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Configuration Analysis of Mainsteam Stop Valve-Governor Valve in a Supercritical Steam Turbine Using Computational Fluid Dynamics to Minimize Throttle Losses

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Abstract. *Quantification of losses and prediction of the technical feasibility of a proposed modified system through computational fluid dynamics (CFD) numerical analysis constitute the focus of this study. The system involves a configuration of four sets of Main Steam Stop Valves and Governor Valves (4MSVGV), which are known to operate at low valve openings (<30%) to serve normal load demand. Using CFD modeling, the proposed 2MSVGV configuration demonstrates the ability to serve all load ranges with a general requirement of GV lifting below 82mm. The principles of entropy generation and exergy destruction are employed to quantify throttle losses. 2MSVGV valves configuration, consistently exhibits lower exergy destruction than the 4MSVGV configuration. Further review of the 4MSVGV configuration reveals that the low-opening 4GV contributes in higher exergy destruction compared to the MSV that indicate higher throttle lossess. The total average improvement in exergy destruction rate with the modified 2MSVGV system is about 2.05MW. The validated CFD-based analysis method in this study facilitates predictive evaluation of system behavior, offering a practical means to investigate various operational issues without resorting to expensive and high-risk experimental trials.*

Keywords – *CFD; Throttle Losses; Governor Valves; Entropy; Exergy.*

INTRODUCTION

Improving the efficiency of energy conversion systems to achieve optimal utilization of energy resources remains a persistent and increasingly significant challenge in response to the growing global energy demand. Although the prevalence of Coal-Fired Power Plants (CFPPs) is projected to decline or face operational limitations in the future, the steam-based energy conversion component, such as steam turbines, is expected to retain a significant role in harnessing various emerging renewable thermal energy sources, including nuclear, geothermal, and solar thermal power. This highlights the continuing strategic relevance of steam turbines in the transition toward more sustainable energy systems.

For power plants that are already integrated into the national grid, enhancing the efficiency of large-capacity steam turbines through process optimization within the existing equipment configuration offers a better approach compared to extensive design modifications or component replacements. This method offers cost-effectiveness, operational continuity, and implementation feasibility, while efficiency improvements are primarily achieved through process modifications that minimize energy losses in the power conversion cycle.

The present study investigates the potential of modifying the control process of the main steam stop valve (MSV) and governor valve (GV) with the aim of minimizing throttling losses. Adjustments in the regulation of steam flow are anticipated to improve the thermal efficiency of the system and thereby contribute to an increase in the generated electrical load. Consequently, the findings of this research are expected to provide a tangible contribution to strategies for improving energy efficiency in steam-based power plants, both within the framework of energy transition and the optimization of existing technologies.

Table 1. Relationship between load, main thermodynamic parameters, and GV opening

Load	400 MW	500 MW	600 MW	700 MW	800 MW	900 MW	950 MW
GVOpening (%)	22.3	34.6	27.5	26.7	24.6	26.7	25.7
Mass Flow (Kg/s)	346	407	481	569	637	735	774
Pressure (Mpa)	11.28	12.09	15.10	19.69	19.69	22.06	23.09

The turbine studied utilizes four MSVGV assemblies to direct steam to the high-pressure (HP) turbine. Operational records show that the governor valves (GV) operate at low opening during normal operation, as shown in Table 1 that outlines the steam admission configuration of the investigated turbine, noting the typical operating conditions of its governor valves.

The study by Yoo Y, Suh K (2011), on paper with title: Engineering analysis of mass flow rate for turbine system control and design investigated the characteristics and modifications of turbine stop and governor valves through experimental and numerical analysis, demonstrating that a mass flow methodology simplified both programming and experimental procedures for characterizing turbine systems[1]. By that paper result, this study was emerged for typical system with supercritical steam. In operational practice, with a fixed setpoint temperature and sliding pressure arrangement in response to the load setpoint, the magnitude of the mass flow directly determines the amount of energy that can be converted into electricity by the turbine-generator system. In this study, the numerical analysis was validated using actual operational data. Furthermore, numerical analysis was subsequently conducted on a modified MSVGV configuration, and quantify the reduction in losses achieved by both configuration.

METHODS

The main challenge in process modification in large system is lies in demonstrating the practical applicability of the proposed system and validating the approach of solution methods employed to ensure that the predicted process outcomes will closely align with actual results. The inherent complexity of valve geometries poses a significant challenge to analytical approaches, thereby making it difficult to acquire data on specific operational parameters at desired spatial locations. On the other hand, numerical solution approach, which allow detailed data extraction at various points thus enabling further in-depth analysis, depend on the accuracy of the input parameters and the solution method to provide more accurate results. Figure 1, depicting the layout of MSVGV configuration to service high pressure (HP) turbine.

