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Instability due to steam whirl in large rotating machinery

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Abstract: A 400MW steam turbo generator of a power plant showed an unstable vibrational behaviour, which was attributed to steam whirl (or steam flow excited instability) during the start up of the plant.
The combined effect of steam excitation in bladed rows, in the steam glands and of low damping in some of the oil film bearings is most likely the main cause of the observed malfunction. The phenomenon has then been overcome by changing the alignment of the bearings.
This experimental case is first discussed and then the instability causes are analyzed in the paper. Standard stability criteria based on the calculation of the real and imaginary parts of the eigenvalues of the turbo generator showed a consistent stability margin, and the machines of the same design, in similar plants, did not show any instability problem. To get deeper insight in the mechanism of steam whirl, the presented research has been developed to define suitable models for the steam whirl excitation and a new method for the stability evaluation, which is described in the paper, has been implemented. The method is derived from previous studies on oil whirl instability and is based on the energy balance when the rotor, rotating at its normal operating speed, is excited by a small exciting force at the frequency corresponding to its first critical speed. The energy introduced during one cycle by the steam forces is compared to the energy dissipated by the oil film bearings.

Keywords: Steam whirl, instability, steam glands, rotor dynamics.

INTRODUCTION

Turbo-machinery like steam turbines and gas compressors may experience unstable vibrations due to an interaction between the whirling shaft and the surrounding fluid forces in correspondence of seals where the clearance between rotating and stationary parts are small. These unstable vibrations arise in steam turbines most likely close to the maximum power condition, when pressures and consequent flows are maximum. In gas compressors instability may occur also in off design conditions. The vibration amplitudes reach very quickly high unacceptable values, the machines are shut down due to excessive vibration levels and can then be operated only at reduced power levels. The unstable vibrations occur at the shaft first bending natural frequency, and can be controlled only by the damping introduced by the oil film bearings, or by some additional external damping. When damping is insufficient, instability can be overcome only by modifying the seal geometry. Since the occurrence of this type of instability has limited the development of performances of turbo-machinery, many studies have been devoted in the last 45 years, depending on the development needs of high performance turbo-machinery, either to the analysis of steam whirl in steam turbines, or to instabilities excited by air flow in the axial compressors of gas turbines, or to instabilities excited by the fluid in labyrinth seals of high performance centrifugal gas compressors or finally instability excited by the liquid in the seals of high performance centrifugal pumps.

Despite the fact that there is a consolidated knowledge about these problems at least in the scientific community, there are still some weak points in modelling the steam whirl excitation, so that predictions of stability margins sometimes can fail.

The motivation of the present study is exactly a case history of a steam turbine that experienced a heavy steam whirl instability though the stability margin in design conditions, calculated by the turbine manufacturer according to a consolidated methodology, was sufficiently high. A twin turbine operated in the same conditions had no stability problems at all.

First a historical overview on studies about instability excited by fluid flow over seals in turbo-machinery is presented. Then the recent steam whirl instability event on the turbine is described, the possible causes of inaccuracies in standard modelling and stability margin evaluation are analysed, and finally some proposal for increasing the accuracy in instability predictions is presented.

HISTORICAL NOTES

First approaches

Unstable behaviour occurring in HP steam turbines was first analysed by Thomas (1958) in Germany. He realized that the excitation was mainly due to the leakage steam flow over blade rows. In the same years in Russia Lomakin

(1958) discovered that fluid forces in seals of hydraulic machines affect critical speeds and stability. The self-excited rotor whirl in turbo-machinery has then been studied by Alford (1965) in the USA 7 years later. He derived formulas for the excitation through unequal clearances over blade rows which are similar to the formulation by Thomas, and considered also the leakage flow through a single vane labyrinth seal, neglecting the circumferential flow. He found instability when the outlet clearance is smaller than the inlet clearance. Since then these fluid forces generating instability have been named often in English literature "Alford forces".

Krämer (1968) used the model of Thomas for the blade row steam leakage forces, which excite forward whirl, and introduced the model of oil film bearings showing the combined effect on stability. He defined a linearized cross coupling stiffness coefficient K_{xy} (force in x direction due to shaft displacement in y direction) proportional to the equivalent tangential driving force acting on the stage divided by the length l of the blade:

$$K_{xy} = k \frac{N}{\Omega D_m l} \tag{1}$$

where N is the power of the stage, D_m the mean diameter of the blade row, Ω the rotating speed, and k a parameter which has to be calculated modelling the leakage flow over the blades (which depends obviously on the clearance). Reliable values of k are not given, but can be calculated with thermodynamic models of flow through seals.

