTTLE FLOW (LB/HR)	GENERATOR OUTPUT (KW)	EXHAUST SATURATION TEMP (℉)	EXHAUST TEMP (℉)	EXHAUST SUPERHEAT (℉)	OVERALL T-G EFFICIENCY (%)
1	820	725	62.5	140,000	5808	310	347	38	63.7
2	820	725	62.5	162,000	6905	310	340	31	65.4
3	820	725	62.5	175,000	7548	310	337	27	66.2
4	820	725	62.5	185,000	8031	310	335	26	66.7
5	820	725	62.5	200,000	8699	310	335	25	66.8
6	820	725	65	140,000	5738	312	351	40	63.6
7	820	725	65	162,000	6831	312	344	32	65.5
8	820	725	65	175,000	7468	312	341	29	66.2
9	820	725	65	185,000	7952	312	339	27	66.7
10	820	725	65	200,000	8647	312	338	26	67.1
11	800	700	58	200,000	8570	305	311	6	66.5
DR-1	800	700	65	150,000	6047	312	330	18	64.6
DR-2	800	700	65	162,000	6624	312	326	14	65.5
DR-3	800	700	65	175,000	7246	312	323	11	66.3
DR-4	800	700	65	185,000	7714	312	321	10	66.8
DR-5	820	720	65	185,000	7911	312	335	23	66.7
DR-6	820	720	65	162,000	6794	312	340	28	65.4

Note that operating point #11 in the above table is representative of the operating conditions which would be expected to yield the minimum superheat in the exhaust.

BLADE ANALYSIS

Steady State stresses:

Steady state and alternating stresses were evaluated for all critical high stress including the vanes, dovetails, bands and tenons. The steam path was specifically designed to minimize stress at all critical locations, in the wheel and blade, as well as avoid blade resonances wherever possible. When it was not possible to avoid blade resonances, the stage was thoroughly evaluated to ensure that the resulting stress levels were well below industry accepted standards for reliable long term service. All nozzle and rotating blade airfoils will be constructed of corrosion and erosion resistant stainless steel.

Calculated steady state stresses resulting from the centrifugal loading in the blade and wheel for each of the evaluated blade locations is outlined table # 2 below. Stresses were calculated using 12 chrome stainless steel shroud bands for stages #1 and #2 and titanium material for stages # 3 and #4. Titanium is an excellent shroud material in that it combines the strength of steel with an approximate 50% reduction in weight and exhibits excellent erosion and corrosion resistance. The use of titanium on results in a significant reduction in the stage #3 and #4 band stresses. The expanded table below includes a summary of blade stresses at specific locations in the wheel and blades and more precisely defines the location and magnitude of the maximum stresses. Note that some values have changed due to revisions in both material, assumed operating stage temperature and the configuration of the shroud band in selected high stress locations to additionally reduce peak stresses. Note that the blade tenon tension, which represents the tensile load on the shroud rivet head, and the one pitch bending, which is representative of the tendency of the shroud band to lift between rivet heads, are both extremely low, i.e. less than 15% of the permissible limits. The listed stage temperatures represent the calculated operating temperatures for the rotating blades.

	Stg #1	**Stg #2**	**Stg #3**	**Stg # 4**
Rotating Blade operating temperature @ max rated load.	630 F	515 F	435 F	350 F
Stress Location				
Rotating Blade Vane	7 %	8 %	12 %	13 %
Blade dovetail tension	11 %	14 %	15 %	12 %
Wheel dovetail tension	32 %	26 %	25 %	29. %
Blade dovetail hook shear	18 %	23 %	23 %	18 %
Wheel dovetail hook shear	32 %	26 %	25 %	30 %
Blade Tenon tension	11 %	15 %	14 %	6 %
Shroud corner bending w/ cutback	52 %	46 %	34 %	34 %
One pitch Shroud bending	11 %	9 %	6 %	7 %

Table #2

Steady State Stresses in the wheel and blade dovetails

Component	Material
Stage 1 Blading	AISI 422 SS blading and shroud band
Stage 2 Blading	AISI 403 SS blading and 422 SS shroud band
Stage 3 Blading	AISI 403 SS blading and Titanium band
Stage 4 Blading	M152 with titanium shroud band
Nozzle / Diaphragms	C Stl Inner/Outer Ring; 410 SS Vanes
Rotor Forging	ASTM A470 Class 4
Packing	Bronze

Table #3

Component Material Chart

Alternating / Vibratory Stresses:

In addition to steady stresses, alternating stress levels were also investigated. Alternating stresses are the result of the stimulus forces imposed on the rotating blade by the upstream nozzles. These impulses can become especially problematic when they cause the blade row to vibrate on one of its' resonant natural frequencies. The primary method of avoiding this issue is to vary the stage geometry where possible to avoid running on resonance. In cases where this is unavoidable, the resulting stresses are calculated to ensure that the response at resonance does not result in an overstress conditions. The results of this review are represented in the following Table #4 and on the accompanying Campbell diagrams included in appendix "A".