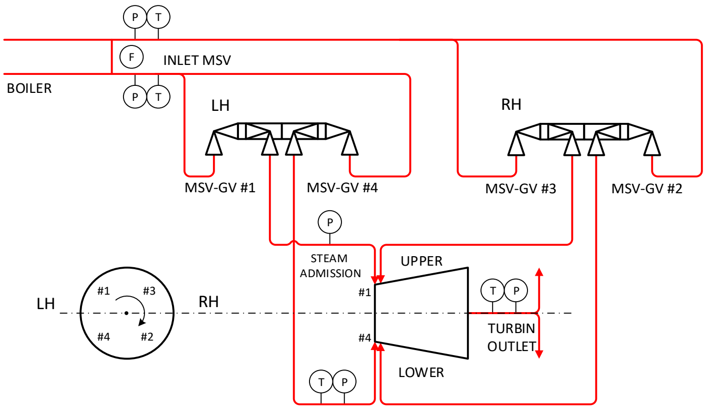

Figure 1. MSVGV and HP Turbine configuration

This study aims to reduce throttle losses of steam turbine MSVGV by modifying its configuration from 4 assembly to become 2 assembly with the analysis method as depicted in figure 2, with initial step is problem acknowledgment, followed by:

1. Data collection for geometry development, supplemented by daily operational data, and reference books;
2. Geometry modeling using Ansys Fluent software (fluent with fluent meshing) with GV lift (opening) basis for the original 4MSVGV configuration;
3. The results of the numerical analysis process are compared with actual operational data as a model validation stage with standardized software variable input;
4. Variable input is used for configuration analysis of the 2MSVGV assembly with Mass flow rate basis at a certain load to obtain the required valve stem height.
5. Entropy generation and exergy destruction analysis for the 4MSVGV and 2MSVGV system configuration.

The new GV lift position for 2MSVGV will provides a reference for the new GV control system input.

RESULTS AND DISCUSSION

A. Governing Equation

Continuity and energy balance equation

The main task of the MSVGV is the control of mass flow quantity using defined parameters to achieve the setpoint dictated by the turbine-generator's speed-load controller. The fundamental equations governing this process are based on the continuity equation (1) and the energy balance equation (2).

$$\frac{dM}{dt} = \sum_{i=1}^{n\,stream} \dot{m}_i \tag{1}$$

$$\frac{d}{dt}\left(U + \frac{1}{2}M \cdot v^2 + M \cdot g \cdot h\right) = \sum_{i=1}^{n\,stream} \dot{m}_i \cdot \left(\hat{H}_i + \frac{1}{2} \cdot v_i^2 + g \cdot h_i\right) + \dot{Q} + \dot{W} \tag{2}$$

Under steady-state, the change of system internal (U) and external (kinetik+potential) energy with time are zero. Given negligible rates of energy at the system outlet due to kinetic and potential energy, and most real-world throttling processes in control valves are assumed to be approximately adiabatic ($\dot{Q} \approx 0$) because they happen rapidly and both valves are insulated to prevent significant heat exchange with the surroundings [2], it follows that:

