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Temporary CO_2 capture shut down: Implications on low pressure steam turbine design and efficiency

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Abstract

The Natural gas Combined Cycle (NGCC) with post combustion capture using liquid solvents may in some cases be of interest to design with a flexible steam bottoming cycle, so that it can operate both with and without CO_2 capture. It is then important that the choice of the low pressure (LP) steam turbine exhaust size is made accordingly. The paper describes why a flexible NGCC requires a LP steam turbine with smaller exhaust than the corresponding NGCC without CO_2 capture, and how this will affect the LP turbine exhaust loss and NGCC process efficiency. Handling large variations in LP steam flow is in fact well-known technology in combined heat and power (CHP) plants, and the use of 3D simulation tools can further help making the best LP steam turbine design choice.

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Keywords: post-combustion capture, LP steam turbine, exhaust loss, capture flexibility, turn-up mode

1. Introduction

The post-combustion capture route may offer an opportunity for flexibility in power generation if it is possible to temporarily shut down the CO_2 capture unit. This would enable responding to short periods of high electric power demand in the power grid. When a CO_2 capture unit using liquid solvents is taken out of operation, this corresponds to an increase in power generation in the low-pressure (LP) steam turbine, since no steam is required for solvent regeneration. An alternative for temporary increase in power generation, described in [1] could be to

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continue capturing CO_2, temporarily store CO_2-rich solvent and regenerate solvent when electric power demand is low. This solution corresponds to extracting additional steam from the crossover between the intermediate pressure (IP) turbine and LP turbine, i.e. steam flow through the LP turbine is reduced when regenerating stored solvent.

In the case of coal-fired steam power plants, the issue with variable steam flows through the LP turbine section could be resolved through the use of a SSS-clutch for one of the low pressure turbines. This means that one of the low pressure turbines could be either completely shut-down or in operation, depending on the heat demand from the CO_2 capture plant.

In the case of natural-gas fired combined cycles, however, there is (usually) only one LP double flow steam turbine, and hence there is no flexibility in terms of possibility to shut down an LP turbine. In earlier work [2], it was found that the LP turbine inlet steam flow for a triple-pressure 430 MW NGCC without CO_2 capture is 111 kg/s at full load, and 72.3 kg/s at 47.9% load . For the corresponding triple pressure NGCC with post-combustion CO_2 capture with MEA (capture rate 90%) and with steam extraction for solvent regeneration at the IP-LP crossover, the LP inlet steam flow at 100% load was calculated to be 52.5 kg/s, i.e. the LP steam turbine inlet flow is lower at full load with capture, that at 47.9% load without CO_2 capture. If an NGCC is to be designed so that it is able to handle operation without CO_2 capture as well as operation at full load and part load with CO_2 capture, this will put specific demands on LP steam turbine design, since varying LP turbine flow will change the last-stage loading and may turn the stage into "turn-up" (or recompression) mode, as described in section 2 of this paper.

It should first of all be noted that LP steam turbines capable of handling large variations in steam flow is not a new technology. There are indeed several steam turbines with large LP volumetric flow variations that are already in operation. These turbines were designed and installed from the 1970's and 1980's and onwards in countries like Sweden and Denmark, where combined heat and power plants (CHP) are common. These CHP plants typically operate against both high- and low heat loads on a daily cycling basis, which offers analogies to a power plant with flexible CO_2 capture.

Altogether, the purpose of this paper is to shed light on the requirements on and performance of the natural gas combined cycle (NGCC) LP steam turbine if operating with large variations in steam flow. The paper provides a view on how steam turbine theory, knowledge from existing LP steam turbines and modern 2D and 3D turbomachinery simulation tools can be applied. In addition, process simulations with a LP turbine exhaust loss model are provided to illustrate the impact on the LP steam turbine efficiency on power process performance.

