Modelling and Dimensional Analysis of Spring-Loaded Pressure Regulator for Performance Prediction and Characterization

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Abstract

The successful performance of a typical launch vehicle depends on fault free and reliable performance of different propulsion stages which uses various fluid control systems to control flow, pressure and direction of propellant and pressurant. Liquid propulsion stages require almost constant propellant tank pressure to maintain desired propellant flow into engines and to provide a positive pump suction head to avoid cavitation. In addition, these stages require constant pressure command gas source to operate engine propellant flow control valves, gas generator valves, purge valves etc. Pressure regulator is a variable-area pressure reduction valve, which provides constant outlet pressure at its downstream, irrespective of variations in inlet pressure and flow demand. In active Pressurisation systems, the design of pressure regulator is complex due to the large band of inlet pressure variation at high flow rate demand. A non-linear dynamic mathematical model of pressure regulator is required for predicting the performance in the specified working conditions with respect to initial transients, temperature effects, variation in input/ output conditions, inherent oscillations etc. The model is also effective to carry out post-flight analysis by simulating anomalies seen during ground tests and during flight. The upper stage of PSLV uses liquid pressure-fed engines which has two-stage regulated pressurisation system to maintain constant tank pressure with regulators in series. First rough regulator regulates high pressure GHe stored in gas bottle to intermediate pressure and then precision second regulator regulates to final required tank pressure. A non-linear and 1D time-domain mathematical model of high-pressure regulator simulating transients and working range was made using AMESim software and validated with experiment results. One of the anomalous observations in regulator during ground testing was studied with this model. Later these observations were confirmed by NDT techniques on the hardware. Paper presents the details of the model and parametric study conducted on the regulator. The validation of model with test results and characterization of the model to recreate the anomaly is also presented.

Keywords: *Pressure Regulator, AMESim Model, Performance analysis*

1. Introduction

Liquid propulsion stages require active control of propellant tank internal pressure. To reduce weight, propellants are stored in thin-walled tanks at low pressure and then injected into the engine at high pressure using turbo pumps. To ensure proper pump performance and avoid cavitation, the pump inlet pressure should be kept above the NPSP. Also, the tank internal pressure should be kept sufficiently higher than the atmospheric pressure to prevent structural failure by buckling due to low pressure difference. Tank pressures are controlled by admitting propellant compatible gas into the ullage volume to volumetrically replace the propellants consumed by the engine. Helium gas is typically used because of its inert nature and compatibility with propellants, low molecular weight which provides high pressure per unit mass and volume. Initially the tanks will be pressurized at ground through a pre-pressurization circuit for engine start-up. The on-board pressurization system takes over for the flight propulsive phase. In a typical Pressurisation system, the pressurant are stored in gas bottles at very high pressure. Isolation valve is used to isolate the high-pressure gas from pressure regulator and downstream elements. Once the isolation valve is opened, pressure is

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communicated to regulator which regulates it to required outlet pressure. This regulator can be in single stage or multi-stage with two or more pressure regulators in series. Figure-1 shows the typical configuration of a single stage pressure regulation system.

