Proactive product & process qualification in steam turbine development

Proactive qualification is today an integral part of the development process for steam turbines used in steam and combined cycle power plants. By providing experimentally based 'random' variables, such as the steam path efficiency and blading reliability, it makes a valuable contribution to the economic 'random' variables of the power plant, eg the rate of return, as viewed by the customer. One of the areas in which proactive product and process qualification plays an important role is the development of LP steam paths.

Ideally, product development in the power plant sector should rigorously target the wishes and requirements of potential customers. In this direct interpretation of ABB's well-established Customer Focus programme, both economic and ecological investigations play an important role [1]. Also, the idealistic, deterministic point of view increasingly is being replaced by a more realistic, probabilistic viewpoint [2]. This trend has been especially noticeable since the energy markets started to deregulate and ecological factors began to be looked at from an economic standpoint. In the power plant context, the economic random variables that serve as 'global' measurands are the rate of return on equity (ROR), the net present value index (NPVI) and the payback time (Y) [2], calculated in each case for the time from tendering to the end of utilization. Because of this long period, only the probabilistic approach ensures adequate results, especially with regard to the monetary risks.

Which of the three economic random variables is most important for the appraisal will depend upon the standpoint of the individual customer. The ROR additionally allows a comparison of the power plant investment with alternative financial investments. A complete characterization of the associated populations of ROR, NPVI and Y is given by the sample means, variances and sizes as well as the chosen confidence level. Usually, normal or log-normal distributions are assumed.

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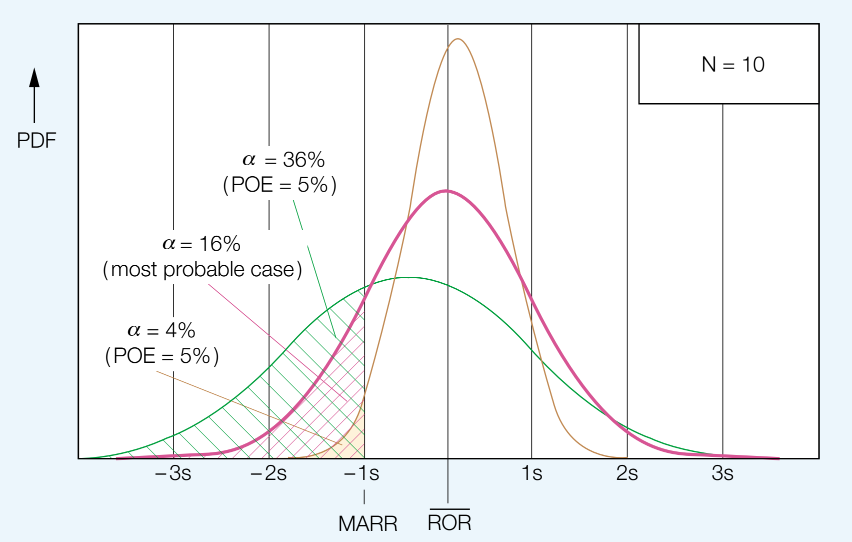
Given the ROR, NPVI or Y considered by the customer to be the 'minimal attractive' values, the probability of non-conformance can be determined from the economic random variables of the power plant using familiar statistical methods. Since the sample mean and variance are both random, the probability of non-conformance can only be given in terms of the mode (most probable value) and the confidence limits for a chosen confidence level. This situation is represented in **1**, in which the ROR is taken as an example.

In the diagram the sample size is 10 and the confidence level 90 %. Of the total population, 16 % most likely lie below the Minimal Attractive ROR (MARR). In the worst case this value can rise for the chosen confidence level to 36 %, which is the upper confidence limit, and in the best case drop to 4 %, representing the lower confidence limit. Even with a sample size of 50, the confidence limits would still lie at 25 % and 10 % given the same mode.

This example shows that the sample size has only a minor influence on the confidence limits. Since power plants are not mass produced, the best way to reduce the probability of non-conformance is to reduce the sample variance. If in the above example the sample variance is made approximately 4 times smaller, the mode will lie at 2.5 % and the confidence limits at 15 % and 0.2 %, respectively. With the help of the NPVI the values can also be set in relation to the investment, allowing them to be expressed in monetary terms, and especially in terms of monetary risk.

A superior product from the customer's point of view, ie one with the best *total quality*, is thus better than the products with which it competes in that it exhibits, for the specified MARR, the lowest confidence limits for the probability of non-conformance while complying with ever-stricter environmental legislation.



Probability of non-conformance for the MARR of a power plant, as seen by customers **1**

ROR *Rate of return (sample mean: $\overline{ROR}$)*
PDF *Probability density function*
MARR *Minimal attractive ROR*
N *Size of sample*
POE *Probability of limit being exceeded*
s *Sample standard deviation of ROR*
α *Probability of non-conformance*

Analysis of the probability of non-conformance

In view of the extreme economic importance of the probability of non-conformance for the power plant appraisal, the input random variables for the Monte-Carlo simulation used to investigate the ROR, NPVI and Y, must be known. This also generally presumes knowledge of the sample means, variances and sizes as well as the confidence level for the populations.

In the case of thermal power plants, a distinction has to be made between the following input random variables:

- *Market-specific random variables*, such as the interest rate on debt, income tax rate, capital tax and insurance, capacity factor (during operation), specific fuel and reserve power costs, specific revenues, environmental limits (eg, noise and flue-gas emissions).
- *Customer-specific random variables*, such as the equity ratio, interest on equity, period of utilization, reserve power ratio (depends on size and reliability of the utility network), scheduled service factor and cycling rate.
- *Vendor-specific random variables*, eg the specific investment costs at the time of signing of the contract, the time for erection and commissioning, heat rate, forced, maintenance and planned outage factors and their respective costs, as well as the specific cost of operation.

It is important in the case of the third group of variables to distinguish between the forced, maintenance and planned outages, as their respective economic weight depends greatly on the specific reserve power costs.

Proactive versus reactive qualification

It may be assumed that the best power plant vendors or suppliers of power plant components will know both the mean and variance of each of 'their' random variables within the framework of a continuous qualification process. An important role will also be played by the target values and the probability of non-conformance. Thus, the supplier obtains an important link in the process of continuous improvement, and one which also has to be managed from the economic viewpoint.

A qualification process, by definition, only admits factual evidence. Besides being reactive, it should also allow proactive qualification in the case of the especially sensitive random variables. The alternative to this is the verification process, which is characterized by theoretical or numerical reasoning.

