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AUTOMATED ANALYSIS OF STALL INCEPTION IN MULTISTAGE COMPRESSORS USING 1D METHODS

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ABSTRACT

A compressor operating in stall can lead to large aerodynamic losses and potentially compressor surge. Understanding the operating point where stall first occurs is of importance to turbomachinery design engineers as they iterate between designs during the design process. A methodology has been developed which predicts the onset of stall using a 1D-solver approach. This methodology shows reasonable agreement in the stall inception point compared to both 3D CFD and experimental approaches. Utilizing this 1D approach allows for the prediction of the stall point, as well as the identification of the compressor stage where stall inception occurs, all in a matter of minutes. This allows for more rapid and targeted design changes earlier on in the engineering design process.

Keywords: Stall; multistage compressor; 1D analysis

1. INTRODUCTION

A methodology was developed for determining the inception of stall in multistage compressors. Operating in stall leads to large aerodynamic losses and is associated with compressor surge, potentially causing vibrations and damage to the compressor. The developed methodology finds the stall point utilizing a 1D solver approach in AxSTREAM™. This allows the stall point to be found in a matter of minutes, rather than hours or days utilizing a CFD approach. The 1D approach can be modified to find the stall point on machines with any number of stages. By rapidly finding the stall point, design iterations may be performed much more quickly and earlier on in the design process. It should be noted that while the accuracy of this 1D stall modelling approach may be more limited when compared to that of a CFD approach, the 1D approach provides a preliminary estimate of stall which otherwise would not be available so early in the design process. If there is an issue with stall margin for a given design, it is useful to understand this as early in the design process as possible. Knowledge of the approximate location of surge allows a designer to perform trade-off studies on various overall architectures, configurations, and overall geometries, especially when particularly sensitive or impactful stages are identified.

An example of a typical compressor performance map is shown in FIGURE 1, which relates the compressor's pressure rise to its mass flow rate for various operating speeds. Each performance curve is limited at low flow rates at the stall line and limited at high flow rates at the choke line.

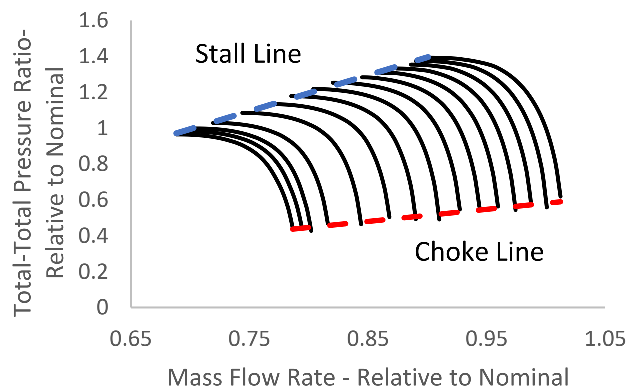

FIGURE 1: EXAMPLE OF A COMPRESSOR PERFORMANCE MAP

When a compressor is exhibiting stall, there is a low flow rate and high pressure rise across the machine. The fluid naturally wants to move against the flow direction due to the fluid pressure gradient but is moved through the machine by the rotors. At stall, the fluid pressure gradient becomes so great that not all of the fluid is moved through the machine by the rotors. Instead, small pockets of working fluid (called stall cells) become stagnant, leading to local regions of flow recirculation within the flow path. Eventually, these stall cells grow large enough that there is reversed flow at the compressor inlet, leading to a sudden surge of fluid out of the compressor inlet. This phenomenon is called compressor surge. Thus, at a particular operating speed, there is a minimum flow rate at which the compressor can operate before being limited by stall and surge behavior. The collection of these minimum flow rate operating points makes up the compressor's stall line.

Alternatively, the maximum flow rate that the compressor can achieve at a particular speed is called the choke (or stonewall) point, and the collection of maximum flow rate operating points make up the choke line. The choke point is caused by the flow speed reaching the speed of sound, which limits further increases of mass flow rate.

Understanding the location of the stall inception operating point helps ensure safe operation of a compressor, as well as the implementation of anti-surge control systems. Anti-surge systems use existing surge limit models and system control algorithms in conjunction with sensors detecting system parameters to determine whether the onset of surge is occurring and intelligently controlling an anti-surge valve to counteract this onset [1]. These surge limit models are typically based off previous experimental testing data as well as some desired surge margin. Typically, sensors will be located either near the rotor leading edge [2] to monitor tip vortex interactions at the stall point, as well as directly upstream and downstream of the rotor at various channel spans [3]. By having a preliminary understanding of where the stall line is during the design process, a certain amount of surge margin at the design point can be accounted for, allowing for easier surge control behavior during normal operation.

Also, it should be noted that stall and surge inception typically originate as a localized flow disturbance before growing to a fully developed stall cell [4]. This developed stall cell may only fully block the flow through a single cascade cell of a given compressor stage. Additionally, since the stall propagation speed is typically not the same as the rotor speed, the stalled cascade cell will move from one cascade cell to another. This phenomenon is demonstrated in FIGURE 2. As local stall transitions to surge and the stall cell blocks flow through the entire cell, this causes the propagation of shock waves through the rest of the machine and generates more flow instabilities. It is of importance to design engineers to not only determine the operating point at which stall inception occurs, but also the compressor stage at which local stall originates to allow for more targeted design modifications.