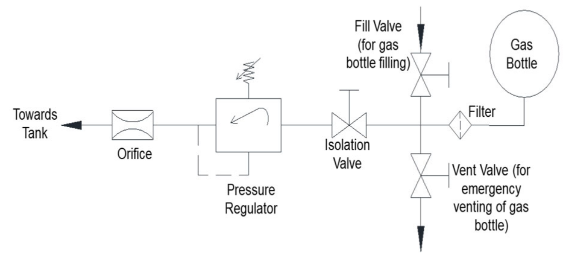

Figure 1: Typical Single-stage Pressure Regulation System

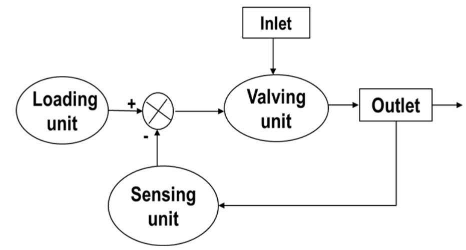

Figure 2: Control System Diagram of a Pressure Regulator

A pressure regulator is a control valve which maintains the pressure to a required level by varying flow across the valve as a function of difference in actual & desired outlet pressure. This is a self-contained close loop control system which senses any difference in outlet pressure w.r.t required outlet pressure and vary the flow area across valve with help of Valving element movement. Control system diagram of a pressure regulator is shown in Figure-2. Pressure regulator is a common device used in many industrial applications, domestic usage and widely used in aerospace application too. A pressure regulator is a self-actuated device which does not need any additional power source to operate which makes it a preferable candidate for pressure controlling in launch vehicle application. Pressure regulator can be broadly classified as modulating pressure regulator and non-modulating pressure regulator [1]. A modulating pressure regulator maintains required regulator outlet pressure at a chosen set point. However, non-modulating pressure regulator maintains regulator outlet pressure between upper and lower limit of required pressure.

Pressuiration system selected for study is a two-stage regulated ambient gas pressurization system which regulates high pressure Helium gas from 330 bar to 35 bar in first stage by primary pressure regulator (PPR) and then finally to 19 bar by secondary pressure regulator (SPR). PPR of first stage regulation is considered for study which a direct acting, Belleville disc loaded pressure regulator. A one-dimensional time domain mathematical model of PPR is developed using AMESim to study the complete behavior of pressure regulator and validated with experimental results. Later, abnormal performance seen during lab test was simulated by model characterization and design of regulator was modified based on model results.

2. Mathematical Modelling of Spring Loaded Pressure Regulator

2.1 Configuration of pressure regulator

Primary Pressure regulator (PPR) of 2-stage pressure regulation system of upper stage of PSLV is selected. Figure 3 shows the schematic view of PPR. This is a direct acting, Belleville spring loaded, balanced type, normally open regulator. Regulator has conical hard on soft seat configuration. Piston with 'O' ring seals is used as a sensing element. Downstream pressure is maintained constant by regulating the flow across the seat. Belleville spring load will be adjusted to achieve required outlet pressure. Regulator has ambient pressure communication on the loading side which makes a reduction 1 bar in set pressure from ground to flight.

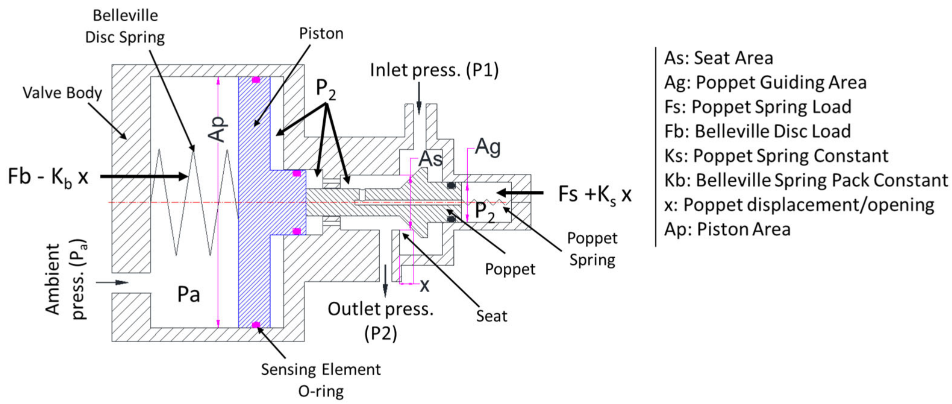

Figure 3: Free-body diagram of Primary Pressure Regulator

Following is the force balance equation of pressure regulator at a particular opening (x):

$$P_1 \times (As - Ag) + Fs + Ks \times x - Fb + Kb \times \mathrm{x} + P_2 \times Ap - P_2 \times (As - Ag) - Pa \times Ap = 0 \quad (1)$$

This regulation system has Helium gas stored at high pressure in upstream of PPR and isolated with isolation valve. In downstream of PPR, another fine secondary pressure regulator (SPR) is mounted followed by solenoid valve (SV) for isolation in downstream which allows the flow through regulator when opened. Initially SV is closed and when isolation valve is opened, the high-pressure helium is communicated to downstream regulators and regulator goes in lock-up mode providing required outlet pressure at downstream. However, there is no flow across regulators since the downstream SV is closed. Once the SV is opened, flow stars and regulator comes into function providing required pre-set outlet pressure.