$$0 = \dot{m}_1 \hat{H}_1 - \dot{m}_2 \hat{H}_2 \tag{3}$$

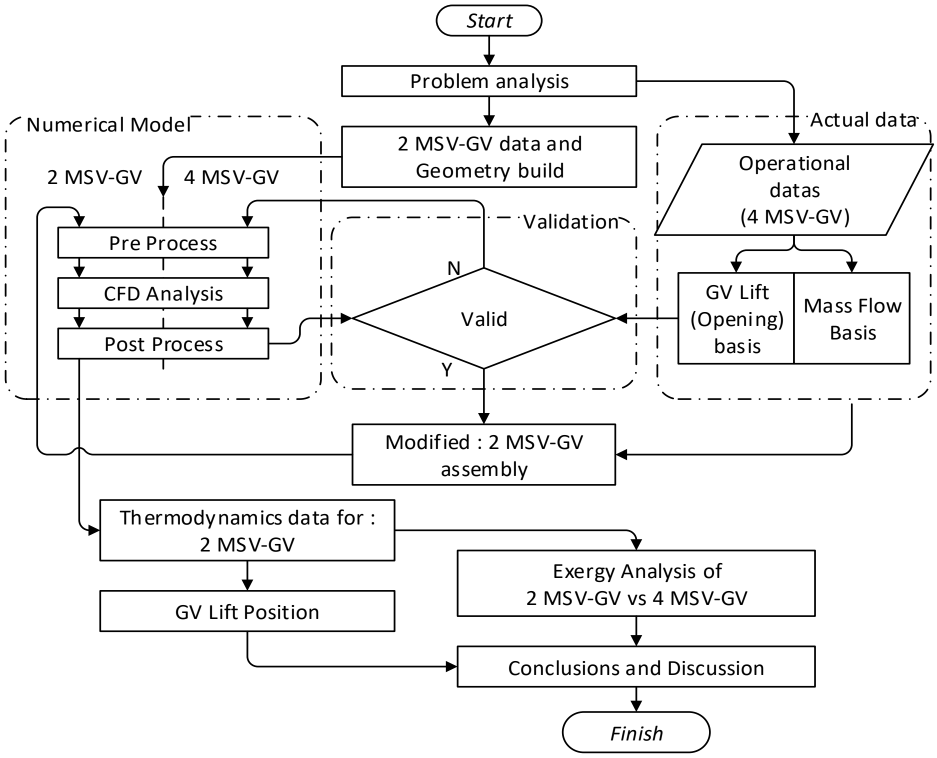

Figure 2. Flow chart of analysis methods framework

The throttling effect in control valves refers to fluid flow through a reduced geometry under isenthalpic conditions, meaning no heat transfer occurs and no work is performed, also known as an isenthalpic process. The fluid passes through the restriction from side 1 to side 2, experiencing a pressure drop and an increase in volume; however, throughout this flow, the enthalpy remains constant, such that $\hat{H}_1 = \hat{H}_2$ [3].

Based on Equation (3), three primary parameters determine the amount of steam energy entering the turbine: steam enthalpy (defined by pressure and temperature) and mass flow rate. Within the distributed control system the responsibility for achieving these parameters is distributed between the turbine master control and the boiler master control (boiler load demand). The boiler master control is tasked with supplying a steam at the target flow, temperature and pressure to MSV inlet. The temperature setpoint is maintained constant at a defined value (565°C for all load targets) through combustion control, heat distribution management, and desuperheater spray control.

Entropy generation

Governor valve modulation induces a throttling effect across the valve, as an unavoidable irreversible process. The associated throttling loss can be quantified by the differential pressure measured upstream and downstream of the control valve. A fundamental characteristic of such valves is the inverse relationship between the valve's effective flow area and the resultant pressure drop; a smaller opening area leads to a greater pressure drop and, consequently, increased energy dissipation. Equation (4), derived from the principle of entropy balance where S_{gen} represents the magnitude of entropy generation, explain this phenomenon [4].

$$\frac{dS}{dt} = \sum_{i=1}^{n\,stream} \dot{m}_i \hat{S}_i + \sum_{all} \frac{\dot{Q}}{T} + \dot{S}_{gen} \tag{4}$$

With steady state and adiabatic conditions (no heat transfer), then

$$\dot{S}_{gen} = \dot{m} \cdot (\hat{S}_2(T_2, P_2) - \hat{S}_1(T_1, P_1)) \tag{5}$$

In the context of compressible fluids, variations in valve opening also affect flow velocity, pressure distribution, as well as expansion and throttling phenomena along the valve passage. Conversely, when the valve opening is smaller, the flow area narrows, leading to a reduction in mass flow rate and an increase in pressure losses due to intensified throttling effects. This condition not only decreases the efficiency of energy conversion within the system but may also induce significant local turbulence, particularly under non-partial load (low-load) operating conditions [5].

Exergy Analysis

The application of exergy balance to a control volume of assembly of MSVGV operating at steady state is corespond to above chapter assumption (W = 0, Q = 0, Kinetick and potential energy are neglected). Exergy balance equation is given in equation (6) considering exergy flow entering ($E_{x,in}$), exiting, ($E_{x,out}$), and exergy destruction ($E_{x,dest}$), within the control volume due to irreversibility.

