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Increasing the performance of steam turbines at part load by optimizing the control system during operation

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ABSTRACT

Steam turbines are subject to process-related operational fluctuations. It is mandatory for the machines to react flexibly to process changes and to operate efficiently in part load or with fluctuating steam parameters. In impulse turbines the power is controlled by valves that regulate the steam flow and pressure approaching the stator nozzle. To optimize part load behavior of the steam turbines the nozzle ring is separated into segments that can be individually supplied with steam. The division of the nozzle ring into segments is defined during the design phase of the machine and cannot be changed without major re-work. Experience shows that the actual operating points deviate from the data originally specified.

In order to optimize part load performance a method is developed to optimize the turbine and the control logic to the actual part load operating condition. To do this, operating data is continuously collected using Howden's proprietary operating data acquisition system "Uptime", stored in the cloud and analyzed by experts. If there is potential in terms of efficiency improvement, the controller is adjusted accordingly and the control of the valves is optimized. The changeover of the control system and the associated partial load optimization can be carried out during ongoing plant operation and usually without any mechanical changes to components. The performance increase is measurable and can be proven, for example, via “Uptime”.

In this work, the findings from development of the flexible valve system, simulation of the turbine in the controller model and from operation will be presented.

ROLE OF THE STEAM TURBINE IN THE RENEWABLE ENERGY ENVIRONMENT

In the future, the share of large base-load power plants in electricity generation will decrease and be replaced by many smaller decentralized power generating units [1].

Therefore, the applications and fields of use for steam turbines as turbogenerators are and will remain diverse. In addition to their use in traditional power plants or in combined heat and power generation, backpressure turbines are used in industries with fluctuating heat demand, such as the food and beverage industry or the paper industry. In these industrial applications, the focus is on controlling the steam process parameters. Strongly fluctuating live steam parameters or heat supply are typical for biomass, waste or sewage sludge incineration or for the conversion of waste heat to electricity in the steel and glass industry [2][3].

For all these applications and to meet the requirements of a volatile and decentralized electricity market, steam turbines with high flexibility, easy operation and good maintainability are needed. High flexibility in this context means that the control of the turbine must be fast, reliable and the turbine must also be operated at partial load for longer periods of time.

TECHNICAL DESIGN: TYPES AND DESIGNS OF INDUSTRIAL STEAM TURBINES

In accordance with the multitude of applications, the design of industrial steam turbines is also diverse. Depending on the customer, the application and the legal framework conditions, the focus of the users is not only on the plant efficiency, but also on a high availability, quick-start capability, the use of the residual heat CHP for heating purposes or the possibility of controlled extraction of steam at specified pressure levels [4].

Figure 1 shows a steam turbine with electromechanical control valves. The compact design with integrated gearbox on a base plate is clearly visible. When several steam turbine housings are mounted on a common gearbox, it is also possible to provide controlled steam extraction in the connecting lines of the individual steam components. These advantages allow a high degree of flexibility in combined heat and power plants or in industrial applications for the conversion of residual steam into electricity.

The operationally proven principle of the impulse turbine arranged overhanging on an integrated gearbox allows the quick-start capability required in industrial applications and simple commissioning and maintenance. There is no need for a rotating device or complex start-up and shut-down procedures for pre-heating or cooling the turbine housing and rotor [5].