Reactive qualification

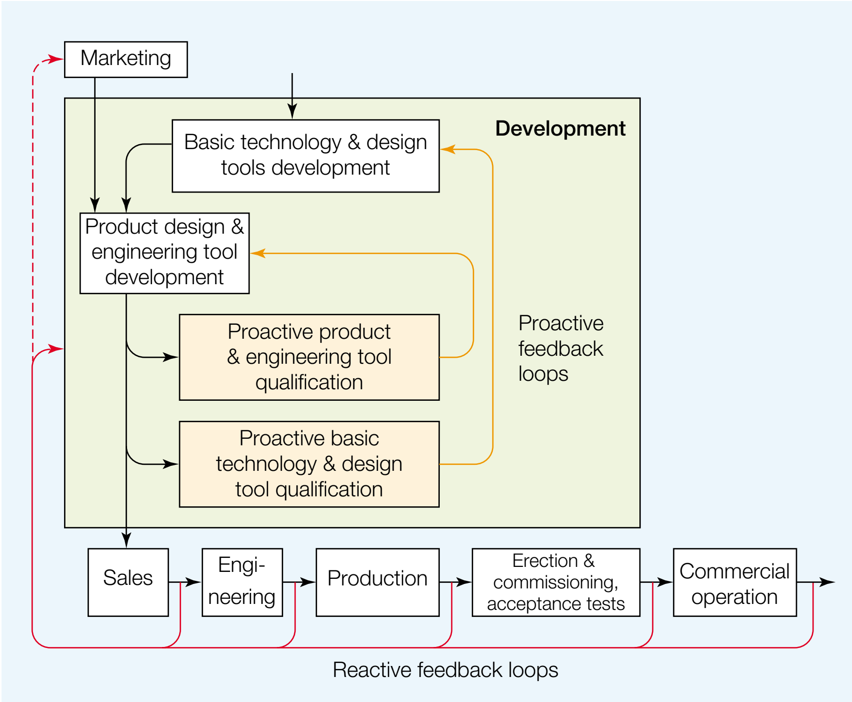
In reactive qualification, the supplier obtains his random variables from the statistical evaluation of the experience accumulated during the order handling, in particular between the production phase and the start-up of commercial operation. Non-conformance reports (NCRs) and the reactive feedback loops provide the bulk of the data **2**.

Experience has shown, however, that the variances for the different random variables will be considerable even after corrections have been made to take account of the known deviations of the components, their operating conditions and the systematic errors in the measurements. Many reasons exist for the larger variances, which are composed of random and unknown systematic errors: differences in the operation geometry, in the properties of the materials and in the operating conditions of the power plant, as well as in the test methods, test equipment and test personnel.

For example, IEC 953-2 allows relatively large fluctuations in the operating conditions: initial steam pressure and temperature ± 2.5 % and ± 7.5 K, respectively; exhaust pressure ± 12.5 %; extraction flow rate ± 5 %; speed ± 1 %. Under these conditions the output and initial steam flow can fluctuate by ± 3 %. For the heat rate of power plants with extraction condensing turbines the confidence limits that apply are ± 0.9 to 1.2 %; for the thermodynamic efficiency of the turbine (corresponding to an isentropic efficiency) a value of ± 1.3 to 2.0 % is even given.

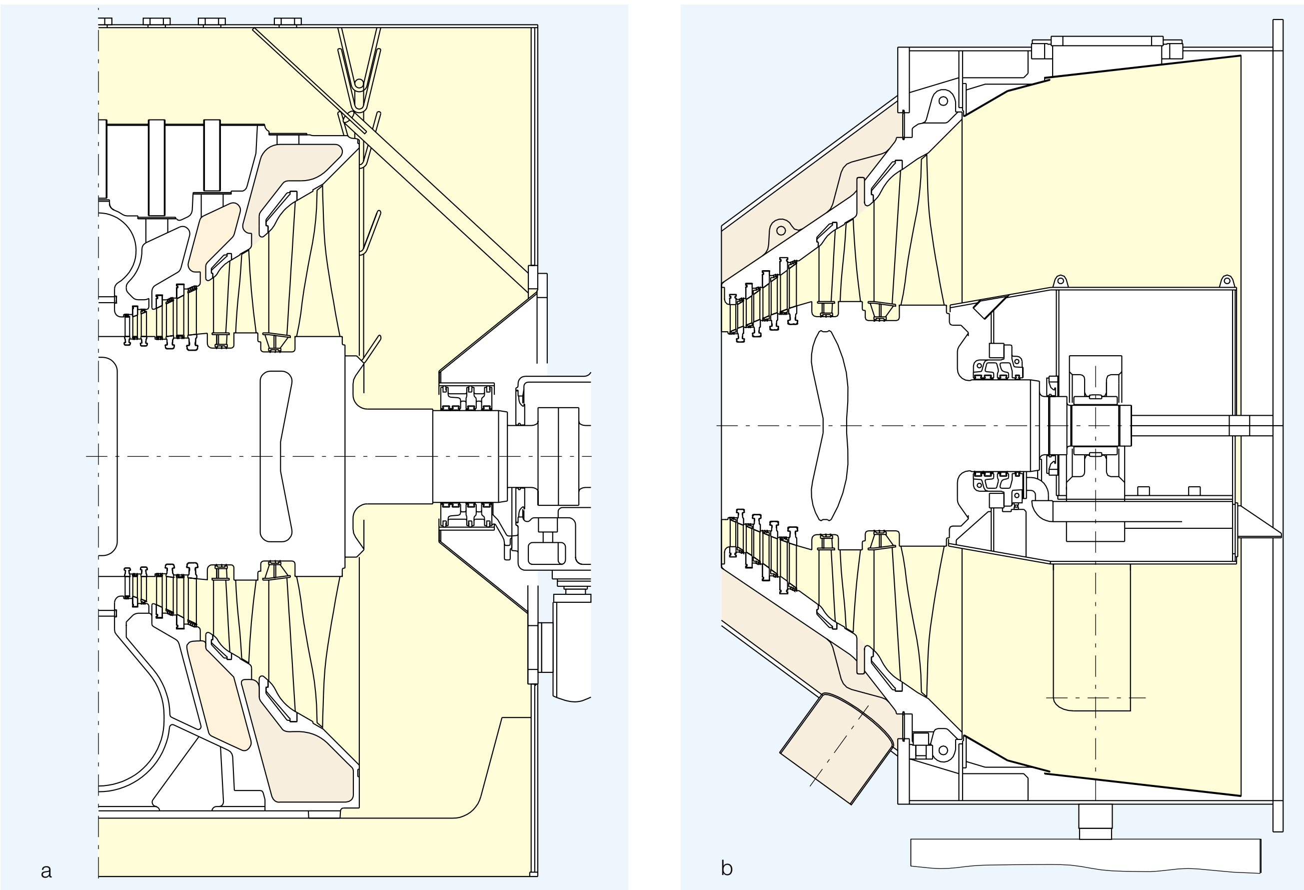
In addition, only a small number of power plants of the same type are manufactured per time-unit, leading to large confidence intervals, particularly during the market introduction, and making a systematic, quick analysis and possible improvement of the product and the related processes extremely difficult.