In the same year in the USA Ehrich (1968) analyses instability excited by labyrinth seals.

Kostyuk (1972) calculated the pressure distribution in the seal taking account of the rotational shaft speed and of the circumferential flow, but his results are not suitable to calculate the excitation constant.

In the 70's gas-dynamic calculations by German researchers (Spurk and Keiper, 1974), in which the effect of circumferential flow was also considered, showed that Alford's results regarding converging and diverging seals were uncorrect; that instability may occur when the outlet clearance is larger than inlet clearance (diverging seal). The leakage flow excitation was analysed more deeply, comparing theory to experiments (Ulrichs, 1976, Thomas et al., 1976). Values of k can be derived from diagrams in Thomas et al. (1976). These first experimental results, obtained with an offset rotating shaft, and not with a whirling shaft, showed that the forces are not linear with the shaft displacement, and are much higher for shrouded blades than for single standing blades, revealing the influence of the circumferential flow, which is also sometimes called the "gas bearing effect". This effect, which is present and well known in all hydraulic machinery, is obviously the only one present in the labyrinth seals where no blades are present (such as the balancing drum in HP steam turbines); it is mainly due to the rotation of the shaft and could be easily separated from leakage over blades effect which is apparently independent from rotation.

In the paper of Greathead and Bastow (1976) results are reported for a heavy steam whirl in an industrial HP steam turbine of a large turbo-generator, resulting in a limitation in load to 80%, which was overcome by changing design of oil film bearings and increasing the clearances in the seals. In Wright (1978) experiments for determining labyrinth forces with an air test rig and a single cavity seal are presented: the results, which were probably the first results obtained with a whirling shaft (instead of an offset shaft), confirm that diverging seals are unstable strongly exciting backward whirl when disk is rotating, converging seals are stable and straight seals excite only weakly backward whirl. It is further confirmed that whirl excitation on rotating disk is much stronger with respect to non rotating disk, and that the prediction of excitation constants is still not reliable. The excitation constant depends on pressure drop, on backpressure value, and in real machines on many other parameters such as gas density, friction, surface speed, pre-swirl velocity.

Pollman et al. (1978) presented an overview of different excitation mechanisms, and compared calculated and measured excitation factors for 3 different types: single 50% reaction stage, 3 stage 50% reaction and impulse stage. The agreement is not very good, measured values are much higher than calculated values for the impulse stage, which is attributed to the gas bearing effect, are generally lower for the single 50% reaction stage and can be higher or lower for the 3 stage model, depending on pressure levels. Applications to real turbo-groups are also shown, and prediction of stable or unstable behaviour is claimed to be possible at design stage.

Further developments

In 1980 the steam whirl phenomena stimulated the production of 4 papers from 4 different countries, all presented in the same IMechE Conference. Further experimental results in air are given in Greathead and Slowcombe (1980) for multi-cell shaft labyrinth glands, showing the development of consistent unbalanced forces, which contribute to steam whirl instability. In Murphy and Vance (1980) a theoretical investigations about converging and diverging seals, which ignored German literature, showed that neglecting circumferential flow a diverging seal introduces damping and converging seals can generate instability, which is exactly opposite to the results of Spurk and Keiper (1974) and confirmed Alford's findings. The authors realized that opposite results have been found by Wright (1978), but attributed these results to low frequency whirl; in high frequency whirl these results should not be valid, due to presence of inertia forces.

In Kurohashi et al. (1980) appears the first Japanese approach to the problem written in English, the approach of Kostyuk (1972) is used and simplified and corrections are introduced to take into account the effects of flow contraction and of carry over flow. Stiffness and damping coefficients are calculated by means of an approximated approach, the agreement with experimental results is only qualitative and not quantitative, theory and experiments confirm that diverging seals cause instability (backward whirl), the opposite occurs with converging seals. Benckert and Wachter (1980) report accurate measurements made in different labyrinth seal geometries and attributes the exciting lateral forces to swirl entry flow (for short seals) and to circumferential flow and related drag of rotating shaft (for long seals). These effects are shown separately. Linear behaviour is observed up to eccentricities of 40-60% of clearance, which is partially in contrast with the findings of Thomas et al. (1976). Results are applied successfully to steam turbines and compressors.