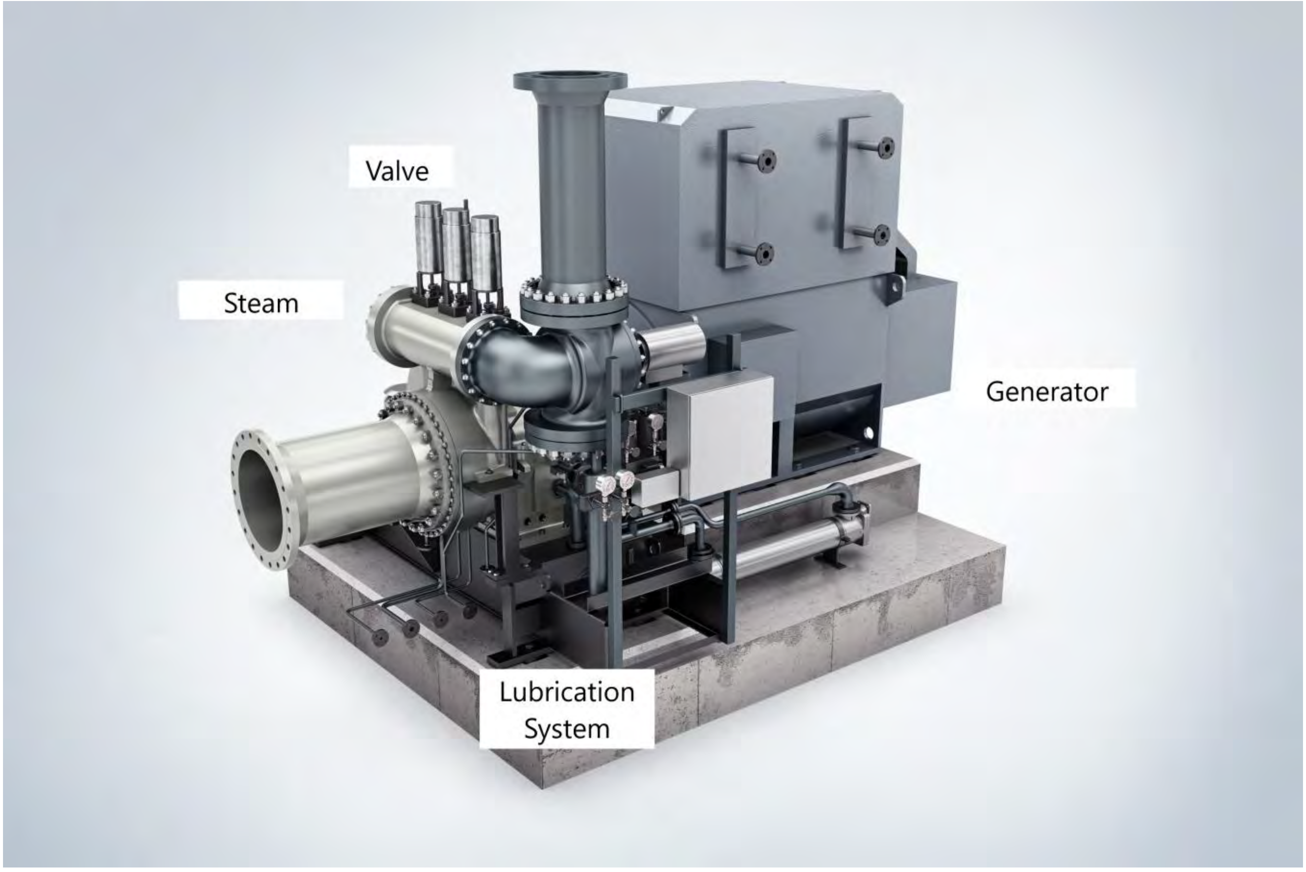

Figure 1: MONO AFA6 turbine (Howden/ KK&K design)

CONTROL OF THE INDUSTRIAL STEAM TURBINE

The task of process control for industrial steam turbines is often to keep the pressure in the steam system constant on the fresh steam or exhaust steam side. This is realized by valves that continuously throttle the pressure upstream of the nozzle from the highest pressure for full load to the lowest pressure for idle, thus adjusting the steam mass flow. However, the throttling process is also associated with a reduction in the enthalpy drop Δh, which is available for conversion to mechanical work. In these heat-led processes, power generation is always dependent on the steam production of the boiler or the heat demand of the production process [6].

Throttle control

Throttle control consists of only one control valve, so that the turbine output is linearly dependent on the mass flow. During operation, the entire nozzle area is always pressurized with steam. As a result, the nozzle pressure of all nozzles is lowered in the partial load range, which reduces the enthalpy drop and consequently leads to high throttling losses.

Nozzle group control

Nozzle group control is realized by distributing the live steam to different nozzle groups, which are selectively closed in the part-load area. The nozzle area is divided into nozzle groups, each of which is controlled by an associated nozzle group valve. To reduce the output, individual nozzle groups are switched off so that the effective nozzle area is reduced. This allows an operating point in the part-load range to be approached with lower throttle losses, since the available enthalpy gradient Δh is reduced only for the respective active nozzle group. This eliminates the proportional throttling losses for the non-active groups. Typically, KK&K turbines are designed with 3 to 4 nozzle group valves.

Comparison of the control concepts

Figure 2 shows the steam consumption versus turbine power. Clearly visible is the significantly lower steam consumption at partial load with the same power for the nozzle group control. At full load, the additional valve losses are evident in the slightly higher steam consumption for the nozzle group control. The valve points can be adapted to the main design points of the turbine system by specifying nozzle areas for each nozzle group.

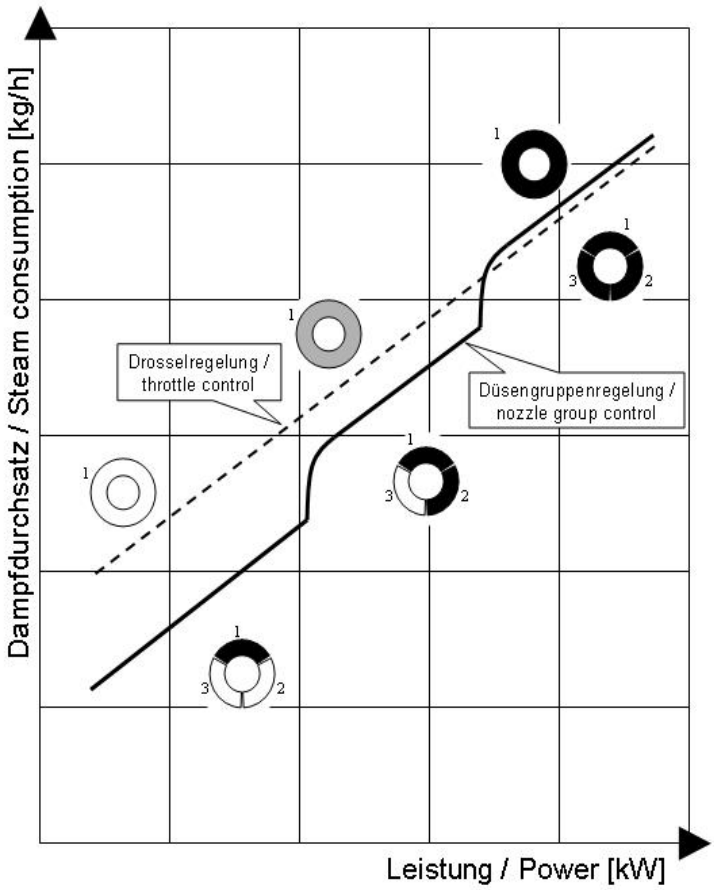

Figure 2: Steam consumption diagram for throttle control and nozzle group control