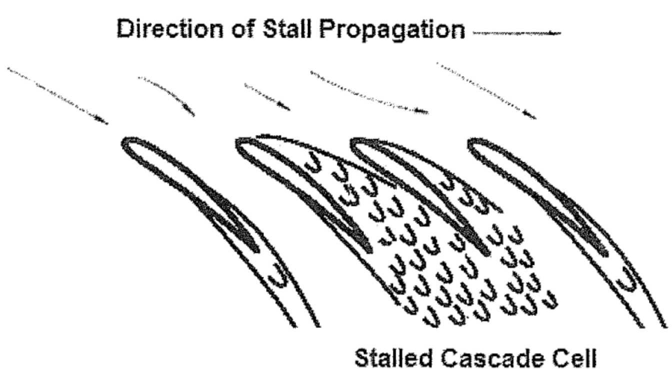

FIGURE 2: ROTATING STALL PROPAGATION PHENOMENON [5]

Several methodologies have been used previously in modelling stall inception. Some methodologies involve the use of unsteady pressure envelopes (waves) and either analyzing a pressure amplitude frequency spectrum [6] or solving an eigenvalue problem to determine if a solution incorporating certain pressure, vortex, and entropy waves have a stable solution [7]. Other methodologies modify the equations solved in CFD to better predict stall inception [8]. However, some of these methods require certain flow assumptions to be met (e.g., relatively large circumferential flow disturbances) and/or require the use of CFD, which slows down the turbomachinery design process.

Additionally, 1D methods have previously been developed to simulate surge behavior [3]. However, this method calculated the unsteady flow field by solving the unsteady conservation laws for compressible, inviscid flows. Due to the transient nature of this method, system properties were calculated for individual rotor revolutions during the different phases of a surge cycle. While this sort of model provides key insights of the transient surge behavior, the model is overly complex for use early in the design process when compared to 1D steady-state methods.

The newly developed 1D methodology was originally implemented in the design of a multistage compressor for a power plant application. The methodology was utilized during the design and analysis of several compressor design variants, including two different multistage axial-centrifugal compressors and a multistage integrally geared centrifugal compressor. Once the methodology was better understood, a study validating the methodology was performed using a 2-stage transonic axial compressor from literature.

2. 1D STALL INCEPTION ANALYSIS PROCESS

During the application of this methodology, several operating points need to be analyzed and compared along a single speedline. An automated workflow approach utilizing AxSTREAM ION, which is a commercial computational workflow automation software developed by the authors' company, was developed to automate the 1D analysis of several operating points. Based on the results of the previous 1D analysis, the compressor operating point would be updated for the next analysis iteration, or the operating point would be determined to have stalled.

To start the methodology, a high mass flow rate (MFR) is initially tested. The initial flow rate should be large to ensure that the compressor is not stalling during the first iteration. A good operating condition to use would typically be the design point. The mass flow rate is then slightly decreased during the next solver iteration. This provides a solution for two different operating points.

The total-static pressure ratios are extracted for each of the stages for both analyzed operating points. A compressor stage is said to be stalling if its total-static pressure ratio decreases as mass flow rate through the compressor decreases. This type of definition of stall has been used previously by Hipple, et. al [9]. While other stall estimation techniques exist which involve estimating the stall point based on the relationship of the maximum static pressure rise and cascade geometric parameters such as the limiting area ratio [10], diffuser width ratio, contraction ratio, and diffuser inlet shape [11], the 1D approach used in this study is different in that it finds the maximum total-static pressure ratio by directly calculating the aerodynamic performance of a given machine geometry.

If the stalling condition is not met for any of the stages, then mass flow rate is again decreased slightly for the next operating point. The two lowest mass flow rate operating points again compare their total-static pressure ratios to check for stall. This process occurs repeatedly until an operating point is found where the conditions for stall are met for at least one stage. The output of this analysis is not only the operating point at which stall inception occurs, but also the compressor stage which is stalling.

This process can be optimized by carefully controlling the step size for changing mass flow rate between operating points. Initially, the step sizes can be relatively large. Once an initial prediction for the stall point is generated with large flow rate step sizes, the step size can be reduced, and the range of operating points analyzed can be narrowed. This process can be repeated to home in on the stall value to any desired accuracy, constrained of course by the accuracy of the underlying performance prediction models. The conceptual workflow algorithm is shown in FIGURE 3.