Calculations of stability margins in the design stage of steam turbines have been introduced in the design computer codes of steam turbine manufacturers (ABB, 1994): these are based on the evaluation of the real part of the eigenfrequencies of the rotor system. For this calculation the evaluation of the excitation constant for each reaction blade row (with corrections for shrouded rows) and for the different types of labyrinth seals is based on the German research stream results (Thomas,1958, Krämer, 1968, Urlichs, 1976, Thomas et al., 1976, Benckert and Wachter, 1980). Seals and bearings, represented respectively by these excitation constants (also called cross coupling stiffness) and by the stiffness and damping coefficients of the oil film bearings, constitute the connections between stationary casing and the rotating shaft, where they can cause unstable whirling motion, or stable behaviour.

In Hauck (1982) and (1982) the author resumes main excitations related to unequal circumferential pressure distribution in the seals of a deflected shaft, and to entry swirl. He presents accurate pressure and velocity measurements in the cavities between seal strips showing the presence of different flows: one mainly peripheral and another mainly axial. He shows also results of typical shrouded turbine stages with labyrinth seals.

In the following years (between 1980 and 1994) many researches were presented on the excitation due to leakage in seals and related instability mainly of compressors or pumps; these were published in the proceedings of a workshop about “Rotordynamic Instability Problems in High Performance Turbomachinery” which took place every 2 years in between 1980 and 1990 published as NASA Conference Publications. Main interest was on pump seals, but also compressor instabilities have been reported and the stabilizing effect of honeycomb seals and swirl brakes has been shown both theoretically and experimentally on real machines. Only few papers on steam whirl were presented. Main contributions were from Japanese, from Americans and from Germans.

Ehrich (1988) in a well known Handbook explains briefly how forward whirl could be excited both in steam turbines and in compressors due to unsymmetrical pressure distribution in the blade row of a deflected shaft.

But backward whirl in compressors is more likely to occur as shown in Storace et al. (2001) and Ehrich et al. (2001). Here the 8 authors present experimental and calculated results of whirl inducing forces in axial flow compressors of gas turbines, measuring also the whirl exciting coefficient. Experiments are made on a rotating (non-whirling) eccentric bladed shaft, with one inter-stage tooth seal on the shaft. This coefficient depends strongly on the operating point of the machine. Also forward and backward whirl excitation is discussed: whilst in steam turbines only forward whirl is excited, in compressors both forward or backward whirl can be excited, depending on operating condition, but backward whirl excitation constant is much higher in low flow conditions. Accurate pressure distribution measurements have been made on stationary blades in different angular positions, allowing to evaluate the Alford forces, and the excitation constant. The effect of a whirling motion of the shaft is analysed theoretically (by means of simplified models) and some effects are described: a phase shift of maximum pressure with respect to minimum clearance in the seal is shown as well as a phase shift of maximum circumferential blade force with respect to minimum clearance are both attributed to inertia effects. The effect of these phase shifts is obviously to change the excitation constant which have generally been evaluated with static measurements. Childs (1993) describes exhaustingly all results obtained and models used in liquid and gas annular seals and turbines and pump impellers.

Recent studies involving also CFD

In recent years theoretical investigations have been made calculating with rather simple expressions excitation forces due to unequal pressure distributions in the control stage of a steam turbine caused by different nozzle admissions (Chai et al., 2001). The time domain calculation of the dynamical behaviour of a Jeffcott rotor excited by the flow in the clearance over a blade row, with a rather simple flow model in Kim et al. (2002), (2003) gives no further insight in the excitation mechanism and seems to be more an application exercise of time integration methods.

Useful results have been obtained by Kwanka with his test-rigs for measuring direct stiffness, damping and excitation constant in whirling rotors Kwanka et al. (1995). He c)ompares also measured values with calculated values, recently also by means of CFD (Schettel et al., 2005). Test results are also compared to calculated results obtained with two different CFD codes in Schettel and Nordmann (2004), giving insight in capabilities of CFD codes to represent experimental results.