2.2 MATHEMATICAL MODEL

In a pressurisation system, it is of great importance to predict the behavior of pressure regulator during transient and steady state. In addition to basic governing equations for flow and force balance, the dynamic characteristic of a pressure regulator is highly dependent on various geometrical and physical parameters like static and dynamic friction in sliding pairs, clearances and guidance of various critical moving pairs, lubrication stribeck effects, mass inertia of moving elements, Valving element geometry, coulomb & pneumatic damping, spring stiffness, flow losses, physical properties of working medium, effects of temperature variation etc. This makes the conventional mathematical model complex and difficult to solve by conventional governing differential equations and then solving them by either numerical techniques or conventional software codes. Hence, a one-dimensional time domain mathematical model of PPR is developed using *"Advanced Modelling Environment for performing Simulation for Engineering systems (AMESim)"* to study the complete behavior of pressure regulator. The ability, accuracy, pictorial representation, less post-processing time of modeling tool software's for prediction of system dynamics and its behavior at various operating conditions makes it more popular and convenient to use.

LMS Imagine AMESim® helps in 1D lumped parameter time domain analysis of hydraulic and pneumatic systems. The software interfaces uses symbols to represent individual components in a easily recognizable pictorial form. Each component has its own governing equations to simulate the physical phenomena related to that component in form of algebraic and partial differential equations. It converts a given set of partial differential

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equations to a system of either ordinary differential equations or algebraic equations and solves them using numerical method such as explicit Runge-Kutta or linear multi step method like adams bashforth or adams moulton method.

Pressure regulator is divided in various sub-system chambers from free body diagram is made and suitable algorithms are selected from software [4]. Figure-4 shows the equivalent mathematical model of PPR along with various upstream and downstream components used during testing. In PPR model, moving element's inertia is simulated by mass simulator module with maximum movement of poppet restricted to its maximum travel possible in actual hardware. A standard O-ring model is used for simulating O-ring static and sliding friction providing the actual geometrical parameters for O-ring and corresponding groove [3]. Spring stiffness, piston, downstream orifice etc. is simulated with respective sub-models. Since the expansion of high pressure gas in upstream gas bottle depends on heat exchange across gas bottle which needs further detailed thermal model of gas bottle, direct pressure & temperature measurement from the flow test is given as input to model to validate it with experimental results. A standard ideal pressure regulator for simulating SPR and orifices & flow lines for simulating downstream components which controls the flow demand are selected.