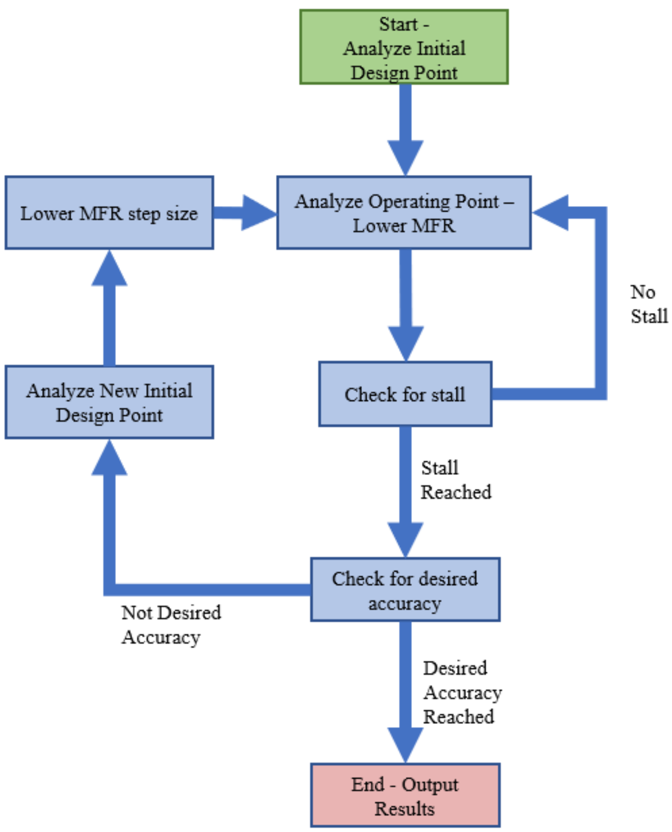

FIGURE 3: FLOW DIAGRAM FOR FINDING STALL INCEPTION

This process was performed autonomously utilizing AxSTREAM ION. The workflow diagram implemented in the software is shown in FIGURE 4. Each block shown corresponds to some operation being performed in the workflow.

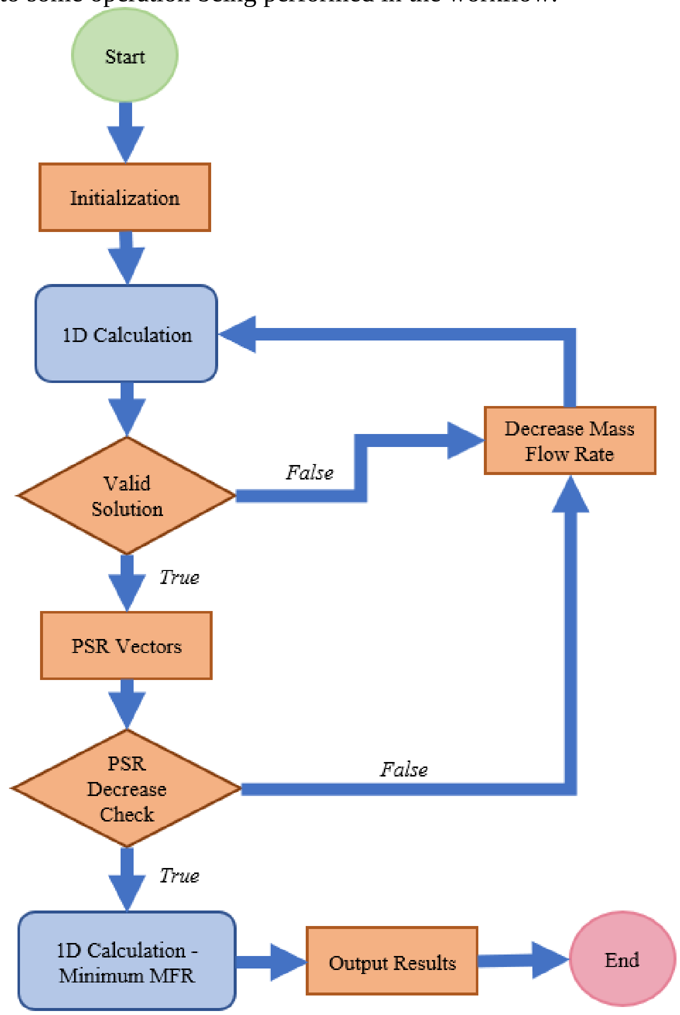

FIGURE 4: WORKFLOW DIAGRAM FROM AXSTREAM ION

The operation of each workflow block is described below.

Start: Starts the workflow

Initialization: Sets initial value of the mass flow rate to be analyzed as well as the step size used when incrementally decreasing the mass flow rate.

1D Calculation: Performs a 1D analysis of the machine at the specified mass flow rate. Note that other operating parameters such as inlet pressure and rotational speed are fixed for all 1D calculations performed in a given workflow.

Valid Solution: Checks whether a valid solution was successfully generated during the 1D Calculation operation. If the solution is invalid, the *False* branch is chosen so a new flow rate may be analyzed. Otherwise, the *True* branch is chosen.

PSR Vectors: Each time a calculation is successfully performed, the total-static pressure ratio (PSR) of each stage is stored in a vector. A single vector stores the PSR of a given stage for all successfully analyzed operating points.

PSR Decrease Check: Each PSR vector is checked to determine whether the conditions for stall have been met. If stall hasn't been achieved, the *False* branch is selected so a new flow rate may be analyzed. If stall has been achieved, then the *True* branch is chosen.

