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# STRETCHING THE SIZE OF GEOTHERMAL STEAM TURBINES

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**ABSTRACT**

Developers have an incentive towards increased size to gain benefits from economy of scale - for example, US$850/kW for 110 MW units compared with US$1,000/kW for the same station size with 55 MW units. Currently, the largest geothermal turbines in single cylinder, double-flow configuration are 110 MW at 50 Hz and 77.5 MW at 60 Hz. These are of equivalent size, after accounting for the different operating speeds.

Increased size presents challenges for both the designer and the developer's engineer to ensure that both the developer and the operator maximise the return on their investment over the life of the plant. These challenges arise largely as a result of increased wetness through the steam path, together with increased stresses and corrosion fatigue potential from longer last row blades. This paper discusses the problems exacerbated by large sized units and how these can be alleviated by selection of appropriate materials and design configuration. Optimisation of the main design parameters for the power plant and steam-field, in conjunction with the supplier, also has a major influence. This involves selection of the steam inlet pressure, design cooling tower wet-bulb temperature, cooling tower approach temperature and condenser pressure. Optimisation of the steam-field, leading to selection of the steam inlet pressure, needs to consider also the likelihood of resource run-down of pressure and flow and the policy with make-up wells.

Some recent examples of large size units are reviewed. At Darajat in Indonesia, the rated size is 81.3 MW, with capability to 101.7 MW. Wayang Windu in Indonesia is rated at 110 MW. Recent inspection of the three 77.5 MW turbines at Malitbog in The Philippines revealed these to be in excellent condition after operation since 1997, with minimal scaling, no significant erosion and no blading defects.

It is concluded that the technology exists to further increase size, even before resorting to the more expensive titanium last row blades. However, the limit may be imposed instead by transportation restrictions of large/heavy items, or transmission limitations.

## 1. INTRODUCTION

Developers have an incentive towards increased size of generating units to gain benefits from economy of scale. This can be readily appreciated by comparing typical 50 Hz power station costs as follows:

| | |
|---|---|
| 4 x 55 MW Units | US$1,000/kW |
| 2 x 110 MW Units | US$825/kW |

A small reduction can be obtained with two of the 55 MW turbines in tandem with a 110 MW generator.

Recently, there have been significant increases in size of geothermal steam turbines. Currently, the largest single cylinder, double-flow turbines are shown in Table 1.

The smaller 77.5 MW 3,600 rpm turbine is the equivalent of the 110 MW 3,000 rpm turbine because, for the same steam conditions, stresses, steam velocities and blade velocity triangles, the following scaling factors apply for a 3,000 rpm turbine in relation to a 3,600 rpm turbine:

| | |
|---|---|
| Steam flow | $n^2$ |
| Output | $n^2$ |
| Blade size | n |
| Mean blade diameter | n |

where $n = 3{,}600/3{,}000 = 1.2$

As a result, countries with power systems at 50 Hz have a significant advantage over those at 60 Hz, in achieving economy of scale for geothermal turbines with two-pole generators.

For geothermal turbines, increased size presents special challenges for the designer, to preserve operating life and reliability and to keep maintenance costs within bounds. Increased size is mainly obtained through:

- increased steam inlet pressure
- decreased exhaust pressure
- increased steam flow.

There is a temptation to set these parameters in favour of increased size, without considering the problems they introduce. Causes, effects and mitigation are discussed under two broad headings:

- steam wetness in the steam path
- last row blade length

Two further aspects to further mitigate the problems and to ensure the developer gains most benefit long-term are then considered:

- optimisation of the cooling system/condenser pressure
- considering steam-field run-down.

The paper ends with examples and a look into the future.

## 2. STEAM WETNESS IN THE STEAM PATH

Increase of steam wetness and its effects in the steam path are a major impediment to increase in size of geothermal generating units.

### 2.1 Causes and Effects of Steam Wetness

Causes/Severity of Erosion from Steam Wetness
(Ansari, 1986), (Moore and Sieverding, 1976)
Erosion from steam wetness arises from water drops entrained in the steam flow impinging onto moving blade surfaces. The drops come from liquid films and rivulets on the fixed blade. The stripping action of the steam on the surface water from the trailing edge of the blade leads to drops entrained in the steam flow. The impingement of the drops on the downstream moving blade gives rise to high, localised stresses, eventually leading to fatigue failure and cracking. This causes gradual breakdown of the blade material. Its severity is related to:

- steam wetness , typically 12% at the exhaust
- drop size, typically 20 – 200 µm (<50µm are harmless)

- blade speed typically 390 – 470 m/s in geothermal turbines (compared with up to 600 m/s in large conventional turbines with similar wetness levels)
- angle of impact
- pressure upstream of the stage in question.