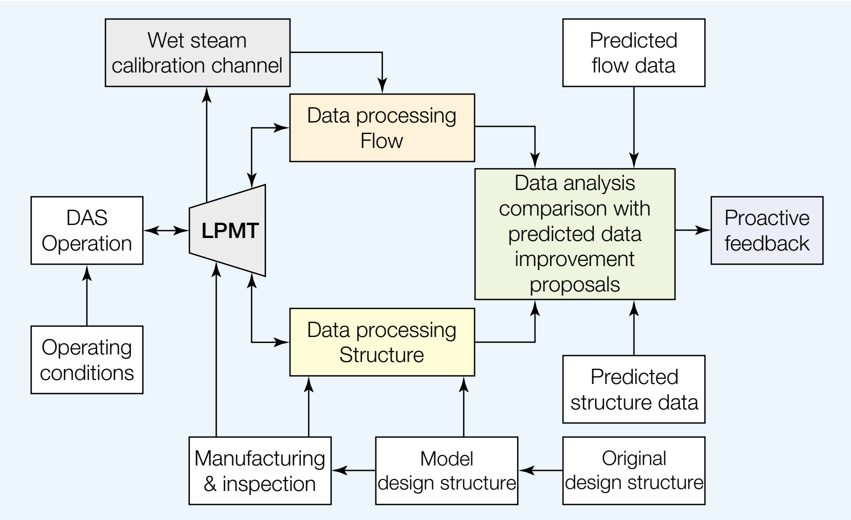
A further sub-division of the random variables (eg, of the heat rate) into the efficiency of the HP, IP and LP turbines, is only possible with large variances due to the reduced metrological discretization and the usual absence of direct measurement of the physical (mass, mechanical and ther-



Business process for steam power plants **2**

Section of a double-flow LP turbine (ND41) with axial/radial exhaust (a) and of a single-flow LP turbine (ND41) with axial/axial exhaust (b) **3**





Proactive qualification process for LP steam paths **4**

LPMT *Low-pressure model turbine* DAS *Data acquisition system*

mal) flows. The situation becomes even more extreme when the steam paths also have to be sub-divided into their components, namely the inlet, blading, outlet and extractions.

Thus, no experimental results exhibiting just a small variance are available which could be used for, eg, the local dissipation power and therefore for the qualification of the product, the basic technology, the design or the engineering tools. Active development control is therefore made impossible.

Proactive qualification

In contrast to reactive qualification, proactive qualification begins already during development, ie long before commercial operation. As a rule, this 'concurrent engineering' phase ends before the first newly designed components leave the works **2**.

Unlike reactive qualification, which covers the total, ever-growing sample size of power plant components, proactive qualification is based on individual, representative and precisely known random samples values. It involves rigorously reducing the systematic and random measurement errors enough to allow approximately true values of the relevant measurands to be obtained. By matching them to the individual random variables from the reactive qualification, the associated confidence intervals are considerably reduced, since the results with low variance dominate.

Neither does the proactive qualification cover all the vendor-specific random variables. It targets primarily the contributions made by the steam paths to the following key economic and vendor-specific random variables:

- *Specific investment costs*, based on fluid mechanics investigations to determine the mechanical power
- *Heat rate*, based on fluid mechanics investigations to determine the efficiencies
- *Forced outage factor*, based on structural investigations to determine the reliability of the blading

Only a short-term analysis of the lifetime-reducing phenomena and a subsequent projection are possible. For example, the reliability of the blading can be predicted on the basis of blade vibration measurements.

The special strength of the proactive qualification is therefore the fast, proactive feedback loop with high-quality experimental data. As a rule, these data are obtained for individual steam paths under laboratory conditions and are characterized by extremely small systematic errors and variances. The measurements are carried out under precisely known geometric and operational boundary conditions. A special status is given to the interaction between the fluid and the structure and between the actual flow domains. Other important factors are the use of advanced test methods and equipment plus the fact that the personnel carrying out the tests are the same each time.

Consequently, proactive qualification of the product is based on a single model. The same also applies to the basic technology and the design and engineering tools. An example of proactive qualification is given in the following, with reference to the LP steam paths of large steam turbines employed in power plants.

Realization of a proactive qualification process for LP steam paths

Existing LP steam paths

Geometry

It is normal for large extraction condensing steam turbines to have 1 to 3 double-flow LP turbines. The development of the latest generation of such turbines has been described on numerous occasions (eg [3]).

3a shows an elementary steam path with one half of the inlet scroll and the radial/axial stage, three axial front stages with cylindrical hub, two final stages (L-1 and L-0), the axial/radial diffusor and the exhaust-steam casing. Also shown are the extractions A1 to A3 (numbered counter to the flow direction). The steam paths, of which there may be 6, are identical except for the direction of rotation (viewed in the direction of the flow) and the geometry of the blade labyrinths or blade clearances (L-0, L-1).

In the fleet leaders now in operation – ND41 for 50 Hz and ND34 for 60 Hz – the last-stage blade lengths are 1050 mm (approx 41 inches) and 880 mm (approx 34 inches), respectively. LP steam paths of the described kind have already been used in retrofit projects [4].

Single-flow LP turbines are also used [5] in small extraction-condensing turbines and combined cycle plants featuring small exhaust-steam volume flows. The steam path begins in these cases with a crossover channel, axial front stages (eg, 4) with a conical hub, two last stages (as in the double-flow turbines), an axial/radial diffusor (or axial/axial, as an option) plus the exhaust-steam casing.

3b shows a single-flow LP turbine with axial/axial diffusor and a single extraction, as is commonly used in combined cycle applications.

Both types of LP turbine are adapted to the required mass flow and pressure by varying the geometry of the front stages and, where applicable, the radial/axial stage.

Operating conditions

The inlet state data for conventional steam power plants lie, as a rule, in the range of 4 to 6 bar and 150 to 320 °C; the backpressure lies between 30 and 120 mbar, and the individual extraction mass flows in the range 3 to 7 % of the inlet mass flows. For the large ND41 and ND34 turbines, the speed is fixed at 50 rev/s and 60 rev/s, respectively.

Structure

The flow-governing geometry varies, due to the fluid-structure interaction, between assembly and operation. For example, the stationary, axial differential expansion in a 6th steam path can be greater than 10 mm.

