Optical vibration measuring system for long, free-standing LP rotor blades

The increasing use of large low-pressure steam turbines in combined cycle power plants has led to the requirement for these machines to be designed for a wider range of applications, eg to take account of larger speed variations, more start-up cycles and higher condenser pressures. Among those parts of the turbines which have to withstand the highest mechanical stresses are the low-pressure last-stage blades. While it has long been standard practice to calculate the static stresses and natural frequencies in the centrifugal force field, the stochastic excitation forces occurring during part-load operation and at increased condenser pressures are still determined experimentally. A new technology employing optical probes allows these forces to be measured without causing any major disturbance to normal operation of the turbine. Measurements carried out confirm that free-standing last-stage blades from ABB easily meet all the requirements for applications in combined cycle plants.

Power plant operators make very high demands on the availability of their steam turbines. For example, intervals between overhauls of up to 100,000 hours are expected for the low-pressure turbines. The rotor blades play an important role in this on account of their size, which causes them to be subjected to very high mechanical loads [1].

ABB has designed the last stage of its low-pressure turbines with free-standing rotor blades **1** for more than 30 years. Normally, precision forgings made of 12 % chromium steel are used for these blades, which are fixed to the shaft by means of a milled firtree root.

To guarantee safe and reliable operation under all possible service conditions and at every load point, great care has to be given to the mechanical dimensioning of the blades, with special attention paid to the static and dynamic stresses.

Using advanced calculation software, the static stress in the centrifugal force field due to the steam flow can be determined with a sufficient degree of accuracy even for the more complex zones of the blade fixation.

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When dimensioning for the dynamic stresses, it is essential to avoid blade resonances with high amplitudes in the lower eigenmodes. Here, the design with free-standing blades has a significant advantage over the coupled designs since the the calculation of the frequency is made easier by the simpler boundary conditions. The only boundary conditions that need to be taken into account involve the fixation of the blade root, and play a lesser role in the case of long blades.

Even when the blades are designed to avoid resonance, certain operating points can still give rise to dynamic stresses which cannot be neglected, for example when:

- Passing through resonances: passage through resonances with increased vibration amplitudes is unavoidable during start-up and shut-down.
- The condenser pressure is increased: higher steam densities cause the stochastic excitation forces to be increased.
- Volumetric flow rates are very low: these can give rise to incidences and flow separation, resulting in higher stochastic excitation forces.
- Volumetric flow rates are very high: blades which are very compliant are susceptible to flutter. The usual result is an increase in blade vibration at the fundamental frequency.
- Operating at overspeed and underspeed: resonant frequencies are more likely to be approached. This is a typical phenomenon in weak electrical power systems.

The above operating points have taken on new importance in recent years, especially since combined cycle facilities are started up and shut down more often than conventional steam power plants. Also, when dry cooling towers are installed or heat is extracted, the volumetric flow rates and condenser pressures in the LP turbine depend heavily on the ambient temperatures.

Free-standing last-stage blade in the low-pressure stage of a large steam turbine. Such blades have lengths of up to 1200 mm. **1**

In order to better understand the vibrational behaviour of the last-stage rotor blades and to improve the software used for its analysis, measurements therefore need to be carried out in full-scale, operating power plants.

Optical blade vibration measurement system

ABB began already in the 1970s to develop an optical blade vibration measurement system, which it designated OSS [2]. The system has since been employed in a range of power plants, resulting in continual improvement [3].

The optical blade vibration measurement system offers several characteristic benefits:

- No components need to be fixed to the rotating parts.
- All blades in a row are measured.
- Sensitivity is greater than with electromagnetic sensors.