Decrease Mass Flow Rate: Updates the mass flow rate by decreasing it by a small step size. This new mass flow rate is used during the next 1D calculation.

1D Calculation – Minimum MFR: Once the flow rate at which stall occurs is found, this mass flow rate (MFR) is analyzed again in a 1D analysis. After this analysis is performed, outputs (such overall machine pressure ratios, stage pressure ratios, power, efficiency, etc.) at the stall point may be extracted.

Output Results: Outputs are saved to an external file.

End: Ends the workflow

3. METHODOLOGY APPLICATION AND VALIDATION

3.1 Methodology Initial Application

This methodology was initially used during the development of several multistage compressor design variations for a power plant application. Part of this analysis included determining the turndown capability of three separate compressor designs. It is important to note that the generated designs were developed first, and then the stall point analysis methodology was used to evaluate these generated designs. The first design is a multistage axial-centrifugal configuration which compresses the nominal flow rate of air up to the design pressure ratio. Due to the nominal flow rate and size of the compressor being large, two additional designs were generated and analyzed, each sized for 50% of the nominal flow rate at the design point. These two additional compressor designs used different configurations: a multistage axial-centrifugal configuration for Design 2 and a multistage integrally geared centrifugal configuration for Design 3.

The compressor analyses involved generating a family of performance curves for different inlet guide vane (IGV) positions of the given compressor being analyzed. For each performance curve, the range of performance was found automatically by finding the stall inception point and similarly the choke point using the previously described process. By understanding the location of the stall and choke points, performance maps could be more easily generated. Figures showing the analyzed compressor geometries as well as their corresponding performance maps are shown below. It will be noted how the surge lines (and choke lines) can be easily visualized based on the extrema of the performance curves.

