Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45672

MODELLING AND VALIDATION OF WET STEAM FLOW IN A LOW PRESSURE STEAM TURBINE

Jörg Starzmann, M. Schatz, M. V. Casey and J. F. Mayer

ITSM - Institute of Thermal Turbomachinery and Machinery Laboratory
Universität Stuttgart, Germany

Frank Sieverding

Siemens AG, Energy Sector, Fossil Power Generation
Mülheim/Ruhr, Germany

ABSTRACT

Results of numerical investigations of the wet steam flow in a three stage low pressure steam turbine test rig are presented. The test rig is a scale model of a modern steam turbine design and provides flow measurements over a range of operating conditions which are used for detailed comparisons with the numerical results.

For the numerical analysis a modern CFD code with user defined models for specific wet steam modelling is used. The effect of different theoretical models for nucleation and droplet growth are examined. It is shown that heterogeneous condensation is highly dependent on steam quality and, in this model turbine with high quality steam, a homogeneous theory appears to be the best choice. The homogeneous theory gives good agreement between the test rig traverse measurements and the numerical results. The differences in the droplet size distribution of the three stage turbine are shown for different loads and modelling assumptions. The different droplet growth models can influence the droplet size by a factor of two. An estimate of the influence of unsteady effects is made by means of an unsteady two-dimensional simulation. The unsteady modelling leads to a shift of nucleation into the next blade row. For the investigated three stage turbine the influence due to wake chopping on the condensation process is weak but to confirm this conclusion further investigations are needed in complete three dimensions and on turbines with more stages.

NOMENCLATURE

c_p	specific heat at constant pressure (J/kg·K)
J	nucleation rate (1/m^3·s)
K	Boltzmann constant (=1.3807·10^{-23} J/kg)
Kn	Knudsen number (-)
L	latent heat (J/kg)
$\bar{l}$	mean free path length (m)
m	mass of a water molecule (kg)
m_{het}	cosinus of the contact angle (-)
Nu	Nusselt number (-)
p	static pressure (Pa)
q_c	condensation coefficient (-)
r	radius (m)
R	gas constant (=461.4 J/kg·K)
S	supersaturation (-)
T	temperature (K)
ΔT	subcooling (K)
x_{het}	ratio of droplet radius (r_{het} / r_{crit})
y	liquid mass fraction (-)

Greek symbols

α	absolute flow angle (°)
α	Young's [11] droplet growth parameter (-)
α_d	heat transfer coefficient (W/K·m^2)
γ	ratio of the specific heats of the vapour (-)
λ	thermal conductivity (W/K·m)
μ	dynamic viscosity (kg/m·s)
ρ	density (kg/m³)
σ	planar surface tension (N/m)
ϕ	Kantrowitz [16] non-isothermal correction (-)

Subscripts

crit	critical condition
d	droplet
g	gas (vapour)
het	heterogeneous
ref	reference
s	saturated

INTRODUCTION

Generally the last stages of low pressure steam turbines operate in a condensing wet steam flow. The problems associated with non-equilibrium condensation in the flow of low pressure steam turbines are efficiency reduction, droplet erosion and corrosion and have been a topic of research since Stodola [1] in 1910. Turbine manufacturers are still strongly interested in modelling the effects of condensation on the flow field and the performance of the turbine, as this remains an area where further improvements in efficiency or reliability can be expected. Here there are two main effects. Firstly the condensation itself is a source of thermodynamic relaxation loss. Secondly, and perhaps more importantly, the condensation process changes the flow field and as this effect is usually not considered in the blade profile design process, additional aerodynamic losses are caused, see Starzmann [2].

A comprehensive theory to describe the flow physics and the influence on turbine behaviour goes back to Gyarmathy [3] and Kirillov [4]. Their findings prepared the foundation for most research up to the present day. In the meantime, several efforts have been undertaken to implement wet steam models in modern 2D or 3D CFD codes. A broad overview of the literature is given in Bakhtar [5] and in the many sources referred to later in the current paper. The models differ in the way the wetness equations are used in the conservation equations, that is if the wetness equations are solved in a Lagrangian or Eulerian frame of reference. Further attributes are whether the flow model takes viscous effects into account, if the wetness dispersion is treated in a mono- or polydispersed (e.g. Gerber [6]) way, which is discussed by White [7], or if unsteady effects can be represented.

In most cases the aim of earlier numerical studies was to enhance knowledge about the complicated physics in rapid expanding and condensing flows and to improve the prospects of flow modelling. More recently, the effect of non-equilibrium condensation on the flow field has been investigated by comparing non-equilibrium and equilibrium three dimensional CFD simulations (Wroblewski et al. [8], Gerber et al. [9]). Starzmann [2] presented a method to determine the thermodynamic relaxation loss from a non-equilibrium solution and showed that this loss not only occurs in the stage where nucleation takes place but also in the stages where the droplets are growing. This paper describes further development and validation of the models described by Gerber et al. [9], where a more detailed description of the models can be found.

In the present study a three stage low pressure steam turbine tested in the turbine test rig at the Institute of Thermal Turbomachinery (ITSM) at the University of Stuttgart was investigated. The test rig is a scale model with a factor of approx. 4 of a modern steam turbine design. Flow measurements have been done by Völker [10] over a range of operating conditions which can be used for detailed comparisons with CFD results.

Steady three dimensional and unsteady two dimensional CFD calculations were performed. At least two additional models are necessary compared to a standard RANS-solver to model non-equilibrium condensing flows; a nucleation model to determine the droplet formation and a droplet-growth model must be introduced. These additional models are linked to the mass, momentum and energy conservation equations via source terms. In the present paper different models for these aspects are examined and the limitations in heterogeneous nucleation modelling are discussed. Regarding the droplet growth modelling the method of Gyarmathy [3] is compared with the extended model from Young [11], both of which are widely used.

NUMERICAL MODELLING

The commercial solver ANSYS CFX 12.1 was used in the present study. The real gas handling is based on the IAPWS-97 standard which includes the extrapolation into the metastable steam region. The implemented non-equilibrium steam (NES) model accounts for subcooling and condensation effects. Earlier published work [9], [2] from the same research group has already presented the capacity of the non-equilibrium CFD models used here. A special feature of the model is that the user is able to define several liquid phases and allow each liquid phase to nucleate for each of the simulation domains separately. This retains the information about where a certain droplet group first appears and how this class of droplets grow during their flight through the turbine. Four liquid phases (P) were used (see Fig. 1), to model homogeneous nucleation and for heterogeneous nucleation modelling an additional phase (P5) was introduced. For example, in the first three blade rows homogeneous nucleation is only allowed for the first liquid phase P1, downstream of the second stator S2 these droplets of the phase P1 are only allowed to grow in size. All newly formed droplets in the rotor R2 belong to the second liquid-phase P2 and these droplets are also only allowed to grow in the subsequent blade rows.

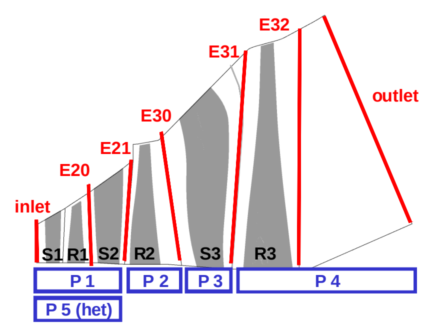

Fig. 1: Sketch of the investigated model steam turbine

For the present approach it was assumed that the droplets are moving with the vapour flow field. According to Gyarmathy [3] this slip can be neglected for small droplets below 1 µm. The droplets which are formed by spontaneous condensation are even smaller and coarse water formation was not considered in this study. The comparison between the mechanical relaxation time (Moore [12]) of a droplet in the flow equates to 3.9 µs to 8.1 µs (depending on the chosen streamline) and is much smaller than the 500 µs, which is the time the flow needs for passing through the third rotor blade row R3. This shows clearly that such small droplets accelerate up to the steam velocity very fast and thus the slip can be neglected. A three dimensional CFD calculation, in which additional momentum equations for the liquid phase are solved, but not reported here, confirmed the analytical findings above. Three different levels of grid refinement (Tab. 1), all with similar O-grids around the blades were generated for which common quality requirements [13] were considered. The y+-values given in the table were averaged over all wetted surfaces. The radial clearances between the casing and rotor blades were considered in the grid generation and for turbulence modelling the SST-model was used. For the steady flow simulations mixing planes are necessary between the rotating and stationary parts of the turbine. The last domain includes the last rotor R3 and the modelled part of the diffusor (Fig. 1), and this ensures that the trailing edge flow of R3 is not mixed out due to a mixing plane.