ANALYSIS OF THE STEAM TURBINE OPERATION

The analysis of the steam turbine operation is based on the operation data recording over one year. The so-called annual duration curve indicates the cumulative duration of various energy flows, usually in the form of heat demand or steam mass flow. The pattern of the annual duration curve for a heat supply plant is typically similar to the example shown in the left part of Figure 3. The plant is operated for a few hours at very high or very low mass flow, but for most of the annual hours the turbine operates in the medium load range.

Another type of representation is the frequency distribution of certain fluid quantities (Figure 3 right). This shows clearly how many hours per year a particular mass flow demand is present. The frequency of the occurring quantities usually follows the Gaussian normal distribution. In a mathematical sense, this is the derivative of the inverse function of the annual duration curve.

Also of interest is the time course of the quantities of steam occurring. In this way, stronger load changes - see example on cyclic industrial operation - or seasonal fluctuations - see summer-winter operation - can be recognized and the turbine operation can be optimized for these conditions.

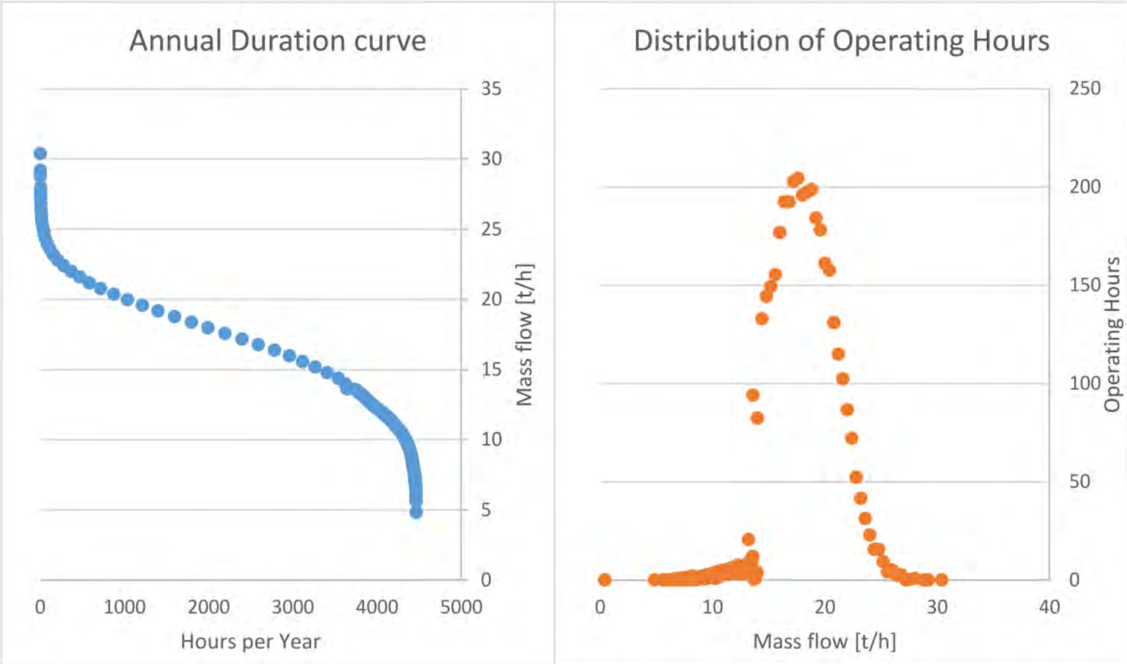

Figure 3: Exemplary example of the annual duration curve and the operating hours distribution

Example of summer-winter operation of a cogeneration plant

The waste heat from turbine operation is often used to provide local heating for a neighborhood. Due to the lower outdoor temperatures, the minimum steam consumption is also higher in winter than in summer. This results in two different load cases in summer and winter operation. Typically, the first group of nozzles is optimized for the customer's minimum load. The customer's experience shows that if the turbine is optimized only for winter operation, then for the load case summer operation either the minimum load - without a customer - must be kept at the winter level, with corresponding economic losses, since the excess steam in this case is often discharged into the atmosphere. Or the turbine must even be switched off completely in summer. Conversely, if the turbine is optimized for the base load in summer, corresponding losses in power generation in winter must be expected.

This correlation makes it necessary to switch the nozzle opening sequence. Thus, one valve can be optimized for the summer minimum load and another for the winter minimum load. When the days become warmer, the opening sequence is changed and the valve that opens first is adapted to the lower heat load. This ensures that the base load is always covered by the first valve.

Example: Strongly cyclic industrial operation

Figure 4 shows the daily load curve of an exemplary industrial steam turbine. The significant hourly changes in the amount of steam supplied to the turbine require a fast and flexible response from the turbine, but also pose the challenge of achieving high partial load efficiency at as many load points as possible. To this end, the nozzle area exposed to steam must be adapted as variably as possible to the available steam quantity. The variability of the turbine can be increased on the one hand by designing the nozzle ring in such a way that the individual nozzle groups are dimensioned differently. To take advantage of this variability, the opening sequence must be freely selectable so that the nozzle area optimized for the respective operating point can be pressurized. Thanks to the new dynamic valve coordinator, any combination of valve openings can be used in the future. Together with an operating data acquisition system that determines the best valve combination for the current operating point, this results in maximum flexibility and a significant increase in power yield.