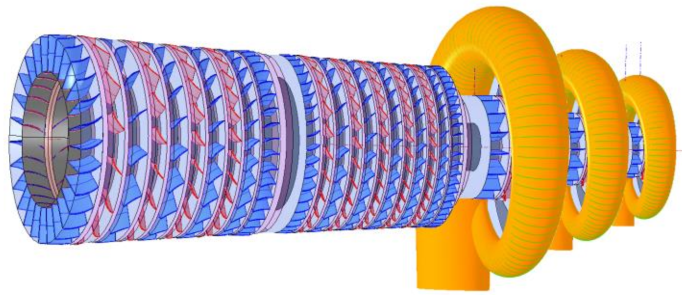

FIGURE 5: MULTISTAGE AXIAL-CENTRIFUGAL COMPRESSOR, DESIGN 1

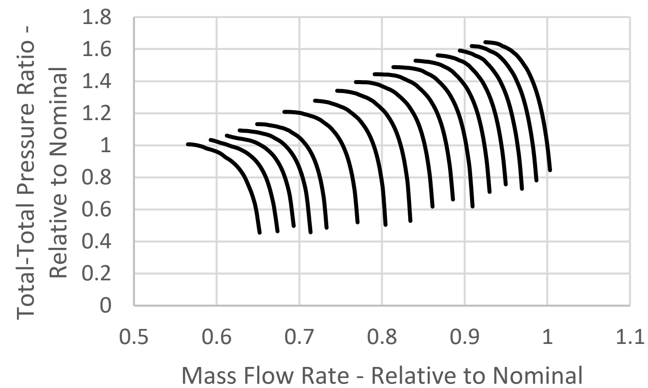

FIGURE 6: PERFORMANCE MAP, MULTISTAGE AXIAL-CENTRIFUGAL COMPRESSOR, DESIGN 1

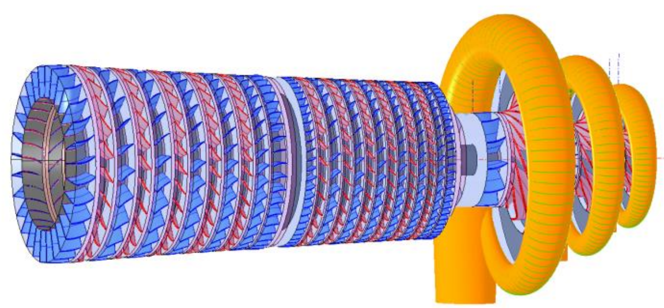

FIGURE 7: MULTISTAGE AXIAL-CENTRIFUGAL COMPRESSOR, DESIGN 2

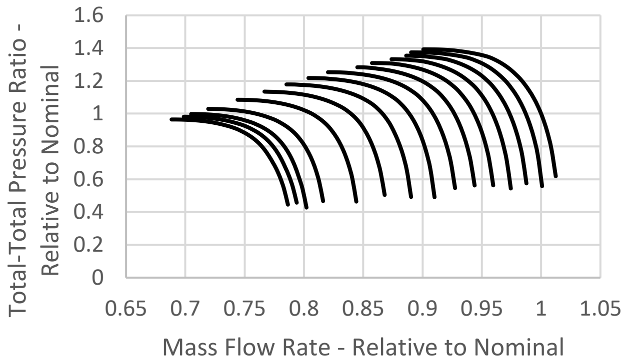

FIGURE 8: PERFORMANCE MAP, MULTISTAGE AXIAL-CENTRIFUGAL COMPRESSOR, DESIGN 2

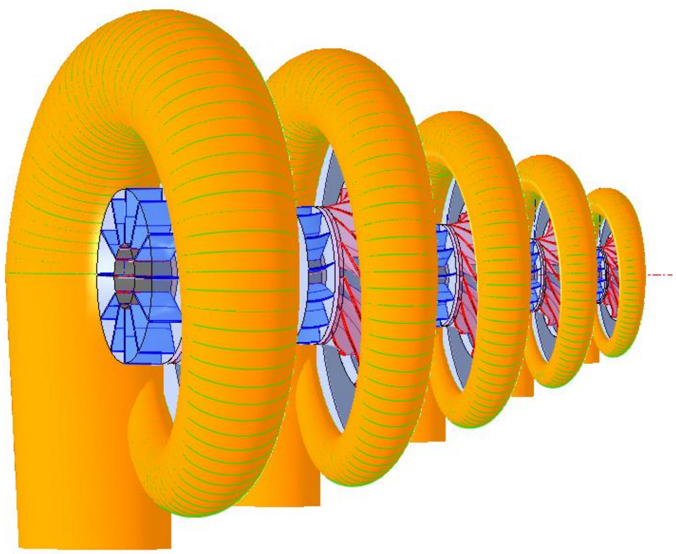

FIGURE 9: MULTISTAGE INTEGRALLY GEARED CENTRIFUGAL COMPRESSOR, DESIGN 3

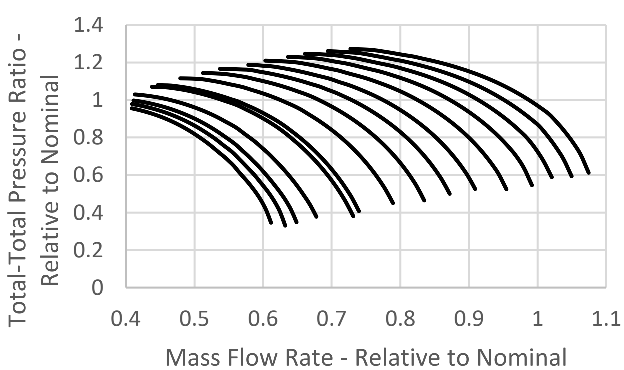

FIGURE 10: PERFORMANCE MAP, MULTISTAGE INTEGRALLY GEARED CENTRIFUGAL COMPRESSOR, DESIGN 3

3.2 Methodology Validation

The methodology was validated using an existing design of a 2-stage transonic axial compressor [12]. The geometry was imported into AxSTREAM as specified in literature. This geometry is shown in the below figures. The operating conditions of this compressor were also specified from literature and are presented in the following table.

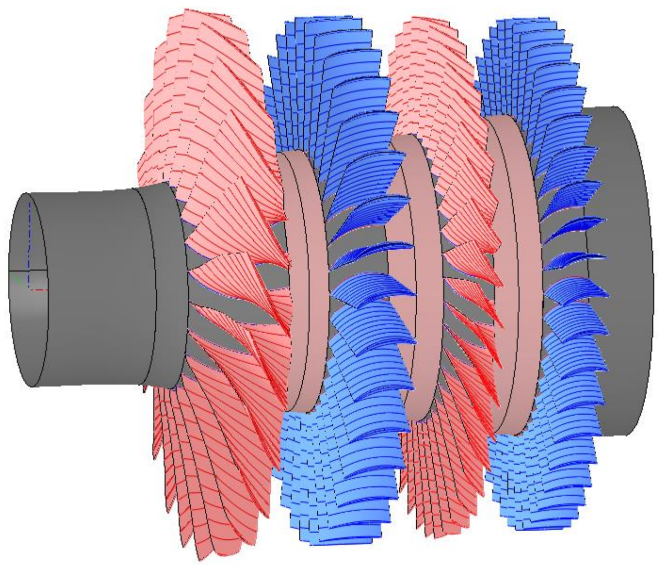

FIGURE 11: 2-STAGE TRANSONIC AXIAL COMPRESSOR, 3D VIEW

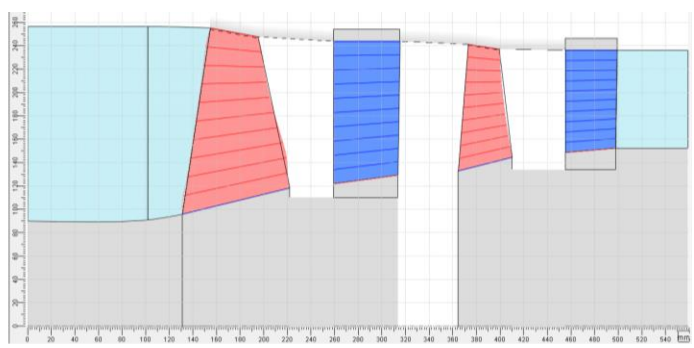

FIGURE 12: 2-STAGE TRANSONIC AXIAL COMPRESSOR, MERIDONIAL VIEW

TABLE 1: OPERATING CONDITIONS OF 2-STAGE TRANSONIC AXIAL COMPRESSOR AT DESIGN POINT

Operating Parameter	Value
Working Fluid	Air
Total Inlet Temperature	15 °C
Total Inlet Pressure	101.325 kPa
Total Pressure Ratio	2.399
Mass Flow Rate	33.248 kg/s
Rotational Speed	16042.8 rpm

An automated workflow process in AxSTREAM ION was implemented and the stall point was found using the 1D solver approach. Additionally, 3D CFD simulations were performed in STAR-CCM+ and Ansys CFX to compare the stall point analytically to higher-fidelity modeling methods. Mesh, physics, and domain properties of the two CFD simulations used in this validation are provided in the below table.