grid	coarse	medium	fine
no. of elements [mio]	0.65	1.6	5.2
mean y+-values	17	10	5

Tab. 1: Grid size and y+-values

For the grid study the classical homogeneous nucleation model given in eq. (1) and the original droplet growth model from Gyarmathy as stated in the corresponding section were used. The grid independency study was conducted for the design load case and the results are included in the related diagrams in Fig. 9 and Fig. 10. The circumferential averaged flow variables after the second rotor R2 in plane E30 shows negligible differences between the three grids, the same can be stated for the evaluation plane E32 downstream of the last rotor. For the wetness related variables such as the wetness fraction and the droplet diameter the differences are more pronounced but also small. In Fig. 2 the nucleation rate and the subcooling are shown along a mid-passage streamline through stator S2 at approx. 50 % span. The nucleation and resulting rapid condensation process takes place mainly in this blade row and actually in this sensitive flow region the grid resolution is sufficient. The differences in the predicted power output of the three stage turbine is 1.2 % between the coarse and the medium grid and 0.09 % between the medium and the fine grid. It can be concluded that for real grid independence a much finer grid is needed but the changes in the flow field are very small. All the further results presented are obtained with the fine grid.

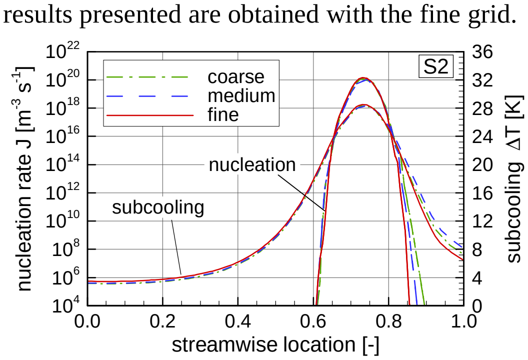

Fig. 2: Nucleation and subcooling on a mid-passage streamline in stator S2 at different levels of grid refinement.

NUCLEATION MODEL

In low pressure steam turbines water droplets are in general not formed at saturated conditions. To overcome the energy barrier, generated due to the surface tension of newly built droplets, subcooled steam conditions are needed before a spontaneous condensation brings the steam back to nearly equilibrium conditions. The non-equilibrium condensation influences the flow field of steam turbines and induces a thermodynamic relaxation loss e.g. Starzmann [2] has shown.

For a pure steam without foreign nuclei a homogeneous nucleation takes place. The classical nucleation model described by McDonald [14] or Bakhtar et al. [15], is commonly used to describe this process, as follows

$$J = \frac{q_c}{1+\phi}\sqrt{\frac{2\sigma}{\pi m^3}}\,\frac{\rho_g^2}{\rho_d}\cdot\exp\left(-\frac{4\pi\sigma r_{crit}^2}{3KT_g}\right) \qquad (1)$$

In this equation q_c is the condensation coefficient (generally $q_c = 1$) and ϕ equates to the non-isothermal correction by Kantrowitz [16]. Further parameters are m, which signifies the mass of a single water molecule and K, which is the Boltzmann constant. This model is used by many authors and in the present work it has been implemented as a user defined model in ANSYS CFX 12.1. The implementation in this way allows parameter studies based on different model assumptions and coefficients. (It is interesting to note that this procedure also identified an incorrect implementation in ANSYS CFX 12.1 and earlier versions).

For steam containing impurities, heterogeneous condensation, i.e. a phase change on existing nuclei, can also have a considerable effect on the condensation process and the resulting two-phase flow throughout the turbine. The section below discusses the modelling procedures for heterogeneous condensation as the modelling strategy for this is less clear and different models have been examined here. Considerable research on heterogeneous nucleation was initialized by the "Electric Power Research Institute" EPRI at the end of the last century [17], [18]. Within this framework impurity concentrations were measured in 21 power plant units and tests in nozzles and turbines were made to investigate the influence of different impurities and concentrations on the condensation process. The nozzle tests of Petr and Kolovratnik, which are reported in [17], were made at expansion rates of 1000 1/s and 4500 1/s and the results show that the role of heterogeneous nucleation is more important for low expansion rates. But even for low expansion rates the droplet size decreased only by approx. 8 % for high concentrations of ammonia. Due to the injection of NaCl the Sauter averaged droplet size increased by up to 15 % if using a high concentration of about 150 ppb. However, a common limit of NaCl in the feedwater of steam power plants is of about 5 ppb [19]. Similar small effects due to different concentrations of impurities have been observed in a model steam turbine located in the Moscow Power Institute [17]. Bakhtar could not find any influence on the measured blade pressure profile in tests in a 2D cascade with injection of ammonia [20]. Without doubt, heterogeneous nucleation can occur in low pressure steam turbines but its significance is small and depends on having low expansion rates and poor steam quality.

The modelling of heterogeneous nucleation is currently unsatisfactory, as either the models are strongly simplified or too complex to be used due to the lack of physical knowledge of the behaviour of the impurities in the steam flow [21]. For example one open question relates to the solubility of NaCl or NaOH near the saturation line under the fast changing flow conditions in a low pressure steam turbine [19]. Several researchers have concluded that if detailed knowledge about the process of heterogeneous nucleation is of interest, then further work is required [21], [22].

For the investigated ITSM model turbine the steam is provided by the combined heat and power plant of the University of Stuttgart. The water chemistry of the steam cycle is controlled by ammonia to ensure a pH-value of 8.3 to 9.3 and desalinated water is used. The conductivity of the feedwater is less than 0.1 µS/cm. Silicone oxide (SiO_2) is an insoluble substance and is the only known significant contamination in the steam with a concentration of approximately 10 mg/l. In the present investigation different models were used to obtain an estimate about the occurrence and significance of heterogeneous nucleation in the turbine test rig.

Despite the use of very pure water a certain NaCl contamination is possible. According to Stastny [23] nucleation on NaCl clusters already starts before reaching the saturation line in the so-called salt solution zone, where NaCl becomes soluble in steam. Under these assumptions small droplets persist at saturated conditions and the number of droplets depends on the concentration of NaCl. In this implementation of the Stastny model such droplets are given as a boundary condition and when saturated conditions are reached they are allowed to grow in size. In the present study values of NaCl concentration are obtained from Wroblewski et al. [8] and from Petr and Kolovratnik [21]. For a concentration of 2 ppb, a diameter of 4 nm (clusters of about 100 molecules) and thus $2{\cdot}10^{14}$ droplets per kg steam at the saturated conditions are assumed. The results show that with this model the condensation on existing particles cannot suppress homogeneous nucleation. The modelling leads to an extended homogeneous nucleation in the stator S2 because the maximum value of the homogeneous modelling cannot be reached. As a consequence the subcooling at the outlet of the S2 domain remains higher than for purely homogeneous nucleation. This is shown along a mid-passage streamline at 50 % span in Fig. 3. The resulting influence on wetness fraction, droplet diameter and droplet number is small, as the circumferential averaged distributions in plane E21 shows, see Fig. 4.

For the influence of insoluble particles such as SiO_2 a similar examination can be realized, see also Wroblewski et al. [8]. For the present study droplets with the size of solid particles are assumed as a boundary condition. It is assumed that condensation on the surface of existing particles is possible if saturated conditions are reached. The energy barrier is reduced due to the provided surface for condensation but not removed completely. That is why the assumption that condensation occurs already at saturated conditions is strongly simplified. According to the EPRI report [17] the mean size of solid impurities was estimated at 0.1 µm and therefore a number of 10^{13} nuclei exists at the saturation line. Similar results as for the soluble substance NaCl are obtained here (see Fig. 3 and Fig. 4). For both cases it can be concluded that the total mass fraction of the heterogeneous formed liquid phase is too small to influence the flow field significantly. However, if the wetness dispersion is evaluated by a volume weighted droplet spectrum the additional heterogeneous formed droplets are visible, as the diameter chart in Fig. 4 shows.