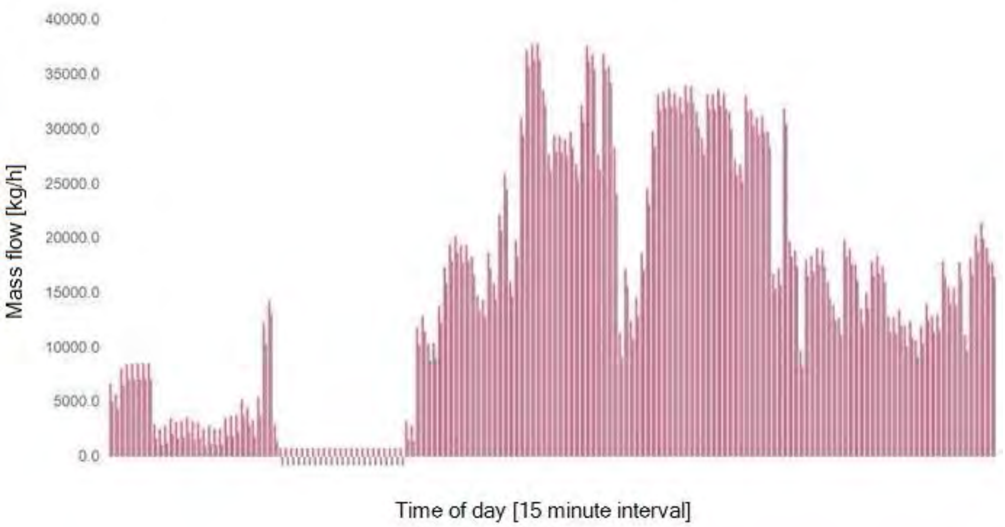

Figure 4: Example of a heavily cyclic industrial operation

Automatic analysis in Howden Uptime

Up to now, the collection of operating data has usually been carried out at the customer's site, who in turn makes this data available for evaluation on request. The evaluation itself is carried out manually by an engineer. In the future, this task will be handled by means of automatic analysis in Uptime. The acquisition tool continuously analyzes the operating data and identifies the potential for optimization and the resulting increase in turbine performance.

POSSIBILITIES FOR ADAPTATION TO CHANGED PROCESS DATA

Industrial steam turbines are always part of an overall plant. When planning a plant, the operator makes assumptions in order to be able to dimension the individual plant components and the turbine. For a variety of reasons, these assumptions can sometimes deviate very significantly from the subsequent mode of operation, and it becomes necessary to adapt the turbine to the - now known - process data after a certain period of operation. Basically, 4 scenarios are conceivable.

1. the turbine runs as planned in the design phase.
2. the steam parameters have changed compared to the planning.
3. the actual amount of steam available is lower than planned
4. the actual amount of steam available is (temporarily) greater than planned

In the first case, no changes to the machine are necessary.

In the second case - especially in the case of strongly fluctuating parameters - it is sufficient under certain conditions to adjust the interlock values of the turbine, provided that safe operation can be guaranteed with the adjusted values. However, if the deviation from the design data becomes too great and the planned thermodynamic parameters cannot be maintained, especially in continuous operation, mechanical adjustments to the turbine are often advisable. Usually, it is sufficient to replace the valves and the nozzle ring, sometimes the impeller must also be replaced.

If the actual amount of steam available is lower than initially assumed (case 3), there is no acute need for action. However, a detailed evaluation of the operating data is often worthwhile, as significant increases in performance are possible. The nozzle area is adapted to the steam quantity: Reducing the nozzle area increases the nozzle pressure and thus the usable enthalpy drop for the low steam quantity and thus the output power and efficiency of the turbine. Often, this modification pays for itself after only a few years.

If the actual steam quantity is greater than assumed in the design, the same conversion parts are required from a fluidic point of view as in case 3 with a lower steam quantity. In addition, however, the mechanics and strength of the turbine must be checked, since the design capacity of the turbine is exceeded by the increased mass flow. The bearing and gearbox loadings, as well as the oil supply and generator, must be rechecked to ensure that, despite the increased power, the machine operates within its operating limits. Depending on the results, various additional rebuilds or a change in boiler parameters may be required.

These scenarios are used whenever the main design point changes. However, if the operation varies greatly over the course of the year, a mechanical conversion does not make sense.

NEW SMART EFFICIENCY INCREASE THROUGH THE DYNAMIC VALVE COORDINATOR

In the dynamic valve coordinator, the advantages of the nozzle group control mentioned in the previous chapters are combined with the advantages of a conversion to lower steam quantities. The thermodynamic principles and the general procedure on which the dynamic valve coordinator is based are described in the patent specification [7]. With the intelligent real-time operation optimization, it is possible to optimize the turbine behavior in the part-load range in such a way that, depending on the dimensioning of the turbine and the operating point, considerable increases in the generated power can be realized [8].