Most problems occur in the last rows of the moving blades towards the periphery, where the steam wetness and blade tip speed are highest.

Effect of Steam Inlet Pressure on Steam Path Wetness

Size increase by means of increased steam inlet pressure leads to higher wetness in the steam path through the turbine, increasing potential for erosion. Figure 1 illustrates this by comparing two turbine expansion lines on the Mollier diagram, ignoring any reduction caused by inter-stage drainage. With both turbines A and B at the same exhaust pressure, turbine A, which has a higher steam inlet pressure, exhausts at 17% wetness compared with 16% for turbine B.

Effect of Exhaust Pressure on Steam Wetness

Size increase by means of decreased exhaust pressure leads to higher wetness in the steam path through the turbine, increasing potential for erosion. This is illustrated in Figure 2, which indicates that a drop of exhaust pressure from 0.12 bara to 0.08 bara is accompanied by an increase of wetness from 16% to 17%, ignoring the effect of inter-stage drainage.

Effect of Increase of Steam Flow

Output increases in proportion to steam flow. This results in a higher mass flow of droplets or higher blade tip velocities if size is increased, resulting in greater potential for erosion.

Increased Losses from Steam Wetness

Various methods have been proposed to allow for wetness losses, but the simplest, which can be applied for low pressure situations applicable for geothermal turbines, is a loss of 1 per cent in stage efficiency per 1 percent of mean stage wetness (Craig and Cox, 1971).

### 2.2 Mitigation of Steam Wetness Effects

Reduction of Steam Wetness/Water Drops by Water Catching/Inter-stage Drainage

The water drops tend to be more concentrated around the periphery of the steam path through the turbine, due to the centrifugal action of rotation. This can be used to reduce steam wetness, with one or more of the following:

- Inter-stage drainage, aided by water catching lips around the casing
- Grooves on the stationary blades to direct water into the inter-stage drains
- Use of slots to direct water into hollow stationary blades and then to drains.

These methods ensure that irrespective of the steam inlet pressure or exhaust pressure, the wetness at the exhaust can be reduced to about the same level as normally applies for condensing turbines: about 12 to 13%. However, increased steam inlet and/or decreased exhaust pressure results in creation of greater quantities of water, which has the potential to form into destructive drops before removal in the inter-stage drains. Also, this water, disposed of in inter-stage drains, represents a loss of efficiency.

Reduction of Erosion Effects

The effects of erosion can be reduced by:

- Minimising last row blade tip speed
- Use of large axial space between stationary and moving blades, to encourage break-up of large drops in the flow
- Thin stationary blade trailing edge to reduce drop size stripped off by the steam flow
- Use of erosion resistant material at the outer leading edge of the last one or two rows of moving blades
- Use of moving blade geometry to ensure that the water drops impinge on the erosion resistant material, rather than beyond it (noting that the drops have a greater inertia compared with steam and do not readily change direction in the upstream fixed blade passages).

Rate of Erosion

The rate of erosion decreases considerably with time (Moore and Sieverding, 1976). This has been borne out by experience at Wairakei Power Station in New Zealand. This is thought to be the result of:

- Cushioning effect of water trapped in the eroded area
- Greater surface area as a result of erosion
- Reduced impact angle of water droplets due to the peaks of the eroded areas.

## 3. LAST ROW BLADE LENGTH

Size increase of geothermal generating units necessitates increase of exhaust annulus area and consequently last row blade length. The endurance limit of blade steels is significantly less in geothermal steam containing $H_2S$ than in air, so the last row blade length constitutes one of the main impediments to size increase of geothermal generating units.

Typical last row blade lengths in relation to annulus areas for large geothermal turbines are as in Figure 3.

### 3.1 Effect of Exhaust Pressure on Annulus Area/Blade Length at Exhaust End

Exhaust pressure has a marked effect on annulus area/last row blade length due to the effect on volume flow. This can be illustrated by comparing turbines with the same exhaust steam mass flow and velocity, in Figure 4.