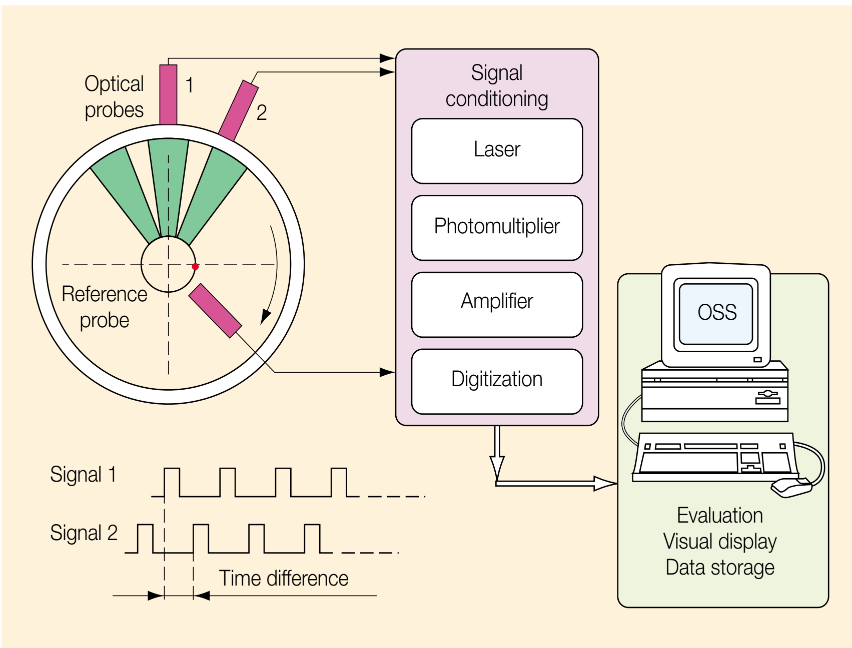
The OSS system is based on measurement of the difference in the time taken by the tip of a vibrating blade and of a non-vibrating blade to pass through a defined circumferential angle. The measurement is carried out with the help of two optical probes which are mounted inside the turbine casing above the rotor blade row. These two probes transmit a laser beam to the tips of the blades. A light signal reflected by the tips is received by the probes and sent via a fiber-optic link to a photomultiplier **2**. The time intervals between the reflected light pulses are a measure of the blade vibration. Another sensing element, known as a key phasor, generates the speed and a reference signal with which the transmitted pulses can be assigned to the individual blades. The measured time differences are stored, analyzed and displayed on a personal computer, making on-line monitoring as well as detailed analyses possible.

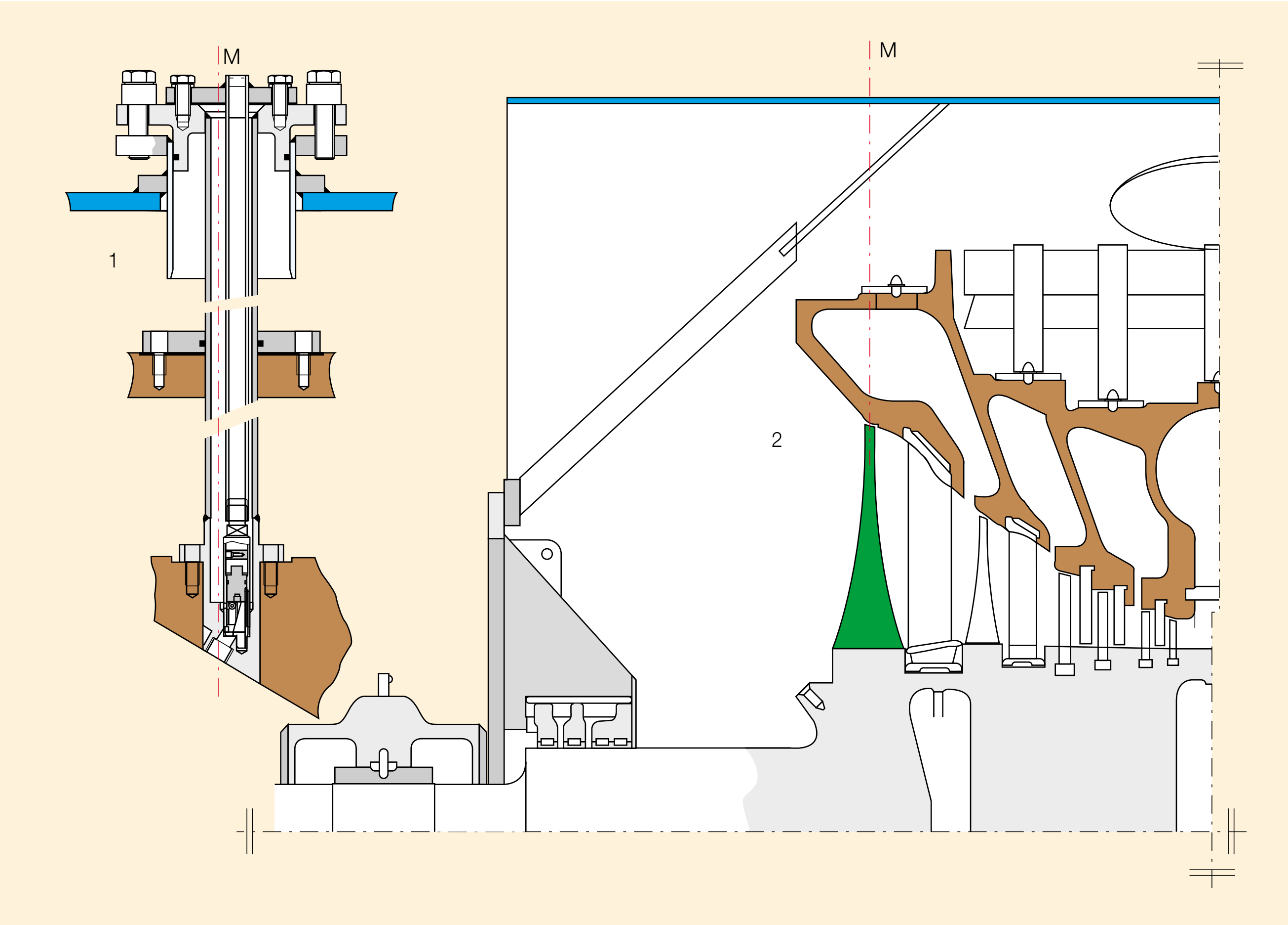
Measurements performed on a CCPP steam turbine