Fluid flow

The LP steam paths are responsible for approximately 35 % of the power in conventional steam power plants with reheat and approximately 45 % in steam turbines without reheat (eg, for combined cycle plants). The maximum power rating of one steam path can reach 130 MW, while the mass flow at the inlet can be as high as 210 kg/s.

LP steam paths are located at the end of the expansion, which means that 2/3 of the dissipation power has to be written off as exergy loss flux, and therefore as a power loss. Due to the difference in the operation geometry of the LP steam paths, especially in the area of the blade tip section, the individual steam paths (eg, for a turbine with 6 steam paths) will exhibit different flow patterns.

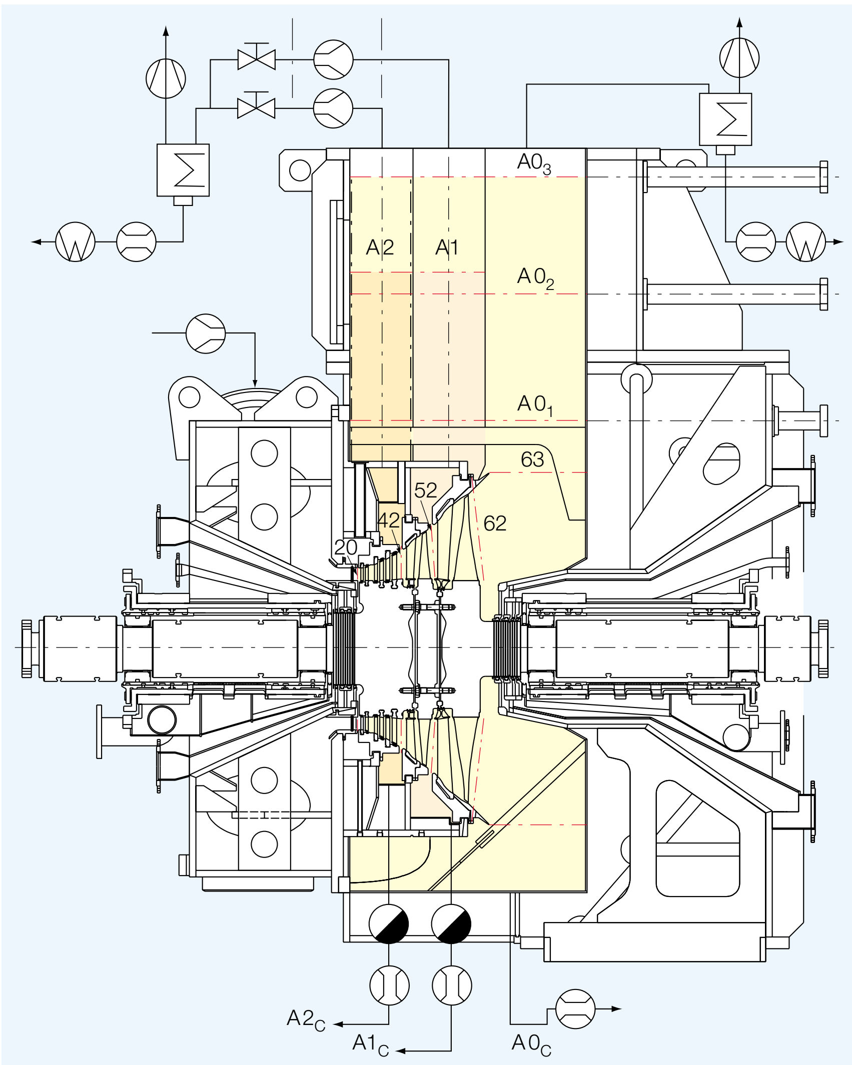
Under certain inlet conditions the saturation line will be crossed in the stages L-4 to L-1. The relaxation effects occurring as a result are particularly difficult to control, and are a cause of variance in the prediction of the dissipation power.

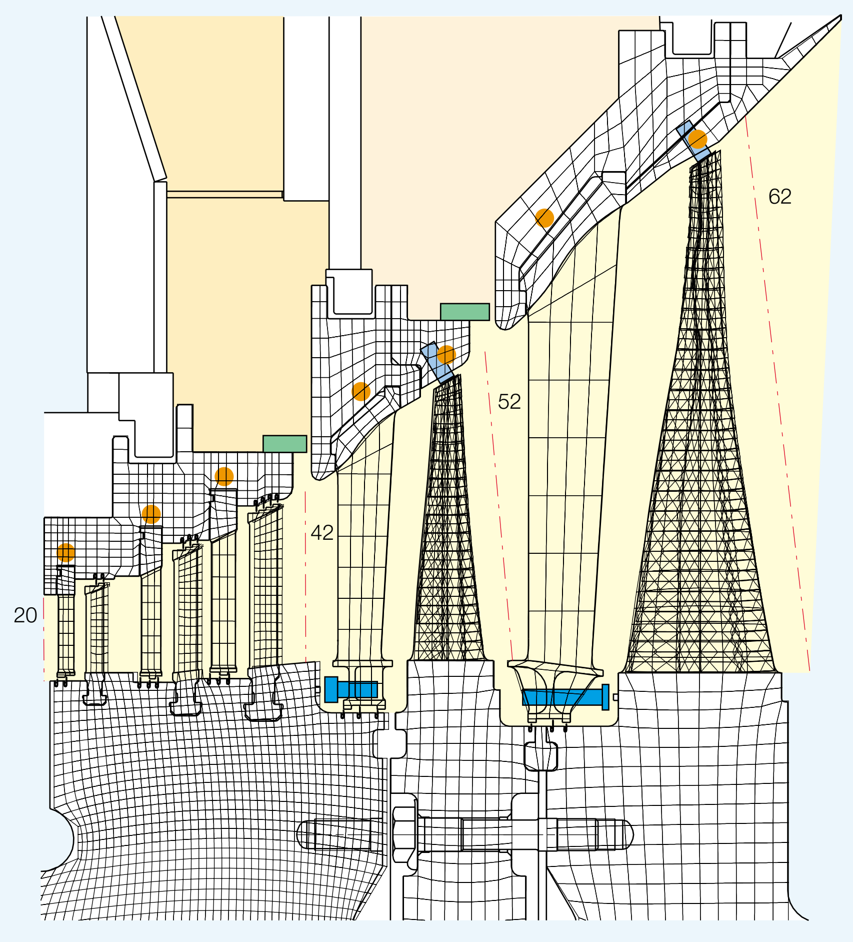
Concept for a proactive qualification process for LP turbines

It has already been shown that the established acceptance measurements them-

LP model steam path with control surfaces (CS), shown in the LP model turbine (ND41) **5**

CS 20, 42, 52, 62	*Blading*	CS A1, A2	*Extractions*
CS 63, $A0_1$, $A0_2$, $A0_3$	*Diffusor and exhaust*	CS $A0_C$, $A1_C$, $A2_C$	*Coarse water drains*





Blading for the LP model steam path with control surfaces (ND41). The grid is shown for operating conditions. **6**

Temperature
Vane carrier gaps
Rotor/stator gaps
Clearance & blade vibrations, blade deflection (L-0)

selves allow reasonably reliable predictions of the overall power plant performance. However, even the determination of the overall turbine performance, covering the HP, IP and LP steam paths, is accompanied by large variances. And the variances increase further when attention turns to the performance of the individual steam paths. Obviously, a detailed look at the performance of the components of the individual steam paths would show even more extreme variance.

