MATEC Web of Conferences **367**, 00013 (2022) https://doi.org/10.1051/matecconf/202236700013
Power System Engineering 2022

Preliminary Experimental Investigation of the Design-Overloaded Stage in Two-Stage Axial Turbine Test Rig

Tomáš Jelínek[1*], *Martin* Němec[1], *Petr* Milčák[2] and *Richard* Fichtl[2]

[1] VZLU – Czech Aerospace Research Centre, Beranovych 130, Prague, Czech Republic
[2] Doosan Škoda Power, Tylova 1/57, Pilsen, Czech Republic

Abstract. The article deals with the issue of experimental research of the so-called design overloaded stage. A new experimental device enabling measurements on two-stage axial turbine is advantageously used in this research task. The experimental setup is designed so that a preparatory stage is included in front of the examined stage. The stage design introduces some aerodynamic overload already in the flow path design phase. This research program is focused primarily on the effect of leakage flows and their mixing with the mainstream. This paper describes the verification measurements on a newly introduced two-stage arrangement and reveals the first data obtained.

1 Introduction

The global steam turbine market is evolving due to composition of electricity generation sources, the need for greater decentralization of power production, and the requirements for cogeneration of electricity and heat. Demand for large, continuously operated units is moving towards smaller units with greater operating flexibility. High competition in the market pushes to reduce prices, but at the same time it does not slow down the requirements of achieving high efficiency. This encourages the producing companies to research new technical solutions.

The above market requirements direct the design of turbines to smaller, often single-casing, machines (power up to 100 MW), where the pressure on the price affects the design options. One of possible solutions applied by Doosan Skoda Power Company is the deployment of so-called design-overloaded stages. However, it is not an aerodynamic overload stage in the sense of operation in non-design mode. On the contrary, this overload regime is already considered when designing the stage, respectively the whole flow path. This aerodynamic overload is caused, for example, by the design of the flow section with a reduced number of stages compared to the so-called optimal loading. The motivation of such a design can be, for example, a demand to reduce the bearing span and the price reduction mentioned before. The designed stages thus work in the nominal (design) mode, but with certain aerodynamic overload.

* Corresponding author: jelinek@vzlu.cz

The optimal load of the stage is usually understood as a load at which the stage reaches the highest efficiency. When an isolated stages is considering, this mode is close to the regime when the outlet stage velocity is minimal, i.e., in the axial direction. Aerodynamic overload of the stage leads, among other things, to a deviation of the output speed from the axial direction - the output angles reach values of 60° or less. In a multi-stage arrangement of stages, the design of the downstream stage must be subject to such a "design regime". Due to the gradual expansion and construction of the flow path from similar stages (with a similar level of reaction), this stage is also aerodynamically overloaded. The situation for the downstream stage is thus repeated, i.e. the design-overloaded stages usually have a non-axial input and output.

Doosan Škoda Power Company has implemented the extensive development program over the past decade, in which it has developed and tested a wide portfolio of blade geometries for the design of stages ranging from low reaction to full reaction stages. Experimental research of these stages was carried out in cooperation with VZLÚ - Czech Aerospace Research Centre. At the same time, loss models and procedures for selecting a suitable blade type or blade combination for a given application were prepared [1]. The choice of the stage loading resp. the processing rate of the thermal gradient was primarily solved through the selection of the appropriate level of stage reaction in combination with the reference design patterns. The aerodynamic stage overloading was chosen to be rather moderate; if greater thermal gradient processing was required, a reduction in the level of reaction was preferred.

From the above requirements for higher aerodynamic stage loading, questions logically arise regarding the appropriate level of aerodynamic overload compared to the reduction of the reaction of the stage, resp. a combination of both approaches.

2 Theoretical background

The dependency of the stage efficiency on the stage loading represented by a velocity ratio u/c is the basic axial turbine characteristic. The optimal loading usually corresponds to the regime with axial outlet velocity, i.e. smallest leaving loss.