Table 2. Boundary condition of model

Geometri	Boundary Type	Value					Unit
Load	n/a	951	720	552	639	855	MW
Inlet	Pressure Inlet (MSV)	23.13	17.42	13.83	15.22	21.63	MPa
Outlet	Pressure Outlet (GV)	21.26	16.07	12.47	14.25	18.92	MPa
Reynold Number	Inlet (MSV)	1959410	1053131	646427	797934	1687719	-
Turbulennt Intensity	Inlet (D_H= 0.262m)	2.615	2.826	3.004	2.926	2.665	m

$$E_{x,in} - E_{x,out} - E_{x,dest} = 0 \tag{6}$$

The specific exergy [6], the rate of physical exergy flow per unit mass of fluid at a point can be expressed as equation (7), where h_o, s_o, T_o is the enthalpy, entropi and temperature (in K) of the fluid at dead state.:

$$\varepsilon_x = (h - h_o) - T_o(s - s_o) + \frac{{V_i}^2}{2} + gh_i \tag{7}$$

As velocity and elevation can be neglected, so the physical exergy per unit mass equation becomes:

$$\varepsilon_x = (h - h_o) - T_o(s - s_o) \tag{8}$$

The destruction of exergy power as a result of throttling of the steam passing through the MSVGV assembly is calculated by [7]:

$$E_{dest} = \ \varepsilon_x \cdot \sum_{i=1}^{n} \dot{m}_i \tag{9}$$

And corespond to equation (2), where $\dot{W} = 0, \dot{Q} = 0$, thus, the rate of exergy destruction that directly measures the energy that is "lost" or no longer available to do useful work due to the irreversible throttling process, is:

$$\dot{E}_{x.dest} = \dot{m}(\varepsilon_{x,1} - \varepsilon_{x,2}) = \dot{m}[(h_1 - h_0) - T_0(s_1 - s_0) - (h_2 - h_0) + T_0(s_2 - s_0)] \tag{10}$$

$$\dot{E}_{x.dest} = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) \tag{11}$$

Combine with equation (2) above:

$$\dot{E}_{x.dest} = \dot{m}[- T_0(s_1 - s_2)] \tag{12}$$

B. Numerical Study

Numerical approach model validation and code verification are important aspects of a CFD method. However, they are not the only ones that have to be considered in order to achieve high-quality CFD results. One is by following guidelines, “To offer roughly those 20% of the most important general rules of advice that cover roughly 80% of the problems, the technical content for the guidelines is limited to single-phase, compressible and incompressible, steady and unsteady, turbulent and laminar flow with and without heat transfer and relevant to many mechanical, aeronautical, automotive, power, environmental, medical and process engineering applications”[8]. Thus, if relative factors were defined between numerical analysis results and experimental results in the early stage of development, it should be possible to produce the characteristic curve and to predict the actual performance of turbine systems wholly by numerical method without additional experiments [9].

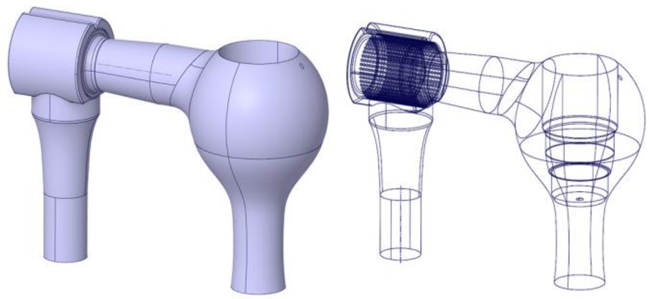

Figure 3. MSVGV volume control as a steam fluid path

Geometri Modelling

The modeled geometry defines the steam flow path from the MSV inlet header to GV diffuser, upstream of the first turbine nozzle. Key components integrated into this geometry are inlet of Main Stop Valve (MSV), MSV internal section (with strainer), MSV throat, MSV diffuser, Governor Valve (GV) internal section, GV throat, and GV diffuser.

Due to the system is considered no heat transfer, the complexities of modeling and calculation setup are reduced. This less-complex system makes the setup of viscosity model and solution method relatively straightforward and efficient from a computational effort standpoint. Figure 3. shows the layout geometry of one of the 4MSVGV assembly sets and the position of the data collection area and the position of the data collection area.