The instability due to fluid flow that can occur in compressors and pumps as well as in steam turbines is now a well known phenomenon. Also standards as API 617 (2002) have recognized these potential unsafe behaviours and have given simplified formulas for checking stability based on expression Thomas (1968) where the k coefficient is assumed to be 3.14 for an unexplained reason.

STEAM WHIRL CASE HISTORY

The machine-train of a 425 MW power unit was composed of a single-flow high pressure turbine (HP), a single-flow intermediate pressure turbine (IP), a double-flow low pressure turbine (LP) and a generator. The shafts of the machine-train were directly coupled each other by means of rigid couplings. The operating speed of this reheat unit of reaction design was 3000 rpm. The steam admission to the HP section was full arc. The shaft-train was mounted on six main oil-film journal bearings having nearly the same geometrical properties. They were two-lobe journal bearings which showed two large pockets located at the horizontal partition of the bearing casing. The pre-load of the lower lobe was 0.3. Figure 1 shows the layout of the machine-train and the bearing numbers. Each support was equipped with a pair of XY proximity probes and one vertical seismic transducer.

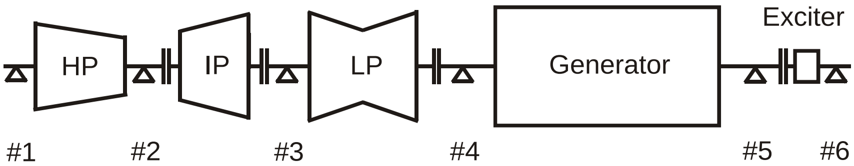

Figure 1: Machine-train layout and bearing numbers.

The HP turbine was mounted on two journal bearings whose lower lobe showed an angular width of only 60 degrees. The experimental vibrations measured during a run down transient at bearing #1 (Figure 2) showed that the first critical speed of the HP turbine was split into two values: 1850 rpm and 1970 rpm.

Few months after the first rollout of the power unit the HP turbine showed multiple events of high vibration levels that occurred in operating condition and caused several machine trips. These high vibrations always occurred when the electrical load exceeded the 80% of the nominal rating, that is when the opening degree of the main control valves of the HP section reached its upper range.

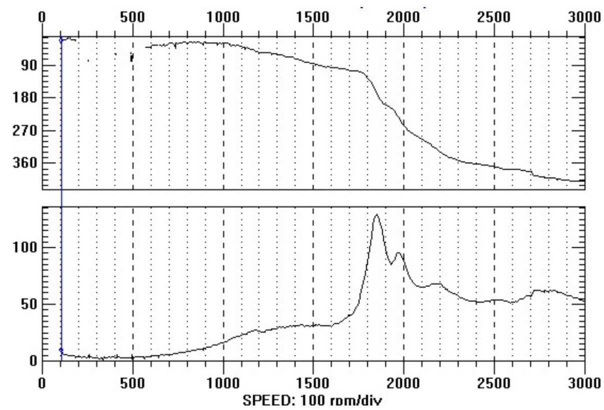

Figure 2: Synchronous transient vibrations (1X) measured at bearing #1 of the HP turbine: passing through the first critical speed.

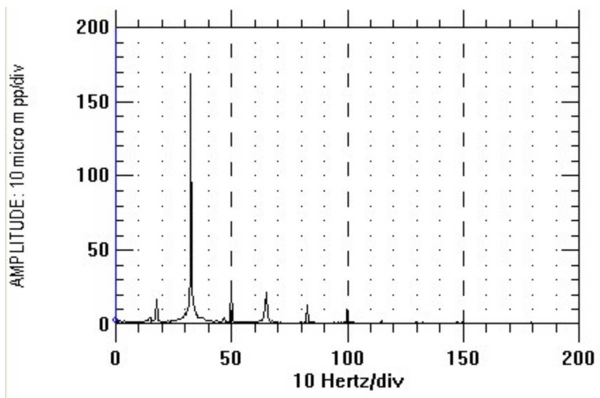

Figure 3: Spectrum of the shaft vibrations measured at bearing #1 in operating condition before the onset of the instability.