2. LP turbine exhaust design

A typical state-of-the-art large size NGCC has quite high steam data and it is not uncommon to find high-pressure (HP) turbine admission data in the order of 160 bar and 600°C. The condenser pressure is dependent on the site conditions and available cooling (air or water). This means that the volumetric flow ratio over the turbine section is high with a low volumetric flow at the turbine inlet and high volumetric flow at the turbine exhaust. This means short blades at the inlet and long blades at the exhaust side just upstream of the condenser. Both these blade length extremes have, with respect to efficiency, their own set of implications. A short blade at the inlet is relatively more sensitive to tip clearance and secondary flow effects, whilst the rear stages, and in particular the exhaust stage, experience moisture, span-wise variations of flow properties and an exhaust loss. The exhaust loss is the exit velocity at the exhaust and it is a loss since the kinetic energy of the exhaust steam flow will be lost in the condenser. A part of the kinetic energy is recovered in the turbine diffuser resulting in a lower exit pressure (i.e. lower static pressure after the last rotor blade than in the condenser). Since the exit velocity is volumetric flow over the available flow area, the need to maximize the area and thus reduce outlet flow velocity is imminent. The available annular flow area is limited by the permissible centrifugal stress level of the exhaust stage blades and disks. A modern steam turbine steel blade material may have as design value a tensile hub stress level of 620 MPa. The blade material density is approximately 8,000 kg/m^3 for steel blades. One way of providing stress-relief is to introduce taper where the blade cross-sectional area is reduced toward the tip. It is typical to have a taper-ratio (area at the hub over the tip area) of ten, which typically results in a stress factor K_{Taper} of 0.4 (i.e. a stress reduction of 60 percent). The maximum area can be evaluated with the equation:

$$A = \frac{\sigma_{Root}}{K_{Taper} \cdot 2 \cdot \pi \cdot \rho \cdot (n/60)^2} \tag{1}$$

Typical available exhaust areas as a function of rotational speed can be seen in Fig. 1. All large size turbines (above say 100 MW) are direct drive and the only freedom is to choose either full- or half speed by using either two- or four pole alternators, respectively. This means that a 50 Hz turbine in the utility size bracket either spins of 3,000 or 1,500 rpm. The available area is a non-linear function of the speed and one can show that the size increases by a factor of four when the speed is reduced to 1,500 rpm – still maintaining the same stress level. The drawback, however, is a severe increment in the turbine stage count and associated rotor length. A detailed discussion is out of the scope of the paper but the available stage heat drops is proportional to the blade speed squared. It is therefore uncommon to find half-speeders outside the nuclear industry where the sheer number of exhaust of full-speed LP turbines would be prohibitive in many cases. Most 3,000 rpm steel exhausts are around 10-12 square meters whilst titanium exhausts may reach 16 square meters (Refer to Fig. 1). The titanium blades are typically alloys like Ti-5Al or Ti-6Al-6V-2Sn rather than pure titanium and typically offer a stress reduction by a factor of 1.8. The reason for introducing such a very large exhaust stage into a NGCC LP is mainly driven by the fact that a "butterfly" or two-flow low-pressure turbine with two exhausts can be replaced with a single flow unit. A single flow unit should potentially have higher efficiency and certainly lower costs.

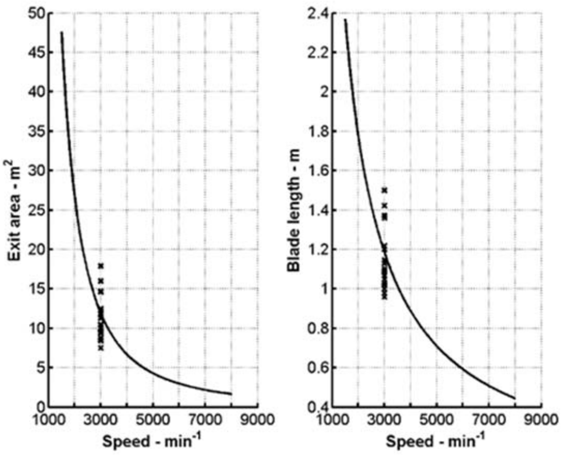

Fig. 1. LP steam turbine exhaust areas as function of steam turbine rotational speed