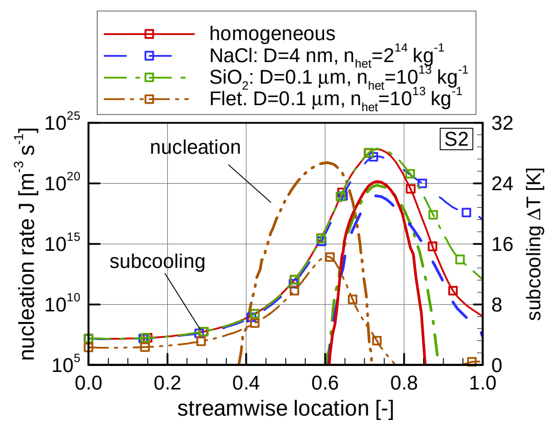

Fig. 3: Nucleation and subcooling for various nucleation models in stator S2.

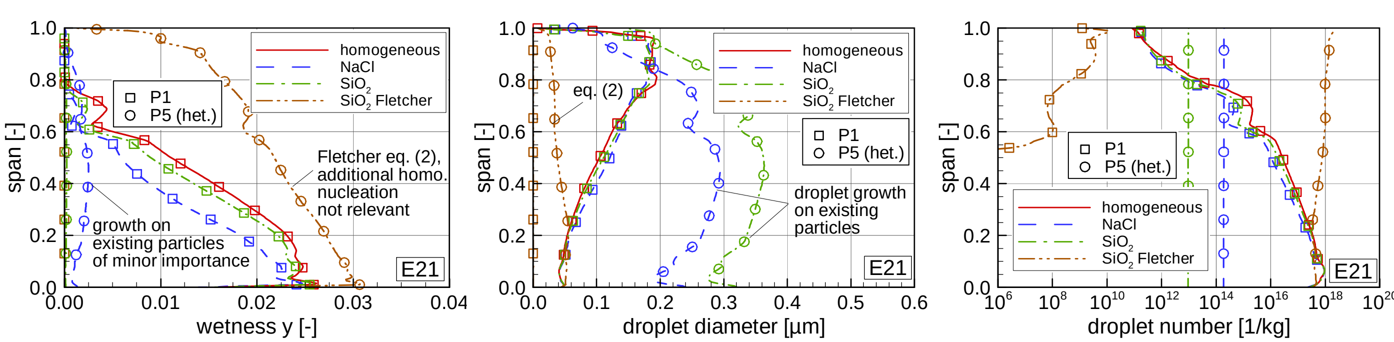

Fig. 4: Homogeneous (P1 □) and heterogeneous (P5 o) modelled wetness dispersion after the blade row with first condensation

If the heterogeneous nucleation on insoluble surfaces is modelled by the classical model of Fletcher [24] a notable difference in the wetness dispersion was found. This model uses a nucleation rate definition, in which the energy barrier is reduced based on a geometric relation $f(m_{\mathrm{het}},x_{\mathrm{het}})$, which accounts for the formation of a cap of water on the insoluble core. For the implementation the equations given by Gerber [6] were used, as follows

$$J_{\mathrm{het}} = 10^{21} \cdot 4\pi r_{\mathrm{het}}\, n_{\mathrm{het}} \cdot \exp\left(-\frac{4\pi\sigma r_{\mathrm{crit}}^{2}}{3KT_{\mathrm{g}}} \cdot f(m_{\mathrm{het}}, x_{\mathrm{het}}) \right) \quad (2)$$

The contact angle between the liquid cap and the particle was assumed to be 40° ($m_{\mathrm{het}} = \cos 40°$) and x_{het} means the relation between the radius r_{het} and r_{crit}. In this case the nucleation starts earlier (Fig. 3) which leads to smaller droplets (Fig. 4) which again leads to a more equilibrated steam flow. From the physical point of view it is not comprehensible that such small droplets are meaningful where particles with about 0.1 µm still exist, on which condensation could occur.

Nucleation modelling with the Fletcher model also gives an influence on the flow field. Due to the lower energy barrier, compared to the homogeneous theory, nucleation starts at lower subcooling. Thus, effect on the flow field (such as changes in pressure or flow angle) is smaller than with homogenous nucleation and closer to an equilibrium model. The maximal difference between the calculation with the Fletcher model and the other models can be found after nucleation in plane E21, where a difference in flow angle of 5° exists. The overall influence on performance was determined to be 0.01 %-points in efficiency and can be neglected.

A further aspect which should be considered is that it must be expected that the test cases which are available to validate the homogeneous model also operate with steam of different purity. Unfortunately in most test cases nothing is reported about the steam quality. In addition to the earlier work of Young [11], the most valuable work regarding the validation of the homogeneous nucleation model comes from Wroblewski et al. [25] in which extensive calculations of most existing test cases are published. In this work it is concluded that the uncertainty of measurement data makes a validation of the models difficult a change of 1 K in inlet conditions can already has a significant influence on the nucleation process.

An example of validation of the classical homogeneous model used here is given in Fig. 5. This diagram shows the calculated pressure distribution of the Bakhtar rotor cascade [26] with an outlet wetness of 5 % and a pressure ratio of 2.34. The location of the pressure hump at an axial chord of 0.4 (related to the condensation) agrees well with the experiments and suggests that the model predicts the condensation correctly.

At this point it is worthwhile focussing on the results with regard to the droplet formation models. The comparison between the different heterogeneous models demonstrates that the application of heterogeneous theories in turbine flows is poorly conceived. The modelling process lacks clarity, with regard to size and number of particles, as well as the condensation factor

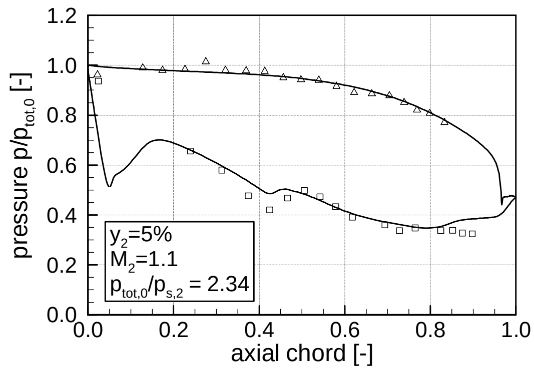

Fig. 5: Predicted pressure profile for the Bakhtar cascade [26]

used in the nucleation model given in eq. (2), and in the contact angle between the liquid cap and the particle. Above all a validation of the process is completely missing in wet steam flows. On the basis of this, and in agreement with the results of Petr and Kolovratnik [21] which also attest the insufficiency of heterogeneous nucleation models, currently the classical homogeneous nucleation theory must be regarded as the most appropriate model for condensing steam flows in low pressure steam turbines. This is especially valid for the scaled turbine investigated here with high expansion rates between 8000 1/s and 12000 1/s in the Wilson zones, and steam of high quality.

DROPLET GROWTH MODEL

Several droplet growth models exist with different complexity. It is not the scope of the present paper to discuss the complexity of the droplet growth in detail, for which the work of Lamanna [27] can be referred to. From Lamanna it can be concluded that simple models are able to predict the droplet growth with sufficient precision. Above all, if the accuracy of droplet size measurements is taken into account. The most established models for droplet growth in wet steam flows are the original model from Gyarmathy [3] and an extended model from Young [11]. Both models are well-known and easy to implement in CFD codes. One objective of this paper is to compare these models by means of the flow conditions in the low pressure steam turbine investigated here.

The droplet growth process mainly depends on the heat transfer between the droplet and the steam. This is also the reason why it can be expected that condensation on solid surfaces (such as the casing and blades) is of minor importance. The heat transport potential is simply not high enough, because on one hand the casing of the turbine is not cooled from the exterior and on the other hand the solid surfaces are even heated due to the friction in the boundary layer.