TABLE 2: PROPERTIES OF CFD SIMULATIONS

Property	STAR-CCM+	Ansys CFX
Mesh Type	Polyhedral	Hexahedral
Number of Elements	$\sim 2.5 \times 10^6$	$\sim 4.9 \times 10^6$
Turbulence Model	k-ε	k-ω SST
Equation of State	Ideal Gas	
Energy	Coupled Energy Modeling	
Reference Frame	Moving Reference Frame	
Type of Geometry Modeled	Single Rotationally Periodic Blade Passages	
Interfaces	Mixing Plane interfaces between rotating and stationary domains Periodic interfaces at periodic boundaries	
Inlet Boundary Condition	Total Pressure and Temperature	
Outlet Boundary Condition	Static Pressure	

Representative figures showing the computational mesh used in the two CFD packages are shown below. In both cases, the mesh grid was refined near the walls and in any tip gaps.

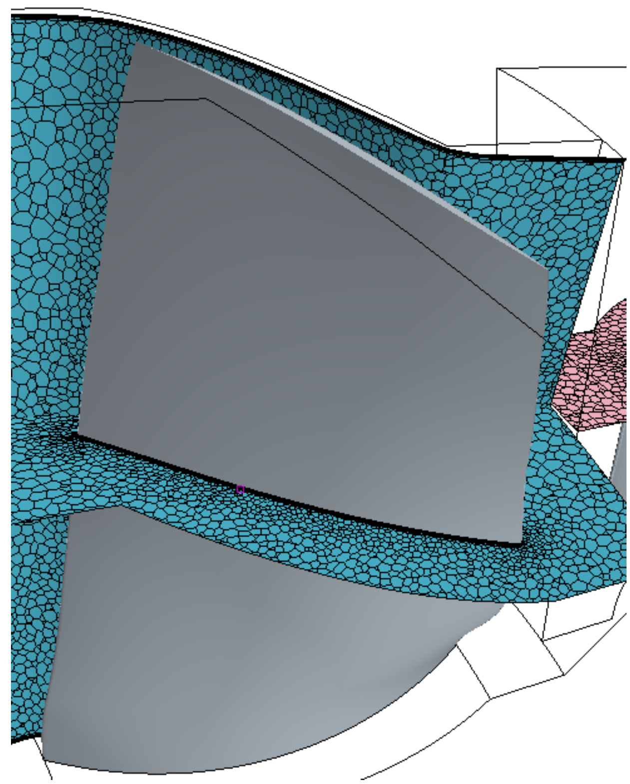

Figure 13: REPRESENTATIVE UNSTRUCTURED POLYHEDRAL MESH FROM STAR-CCM+

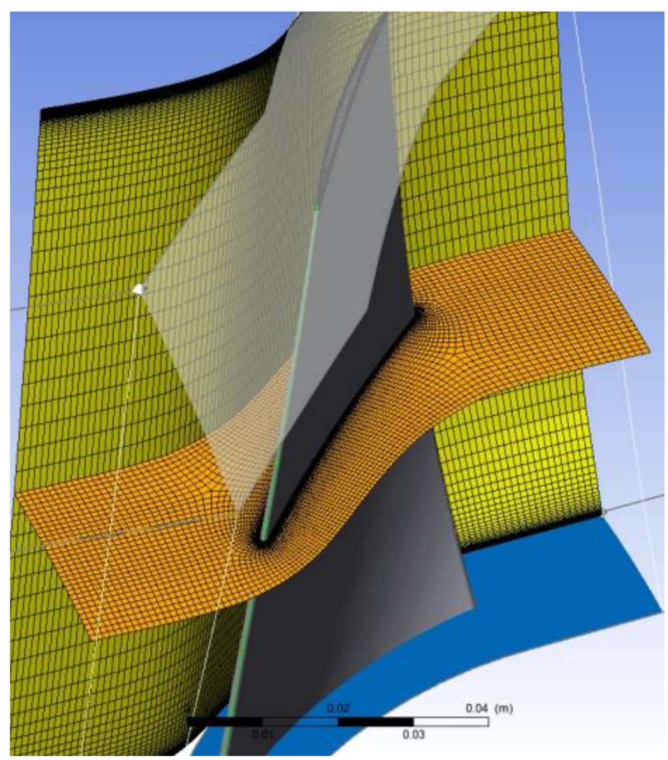

Figure 14: REPRESENTATIVE HEXAHEDRAL STRUCTURED MESH FROM ANSYS CFX

The results of this analysis were plots of total-static pressure ratio vs. flow rate for each compressor stage, as well as a performance curve of total-static pressure rise vs. flow rate for the total machine. The experimental results from literature were included in the plots for comparison as well. These plots are shown in the below figures.