### 3.2 Effects of Last Row Blade Length Increase

Last Row Blade End-Loading

Last row blade end loading is defined as steam flow per unit area of the last row annulus. An increase of last row blade end-loading means increase of:

- Water flow per unit area (affecting blade erosion – Section 2 above)
- Steam bending force on blades and vibration exciting forces (affecting stress and corrosion fatigue – Section 3.3 below)
- Exit loss due to higher velocity (affecting steam rate).

Typical last row blade end-loadings are as follows:

| | |
|---|---|
| 50 Hz turbines: | 50 to 75 t/h $m^2$ |
| 60 Hz turbines: | 45 to 60 t/h $m^2$ |

With large geothermal turbines the loading is more likely to be at the high end of the range, very close to design limits.

Materials/Stress/Corrosion Fatigue

Blading is subject to vibration stress superimposed on steady combined centrifugal and bending stress. This situation can be examined at the point of maximum stress, usually near the blade root. The maximum combined stress must be well inside the line linking the endurance limit and yield strength on the typical Goodman diagram for 13% Cr blade steel in Figure 5.

Use of 17-4 PH steel, which has a higher corrosion fatigue strength, allows greater latitude than 13% Cr steel for increased blade length, if needed to achieve the required output.

Referring to the relationships in Section 1, the 697 mm last row blade for Wayang Windu in Table 1 approximates to a scaled-up version of the 565 mm blade used widely by the manufacturer in 3,600 rpm geothermal turbines:

| | | |
|---|---|---|
| Ratio of lengths | 697/565 | = 1.23 |
| Ratio of speeds | 3,600/3,000 | = 1.2 |

The last row blade at Malitbog is 658 mm, again used on other 3,600 rpm geothermal turbines. When scaled up by 1.2 for 3,000 rpm, this is equivalent to a larger last row blade than for Wayang Windu.

Titanium alloy has a density about 60% that of steel, so a blade nearly 40% longer can be used, without increasing stress in the blade and rotor. A further advantage of titanium alloy is its superior fatigue, corrosion and erosion resistance. Experience dates back to the 1950s in Russia. The increased cost of titanium alloy blades (up to four times that of steel) is partially offset by the increased efficiency due to reduced leaving loss from the longer blade (Nedeljkovic *et al*., 1991).

Vibration Characteristics

The natural vibration frequency of the blading in its various modes should not coincide with a harmonic corresponding to either the operating speed range, or the blade passing frequency, to avoid resonance. The operating range should take into account under or over frequency operation (usually between 95% to 103%, depending on the transmission system). These vibration characteristics of all the blades can be examined on a Campbell diagram. For large size turbines, the main interest is in the last row blades. A typical Campbell diagram for these at the operating speed range for a 3,600 rpm turbine is in Figure 6.

It is difficult to calculate blade natural frequencies, particularly if groups of blades are tied together. For example, some manufacturers have an integral shroud, which locks on to the corresponding shroud of the next blade when the twisted blade untwists slightly under centrifugal load. A damping snubber may also be incorporated. If the design is new, it is important to ensure that the manufacturer has checked the calculated natural frequencies in a model test rig. Otherwise, there is a need to ensure that there is adequate operating experience of identical blading at the same frequency (or appropriately scaled for a different frequency).

## 4. COOLING SYSTEM/CONDENSER PRESSURE OPTIMISATION

Programmes have been developed to assist in identifying the optimum condenser pressure and cooling tower approach temperature for a given design wet-bulb temperature. The programme carries out an initial design of the turbine, condenser, main and auxiliary cooling water systems, gas extraction system and cooling tower to evaluate performance and capital/operating costs for estimation of net present value.

The optimisation needs to be carried out in conjunction with the turbine plant designer, to ensure compatibility with the equipment he can offer and that there is agreement about the results. For example, the manufacturer can assist with the optimisation of the exhaust end in relation to leaving losses and blade length (Yokota and Saito, 1997).

However, it is necessary to consider other factors. For example, for Wayang Windu the optimisation resulted in a comparatively high exhaust pressure in the region of 0.12 bara. Apart from purely financial considerations, this also allowed the last row blades not to be pressed too much, which allayed concerns about long blades and steam wetness. Another benefit is that this has allowed the size of the main cooling system to be kept down, including the hot-well pumps and main cooling water pipes, with attendant deep excavations close to major foundations. The condenser, under the turbine, was also kept down in size, reducing foundation size/cost.

The selection of the design wet-bulb temperature has a major influence on the design and performance. It is essential to obtain reliable weather records applicable to the power station site to enable a plot of wet bulb temperature against percentage occurrence.