The areas marked in yellow in Figure 5 correspond to the operating areas with optimization potential. In this example, with a steam consumption of 20t/h, the output can be increased from 300 kW to 450 kW. This increase in output can be achieved during operation by control optimization and smart control of the individual nozzle groups alone.

With the dynamic valve coordinator and selective control of individual valves, it is also possible to swap groups during operation. Since the nozzle groups - and thus the nozzle areas - are dimensioned differently, swapping the valves results in combinations of cross-sectional areas that are optimal for different steam quantities. A simple calculation example: If nozzle group 1 is optimized for 5 t/h of steam and nozzle group 2 for 8 t/h, then serial opening results in optimized operating points for 5 t/h and for 13 t/h. Optimum operation can also be achieved for 8 t/h by exchanging the valves.

The major challenges for implementing the optimized control behavior lie in the determination of the optimum switchover point in-situ and in the execution or initiation of the switchover.

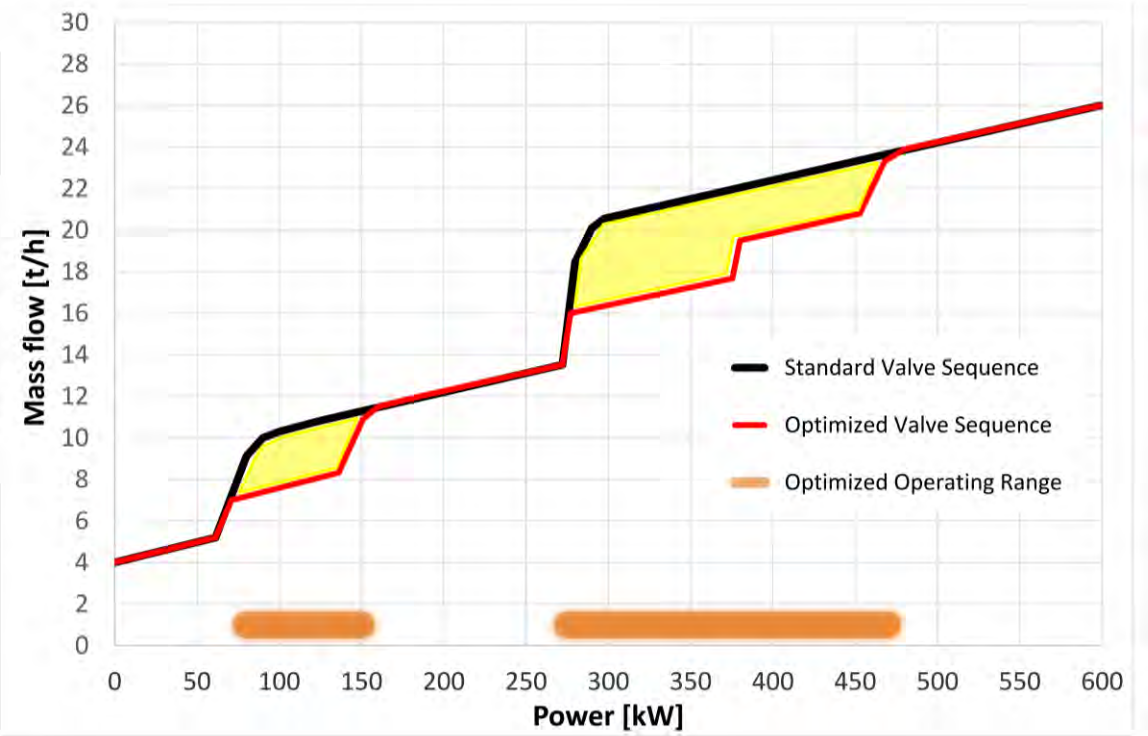

Figure 5: Comparison of turbine efficiency with serial opening and with dynamic valve coordinator

SIMULATION OF THE PARTIAL LOAD BEHAVIOR AND THE CONTROL INTERVENTIONS.

A hardware-in-the-loop (HIL) simulation programmed in LabView was applied for the design and optimization of the control behavior. In this simulation, the hardware turbine controller is integrated into a simulation of the steam turbine. For this purpose, the sensor inputs and outputs of the controller are connected to the simulation computer via an electrical interface. The simulation computer calculates the behavior of the steam turbine in real time, which allows the use of steady-state turbine equations when assuming a quasi-stationary operating point. These are based on literature sources such as aus der Wiesche [9], Traupel [10], [11] or Dietzel [12] and are also empirically verified. Thus, the highest accuracy of the turbine model is achieved [13].

Figure 6 shows the schematic structure of a hardware-in-the-loop test system.