Table # 4 shows the results of the vibratory stress analysis and provides a summary of maximum calculated vibratory stresses and corresponding vibration mode for each of the four redesigned stages. Stresses are presented as a percentage of TurboCare and maximum industry accepted design allowable based on the selected material properties and the stage operating conditions. Maximum stresses are presented based on their identified mode shape and the magnitude of the stress and its location on the stage. (i.e. its value at the blade dovetail, vane and tenon)

Stage	Dovetail Stress % Allowable / (Mode)	Vibratory Stresses Vane Stress % Allowable / (Mode)	Tenon Stress % Allowable /(Mode)
1	5 % / (2x 1An)	4 % (2x 1An)	15 % (2x 1An)
2	6 % / (2x 1An)	5 % (2x 1An)	14% (2x 1An)
3	13 % / (2x 1An)	14% (2x 1An)	11% (2x 1An)
4	24 % / (1x 1An)	28 % (2x 1Tn)	19 % (2x 1Rn)

Table # 4

Maximum vibratory stresses by stage and location

Comments on blade design practices:

Each rotating blade has a number of discrete natural frequencies that are determined by a variety of factors, including dovetail type, vane geometry, blade dovetail fixation, turbine operating speed and cove/shroud connection. Nozzle passing stimulus is the stimulus that results from the interaction between the stimulus that results from the number of nozzles in the adjacent diaphragm and the natural frequency of the rotating blade. This nozzle passing stimulus is a primary concern in the design of short height blades like those applied on the New Page unit, where the blades pass in front of the stationary nozzles in the diaphragm and are impacted by the relatively high frequency individual nozzle stimulus.

Nozzle passing resonances are typically depicted on a Campbell diagram and identified as individual families and are classified by the mode shape of the resulting grouped natural frequency for the blade group. Typical mode shapes include:

- The fundamental tangential or 1To mode
- The second tangential range or 1Tn group
- The torsional modes or 1Rn group
- The axial modes or 1An group

The frequency ranges that correspond to the above groups are shown as shaped boxes on the right margin of the diagram. The stimulating frequency is represented as a diagonal line representing the number of nozzles and the turbine speed. When there is a coincidence between the diagonal stimulus line and the indicated family of blade resonances, that mode is considered to be in a resonant condition. This indicates that resonance, excitation and elevated stresses may exist at the identified resonant frequency. When resonance is indicated, stresses representing the resulting stresses are calculated and presented in the stress summary. The steady stresses represented on the Goodman diagrams are representative of the stresses in the blade/wheel dovetail, which typically represents the highest stress location.

Comments on the design of the individual stages:

Stage #1

The current inlet control stage uses a two row Curtis configuration, with reamed high pressure inlet nozzles and a fabricated reversing ring that feeds the “B” row. This arrangement is typically used in high pressure applications, where part load performance is considered a key design parameter, and was not viewed by TurboCare as the optimum design for the currently specified operating conditions.

The redesign will replace the existing Curtis control stage with a single row Rateau design that includes the use of a modern aerodynamic nozzle profile and fabricated nozzle plate construction for improved performance and reduced nozzle stimulus. The stage will maintain use of the current partial arc inlet configuration, where throttle steam is admitted through the existing inlet control valves via four individual nozzle arcs in the upper half and two in the lower half of the high pressure casing.

As shown on the stage #1 Campbell diagram in Appendix "A", the selected nozzle pitching results in a coincidence between the 1An blade natural frequency and two times the nozzle passing frequency (2X) at the operating speed of 5707 RPM. The resulting stresses for the identified resonance mode are extremely low at 15% of the maximum permissible design limit. The stage also uses 422 Stainless steel blade material for elevated strength at the elevated inlet temperature.

The combined steady state and vibratory stresses are graphically represented on the accompanying Goodman/Soderberg diaphragm in Appendix "B", along with the permissible design limits and appropriate properties. The Goodman diagrams plot the combined alternating and steady stresses in comparison with the yield and tensile strength of the blade material. The Goodman diagrams for each stage in the redesigned steam path are presented in Appendix B. The overall stress profile for each stage is significantly below the maximum design limits.

Stage #2

Due to their physical similarity, stage #2 exhibits a comparable resonance profile to stage #1, with a second order (2X) nozzle passing resonance with the 1An mode. The highest stress occurs in the blade tenon and is approximately 14% of permissible design limits.

Stage #3

As blade frequency decreases with increasing blade height, stage #3 requires a reduction in nozzle count to avoid problematic high stress resonance modes. The selected nozzle results in the highest stress occurring in the vane of the blade at a very nominal 14% of maximum design limits. This stage also uses titanium shroud bands to reduce the steady stress present in the tenon / blade interface.

Stage #4

As a result of concerns about the previous failure, and the inability to avoid low stress blade resonances, the stage #4 blade has been fully evaluated for potential steady state and vibratory stress issues and will use a proven, redesigned two hook external TurboCare dovetail configuration for maximum reliability. All of the operating stresses have been evaluated and found to be well within proven design limits for reliable long term service. Although the stage has been designed to operate in a thoroughly dry operating environment due to its previous failure history and experience with operating in a corrosive steam environment, the stage will also use high strength corrosion resistant 12 Chrome Jetheate/M-152 blade material to provide additional margin against corrosive attack. This material is widely used on exhaust stages of large steam turbine generators due to its excellent corrosion resistance and fatigue strength in a wet corrosive environment. Although not required for the currently specified operating conditions, this enhanced performance material is being applied in this case to provide an additional level of conservatism in the design of the stage and to provide the maximum possible reliability margin. Both the 3rd and 4th stages will also use titanium shroud bands for increased resistance to corrosive attack and reduce centrifugal loading in the blade/tenon area.

The optimum nozzle pitching for stage #4 results in a 1X 1An resonance and both a 2X 1Tn and 2 X 1Rn resonances with the blade natural frequency at the 5707 RPM operating speed. The maximum stress occurs in the blade vane and is at 28% of the maximum permissible design limits.

Comments on the TurboCare blade stress calculation system

The maximum allowable design limits referenced in this report are based on both the material properties and TurboCare's proven and calibrated blade stress analysis system. The maximum permissible design limits have been established according to industry accepted design practices for turbine blade and wheel applications. These limits are based on the material tensile strength at maximum operating temperature and include design factors for geometry and service induced degradation due to operation in a steam environment. The system has been proven to yield reliable blade designs suitable for a planned 30 year operating life when applied at up to 100 % of the specified allowable limits.