A way out of this problem is to use experimental turbines designed specifically for proactive qualification. The most important issue involved is the geometric and physical modelling of the described LP steam paths.

Geometric modelling

The choice here is between a complete or partial representation. The ideal approach would of course be a full representation of the LP steam paths. Only this approach will ensure complete geometric similarity. In a first, reliable simplification the inlet scroll and the axial/radial stage in **3a** are eliminated. Both of these elements have already been studied separately with the help of an air flow [3]. A second simplification involves extraction A3, which can be omitted. Both of these simplifications presume corrections, either by theoretical or numerical means, to the geometry of the first axial (L-4) stage. These are possible today in the dry steam region. For the same reason, and to allow a direct comparison, the L-5 stage in **3b** can also be eliminated.

Further simplifications to the flow-governing geometry may not be made as they could affect the measurements (the steam upstream of the blading must always be dry), the conditioning of the turbomachine-specific wet steam, which is problematic, and the interaction of certain components (front stages, stage L-1, stage L-0, diffusor, exhaust-steam casing, extractions).

Physical modelling

The physical similarity of the flow calls for equality of the similarity characteristics. In the dry-steam zone these are the isentropic exponent, Rossby number (flow/circumferential velocity), Reynolds and Mach number as well as the time-related inhomogeneities at all points in the flow field. In the wet-steam zone, Damköhler numbers (relaxation time of the fluid over the residence time of the flow), etc, are used to characterize the thermodynamic and kinematic relaxation process of the wet steam [6]. Both relaxation processes are responsible for increasing the dissipation power [7]; in addition, the kinematic relaxation process is also responsible for erosion.

A geometric reduction of the steam path by means of a scaling factor therefore has to be carried out with the rotational speed increased by the same factor and under the same conditions as for the original steam path. However, the Reynolds and Damköhler numbers will then always be different to those for the original steam path.

The correct scaling factor is obtained ultimately from an economic appraisal based on the ROR as viewed by the supplier. The manufacturing and operating costs of the experimental turbine as well as the achievable variances and their economical value play the main role here. A scaling factor of 0.316 was chosen for the ND41 considered here. As a result, the model Reynolds number will be 1.5 times smaller than the limit Reynolds number of 4.10⁵. The model Damköhler numbers will be 1.5 times larger than the original Damköhler numbers thanks to the simultaneous reduction in droplet size and residence time.

The qualification process, beginning with the adoption of the flow-governing design geometry, the operating conditions and the advance calculations, and ending with activation of the proactive feedback loop, is shown in **4**. Its most important elements are described below.

Design of experiment

Model steam path

5 shows the LP model steam path derived from the LP original steam path with axial/radial exhaust **3a**. The model is formed from the core components of the flow-governing geometry. **6** shows the blading enlarged. It must be emphasized here that the production tolerances and the surface roughness also have to be reduced by the scaling factor.

A feature worth noting is that the main steam outlet and extraction flows A1 and A2 are all located at the top. Because of this, the quantities of coarse water separated by the flow-governing geometry are extracted downwards and, most importantly, can be measured.

Operating conditions

The operating conditions derived from the original (full-scale) turbines are shown in an *h-s* diagram **7**. The quality of the steam also plays an important role in this. For the LP model steam path the fluctuations are several times smaller than those accepted by IEC 953-2.

Flow field

The controlled flow field (ie, the control volume) extend from control surface (CS) 20 at the main flow inlet to control surfaces $A0_1$ to $A0_3$ at the main flow outlet and A1 and A2 at the extraction flow outlet. To these must be added the 3 outlets for the coarse water flows $A0_c$, $A1_c$ and $A2_c$ **5**.

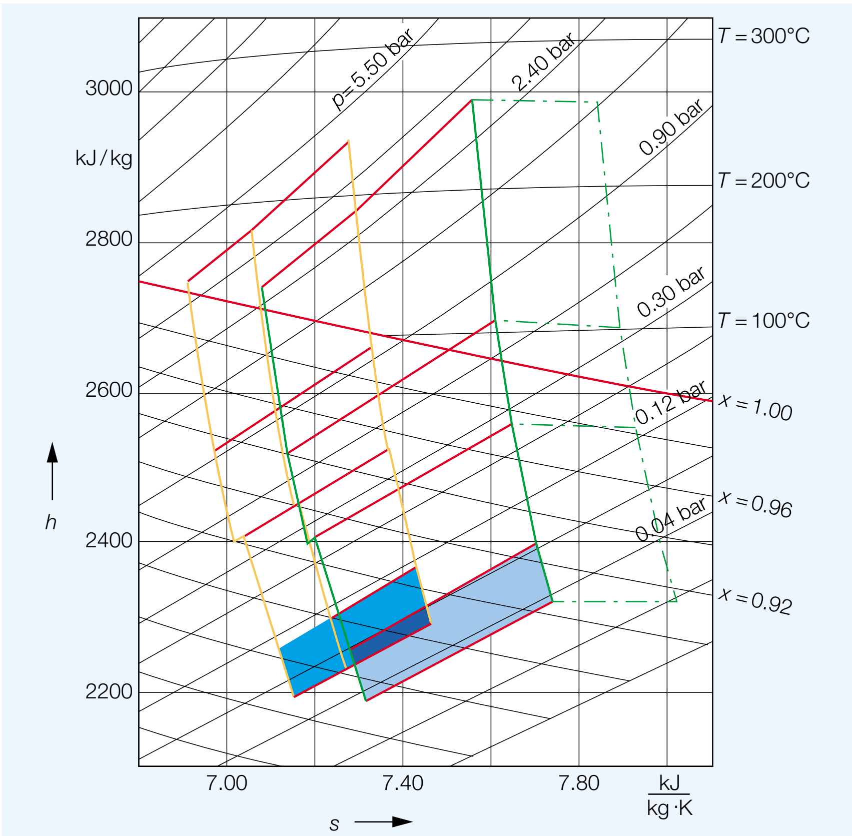
The additional control surfaces 42, 52, 62 and 63 allow the fluid-dynamic/thermodynamic subdivision of the steam path into its 7 components: front stages, stage L-1, stage L-0, diffusor, exhaust-steam casing (as far as CS $A0_1$ to $A0_3$, respectively), extraction A1 and extraction A2. A further subdivision, which would have made it necessary to introduce control surfaces after the vanes, was not undertaken due to the limited present-day capability of disturbance-free metrological systems.