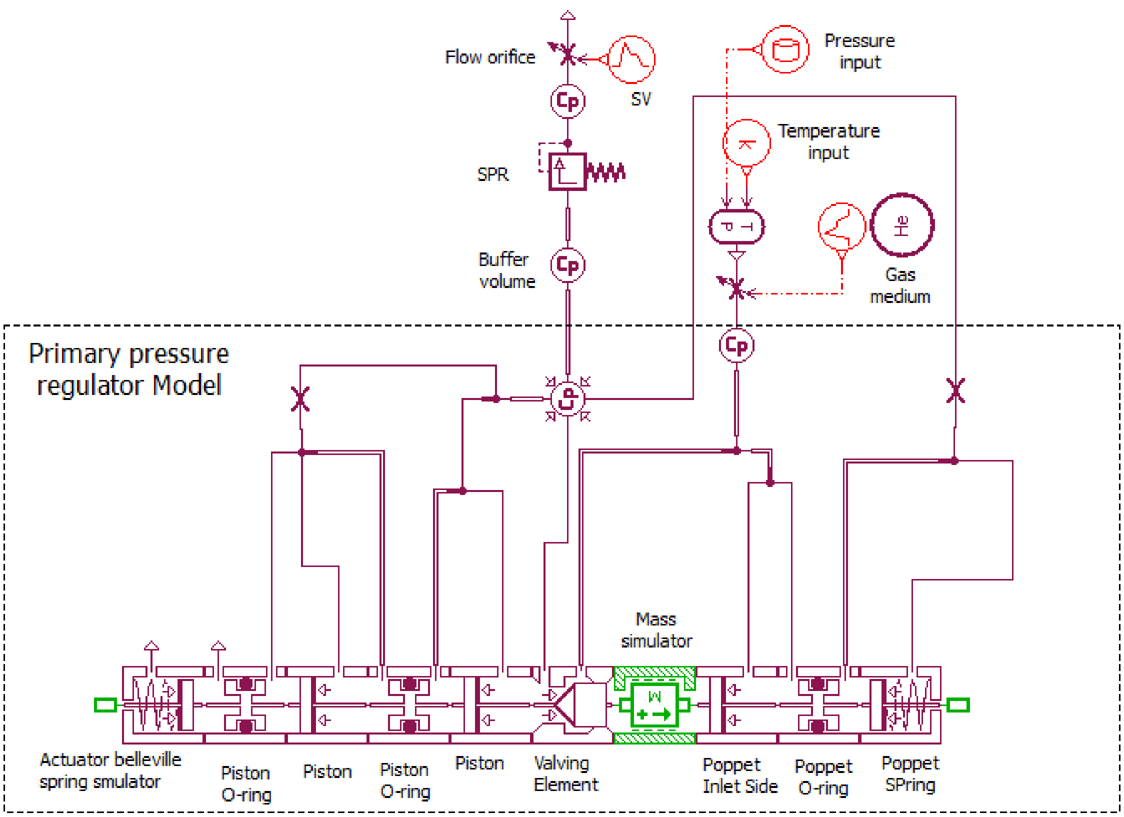

Figure 4: Mathematical Model of Primary Pressure Regulator

For simulating working medium, a real gas sub-model using Vander wall equation of state is selected as follows:

$$\left(P + \frac{a}{V^2}\right) \cdot (V - b) - RT = 0 \qquad (2)$$

where, $a = \frac{27.R^2.T_c^2}{64.P_c}$ and $b = \frac{R.T_c}{8.P_c}$; P: Pressure; V: Volume; T: Temperature; R: Characteristic gas constant; Tc: Critical Temperature; Pc: Critical Pressure

The temperature and pressure time-derivatives are evaluated resolving the following system of two equations traducing the mass balance and enthalpy balance:

$$\begin{bmatrix} V.\left(\frac{\partial\rho}{\partial p}\right)_T & V.\left(\frac{\partial\rho}{\partial T}\right)_p \\ m.\left(\frac{\partial h}{\partial p}\right)_T - V & m.\left(\frac{\partial h}{\partial T}\right)_p \end{bmatrix} \cdot \begin{bmatrix} \frac{dp}{dt} \\ \frac{dT}{dt} \end{bmatrix} = \begin{bmatrix} \sum \frac{dm_i}{dt} - \rho.\frac{dV}{dt} \\ \sum \frac{dm_i}{dt}.h_i - h.\sum \frac{dm_i}{dt} + \delta Q \end{bmatrix} \quad (3)$$

where density (ρ) and enthalpy (h) partial derivatives depend on the equation of state and m is mass.

These equations are now solved using numerical techniques. The heat exchange is computed using the following relation:

$$\dot{Q} = k_{th} \cdot S \cdot (T_{ext} - T_w) \quad (4)$$

Where k_{th} is the thermal exchange coefficient, S is the thermal exchange area, T_{ext} is the external temperature and T_w the working temperature.

2.3 Model validation with experiments

To validate the model, lab test was conducted on same hardware and model results are compared with lab test results. Helium is stored in 0.07m^3 gas bottle at 330 bar and fed to PPR. The flow rate in regulator downstream is controlled by ø3.2mm orifice. Comparison of simulation results with actual hardware test results shows a close match during the transients and in steady phase.