Conclusion/Summary:

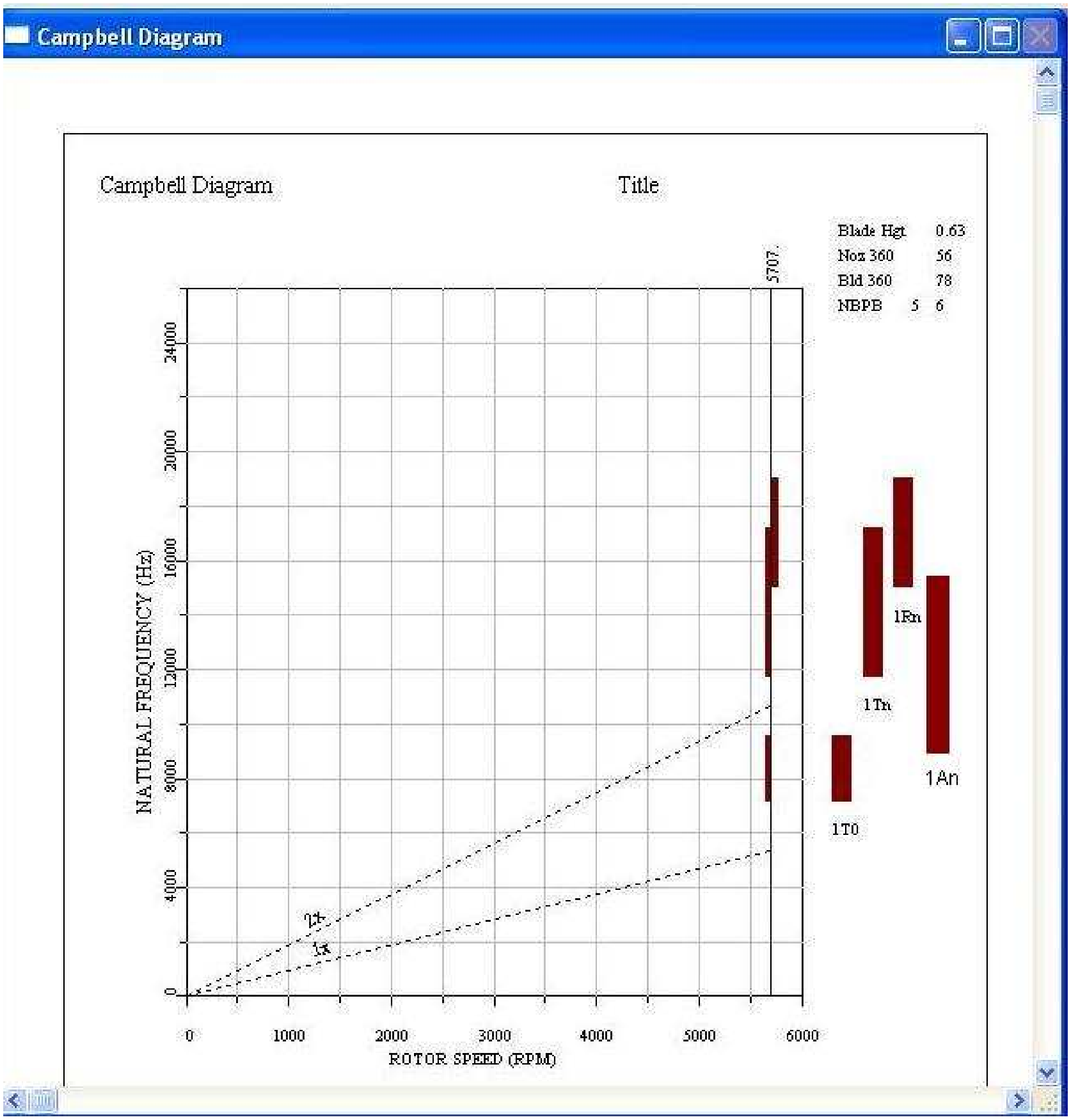
The results of the above analysis combined with TurboCare's experience with the design of both variable speed mechanical drive and constant speed turbine/ generator drives, indicates that the proposed turbine rerate represents a highly reliable design that will satisfy both the specified thermodynamic and mechanical requirements.

The replacement steam path will take advantage of modern design practices and proven analysis methods to minimize both steady state and vibratory stresses. As a result all stresses will be below the maximum listed value of 28% of industry accepted limits. These very low stress values will ensure that the unit will be able to achieve reliable operation over the specified range of operating limits. The steam path has also been designed to achieve the specified 8.6 MW at the maximum rating and to accommodate the process steam requirement that the exhaust steam maintains a minimum of +15 F of superheat for process use over the specified load range.

