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Modelling of Rapid Transients in Natural Gas Pipelines

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Abstract

Many things can cause rapid transients in gas pipelines, such as the sudden shutdown of compressors, or the abrupt changes in flow rate due to leaks. The hydraulic simulation of these rapid transients requires a mathematical model capable of modelling the detailed rarefaction wave. This also enables the accurate analysis of the impact of sequential valve closures along the pipeline. To minimize potential flow interruptions the highly accurate modelling of the pipeline dynamics can be used to understand and tune the behavior of the automatic shutdown valve (SDV's). By simulating the pipeline behavior, offline analysis can be performed in a safe environment without interrupting pipeline operations.

Automatic shutdown valves are designed to automatically close mainline valves in the case of a major pipeline leak or rupture to minimize pipeline leakage. Calibrated to detect a sudden pressure differential drop between a reference reservoir and the pipeline, automatic shutdown valves can respond to more than just pipeline leaks. Improper actuation of the line break mechanisms during pipeline operations can result in unwanted interruption of the pipeline flow if the calibration is not optimized.

This paper discusses the pipeline behavior during the transient, the dynamic effects on the automatic shutdown valves, and covers an analysis of first and second order mathematical solvers. The results of the simulations are validated throughout with a comparison with field test measurements.

Introduction

A piping system is the most common method for transporting fluids between two locations. There are many challenges in the safe transportation of fluids in pipeline systems, particularly during rapid transients. The use of mathematical models to simulate these rapid transients makes pipeline operations safer.

Mathematical models enable detailed analysis of transients following sudden compressor shutdowns, and abrupt flow rate changes caused by leakages. This detailed analysis provides an understanding of the behavior of the rarefaction wave as it propagates from the source of the change.

Rarefaction is the reduction of the fluids density caused by decompression, and rarefaction travels in waves. This rarefaction wave creates an area of low relative pressure within the pipeline that moves at the fluid speed of sound. The decompressed low-pressure wave expands with time and a compressed section of high-pressure fluid narrows as it is further compressed.

The mathematical modelling of the rarefaction waves provides an opportunity to understand the detailed characteristics of rapid transients. This understanding enables the differentiation between different pipeline operations such as compressor shut-downs and valve closures and the effects of pipeline leaks and ruptures. Differentiating between normal pipeline operations and abnormal conditions allows the pipeline operator to tune the automatic controls for different pipeline states. A pipeline could, for example, shut-down by automatically closing valves around a pipeline segment to isolate it from the rest of the network in the case of a rupture. The pipeline however would remain operational and within a running state following the shut-down of a compressor (for example running with a reduced flow rate). The ability to differentiate between these states enables the tuning of automatic shut-down valves (Figure 1). Tuning optimizes the system response to pipeline ruptures and compressor shut-downs.

Rapid Transients

Events such as the sudden shut-down of a compressor or pipeline ruptures change the velocity of the moving fluid, causing rapid transients. The effects of these are considered below.

Sudden Shut-Down of Compressors

Compressors are an essential element in natural gas pipeline networks. As the natural gas travels along the pipeline, friction and elevation differences reduce the fluid pressure. Compressors are strategically placed along the pipeline to maintain the pressure and flow of the gas.

A variety of reasons can cause a sudden shut-down of a compressor unit or a compressor station. These causes include scenarios such as a power outage or a defect on the compressor. While running, the discharge pressure of a compressor on a large transmission pipeline is significantly higher than the pressure would be if the compressor stopped. A sudden compressor shut-down causes an abrupt reduction in the discharge pressure. This sudden drop in pressure creates a pressure wave that propagates along the pipeline.

Compressor stations integrate a selection of safety systems and practices, including automatic shut-down systems. These systems can detect abnormal conditions such as unanticipated pressure drops and pipeline leaks and ruptures. These systems automatically stop the compressor units and isolate the section of the pipeline to limit environmental damage in the case of a pipeline rupture. Regulations require operators to periodically test compressor stations and maintain the emergency shutdown system to ensure reliability.

Leaks

A natural gas pipeline can act as a storage facility for the gas, through packing and unpacking the pipeline. This allows the pipeline operator to meet the needs of consumers while maintaining a steady supply of gas to the pipeline. As the pipeline packs, the gas compresses, and the pressure within the pipeline increases.

A leak or rupture exposes the pipeline to atmospheric conditions at the location of the leak. The pressure within the pipeline at the leak location then drops as it tries to equalize with the atmospheric pressure. As the pressure drops, this pressure drop propagates along the pipeline as a pressure wave.

In the case of a large leak in a typical gas pipeline the flow chokes. If the leak is a large enough rupture along some distance of pipe, flow may become choked in the pipe. In choked flow the pipe pressure drops to about twice the ambient pressure, while pipe flow just reaches sonic velocity. This provides an upper limit to the flow rate in a rupture.

The Effects and Consequences of Rapid Transients

Rapid transients cause pressure waves that propagate along the pipeline. These waves create an area of relative decompressed low-pressure and an area of compressed high-pressure. As these waves travel along the pipeline, the high and low-pressure regions can interfere with pipeline equipment. Equipment such as automatic shutdown valves can react incorrectly to these transients during normal operational conditions such as a compressor station shut-down. In addition to causing the incorrect activation of automatic shut-down valves, the propagation of rapid transients can also cause adverse effects to pipeline equipment such as:

- Failure of static components through fatigue
- Failure of dynamic components leading to high fatigue loads on other components
- Failure of the piping system due to extreme pressures
- Ruptures on high-pressure gas pipelines can lead to very low temperatures due to Joule-Thomson cooling. These very low temperatures cause a brittle transition in the steel pipe, requiring the replacement of significant lengths of the pipeline.

The extreme pressures and high fatigue loads are observed during the expansion of the decompressed low-pressure rarefaction wave. As the decompressed area expands, a compressed section of high-pressure fluid narrows as it is further compressed.

Mitigating the Effects of Rapid Transients

Following a pipeline rupture, it is essential for the emergency shut-down controls to respond correctly. Failure to close the automatic shut-down valves can result in the unnecessary leakage of harmful emissions into the environment, which if ignited could result in an explosion. An uncontrolled stoppage can cause damage to pipeline equipment such as compressors that could be costly to repair.

A planned compressor stop is of paramount importance and must occur within a minimal time frame. Automated responses can be problematic. Automatic shut-down valves close main line block valves if pressure sensing mechanisms detect an unexpected drop within the pipeline pressure. This can lead to the incorrect actuation of the valves during normal operating procedures (e.g. a planned compressor shut-down) resulting in the unnecessary closure of pipeline mainline block valves. This can result in a costly restart procedure, combined with the loss of earnings while the pipeline is not operational. The automatic shut-down controls therefore must be adequately planned, and the equipment calibrated to avoid these scenarios.

Automatic Shutdown Valves

Automatic shut-down valves (also referred to as automatic line-break controls) act to promote the rapid isolation of sections adjacent to the area where there is a massive gas leak. Its main function is to reduce the release of natural gas, minimizing the economic impact, mitigating the associated environmental damages and avoiding negative publicity during an accident. Furthermore, recognized standards addressing the design of onshore transmission pipelines require the utilization of shutdown valves at spaced intervals defined per population density of the region.