When the stage loading changes, the stage works in an overloaded or underloaded regime; correspondingly the flow with off-design direction hits the profiles. This change is associated with an increase in profile (incidence) losses and thus losses of the entire stage. In the off-design regime, the stage outlet velocity is non-axial and thus the outlet velocity (leaving loss) is increased. These basic ideas can be observed broadly in literature as a chart with the efficiency depending on the u/c ratio, where the efficiency in the total-to-static definition changes more significantly due to the change in outlet velocity. On the contrary, such a dependence of the efficiency in the total-to-total definition is relatively flat, as the change in row efficiency due to incidence is mainly reflected.

Moreover, the change in the stage loading will also induce a change in the stage reaction (distribution of the thermal gradient between the stator and the rotor) and thus other related changes. When the complex stage concept is considered, changes in the seal leakage rates are evident, as also their downstream mixing with the main flow from the blades.

Consider now a multistage flow path consisting of similar stages. The shape of stator blades should correspond to the outlet flow of the upstream stage. This means that a higher outlet velocity, caused by the off-design regime, might be fully utilized as higher inlet energy in the further stage and therefore we can deal only with the total-to-total efficiency. Also, the shape of rotor blades should be appropriately adjusted during the design to the considered stage loading regime. When speaking about design-overloaded stages, it means, these stages are purposely designed for higher loading (compared to the nominal loading with axial outlet flow). Such stages do not suffer from incidences in this design-overloaded regime and a higher outlet velocity means no additional loss. On the other hand, the profiles for design-overloaded stages have higher flow turning which might result in higher both profile and secondary losses. Finally, the behaviour of design-overloaded stages in off-design conditions is also a topic of interest. The assessment of increased stage power from overloading versus increased stress resistance of more bended profiles is also an interesting topic mainly from the point of aspect ratio (blade length to chord) and its effect on losses.

Experimental verification of the behaviour of design-overloaded stages in various operational conditions is the subject of this paper.

3 Experimental setup

3.1 Experimental equipment description

To experimentally verify the design of the design-overloaded stage, it is necessary to achieve a non-axial inlet into the tested stage. For this purpose, it is possible to advantageously use a two-stage arrangement in the test rig, where the design-overloaded stage is preceded by another "preparatory" stage. This stage can also be used to change the inlet conditions for tested stage; for example by changing the clearance in the shaft and shroud seal (see previous research at the single?-stage and 1.5-stage test rings presented in [2] and [3]).

A proper test rig has been created at VZLÚ, which enables the measurement of two axial turbine stages. The device was described in detail in [4] and its cross-section is shown in Fig. 1. Both stages are mounted on the overhanging end of the shaft, which is connected on the opposite side via a clutch to a torque meter. It is more accurate than the water brake (dynamometer) that is used only to set and keep specific testing RPM. In this way, the overall performance of both stages can be measured. To determine the performance of the individual stages, it is also necessary to measure the differential torque between the two stages. For this purpose, a special hub has been designed, on which the wheels of both stages are mounted, and which measures the torque of the second stage using a measuring element equipped with strain gauges. The signal from this measuring element is lead out via a hollow shaft to the side of the dynamometer and is scanned employing a telemetry device.

A detailed investigation of the flow field parameters can also be performed by this experimental device. The flow parameters are measured by the pressure probes in the planes behind the first and second stage, or the planes between the stator and the rotor.