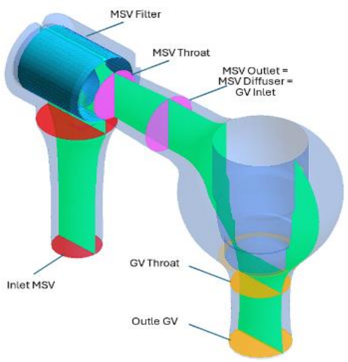

Figure 4. Area of data extraction

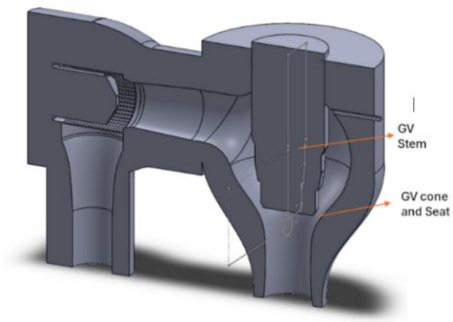

Figure 5. GV lift position as mass flow controller

Turbulence Model

Turbulent flows feature fluctuating velocity that mix momentum and energy, but their small scale and high frequency make direct simulation computationally impractical. The Reynolds-Averaged Navier-Stokes (RANS) equations model all turbulence scales, focusing on mean flow quantities, and can yield steady-state solutions for steady mean flows[10]. The RNG k–ε model takes on a larger of the turbulent viscosity values with a slight gradient than the standard model and, inversely, takes on a smaller amount with a steep gradient. Thus, the RNG k–ε model appears to be more suitable for conducting numerical analysis of bent pipes and complex contours [1]. The RNG k-ε turbulent model was selected for subsequent studies for reasons of computational effort efficiency and accuracy[11] of this work. Table 3 provides detailed settings applied to the analysis after obtaining settings validation. The advantage of using numerical analysis is that it can determine the location of data to be extracted from the model quickly and repeatedly, as shown in Figure 4.

Table 3. Numerical analysis configuration setup

Model Preset	Selection
- Energy Equation	Enabled
- Solver	Pressure based, absolute formulation, steady state
- Near-Wall Treatment	Standard Wall Function (No Slip)
- Gradient option	Least Square Based
- Heat transfer	Isothermal
- Viscous Model	k-ε (2 equation) RNG
- Fluids	Ideal Gas

An ideal gas was selected as the working fluid in the software model to maintain consistency between the numerical analysis and subsequent loss calculations, the difference in throttling efficiency between real and ideal gases is less than 1.037% provided that the initial pressure is below 30 MPa [12]. This choice is both efficient and appropriate for simulating flow behavior, offering a balance between accuracy and computational cost. At the other hand, according to Luo and Wang,. This is because the heat exergy generated by throttling contributes only marginally to the total exergy when compared to the exergy at the initial throttling pressure. Therefore, the use of an ideal gas is recommended for calculating throttling efficiency.

Boundary condition

The boundary conditions for this simulation were defined based on software best practices outlined in the user manual, and relevant preceding publications. The MSVGV system was modeled under the assumption of an isentropic throttling process, with no heat transfer. This simplification allowed for the calculation of total steam densities throughout the system using the initial total pressure and temperature, with these values subsequently held constant according to the ideal gas law.

The Reynolds Number was determined using Equation (6), and the turbulence intensity for fully-developed flow can be estimated from the following formula derived from an empirical correlation for pipe flows (7) [13]. Table 3. Shows boundary condition setup.

$$R_e = \frac{inertial\ force}{viscous\ force} \tag{6}$$

$$I = 0.16(Re_{Dh})^{-1/8} \tag{7}$$

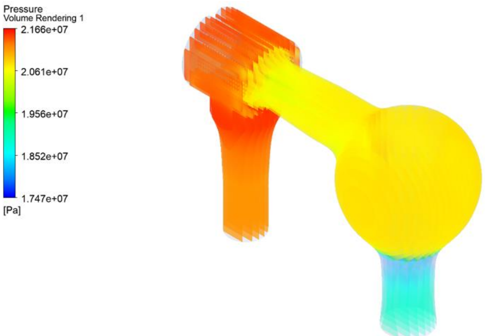

Figure 6. Pressure distribution with volume rendering

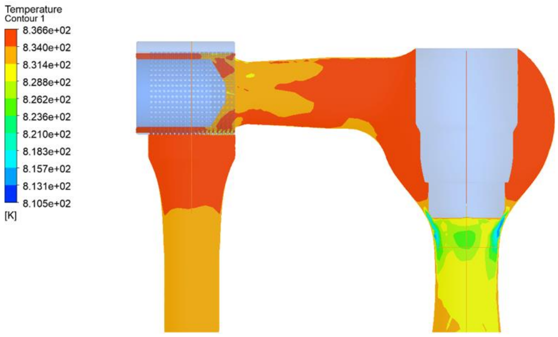

Figure 7. Temperature Contour

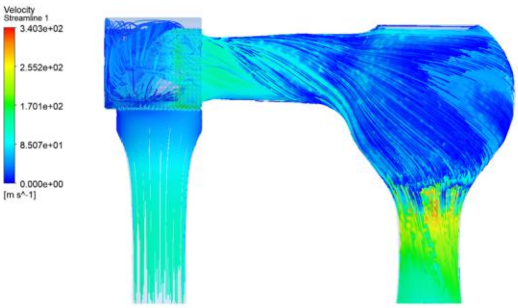

Figure 8. Velocity stream line (left side)