The costs associated with developing the last stage(s) of a steam turbine are indeed high and most manufacturers have certain available stages covering their entire application spectra. Despite most manufacturers striving to have standardized values for e.g. admission conditions, the exhaust condition is set by the local availability of a heat sink. The condenser pressure for a sea-water plant in e.g. a Nordic area may be as low as 0.02 bar whilst an air-cooled plant in e.g. Germany could have a condenser pressure of 0.048 bar. This means that for the same admission condition, there is a need for 2.4 times more exhaust area in the Nordic plant, but also that the Nordic LP steam turbine can generate more power. All manufacturers probably have ways for optimizing the condenser pressure to fit a certain optimum exhaust and no firm engineering rules prevails. It is, however, possible to discuss the principles in a general way. All exhausts have a loss bucket, when plotted against the volumetric flow (or similar), with a certain minimum value that is dependent on the exhaust stage geometry. Above the "optimum" volumetric flow the exhaust loss ($c^2/2$) increases according to some function of the volumetric flow (see equation 5 below). At lower than optimum flow, the loss increases. The underlying mechanism is different from the normal exit velocity since the last- and perhaps also the penultimate stage will begin to feed energy into the steam. This phenomenon is normally referred to as “turn-up mode”. At low loads, most LP steam turbines require cooling by spray injection to prevent from too high blade temperatures, where the water sprayed into the last turbine stage(s) typically is taken from the condenser. Spray injection could typically be applied up to 10-15 percent load.

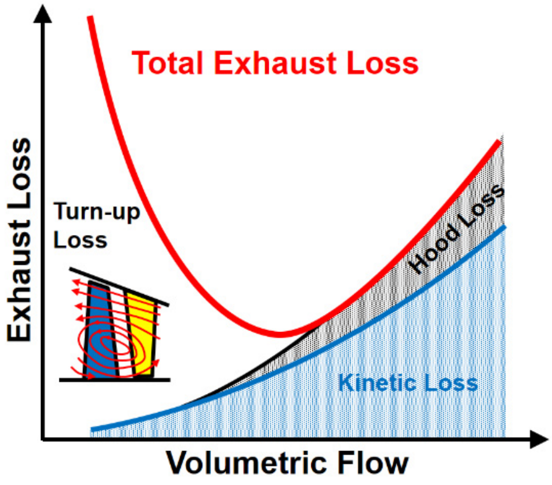

Fig. 2. Schematic illustration of LP steam turbine exhaust loss curve as a function of volumetric flow.

A detailed explanation of the involved aerodynamics is outside the scope of the present paper, where only the very basic principles are discussed. All turbine stages experience a centrifugal accelerating type of flow field by virtue of the working principle. This is commonly referred to as the “radial equilibrium” and its associated radial static pressure gradient. The pressure is always higher at the outer casing that at the inner casing and both rotors and stators experience the same type of pressure gradient, but at different levels. This is strongly related to the stage reaction gradient which causes the high tip- and low hub reaction. It can be shown that the pressure upstream each stage is proportional to the mass flow (cf. a normal nozzle) by introducing a capacity function. Since the pressure is proportional to the mass flow, a reduction of e.g. 50 percent mass flow translates into a (more-or-less) 50 percent reduction in the pressure upstream each stage. The condenser, however, does not obey the same fluid dynamic principle since the pressure is governed by the heat duty (LMTD), available surface, and heat transfer coefficients. This means that all stages - except for the last - will have more-or-less unchanged pressure ratios. It is possible to translate pressure ratio into velocity triangles, which govern the radial variation in static pressure. When the turbine is off-loaded, the last stage will initially start to reduce and upset the radial pressure gradient. The last stage will be followed by the penultimate stage when the load is reduced further and so on. At some point, the pressure at the lower part of the blade will be in a relative sense too high, which will cause too low flow velocities. This phenomenon will set-off the fairly large scale re-circulation zone(s) where work is being fed into the “trapped steam” causing heating – and lost shaft work. Despite the maturity of steam turbines, a complete understanding of the flow field at very low loads is still lacking. It should also be mentioned that “turn-up mode” is an in-stationary phenomena that can cause all kinds of in-stationary forces and ultimately forced-response and flutter issues.

From an engineering perspective the optimum design point is slightly to the right of the minimum of the total exhaust loss curve shown in Fig. 2, which reflects the right balance between performance and operability. A good rule of thumb could be to assume a velocity of 230-240 m/s for selecting the correct exhaust size. It should be noted, however, that this is the annular velocity rather than the true exit velocity as shown in the equations below.

Operation with high heat load (large LP steam extractions for district heating generation) requires a minimum cooling flow in the LP turbine for maintaining the outlet temperature at a reasonable level, as stated above. The required steam cooling is provided through a hole in the control valve mounted in the cross-over pipe between the intermediate pressure (IP) turbine and the LP turbine. In addition, water is sprayed from the condensate system for cooling of the last turbine blade row when operating below 10-15 per cent load. It is quite typical to find a grey-coloured section at the trailing edge of the last rotor where light erosion has been caused by the cooling water for LP turbines that regularly operate at low loads.