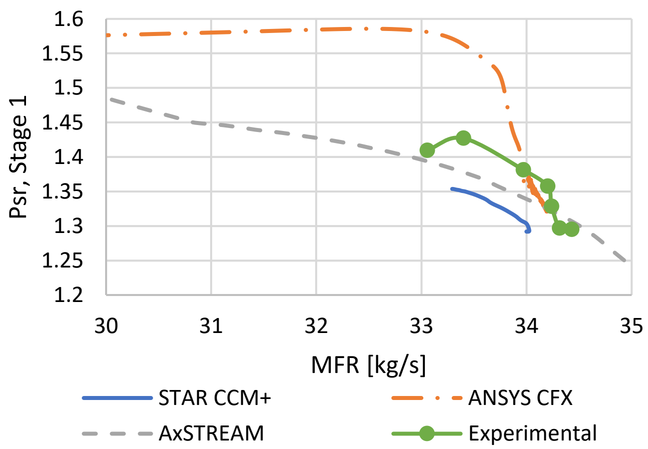

FIGURE 15: TOTAL-STATIC PRESSURE RATIO VS. MASS FLOW RATE, STAGE 1

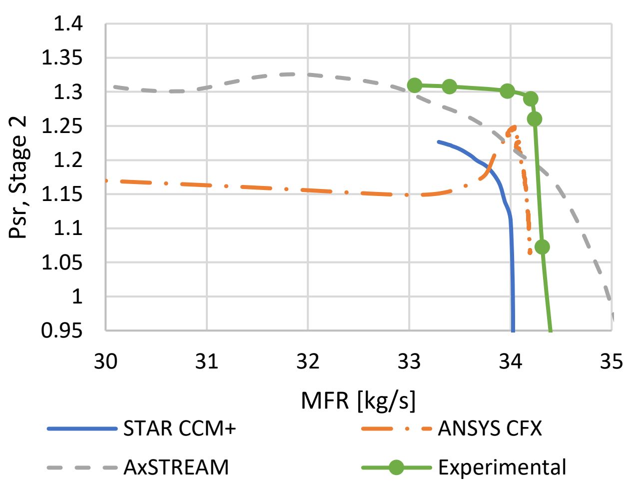

FIGURE 16: TOTAL-STATIC PRESSURE RATIO VS. MASS FLOW RATE, STAGE 2

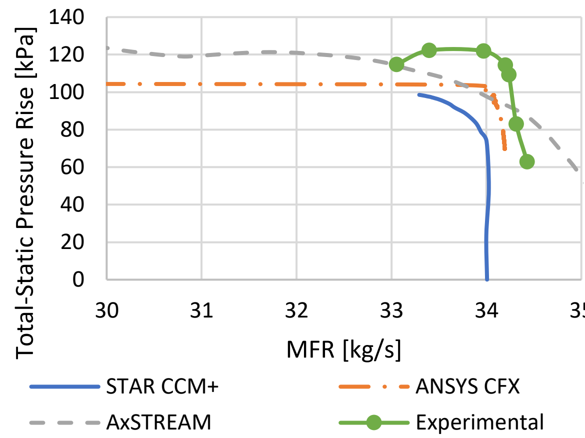

FIGURE 17: TOTAL-STATIC PRESSURE RISE VS. MASS FLOW RATE, OVERALL MACHINE

By analyzing the plots of total-static pressure ratio (psr) vs. mass flow rate for each stage, an approximate location of the stall point can be determined where the derivative of one of the psr vs. mass flow rate plots goes to zero. These values of the stall point are provided in the below table. Note that the experimental stall point was reported in literature as 'near stall' [12] and was therefore not produced using the graphical process previously described.

TABLE 3: STALL POINT USING DIFFERENT ANALYSIS METHODS

Analysis Method	Stall Point Mass Flow Rate
AxSTREAM – 1D Solver	32.00 kg/s
STAR CCM+ - 3D CFD	33.29 kg/s
ANSYS CFX – 3D CFD	33.97 kg/s
Literature Experimental Value [12]	34.01 kg/s

One thing to note is that there is some deviation between the results from the CFD analyses. This may be a result of the difference in turbulence models being used or different mesh types and sizes, especially near the boundary layer. The experiment also predicts higher overall total-static pressure rise compared to analytical methods. It is likely that this is due to the experimental compressor outlet conditions being evaluated very close to the outlet stator trailing edge [12], whereas the CFD simulations included an annular space several blade chords in length which would have incurred mixing and swirling losses. Even so, the validation study shows that there is reasonable level of agreement between 1D and CFD results such that the results of the 1D analysis may be used as a tool to aid preliminary design decisions. Once the preliminary design has been sufficiently refined based on the results of 1D analyses, it is good practice to still validate the performance predictions of a given design in CFD later in the design process.