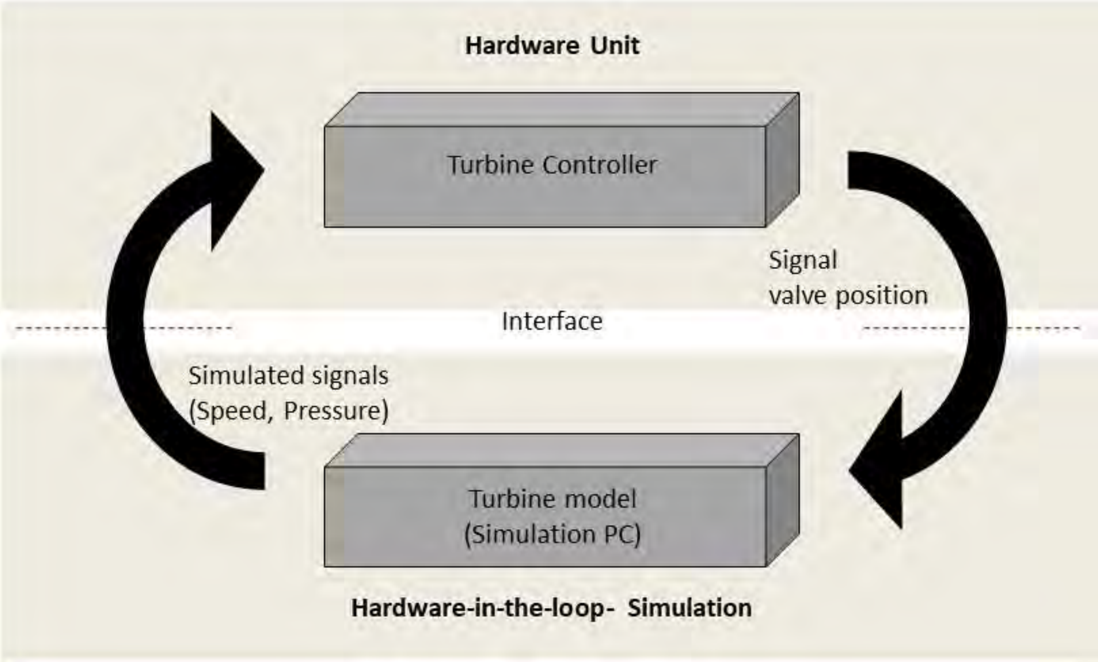

Figure 6: Schematic structure of a HIL simulator for steam turbine controllers [13].

Determination of the optimum switchover mass flow

Up to now - as explained above - the valves opened one after the other in a specific sequence during operation. With serial valve control, the power behaves in a basically linear manner over the valve stroke. However, if one valve is fully open and another takes over control, the idle stroke of this valve must first be exceeded (no power increase in the lower valve opening range) before there is again a linear relationship between power and valve stroke (see also Figure 5).

Valve replacement opens up additional optimization potential in the part-load range. From the combinatorics, the number of sensible opening variants Z_v results from the number of nozzle groups Z_{DG} and the respective number of nozzle groups with equal numbers of nozzles Z_{Si}.

$$Z_v = \frac{Z_{DG}!}{Z_{s1!} * Z_{s2}!}$$

Example 1:
The turbine has Z_{DG}= 3 nozzle groups, of which all are different (13-8-5)

$$Z_v = \frac{3!}{1} = 6$$

Example 2:
The turbine has Z_{DG}= 3 nozzle groups, two of which are the same (12-12-8)

$$Z_v = \frac{3!}{2!} = 3$$

Example 3:
The turbine has Z_{DG}= 4 nozzle groups, two of which are the same (13-13-8-5)

$$Z_v = \frac{4!}{2!} = 12$$

Switching takes place if more power can be generated with another variant at a certain operating point (= mass flow) than with the variant that is currently engaged. The optimum switchover point can be defined via the steam quantity at which both valve variants generate the same internal power. If the previously calculated steam quantity is exceeded/undershot, switchover takes place.

To determine the optimum switching points and to optimize the hardware controller, the turbine HIL-simulator model is applied.

First, the internal power of the turbine is calculated for the smallest valve when the valve is fully open. Then, for the next largest valve, this power is specified and the mass flow at which this power is achieved is determined. This point is the optimum switching point. Since the nozzle area associated with the valve is larger than for the smaller valve, the required mass flow will be larger. If the mass flow is greater than the calculated mass flow, then more power can be generated with the larger valve opening alone than with the small valve fully open and the larger valve partially open. If the mass flow is less, the smaller valve alone is sufficient and most efficient. In this way, the switching points are calculated for all valve exchange combinations and defined based on the mass flow. The mass flow passed through and the valve lift of the respective valve represent a linear function, differing from valve to valve only by the maximum mass flow and the straight line slope. The straight line equation for each valve is known and can be implemented in the controller. During operation, the current mass flow/operating point is then recorded via position feedback from the valves and a selection table is used to compare whether the turbine is operating at the optimum operating point. In the event of a deviation, the system switches over.

Hysteresis control

As described above, the optimum switchover point can be calculated and stored in the controller accordingly. There are two basic approaches to avoid frequent changeovers. On the one hand, a time specification after a changeover can be used to prevent a renewed valve changeover within a period of time. On the other hand, it is possible to define a hysteresis based on the optimum switchover point within which no switchover takes place. Instead of one changeover point, there are then two changeover points; the changeover characteristic is shown in Figure 7. If the operating range lies between the two hysteresis points, no changeover takes place; the changeover is only activated when one of the hysteresis points is exceeded or undershot. This compensates for operational fluctuations to a certain extent. As a compromise to this, the turbine efficiency within the hysteresis points is worse than with direct switching at the optimum point. In addition, the two different hysteresis points result in a different power minimum at the respective switchover point. Therefore, time-limited switching is usually preferable to hysteresis control.