The optical blade vibration measurements described in the following were performed on a steam turbine in a combined cycle power plant (CCPP) in the USA. The unit, with a nominal output of 134 MW, consists of a single-flow, single-casing HP/IP turbine and a double-flow LP turbine of type ND34 **3**. The last-stage blades are approximately 880 mm long. After passing through the last stage, the steam flows downwards to the condenser. A wet cooling tower is used to cool the water circuit.

For the purpose of the measurements the steam turbine was operated at different load points, the majority of which rep-

Principle of operation of the optical blade vibration measuring system OSS **2**



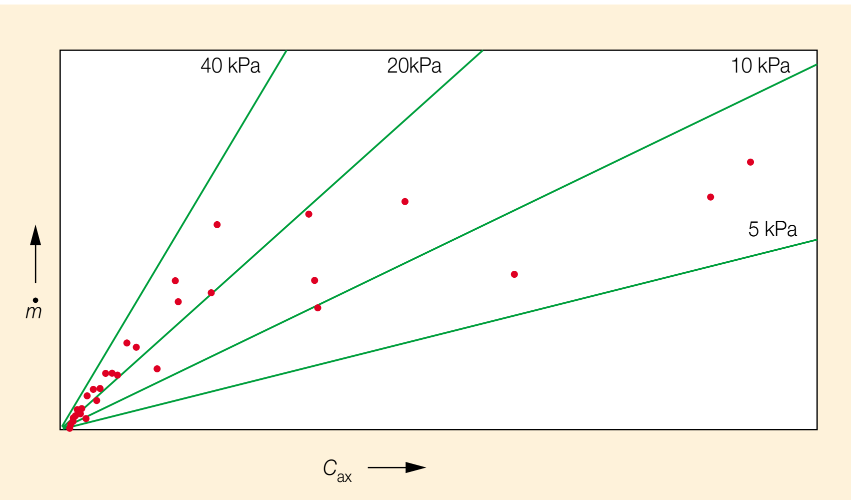


Section through half of a double-flow LP turbine on which optical vibration measurements were carried out **3**

1 *Optical probe, including holder*
2 *Section through half of the LP turbine*
M *Measuring plane*