To achieve the required balance it is necessary to measure the pressure and velocity vector as well as the temperature (for dry steam) or wetness and Sauter diameter (for wet steam) in the introduced control surfaces. The Sauter diameter, which is averaged via the droplet surfaces, characterizes the dispersion of the wetness and also contributes to the formation of the Damköhler numbers. The wall surface pressures are measured in the control surfaces as well as at distributed locations over the entire control volume, eg after the vanes, on the vanes and on the walls of the diffusor. The heat transfer coefficients are determined numerically, as will be explained later.

Although the mass fluxes through the different control surfaces can be determined from the field variables by numerical

Operating conditions for the LP model steam path (ND41) **7**

h *Enthalpy*
s *Entropy*
p *Pressure*
T *Temperature*
x *Vapour fraction*



integration, the mass fluxes through CS 20 (steam side), $A0_1$ to $A0_3$, A1 and A2 as well as $A0_c$, $A1_c$ and $A2_c$ (water side) are measured directly to ensure the required accuracy. Controlled blocking of the innermost labyrinths prevents parasitic steam from entering or exiting at this location.

To ensure accurate results the mechanical powers of the three front stages transferred via the rotating wall surfaces and of stages L-1 and L-0 are also measured directly. To this end it is necessary to fix the rotor part of stage L-1 once to the front-stage rotor and once to the end-stage rotor. Measurements are then carried out on the two rotors via the balancing casings of the water brakes and the hydrodynamic rotor bearings **5**. In addition, the rear rotor can be entrained by a Curtis turbine when necessary. The heat fluxes, which are also transferred via the wall surfaces, are determined numerically.

Displacement field

The assembly and operation geometry are partly different on account of the strong interaction between the fluid and structure. Particularly strong are the differences in the blade clearances, the axial distances between the blade rows and the shape of the last blade row. Blade vibrations, which can be resolved down to the individual blades, are also possible. Direct measurement of these variables is therefore advisable **6**. A finite element (FE) analysis is carried out to reconstruct the complete operation geometry.

LPMT, measuring and calibration equipment

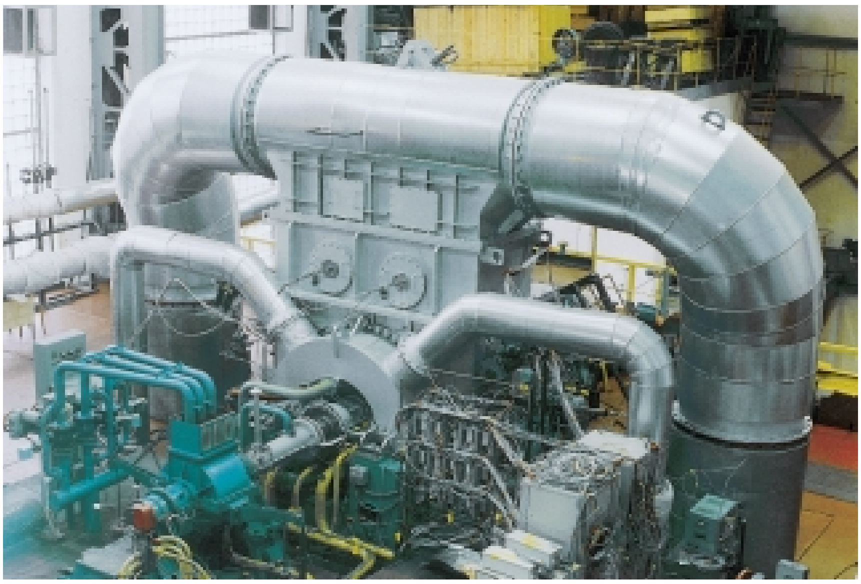
Low-pressure model turbine (LPMT)

The LPMT **8** is designed to accommodate the two representative LP steam paths shown in **3a** and **3b**. The configuration used is based on **3a** and **5**. In the case of the steam path in **3b** , the top and bottom exit flows are symmetrical to avoid fluid asymmetry **5**.

Flow-measuring probes

The head of the 5-hole probe used in control surfaces 42, 52 and 62 has a small angle due to the high Mach numbers. The same probe type is also used in control surfaces 63 and $A0_1$ to $A0_3$, however with a head with a larger angle. 3-hole probes can also be used.

View of the test facility, inlet side **8**



A combined extinction/temperature probe is used to measure the steam wetness, droplet size and temperature. This probe employs the principle by which light is attenuated on its passage through the wet steam [8]. Two colours, red and blue, are used. The method allows droplet sizes from 0.1 to 2 μm as well as the wetness to be measured with an uncertainty of 15 %. Such probes are used in control surfaces 42, 52 and 62.

Since the probes are not capable of time-based resolution, the measured field variables are assumed to be irreversibly time-averaged [9].

