<u>KATECH & PERC Research Program Performance Report</u>

Reporting Period: Sept 13th, 2021 – August 31st, 2023

PROJECT TITLE

Assessment of Liquid
Direct Injected Fuel Systems for Propane Engines

DOCKET NUMBER

23027

PROJECT PERIOD

September 2021 – August 2023

RECEPIENT ORGANIZATION

Katech Engineering LLC.
24324 Sorrentino Ct., Clinton Township, MI 48035
https://katechengines.com/
DUNS: 117558675

PRINCIPAL INVESTIGATOR

Mr. Eric Suits
Lead R&D / Motorsport Engineer
Katech Engineering LLC.
C: (616) 834-8228 O:(586) 791-4120
eric@katechengines.com

SUPPORTING INVESTIGATORS

Mr. Mehar Babe (Katech Engineering LLC)
Mr. Derek Peria (Katech Engineering LLC)
Mr. Srinu Gunturu (Stanadyne Operationg Company LLC)
Mr. Jacob Irwin (Katech Engineering LLC)

SUBMISSION DATE

August 31st, 2023

Executive Summary

Since September 2021, Katech worked on the feasibility of Liquid Propane Direct Injection as a viable option for medium-duty (class 3 to 7) automotive engines. To meet this objective, Katech initially developed a robust fuel system with a low-side pump, a high-side pump, and injectors using off-the-shelf components. To prevent vapor lock barriers that are specific to propane, Katech added vapor lock inhibitor technologies both at hardware and software levels. The proposed fuel system schematic with vapor lock inhibitor components leveraged the use of Katech's extensive prior works, and several unique and innovative fuel-related technologies developed for both part-load and full-load operating conditions. As evident from the testing results, these technologies enabled stable operation of the engine even during hot-start, cold-start, hot-soak, and hot-idle conditions.

The proposed DI fuel system was tested in two phases: the non-firing phase and the firing-phase in accordance with the SAE standards. In the non-firing phase, the proposed fuel system was tested on a full engine rotating assembly setup without operating the vapor lock inhibitor hardware. A total of three fuel pumps (Stanadyne, Bosch, and Stanadyne Development) and three fuel injectors (Delphi, Bosch, and Stanadyne) were tested individually in terms of flow/pressure capabilities, volumetric efficiency, minimum and maximum fuel injection rate, etc. The non-firing testing without the vapor-lock inhibitor hardware proved the steady operation of the proposed fuel system at medium engine loads. However, at low and high loads, due to excessive localized heat and cavitation, the flow/pressure capabilities were unsteady. This problem of unsteady operation was eliminated in the firing phase with the addition of vapor lock inhibitor hardware and software. In addition to unsteady operation, compared to gasoline operation, the Katech team also identified a 0% to 35% reduction in the volumetric efficiency of the pumps depending upon the operating condition without the vapor lock system. As the delivery pressure to the rail increased, a steeper decline in pumps' volumetric efficiency with propane was observed when compared to gasoline. Other major testing results include the dominant effect of delivery pressure on pumps' fuel flow rate and volumetric efficiency followed by engine speed and fuel temperature. Overall, the non-firing testing highlighted the need for incorporating vapor lock inhibitor technologies in the proposed fuel system design both in terms of stability and performance.

Prior to the firing phase testing, the team started working directly with Stanadyne, a Tier I automotive fuel system OEM, to address the modifications made to the fuel pumps and injectors enabling the opportunity for widespread production and commercialization. Based on the modifications, Stanadyne developed and provided 9x propane - specific injectors and 1x high-side DI pump for firing phase testing. In the firing phase, the proposed fuel system with Stanadyne 200 bar pump and Delphi injectors were installed on General Motors' L8T 6.6L GDI engine. This combination was tested with 87 AKI gasoline operation to understand the baseline torque and power characteristics.