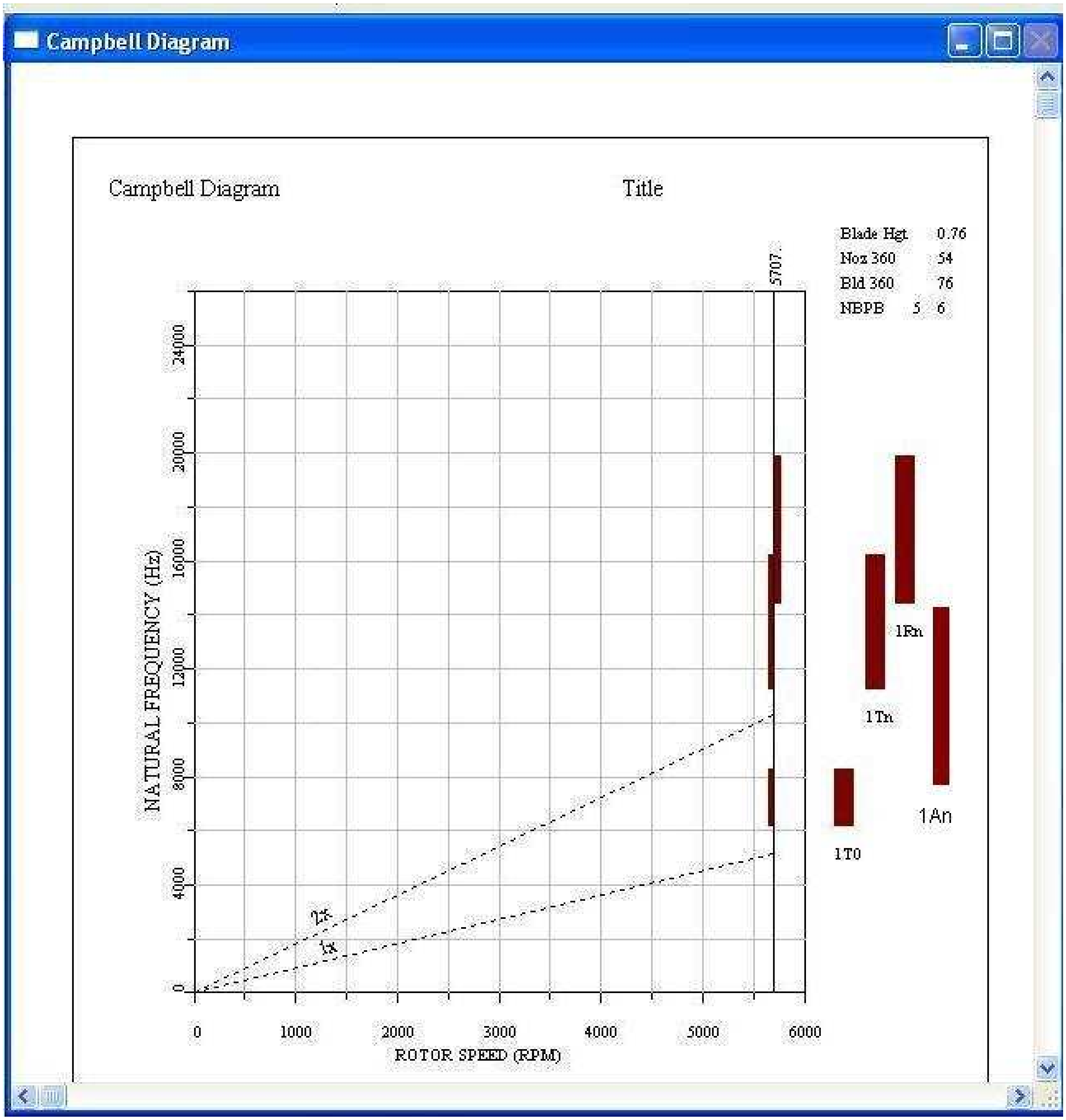
The replacement steam path will include the supply of a new fully bladed one piece forged rotor, replacement inlet nozzle plate, a set of three replacement diaphragms and a full set of interstage and shaft end seals. The new steam path will be designed to be a drop in replacement suitable for installation into the existing casing. All components will be custom machined to ensure proper alignment and clearances during unit reassembly. The design will use the existing casing and inlet valve gear. It is also TurboCare's intention to reapply the existing thrust and journal bearings, provided that they are proven upon inspection to be in serviceable operating condition.