4. DISCUSSION

A primary benefit of using the automated 1D solver approach is that finding the stall point of a particular compressor design takes minutes instead of hours or days using a CFD approach. This allows engineers to make rapid changes to their compressor design and quickly compare their performance. If a given machine has issues with stall margin, it is better to understand this as early in the design process as possible so targeted changes can be made more easily.

During the validation study, all the analytical approaches indicated that stall inception occurred in the second compressor stage. Additionally, when compared to the experimental stall inception point from literature, the 1D approach indicated a 5.9% difference in the mass flow rate at stall inception, while CFD approaches showed 2.1% (STAR-CCM+) and 0.1% (ANSYS CFX) difference. While the 1D approach had the largest error, it used standard non-tuned loss models when calculating performance. Tuning the 1D loss models to the experimental data and this family of compressors would result in even better agreement between 1D and experimental performance.

These 1D loss models are based on the work of Lieblein [13] who developed rules and relations to predict the total-pressure loss as well as reference deviation and incidence angles for compressor blade profiles. These relations were further developed by Aungier [14] [15] for several types of machines, including axial and centrifugal compressor stages. The loss models developed by Aungier are the ones which were implemented during the 1D analysis. These types of loss models have been shown to have satisfactory agreement with experimental performance data [16], especially once the models are tuned for a given machine type and system environment.

5. CONCLUSION

A methodology using a 1D solver approach allowed for an approximation of the compressor stall point to be generated in a matter of minutes. There is also a reasonably low level of error in the stall point from the 1D approach when compared to experimental values from literature. This approach allows engineers to rapidly iterate between compressor designs early in the design process while taking the stall point into account.

ACKNOWLEDGEMENTS

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REFERENCES

[1] K. Brun and M. G. Nored, *Application Guideline for Centrifugal Compressor Surge Control Systems,* Southwest Research Institute, 2008.

[2] N. Tahara, T. Nakajima, M. Kurosaki, Y. Ohta, E. Outa and T. Nishikawa, "Active Stall Control with Practicable Stall Prediction System Using Auto-Correlation Coefficient," American Institute of Aeronautics & Astronautics, 2001.

[3] P. Waniczek, H. Schoenenborn and P. Jescke, "Experimental and Analytical Surge Cycle Analysis of a High Pressure Aero Engine compressor," in *ASME Turbo Expo*, Copenhagen, Denmark, 2012.

[4] P. X. Huang and J. Yin, "Transient Surge Dynamics: A Shock Tube Theory & Experimental Comparison," in *ASME Turbo Expo*, Düsseldorf, Germany, 2014.

[5] H. W. Emmons, C. E. Pearson and H. P. Grant, "Compressor Surge and Stall Propagation," *Transactions of ASME*, vol. 77, no. 4, pp. 455-467, 1955.

[6] R. L. Davis and J. Yao, "Computational Approach for Predicting Stall Inception in Multistage Axial Compressors," *Journal of Propulsion and Power,* vol. 23, no. 2, March-April 2007.

[7] S. Xiaofeng, S. Dakun and Y. Weiwei, "A Model to Predict Stall Inception of Transonic Axial Flow Fan/Compressors," *Chinese Journal of Aeronautics,* vol. 24, pp. 687-700, 2011.

[8] M. Righi, V. Pachidis, L. Könözsy and L. Pawsey, "Three-dimensional through-flow modelling of axial flow compressor rotating stall and surge," *Aerospace Science and Technology,* vol. 78, pp. 271-279, July 2018.

[9] S. M. Hipple, H. Bonilla-Alvarado, P. Pezzini, L. Shadle and K. M. Bryden, "Using Machine Learning Tools to Predict Compressor Stall," *Journal of Energy Resources Technology,* vol. 142, July 2020.

[10] C. Koch, "Stalling Pressure Rise Capability of Axial Flow Compressor Stages," *Journal of Engineering Power,* vol. 103, pp. 645-656, 1981.

[11] H. Kobayashi, H. Nishida, T. Takagi and Y. Fukushima, "A Study on the Rotating Stall of Centrifugal Compressors," *JSME Transactions,* vol. 56, pp. 2646-2651, 1990.

[12] C. L. Ball, L. Reid and J. F. Schmidt, "End-Wall Boundary Layer Measurements in a Two-Stage Fan (NASA-TM-83409)," NASA, 1983.

[13] I. A. Johnsen and R. O. Bullock, Aerodynamic Design of Axial-Flow Compressors, vol. 36, Scientific and Technical Information Division, National Aeronautics and Space Administration, 1965, pp. 183-226.

[14] R. H. Aungier, Axial-Flow Compressors: A Strategy for Aerodynamic Design and Analysis, 2003.

[15] R. H. Aungier, Centrifugal Compressors, New York: ASME, 2000.

[16] L. Moroz, Y. Govoruschenko, P. Pagur and L. Romanenko, "Integrated Conceptual Design Environment for Centrifugal Compressors Flow Path Design," in *ASME*

International Mechanical Engineering Congress and Exposition, Boston, 2008.