An Approach to Prevent Tip-Out Surge in a Turbocharger Compressor

Thesis

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Abstract

At low mass flow rates, a turbocharger's performance is limited by compressor surge which is an instability characterized by self-sustained oscillations in the compression system. Compressor throttle valves (CTVs) are used to regulate compressor flow, and a rapid closure of the CTV is called a “tip-out”. Bypass valves are typically used to prevent surge during tip-outs by venting excess compressor exit pressure through an alternative path. To reduce complexity, however, there is interest in preventing surge without a bypass valve. One method is to optimize the CTV closure profile: earlier attempts on this approach focused on single tip-out cases where the profile was manually determined. The objective of this research was to prevent surge for a range of tip-outs without a bypass valve by controlling the closing profile of a CTV. A turbocharger test stand was first used to determine the CTV angles that resulted in surge for various rotational speeds. These CTV angles are called the “critical angles” and fall on the stability limit of the turbocharger. Thus, actual CTV angles smaller than the critical angle for a given rotational speed are expected to result in surge as further reducing the flow rate pushes the compressor into the unstable region. The controller was then implemented into LabVIEW, and it prevents surge by ensuring the actual CTV angle is never smaller

than the critical angle for a given rotational speed. The results showed that surge was prevented for all tip-out cases automatically within a selected range. The controller was tuned such that the compressor operated near the surge boundary indicating that the closure time of the controlled CTV profiles is near optimal for the current logic implementation. This study benefits the automotive industry by demonstrating that turbochargers can be kept out of surge during tip-outs without a bypass valve.

Acknowledgments

I would first like to thank Professor Selamet for giving me the opportunity to conduct research in the Flow, Engine, Acoustics, and Turbocharger Research Laboratories. His mentorship and expertise throughout the construction of this thesis has contributed immensely to my growth as a researcher, and I have benefitted greatly from his patience in reviewing and correcting this thesis. I also thank Dr. Pranav Sriganesh for the instrumental role he played in conducting the experimental testing and interpreting the results of this study. Furthermore, I thank Professor Jen-Ping Chen for taking the time to serve on my defense committee. Lastly, I would like to thank my family for their unwavering support and encouragement throughout my undergraduate studies.

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Chapter 1. Introduction

1.1 Background

Turbochargers increase the quantity of air fed to an engine by using waste heat to spin a turbine which drives a compressor. Air from the compressor is cooled and fed to the engine’s intake manifold at pressures higher than atmospheric. The higher pressure results in increased intake air density which means more fuel can combust for a given engine displacement. Using a turbocharger on a downsized engine is employed by automakers as an effective method for reducing fuel consumption and greenhouse gas emissions of internal combustion engines (Kirwan et al., 2010).

The performance and operating range of centrifugal compressors is commonly conveyed in terms of the pressure ratio as a function of mass flow rate for constant rotational speeds. A compressor characteristic is a plot of pressure ratio as a function of mass flow rate for a single constant rotational speed, and a compressor map is a collection of such compressor characteristics. A typical performance map for a centrifugal compressor is shown in Fig. 1.1.

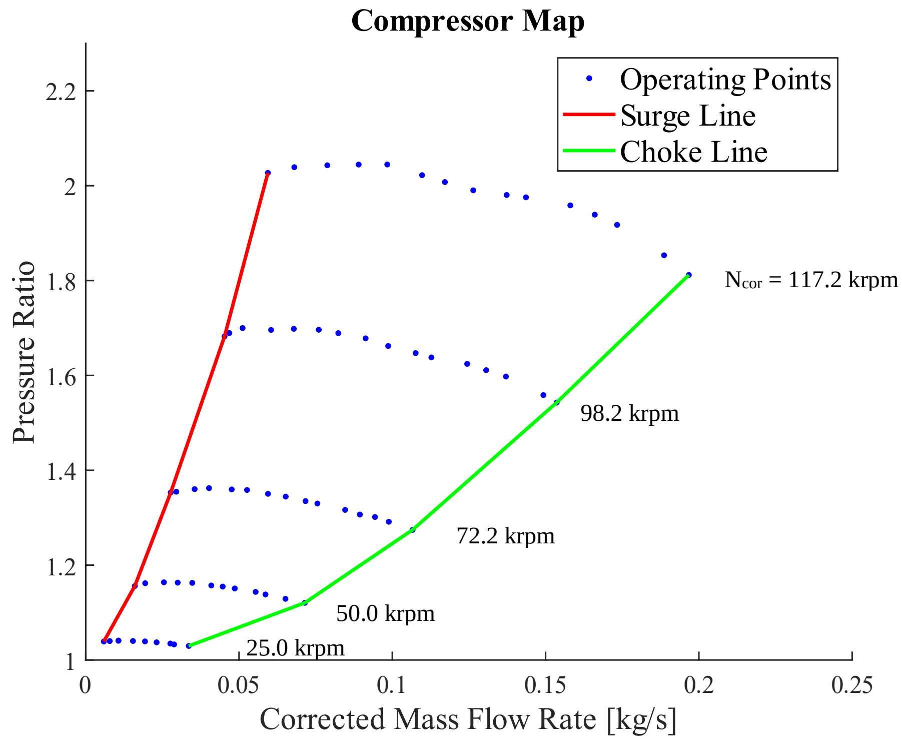

Figure 1.1: Sample compressor map.

A compressor map commonly uses corrected parameters that adjust for changes in temperature and pressure to reflect a standard condition. The corrected rotational speed

$$N_{cor} = \frac{N}{\sqrt{T_{01}/T_{ref}}} \tag{1.1}$$

adjusts rotational speed N for the total compressor inlet temperature T_{01} where $T_{ref} = 298\ K$ is the reference temperature. Similarly, the corrected compressor mass flow rate

.

$$\dot{m}_{cor} = \dot{m}\sqrt{\frac{T_{01}}{T_{ref}}} * \frac{P_{ref}}{P_{01}} \tag{1.2}$$

adjusts the mass flow rate $\dot{m}$ for T_{01} and the total compressor inlet pressure P_{01} where $P_{ref} = 100\ kPa$ is the reference pressure.

On a compressor map, operating points are plotted along lines of constant rotational speed, and the operating range for a given speed is defined at low flow rates by surge and at high flow rates by compressor choke. Surge is a low flow instability characterized by self-sustained oscillations in the compression system. Choked flow is a compressible fluid effect where the flow becomes sonic at the point of minimum cross-sectional area.

Compressor characteristics exhibit downward concavity meaning there will be a maximum pressure ratio dividing positively and negatively sloped portions of the characteristic line. The compressor system is stable for all operating points on the negatively sloped portion of the compressor characteristic (Greitzer, 1981). For flow rates below the maximum pressure ratio, a gradual reduction in compressor pressure ratio along with decreasing flow rate is termed progressive stall, and at low flow rates can result in compressor stall or surge (Cumpsty 1989).

Surge is a self-excited, system oscillation which can be classified as *mild* or *deep* depending on the severity of the mass flow fluctuations. In mild surge, the annulus average mass flow oscillation remains in the forward direction while deep surge is characterized by reversal of the mean flow for some duration of the cycle (Dehner 2016). Operating a compressor in deep surge should be avoided as the large-scale pressure fluctuations increase noise, deteriorate compressor performance, and can lead to premature mechanical failure.

Compressor throttle valves (CTVs) regulate the mass flow rate of air delivered by the compressor to the intake manifold of an engine. A tip-out is an actuation case where the CTV is rapidly closed, reducing the mass flow rate delivered to the engine. Rapidly closing the CTV can induce compressor surge as the swift decrease in mass flow rate drives the operating point towards the surge line. To prevent the compressor from entering surge during tip-outs, compressor bypass valves (CBVs) or similar devices can be implemented. CBVs connect the compressor outlet and inlet allowing excess pressure to be vented by driving the flow through an alternative path instead of backwards through the compressor. By providing an open path between the compressor outlet and inlet, the outlet pressure is decreased thus reducing the overall pressure ratio. By reducing the pressure ratio, lower mass flow rates are required to enter surge. With proper implementation of a CBV, surge can be mitigated during tip-outs.

1.2 Literature Review

A common method of preventing surge is known as surge avoidance where the compressor is prevented from entering the unstable region left of the surge line. A surge avoidance line is typically defined to the right of the surge line to act as a buffer between the operating region and unstable region. Surge margin indicates how close the surge avoidance line is to the true surge line. A simple illustration is shown in Fig. 1.2.