A basic automatic shut-down valve (Figure 2) consists of a block valve, a pressure sensing mechanism and an actuator that can physically open and close the block valve. The pressure sensing system comprises a pressure reference reservoir and an orifice located between the pipeline and the pressure reservoir. Two ports reside within the pressure sensing mechanism, one connects to the reference reservoir, and the second connects downstream of the orifice on the pipeline.

The pressure reservoir is normally pressurized to the pipeline pressure. A rapid decrease in the pipeline pressure creates a pressure differential between the pipeline and the reservoir. The orifice increasing the depressurization time of the reservoir compared with the pipeline causes this effect. Once the pressure differential reaches a pre-set level it triggers the actuator to close the block valve. The closure of the valve is usually pneumatic, instead of electric, using the transported gas itself to power the actuator that closes the valve.

The Problems with Automatic Shutdown Valves

Shutdown valves rely on automatic mechanisms of flow blocking, that fall into two categories, ‘low-pressure’ (or ‘minimum pressure) and ‘high rate of pressure drop’ (Figure 3). Modern automatic shut-down valves incorporate both closure trigger categories. The 'low-pressure' type, close the valve if the pipeline pressure falls below a pre-set level. The low-pressure setting is based on assumptions defined during the pipeline design project. The 'high rate of pressure drop' type, close the valve if the pressure drops faster than a pre-set rate. The adjustment of the set point used within this category can create serious problems for the pipeline operator, such as the incorrect closure of the shut-down valve, especially when the pipeline is prone to experience large operational transients.

The speed of the pressure drop in a pipeline varies depending on the operational conditions and causes problems with the 'high rate of pressure drop' automatic shut-down valves. Transients imposed by a compressor station or an increase in gas withdrawn from the main line can significantly affect the pipeline. The rate of pressure drops caused by these operational conditions can be comparable in magnitude to pipeline leaks. This creates a challenge to determine and characterize the pressure drops caused by operational transients and the ones caused by leaks.

The line break mechanism must be sufficiently sensitive to identify and act in the event of a leak. The mechanism must not promote improper closure of the valve as, for example, in the event of a rapid stop of compressor stations or the sudden increase of flow at a demand point.

Calibrate Automatic Shutdown Valves

The behavior of the pressure sensing mechanism within the automatic shut-down valve can be calibrated to detect a pipeline rupture or major leak. The pressure sensing mechanism that monitors the pipeline pressure is compared with the pressure in a reference reservoir. Following a decrease in the pipeline pressure, the pressure in the reference reservoir also decreases. The rate of the reference reservoir pressure decreases can be controlled by a calibrated orifice. The calibration of the orifice enables a 'set-point' to be placed at which an actuator is activated to close the block valve.

Traditionally the calibration of the automatic shutdown involves performing an operation on the physical pipeline whereby a line break is simulated. The pressure levels in the pipeline and the reference reservoir are then monitored to determine a pressure differential across the orifice. This is a trial and error approach that does not cater for the behavior of the pipeline during other pipeline operations, and as such calibration can overlook these. Pipeline operations may cause the unnecessary closure of pipeline valves.

Testing and monitoring of pipeline operations during the calibration of the automatic shutdown valves on the physical pipeline can be extremely costly, time consuming and hazardous. The only practical way to evaluate the rate of pressure drop and to define the required pressure sensing mechanism sensitivity is by mathematical simulation due to the high costs associated with running studies on the physical pipeline. This enables the relative magnitude of ruptures to be analyzed and evaluated against other pipeline operations. Pipeline simulations should therefore be used to simulate these scenarios to achieve satisfactory calibration.

The calibration of the orifice involves adjusting the resistance coefficient across the orifice. Increasing the resistance reduces the fluid velocity. This drains the pressure reservoir at a slower rate than the pipeline, creating a pressure difference. The magnitude of the pressure difference can then be used to trigger the actuator to close the block valve.

Analysis of Rapid Transients

Automatic line break control modules reside in the actuators of automatic shut-down valves. When the pipeline pressure suddenly falls because of a large leak (like those caused by a pipeline rupture), the control module automatically closes the valve. The control module of the line break detects the sudden pressure fall.

Mathematical models can demonstrate the sensitivity of the pneumatic line. The model simulates and predicts the dynamic response of the control module in terms of the pressure difference between the pressure reservoir and the pipeline.

Simulation of pipeline conditions is a powerful tool for the pipeline operator. The pipeline operator can use the simulation results to assist with decision making. Simulation enables the operator to access unmetered areas of the pipeline and provides accurate prediction and planning capability. Pipeline simulation provides the operators with the ability to respond quickly to ongoing changes in supply and demand. Simulation helps operators run the pipelines safely and cost effectively always.

Real-time transient analysis evaluates the physical pipeline measurements against model calculated flows and pressures. By tuning the pipeline simulation, the physical pipeline measurements and the model calculated values converge. This provides confidence within the simulated values. Comparing the pipeline capacity and throughput time values further increases this confidence. Alongside real-time modelling, predictive capabilities are a key feature of the pipeline simulation systems. They use the current state as a starting point for analysis of what will happen in the future.

The planning and predictive capabilities can also assist with the calibration and planning of emergency shut-down controls. By taking actual operational states of the pipeline, pipeline operations can be simulated to a high degree of accuracy. This is achieved by taking a pipeline state from an online real-time model and restoring the state to an offline simulation. The results of these highly accurate simulations can then be used to calibrate automatic shut-down valves. Operational scenarios such as compressor shut-downs can be simulated, and the resulting pressure wave analyzed. These operations can then be evaluated against simulated leak data. The data from this analysis is used to differentiate operational scenarios from pipeline leakages. These results can subsequently be used to identify desired calibration ranges for each automatic shut-down valve.

Case Study Pipeline Details

The physical case study pipeline is a 29 km (18 mile), 24-inch segment of a natural gas pipeline and is detailed in Figure 4. At the inlet of the pipeline lies a compressor station with a shutdown valve immediately downstream. At the 2km (1.24 mile), 16km (9.94 mile) and 29km (18 mile) points along the mainline are metered pipeline offtakes. Between the first and second offtake, located at the 11km (6.84 mile) point is an automatic shut-down valve.

Testing Procedure

The testing procedure consisted of a series of simulated pipeline ruptures and an online analysis of a compressor shut-down at the compressor station at the inlet to the pipeline for model tuning purposes.

The simulated pipeline ruptures enable the user to analyze the effects on the rest of the pipeline segment. Most importantly for this study, the user can observe the pipeline dynamics at the location of the automatic shut-down valve.

Experimental Data

High-resolution data from the physical pipeline was measured at the location of the automatic shut-down valve. This data was sampled at a rate of 0.1s. The typical pipeline measurement data received from the pipeline instrumentation for the network has a sampling rate of 10s. The objectives of the case study are to obtain a clear reference of the actual rate-of-change of the pressure at the location of the shut-down valve.

Figure 3 shows a representation of the setup and some details about the hardware used.