Appendix A: Campbell Diagrams

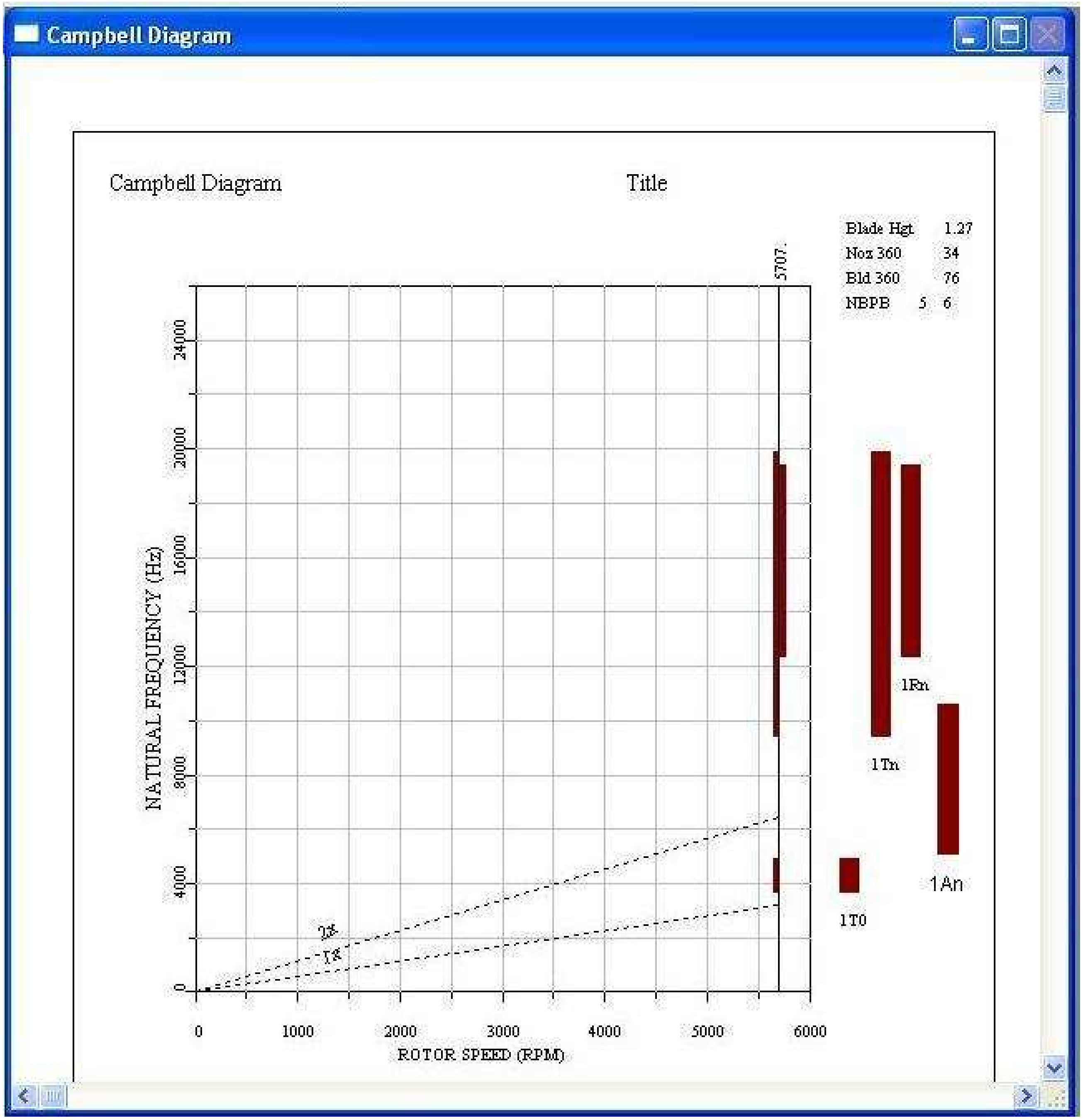
Stage 1



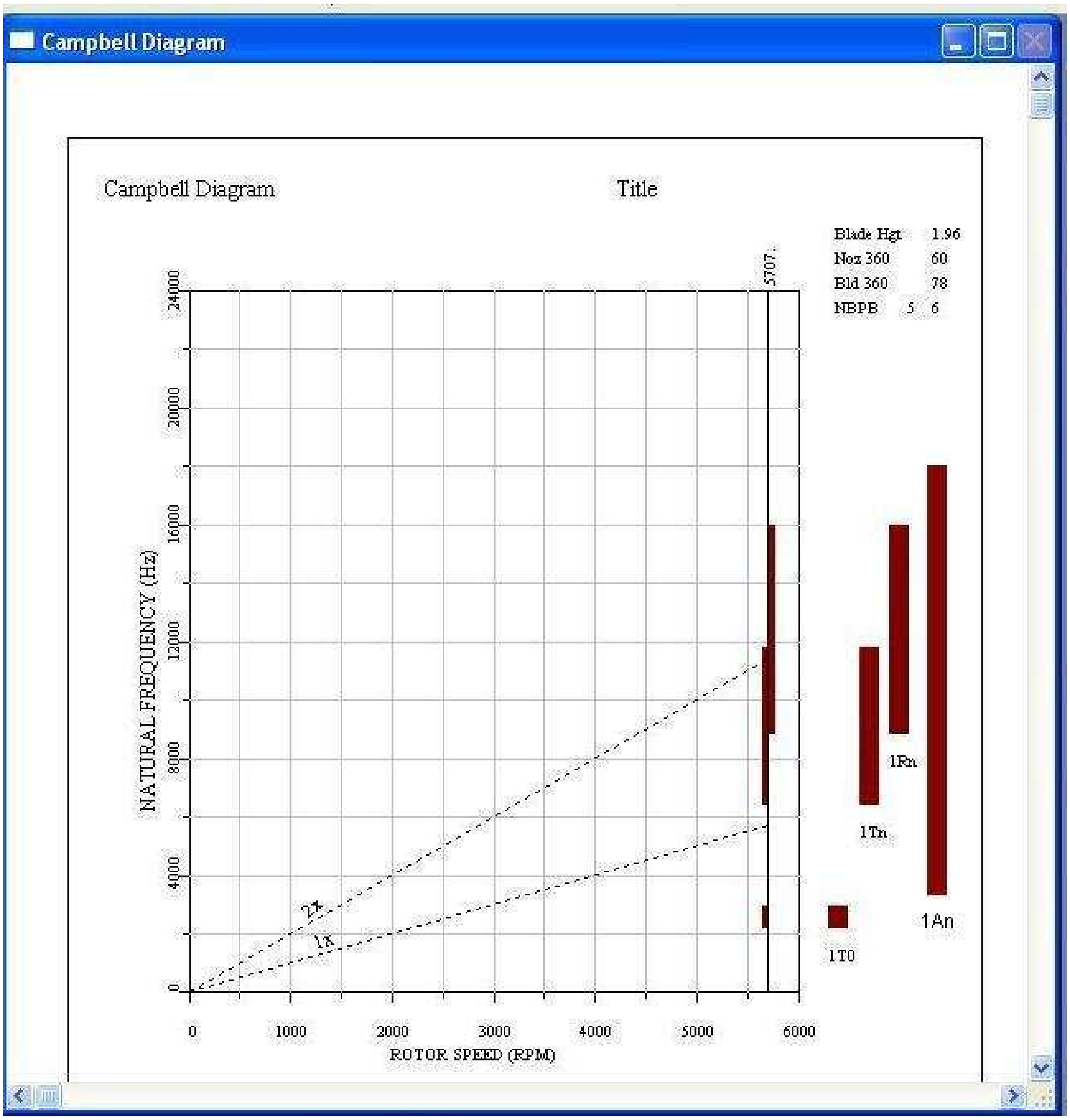
Stage 2


Campbell Diagram
Campbell Diagram
Title
Blade Hgt 0.76
Noz 360 54
Bld 360 76
NBPB 5 6
5707
NATURAL FREQUENCY (Hz)
0
4000
8000
12000
16000
20000
24000
0
1000
2000
3000
4000
5000
6000
ROTOR SPEED (RPM)
2x
1x
1Fn
1Tn
1An
1T0