The origin for each droplet growth model is the energy balance around a single droplet. Assuming that the droplet has a perfect spherical shape and that the internal droplet heat transfer can be neglected compared to the heat transfer across the droplet surface, the following equation can be derived, [28].

$$\mathrm{L}\rho_{\mathrm{d}} \frac{\mathrm{d}r}{\mathrm{d}t} = \alpha_{\mathrm{d}} \cdot (T_{\mathrm{d}} - T_{\mathrm{g}}) \quad (3)$$

This shows that the droplet growth rate depends on the latent heat L, the density of the droplet ρ_d, the temperature difference between the surface of the droplet and the gaseous steam and the heat transfer coefficient α_d. According to Gyarmathy [3] the droplet surface temperature for small droplets (< 1 µm) can be calculated by the subcooling and the critical radius.

$$T_d - T_g = \left(T_s(p) - T_g\right)\left(1 - \frac{r_{crit}}{r}\right) \tag{4}$$

If the Nusselt number ($Nu = \alpha_d \cdot 2r / \lambda_g$) is introduced and the temperature difference between the droplet surface and the vapour is replaced by eq. (4) the following equation is obtained.

$$L\rho_d \frac{dr}{dt} = \frac{Nu \cdot \lambda_g}{2r}(T_s(p) - T_g) \cdot \left(1 - \frac{r_{crit}}{r}\right) \tag{5}$$

The heat transfer coefficient resp. the Nusselt number depends on the local flow conditions and the flow regime. The molecular clusters formed during nucleation and also the spontaneous formed droplets are of comparable magnitude to the mean free path length $\bar{l}$ of the steam (typical values are in the order $r = 10^{-8}$ m). This leads to Knudsen numbers ($Kn = \bar{l}/2r$) in the range of 1 and higher, therefore a flow regime between a continuum flow and a free molecular flow exits. Gyarmathy [3] solved this problem by a universal equation, which can be used in this transition regime and thus over a wide range of Knudsen numbers given by

$$Nu = \frac{2}{1 + c \cdot Kn} \tag{6}$$

In the original model from Gyarmathy [3] the constant c is derived to have a value of 3.18. Although Young [11] has subsequently derived a value of 3.98 for the constant c, the original model of Gyarmathy is still widely used, e.g. by Wroblewski et al. [8] or Stastny [29].

Young presented in [11] an extended model based on the formulation of Gyarmathy, by the following two equations.

$$Nu = \frac{2}{1 + 3.78(1-\nu)Kn/Pr} \tag{7}$$

$$\nu = \frac{RT_s(p)}{L}\left(\alpha - 0.5 - \frac{2 - q_c}{2q_c}\left(\frac{\gamma + 1}{2\gamma}\right)\left(\frac{c_p T_s(p)}{L}\right)\right) \tag{8}$$

The factor $(1-\nu)$ was introduced to get a better agreement with steam nozzles experiments for low Wilson zone pressures. The Prandtl number is calculated by $Pr = \mu_d c_p / \lambda_d$, whereas μ_d is the dynamic viscosity of the steam. The coefficient α in eq. (8) is an empirical constant and for the present investigation a value of zero (White and Young [30]) and a value of 9 (Young [11]) is used. More physical details about this model are given by Young in [11].

Finally, the combination of equations (5) and (6) builds the original Gyarmathy model (Gy.) and equations (5), (7) and (8) give the droplet growth model of Young. Both models were implemented by user expressions in ANSYS CFX. The default CFX implementation uses the original Gyarmathy model, which was first rebuilt by a user defined implementation to check consistence.

The differences between the models are shown by calculation of a condensing flow through different nozzles tested by Moore [31]. It was found that the quality of the results depends strongly on the grid solution and for the droplet diameter up to a size of 60,000 elements no grid independency can be obtained. Furthermore, the results are obtained without any calibration e.g. a temperature fit as done by Gerber [6], which influence the results considerably. Tab. 2 shows the agreement between the predicted and the measured droplet sizes at the outlet of the investigated nozzle.

droplet diameter [µm]	**Exp. [31]**	**Gy.**	**Young α=0**	**Young α=9**
nozzle A	0.050	0.026	0.025	0.036
nozzle B	0.100	0.070	0.068	0.077
nozzle C	0.150	0.065	0.057	0.132
nozzle D	0.140	0.103	0.090	0.210

Tab. 2: Droplet size prediction for the Moore nozzles [31]

The droplet size predicted with the original Gyarmathy model (Gy.) is somewhat too low. This result agrees with the comprehensive work of Wroblewski et al. [25], where the original Gyarmathy model is used and the results shows that the droplet size is predicted to be smaller when compared to the experimental data given in the literature for low pressures. This shortcoming of the Gyarmathy model, which underpredicts the droplet growth rate in the region of small Knudsen number (free molecular regime), is attributed by Lamanna [27] to an inaccuracy in the droplet temperature estimation. This should be improved by the Young model. Note that the determination of the mean free path length $\bar{l}$ in CFX 12.1 is given by

$$\bar{l}_{CFX} = 3\sqrt{\frac{\pi}{8}} \cdot \frac{\mu_d\sqrt{RT_g}}{p} \;,\quad \bar{l}_{Gy} = 1.5 \cdot \frac{\mu_d\sqrt{RT_g}}{p} \tag{9}$$

This differs slightly from the formula which is used by Gyarmathy [3] but tests have shown that the influence is small.

The calculations of the three stage model steam turbine with different droplet models are performed using the homogeneous nucleation theory and for the design load case for which nucleation is predicted at 50 % span in stator S2 (Fig. 6). For the original Gyarmathy model and the Young model with $\alpha = 0$ the initial droplet diameter rises in S2 to a value of 0.1 µm in plane E21. The higher droplet growth rate in the free molecular regime for the Young model with $\alpha = 9$ leads to a 50 % higher droplet diameter compared to the Gyarmathy model. The circumferentially averaged droplet sizes at a channel height of 50 % are shown in Tab. 3. At the stage outlet plane E32 there is a 54 % difference between the Young model ($\alpha = 9$) and the Gyarmathy droplet model. The distribution of the diameter over the span height is shown in Fig. 7 and Fig. 8 in the context of the following section.

Location	Droplet diameter [µm]		
	Gyarmathy	Young (α=0)	Young (α=9)
E21 (50 % span)	0.105	0.096	0.159
E30 (50 % span)	0.112	0.104	0.176
E32 (50 % span)	0.158	0.146	0.244

Tab. 3: Predicted droplet sizes for the ITSM steam turbine

In summary, the influence of the different droplet growth models on the overall wetness generation and thus on the thermodynamic relaxation of the steam flow is small and only of local importance. The simulation with the original Gyarmathy model leads to a wetness of 1.1 % whereas with the Young model ($\alpha = 9$) a wetness of 1.36 % is reached in plane E21 at 50 % span. This leads to a difference in subcooling between the calculations of 3 K, which is not enough to cause a noticeable difference on the pressure or the flow angle distribution between the droplet growth models.

FLOW FIELD OF THE THREE STAGE TURBINE

The results of the previous section have shown that the classical homogenous condensation model is superior to heterogeneous condensation models and that the differences in the droplet growth models are too weak to influence the flow field significantly. For this reason, the results discussed below were obtained with the homogeneous nucleation model and the original droplet growth of Gyarmathy, expect for diagrams with a special legend.

The first part of this section shows how the position of the Wilson zone varies for different load cases. Depending on the load case the nucleation is shifted between the stator S2 and the rotor R2 of the second stage for the turbine investigated here. Because of the radial pressure distribution which gives a lower pressure close to the hub the nucleation starts earliest in the hub region and later towards the casing. The nucleation rates for three different loads at the selected blade heights at 10 %, 50 % and 90 % span are shown in Fig. 6. For the part load case a significant part of nucleation takes place in rotor R2. For the design load case nucleation takes place almost over the whole blade height of the stator, except from a span height of 90 % up to the shroud as Fig. 6 shows. Near the shroud the nucleation is shifted into the rotor. For the over load case the whole nucleation process takes place in the stator. Any significant further nucleation in the third stage could not be predicted. Even in flow region with high expansion rates, condensation on existing droplets holds the subcooling level at moderate values.