The hardware, shown in Figure 3, consists of:

- Differential pressure transmitter Rosemount 3051CD3 (-2.5 to 2.5 bar);
- Pressure transmitter Rosemount 3051TG4 (-1 to 276 bar);
- Controller Fisher ROC-312;
- 24 V battery (in fact a box containing two 12 V batteries in series);
- Serial crossover cable;

- Serial/USB adapter cable (they may look all the same, but some adapters tested did not work well – the model used in the tests here were the USB/serial 9037 of the Brazilian manufacturer Comtac);
- Portable micro-computer (notebook) with a Microsoft Windows 32 bits operating system.

Evaluating the Experimental Data

When evaluating the experimental data, it was difficult to discern the compressor shut-down event from the normal instrument reading noise. This can be observed within the Rate-of-Change (ROC) of pressure per second for the raw pressure data from the first compressor trip in Figure 5.

The difficulty in determining the shut-down event from the instrument reading noise was true for all trip events where data was collected at the shut-down valve location. Upon inspection of the raw data it was observed that some discretization effects were present on the data, such as adjacent samples with the same reading, some jumps with repeating patterns and high noise to signal ratio. To address the discretization effects an average filter was applied to the raw pressure and then a rate-of-change was calculated using this filtered pressure. Figure 6 shows the resulting signal. This filtering clarified the data and the moment at which the compressor trip occurred could now be identified.

Setup and Validation of the Case Study Model

A first approach was made by importing the low-resolution pipeline instrumentation data into the pipeline simulation model. This data provided the boundary conditions for the model (given that high-resolution data was only available at the automatic shutdown location). The simulation was run at 0.1 second intervals to match the high-frequency data. As the input data was given in 10 second intervals, this data was interpolated into 0.1 second inputs.

When running the simulation under these settings, the interpolation smoothed the rate-of-change. This made the simulation calculate rates-of-changes at the valve location much smaller than those measured by the high-resolution data.

The issue with the interpolation is that the nature of a rapid transient such as a compressor shut-down can occur in a much shorter time-period than the sample rate interval. Interpolating the values of the low-resolution data into smaller steps, decreases the rate-of-change, generating a much gentler rate-of-change over the 10s. This was because only data at the shutdown valve, but not at the compressor that tripped, had a 0.1 second resolution. It was therefore not known how quickly the trip had occurred; just that it took less than one 10 second scan.

Validating the Case Study Model - Method Two

A second run was then made, by discretizing the data in such a way that the pressure drop occurred over a single time step interval of 0.1s. This maximizes the rate-of-change for the low-resolution data and provides a practical upper limit to the rate-of-change that truly occurred since it is not physically plausible that the compressor spun down faster than this.

Figure 7 compares the average pressure at the compressor discharge pressure (as measured by the low-resolution pipeline measurement data) against the signal that was discretized (10s pressure change occurring over a single 0.1s step).

Figure 8 shows a chart that compares the rate-of-change measured at the valve against the rate-of-changes calculated by the model when using the averaged low-resolution pipeline measurement data against the discretized 0.1s changing pressure.

The calculated rate-of-change in the series “Averaged Model Rate-of-Change” of 0.146 kgf/cm2 (2.08 psi) was too low when compared to the actual rate-of-change measured in the field. This was found to be 0.4356 kgf/cm2 (6.19 psi) per second. By looking at the series named “Discretized Model Rate-of-Change”, the value obtained matches the measured rate-of-change at the valve of 0.4356 kgf/cm2 (6.20 psi) per second.

This was a coincidence, since the simulated pressure rate-of-change at the valve depended on the model time-step and distance-step, and there is no reason to believe that the compressor trip occurred over exactly one 0.1 second high-resolution scan interval. It clearly indicated that the rate-of-change encoded in the low-resolution measurement data is not representative of the actual pressure rate-of-change that occurred.

Validating the Case Study Model - Method Three

The study examined a third approach. The high-resolution data measurement from the shut-down valve, 11km (6.84 mile) downstream of the compressor, is likely to be a more accurate representation of the real transient that took place at the compressor discharge compared to the lower resolution pipeline measurement data at the compressor location.

This third approach involves constructing a pressure wave by applying a time offset to the high-resolution pressure measurement from the shut-down valve location. The data was also assigned a pressure offset to compensate for the pressure drop over the 11km (6.84 mile) segment that separates the shut-down valve from the compressor location. Figure 9 shows the modified discharge pressure used in the model under this approach.

Note how the absolute pressure drop as indicated by the low-resolution pipeline measurement data is larger than the one in the modified pressure data. This was acceptable for this study, since the study focuses on the rate-of-change of the pressure.

The rate-of-change obtained under this test was higher than the case where the compressor discharge pressure was averaged over the 0.1s steps, however it was not as large as the actual measured rate-of-change shown in Figure 10.

Validation Results

The results of the third method show that the pressure rate-of-change used as the model boundary condition at the compressor discharge decayed significantly on its way to the measurement point 11km (6.84 mile) downstream. This pressure wave decay was significant, and the impact of numerical dispersion and the impact of the actual decay of the wave were analyzed.

The pressure drop taken from the high-resolution data measured at the valve was used to validate the expected pressure wave decay as described by the Kirchoff equation (Ref 1).

The Kirchoff equation (Ref 1) represents the viscothermal losses at the pipe wall, while interactions of a pressure surge with bulk turbulence in a flowing gas pipeline are representations by, Peters (Ref 2) and Howe (Ref 3). These three references represent decay due to complex physical processes that are not included in the pipeline model, but the effects of which can be computed. The simulator includes three mechanisms of decay of a sharp surge: numerical dispersion (which is not physically present in the real fluid, but instead is an artifact of the simulator), friction (which we believe to be correct at long timescales, but which may or may not be correct at timescales of a second or less - at such short timescales the decay mechanisms described in References 1 - 3 may be more important). The decay resulting from the nonlinear convection term in the momentum equation is important for large pressure surges.

The simulator produced numerical dispersion based on the model order. Pipeline simulators solve sets of partial differential equations in space and time; as such, they have a mesh on which space is discretized and a mesh on which time is discretized. These are characterized by a spatial mesh step, also known as "knot spacing", and a time mesh step, also called the "time step". Generally - in the absence of stability issues - the model results will be more accurate with a smaller time step and with a smaller knot spacing. The order of the model indicates how the accuracy improves as the time step and knot spacing decrease. A model that is first-order in space will generally produce half the error if the knot spacing is cut in half, while a model that is second-order in space will produce a quarter the error if the knot spacing is cut in half. Similarly, models can be described as first-order, second-order, or some other order in time, which indicates how the error decreases as the time step is reduced.

The simulator used for this study has an automatic mechanism to maintain stability by picking an appropriate knot spacing for a given time step. No stability problems were observed in the model runs reported here with the second-order model. There were some stability problems observed when using the lower-order models which are discussed in the next section.