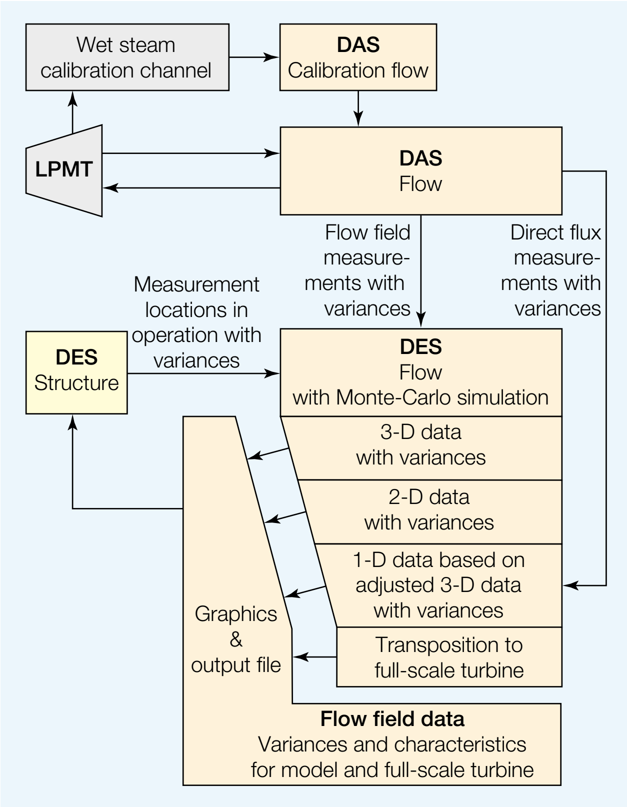
Calibration of flow-measuring probes

The calibration of the 5-hole, 3-hole and combined probes was given a high priority. Calibration in the artificially prepared wet steam was ruled out as this would not allow correct Damköhler numbers to be obtained. Calibration finally took place directly in the wet steam of the model steam path, with a calibration channel integrated in extraction line A1. The channel features a so-called slotted nozzle which can be used continuously up to Mach numbers of approximately 1.6 [10]. Before being used, it was calibrated up to Mach numbers of 1.6. The calibrated probes ensure that the wet steam measurements are of the highest possible accuracy (the measurement represents a complete inversion of the calibration!).

Structure measurement equipment

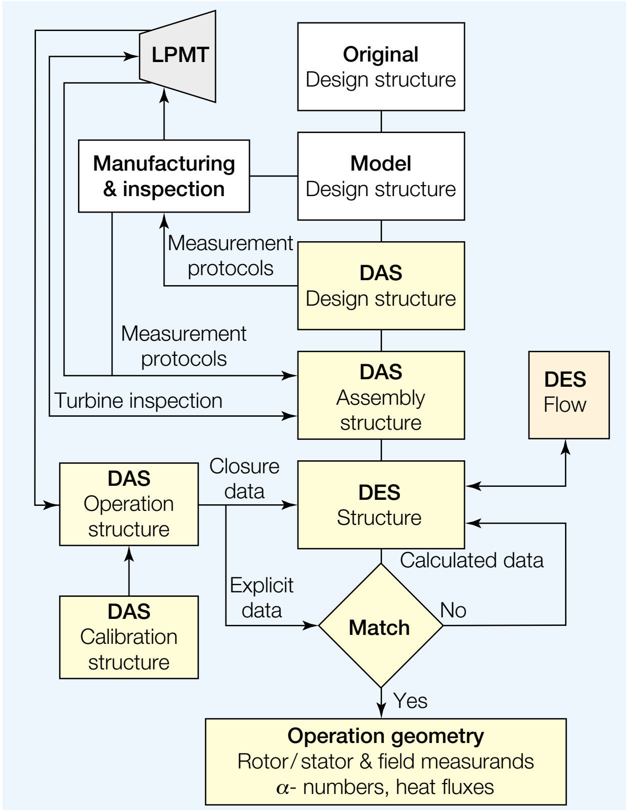
Inductive sensors were fitted for the measurement of the blade clearance in the two last blade rows **6**. To compensate for any possible measuring error, an additional sensor is installed in each case. The same sensors are also used to measure the blade deflection (L-0) and any blade vibration that might be present.

The axial distance between the blade carriers and between the vanes and the



Data processing, flow 9

LPMT *Low-pressure model turbine*
DAS *Data acquisition system*
DES *Data evaluation system*



Data processing, structure 10

LPMT *Low-pressure model turbine*
DAS *Data acquisition system*
DES *Data evaluation system*
α *Heat transfer coefficient*

front and rear rotor is determined with the help of eddy current and inductive sensors, respectively. Temperature measurements were carried out on the three vane carriers to support the FE calculations.

Calibration of structure measurement equipment

The inductive sensors were calibrated on a rotating bladed wheel. Conventional methods were used to calibrate the eddy current sensors and the thermocouples.

Data processing, flow

The data acquisition system DAS Flow 9 receives data from 5-hole and 3-hole probes, combined probes, plus rakes for the total pressure and temperature, wall pressures, pressure and temperature at the inlet mass flow nozzle, torques at the casings of the water brakes and the hydrodynamic bearings, etc. DAS Flow also transmits the data for the probe positioning and the discontinuous purging of the pressure measuring holes [11], and is responsible for evaluating the measured data and converting them into field variables and fluxes with the help of the calibration curves. Account is taken of the variances at all stages.

The data evaluation system DES Flow integrates the field variables in time and space and compares the resulting mass flux and total enthalpy fluxes with the directly measured values. Time-integration ensures that the fluxes and flux densities are correctly averaged (the traditional averaging of the quantities leads to inconsistencies and increases the variances) [9]. Generally, this shows differences that call for adjustment of the field variables: pressure and specific enthalpy. The definitive evaluation of the flow field is carried out on the basis of 3-D, 2-D and 1-D field mean values and characteristics [12].

In the above, account is taken of the fact that the (iterative) evaluations have to be based on the operation geometry, as does the position of the probe heads.

The evaluation is concluded with the performance characteristics of the steam path and its components. Every step in the DES Flow process is accompanied by variances, which are determined by means of a Monte-Carlo simulation.

The determination of the performance characteristics of the original steam path relies on transfer of the flow-governing operation geometry according to the laws of similarity. As already mentioned, the main role here is played by the Reynolds numbers for the front stages as well as the Damköhler numbers for the wet-steam zone. The investigations described in [13] were carried out on an original steam path to allow the method of transference to be checked.

Data processing, structure

The *data acquisition system DAS Design Structure* **10** provides the inspectors who check the manufacture of the core components with data relating to the flow-governing design geometry (including the tolerances) and the design material properties. They give, in a comparable form, the measured flow-governing assembly geometry data (including the variances) as well as the (possibly) measured assembly material properties, and transfer the results to DAS Assembly Structure. Further flow-governing assembly geometry data (eg, assembly blade clearances) are expected from LPMT measurements.

DES Structure is at the center of the structure-oriented evaluation. Its initial task is to determine the flow-governing operation geometry (including the variances) on the basis of an FE analysis of the core components. Also to be determined are the heat transfer coefficients and the heat flows on and over the flow-governing surface. These variables complete the flow field.

The variances in the FE analysis are reduced by using the measured data collected in DAS Operation Structure as well as the relevant functions from DAS Calibration Structure. Also used are the flow field data from DES Flow. The only degree of freedom lies in the heat transfer coefficients, which are adapted iteratively until all the measured data are fulfilled.