The initial droplet size depends on the expansion rate during the nucleation process. In the part load case the droplets which nucleate in the stator (containing to phase P1) are bigger

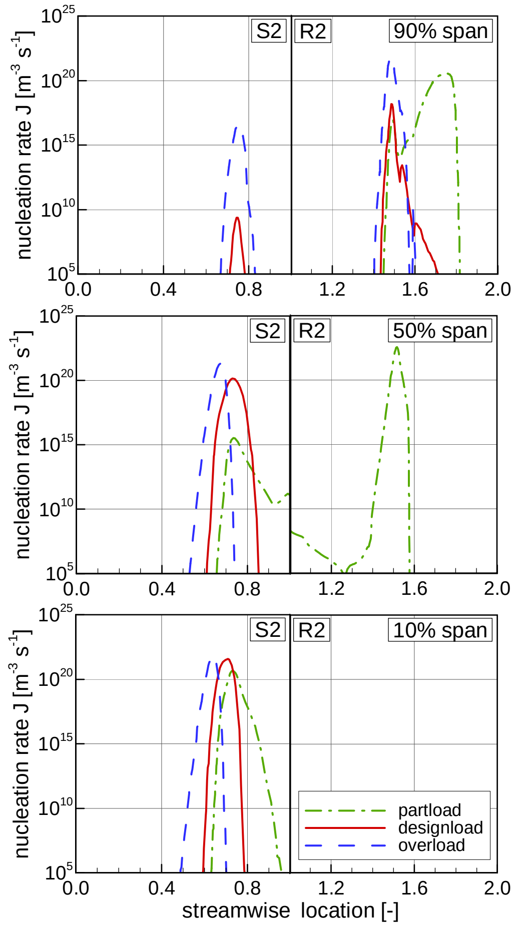

Fig. 6: Homogeneous nucleation along mid-streamlines in stator S2 and rotor R2

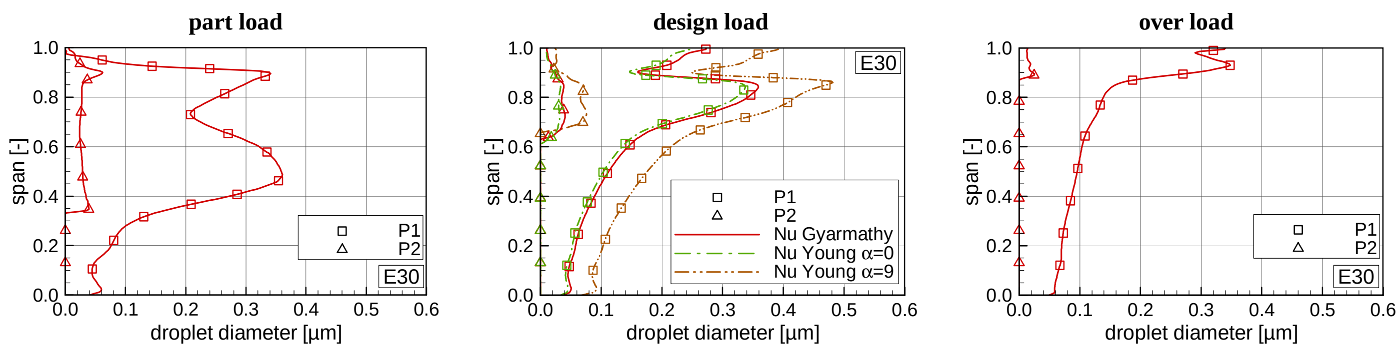

Fig. 7: Droplet diameter in plane E30 for different loads and droplet models

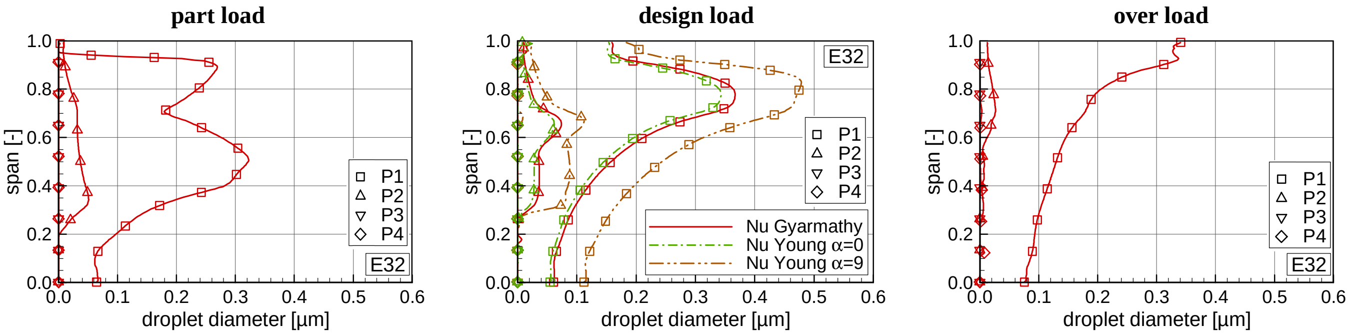

Fig. 8: Droplet diameter in plane E32 for different loads and droplet models

than in the other load cases as can be seen in Fig. 7 by means of the circumferential averaged droplet diameter in plane E30 after the rotor. For this part load case the nucleation starts more downstream of the stator and thus the droplet formation is extended in the unbladed space between stator and rotor with low expansion rates. The droplets formed in the rotor R2 are represented by the second liquid phase (P2) in the CFD solution and are much smaller. The growth of the droplets is weak, as is shown by the distributions of the droplet diameter in plane E32, after the third stage of the turbine, in Fig. 8.

The non-equilibrium phase change influences the flow field of the first and the second stage of the steam turbine, as was already pointed out by a comparison between equilibrium and non-equilibrium simulations for the present turbine geometry in Starzmann [2]. The available flow field measurements in the last stage can be used to validate the CFD-solution. In Fig. 9 and Fig. 10 the pressures and the absolute flow angles obtained from experiments and CFD-simulations are compared. The traverse measurements are conducted with pneumatic four-hole probes; in addition static wall pressure