The design wet-bulb temperature should also be selected by economic optimisation, although it is sometimes arbitrarily set at the temperature that is exceeded only 5% of the time, after additions of margins for cooling tower recirculation and possible global warming. However, optimisation generally results in selection of a design wet-bulb temperature closer to the annual average ambient wet bulb temperature, after addition of the above margins.

Selection of the “annual average” basis means that the output will float above and below nominal output throughout the year. However, the turbine design usually includes an inlet pressure margin to compensate for first stage nozzle scaling, so this margin can be used to avoid some of the load reduction on hot days.

Selection of “5% exceedence” means that for much of the year the condenser will be operating at a lower-than-design pressure, with the turbine capable of increased overload operation – unless adjustments are made at the cooling tower (for example, by variable speed operation or by turning off fans/cells). Although this means that the nominal output will always be obtained, capital cost is higher. There is also more potential for problems, such as erosion, due to off-design operation, if adjustments are not made at the cooling tower.

## 5. STEAMFIELD OPTIMISATION/RUN-DOWN

Mills (1997) discussed the selection of the steam pressure at the power plant boundary, along with steam turbine-generator output. This needs to be considered in conjunction with:

- optimisation of steam-field plant
- likelihood of resource run-down of pressure and flow
- policy with drilling make-up wells.

For a high-pressure wet resource, the possibility of double flash needs to be considered, as this allows higher turbine output for a given steam flow. This needs to be balanced against the extra capital cost and the possibility of silica deposition at the low re-injection temperature from the second- flash stage (Yokota and Saito, 1997).

For a developer, the financial life of the project may have an over-riding influence. For example, if the financial life is only ten years, there is an encouragement to maximise output and extract the maximum from the resource, with less regard for its longer-term performance.

## 6. SOME EXAMPLES OF LARGE GEOTHERMAL STEAM TURBINES

Examples of large geothermal steam turbines are given in Table 1. The turbines at Darajat (Saito *et al*., 1998) and Wayang Windu are in the same district at similar altitude, yet the design condenser pressures are radically different, at 60.6 mbara and 120 mbara respectively. This has necessitated a longer last row blade at Darajat, at 762 mm compared with 697 mm at Wayang Windu.

The three turbines at Malitbog have had the top covers removed after a year of full load operation in addition to 6 – 12 months part-load operation. As illustrated in Figure 7, the steam path was in excellent condition. A small amount of scaling was present, but was easily removed. There was no significant erosion. Non-destructive testing of blading did not reveal any defects.

## 7. THE FUTURE

The question arises: will the size of single cylinder double-flow geothermal steam turbines continue to rise? The incentive of economy of scale will push for this. The technology exists to further increase size even before resorting to the more expensive titanium last row blades, given sufficient inlet pressure.

A geothermal turbine cylinder is analagous to the low pressure cylinder for a conventional steam turbine in a fossil fuel fired station. In 60 Hz areas size increase was accomplished by use of cross-compound units with the LP turbine coupled to a generator at 1,800 rpm. Use of the scaling relationships outlined in Section 1 means a substantial increase in output/size is then possible. However, a four-pole generator is at least 20 - 30% more expensive than one with two-poles, which may nullify the advantage of scale.

The limit may be imposed instead by transportation restrictions of large/heavy items, geothermal plant often being in mountainous, remote regions. There may also be transmission system limitations.

## 8. SUMMARY

The significant increase in size of the single cylinder double-flow geothermal steam turbine has resulted from developers seeking lower capital costs from economy of scale. Manufacturers have readily taken up the challenge and demonstrated that such size increase is practicable.

However, size increase means that the equipment items are operating much closer to their design limits. The role of the owner's engineer in the project is, therefore, vital in plant selection and review. This should include selection of the design parameters in conjunction with the supplier. The engineer has optimisation tools to achieve this, for the power plant and steam-field, considering also field run-down. The engineer also needs to critically examine the design proposed by the supplier, particularly methods to deal with steam wetness effects and materials/stress/corrosion fatigue and vibration at the last row blades. The experience and testing facilities of the supplier are an essential part of the review.

This will then ensure that the developer and the operator maximise the return on their investment over the life of the plant.