Stage 3

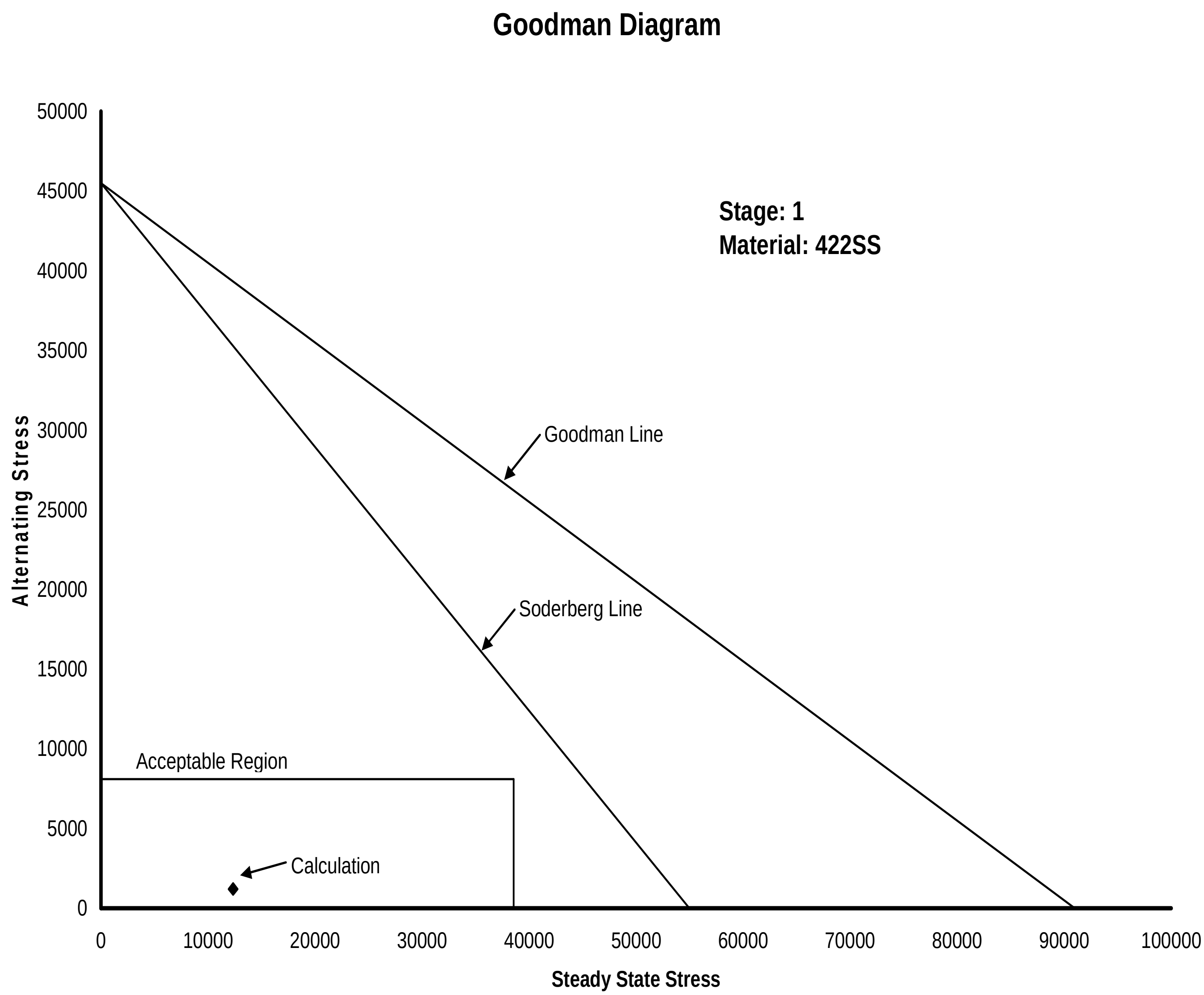

Campbell Diagram
Campbell Diagram
Title
Blade Hgt 1.27
Noz 360 34
Bld 360 76
NBPB 5 6
5707.
NATURAL FREQUENCY (Hz)
0
4000
8000
12000
16000
20000
24000
0
1000
2000
3000
4000
5000
6000
ROTOR SPEED (RPM)
2X
1X
1Fn
1Tn
1T0
1An