Following the results of baseline testing, the engine was updated to the Stanadyne LPG specific components and operated on LPG DI. Peak torque was increased by 2.6% with propane when compared to gasoline with refined engine calibration, air-fuel control, and maximum brake torque timing. The peak torque location occurred 200RPM earlier for propane than 87AKI gasoline, likely due to higher octane rating and early ignition timing. To understand the robustness of the proposed fuel system design, a 250-hour durability test simulating on-road, low flow, idle, heat soak, refueling, and restarting conditions, through a 30-minute cycle (repeated 500 times i.e., 500 x 0.5 hr. =250 hrs.) was performed on the fuel system components on a firing engine. In short, the pump and each injector were tested for 50,220,000 pump cycles and 16,740,000 injection cycles respectively. The high-side DI pump, often the first source of failure in LPG DI till today, exhibited no degradation in performance during durability testing. The consistency in pump performance was verified by using engine performance parameters (brake specific fuel consumption, fuel flow rate, etc.), post-durability pump performance, and pump disassembly tests. The fuel injectors, unlike high-side fuel pump performance, showed minimal flow shift on a few injectors. Upon post-processing the test data, the team attributed the flow shift to internal injector factors. Internal to injector design, after root cause analysis, the team identified the decrease in injector lift due to wear between the retainer and armature interface.

The design for the retainer and armature interface was modified (at the OEM level) to prevent this wear in the future. More information and run times are needed to estimate the wear or degradation rate for these components with liquid propane operation. Overall, the proposed fuel system with in-house developed vapor lock inhibitor hardware and software addressed the design limitations faced by other LPG DI applicators till today, and also proved the efficacy of liquid propane as a fuel for direct injection.

Table of Contents

Assessment of Liquid Direct Injected Fuel Systems for Propane Engines

1. Background & Introduction

To drastically reduce greenhouse gas (GHG) emissions and to meet stringent future emission standards, it is of utmost importance to transition to better and cleaner mobility options and cleaner burning fuels. Propane, with lower carbon intensity than gasoline, will reduce GHG emissions when coupled with direct injection (DI) technologies. However, DI of liquid propane (LP) has proven to be difficult until today, owing to technical barriers like vapor lock, undetermined durability, and limited science base. Furthermore, the LP-DI has failed to achieve wider market penetration and commercialization due to high conversion costs and limited customers.

To overcome these technical and commercial barriers, Katech Engineering LLC, since September 2021, started working on developing a robust Liquified Petroleum Gas (LPG) Direct Injected (DI) fueled system using both Commercial Off-The-Shelf (COTS) and custom fuel system components through a three-staged approach. The goals & objectives of these three stages are in line with the Propane Education & Research Council's (PERC) mission of exploring, developing, and commercializing propane technologies for medium-duty (Class 3- 7), off road and standby power applications. In Stage I, Katech and PERC decided to utilize COTS components and leverage their performance using a decision matrix. In Stage II, Katech, depending upon the outcome of Stage I, will engineer and develop custom fuel system components and solutions. After benchmarking & down-selecting the fuel system, Katech will perform a limited-time (250 hr.) rigorous reliability testing in-house in Stage III. Katech focused on all stages for this particular docket.

2. Project Objectives

The main objectives of the project are provided below:

- Identifying COTS DI fuel System components (i.e., high-pressure pumps, injectors) that meet PERC metrics.
- Quantifying the impact of LPG on fuel system operation (i.e., high-pressure pumps, injectors), performance, and component wear.
- Quantifying the impacts of fuel pressure and fuel temperature on fuel system operation and performance metrics.
- Developing propane DI-specific engineering and custom control solutions to inhibit vapor lock issues.

- Developing custom development high-side DI pumps and injectors if the COTS DI fuel systems need modifications.
- Operating a medium-duty engine with the proposed fuel system and understanding the operation and deviations encountered at the engine level.
- Quantifying and identifying the deviation in the performance metrics through a limited 250 hr. in-house reliability testing of the best pump and injector combination set.
- Compiling, documenting, and presenting a detailed technical report at the end of significant milestones.

The success measure of the project would be the successful identification, design, development, integration, and demonstration of a high-pressure (>250 bar) direct propane liquid injection system (including pump and injectors) that is also able to pass the 250 hrs. in-house reliability testing mimicking real-world conditions on a test engine. The work related to the above objectives started in September 2021 based on PERC's funding (Docket Number #23027).

2.1 Fuel Pump & Injector Pass/Fail Criteria

The project objectives mentioned above are at a high level. This subsection highlights the data-based measurable metrics for major fuel system components (high-side fuel pumps, injectors, etc.). The metrics mentioned below are used as a basis for the component decision matrix.

Fuel Pump Metrics:

- Priming: Fuel system priming within 30 seconds.
  - Go / No-Go
- Pump flow rate: >4.2lpm or >70 ml/sec @250 bar.
  - 10%
- Operating pressure: Range of max operating pressure
  - 5%
- Volumetric efficiency: Efficiency of pump at operating conditions
  - 10%
- Pump Stress Testing: Maintain performance up to 50°C.
  - 10%
- Pump Endurance: Maintain performance for 250 hrs.
  - 15%
- Mounting: Ability to integrate with existing DI system.
  - 10%
- Modifications: Post OE manufacturing modifications required.