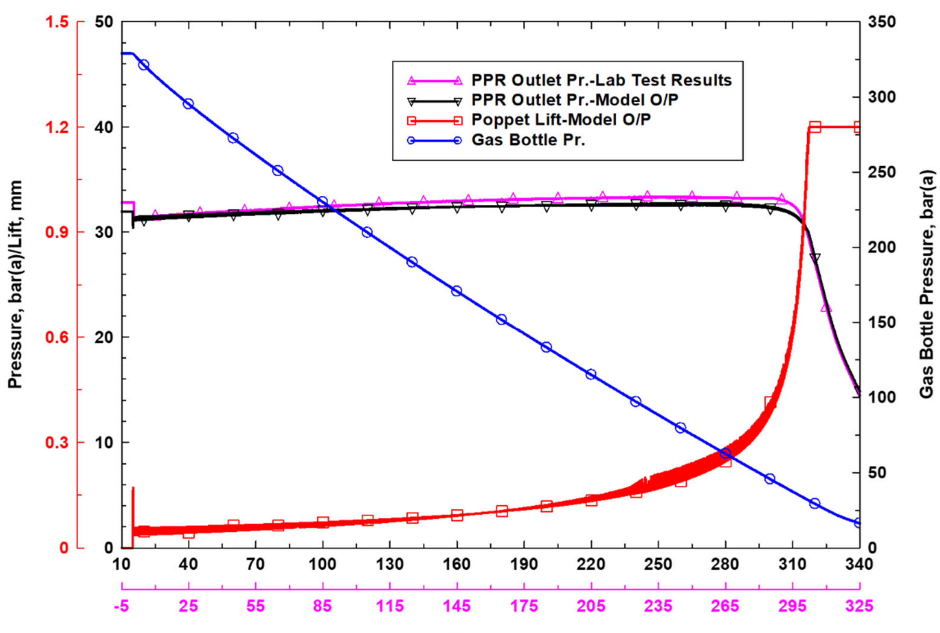

Figure 5: Comparison of Mathematical Model and Experiment Results

3. Performance Prediction and Characterization

3.1 Performance observation in lab test

During lab flow test, pressure spikes of around 0.3-0.4 bar was seen in pressure regulator outlet pressure during flow test of PPR. Subsequently in another hardware also some pressure spikes and roughness were seen during flow test. Figure-6.1 & 6.2 shows the anomalous observation seen during flow test.

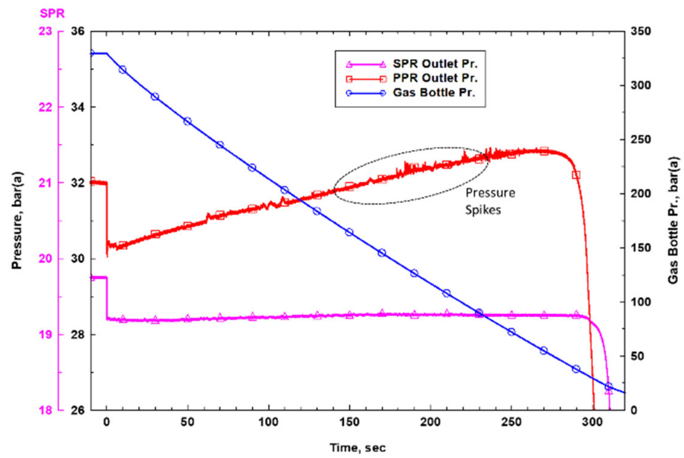

Figure 6.1: Anomalous observation seen in PPR outlet pressure during GHe lab flow test

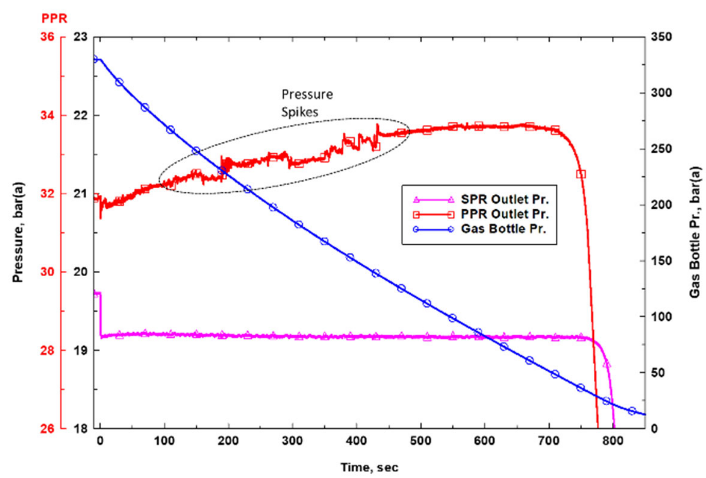

Figure 6.2: Anomalous observation seen in PPR outlet pressure during GN2 lab flow test

3.2 Anomaly simulation

To find out the exact reason of observation a parametric analysis of mathematical model was carried out to see the effect of various parameters on regulator outlet pressure. It was found that the observation seen in regulator outlet pressure is because of intermittent additional friction in sliding elements during motion which leads to pressure spikes when relieved. In mathematical model of regulator, additional friction force was introduced during functioning and after some seconds it was relieved. Model results shows that there is a dip in outlet pressure when there is additional friction force comes in sliding elements and relieved in the form of kink when it is removed. Figure 7 shows the mathematical model results for anomaly simulation. O-ring design, clearances between sliding pairs, surface finish condition of all sliding elements, geometrical parameters to ensure alignment, guidance of moving elements etc. were rechecked and it was understood that there is a design marginality in spring guidance.