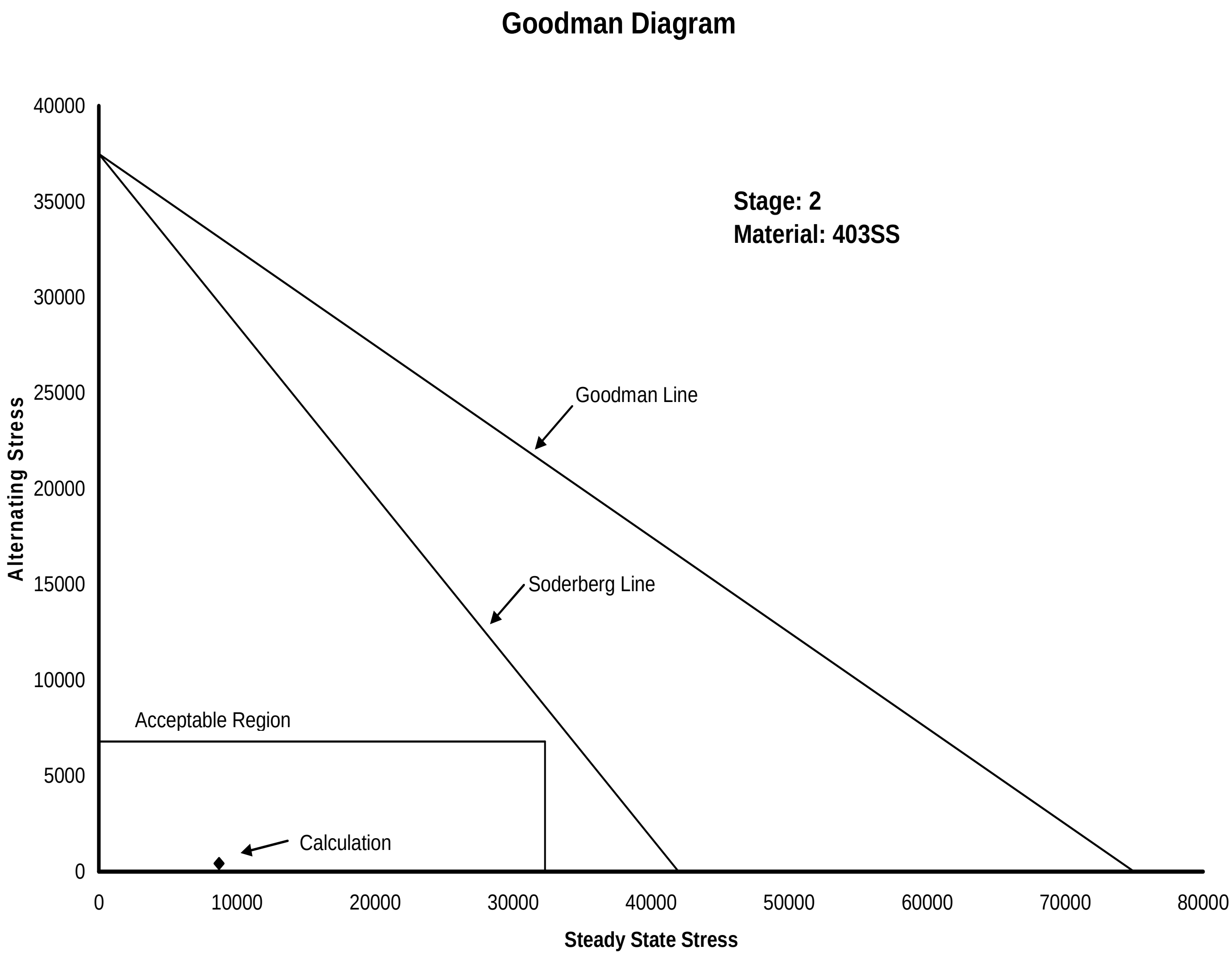
Stage 4


Campbell Diagram
Campbell Diagram
Title
Blade Hgt 1.96
Noz 360 60
Bld 360 78
NBPB 5 6
5707.
NATURAL FREQUENCY (Hz)
24000
20000
16000
12000
8000
4000
0
2x
1x
1T0
1Tn
1Fn
1An
0
1000
2000
3000
4000
5000
6000
ROTOR SPEED (RPM)