    - 10%
- Cost: Compared to OE GDI Pump
    - 10%
- Scalability: OEM level technical support & manufacturing
    - 20%

<u>Fuel Injector Metrics:</u>

- Injection Repeatability: LFR and low pulse-width stability.
    - 10%
- Static flow rate: Exceed 20g/s at 100 bar pressure (0.8SG reference fluid).
    - 15%
- Injector Endurance: Maintain performance for 250 hrs.
    - 15%
- Injector Stress Testing: Maintain performance up to 50°C.
    - 10%
- Mounting: Ability to integrate with existing DI system.
    - 10%
- Modifications: Post OE manufacturing modifications required.
    - 10%
- Cost: Compared to OE GDI Injector
    - 10%
- Scalability: OEM level technical support & manufacturing
    - 20%

2.2 Decision Matrix Scoring System

The decision matrix, defined in this subsection, was used to narrow down one fuel pump and one fuel injector from a competitive list of fuel system options that Katech chose for initial testing. The down-selected combination was integrated onto an engine platform for performance and durability testing. The step-by-step method for choosing the final outcome from the list of options using a decision matrix is provided below:

1. Identifying various fuel pumps and fuel injectors.
2. Brainstorming & determining the requirement metrics/criteria.
3. Weighting the requirement metrics/criteria.
4. Rating scale creation and scoring the options against metrics.
5. Calculating the weighted scores.
6. Identifying the top choice & making the decision.

The fuel system options (i.e., different fuel system components), requirement criteria, and weightage of requirement criteria are listed in section 2.1 . The remaining steps i.e., rating scale creation and identifying the top choice are the focus of this subsection. Katech's team used a 5-Point scale, with 0 being poor and 5 being excellent, for scoring each of the metrics listed in Section 2.1. The definitions & meanings of each of these scores are provided in Table 1 below. The same ratings and definitions were used for the final scores obtained for each option. However, in this case, a higher score of 5 means the option meets all the requirement metrics listed above instead of one criterion. The formulas for calculating weighed and final scores are provided below:

$$Score_W = \frac{\sum_{i=1}^{n} W_i\, S_i}{\sum_{i=1}^{n} W_i}$$

$$Score_f = \frac{\sum_{j=1}^{k} Score_W}{k}$$

In the equations above,

$Score_f$ = Final score of each individual component

$Score_w$ = Weighed score of each metric for each individual component

W_i = Weightage for each metric

S_i = Score assigned to each metric

$n,\ k$ = total number of metrics

Table 1: Decision matrix scoring system ratings scale and definition.

Scale	Rating	Definitions
5 points (Excellent)	Exceptional. Much more than acceptable.	Exceeds or surpassed expectations all the major/essential parameters for that criterion with no deficiencies at all.
4 points (Above Average)	Very Good & above average.	Meets all of the major / essential parameters of that criterion with no major deficiencies.
3 points (Satisfactory)	Good. Acceptable. Average	Good enough to meet essential parameters of that criterion with some deficiencies exist in the areas assessed but none of major concern.
2 points (Weak)	Less than Acceptable.	Insufficient to meet the major or essential parameters with a few major deficiencies.
0 – 1 point (Poor)	Unacceptable. Much less than acceptable	Significantly fails to meet the essential and major parameters with multiple major & minor deficiencies.

The decision matrix scores for the fuel pumps and injectors chosen for this study are provided in Section 6.2. The upcoming sections will provide detailed project scope and tasks, testing procedure, facilities & instrumentation, and results obtained from individual components and engine durability testing.

3. Scope of work

The overall project's scope entails both 1) the integration of COTS components (various combinations of high-pressure pumps and injectors) for developing the best architecture for LPG DI and 2) limited 250 hrs. durability testing on a medium-duty firing engine. The entire project's scope, covering all the project's objectives, is divided into two major sections, namely non-firing and firing section. In the non-firing development and testing section, details like fuel pump and injector characteristics, fuel pump parametric sweeps, fuel injector parametric sweeps, limited time (25 hrs.) fuel system component reliability metrics, vapor lock inhibitor technologies, and their implementation details were provided. In the firing cell development and testing section, details like DI fuel system integration, fuel system calibration, engine control unit calibration, and 250 hrs. durability testing results were provided. Success measures for all these tests were identified, and relevant details were provided in their respective subsections.