Different mechanisms of decay can be characterized by the distance over which they cause reduction in the pressure rate-of-change of a pressure wave front by a factor of two. For this system, viscothermal losses at the pipe wall would only cause this every 40 km per the equations in (Ref 1). The formulations of Peters (Ref 2) and Howe (Ref 3) of the decay caused by the interaction of a pressure wave pulse with the bulk turbulence predicted a factor-of-two rate-of-change decay over distances of 10 km and 11 km, respectively.

It was observed that with the friction term and the nonlinear term in the momentum equation set to zero, and with a sufficiently small time and distance step, and running isothermally, the simulator produced no noticeable numerical dispersion of the pressure wave surge even over 40 km (an isothermal model was used because over short timescales the pipe wall and the earth immediately surrounding the pipe act as a heat buffer, eliminating any short-duration temperature changes in the gas). Both the simulator's friction term - which is based on the Colebrook-White equation - and the convection term in the momentum equation produced dispersion that resulted in considerably shorter characteristic decay distances than 10 km.

MODEL COMPRESSOR TRIPS COMPARISON FIRST AND SECOND ORDER

To evaluate how the different modelling options, behave, different pressure drops were simulated at the compressor location and the resulting modeled rate-of-change at the shutdown valve was recorded.

The following table shows the results for three different tests under the different modelling scenarios.

Stimulus Rate of change (psi/s)	Second order space, second order time (psi/s)	Second order space, first order time (psi/s)	First order space, second order time (psi/s)	First order space, first order time (psi/s)
323	6.21	1.96	2.22	0.78
85	5.51	2.44	2.30	0.94
64	3.12	1.38	1.36	0.49

It is interesting to note that the first order spatial model ran into instabilities when decreasing the time step below 1s, all results are hence for 1s time steps.

The use of a first order over a second order model while keeping the second order time scheme results in a calculation of a pressure drop rate approximately 39% lower for the 11km (6.84 mile) between the compressor location and the shutdown valve location.

As expected the biggest factor in the accuracy of the calculation of the pressure drop rate seems to be tied to the combination of both second order approaches, just relying on second order spatial or second order time collocation scheme does not generate the right results.

COMPRESSOR TRIPS COMPARISON AGAINST SIMULATED LEAKS

It is of interest to contrast the rate-of-pressure drop caused by the compressor trip at the valve location against that of a moderately large leak. For this test, a simulated leak with a diameter of 0.1 m (0.328 ft) was created at the compressor location and the pressure was sampled at every 2 km (1.243 mile) down the line. The simulated leak caused a leak flow of 2.1 times the typical flow in the section. By looking specifically at the automatic shutdown valve location, the high-resolution measurements indicate that the largest rate-of-change in the pressure caused by a compressor trip was of 0.4356 kgf/cm2 (6.19 psi) per second, while the test leak caused a rate-of-change of 1.2847 kgf/sm2 (18.27 psi) per second, which is considerably higher. The results as a function of the distance are depicted in figure 12.

The first 2 km (1.243 mile) show the sharpest decay of the pressure signal, at the leak point the rate-of-change was of 11.56 kgf/cm2 (164.42 psi) per second, just 2 km (1.243 mile) downstream the rate-of-change had already diminished to 2.87 kgf/cm2 (40.82) per second. By looking at the rightmost limit of the rate-of-change vs. distance chart, it can be concluded that a significant leak such as this at the point 30 km (18.64) downstream will closely resemble what a compressor trip might look like at a valve located at 11km (6.84 mile).

CONCLUSIONS

Optimizing an automatic shutdown system requires a careful study of leak pressure waves as they propagate along the pipeline. The decay of the pressure rate-of-change with the distance is significant. This can result in systems that are only sensitive to very large leaks due to the long distances between automatic shutdown valves.

When a sudden change in pressure caused by a compressor trip is used as input to the simulation model, the predicted wave decay downstream is much smaller for a second order spatial model over a first order one. The second order model showed better agreement with the measured data and the theory on wave decay.

Because the rate-of-change decay of a small (compared to the line pressure), rapid (several seconds or less) pressure surge may be dependent on complex physical mechanisms not included in the simulator used here, and not just on the application of the one-dimensional-plus-time momentum equation to the surge boundary condition, it is recommended to observe the propagation of such a surge with high time resolution as a validation check of the model. Large-amplitude surges like big leaks or ruptures do not require this because decay due to the convection term in the momentum equation for such surges is faster than either the decay from the Colebrook-White friction term in the simulator's momentum equation, or the frictional decay mechanisms that act on low-amplitude fast pressure wave pulses as discussed above. Therefore, those frictional mechanisms are relatively unimportant in the case of a rupture, and it is unnecessary to get them exactly right.

The second-order-space second-order-time model was much more stable than the first-order-space model and gave much less numerical dispersion than the first-order-time model, providing the necessary accuracy for this study. It would have been difficult to get any useful results for the propagation of the surge caused by a compressor trip with the lower-order models.

It was found that the behavior of the pressure wave decay over a gas system is a complex phenomenon that is driven by at least two main mechanics. The frictional effects are dominant when the pressure perturbations are smaller such as during a compressor trip.

As the size of the pressure surge grows and approaches the same magnitude as the pipeline pressure, the effects based on the convection term in the momentum equation dominate. The second-order box-scheme model used here is considered accurate for calculating decay of rate-of-change in the pressure front for these larger surges. This method appears to overestimate the decay rate of very small surges and may overestimate the decay rate of intermediate surges like compressor trips. These errors are due to the one-dimensional friction factor that has served the industry so well in modeling steady-state frictional losses and typical pipeline transients with timescales of minutes to hours may not be an accurate representation of the frictional decay of sharp fronts at smaller timescales.

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AUTHOR BIOGRAPHY

Garry Hanmer is a Product Owner of ATMOS International in Manchester, United Kingdom. Garry Hanmer has over 12 years' experience in the pipeline industry with an emphasis on pipeline hydraulic simulation. He also has experience in development and delivery of pipeline operations and integrity management software systems. Garry Hanmer has a Master of Engineering in Aeronautical Engineering (MEng Hons) from the University of Salford, UK.