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**Table 1 – Current Largest Single Cylinder Geothermal Steam Turbines**

| Location | Frequency/ Speed (Hz/rpm) | Rated Output (MW) | Maximum Output (MW) | Inlet Steam Pressure (bara) | Condenser Pressure (mbara) | Last Row Blade Length (mm/inches) |
|---|---|---|---|---|---|---|
| Malitbog, The Philippines | 60/3,600 | 77.5 | 85.25 | 10.3 | 118 | 658/25.9 |
| Wayang Windu, Indonesia | 50/3,000 | 110 | 115.5 | 10.6 | 120 | 697/27.4 |
| Darajat, Indonesia | 50/3,000 | 81.3 | 101.7 | 13.3 | 60.6 | 762/30 |

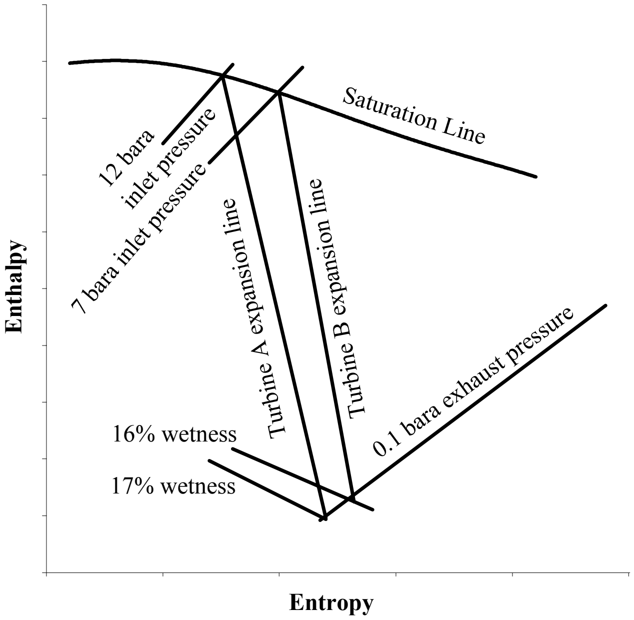