3.1 Non-firing Phase Scope

The scope of work in the non-firing testing phase include:

- Fuel Pumps Identification & Testing processes
- Fuel Injectors Identification & Testing processes
- Limited-time (25 hrs.) Durability Testing
- Vapor lock inhibitor technologies Identification & Testing

Each of these topics are explained in detail in their respective subsections.

3.1.1 Fuel Pumps Identification & Testing processes

While both high and low-side pumps are critical, the primary focus is placed on the high-pressure injection pumps in this project. For the low-side, a commercial-off-the-shelf (COTS) tank-submerged pump, manufactured by TI Automotive, was chosen for the initial non firing testing. The TI pump was updated to a Bosch pump before the firing testing and is described in section 3.2.1. The low-pressure submerged pump, with a maximum pressure of ~112 psig (i.e., 7.72 bar above propane tank pressure), has an outlet diameter of 10 mm, and a length of 130.6 mm. Even though the OEM manufactured the pump for Gasoline and E99, many researchers and propane companies used the same pump for transferring liquid propane. Furthermore, on the low-side pump, the focus was placed especially on the control strategies and system architecture to prevent fuel vaporization in the fuel lines before the high-pressure fuel pump inlet. The Instrumentation & Testing procedure section provides detailed information about these control strategies and system characteristics.

On the high-side pump, based on previous work efforts and discussions with PERC, the Katech team had selected the following pumps:

1. Stanadyne SP1550-200 (200 bar).
2. Bosch HDP6 (350 bar).
3. Stanadyne Prototype Pump (350 bar).

After fuel pump down-selection, the Katech team conducted the design & performance verification testing needed to verify the suitability of these pumps for liquid propane direct injection. The design & performance verification tests were performed according to the SAE J2714 Standards (i.e., Gasoline Direct Injection Pump Standards) [1]. Since no separate standards were available for propane, the project team used SAE J2714 standards as a baseline for the pump testing processes. Additional tests that were not available in the SAE J2714 standards were performed based on Katech's discretion with PERC. In addition to the performance and durability testing requirements from SAE J2714, PERC also proposed some performance and endurance metrics. These metrics are provided below:

- Maintaining the pump flow rate of at least 4.2 lpm at 250 bar.
- Priming of the fuel system should be done within 30 seconds.
- Maintaining the performance of the pumps to 50°C through steady-state temperature testing at 3600 RPM.
- Testing the endurance & performance of the best pump through limited durability testing (i.e., 25 hours on a non-firing bench) and medium-term durability testing (i.e., 250 hours on a firing engine).

In addition to these metrics, the team also worked on understanding the impact of LPG and its compatibility with DI components. Since the COTS fuel system components were developed for 87 octane gasoline, modifications were made to the COTS pumps considering propane's poor lubrication properties and possible fuel vaporization issues [2]. After design & performance verification tests, the team conducted limited durability tests for 25 hrs. for each pump to identify any potential failure modes during LPG operation.

Finally, the success measure for the high-pressure fuel pump is the identification, development, and successful operation of the pump in a 250-hour firing phase while considering all the possible scenarios experienced by the pumps in real-world conditions. For the low-side pump, the success measure is designing and developing control strategies and system architectures for successful operation. These control solutions are essential to prevent vapor lock issues and dry running operation of the high-side pumps.

Preventing vapor lock and dry running operation is critical for meeting the overall fuel system’s performance, operational, and reliability targets defined in the project proposal.

3.1.2 Injector Identification & Testing

Like the high-pressure fuel pumps, the project team also chose three injectors from three different manufacturers. The injectors chosen for this study were given below:

1. Delphi 200 (200 bar).
2. Bosch HDEV6 (350 bar).
3. Stanadyne Prototype LPG Injectors (350 bar).

Using three different high-pressure pumps and three different injector systems will enable Katech & PERC to develop a matrix of different DI injector and pump combinations. All three pumps and three injectors were tested separately before integrating the fuel system onto a firing engine. Of the nine combinations available from the set of components, one was chosen for firing engine testing based on the decision matrix criteria.