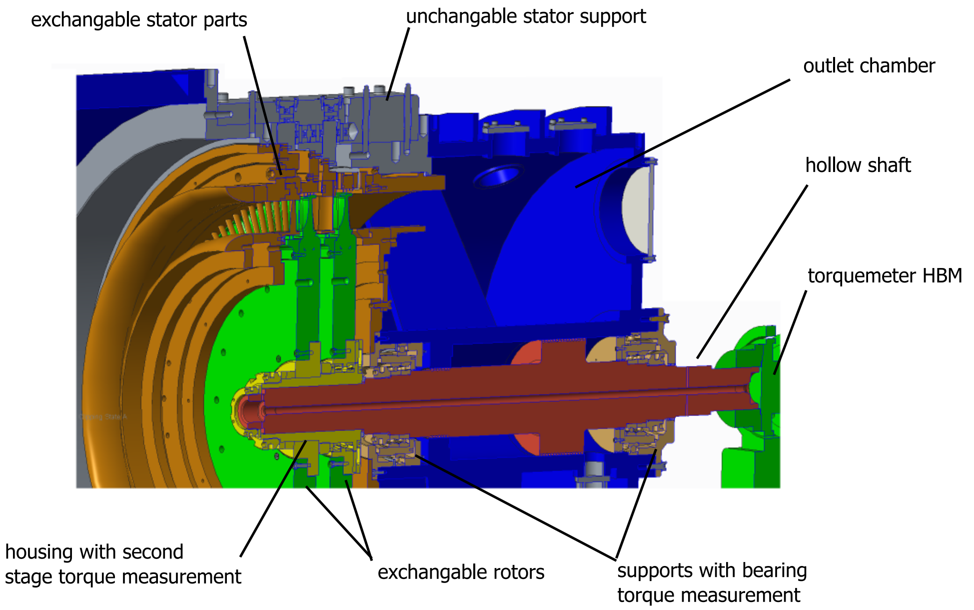

Fig. 1. The cross-section of the two-stage axial test rig.

3.2 Experimental models

The newly designed stage working in aerodynamic design-overloaded mode was installed as the rear stage. The existing stage from the previous research was chosen as the preparatory (front) stage, which works during the tests in the off-design (overloaded) regime so that the output angle is about 70°. The second stage has been designed in two variants of rotor blades to build a full-reaction (HR) overloaded stage and a mid-reaction overloaded stage (MR). Both variants use the same stator blades. This stator wheel (Fig 2) has a special prismatic-blade-based design that corresponds to practical use in small industrial turbines of DSPW. The prismatic shape of blades and their special assembly enabled us to employ 3D printed plastic blades with instrumentation by pressure taps in the bladed wheel.

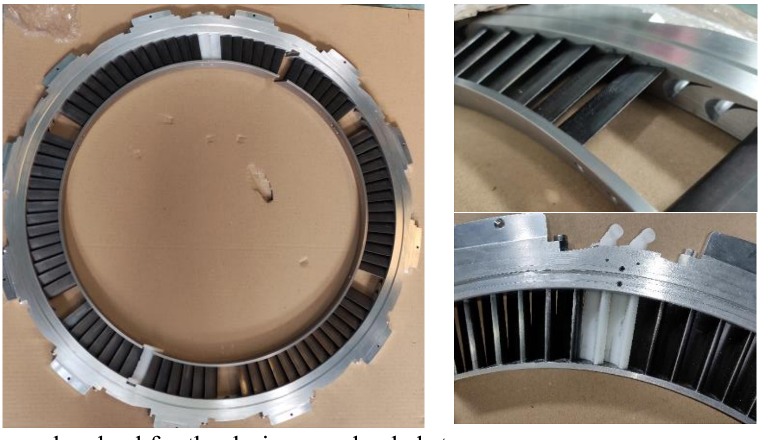

Fig. 2. New nozzle wheel for the design-overloaded stage.

Due to the extensive testing campaign, only the first HR variant tests results are presented. The overview of the stages geometry for the tested HR variant is in Table 1.

Table 1. Summary of stages geometry.

	The existing geometry of 1st stage		The new geometry of 2nd stage	
root diameter [mm]	506.00			
blade length [mm]	45.50	47.00	48.50	50.00
blade count [1]	98	78	96	78
midspan chord [mm]	23.00	28.65	24.00	29.49
stage reaction TS [%]	~ 50		~ 50	
design velocity ratio [1]	~ 0.63			
flow outlet angle [°]	70.00		70.00	

3.3 Experimental work procedure

In the case of measuring a single-stage arrangement, the setting of the operating point (pressure drop) is performed simply based on the measured parameters in front of and behind the stage. Furthermore, to determine the operating characteristics, the stage is gradually loaded and unloaded using a brake (dynamometer) and thus the rotational speed is changed. In the case of a two-stage arrangement of design-overloaded stages, it was essential to find such a pressure drop (mode), when the outlet angle from the 1st stage was identical to the outlet angle from the 2nd stage when the stage load changed. This regime then simulates the design-overloaded stages operation in the flow path. If there was different loading of stages, it will simulate the classic behaviour in off-design states, where the change in load is propagated from the rear stages and only in a few stages. The primary goal is to simulate the design operation of a design-overloaded stage.