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FIGURES

Figure 1. Aerial Part of a Typical SDV

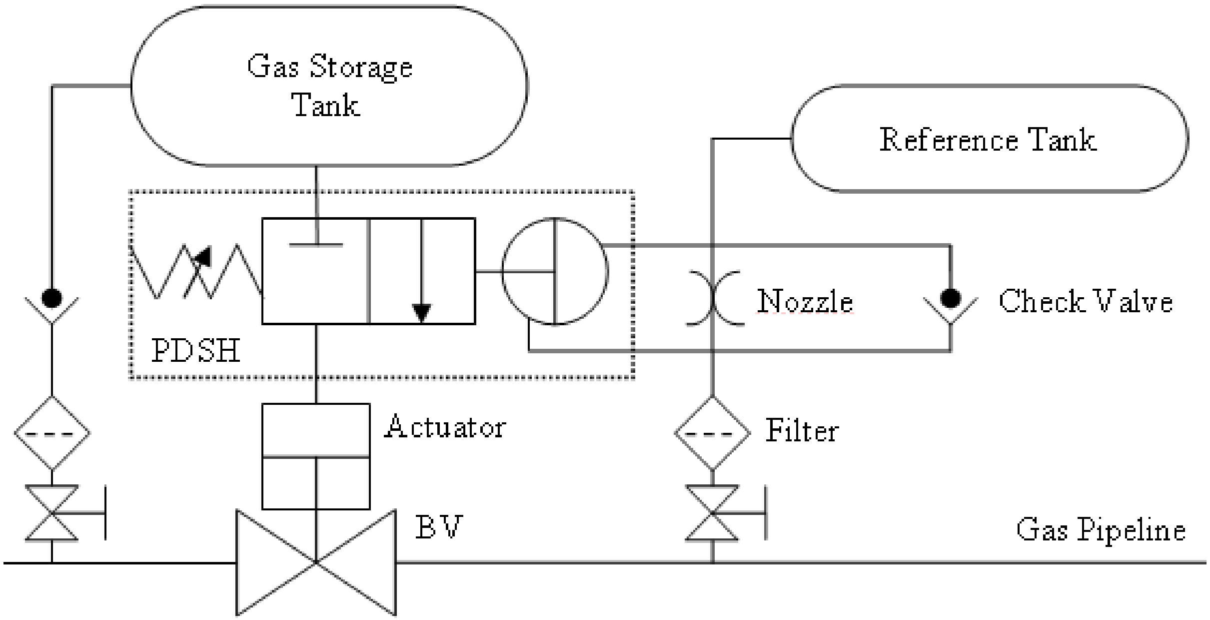

Figure 2. Simplified Diagram of a Pneumatic Line Break Control Module.