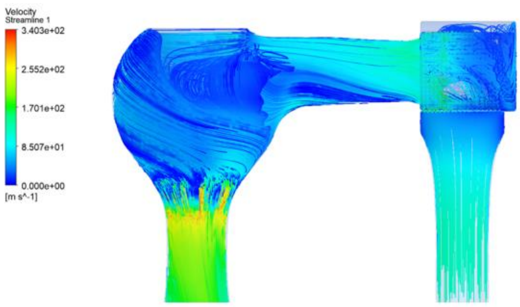

Figure 9. velocity stream line (right side)

C. Validation of the Model and Modification Modelling of 2MSVGV Assembly

By implementing the framework shown in Figure 2, the steam mass flow (including key parameters such as pressure and temperature) in the validated model can be used to predict the mass flow through the MSVGV control volume. Gathering data from relevant experiments to be compared with the simulation results. Result comparison: Comparing the simulation outcomes with the experimental data to evaluate the accuracy and validity of the model [14].

Actual opening demand data (Table 4, Section C) is directly converted into the GV lift position (in millimeters), providing data in Table 4, Section B. In this context, the actual mass flow rate can be compared with the results of numerical analysis for the 4MSVGV modeling. Figure 5. Show GV lift as variable for controlling mass flow rate to achieve load demand. The results from modeling using CFD show a difference in mass flow rate of around 5%. This discrepancy may arise from several factors, one of which is that the mass flow rate values are derived from mass flow rate balance calculations, commencing from the boiler through to the high-pressure (HP) turbine outlet.

The same method as a study conducted by Yoo Y, Suh K (2011) for trial-and-error method is needed to determine the mass flow rate, given the stem lift in the control valve [1]. The calculation is initiated with an assumed mass flow rate at predicted position, and by trial and error to refine GV lift position to achieve desired mass flow rate. Table 4 section A, Give the final result of mass flow that close to actual data at certain load in certain GV lift positions. Figures 6, 7, 8, 9. show the typical distribution of parameters that the fluid has (855MW case).

Table 4. Collection of operational data (C) and Modelling data 4MSVGV (B) vs modified 2MSVGV (A).

Load	Mass Flow	MSV Inlet Press	MSV inlet Temp	MSV Throat Press	MSV Throat Temp	MSV Difsr. Pres	MSV Difsr. Temp	GV Throat Press	GV Throat Temp	GV Outlet Press	GV Outlet Temp	GV Lift <82mm
MW	Kg/s	MPa	K	MPa	K	MPa	K	MPa	K	MPa	K	mm
A. Modified: modelling of 2MSVGV assembly												
951	740.2	22.72	834.7	22.08	834.9	22.21	836.3	21.39	833.2	21.26	832.7	67.30
855	700.8	21.18	833.1	20.58	833.4	20.70	834.8	18.92	829.8	18.40	826.1	45.78
720	566.8	17.08	833.7	16.65	834.1	16.75	835.5	15.85	831.6	16.07	832.6	60.00
639	482.4	14.93	837.6	14.54	837.7	14.62	838.9	14.15	837.1	14.25	837.1	78.67
552	455.2	13.13	835.0	12.72	835.2	12.80	836.8	12.48	835.2	12.49	834.9	70.10
B. Modelling of 4MSVGV assembly												
951	713.6	23.01	837.3	22.88	837.2	22.91	837.4	20.52	831.8	21.26	834.3	17.77
855	691.2	21.50	835.7	21.36	835.7	21.39	835.9	18.03	827.6	18.92	833.6	15.73
720	536.0	17.35	836.4	17.24	836.3	17.26	836.6	15.56	829.2	16.09	833.1	19.80
639	465.6	15.24	840.2	15.15	840.0	15.17	840.3	13.87	835.2	14.28	837.8	21.10
552	459.2	13.73	838.6	13.63	838.5	13.65	838.8	11.95	831.7	12.47	836.7	18.78
C. Actual 4MSVGV												
951	768.0	23.07	837.8	n.a	n.a	n.a	n.a	n.a	n.a	21.26	835.0	17.77
855	707.6	21.59	836.3	n.a	n.a	n.a	n.a	n.a	n.a	18.92	834.3	15.73
720	558.4	17.36	837.4	n.a	n.a	n.a	n.a	n.a	n.a	16.07	832.8	19.80
639	493.2	15.22	840.8	n.a	n.a	n.a	n.a	n.a	n.a	14.28	837.2	21.10
552	439.6	13.74	838.5	n.a	n.a	n.a	n.a	n.a	n.a	12.48	834.0	18.78

Analysis of the modified control system, utilizing only 2MSVGV, demonstrates its capability to effectively manage varying load levels (400~1000MW) under normal operation, with the a-criterion GV Lift travelling below 82mm. The increasing overall pressure ratio for 2MSVGV configuration suggests a reduction in throttle losses in the form of drop pressure as the valve opening increases, which can be further quantified through exergy analysis.