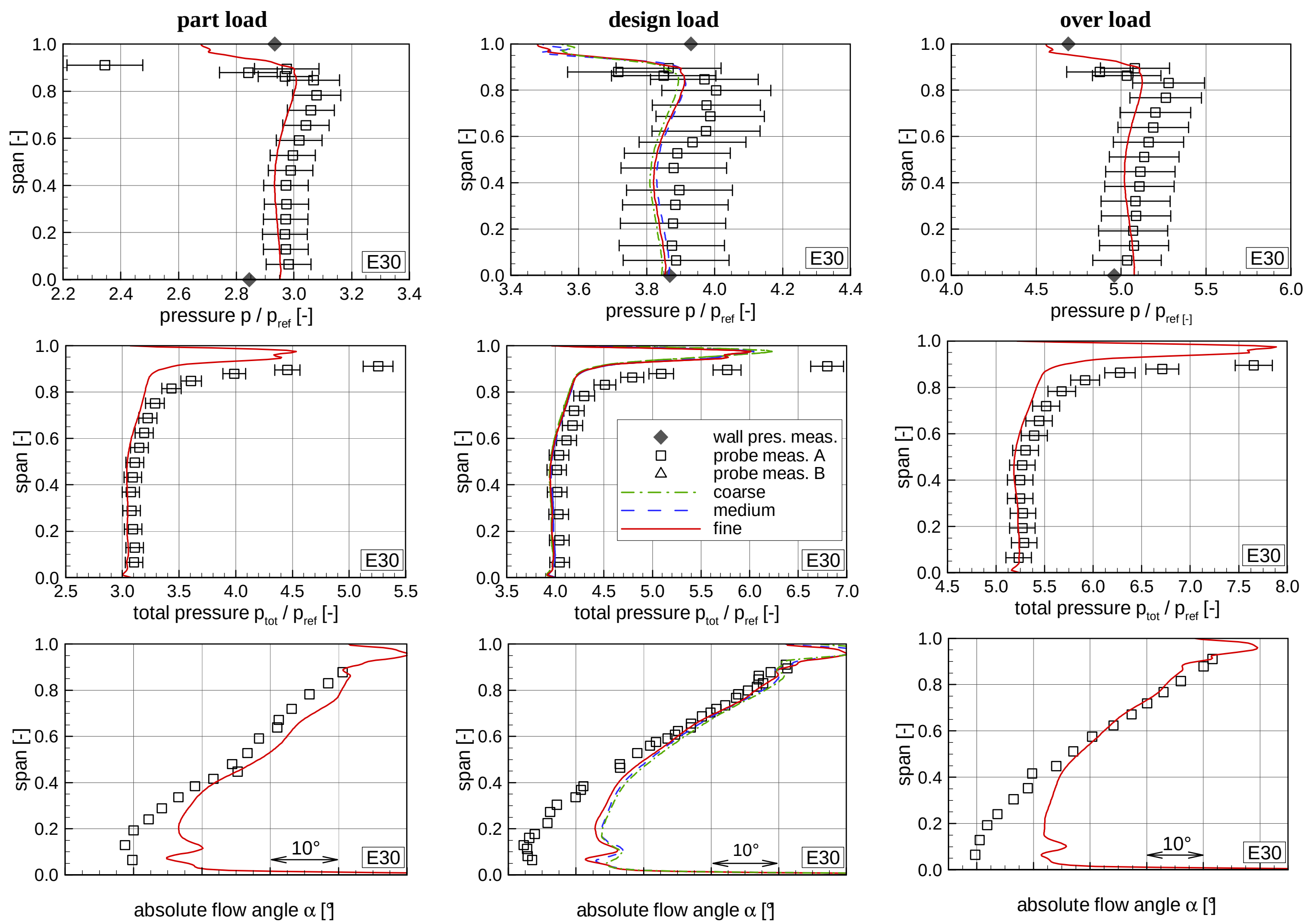

Fig. 9: Predicted and measured flow variables in plane E30 for different loads

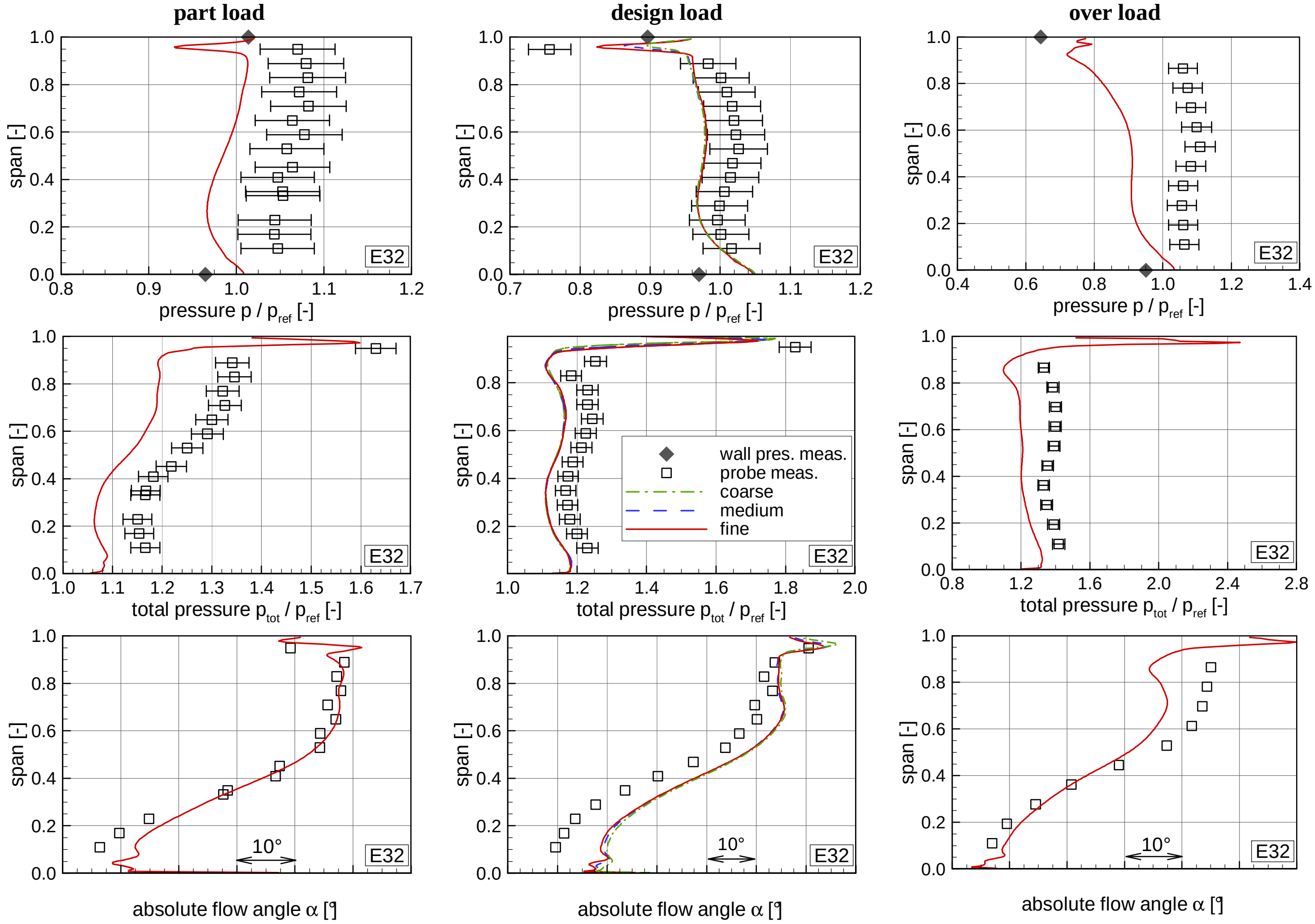

Fig. 10: Predicted and measured flow variables in plane E32 for different loads

measurements are available which represent circumferentially averaged data. The agreement between the measurement and the simulation in plane E30 as well as in plane E31, which is not shown here, is satisfactory for all three load cases. The existing deviations in the flow angle for the hub region of E30 can possibly be attributed to the leakage flow through the cavity related to the following stator S3 which is not modelled in the simulations. Also it might be possible that the probe leads to a significant blockage of the flow in this narrow channel near the hub, as discussed in Völker [10]. The flow field downstream of the last rotor (plane E32) is already influenced by the asymmetric flow in the radial-axial diffusor. The strong deviation of the flow and the blocking due to the exhaust hood leads to higher pressures in the upper part of the diffusor where the probe measurements were made. This effect cannot be modelled by the present numerical analysis, because the model only considers one single blade pitch with a shortened diffusor (Fig. 1) and hence it is assumed that the flow is completely axis-symmetric. Due to this simplified modelling the predicted pressures levels are too low compared to the traverse measurements from the upper part of the diffusor. Especially for the over load case with high mass flows and high dynamic pressure at the outlet the asymmetric effect is considerable. However with respect to the circumferentially averaged wall pressure taps a better agreement can be achieved. Regarding the described conditions in plane E32 the matching of the predicted and the measured flow angles is good. The discrepancy between experiment and simulation of the turbine power output is less than 5 % for all of the three load conditions examined.

UNSTEADY RESULTS

The flow field in a turbine and thus the condensation is influenced by the unsteady rotor-stator interaction. The temperature and pressure fluctuations due to wake chopping were examined by Gyarmathy and Sprengler [32]. Examinations based on their results have shown that from these fluctuations a more polydispersed droplet size distribution can be expected [22]. There are a few further investigations on wake chopping which are summarized by Bakhtar and Heaton in [22].