3. LP steam turbine design with loss model

3.1. Exhaust loss model based on DIN 1943

The most versatile exhaust loss model is the one in DIN 1943 [3] which includes both the exit velocity as well as windage/turn-up. The turbine exit (or exhaust) loss is the kinetic energy associated with the absolute velocity leaving the last rotor blade. The exit absolute velocity vector is defined as:

$$c_2^2 = c_{ax}^2 + c_{2u}^2 \tag{2}$$

The first term can be evaluated directly from the continuity equation whilst the second is more intricate and is a function of both the blade speed and mid-span metal angle. The latter is solved by introducing an axial velocity ($c_{ax,0} = u \cdot \tan(\beta_2)$) that would have given a swirl-free exhaust, then basic *regula de tri* yields:

$$\frac{c_{ax,0}}{u} = \frac{c_{ax} - c_{ax,0}}{c_{2u}} \quad \Rightarrow \quad c_{2u} = \frac{u\left(c_{ax} - c_{ax,0}\right)}{c_{ax,0}} \tag{3}$$

The two preceding equations can be combined to yield:

$$c_2^2 = c_{ax}^2 + u^2\left(\frac{c_{ax}}{c_{ax,0}} - 1\right)^2 \tag{4}$$

The final form is obtained by introducing empirical factors for deviation and ventilation/turn-up work:

$$\frac{c_2^2}{2} = \frac{c_{ax}^2}{2}\overbrace{\left[1 - 0.08\left(\frac{c_{ax}}{380}\right)^4\right]}^{Deviation} + \frac{u^2}{2}\left(\frac{c_{ax}}{u \cdot \tan\beta_2} - 1\right)^2 \underbrace{\overbrace{\frac{u \cdot \tan\beta_2}{c_{ax}}}^{Turn-Up}}_{=1 \; if \; c_{ax} > u \cdot \tan\beta_2} = f\left(c_{ax}, u, \beta_2\right) \tag{5}$$

The axial velocity can be replaced by the continuity equation and the exhaust area (c_{ax}=Q/A). The presented derivation differs slightly from DIN 1943, where the entire second term is dropped at higher axial velocities (i.e. $c_{ax} > u \cdot \tan\beta_2$). The rationale behind the assumption in DIN 1943 is that most steam turbines are designed for rather low swirl levels and the relative importance is therefore smaller at higher flows. In this work, however, a more rigorous approach is taken where both terms are kept and the ventilation component is cancelled at higher velocities.

3.2. Impact on steam turbine performance when applying the loss model

Applying the LP steam turbine exhaust loss model described in section 2, simulations were made to illustrate the behavior of the turbine with varying flow. Condenser pressure at full load was set to 0.048 bar, which is typical for an application where the cooling effect comes from cooling towers. Fig. 3 and Fig. 4 show the exhaust loss vs. volumetric flow and the total-to static efficiency vs. the relative Parson number. The latter is a gauge of the aerodynamic loading† and it is common practice to correlate efficiency as a function of the relative Parson number. The usage of the relative Parson number cancels out the speed dependence for a fix-speed unit. The design point efficiency of 90%, without extraction for the CCS-plant is set at a relative Parson number of unity. The Parson number increases (i.e. a more lightly loaded turbine) to 1.13 when the steam extraction occurs. The extraction system is integrated with the low-pressure steam system and requires a control valve before the Low-pressure turbine to maintain the supply pressure to the CCS system at 4 bars. This means that the intermediate-pressure and low-pressure turbine will operate under varying pressure ratio (and Parson number) over the load range. When the turbine is operated at part load with CCS, the low-pressure turbine relative Parson number reduces to about 1.05 at the lowest load (from 1.13 at full load) – hence an increased efficiency at part load. This is not the case for normal turbine operation (without CCS) at part load and is only a consequence of the large extraction.

The low-pressure turbine efficiency could be seen as the total-to-static with the contribution from the leaving loss. The exit loss is evaluated according to the preceding equation and the exhaust size is set to produce the figure below. The effects due to the wetness loss is included in the work but omitted in the figures below due to clarity reasons.