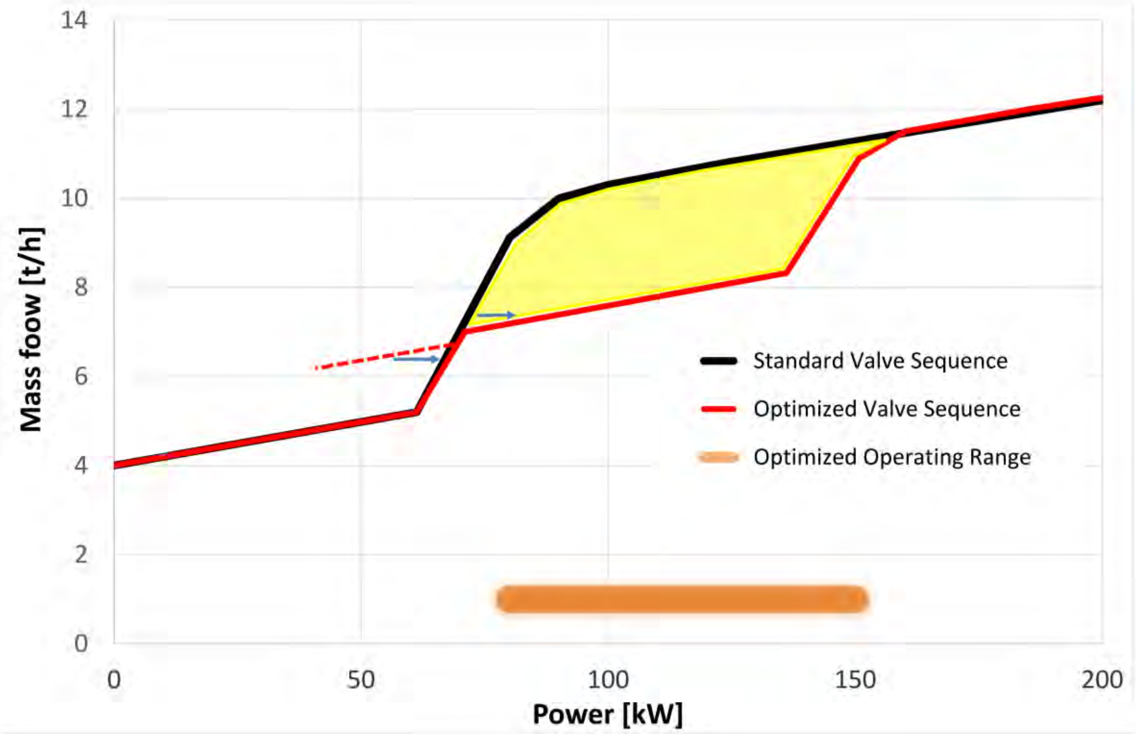

Figure 7 Hysteresis control of the changeover function

BEHAVIOR OF THE TURBINE DURING VALVE CHANGEOVER

In the changeover process, more than one valve is in operation. As a result, the turbine behaves similarly to the throttle control shown in figure 2 and the output decreases while the mass flow remains constant. The inlet and outlet pressures are kept constant by the control system, which is essential in particular for process steam cycles and to protect the boiler from overload or to protect the turbine from water ingress. Due to the specifications from the grid connection directives VDE4105, VDE4110 and VDE4120, a change in the electrical power output must not exceed a certain gradient (typically 0.66% nominal power per second) [14][15].

Simulation of the timed valve transition

Figure 8 shows the results of the simulations for the valve changeover between 2 valves. The switchover times of the first valve were varied between 90, 150 and 300 seconds. The aim of the simulations was to find out whether the switchover time has an influence on the control accuracy and the minimum power at the transition point.

The position of valves 1 and 2 can be seen in the diagram at the bottom. The top left diagram shows the mass flow through the turbine. In the top right diagram, the power curve over time is shown.

As the simulation began, the mass flow was increased. First, valve 1 is fully opened and the power increases from 200 to 400 kW (second 0 to 100). From second 100, valve 2 also opens, but without contributing significantly to an increase in coupling power. At second 280, the automatic valve changeover is initiated. After the switchover, valve 1 is fully closed and the entire mass flow is directed through valve 2. The power in the fully controlled state is approx. 700 kW, which is significantly higher than the power at the start of the changeover. It is easy to see that the minimum power is independent of the changeover time. The minimum for all changeover procedures is approx. 180 kW.

The power minimum in turn has an influence on whether a switchover can be realized, since the protective function "reverse power" of the generator would intervene in the event of insufficient power in grid-parallel operation. In real plant operation, the possible buffer effect of the plant must also be taken into account. In a larger plant, the pressure in front of the turbine drops only slowly, so that in reality an increased amount of steam can flow through the turbine at times and the power does not drop as much as calculated in the simulations. This hypothesis can only be verified by testing in the plant and the result will vary from plant to plant.

In the simulations, the control accuracy was slightly dependent on the duration of the switching process. Thus, the control behavior of the valves could be improved by a slower changeover.