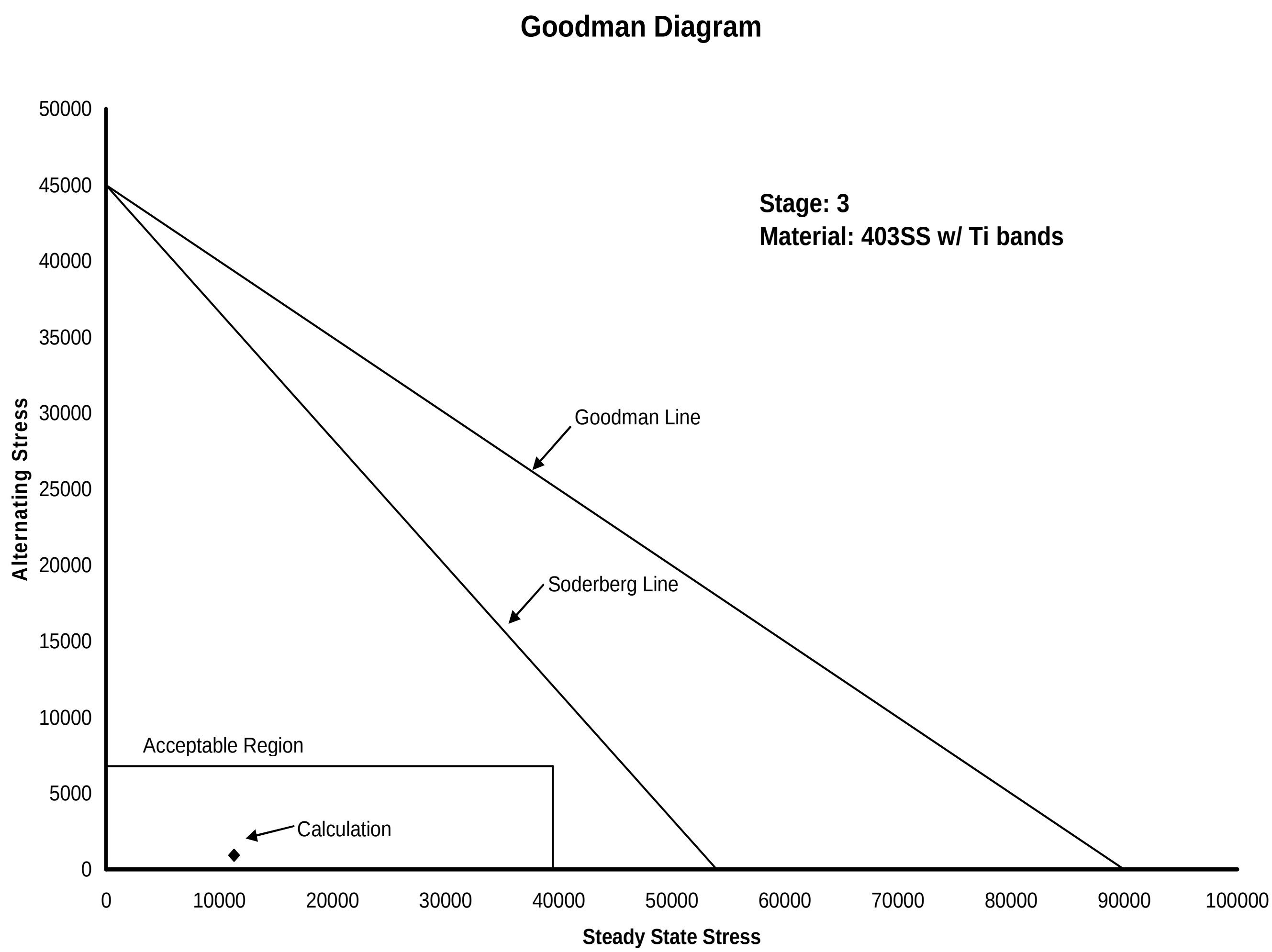
Appendix B: Goodman Diagrams

Stage 1



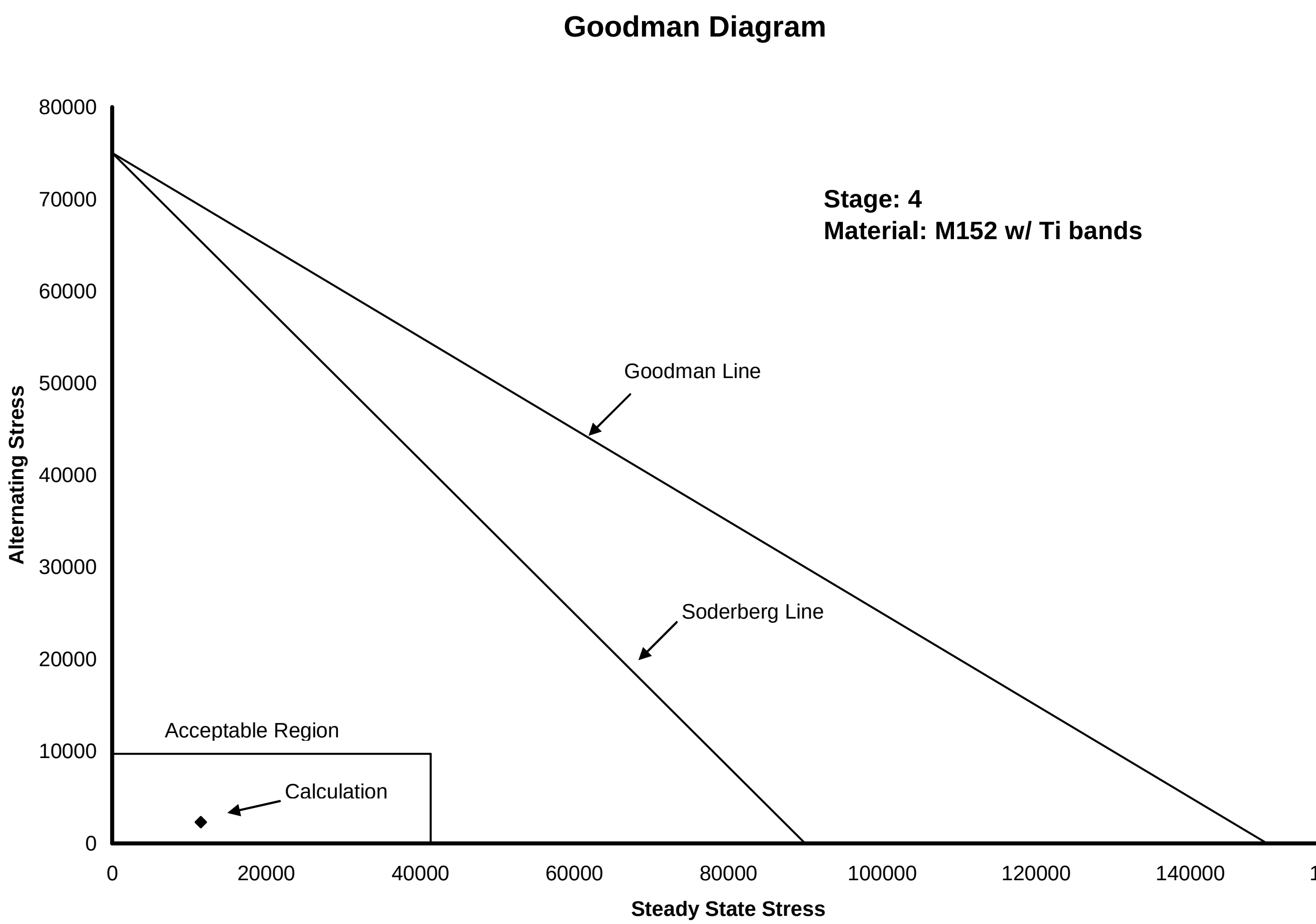

Goodman Diagram
Stage: 2
Material: 403SS
Alternating Stress
40000
35000
30000
25000
20000
15000
10000
5000
0
Goodman Line
Soderberg Line
Acceptable Region
Calculation
0
10000
20000
30000
40000
50000
60000
70000
80000
Steady State Stress

Stage 2


Goodman Diagram
Stage: 3
Material: 403SS w/ Ti bands
Alternating Stress
50000
45000
40000
35000
30000
25000
20000
15000
10000
5000
0
Goodman Line
Soderberg Line
Acceptable Region
Calculation
0
10000
20000
30000
40000
50000
60000
70000
80000
90000
100000
Steady State Stress

Stage 3

Stage 4


Goodman Diagram
Stage: 4
Material: M152 w/ Ti bands
Goodman Line
Soderberg Line
Acceptable Region
Calculation
Alternating Stress
80000
70000
60000
50000
40000
30000
20000
10000
0
0
20000
40000
60000
80000
100000
120000
140000
Steady State Stress