Table 5. Exergy Calculation data

Load	Mass Flow Rate	Entropy Inlet MSV	Entropy Diffuser MSV	Entropy Outlet GV	Specific $\dot{E}_{M.s.dest}$ MSV	Specific $\dot{E}_{G.s.dest}$ GV	Specific $\dot{E}_{Totdest}$ Total	$\dot{E}_{M.t.dest}$ MSV	$\dot{E}_{G.t.dest}$ GV	$\dot{E}_{dest}$ Total
MW	Kg/s	kJ/kg/°C	kJ/kg/°C	kJ/kg/°C	kJ/kg	kJ/kg	kJ/kg	MW	MW	MW
A. *Modified: modelling of 2MSVGV*										
951	740.2	6.294	6.315	6.332	6.382	5.019	11.402	4901.5	3855	8756
855	700.8	6.336	6.357	6.404	6.383	14.051	20.434	4516.5	9943	14459
720	566.8	6.478	6.496	6.512	5.402	4.789	10.191	3016.6	2674	5691
639	482.4	6.573	6.589	6.599	4.965	2.835	7.800	2448.7	1398	3847
552	455.2	6.640	6.661	6.669	6.063	2.456	8.519	2665.4	1080	3745
B. *Modelling of 4MSVGV*										
951	728.8	6.293	6.297	6.340	1.095	12.890	13.985	840.9	9900	10741
855	691.2	6.335	6.339	6.412	1.244	21.771	23.015	880.5	15405	16286
720	536.0	6.477	6.481	6.513	1.168	9.535	10.703	652.4	5324	5977
639	465.6	6.569	6.572	6.600	0.918	8.370	9.288	452.9	4128	4581
552	459.2	6.625	6.630	6.675	1.205	13.663	14.868	529.9	6006	6536
C. *Actual 4MSVGV*										
951	768.0	6.294	6.340	n.a	n.a	n.a	13.607	n.a	n.a	10450
855	707.6	6.334	6.415	n.a	n.a	n.a	23.872	n.a	n.a	16892
720	558.4	6.480	6.512	n.a	n.a	n.a	9.716	n.a	n.a	5425
639	493.2	6.572	6.598	n.a	n.a	n.a	7.827	n.a	n.a	3860
552	439.6	6.625	6.667	n.a	n.a	n.a	12.469	n.a	n.a	5481

D. Entropy Generated and Exergy analysis

The comparison of destruction exergy between control valves in its original configuration (4MSVGV) with modified control characteristic (2MSVGV) will be analyzed to identify the effect of flow restriction level on work potential loss. Using the parameter data obtained at the specified points, an exergy analysis and entropy generation assessment are conducted to determine the contribution of each system component to the occurring losses [15] and to evaluate whether the applied modifications can contribute to reducing those losses. In practical terms, by applying exergy analysis and sustainability indices, refinements in plant design, maintainability procedures, and operational approaches have the ability to improve the overall performance of the plant [16], lowering exergy destruction directly enhances the overall thermodynamic efficiency of the turbine system, meaning that more of the steam's energy is converted into useful mechanical work rather than dissipated as irrecoverable losses.