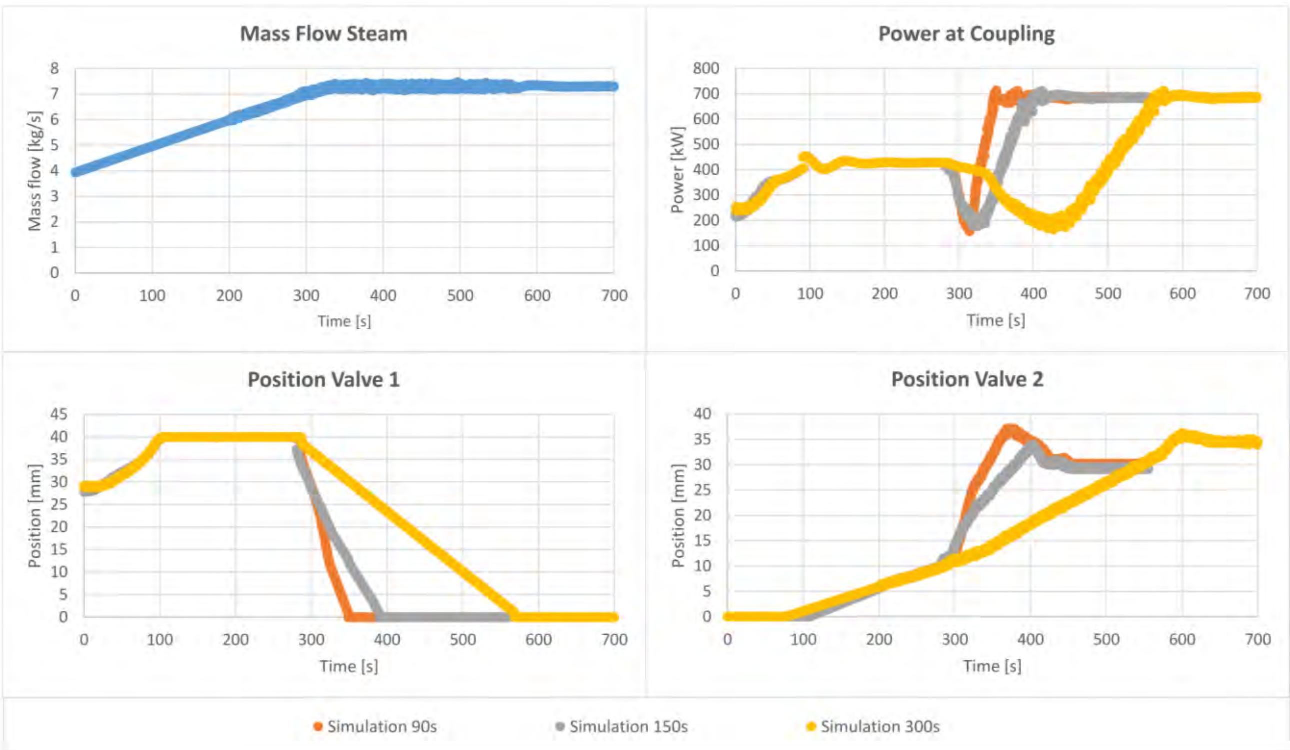

Figure 8: Simulation of the time-controlled switching function - comparison of different closing times

EXPERIENCES FROM INSTALLATION, COMMISSIONING AND OPERATION

After good performance was demonstrated in simulations of the controller in the HIL system, a customer was won for an initial installation of the part-load optimizing controller. To prove the function, an operating data acquisition system and a performance monitoring system were also retrofitted.

The optimum switching points were calculated in advance and simulated again specifically for the turbine. A number of tests were initially run with these preset parameters. The switchover itself ran successfully and due to the buffer effect of the turbine, the real power drop was, as expected, lower than predicted in the simulation. However, due to deviations of the on-site thermodynamic operating data from the turbine design data, the switchover points were initially not optimal in terms of turbine efficiency. This necessitated a more detailed analysis. The nozzle pressure measurement implemented in Howden-Uptime enabled the exact performance of the individual nozzle groups in the "as built" condition to be determined in the first step, and on the basis of this the switchover points were recalculated directly at the customer's site. These were stored in the control system and tested again. With the updated values, the partial load efficiency was once again significantly increased.

SUMMARY AND OUTLOOK

This paper examines a method for improving thermodynamic performance at part load, the effects of different control valve strategies, and shows a way to optimize the steam turbine under part load conditions during operation by using a timed valve switching method.

Modern steam turbines must operate over a wide range of operating conditions and thus with different steam mass flows. To ensure efficient operation, a valve and nozzle design is usually used that divides the inlet into several circulating sections ("nozzle group control"). The sequence of opening of these groups is usually determined by design.

A control function is provided that allows switching between segments while the turbine remains in operation, and how this affects generator performance is shown. Based on simulation results, the transition timing and actual set point for valve switching are discussed. These simulations are based on cloud-based data analysis techniques and it is shown that they are a promising tool to study the interaction of the turbine with the plant and to identify the operational behavior in the real process. With these methods, it is possible to contribute to the increase of efficiency and optimization of the overall process with low resource input and reasonable financial expenditure.

With the successful first commissioning of the dynamic valve coordinator, Howden is demonstrating a way to optimize existing plants and processes and meet the challenges of reducing global CO_2 emissions. This work has shown that even in the technology of a steam turbine, which is often considered mature, there is still potential for improvement that should not be underestimated. Thanks to CFD calculations and computer-aided optimization of blade geometry, the steam turbine itself is indeed a nearly mature product, and further improvement of its overall efficiency would require an immense financial outlay. Instead, the potential lies in the continuous analysis of operating conditions, the optimization of existing techniques and the interaction of plant components. The successful implementation of the dynamic valve coordinator in an existing plant represents a significant contribution to the optimization of plant operation and CO_2 reduction and thus supports operators in achieving their ecological goals.

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