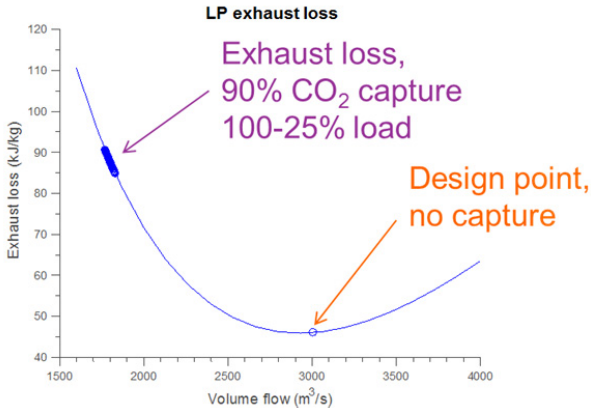

Fig. 3. Exhaust loss variation with varying steam volume flow without and with CO_2 capture

4. Process simulations

4.1. Full load, no CO_2 capture

Process simulations were conducted for a triple-pressure NGCC without and with CO_2 capture, where the gas turbine is modeled to reflect the performance of the GE Frame 9B engine. Simulations were done in the equation-solving process simulator IPSEpro. Key results for the NGCC when simulated without CO_2 capture are given in Table 1.

It should be emphasized that the LP steam turbine geometry was set up so that the exhaust loss was minimized at full load without CO_2 capture, as illustrated in Fig. 3. The reason for this was, for an illustrative purpose, to maximize the impact of exhaust loss increase in the process simulations.

† $X = \frac{\sum_{i=1}^{n} u_i^2}{\Delta h_s}$

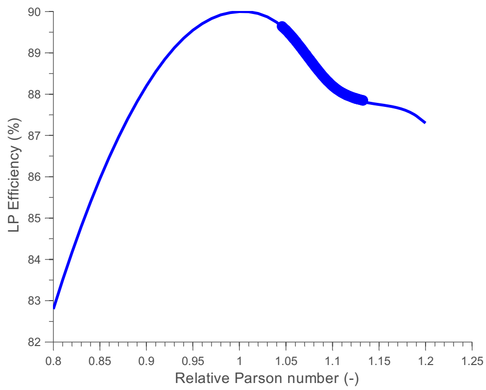

Fig. 4. LP steam turbine efficiency variation wit relative Parson number

Table 1. Key data for the NGCC witout CO_2 capture

Net power output	438.4 MW
Net efficiency	58.7%
LP Turbine isentropic efficiency	90%
LP turbine exhaust loss	46 kJ/kg
Steam pressure at IP/LP crossover	4 bar
Condenser pressure	0.048 bar

4.2. Simulations at full load and part load with CO_2 capture

Simulations were made for the NGCC with CO_2 capture at full load and part load. Such simulations were previously presented for the same gas turbine in [2], but with NGCC process simulations done with GTPro and capture process simulations done in ProTreat. The IPSEpro simulations were done with in-house code, and results for the NGCC have previously been presented in [4], [5] and [6]. The capture unit modeled in IPSEpro for full load and part load simulations was previously applied in [7]. A comparison of results for the LP steam turbine mass flow in the triple-pressure NGCC with steam extraction at the crossover is given in Table 2. It can be seen that there is a deviation between the models concerning how much steam is required for regenerating the solvent at full NGCC load with 90% CO_2-capture. The model applied in [7] requires more steam for solvent regeneration at full load than the more detailed model applied in [2]. Although the model in [2] presumably is more accurate for the calculation of steam requirements with varying load, the IPSEpro model in [7] was maintained in the present work, to emphasize the impact of large variations in steam flow on LP turbine performance.

Table 2.Mass flows in kg/s through the LP turbine for operation without and with CO_2 capture

	Full load, no capture	Full load, 90% capture	47.3% load, 90% capture
This paper, IPSEpro	111.9	43.9	42.1
Jordal et al. [2], GTPro + ProTreat	111.0	52.5	43.3

After extraction at the IP/LP crossover at 4 bar, the steam remaining at the LP turbine inlet was throttled to a pressure that gave constant LP turbine inlet volume flow. The LP steam turbine model was set up to reflect a double-flow configuration. As for a real gas turbine engine, the load of the NGCC model is regulated through variation of the gas turbine inlet guide vane angle in the IPSEpro simulations.