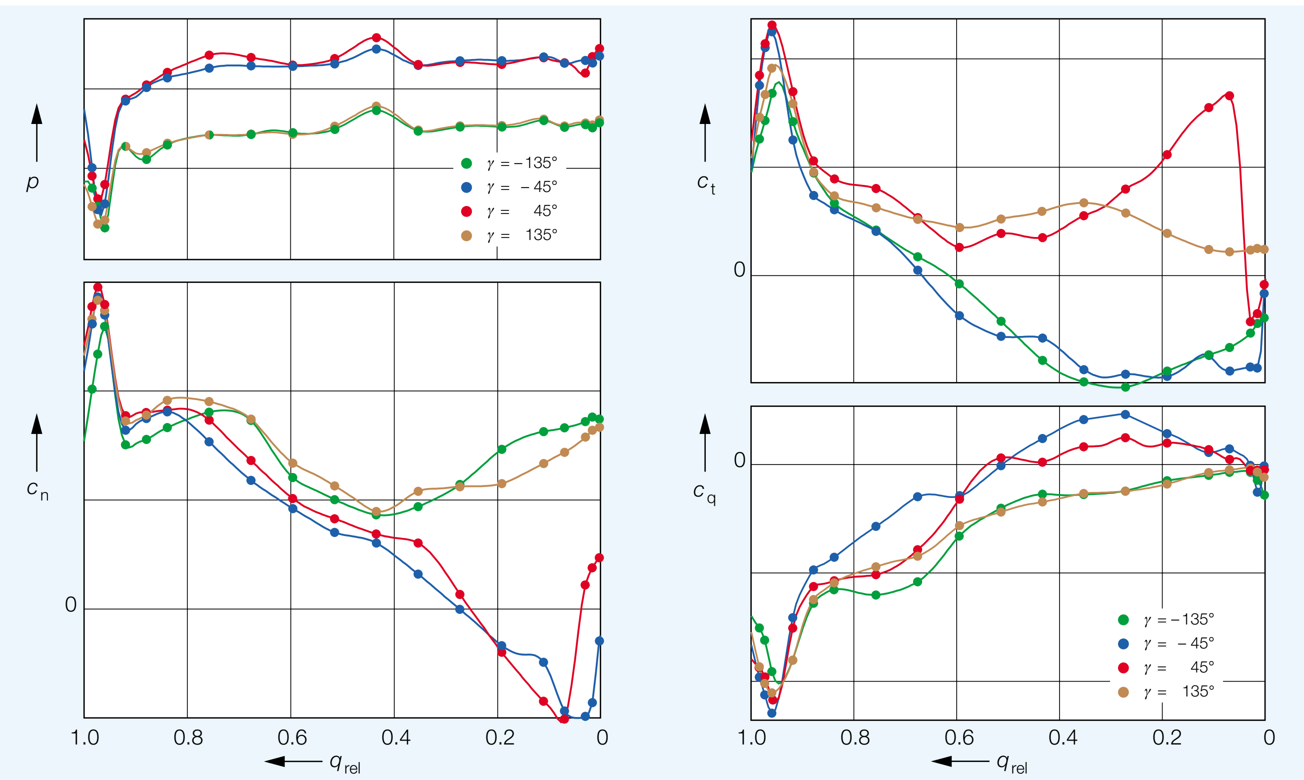
The flow-governing operation geometry is transferred to the original steam path with the help of the scaling factor. The circumferential distribution and the mean blade clearances do not coincide precisely with the operation blade clearances due to certain properties of the LPMT, eg the separated rotors. This problem can be solved by using the data, which are anyway usually available, from the 2nd and 6th steam path. The heat transfer coefficients can be transferred as a function of the Reynolds number.

Data analysis

Using the model steam path in **5** and **6** (this corresponds to the 6th steam path of an ND41 turbine), two examples of the flow field and the displacement field of the steam path are given in the following.

Flow field in control surface 63 (ND41/6th flow) **11**

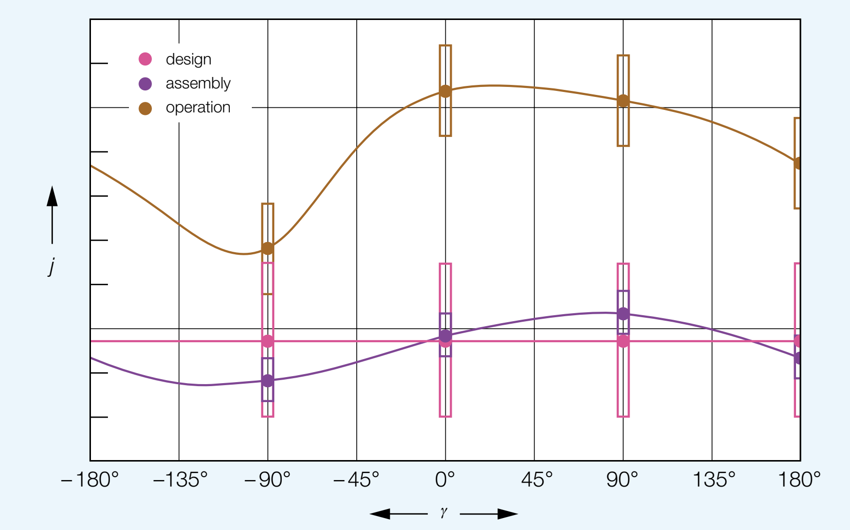
p Pressure
c_n Velocity component, normal
c_t Velocity component, tangential
c_q Velocity component, transverse
q_{rel} Distance from deflector wall relative to channel width
γ Circumferential position



11 shows the flow field in control surface 63 (at the diffusor outlet) for the mean operating conditions in **7**. The static pressure and the velocity components are shown as a function of the dimensionless distance from the deflector wall. The curve parameter is given by the circumferential position of the probes. Position γ = 0 lies opposite the exhaust-steam outlet; viewed in the direction of the flow, γ is positive in terms of the direction of rotation.

The pronounced rotational asymmetry due to the one-sided exhaust steam casing can be clearly seen, as can the jet due to the strong flow through the gaps in the case of large blade clearances.

12 shows the blade clearances at the trailing edge of the last row of blades, averaged over all the blades, as a function of the circumferential position. It can be seen that in the assembly state the rotor is not centered precisely. Moreover, a significant increase in the blade clearances is apparent, mainly as a result of the different axial displacement of the rotor and the blade carrier under operating conditions.



Mean clearance on last blade row (ND41/6th flow) **12**

j *Clearance at trailing edge*

γ *Circumferential position*

Outlook

The described proactive feedback loops for 'product and engineering tools qualification' and 'basic technology & design tools qualification' will lead to a continuous reduction in the differences between the target and expected values as well as in the variances of the LP steam turbine performance. Power plant customers specifying targets for the economic parameters will benefit from these loops in the form of a steady reduction in the probability of nonconformance, plus compliance with ecological needs.

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