Similar to the pump tests, the project team conducted the design & performance verification testing needed to verify the suitability of down-selected DI fuel injectors with LPG. The tests were performed according to the SAE J2713 Standards (i.e., Direct Injection Gasoline Fuel Injector Characterization Standards) [3]. Since no separate standards were available for propane, the project team used SAE J2713 standards as a baseline for the DI injector testing processes. Additional tests that were not available in the SAE J2713 standards were performed based on Katech’s discretion with PERC. In addition to the performance and durability testing requirements from SAE J2713, PERC also proposed some performance and endurance metrics. These metrics are provided below:

- Selecting an injector with a static flow rate of at least 20g/s @ 100bar (0.8 SG reference fluid).
- Maintaining the repeatable flow for injection opening time below 1 ms.
- Identifying the injectors with an operating pressure >250 bar.
- Integrating the down-selected fuel injectors with existing DI pump systems.
- Testing the performance and the endurance of the injector for a limited time (25 hrs. in the non-firing phase) and for a medium time (250 hrs. in the firing phase).

In addition, the project team checked the compatibility of the DI injectors & injector subcomponents for propane. Following the completion of endurance testing in the non-firing and firing phases, the Katech team, assisted by Stanadyne, performed an assessment which documented the condition of injector sub-components, such as, o-ring, needle, armature, and seat. The successful measure for the DI injectors was the efficient and reliable operation of LPG fuel delivery during all testing conditions.

3.1.3 Pump & Injector Reliability Testing

Even though the pumps (both high and low sides) and DI injectors meet the performance and operational requirements individually, it is important to meet the overall fuel system needs under transient and dynamic conditions. To meet this objective, the Katech team performed limited durability tests (25 hrs.) on the individual & the entire fuel system components at a bench level. The entire fuel system included a Bosch low-pressure fuel pump, high-pressure fuel pump, pressure regulating device, pressure recirculation valve, fuel rails, DI injectors, a cylinder mock-up chamber (plenum chamber), feed lines connecting the pumps to the injector, and return lines from the mock-up cylinder chamber to the fuel tank. The detailed schematic of the test setup is provided in the Testing Procedure section, whereas detailed information related to regulating and recirculation valve operation is provided in the Vapor Lock Inhibitor section. The cylinder chamber mock-up chamber (also called the plenum chamber) collected the fuel from all the injectors. The collected fuel was returned to the fuel tank through the return lines connecting the plenum chamber and the fuel tank. In addition, different sensors (pressure, temperature) and flow measurement devices were mounted at different locations to quantify the performance and fuel state at various points in the entire fuel system.

After assembly, the entire fuel system with DI pumps and injectors was tested to see if the proposed fuel system met the operating and performance metrics defined in the previous sections. In addition, the project team identified the flow shift variations for both the high-side pumps and injectors individually. Altogether, the limited reliability testing of the pumps and injectors deepened the scientific base of integrated benchtop fuel systems' operation, flow variation, complexities, and failure modes. Successful completion of this phase in August 2022 provided the confidence to the project team to proceed to the next step (i.e., firing engine testing with the designed/proposed fuel system).

3.1.4 Vapor Lock Inhibitors Design

Due to high vapor pressure and extreme heat transfer conditions (hot soak and hot idle conditions), propane vaporizes inside the high-pressure pumps [2]. Due to this disadvantage, the high-pressure fuel pump, designed for liquid, fails to compress the fuel (due to rapid density & phase change), thereby resulting in an engine stall, failure to maintain power and torque demands and even catastrophic pump failures. To overcome this disadvantage, the project team developed two main countermeasures that inhibit the vapor lock mechanism in the fuel pumps. The first countermeasure was to control the pressure at the inlet of the low pressure fuel pump in order to combat the dynamic vapor pressure of propane at varying fuel temperatures. In addition, choosing a high-pressure fuel pump with a preferential flow design that could be used in conjunction with a regulation valve which controlled the amount of bypassing LPG through the pump. The second countermeasure was to increase low-side fuel pump delivery output, thereby increasing the fuel re-circulation volume at the inlet of the high-pressure pump, maintaining the fuel in liquid state. Due to minimal literature and scarce research, results related to stability, suitability, and effectiveness of individual or the combination of these countermeasures would deepen the understanding and deployment of the DI propane system. Considering that, the project team decided to implement both countermeasures for this project.