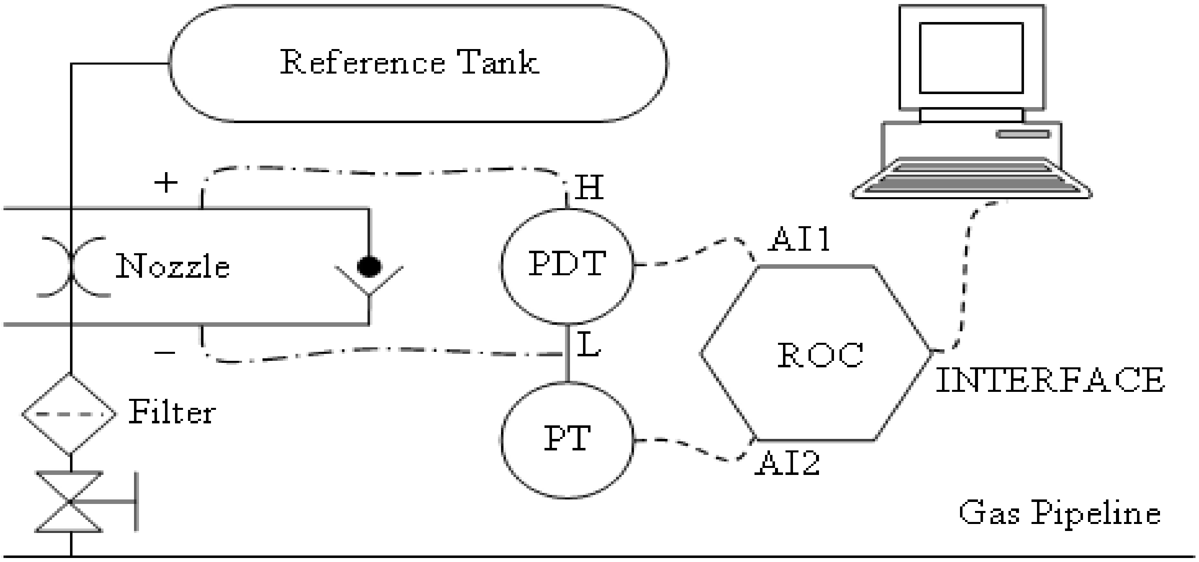

Figure 3. Instrumentation and Connections Diagram

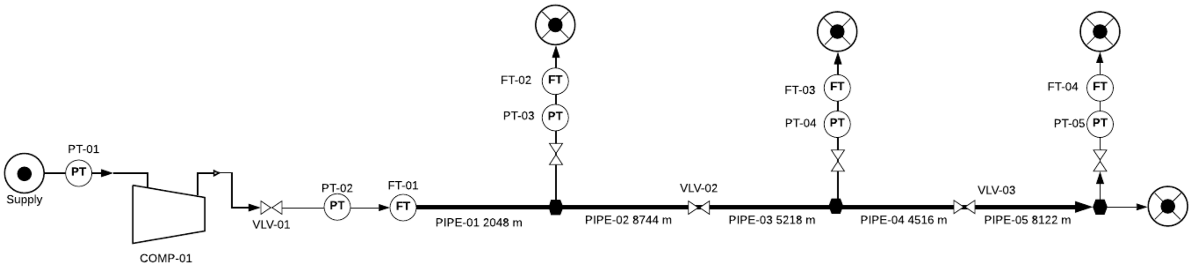

Figure 4. Pipeline Region for Study

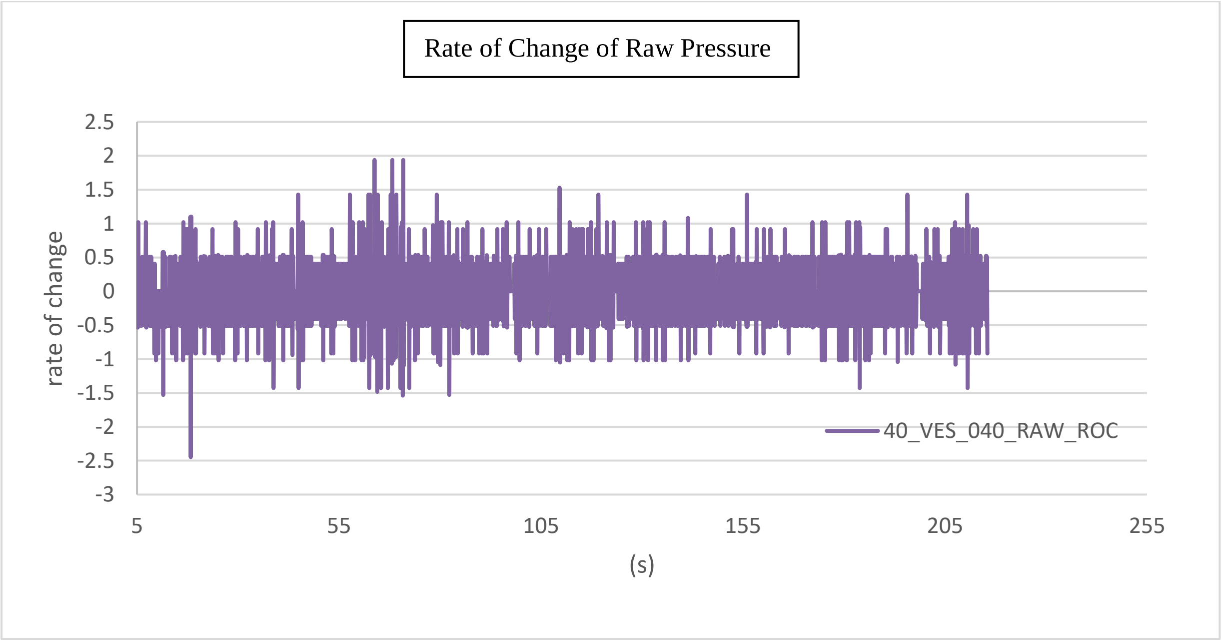

Figure 5. Raw Rate of Change Data

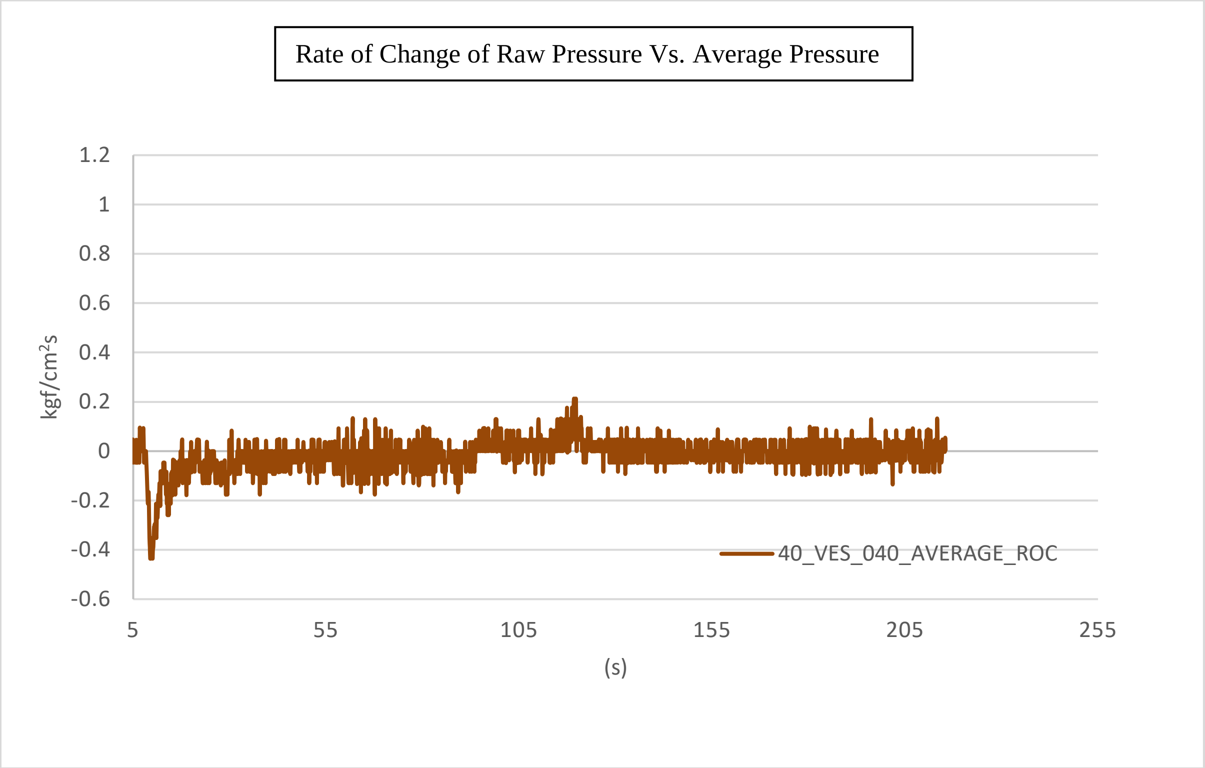