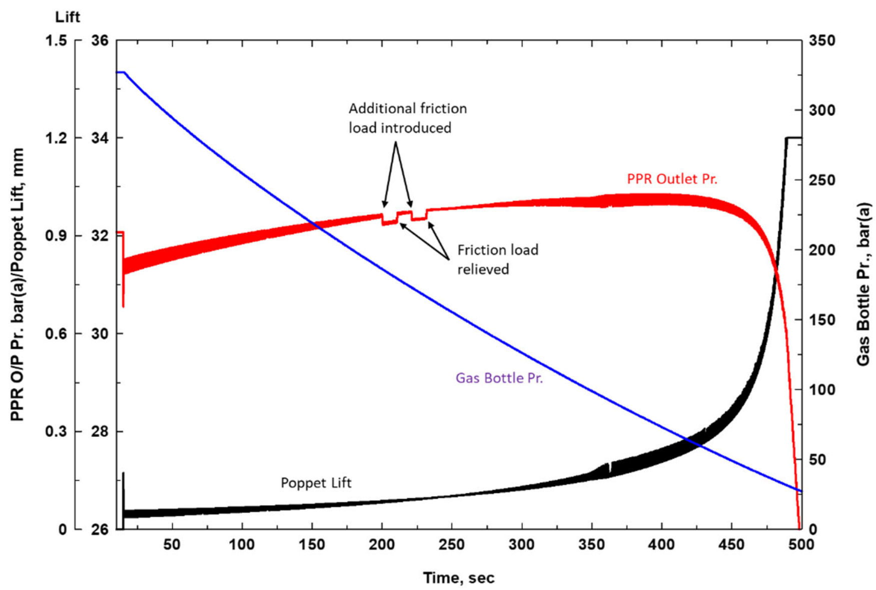

Figure 7: Mathematical Model Simulation results for abnormal lab test observation

To confirm the model results and dimensional analysis results, CT scan of hardware in assembled condition was carried out (results are shown in figure 8) and it was found that spring OD is touching the spring room ID creating the movement restriction and leading to non-smooth movement of sliding elements. Later, this was confirmed when the hardware was dis-assembled and scratches marks were found in spring room ID which was created by spring rubbing during regulator functioning.

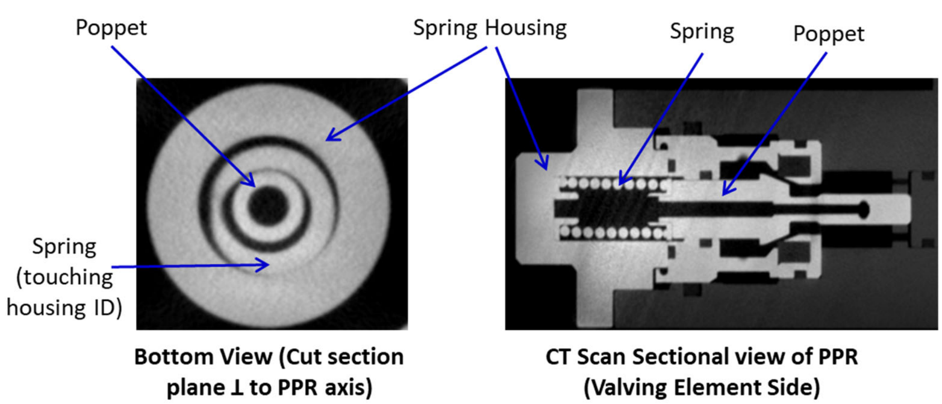

Figure 8: CT scan of PPR in assembled condition

After detailed dimensional analysis, spring was properly guided in ID for a distance of 1.5d-2d mm length at each end with help of spring retainers. It was ensured that with extreme variation in parts dimension and even with lateral shift of spring to extreme position, spring