***Measured operating points for the LP turbine in* 3** **4**

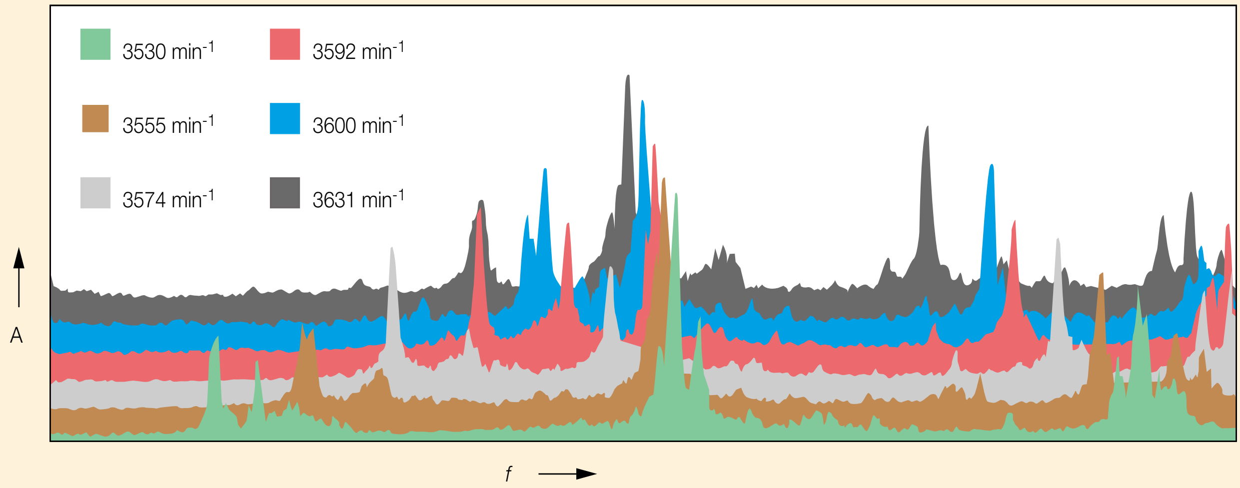
$\dot{m}$ *Volumetric mass flow*
C_{ax} *Axial velocity at outlet*
Green *Condenser pressure*



resented part-load conditions. Since the volumetric flow rate and the steam density at the blading outlet are mainly responsible for the stresses in the final stage, the measuring points are plotted in the $\dot{m}$ versus C_{ax} diagram shown in **4**.

Natural frequencies of blades during operation

As already explained, the last-stage rotor blades are designed such that there is no resonance in the lower eigenmodes. When testing the initial design of new blading in the overspeed pit, it has been usual in the past to verify the natural frequencies of the individual blades experimentally using strain gauges and telem-



Measured frequency spectrum of a last-stage blade for slight variations in speed **5**

A *Amplitude* f *Frequency*

etry for the signal transmission. At the same time, a transmission function has to be derived for the measurement in the zero-speed frequency test rig, which is mandatory for every free-standing blade. This method has been in use for years and seeks to show that the natural frequencies of the blades lie outside of the generously defined, so-called 'unacceptable frequency range'.

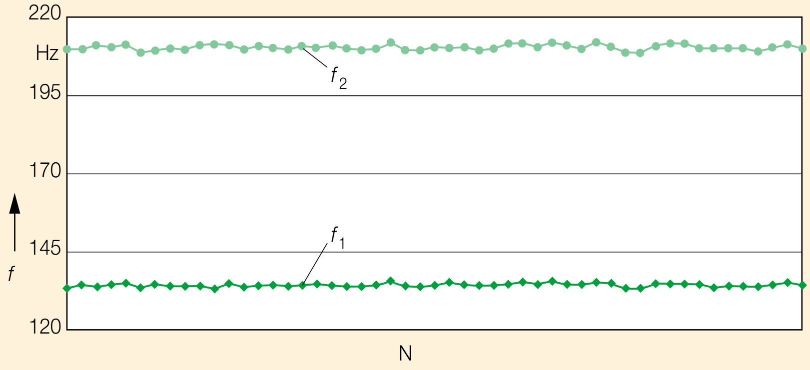
If these reserves are utilized, eg in order to increase the size of the speed 'window', the frequencies of the individual blades can be determined directly by means of measurements carried out on a complete blade ring. In addition, the transmission function for the zero-speed frequency test rig can be improved by measurements performed in service, thereby enabling the safety margin to be estimated more reliably.