Surge avoidance schemes force the compressor to operate to the right of the surge line by actuating a device when the compressor operating point is near the surge avoidance line. Surge margin impacts compressor performance during transient events where the operating point tends to be driven towards the surge line. Thus, one should

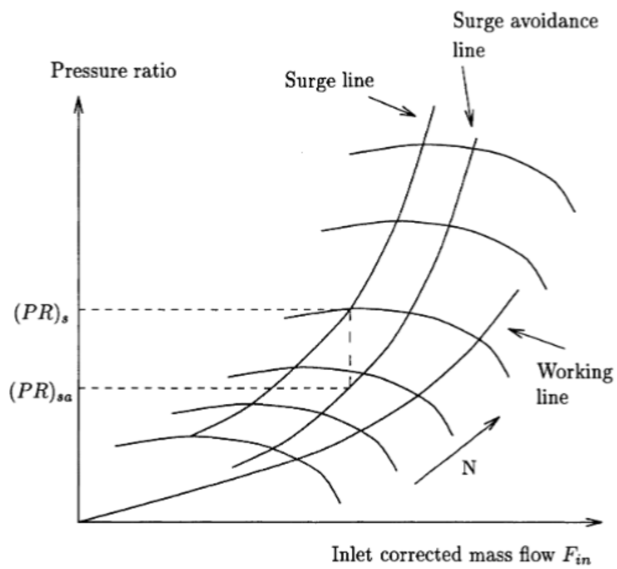

Figure 1.2: Sample compressor map for basic surge avoidance scheme (Gravdahl et al. 1999).

strive to make the surge margin as small as possible to maximize the compressor operating range, especially during transient events.

To constrain the operating point to the stable region in surge avoidance schemes, various methods are possible. A common approach is implementing pressure relief devices such as bleed valves and recirculation devices. Another method of preventing compressor surge is with passive control. Passive control uses detection to avoid surge by measuring certain compressor parameters and controlling the throttle position such that a predetermined value is not exceeded (Gravdahl et al. 1999). An alternative to passive control is active control. Active control uses a feedback system to expand the stable operating range of the compressor. The advantage of this method over passive control is that it can stabilize some portion of the unstable area while also allowing the compressor to use all of the traditionally stable area. Active surge control was introduced into literature by Epstein et al. (1989) where they demonstrated the ability to suppress surge via control of a CTV and moveable plenum wall. Pinsley et al. (1991) managed to use a proportionally controlled CTV to suppress surge without a moveable plenum wall. Active control provides the largest stable regime of any known method but is more challenging to implement and can be highly sensitive to system components.

Another approach is to slow the closing profile of a CTV. Slowing the closing time of the CTV can prevent the onset of surge as mass flow rate is decreased less rapidly. However, slowing the closing time increases the net mass flow rate to the engine compared to the standard profile which negatively impacts engine performance. Galindo et al. (2022) attempted to find the optimal closing profile for a singular tip-out case. They successfully showed that surge could be avoided during rapid load decreases on a spark-ignition engine by only optimizing the closing profile of a throttle valve.

1.3 Objective

There is motivation among auto manufacturers to find ways of preventing surge without CBVs or similar devices to reduce engine complexity and costs. The approach put forth by Galindo et al. of optimizing the closure profile of a throttle valve was particularly interesting due to its relative simplicity and opportunities for advancement. The objective of this thesis is therefore to control the closing profile of a CTV such that surge can be prevented without a CBV for a range of tip-out cases, while maintaining a minimal increase in net mass flow rate when compared to the standard throttle profile.

Chapter 2 discusses the test setup and procedure for conducting tip-out experiments. It also discusses the creation and implementation of the scheme used to control the CTV profile. Chapter 3 provides results showing the functionality of the control scheme and how the new profiles compared to the base profile. Chapter 4 provides final remarks and areas for future improvement.

Chapter 2. Methodology

2.1 Test Bench Overview

All experiments are conducted in the Turbocharger Laboratories at the Center for Automotive Research of The Ohio State University. The turbine and compressor operate on isolated flow paths, and the turbine is driven using air from a screw compressor. Decoupling the mass flow rates extends the operating range over which compressor measurements can be taken. Computer controlled valves are used to alter turbine and compressor conditions. A schematic of the flow paths including valves is shown in Fig. 2.1.

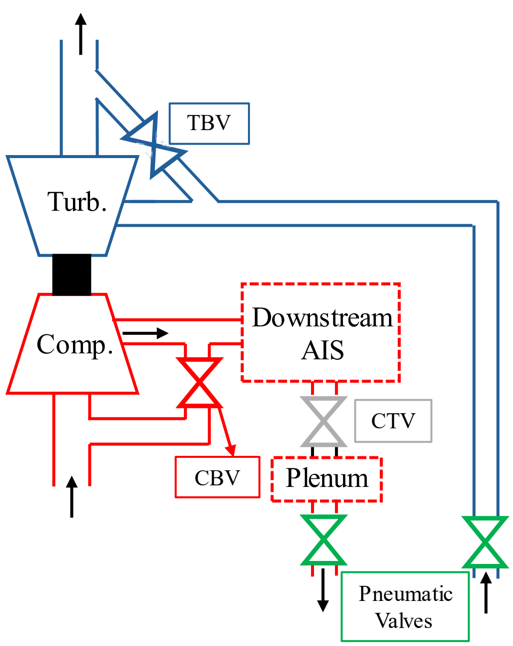

Figure 2.1: Flow circuit schematic of test bench.

During transient events, a turbine bypass valve (TBV) is used to control the flow rate through the turbine by diverting a portion of the flow through a secondary path. The CTV is located downstream of the compressor and is used to control the flow rate through the compressor. Slower acting pneumatic valves can also be used to control the turbine and compressor flow rates. For tip-out tests, the pneumatic valves set the maximum flow rate through each flow circuit and the CTV and TBV are used for finer control during the transient event. The CBV provides an open path between the inlet and outlet of the compressor and is used in this study to vent excess pressure during transient events.

2.2 Tip-Out Test Procedure

A tip-out case can be fully defined by the compressor mass flow rate and rotational speed of the initial and final operating points, along with the CTV, CBV, and TBV actuation profiles. For this study, the initial operating point for all tip-outs had the CTV at 90° or fully open and the TBV 0% open or fully closed. The final operating point had the TBV partially or fully open and the CTV nearly closed. The path traveled from the initial to final operating point is determined by the valve profiles of the CTV, TBV, and CBV. An explanation for how the valve profiles were constructed and how the initial and final operating points were chosen is provided at the end of section 2.3.

To find the initial operating point, the CTV and TBV are set to their initial positions, and the slower acting pneumatic valves for the compressor and turbine are adjusted until the compressor mass flow rate and corrected rotational speed match the target initial operating point. Once the pneumatic valve positions have been determined, they are fixed for the remainder of the test. The pneumatic valves allow the initial CTV and TBV positions to be constant regardless of the initial operating point of the tip-out. To find the final TBV and CTV positions, the TBV and CTV are manually adjusted until the final operating point has been reached.

After the initial and final operating points have been defined, the turbocharger is returned to the initial operating point by fully closing the TBV and fully opening the CTV. Before the transient event begins, the turbocharger is required to reach steady-state which is defined by the following parameters: standard deviations for rotational speed and mass flow rate over two minutes are less than 100 rpm and 2 g/s respectively, and the standard deviation over two minutes for the total-to-total isentropic compressor efficiency is less than 0.2%. Once steady state has been achieved, the CTV, TBV, and CBV are actuated using the pre-determined profiles such that the final operating point is reached. After the necessary data has been saved, the tip-out test has concluded.

2.3 Surge Prevention Approach

The approach taken by Galindo et al. (2022) as outlined in section 1.2 was promising as it showed that surge could be prevented without a CBV by only adjusting the closing profile of the CTV. The major limitation was that the optimal profile was manually determined through iterative testing for a single tip-out case. To generalize this approach to all tip-outs in the compressor's operating range, a method that could automatically control the CTV angle during transient events needed to be developed. To accomplish this objective, a relationship between CTV angle and rotational speed was established. Rotational speed was chosen as this measurement has less noise than that of the pressure ratio.

Since one of the objectives of the control scheme is to minimize the increase in net mass flow rate, it is necessary to operate as close to the surge line as possible during tip-outs. To accomplish this, it was necessary to construct a rough surge line in terms of CTV angle and rotational speed. The CTV angle is the valve's position in degrees from fully closed where the fully open position is 90°. To identify the CTV angles that resulted in surge, rotational speed was held constant, and the CTV was slowly closed until the compressor entered surge. At this point the compressor was moved just out of surge and the CTV angle was noted. This process was completed for five rotational speeds. The angle where the compressor is predicted to enter surge for a certain rotational speed will be referred to as the "critical angle". For rotational speeds where this angle was not determined experimentally, a linear interpolation is used between the experimentally available values. The measured critical angles for the five corrected rotational speeds are listed in Table 2.1.

Table 2.1: Experimentally obtained critical angles at various rotational speeds.

N_{cor} [krpm]	25	50	72.8	98.2	117.2
Critical Angle [deg]	13.4	20	20.1	21.6	22.7

Due to lack of accurate extrapolation, the corrected speed range of 25–117.2 krpm was chosen as none of the current tip-out cases exceed those bounds. Corrected speed was used such that the measured angles would be adjusted to standard conditions.

With the relationship between CTV angle and corrected rotational speed established, the control scheme could be implemented into LabVIEW. The basic premise of the control scheme is to ensure the CTV angle for a certain rotational speed is never smaller than the critical angle. The critical angle is assumed to be on the surge boundary, thus CTV angles smaller than the critical angle would result in flow rates that are predicted to fall left of the surge line. As a result, angles smaller than the critical angle for a certain rotational speed needed to be avoided.