The vibrations measured at bearings from #3 to #6 were nearly unaffected by the abnormal vibrations that occurred at the bearings #1 and #2 of the HP turbine. The analysis of the harmonic content of the vibration signals showed that, in general, in occasion of each machine trip the amplitude of the synchronous component (1X) measured at 3000 rpm was not excessively high. On the contrary, few minutes before each machine trip a sub-harmonic component at 32 Hz suddenly appeared in the vibration signals. This value was rather close to the first flexural critical speed of the HP turbine. The amplitude of this component grew very quickly reaching considerable levels in less than 10 seconds.

Figure 3 shows the frequency spectrum of the shaft vibrations measured at bearing #1 few seconds before the occurrence of a machine trip caused by the sharp growth of the amplitude of the sub-harmonic vibrations of the HP turbine. Figure 4 shows a waterfall diagram of the shaft vibrations measured at bearing #1 in occasion of one of the instability events. It is possible to note that the sub-harmonic vibrations immediately disappeared after the machine trip. The sudden and sharp increase of amplitude at constant speed when maximum power is approached, and still more the sudden and sharp disappearance of the subharmonic component when the power is removed are clear symptoms of steam whirl: other causes of instability (such as oil whirl) could be discarded.

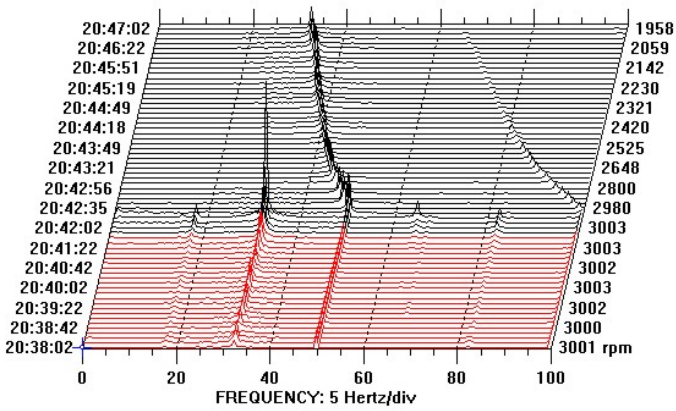

Figure 4: Waterfall plot of the shaft vibrations measured at bearing #1 in occasion of a steam-whirl instability onset and during the subsequent coast-down caused by the machine trip.

The sensitivity of the machine-train to the occurrence of steam-whirl instability phenomena caused by the increase of the seal excitation constant due to the raise of the load has been investigated. This study has been carried out by multiplying all the nominal values of the seal stiffness coefficients of the model by a scaling factor, cs. Different analyses have been performed by varying this scaling factor from 0 to 3. The null value of cs can be associated with the off-load condition while the unity value of cs corresponds to the full-load operating condition. For each value assigned to the constant cs the eigenvalues of the machine model have been evaluated. The real part of each eigenvalue has been used to calculate the instability factor V associated with each flexural critical speed of the shaft-train. Instability factors that exceed the unity indicate the presence of conditions that can cause the occurrence of steam-whirl instability phenomena. Table 1 shows the flexural critical speeds in which the HP steam turbine is involved and the respective instability factors of the machine-train evaluated at the operating speed without considering the stiffness of the seals.

Table 1: Flexural critical speeds and instability factors evaluated with the model n.1, at 3000 rpm, without considering the stiffness of the seals (off-load operating conditions: cs = 0).

Mode n.	Flexural Critical Speed [rpm]	Instability Factor (V)	Mode n.	Flexural Critical Speed [rpm]	Instability Factor (V)
5	1781	0.8862	6	1853	0.7964

As said above, the scaling factors cs that exceed the unity are associated with an unrealistic steam flow which is higher than the nominal rating. However, the results obtained with these arbitrary values of the steam flow allow the threshold level that avoids the occurrence of steam-whirl instability phenomena to be estimated. This study provides an estimate of the safety margin of the machine-train that prevent occurrence of unstable vibrations.

The effects of the seal stiffening on the instability factors associated with the respective critical speeds are shown in Figure 5. When the scaling factor cs exceeds 1.5 the instability factor significantly grows and reaches the unitary limit value when the ideal steam flow is nearly 215% of its nominal value (cs = 2.15).

On the basis of these results, the threshold level of the ideal steam flow that could cause unstable subsynchronous vibrations is significantly higher than the actual maximum nominal value. The results provided by this case study, using the machine nominal model, do not explain the reason for the occurrence of steam-whirl instability phenomena.