For this reason, the experimental research procedure was set up by determining the design pressure ratio first. Here, the aim was to confirm or find the pressure ratio, when the aerodynamic load of both stages is the same, respectively the output angles are the same. Then the design point was determined. For the design pressure ratio, the design point was determined by changing the speed at which the outlet angle from the first stage corresponded to the design angle (identically also the outlet angle from the second stage). This phase also verified the behaviour of both stages in different regimes - the change of output angles should be roughly the same for both stages.

4 Results

4.1 Operating Vibrations

As this measurement was also the first operation of the two-stage configuration of the test rig, it was necessary to measure the operating parameters of the assembly, in particular, the vibrations over the entire operating speed range at first. Both wheels and hub assembly were properly balanced before the operation started.

The initial measurements were aimed primarily at the behaviour of the entire device with the installation of an assembly of two impellers (the weight of the rotor assembly was about 50 kg). In the entire investigated speed range (up to 6 000 rpm), the operation was very calm in terms of vibration. Fig. 3 shows the vibrations as a function of rotational speed for the case of a two-stage turbine in comparison with a single-stage turbine operated during the start-up phase of the device [4]. It demonstrates sufficient dynamic stiffness of the measurement device and indicates the smooth operation of future operation in experiments with a two-stage turbine assembly.

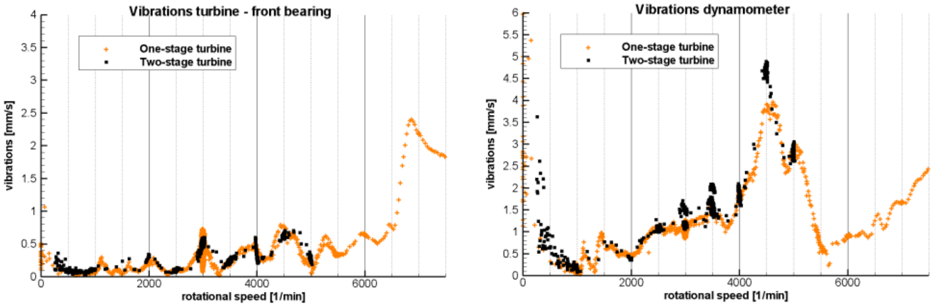

Fig. 3. The measured vibrations as a function of rpm. Comparison of one-stage and two-stage assembly.

4.2 Comparison of first (preparatory) stage parameters

Initial analyses of the results were focused on comparing the flow fields gained from the original measurement on the single-stage turbine with current ones on the two-stage turbine to verify the assumption that results from experiments performed in the past can be used.

Fig. 4 shows the radial distribution of the parameters at the output of the first stage. For this summary, the relative angle at the rotor outlet and the relative mass flow distributions were selected. The usefulness of these parameters is that they blur slight differences in the turbine mode settings. A more significant difference is evident at the blade tip. It is caused by different traversing planes - the current measurement is performed just behind the rotor and therefore there was no mixing of the seal purge flow with the main flow, while in the original measurement the plane is more distant, and the flows were mixed. Despite some differences, the comparisons show that the results of the original measurement can be used to solve the overall efficiency distribution between the two stages.

Fig. 5 shows the comparison of the efficiency obtained from the original measurement on a single-stage turbine and the current results from the two-stage turbine. In the case of the two-stage turbine, the torque of the first stage is computed from the total torque and the second stage torque measurement.