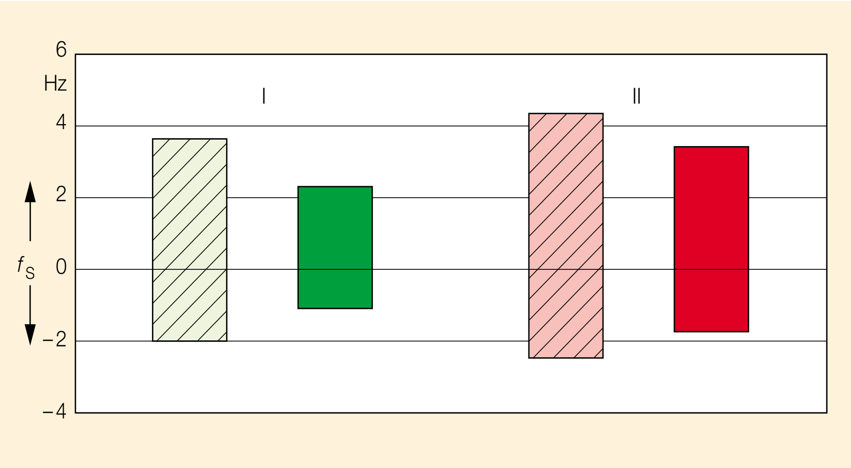
If the time signal received during the blade vibration measurement is shifted by means of a Fourier transform into the frequency domain, the natural frequencies of the blade appear as recognizable signal peaks. Undersampling with only two probes causes all the natural frequencies to be 'folded' into half of the operating speed range. 'Unfolding' of the natural frequencies is theoretically possible when their approximate location is known. However, if the peaks lie close together in the frequency spectrum or the excitation forces are too low for a significant signal to be generated, ie one which stands out from the background noise, an analysis of the signals for slight variations in speed will be worthwhile. Due to the different number of folds for the different natural frequencies, the position of the latter varies strongly within the frequency spectrum, making their allocation easier **5**.

Using this method, it was possible to clearly identify the first five natural frequencies for each blade in operation. **6** shows the distribution of both of the first natural frequencies. The comparison with the measurements carried out on the

Location of the first two natural frequencies of a blade at nominal speed **6**

f *Frequency* N *Blade slot*





Maximum scatter of the natural frequencies of a blade in the zero-speed frequency test rig and in operation **7**

f_S	*Maximum frequency scatter*	Hatched	*Zero-speed frequency test rig*
I, II	*1st and 2nd natural frequency*	Green, red	*Operation*

zero-speed frequency test rig shows that the blade frequency scatter during operation is significantly smaller **7**, ie the centrifugal force occurring in operation reduces the variation in the clamping conditions in the zero-speed frequency test rig, which is a direct result of the manufacturing process. The variation is anyway small enough to be fully covered by the 'unacceptable frequency range' currently applying.

The quality of the optical measuring system is highlighted by **8**. Low volumetric flow rates are the cause of windage in the last-stage zone, as a result of which the blade temperatures rise. The variations in blade frequency resulting from the temperature dependence of the modulus of elasticity can be clearly seen. For the measurement considered here, the frequency resolution was estimated to be 0.1 Hz.

Blade vibrations during the passage through resonance

Free-standing blades have numerous advantages over coupled blades providing the overall design does not strongly excite any of the individual natural frequencies. Since the free-standing blades have no damping mechanisms of any significance of their own, during the passage through resonance excitation forces of the order of magnitude of one percent of the static forces are enough to cause high vibration amplitudes.

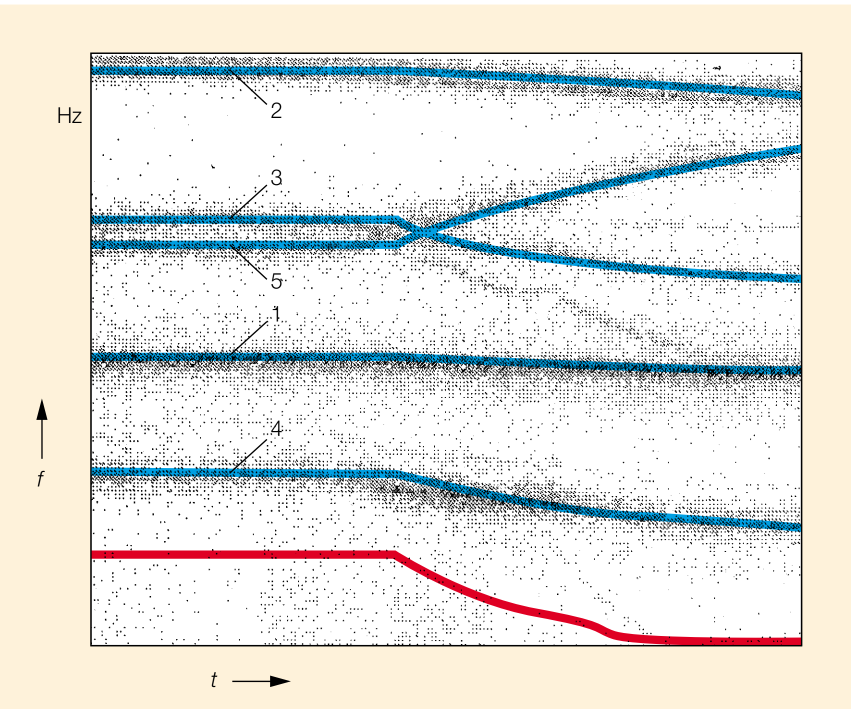
For the evaluation of the measurements during start-ups and shut-downs, the components of the first five natural frequencies were superposed upon one another in order to determine the maximum dynamic stress in the airfoil. It was shown that the maximum dynamic stress during the passage through resonance was not more than about 2 to 3 times higher than the background noise (left-hand part of **9**). Acceleration as well as run-down of the turbine took place with the usual speed ramps.