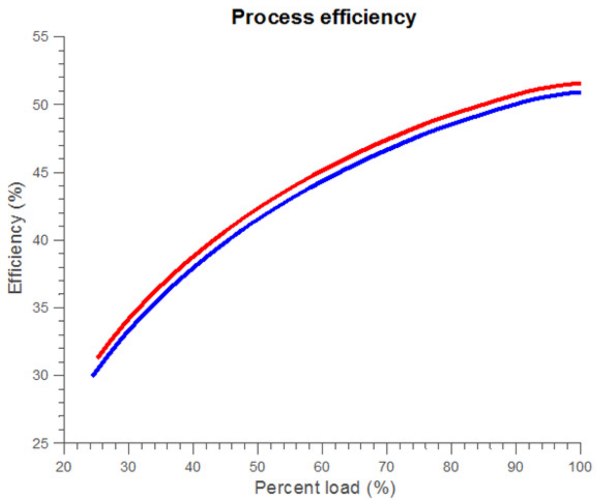

Fig. 5. NGCC process efficiency with 90% CO_2 capture. Red line: constant LP turbine efficiency (90%), blue line: turbine efficiency determined with loss model derived from DIN 1943 [3].

Process efficiency at full load and part load with 90% CO_2 capture can be seen in Fig. 5. This figure clearly shows that the overall efficiency of the NGCC at full load and part load can be overestimated if the turbine efficiency is maintained constant at a value that is valid for operation with no CO_2 capture. It should be recalled, however, that the steam flow required for solvent regeneration probably is overestimated at high loads and that that the turbine model applied in the process simulations was set up with geometry parameters that give a minimum exhaust loss penalty at full load without CO_2 capture. Obviously, this is not the preferred LP turbine exhaust design for an NGCC that should be able to operate both without and with CO_2 capture. The exhaust size should be reduced, as sketched in Fig. 6, to give higher exhaust losses at full load without CO_2 capture. Compared to the results shown in Fig. 5, the difference in overall process efficiency illustrated by the red and blue lines should then presumably be reduced.

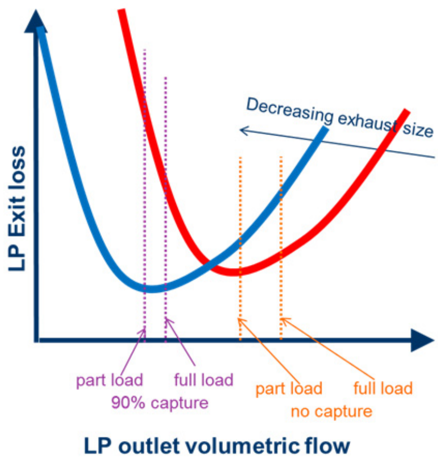

Fig. 6. Sketch of the impact on LP steam turbine exhaust losses due to chosing a smaller LP turbine exhaust size

5. LP steam turbine flow pattern with 3D simulations

The exhaust loss model in equation (5) can describe the shape of the exhaust loss curve that can be seen in Fig. 3. However, this model cannot capture the unsteady behavior of "turn-up" mode, which was mentioned in section 2. By means of applying modern 3D simulation tools, the unsteady behavior at low load can to some extent be indicated. A generic LP turbine last stage was set up and simulated with AxCent™ to illustrate the flow phenomena that occur at low loads. It should be noted in this context that since turn-up mode is unsteady, it cannot be fully captured with a steady state simulation. Instead, one has to use time-resolved calculations to fully capture the behavior.

In Fig. 7, it can be seen that at low loads, there is a recirculation zone on the suction side of the rotor blade, which is a pre-stage to entering turn-up mode. The flow separation indicated in this figure is propagating upstream, and can also enter the turbine stages upstream of the last stage.