Margin was introduced into the control scheme by means of a constant multiplier. After the critical angle is computed from the values in Table 2.1, a multiplier slightly greater than one is applied; the resulting angle will be referred to as the "adjusted critical angle". The multiplier was necessary to allow the scheme to work reliably and for all tip-outs as it can be tuned to ensure sufficiently conservative operation. Increasing the multiplier increases the adjusted critical angle. This pushes the operating range of the control scheme to higher flow rates away from the predicted surge boundary. Some margin was needed to account for error in computing the critical angle, variance between tests, and to ensure the compressor remained stable during aggressive tip-outs. Since a higher multiplier results in a more conservative operation, the aim was to identify the lowest constant multiplier that resulted in stable operation for all tip-out tests.

During the tip-out, the compressor rotational speed and actual CTV angle is measured and input to the controller. Table 2.1 uses corrected rotational speed while the controller uses regular rotational speed; thus, calculating the critical angle involves a small error. Time constraints, however, did not allow an update to use corrected rotational speed. Once the critical angle has been computed, the adjusted critical angle is calculated by applying the constant multiplier to the critical angle. Next, the adjusted critical angle is compared to the current CTV angle set by the base profile. If the CTV angle of the base profile is lower than the adjusted critical angle, the compressor is assumed to be at high risk of entering surge. Thus, the base profile is overwritten, and the CTV position on the test stand is set to the adjusted critical angle. If the CTV angle of the base profile is greater than the adjusted critical angle, such as at the beginning of the tip-out, the compressor is assumed to be stable. Thus, the adjusted critical angle is ignored, and the CTV will be set to the position determined by the base profile. A flow chart for the control scheme is shown in Fig. 2.2.

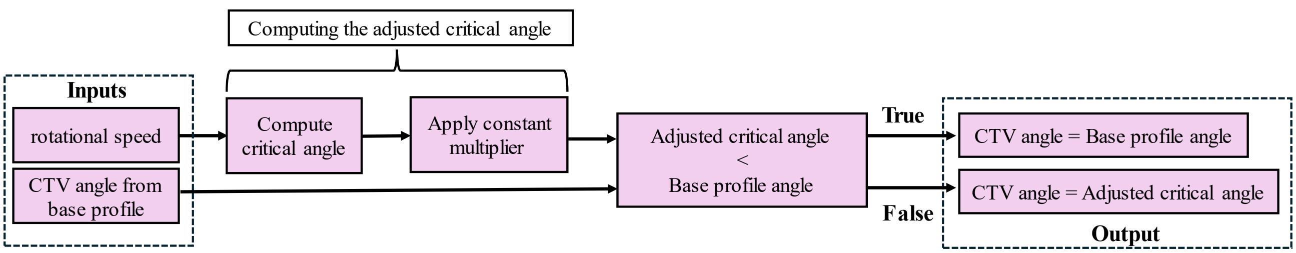

Figure 2.2: Logic flow chart for control scheme.

The control scheme was tested on five different tip-out cases to determine its effectiveness at various points in the compressor's operating range. These cases were primarily chosen to align with current experiments conducted in the Turbocharger Laboratory. The initial and final operating points for the five cases are listed in Table 2.2 and shown on the compressor map in Fig. 2.3.

Table 2.2: Compressor initial and final operating points for all five tip-out cases.

Tip-Out #	$\dot{m}_{\mathrm{cor}}$ [g/s]		N_{cor} [krpm]		$\mathrm{PR}_{\mathrm{tt}}$ [-]	
	Initial	Final	Initial	Final	Initial	Final
1	64.16	8.55	66.90	26.93	1.28	1.05
2	76.71	8.71	85.51	30.17	1.50	1.06
3	91.42	11.38	98.11	38.10	1.68	1.09
4	105.82	16.04	107.64	45.90	1.82	114
5	121.15	20.11	117.20	56.19	2.01	1.20

Tip-out #1 was constructed to match vehicle test data and forms the basis for the other four tip-out cases. The initial operating point for tip-out #5 falls on the fastest constant speed line that is used for testing and is roughly equidistant from the surge and choke points. The initial operating points for tip-outs #2-4 were chosen by spacing them between tip-outs #1 and #5 along a roughly straight line. The final operating points for tip-outs #2–5 were chosen to be as close as possible to the final operating point for tip-out #1. The final operating points are placed near the surge line to ensure the tip-out cases resulted in surge when no CBV was used. All initial and final operating points fall within the speed range of 26.93–117.20 krpm. Thus, the speed range of 25.0–117.2 krpm listed in Table 2.1 is sufficient for computing the critical angle as no extrapolation will be required.

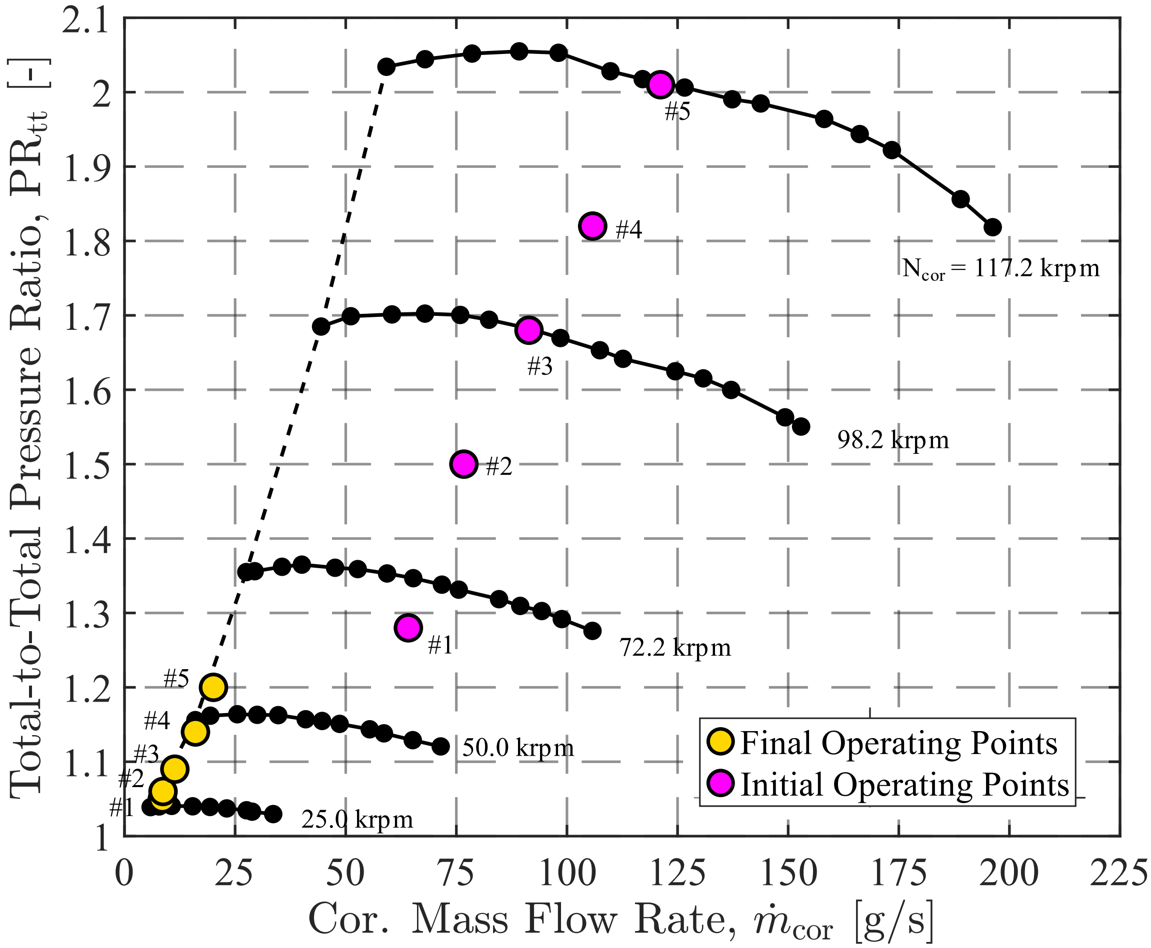

Figure 2.3: Initial and final tip-out operating points on compressor map.

The base CTV profile for tip-out #1 was derived from the vehicle test, and this profile is normalized for the other tip-outs based on the final CTV angle. The initial CTV position is 90° or fully open for each tip-out case. The TBV start position for all tip-outs was 0% open or fully closed and the final position was based on the final operating point. Sample CTV and TBV profiles are shown in Fig. 2.4.

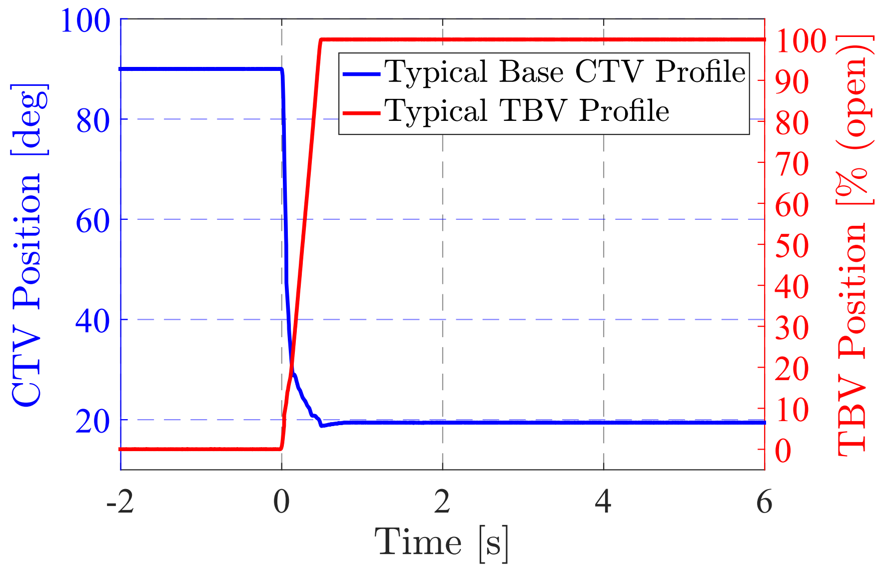

Figure 2.4: Sample CTV base and TBV profiles.