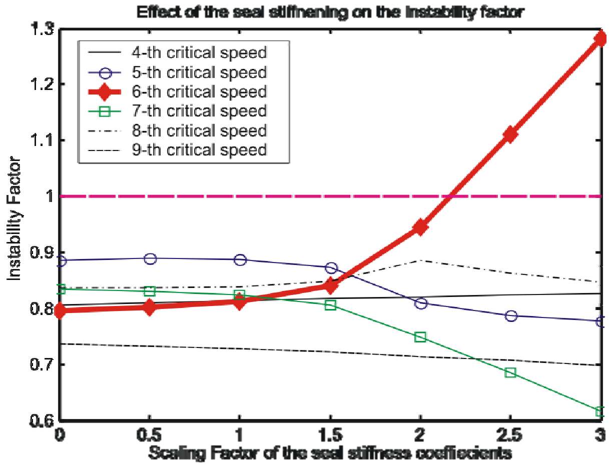

Figure 5: Instability factor versus exciting constant scaling factor

ACCURACY OF ACTUAL STABILITY CALCULATIONS OF STEAM TURBINES

Sometimes accurate calculations made using well assessed procedures for standard design machines are insufficient to avoid steam whirl conditions: this could be attributed to some modelling inaccuracies which are not recognized. Steam turbine manufacturers are developing new design for higher efficiency machines, where different types of seal or tighter clearances are used, for which more accurate modelling is required. Brush seals also have been tested experimentally and their behaviour has been modelled.

For these two reasons (inaccuracies in existing models and need of accurate analysis of new design) the interest on flow excited instability in high performance turbo-machinery has recently increased.

Analysing the standard approach for calculating stability margins of steam turbines (as e.g. in ABB, 1994), one could find following aspects which are neglected, or taken into account by more or less rough approximations or interpolations.

a) The excitation coefficient for blade row is calculated assuming standard fluid dynamic behaviour. This means that:
b) Possible swirl components in circumferential direction are neglected and the circumferential fluid velocity distribution across the clearance is considered linear.
c) flow contraction and carry over flow are probably also neglected.
d) differences due to the dynamic behaviour of the fluid around a whirling shaft instead of eccentric rotating shaft are neglected, because the experiments have mainly be made with eccentric rotating shafts rather than with whirling shafts.
e) The *"gas bearing"* effect due to shrouds on the blade row is represented by a corrective factor which is interpolated between some few experimental results.
f) Non linearity are not taken into account, therefore:
g) the effect of misalignment of shaft with respect to casing, as well as the resulting unequal pressure and velocity distributions in the seal is not considered.
h) the effect of actual orbit size and shape in correspondence of each seal is also completely neglected.
i) Effect of different operating conditions with respect to design cannot be considered.
j) Stability or instability is evaluated calculating the real part of the eigenfrequency of the shaft or of the shaft-line. In this calculation the linearized stiffness and damping coefficients of the oil film bearings give a significant contribution, therefore its values must be well assessed in design conditions. But these values also can change in operating conditions due to different bearing alignment conditions, to different oil temperatures, and to non linear effects since the orbits of the shaft inside the bearing is not vanishing small, which is the condition in which linearized coefficients are calculated.

Regarding the last point it is worth noting that the accuracy of eigenfrequency calculation is not so high as forced motion calculation in the frequency domain: therefore following procedure could be introduced.

A whirling motion at critical speed can be excited by a rotating force; the energies introduced by seals and dissipated by damping in bearings can be evaluated considering the different elliptical orbits in the different locations. If the net energy is positive the system is unstable, if negative stable.

This method could be more robust with respect to eigenfrequency calculation. It would also allow to localize the seals which contribute more to the onset of instability, or the bearings which contribute more to stability.

If desired also nonlinear forces in bearings and in seals could be calculated in some few points of the orbits for having a more accurate evaluation of the energy involved in the assumed whirling motion.

It seems reasonable to decouple the onset of instability from steady state vibrational behaviour, but considering the non-linearity in seals and bearings it could also result that a strong 1X component in bearings and seals, due to unbalance and bow, would be able to overcome the onset of instability.

Summarizing the above comments and proposals, it seems that there is still some development to carry out to increase the accuracy in steam whirl stability margin evaluation.

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