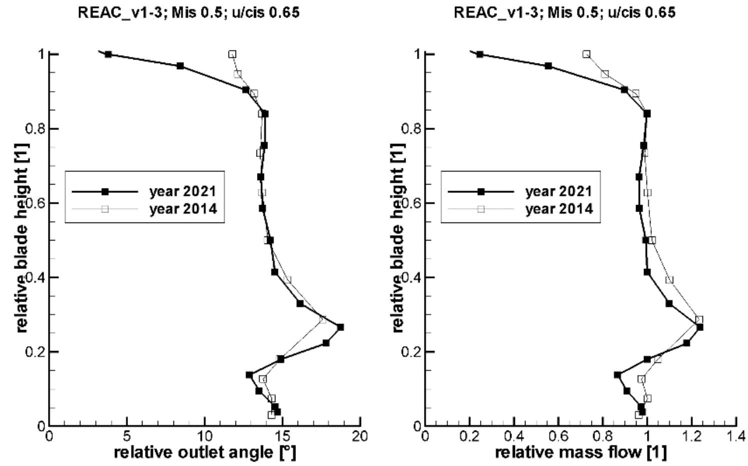

Fig. 4. Radial distribution of relative outlet angle (left) and relative mass flow (right) at the first stage outlet in comparison with the original (2014) results on a single-stage turbine.

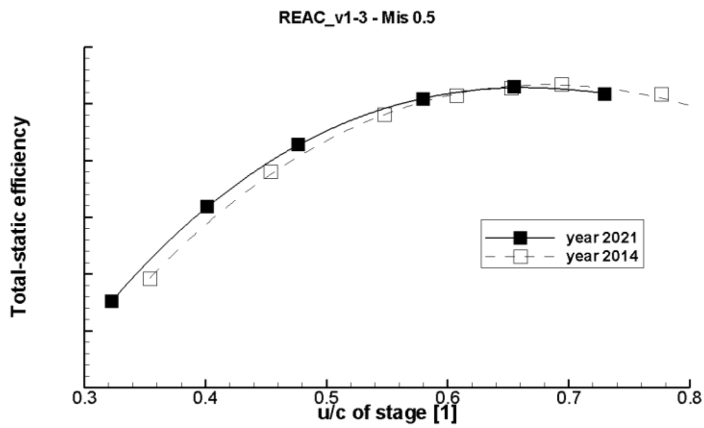

Fig. 5. Comparison of efficiency gained during the original measurement and current results using the differential torque meter

4.3 Design pressure drop determination

It was mentioned in previous paragraphs that the essential initial phase was to determine the design pressure drop where the outlet angles from both stages were the same. The result of this phase is a function of both stages outlet angles shown in Fig. 6. It is seen that at low pressure drops the aerodynamic load of the first stage is higher than the second stage (assessed from the value of the outlet angle) and vice versa for the second stage. The balanced load occurs at a pressure ratio of approx. 1.40–1.43, which corresponds very well to the design value. From the curves of the values of the output angles as a function of rotational speed, it is also possible to see that both stages are overloaded, resp. underloaded simultaneously. At the design pressure ratio, the design RPM (i.e. operating point) is based on values of approximately 3700-3800 RPM, which is only slightly higher than the design considers.

The efficiencies as a function of rotational speed for both stages are shown in Fig. 7. Two approaches to the distribution of torque between stages are compared here. One approach is the distribution based on the calculation of the characteristics of the first stage (the calculation is based on the previous measurement results on a single stage in 2014), the second approach is the determination of the second stage torque using the differential torque meter.

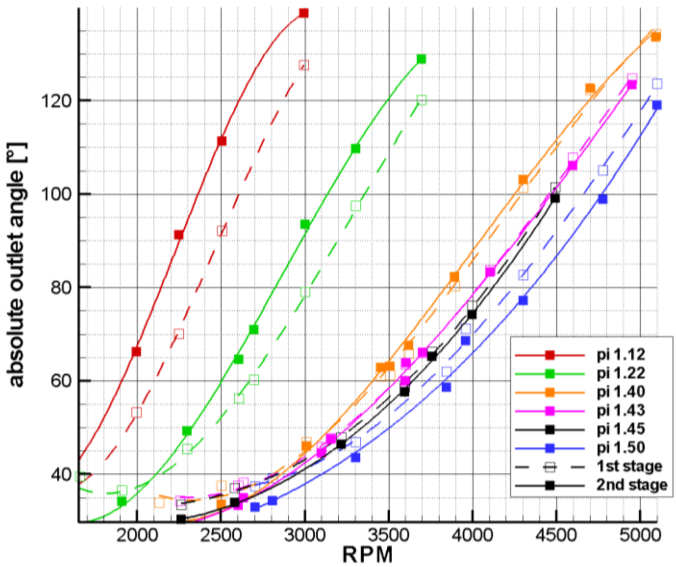

Fig. 6. Output angles from both stages as a function of rotational speed and inlet-to-outlet static pressure ratio