Figure 6. Filtered Rate of Change Data

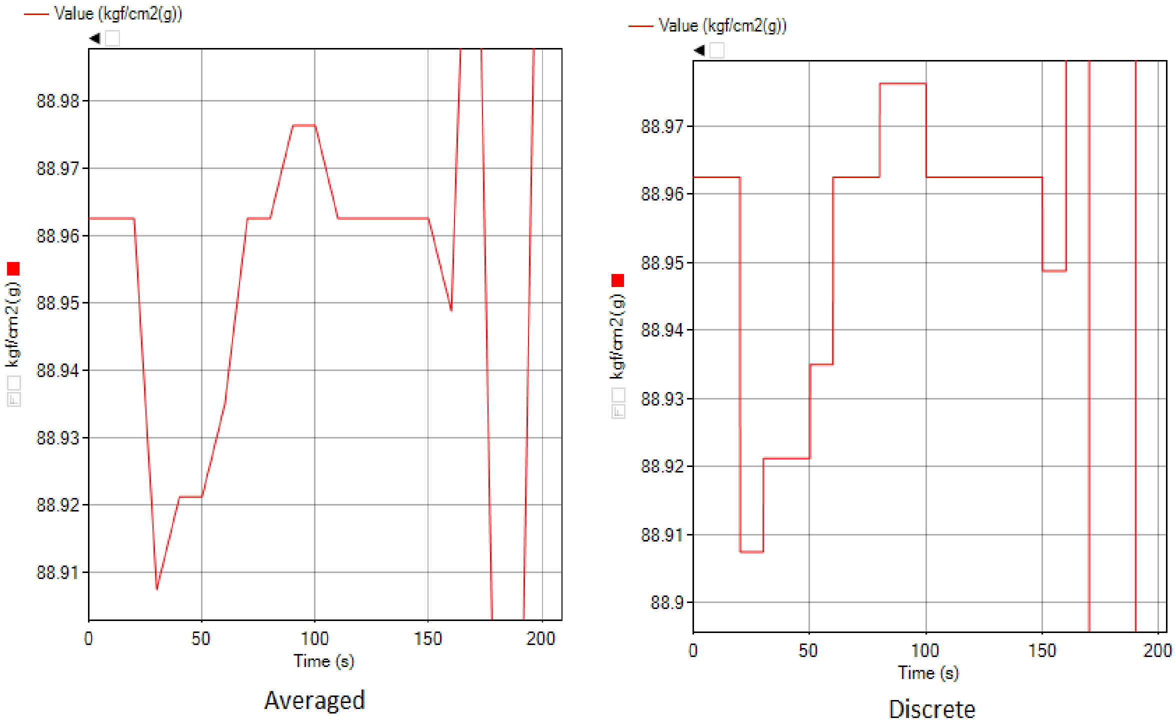

Figure 7. Average compressor discharge pressure

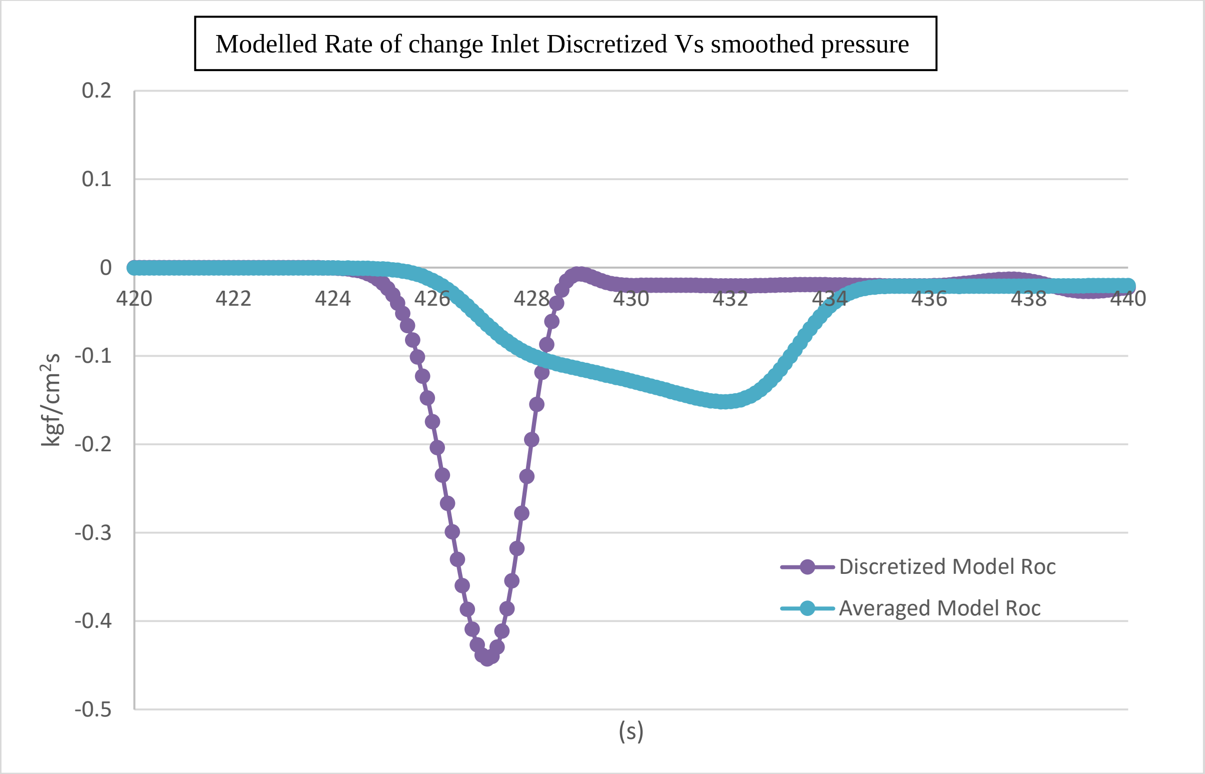

Figure 8. Modelled Rate of Change

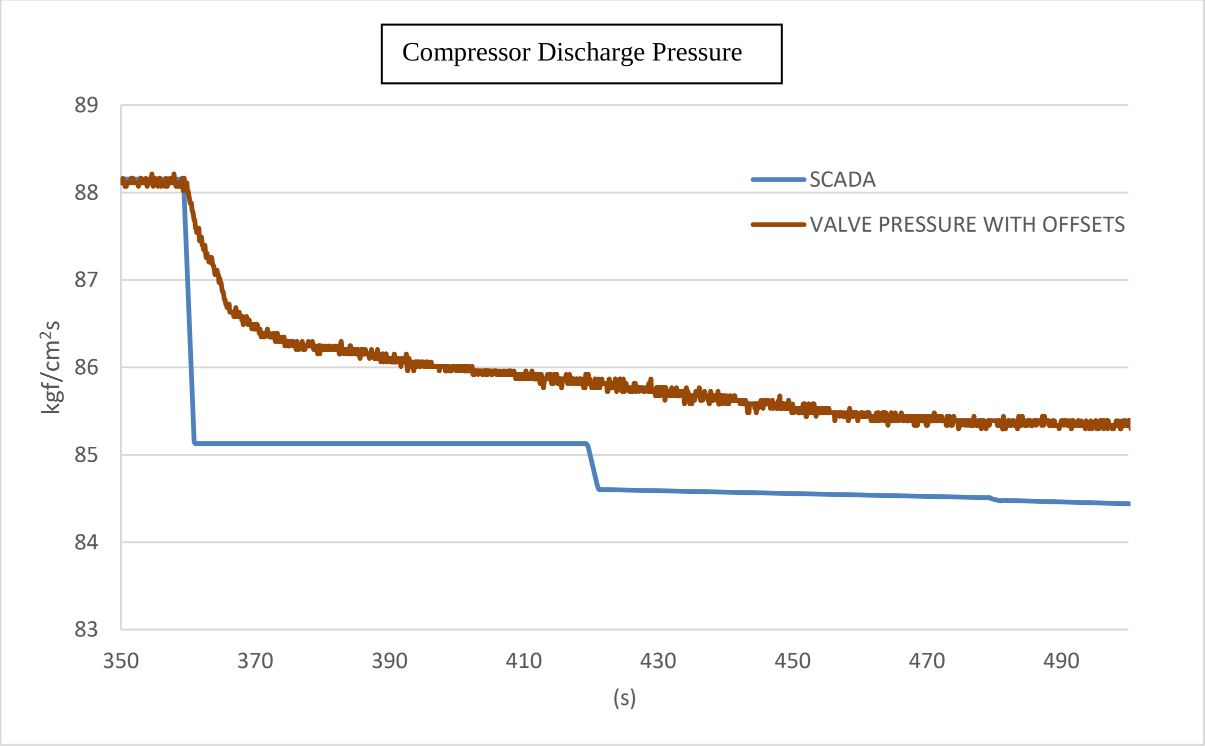

Figure 9. Compressor Discharge Pressure