In the present study a 2D unsteady simulation for the design load case of the three stage turbine was performed as a first study of these effects with the computational models used here. Two meridional streamlines (at 49 % and 51 % span) from the 3D solution provide the upper and lower boundary of the streamtube used for the 2D simulation. A free slip condition was used as a boundary condition on the edges of this streamtube. In contrast to the steady 2D and 3D results using a mixing plane interface, a sliding interface (transient rotor-stator) was used for the unsteady 2D simulation. Circumferential periodicity was assumed which requires a calculation of multiple numbers of blade pitches to ensure the same pitch is modelled for each blade row. Each single blade pitch was resolved by 4500 to 7500 elements depending on the blade row and 11000 elements were used for the last rotor combined with the shortened diffuser. This results in an overall grid size of 430k elements. This grid resolution is comparable with the medium grid from the three-dimensional steady calculation. To reach a solution which is independent of the timestep a very small time discretisation was necessary (1600 steps for one pitch transition). Also approximately eight pitch transitions have to be simulated to reach a converged oscillation solution for the wetness variables. Due to both facts the simulation has taken approximately 40 days with a cluster of 30 CPU's for this 2D simulation.

The influence of the leakage flow and the secondary flow cannot be modelled in the 2D simulation. Due to these effects the expansion characteristic is changed. Above all at a span height of 50 %, the steady 2D simulation cannot be compared directly with the steady 3D result which includes these effects. In the steady 2D simulation the main nucleation occurs slightly further downstream in the second stator S2 than in the 3D steady simulation.

Because of the differences between the 2D and 3D simulation, the unsteady 2D calculation with transient rotor interaction is compared with the steady 2D simulation with mixing planes. The unsteady modelling leads to a shift of nucleation from the stator S2, where nucleation takes place for steady simulation, into the downstream rotor R2. The nucleation rate of the unsteady simulation is shown for one pitch alternation in Fig. 11. Due to the higher expansion rates in the rotor the droplets formed in the rotor in the unsteady case are approximately half the size of the droplets formed in the stator in the steady simulation. This changed wetness dispersion has only a local effect on the flow field of the second stage.

Unsteady modeling includes the effect of wake chopping, which leads to pressure and temperature fluctuations and this unsteadiness in itself should also influence the condensation process. In Fig. 12 the time dependent pressures and temperatures are shown for different monitor points in the second stage, where nucleation takes place. In the diagrams the time positions given by the marked (dots) are the points where the pictures of Fig. 11 are extracted. The temperature fluctuation shown in Fig. 12 reaches 1 K, which appears low compared to estimates by Gyarmathy and Sprengler [32], perhaps because there is only one upstream stage. As can be expected, these fluctuations result in an oscillation of the nucleation zone which in principle can be seen in the pictures of Fig. 11. It is considered that the temperature fluctuation causes only a weak oscillation of the nucleation zone because in this case nucleation occurs in the rotor under very high expansion rates.

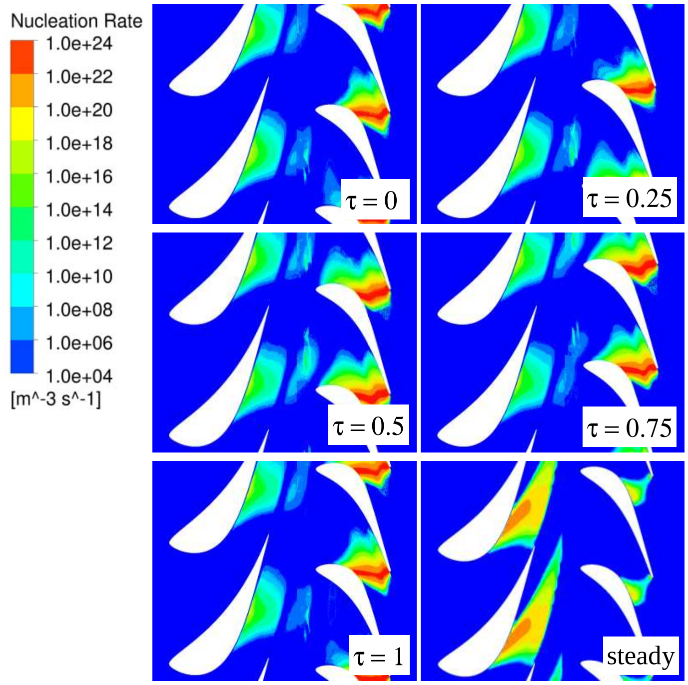

Fig. 11: Nucleation rate for the steady and unsteady 2D simulation in S2 and R2

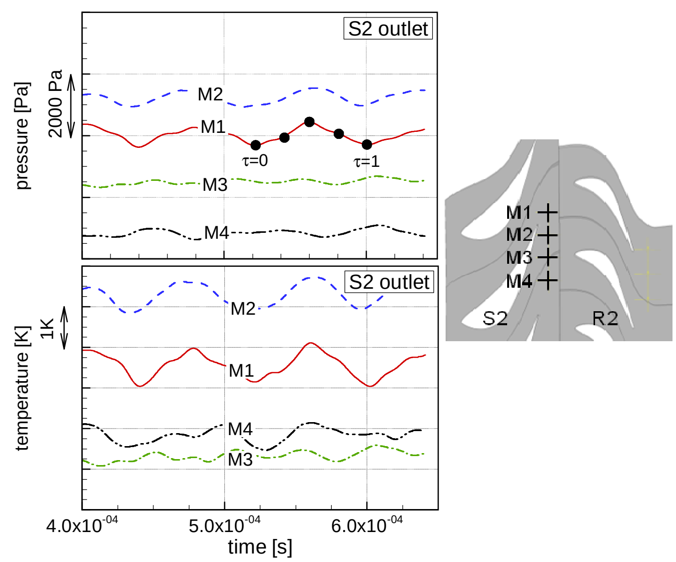

Fig. 12: Time dependent pressure and temperature distributions on monitor points in S2

It can be argued that the temperature fluctuations would be much higher if a turbine with more stages is considered. In fact for several stages upstream, the fluctuations could be high enough such that the nucleation front oscillates between the blade section and the space between the blades. This would lead to a completely different droplet spectrum, as the very low expansion rates between the blades would result in droplets which are one order of magnitude larger than those formed within the blade passages.

Further investigations of these unsteady effects are clearly merited, perhaps also with other codes. Firstly it is necessary to confirm the predicted strong shift in position of nucleation between a modelling with mixing planes and an unsteady simulation. Secondly the unexpected small temperature oscillation needs to be understood.

Future investigations should be realized in three dimensions to obtain more comparable results to the model steam turbine. A complete 3D calculation could demonstrate whether, due to different flow conditions at other positions across the span, the rotor-stator interaction leads to a similar shift of the nucleation position and a stronger oscillation of the nucleation zone. One problem is that this kind of calculation will require a considerable effort in computer resources and, above all, in simulation time (current estimate: 60 CPU's and 60 days), which may make it prohibitive.

CONCLUSIONS

The calculation of non-equilibrium condensation in rapidly expanding wet steam flows with the focus on the flow in low pressure steam turbines has been examined. There are three main conclusions from the work.

Firstly, a classical homogeneous nucleation model and simple heterogeneous nucleation models were implemented in the solver. From the comparison of these results, and with the help of a critical review of the findings in literature, it is concluded that there are certain open questions related to the modelling of heterogeneous nucleation modelling. In general, however, the steam purity is sufficiently high that heterogeneous nucleation does not play a major role. If for other situations a detailed knowledge about the process of heterogeneous nucleation is of interest much further work is required. Even for these cases the homogeneous theory may still be the best choice, especially because the results which are obtained with the homogeneous model are satisfactory from the engineering point of view. For the homogenous theory, and especially for the droplet growth model, further accurate and reliable measured test cases are desirable because, as learnt from [25], nucleation and subsequent condensation is very sensitive to boundary conditions.

Secondly, the complex effects of non-equilibrium condensation on the flow field of a three stage low pressure model steam turbine were numerically investigated using a commercial CFD code. The agreement of the numerical results with the flow measurement is quite good, and with regard to the flow field the numerical modelling of the wet steam flow could be successfully validated. Further work is required to validate the predictions of the droplet sizes.