The TBV has a linear profile from its initial to final position with the opening duration determined to match the closing time of the CTV. The final TBV position for tip-out #1 was determined by matching the final operating point to vehicle test data. For tip-outs #2–5, the final TBV position was fully open to get the final operating point as close as possible to the final operating point for tip-out #1. Due to the higher initial flow rates for tip-outs #2–5, it was not possible to reach the final operating point of tip-out #1. The CBV actuation profile is the same for all tip-out cases. For tests where the CBV was used, it opened fully at the beginning of the tip-out and closed after 2.2 seconds.

3.1 Controller Tuning

The constant multiplier applied to the interpolated critical angle affects the operating range of the controller with higher multipliers resulting in more conservative operation. To minimize the increase in net mass flow rate compared to the base profile, it is necessary to operate as close to the surge line as possible. Thus, the aim was to determine the smallest or least conservative multiplier that resulted in stable operation for all tip-out cases. The controller was first tested with various multipliers on tip-out #2 (recall Table 2.2) as it was a relatively low-speed case. The resulting CTV profiles for different multipliers are shown in Fig. 3.1.

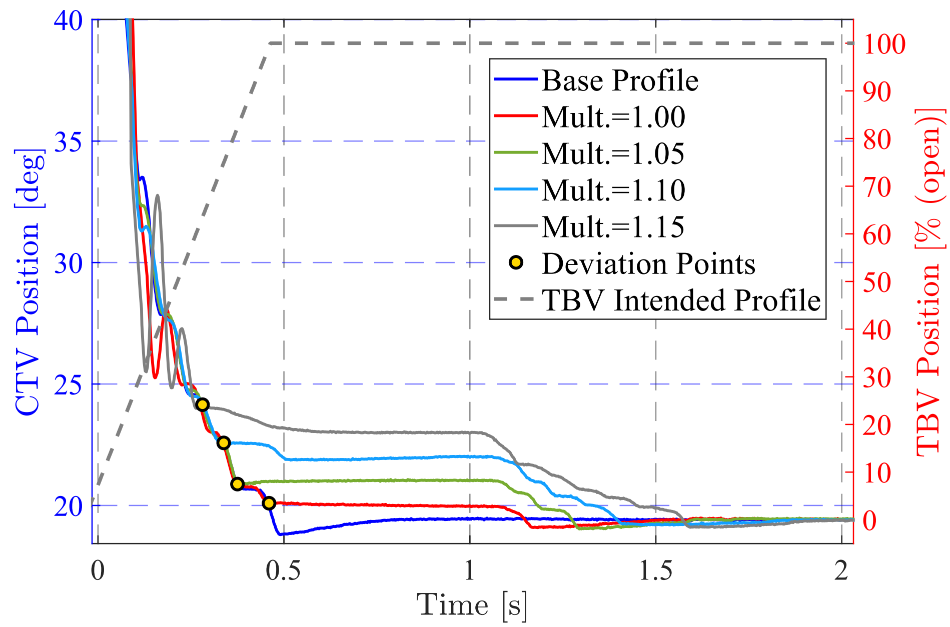

Figure 3.1: Tip-out #2: CTV profiles for multipliers ranging from 1.00–1.15.

The CTV profiles controlled with the newly constructed logic will be called "adjusted profiles". The deviation points are where the adjusted profiles deviate from the base profile, and the CTV angle at which the deviation occurs will be referred to as the "deviation angle". The logic behaved as intended with respect to the varying multipliers. The multiplier changed the CTV angle at which the adjusted profile deviated from the base profile with higher multipliers resulting in higher deviation angles. Additionally, adjusted profiles associated with higher multipliers took longer to reach the final CTV angle. This behavior was expected as profiles with higher multipliers require lower rotational speeds to reach the final CTV angle. The optimal multiplier can be determined by analyzing the compressor outlet pressure and mass flow rate during the tip-out to determine which multipliers resulted in surge. Compressor outlet pressure and mass flow rate for various multipliers are plotted in Fig. 3.2 and Fig. 3.3 respectively.

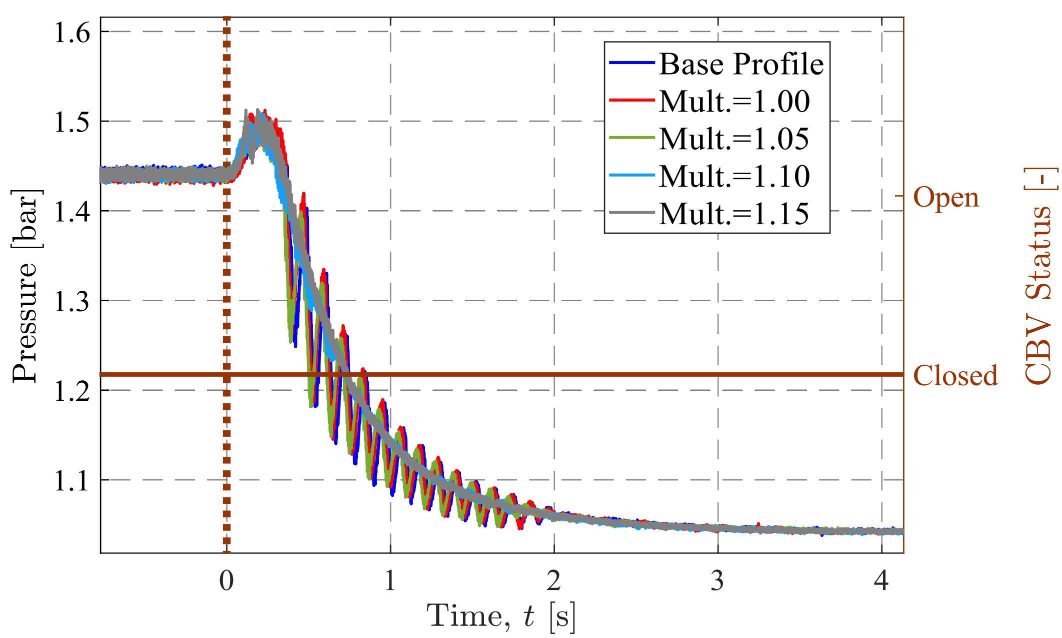

Figure 3.2: Tip-out #2: Compressor outlet pressure for multipliers ranging from 1.00–1.15.

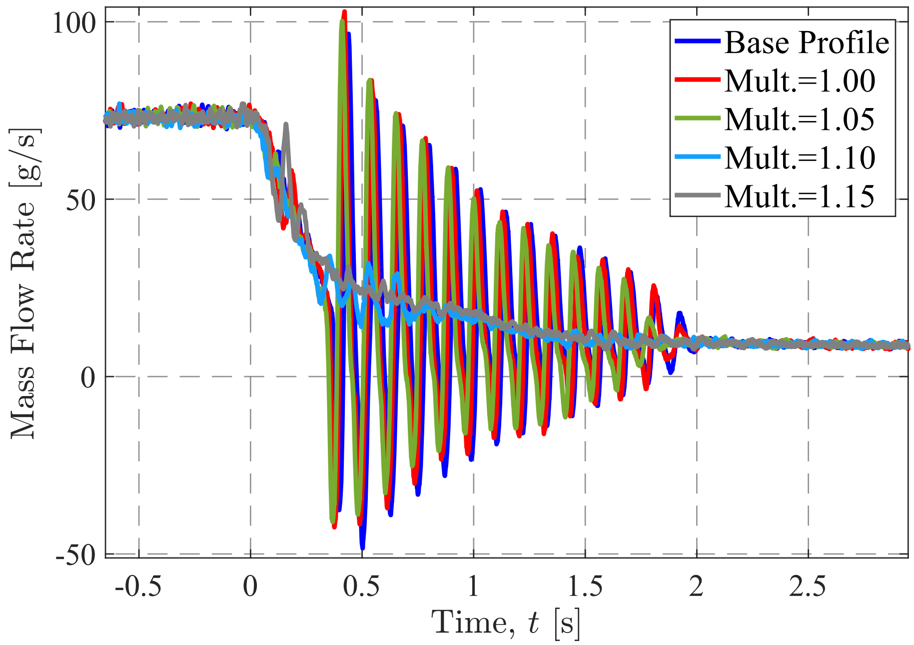

Figure 3.3: Tip-out #2: Mass flow rate for multipliers ranging from 1.00–1.15.