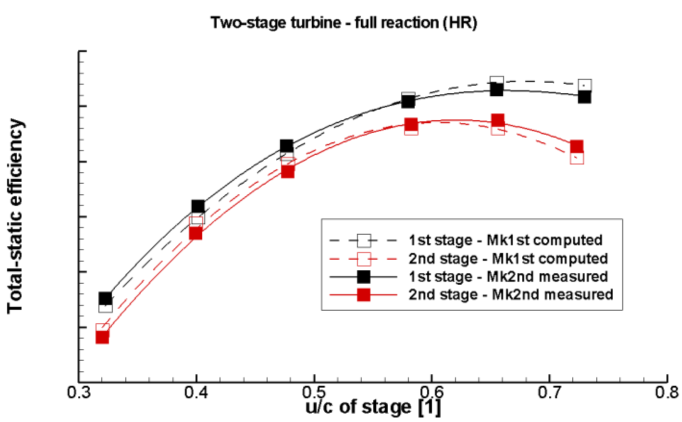

Fig. 7. Total to static efficiency: comparison of determination by dividing the overall torque measurement using the differential torque meter and using the data from the one-stage configuration.

5 Conclusion

This paper reveals part of the present research works that are conducted in the cooperation of Doosan Skoda Power and VZLU research institute. It is focused on design-overloaded stages that represent one of the answers to the changing trends in the steam turbine market.

This new experimental program is possible thanks to a new test rig available at VZLU laboratories – the two-stage axial turbine. New blading has been designed, manufactured, and assembled with some parts from the single-stage test rig to form the design-overloaded stage. First experiments have been performed to verify trouble-free operation of the test rig and to confirm the design point of this new two-stage turbine. An overview of acquired experimental data is presented.

Further experiments are going to focus on the investigation of tip leakage flow mixing with the main stage flow. Various configurations of the tip leakage flows in the first stage will be tested to evaluate the impact on the performance of the second design-overloaded stage. The tests will be performed for several operation regimes and for two stages reaction levels: full-reaction and mid-reaction. The importance of the shroud leakage flow mixing has been recognized and broadly investigated for example by Rosic et al. [5] or Shibata et al. [6].

This extensive experimental program is realized with the financial support of the Technology Agency of the Czech Republic within the THETA 3 program, project No TK03020061.

References

1. J.H. Cheon, P. Milcak, A. Pacak, C.R. Kang, M. Stastny, *Profile Loss Prediction for High Pressure Steam Turbines*, in Proceedings of ASME Turbo Expo 2016, GT2016-56114. (2016)
2. J. Uher, P. Milcak, R. Skach, D. Fenderl, P. Zitek, M. Klimko, *Experimental and Numerical Evaluation of Losses from Turbine Hub Clearance Flow*, in Proceedings of ASME Turbo Expo 2019, GT2019-90726. (2019)
3. M. Nemec, T. Jelinek, J. Uher, P. Milcak, *Effect of Stage Reaction and Shaft Labyrinth Seal in a Stage of an Axial Steam Turbine*, in Proceedings of the ASME Turbo Expo 2020, GT2020-14741 (2020)
4. M. Němec, T. Jelínek, *Adaptable Test Rig for Two-Stage Axial Turbine*, MATEC Web of Conf., 345, 00022. (2021),
5. B. Rosic, J.D. Denton, G. Pullan, *The Importance of Shroud Leakage Modeling in Multistage Turbine Flow Calculations*, J. of Turbomach., 128, 699–707. https://doi.org/10.1115/1.2181999 (2006)
6. T. Shibata, H. Fukushima, K. Segawa, *Improvement of Steam Turbine Stage Efficiency by Controlling Rotor Shroud Leakage Flows: Part I — Design Concept and Typical Performance of a Swirl Breaker*, GT2018-75269 (2018)