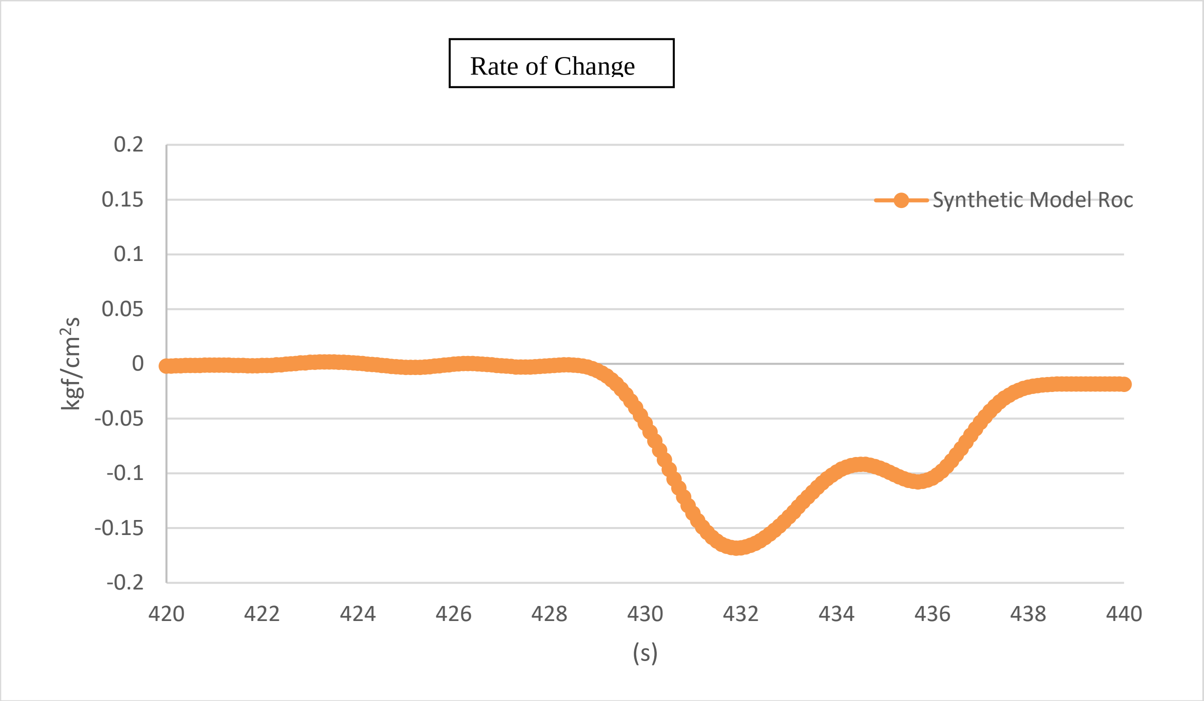

Figure 10. Modelled Rate of Change Synthetic Inlet Pressure Based on Valve Pressure Signal

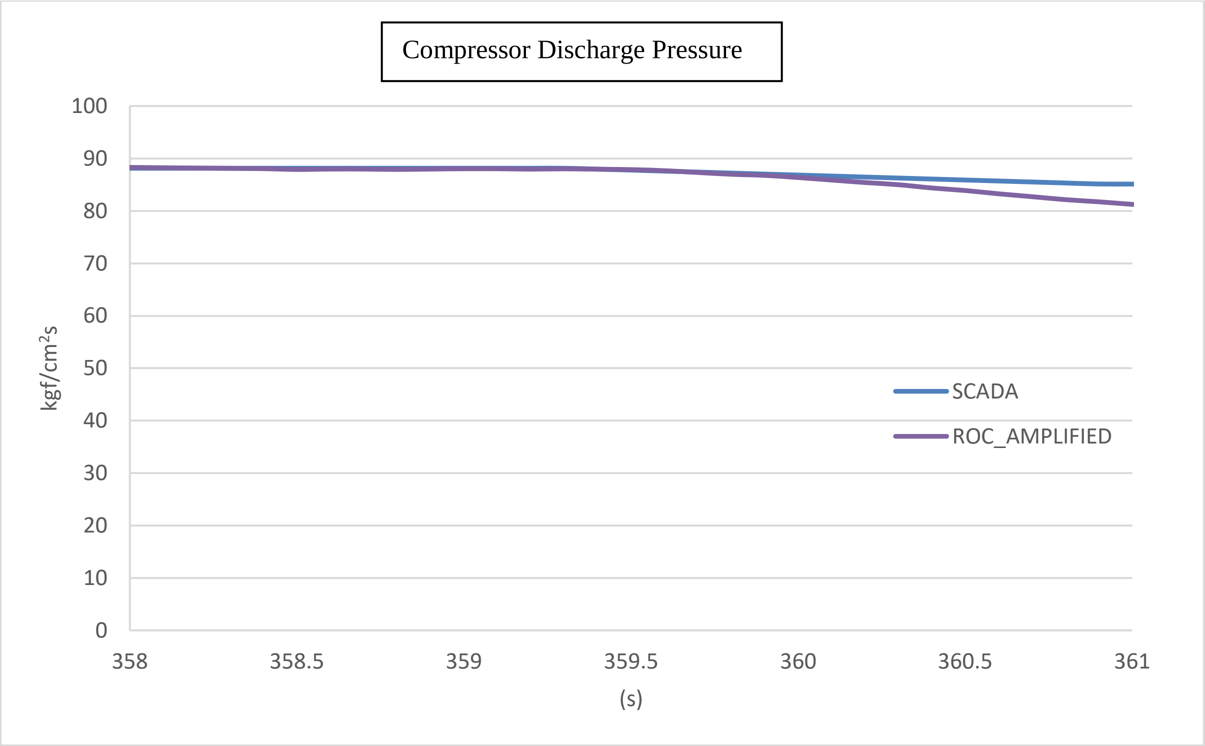

Figure 11. Compressor Discharge Pressure

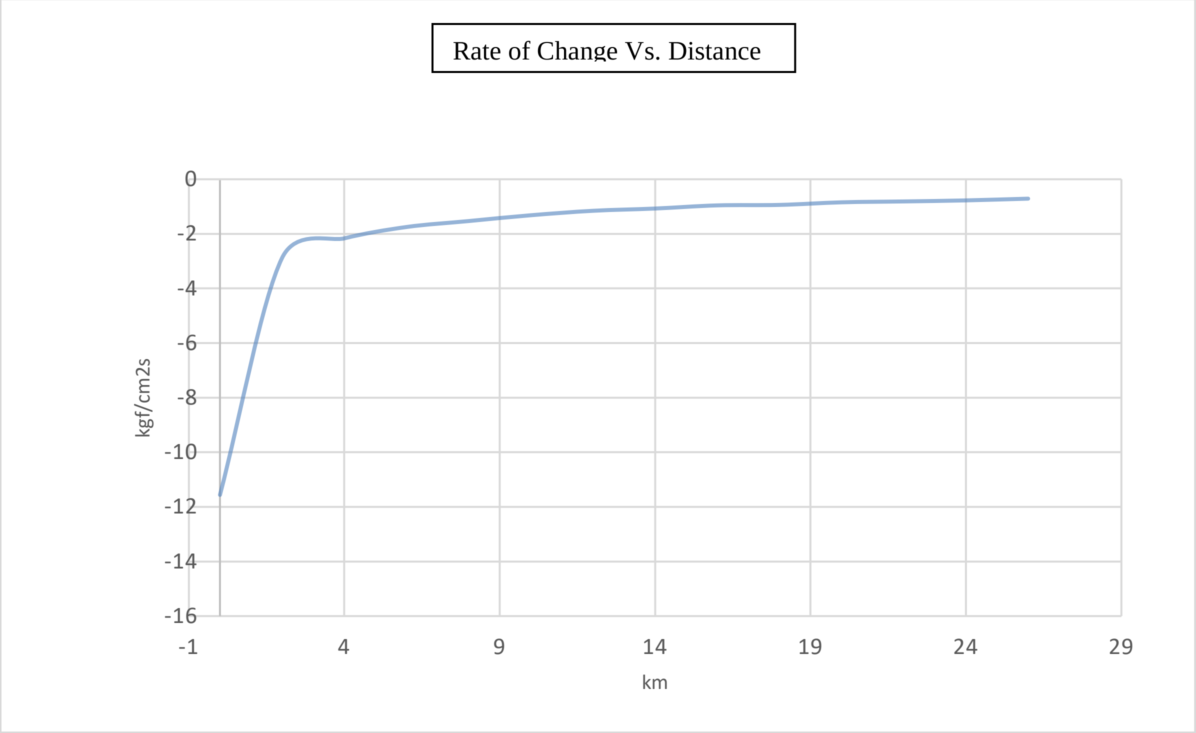

Figure 12. Rate of Change Vs. Distance for First Run Results