The oscillations exhibited by the base profile and two smallest multipliers in Fig. 3.2 and Fig. 3.3 indicate the compressor entered deep surge during the tip-out. By the same logic, multipliers of 1.10 and 1.15 successfullly prevented the compressor from entering deep surge. A multiplier of 1.10 appears to exhibit small amplitude mass flow rate oscillations which could be indicative of mild surge. The tip-out path for multiplier 1.10 was plotted on the compressor map in Fig. 3.4. The mass flow rate does seem to oscillate across the surge line indicating that mild surge was briefly induced. Given that mild surge is less violent than deep surge, a multiplier of 1.10 was temporarily established as the optimal multiplier.

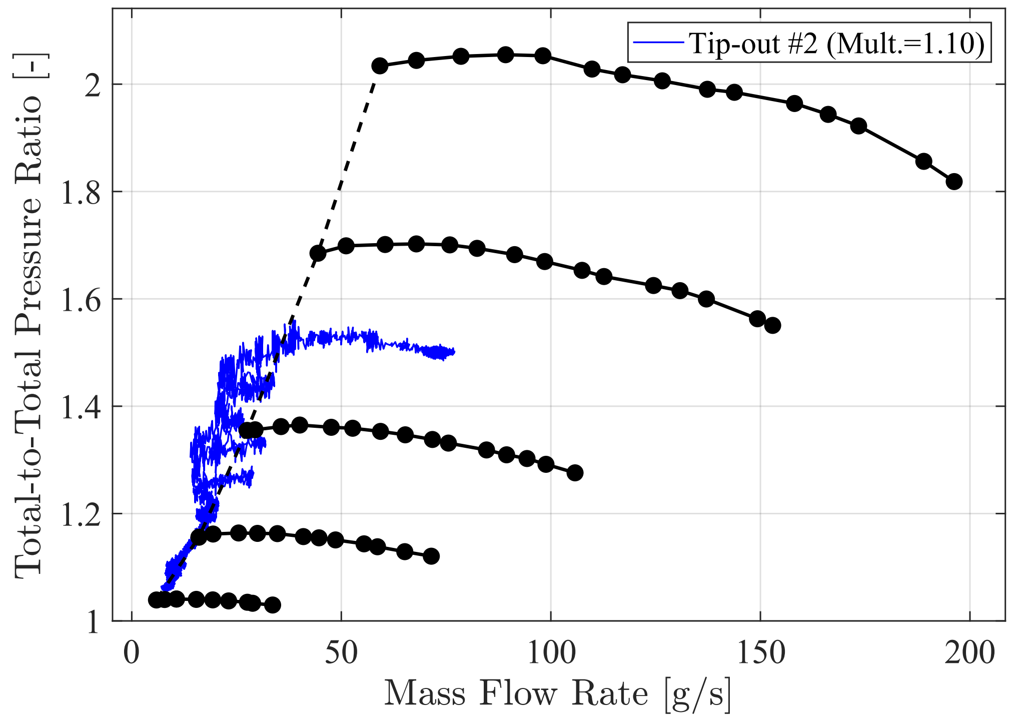

Figure 3.4: Tip-out #2 path plotted on compressor map (Multiplier=1.10).

While a multiplier of 1.10 supressed deep surge for tip-out #2, it was unclear if it would do the same for all tip-out cases. The process for determining the optimal multiplier was repeated for tip-out #5, and the adjusted CTV profiles for the multipliers tested are shown in Fig. 3.5 and Fig. 3.6.

The multipliers again altered the CTV profiles as expected with higher multipliers resulting in higher deviation angles. The main difference from the tip-out #2 results are the rapid closures at 2 seconds and 5 seconds seen in Fig. 3.5. These closures are a result of the logic and how it integrates with the rest of the testing software. For tip-out #5, the final CTV angle is lower than the adjusted critical for the final rotational speed.

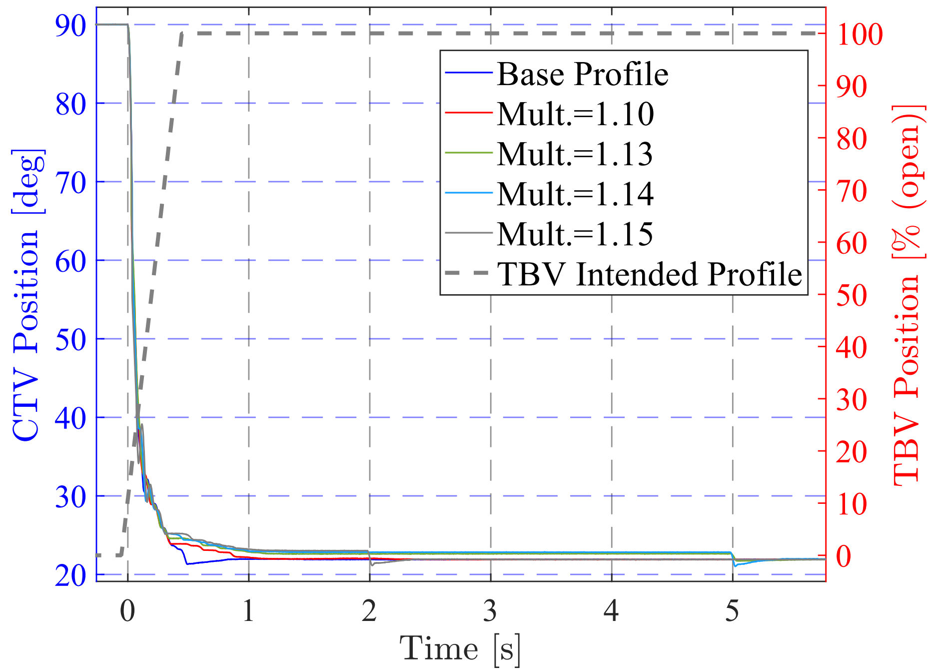

Figure 3.5: Tip-out #5: Full CTV profiles for multipliers ranging from 1.10–1.15.

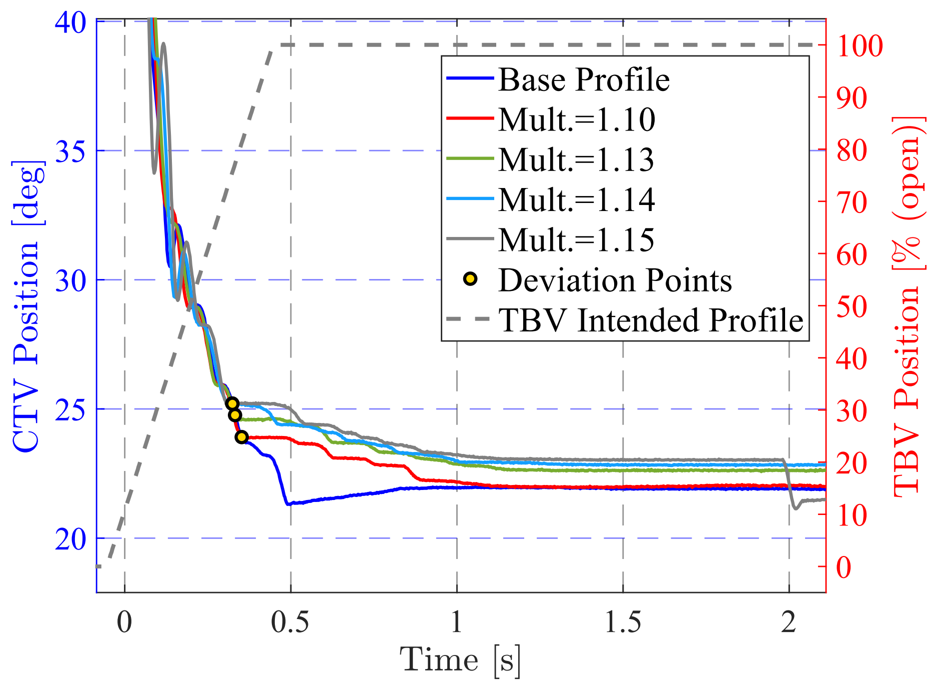

Figure 3.6: Tip-out #5: Section of CTV profiles for multipliers ranging from 1.10–1.15.

Thus, the logic prevents the CTV from reaching the final angle until the logic is deactivated at which point the CTV is forced close to the final position. The compressor outlet pressure and mass flow rate are analyzed to determine which multipliers resulted in surge for tip-out #5. These quantities are plotted in Fig. 3.7 and Fig. 3.8 respectively.