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will not touch spring room ID and hence there is no chance of spring rubbing and restricting smooth movement of sliding elements. The regulator design was modified accordingly and hardware was realised with new dimensions. The flow test results of modified regulator were found to be satisfactory. The hardware was subjected to vibration test also followed by flow test to confirm the ruggedness of design. To ensure the consistency in regulator performance hardware's were realised with min/max tolerance band and flow tests were carried out. Performance of all the hardware was found to be satisfactory during flow tests as shown in figure 9.1 & 9.2.

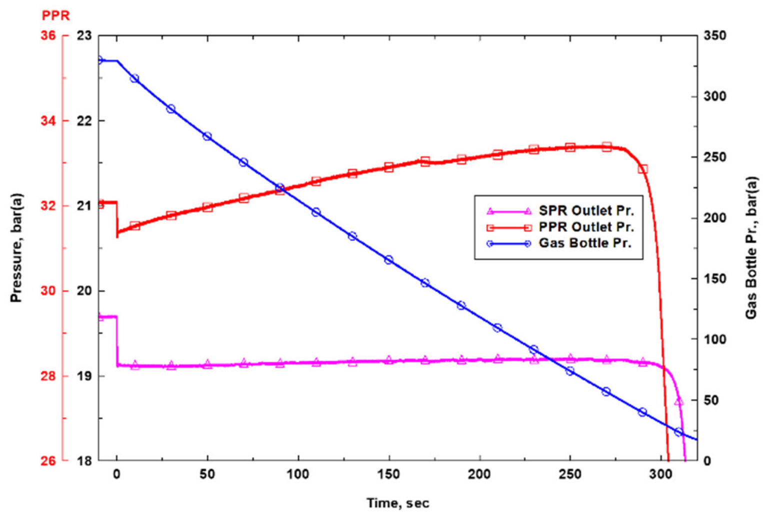

Figure 9.1: Performance of PPR during GHe lab flow test after Design Modification

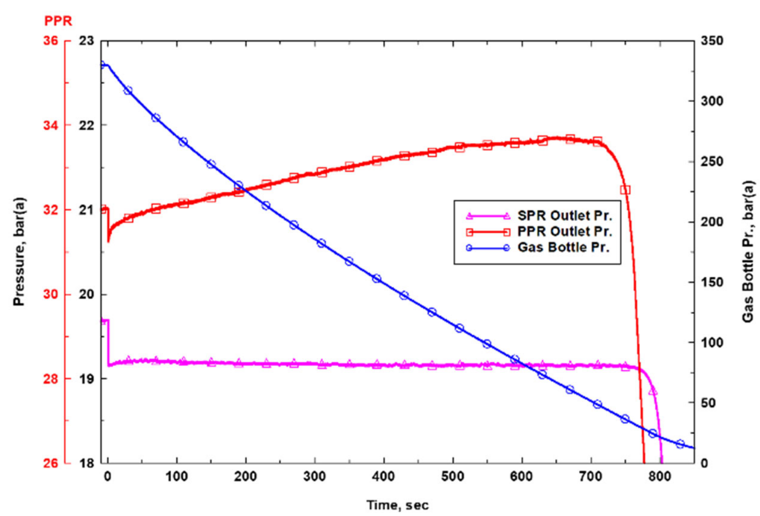

Figure 9.2: Performance of PPR during GN2 lab flow test after Design Modification

4. Conclusion

This paper presents the development of dynamic mathematical model for a spring-loaded pressure regulator using LMS AMESim and comparison of lab level performance with simulation test results. Comparison of simulation results with actual hardware test results shows a close match during the transients and in steady phase. This model is considerably used for performance prediction of pressure regulator during lab test and during flight.

From study, it is concluded that additional momentary friction during the regulator movement can cause sudden dips or roughness in regulator outlet pressure which once relieved can manifest in kink. It is proven with modelling and simulation tests. The major reason for the intermittent friction is found to be design marginality in spring guidance, which was found out by dimensional analysis and later confirmed by NDT techniques. Based on model results and analysis, design of regulator was modified and successfully tested at lab.

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