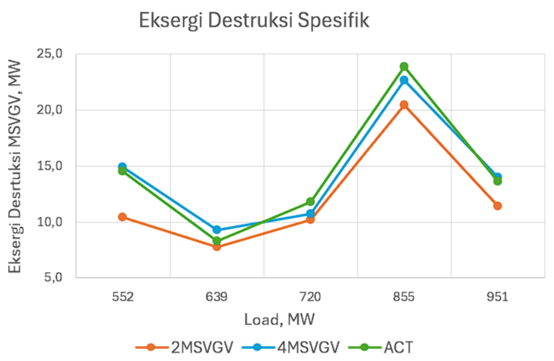

Figure 10. Profile exergy specific total

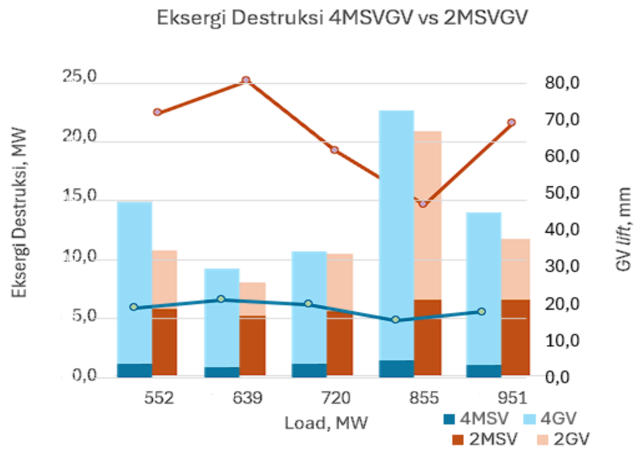

Figure 11. Exergy distribution each assembly

The thermal design of equipment involves two seemingly opposing approaches: enhancing thermal contact to achieve efficient heat transfer or applying insulation to minimize undesirable heat leakage. However, regardless of the specific objective, the core principle of both approaches lies in reducing entropy generation within the thermodynamic system, thereby improving overall efficiency [17]. In practice, the MSVGV body is covered with specialized insulation, and the fluid mass flow passing through the valve is considerably high, such that the heat loss to the environment is relatively small compared to the energy contained in the fluid, rendering negligible heat transfer [18].

Figure 10 illustrates that the overlapping 4MSVGV modeling demonstrates the model's ability to closely approximate actual conditions, thus validating its suitability for subsequent exergy analysis of individual components. The modification employing 2MSVGV consistently exhibits lower exergy destruction compared to the 4MSVGV configuration. The most substantial reduction is observed at a load of 855MW, where the modified configuration achieves an exergy destruction saving of 2.572MW.

Through exergy destruction analysis applied separately to the MSV and GV, as depicted in Figure 11, the compositional contribution of each component to the total exergy destruction was determined. In the 4MSVGV configuration, with an opening of 21mm, the graph indicates significant throttling occurring at the GV valve, thus dominating the exergy destruction. In this configuration, the average exergy destruction in the MSVs ranges from 0.9 to 1.3MW due to low steam velocity rate at 4MSV, with the remainder of the total exergy destruction resulting from throttling effects at 4GV. Figure 11 also reveals that the average exergy destruction for the MSV valves in the 2MSVGV configuration is between 5.4 and 6.4MW, with the remaining exergy destruction occurring in the GVs. Overall, the average reduction in exergy destruction for the 2MSVGV configuration is 2.050MW.

In practical implementation, the proposed 2MSVGV configuration introduces no major trade-offs in safety, operational complexity, or hardware modification, as it merely involves a shift in the valve response characteristic. The valve characteristic adjustment is similar to sequence valves control, allows seamless integration with the existing turbine control logic, preserving all safety interlocks and operational procedures [19]. The modification does not require physical changes to the valve assemblies, ensuring minimal implementation risk. Furthermore, industry practice with the 3-valve governor configuration (3MSVGV) used in routine valve response testing provides supporting evidence that the proposed approach is technically safe and operationally feasible.

CONCLUSION

Utilizing the mass flow rate of steam required to pass through the MSVGV at a specific load setpoint as the basis for determining GV lift has proven effective in simplifying the technical feasibility analysis of the new 2MSVGV configuration. Numerical analysis results for the 2MSVGV configuration indicate its capability to serve all normal operational loads ranging from 400 to 1000MW, meeting the criterion of governor valve lifting <80mm. CFD analysis provides the flexibility to obtain steam property data at all points within the MSVGV control volume, allowing for the analysis of the contribution of each MSVGV component to the overall losses. Quantification of throttling losses irreversibilities within the MSVGV through entropy generation and exergy destruction analysis provides a good understanding of the magnitude of these losses. The results from numerical CFD analysis and exergy destruction calculations reveal that the 4MSVGV configuration experiences significant throttle losses in the GV side compared to the MSV side lossess, attributed to the relatively low GV lift opening. The 2MSVGV configuration demonstrates a better distribution of due to a larger GV lift. Across the calculated load range, the 2MSVGV configuration consistently achieves throttle losses reduction with an average in reduced exergy destruction of 2.05MW.

The present results also indicate that the inlet pressure setpoint upstream of the valve, which is currently operated under a linear control scheme, has a noticeable influence on the fluctuation of exergy destruction across different load conditions. Future studies should therefore focus on investigating a non-linear inlet pressure control strategy, using the minimum achievable exergy destruction as a key optimization criterion. Such an approach could further enhance overall system performance by minimizing unavoidable irreversibilities and improving the thermodynamic efficiency of the steam admission process.

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