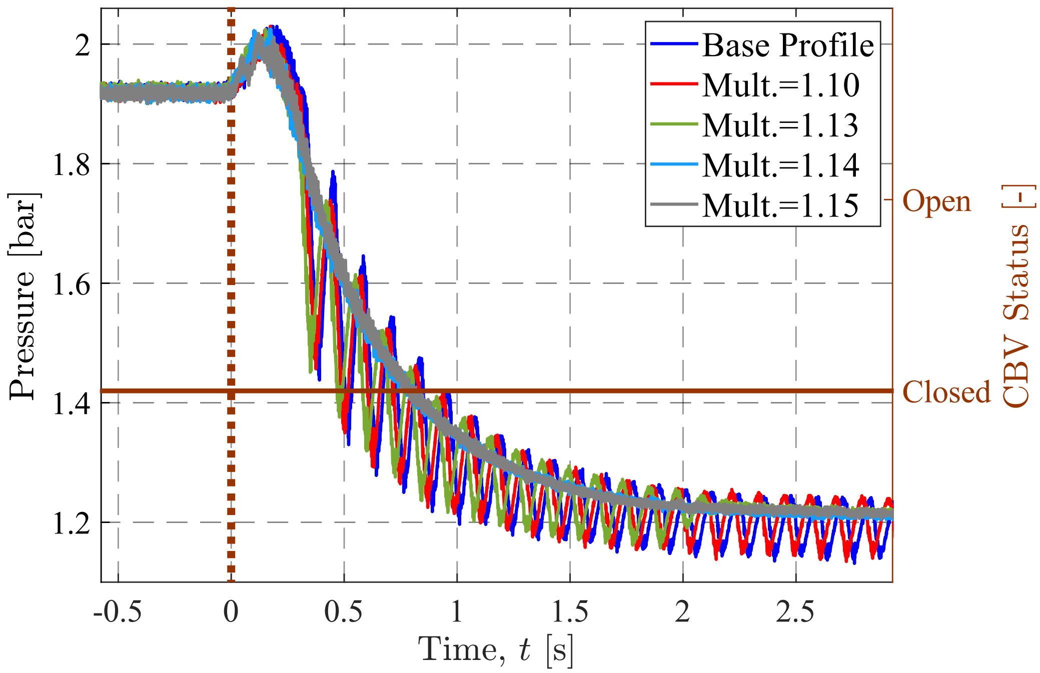

Figure 3.7: Tip-out #5: Compressor outlet pressure for multipliers ranging from 1.10–1.15.

The large amplitude oscillations exhibited by multipliers 1.10–1.13 in Fig. 3.7 and Fig. 3.8 show that the compressor entered deep surge. Conversely, multipliers 1.14 and 1.15 kept the compressor out of deep surge thus both multipliers resulted in stable operation. Since 1.14 was the lowest multiplier to prevent surge, it was determined to be the optimal multiplier for tip-out #5. This was higher than the optimal multiplier of 1.10 that was determined for tip-out #2. This showed that different tip-out cases required

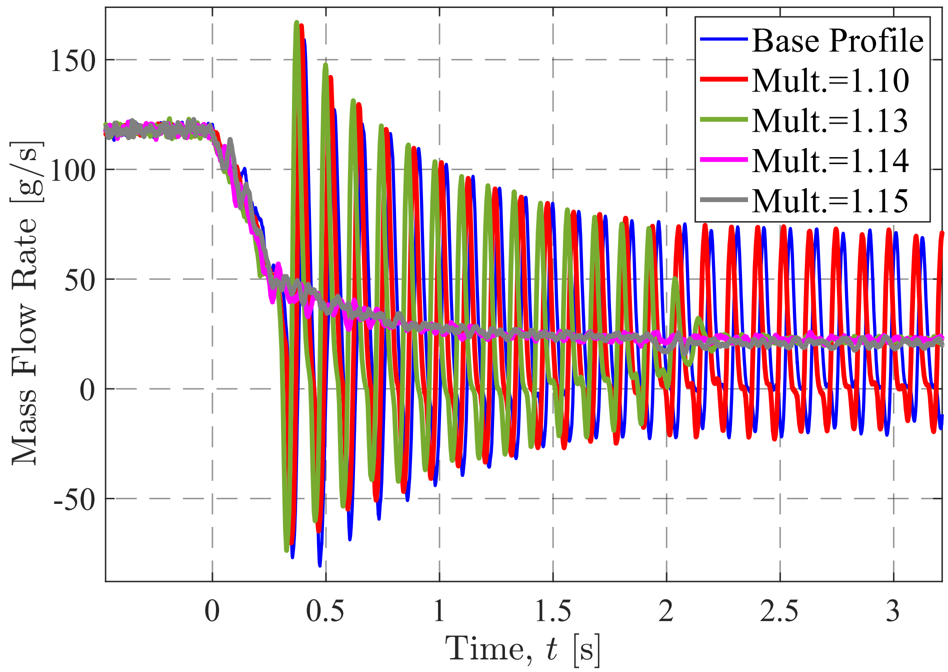

Figure 3.8: Tip-out #5: Mass flow rate for multipliers ranging from 1.00–1.15.

different multipliers. Since tip-out #5 had the highest initial speed, it was assessed that it would be the most aggressive case thus requiring the most conservative multiplier.

With a multiplier of 1.14 able to suppress surge for tip-out #5, it was expected to also prevent surge for the four other tip-out cases. To determine if a multiplier of 1.14 was sufficiently conservative, it was applied to tip-outs #1–4. The compressor outlet pressure and mass flow rate can be analyzed to determine if a multiplier of 1.14 resulted in surge for any of the tip-outs cases. These quantities are plotted in Fig. 3.9 and Fig. 3.10 respectively. From these figures, it can be concluded that a multiplier of 1.14 results in stable compressor operation as there is no evidence of large amplitude oscillations or flow reversal.

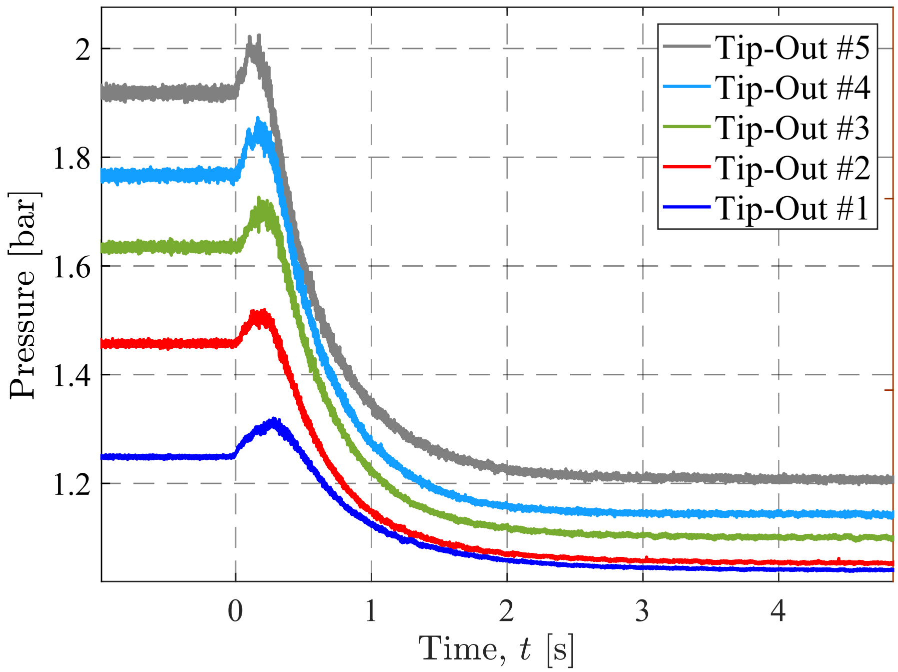

Figure 3.9: Compressor outlet pressure for tip-outs #1–5, (Multiplier=1.14).

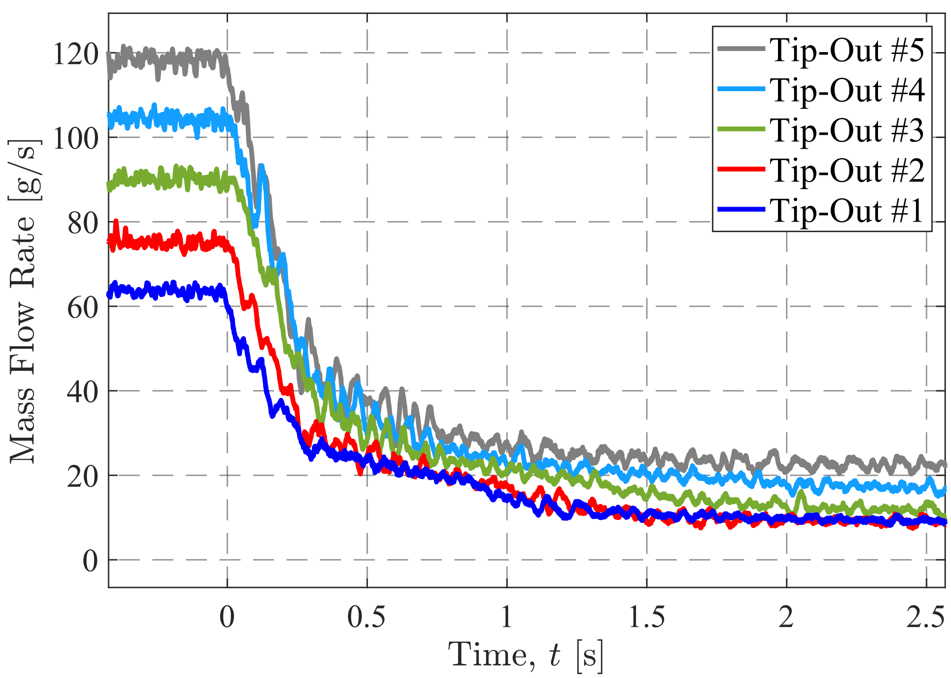

Figure 3.10: Compressor mass flow rate for tip-outs #1–5, (Multiplier=1.14).