Thirdly, a time consuming 2D unsteady simulation with transient rotor-stator interaction of the three stage turbine shows that the position of the main nucleation is shifted into the following blade row compared to steady modelling with mixing planes. The nucleation zone is slightly oscillating due to the inherent unsteadiness of turbine flow. It can be expected that the influence due to temperature fluctuations on the condensation process is higher in turbines with more than three stages and thus investigations are needed on such turbines. In general it is important for the investigation of condensing flows to be able to correctly localize the nucleation site, as the droplet size strongly depends on the supersaturation and thus on the expansion rate. This cannot be reliably attained by a 2D simulation so that further investigations should be fully three-dimensional and unsteady, if possible.

ACKNOWLEDGMENTS

The authors are grateful to ITSM engineering students Annette Seeger for her work on the Fletcher nucleation model and to Tobias Grunert for accomplishing the calculations of the Moore nozzle. Thanks are also given to Siemens Energy for their support and their permission to publish this work. The work presented in this study is funded by KW21 ("Forschungsinitiative Kraftwerke des 21. Jahrhunderts der Länder Baden-Württemberg und Bayern").

REFERENCES

[1] Stodola, A., 1910, *Die Dampfturbinen*, Springer Verlag, Berlin, 4. Edition

[2] Starzmann, J., Casey, M., V. and Sieverding, F, 2010, "Non-Equilibrium Condensation Effects on the Flow Field and the Performance of a Low Pressure Steam Turbine", ASME paper GT-2010-22467, Proceedings Turbo Expo 2010, Glasgow

[3] Gyarmathy, G., 1962, "Grundlagen einer Theorie der Nassdampfturbine", in German, doctoral thesis, ETH Zürich, Juris Verlag Zürich

[4] Kirillov, I. I. and Yablonik, R. M., 1968, "Fundamentals of the Theory of Turbines operating on Wet Steam", Mashinostroyeniye Press, Leningrad, NASA Technical Translation, NASA TT F-611

[5] Bakhtar, F., White, A. J. and Mashmoushy, H., 2005, "Theoretical treatments of two-dimensional two-phase flows of steam and comparison with cascade measurements", Proc. IMechE, Part C: J. Mech. Eng. Sci., Vol. 219 (12), pp. 1335-1355

[6] Gerber, A. and Mousavi, A., 2007, "Representing Polydispersed Droplet Behaviour in Nucleating Steam Flow", Trans. ASME, J. Fluids Eng., Vol. 129 (11), pp. 1404-1414

[7] White, A., J., 2003, "A comparison of modelling methods for polydispersed wet steam flow", Int. J. Numerical Methods in Engineering, Vol. 57, pp. 819-834

[8] Wroblewski, W., Dykas, S., Gardzilewicz, Kolovratnik, M., 2009, "Numerical and Experimental Investigation of Steam Condensation in LP Part of a Large Power Turbine", Trans. ASME, J. Fluids Eng., Vol. 131 (4)

[9] Gerber, A. G., Sigg, R., Völker, L., Casey, M. V. and Sürken, N., 2007, "Predictions of nonequilibrium phase transition in a model low-pressure steam turbine", Proc. IMechE, Part A: J. Power and Energy, Vol. 221 (6), pp. 825-835

[10] Völker, L, 2007, "Neue Aspekte der aerodynamischen Gestaltung von Niederdruck-Endstufen-Beschaufelung-en", in German, Dissertation, Universität Stuttgart, Shaker Verlag

[11] Young, J. B., 1982, "The Spontaneous Condensation of Steam in Supersonic Nozzles", PhysicoChemical hydrodynamics, Vol. 3 (1), pp. 57-82

[12] Moore, M, J. and Sieverding, C. H., 1976, "*Two Phase Steam Flow in Turbines and Separators*", von Karman Institute lecture series, Hemisphere Publishing Corporation Washington, ISBN: 0-07-042992-8

[13] Casey, M., V. and Wintergerste, T., 2000, "Best Practice Guidelines", ERCOFTAC Special Interest Group on Quality and Trust in Industrial CFD

[14] McDonald, J. E., 1962, "Homogeneous Nucleation of Vapour Condensation. I. Thermodynamic Aspects", American Journal of Physics, Vol. 30, pp. 870-877

[15] Bakhtar, F., Young, J. B., White, A. J. and Simpson, A. D., 2005, "Classical nucleation theory and its application to condensing steam flow calculations", Proc. IMechE, Part C: J. Mech. Eng. Sci., Vol. 219 (12), pp. 1315-1333

[16] Kantrowitz, A., 1951, "Nucleation in Very Rapid Expansions", Journal of Chemical Physics, Vol. 19 (9), pp. 1097-1100

[17] Dooley, R. B., 1999, "Steam, Chemistry, and Corrosion in the Phase Transition Zone of Steam Turbines; Volume 1: Key Results, Summary, and Interpretation", EPRI, TR-108184-V1, Palo Alto, California, USA

[18] Rieger, N. F. and Dooley, R. B., 1997, "Moisture Nucleation in Steam Turbines", EPRI, TR-108942, Rochester, New York

[19] McCloskey, T. H., Dooley, R. B. and McNaughton, W.P., 1999, "*Turbine Steam Path Damage: Theory and Practice*", EPRI, ISBN 0-8033-5062-7

[20] Dooley, R. B. and McCloskey, T. H., 2000, "The Role of Heterogeneous Nucleation on the Steam", EPRI, TR-113524, Palo Alto, California, USA

[21] Petr, V. and Kolovratnik, M., 2003, "Diagnostic of Wet Steam in LP Steam Turbines", 5th European Conference on Turbomachinery, Praha, Czech Republic

[22] Bakhtar, F. and Heaton, A. V., 2005, "Effects of wake chopping on droplet sizes in steam", Proc. IMechE, Part C: J. Mech. Eng. Sci., Vol. 219 (12), pp. 1357-1337

[23] Stastny, M. and Sejna, M., 2005, "The effect of expansion rate on the steam flow with hetero-homogeneous condensation", Proc. IMechE, Part A: J. Power and Energy, Vol. 219, pp. 491-497

[24] Fletcher, N., H., 1962, *The Physics of Rainclouds The Condensation of Water-Vapour*, Cambridge University Press, 1962, pp. 37-63

[25] Wroblewski, W., Dykas, S. and Gepert, A., 2009, "Steam condensing flow modeling in turbine channels ", Int. J. Multiphase Flow, Vol. 35, pp. 498-506

[26] Bakhtar, F., Ebrahimi, M. and Webb, R. A., 1995, "On the performance of a cascade of turbine rotor tip section blading in nucleating steam Part 1: surface pressure distributions", Proc. IMechE, Part C: J. Mech. Eng. Sci., Vol. 209 (12), pp. 115-124

[27] Lamanna, G., 2000, "On Nucleation and Droplet Growth in Condensing Nozzle Flows", doctoral thesis, Technical University of Eindhoven (TH), ISBN 90-386-1649-X

[28] Gerber, A. G. and Kermani, M. J., 2004, A pressure based Eulerian-Eulerian multi-phase model for non-equilibrium condensation in transonic steam flow, Int. J. Heat and Mass Transfer, Vol. 47, pp. 2217-2231

[29] Stastny, M. and Sejna, M., 2001, "Two-Population Numerical Model of Heterohomogeneous Condensation of the Steam Flowing in Turbine Cascades", 4th European Conference on Turbomachinery, Firenze, Italy, pp. 803-812

[30] White, A., J. and Young, J. B., 1993, "Time-Marching Method for the Prediction of Two-Dimensional Unsteady Flows of Condensing Steam", AIAA J. Propulsion and Power, Vol. 9 (4), pp. 579-587

[31] Moore, M. J., Walters, P. T., Crane, R. I. and Davidson, B. J., 1973, "Predicting The Fog-Drop Size In Wet-Steam Turbines", Instn. Mech. Eng. Paper C37/73, Conference Publication 3

[32] Gyarmathy, G. and Sprengler, P., 1974, "Über die Strömungsfluktuationen in mehrstufigen thermischen Turbomaschinen", in German, Traupel-Festschrift, pp. 95-141, Juris-Verlag Zürich