**Figure 1 - Effect of Increased Steam Inlet Pressure on Wetness**

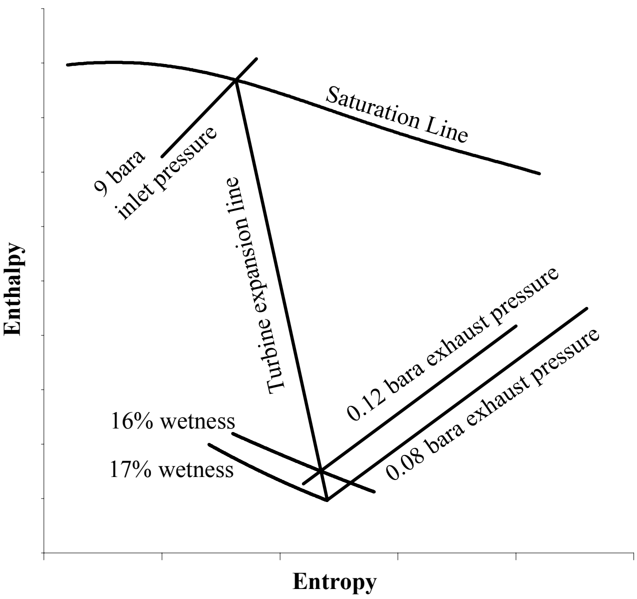


**Figure 2 – Effect of Decreased Exhaust Pressure on Wetness**

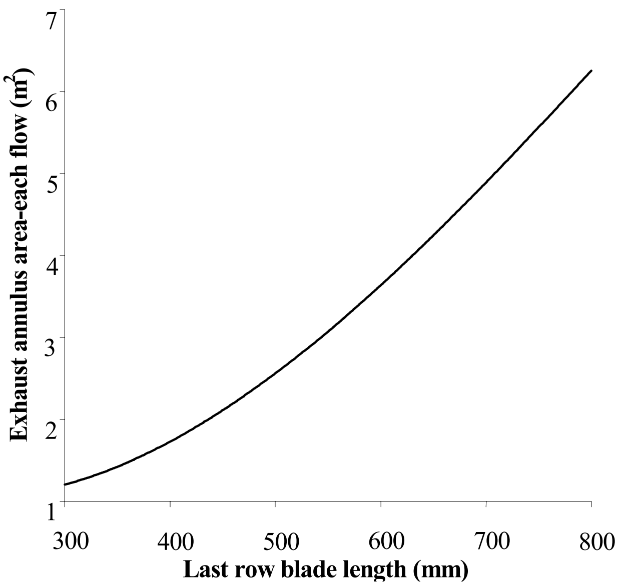


**Figure 3 – Typical Exhaust Annulus Areas in Relation to Last Row Blade Lengths**

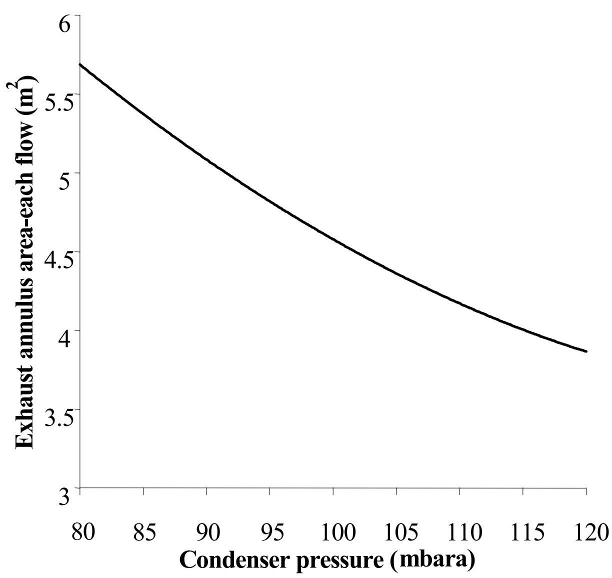


**Figure 4 – Effect of Exhaust Pressure on Annulus Area**

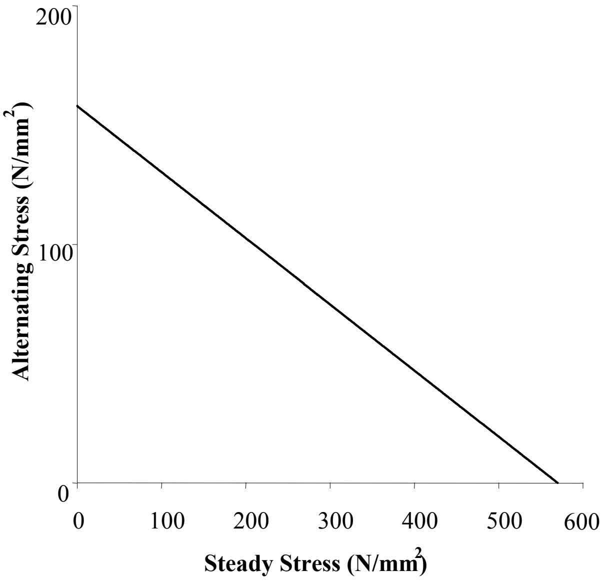


**Figure 5 – Typical Goodman Diagram for 13% Cr Last Row Blade**

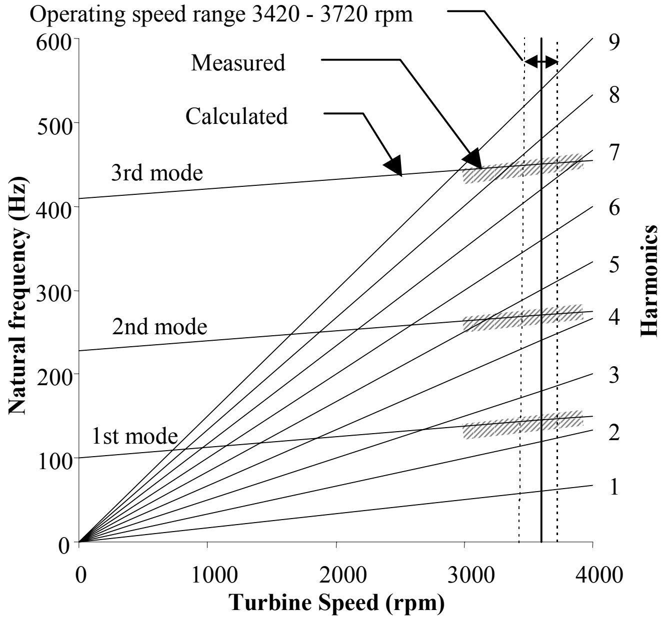


**Figure 6 – Campbell Diagram for Typical Last Row Blade**

**Figure 7 – Malitbog 77.5MW 3,600 rpm Turbine Blading after 18 Months Operation**