In Fig. 3.11, the tip-out paths were plotted on the compressor map to examine how close to the surge line the logic operated.Tip-outs #2–5 operate to the left of the surge line for a substantital portion of the tip-out indicating that the compressor is close to the stability limit.

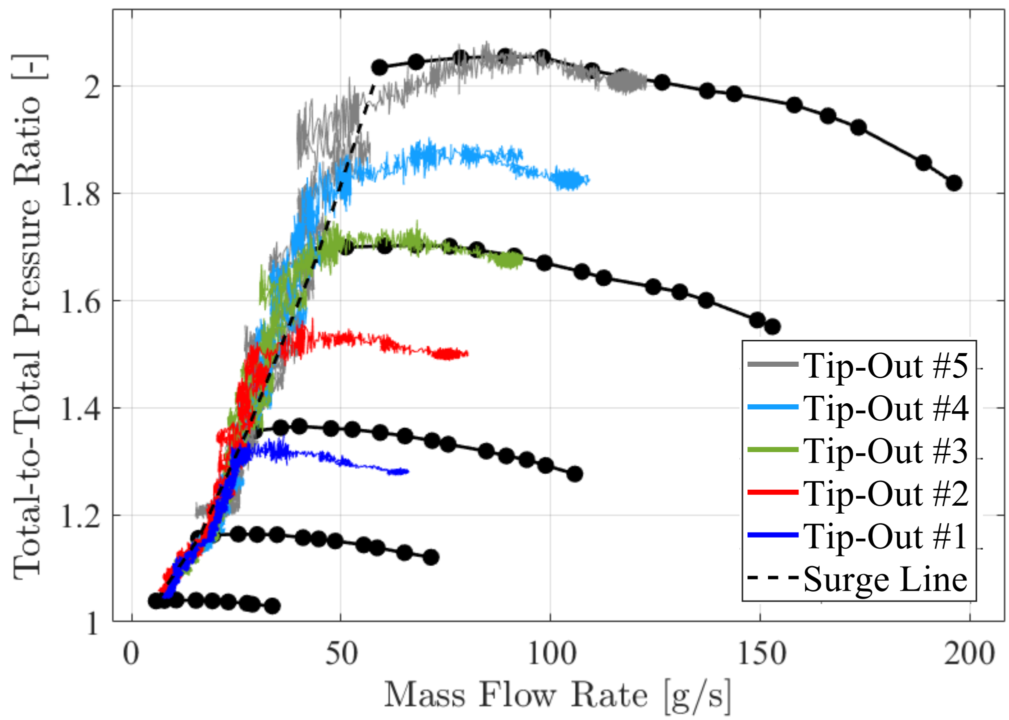

Figure 3.11: Paths for tip-outs #1–5 plotted on compressor map, (Multiplier=1.14).

This shows that a multiplier of 1.14 is near optimal for the current logic implementation as operating in the neighborhood of the stability limit ensures that there is a minimal increase in net mass flow rate when compared to the base profile. Additionally, all tip-outs rapidly approach the surge line from their intial operating points. This is favorable for minimizing mass flow rate as it ensures the compressor operates near the stability boundary for a significant portion of the tip-out.

3.2 Controller Performance Compared to CBV

Given that the controller developed in this study aims to replace the need for a CBV or similar device, it was necessary to compare the performance of the logic to that of the CBV. Only tip-outs #1 and #5 will be analyzed in this section as they form the lower and upper bounds of the experimental data. “New Logic” refers to tests run with the controller developed in this study. All tests using the controller have a multiplier of 1.14 that was established in section 3.1. To understand how the new logic behaves, the adjusted CTV profiles from the new logic are compared to the base CTV profiles with CBV actuation. The profiles for tip-out #1 and tip-out #5 are shown in Fig. 3.12 and Fig. 3.13 respectively.

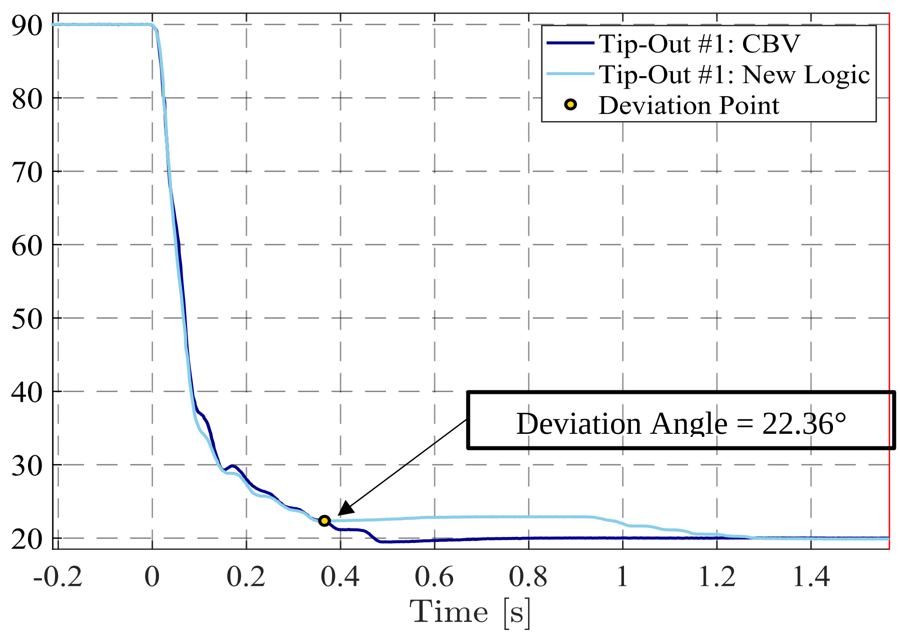

Figure 3.12: Tip-Out #1: CTV profiles for tests with CBV and New Logic

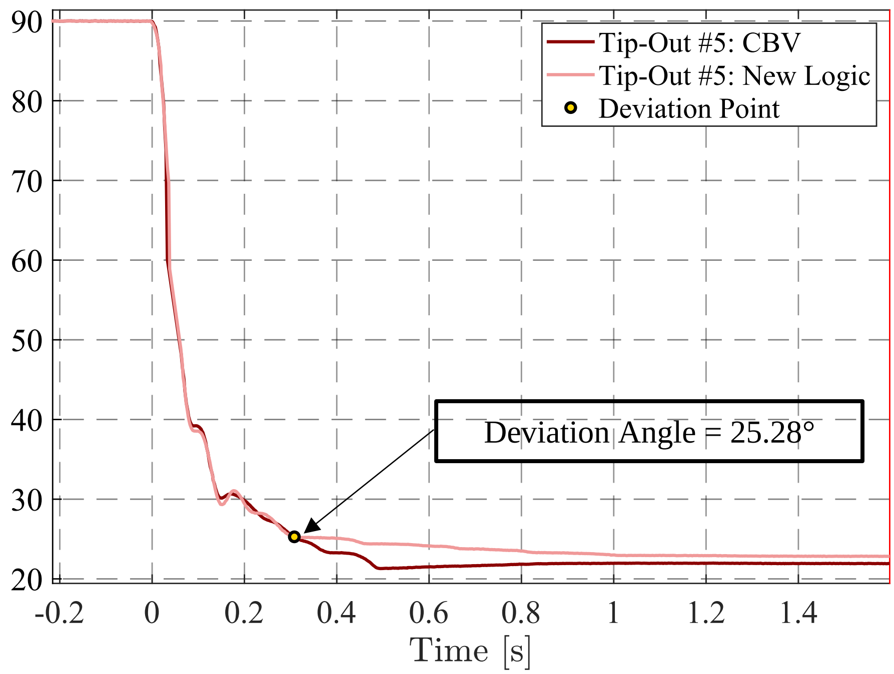

Figure 3.13: Tip-Out #5: CTV profiles for tests with CBV and New Logic

The adjusted profiles take longer to reach the final CTV angle than the base profile. As mentioned earlier for tip-out #5, the final CTV angle for the adjusted profile does not match the final CTV angle for the base profile due to how the controller was constructed. To further examine how the adjusted profiles compare to those of the CBV, compressor rotational speed and compressor outlet pressure were plotted in Fig. 3.14 and Fig. 3.15 respectively.

Fig. 3.14 also shows that the CBV reduces the compressor's rotational speed more rapidly than the new logic. Rapidly reducing rotational speed is favorable for surge prevention during tip-outs as lowering speed decreases the minimum stable mass flow rate. Similarly, the CBV allows the compressor outlet pressure to decrease quicker than the adjusted profiles which is also favorable for surge prevention.