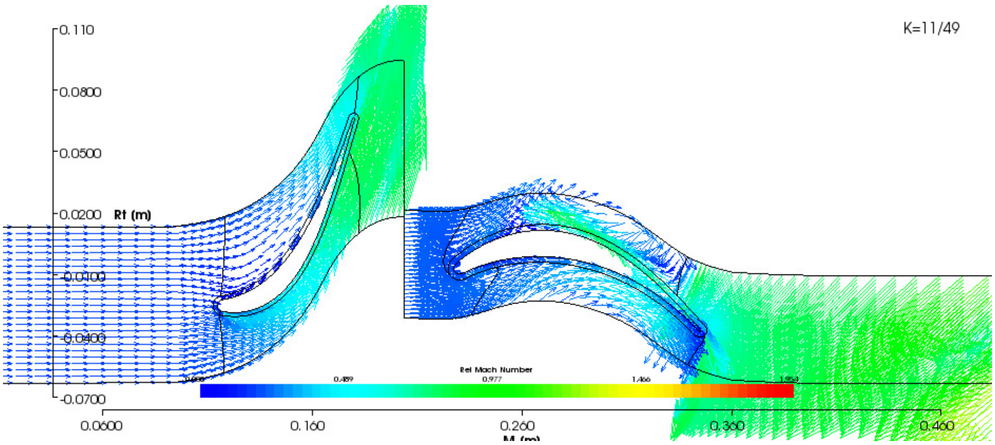

Fig. 7. Cross section of generic LP turbine last stage simulated with AxCent™ at ~15% load.

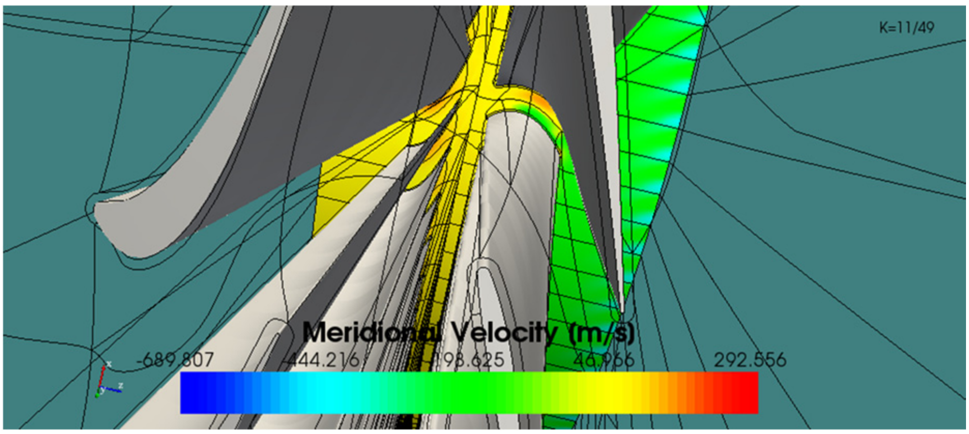

Fig. 8. Meridional velocity in a generic LP turbine last stage simulated with AxCent™ at ~15% load.

The design philosophy will determine the radial pressure gradient at full load for the last stage of the LP turbine. As mentioned previously, the pressure gradient will be reduced at part load, which will cause an imbalance in the flow field. A second view of the flow field in the generic LP steam turbine at low load is shown in Fig. 8. Here, the meridional velocity of the LP turbine is illustrated. It can be seen that at the outlet of the rotor, the flow velocity is negative, i.e. at low loads the last stage of the LP turbine may act as a compressor, with the steam flow propagating upstream. This is clearly an unwanted phenomenon, and the simulation results underline that the LP steam turbine exhaust should be chosen narrow enough to avoid that this phenomenon occurs at NGCC operation with steam extraction for solvent regeneration.

6. Conclusions

This paper has outlined the issue with LP steam turbine operation at full load and part load in flexible NGCC power plants that are intended to be able to operate both with and without CO_2 capture. This requirement can be put on both NGCC power plants that are retrofitted with CO_2 capture, and on NGCC plants with CO_2 capture where the ability to temporarily shut down CO_2 capture is required as a means to respond to peaks in electric power demand.

A proper design choice must be made for the LP turbine exhaust, more specifically the exhaust must be made more narrow than for NGCC plants that are designed for operation without CO_2 capture. Otherwise, at part load or perhaps even at full load with CO_2 capture, the LP turbine may be in the risk of entering "turn-up" mode, which is an unsteady and highly undesired operation mode.

The design of LP steam turbines with large variations in steam flow is a known technology, which has been applied in e.g. large combined heat and power (CHP) plants in e.g. Sweden and Denmark since the 1970's. The application of modern 3D steam turbine design tools should enable further decision support when choosing an appropriate LP steam turbine exhaust size.

Utilities considering to build an NGCC intended for operation both without and with CO_2 capture should be aware of the LP steam turbine exhaust size issue addressed in this paper, in order to be able to enter in a good dialogue with technology providers, so that the most appropriate design choice is made for the LP steam turbine.

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