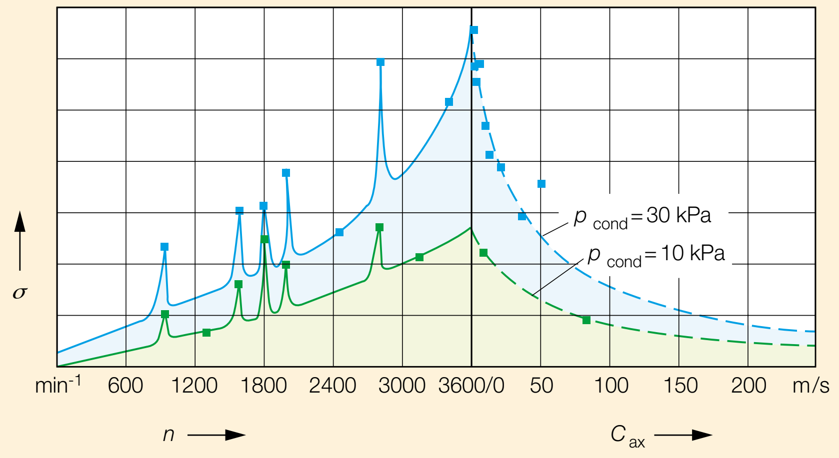
Blade vibrations at increased backpressure

The areas of application for practically all long last-stage blades, whether free-standing or coupled, are all limited by the

Measured load dependence of the natural frequencies of a blade **8**

f	*Frequency (folded)*	1–5	*1st to 5th natural frequency*
t	*Time*	Red	*Generator output*





Measured dynamic loading of a last-stage blade **9**

σ Dynamic load
n Rotational speed
C_{ax} Axial velocity at outlet
p_{cond} Condenser pressure

maximum condenser pressure. This is due to the increase in excitation forces that is caused by the higher steam density and, in association with it, the lower volumetric flow rates with windage in the area of the last stage.

In order to clearly define the maximum possible area of application of the low-pressure turbine, measurements were performed with it running at numerous part-load operating points and with increased condenser pressures.

The results of the measurements can be summarized as follows (right-hand part of **9**):

- When the vacuum is good (p_{cond} < 20 kPa), the blade vibrations are negligible at both very high and very low axial velocities.
- At moderate vacuum (20 kPa < p_{cond} < 30 kPa), a minimum axial velocity has to be ensured for continuous operation; ie, if the flow through the last stage is sufficient, operation at the higher condenser pressures is easily possible.
- When the vacuum is poor (p_{cond} > 30 kPa), continuous operation of the blades is not recommended. Although the experimentally determined maximum stress amplitudes still lie well below the vibration fatigue strength values of the blade material, they should be avoided whenever boundary conditions are unfavourable.

Higher availability as the goal

Extensive measurements carried out at different load points and with a range of condenser pressures, as well as during start-up and shut-down, have confirmed the absolute mechanical integrity of the free-standing blade design under the rigorous operating conditions in combined cycle power plants.

Providing the axial velocity of the steam does not drop below a certain minimum value, thereby ensuring a sufficient steam flow rate in the last stage, such blades can be operated trouble-free up to a vacuum of 30 kPa. The rigid construction makes it possible for them to be operated with volume flow densities significantly higher than 20 kg/sm².

For plant operators the main benefit is the higher turbine availability that results from the improved operational flexibility.

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