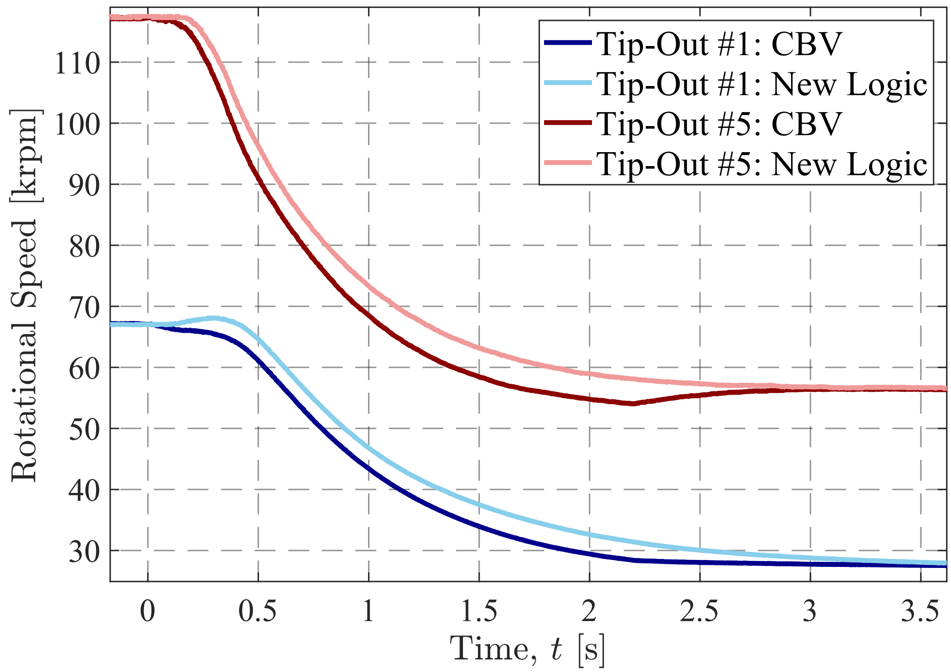

Figure 3.14: Compressor rotational speed comparison between CBV and New Logic for tip-outs #1 and #5.

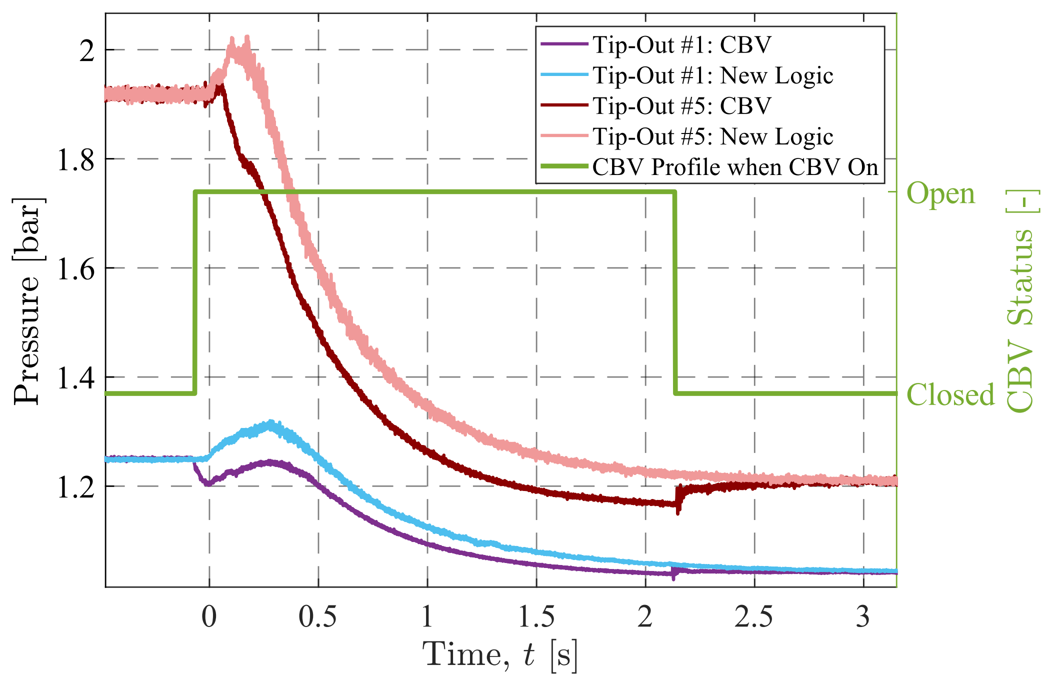

Figure 3.15: Compressor outlet pressure comparison between CBV and New Logic for tip-outs #1 and #5.

Additionally, the adjusted CTV profiles have a brief increase in outlet pressure at the start of the tip-out due to rapidly closing the CTV. This is less prevalent for the CBV profiles as there is an alternative path for venting excess pressure. A similar analysis is done for mass flow rate which is shown in Fig. 3.16.

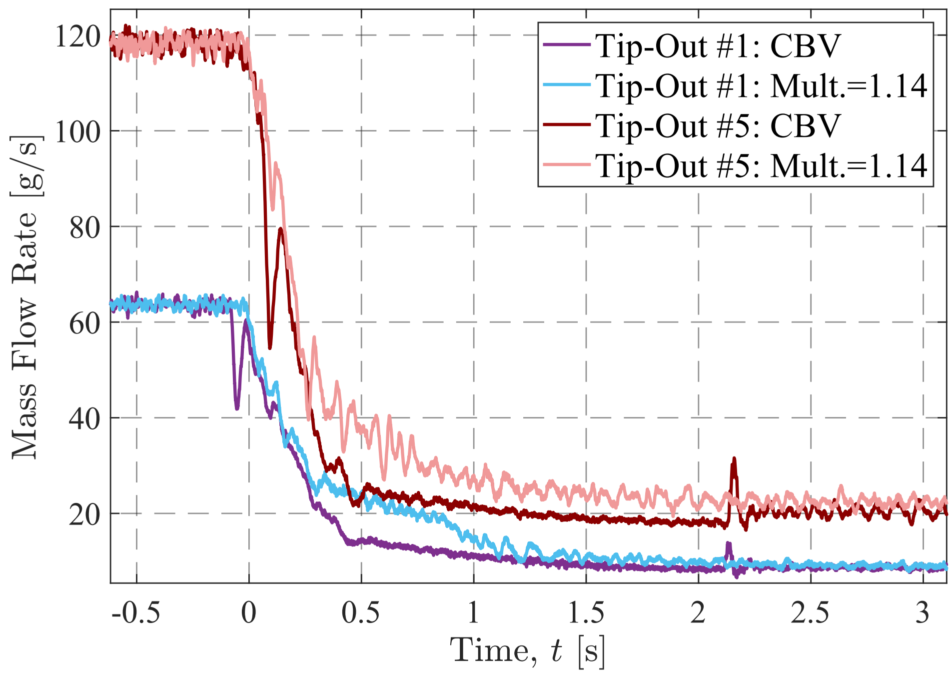

Figure 3.16: Compressor mass flow rate comparison between CBV and New Logic for tip-outs #1 and #5.

The adjusted CTV profiles using the new logic increases the net mass flow rate from the compressor when compared to the CBV profiles. This is undesirable as it would alter the engine inlet manifold conditions. Since the new logic operates near the stability limit of the compressor, the mass flow rate cannot be significantly reduced further without further modifications to the developed controller.

Chapter 4. Conclusions

The objective of this study was to prevent compressor surge during tip-outs without the need for a bypass valve by controlling the closure profile of a CTV. The developed controller resulted in stable operation of the compressor by estimating the CTV angle that would result in surge and ensuring that the actual CTV angle for a given rotational speed was never less than this critical value. This approach improved on earlier attempts by automatically controlling the CTV closure profile which allowed surge to be successfully prevented for a range of tip-out cases. To minimize the increase in net mass flow rate compared to the base CTV profile, the controller was tuned such that the CTV operated near the low flow stability limit during the tip-outs. This indicated that the controller was optimized in its current implementation as further decreasing the closure time would be expected to result in surge for some of the tip-out cases.

A turbocharger test stand was used to conduct the experiments. The CTV angles resulting in surge for five corrected rotational speeds were measured and termed the "critical angles". A linear interpolation was used between the experimental values to determine the critical angle for a given rotational speed. A constant multiplier was applied to the critical angle with higher multipliers resulting in more conservative operation. The multiplier was tuned for tip-out #5 as it was the most aggressive case thus requiring the most conservative multiplier. With the controller tuned, it was tested on five tip-out cases ranging from initial speeds of 66.0–117.2 krpm. Surge was successfully prevented for all five cases.

The controller developed in this study could still be improved. The critical angles are calculated by linearly interpolating within experimental values. Errors associated with the linear interpolation could be reduced by increasing the number of experimentally measured critical angles. The critical angles measured for rotational speeds of 25 krpm and 50 krpm differ by 6.6° while the critical angles measured for speeds of 72.8 krpm and 117.2 krpm differ by 2.6°. Thus, measuring more critical angles in the 25–50 krpm range would have the most significant improvement on the interpolation accuracy as the critical angle varies more over this range than at higher speeds. Secondly, the lookup table for the linear interpolation contains corrected rotational speed while the input to the controller is regular rotational speed. Changing the controller input to corrected rotational speed would decrease error associated with using two different parameters. For tip-outs #4 and #5, the logic prevented the CTV from reaching the final CTV angle. The controller should be modified to allow the CTV to reach the final angle regardless of the tip-out case. Lastly, a lookup table similar to that used for computing the critical angles could be used to compute the optimal multiplier for a given tip-out. The optimal multiplier for tip-out #2 was smaller than the optimal multiplier for tip-out #5. Since the multiplier of tip-out #5 was used in the final implementation, the controlled CTV profile for tip-out #2 would be slightly conservative compared to that of its optimal multiplier.

Thus, adjusting the multiplier based on the tip-out's initial operating point would improve the controller's performance for cases less aggressive than tip-out #5.

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