Under normal conditions, the low-pressure fuel pump was regulated with Pulse Width Modulation (PWM) to deliver the right amount of fuel to the engine. Normal condition refers to the condition where the fuel temperature in the system is below the vaporization temperature at a given pressure. Under certain circumstances, especially at low flow (low throttle/idle conditions/deceleration fuel cut), the fuel temperature becomes too high in the fuel pump and feed system. The pressure regulating valve, mounted after the recirculation outlet of the high-pressure fuel pump, was used to eliminate the fuel vaporization issues in the feed lines between high-side and low-side fuel pumps, as well as, internally in the high-pressure pump. This was accomplished by controlling the flow rates of the low-side fuel pump based on the temperature and pressure. In other words, the low-pressure fuel pump was operated at the maximum (or near maximum) flow rates at low fuel requirement conditions (i.e., low throttle conditions). While only a certain amount of fuel was delivered to the high-pressure fuel pump chamber, the rest of the fuel was returned to the fuel tank using the regulating valve. Flooding and bypassing the fuel at the inlet of the high-pressure fuel pump chamber eliminated the cavitation (to a certain extent) and dry running of the high-pressure fuel pump. Similar to the regulating valve, the pressure recirculation valve (PRV), mounted at the end of the fuel rails, prevents fuel vaporization in the fuel rails connecting the high-side fuel pump and the injectors. The principle of operation of the PRV was similar to that of the regulating valve i.e., bypassing the fuel in the fuel rail (i.e., before the injectors). In other words, both high-side and low-side fuel pumps over delivered additional fuel in this case so that all the fuel in the lines maintained a liquid condition. Furthermore, during the cold-start/hot-start conditions, the low-side fuel pump was programmed to deliver additional fuel to purge the heat-soaked fuel in the lines. Overall, PRV and regulating valves, along with the control strategies, enabled vapor lock free operation of fuel pumps and injectors.

3.2 Firing Phase Scope

The scope of work in the firing phase includes:

- Integration of down-selected fuel pump and injector combination on a running engine.
- Initial control system setup and calibration process.
- Engine power & torque maps for gasoline and propane.
- 250-hour durability testing with propane.

All of these topics were explained in detail in the subsections below.

3.2.1 Engine Performance & Fuel System Durability Testing

After the successful completion of the non-firing testing phase, Katech, in collaboration with PERC, installed the most suitable fuel pump and injector combination (outcome from decision matrix) along with instrumentation on an engine testbed. This testbed enabled the team to mimic and test the final fuel pump and injector combination with propane under the quasi-real-world operating conditions (hot soak, hot start, cold start, etc.). For this purpose, the project team considered a 400hp, 6.6L, DI, naturally aspirated V8 engine. This engine is currently being used in on-road General Motors & Isuzu medium-duty trucks. For testing purposes, the proposed engine, originally a gasoline-fueled GM L8T engine, was modified to accept the proposed DI propane fuel system. The GM L8T engine has a bore, stroke, and stock compression ratio of 103.25 mm (4.065 in), 98.00 mm (3.850 in), and 10.8:1, respectively. The stock engine is rated at 401 hp (299kW) @5200 rpm and 464 lb-ft (629Nm) @4000 rpm with 87 gasoline. The engine dimensions are as follows: L 29.75 in, W 26.10 in, H 31.30 in. For initial testing purposes, the engine will be in OEM form, with no modifications other than what's required to implement the LPG fuel system.

When converting to the LPG engine

Results from previous projects led by Katech, as well as studies conducted by other entities, show that the use of propane in a combustion engine can compromise the sealing capability of the valve seat. This is primarily exhibited in the recession of valve seats into the cylinder head but can also be seen in overall degradation of the valve seat to valve face sealing surface. This phenomenon has been determined to be a result of the lack of lubricity and the increase of valve seat temperature (primarily gaseous LPG port injection) when using LPG**. For this study, the factory L8T valves and seats were maintained, as the use of direct injection of liquid propane has not been proven to exhibit these same effects. In addition, the ability to minimize engine modifications aids in achieving the objectives of this study**. Depending upon the testing results, the cylinder heads will be modified to accept improved valve seats and valves. The potential replacement seats are specific to propane fuel and have been developed and tested with suppliers in other programs that Katech has led.

Prior to the engine testing, it is of utmost importance to develop a robust and reliable DI fuel system. A few modifications to the fuel system design established in the non-firing test phase were made prior to integrating them onto the engine platform. First of all, the COTS in-tank submerged low-side pump was updated from a TI-Automotive to a Bosch fuel pump which allowed for an increase in fuel volume output, and higher delta pressures between the low-side pump outlet and fuel tank. The Bosch fuel pump is a high output single-scroll composite turbine-style pump. Unlike the TI automotive pump, the Bosch pump has a slightly higher PRV activation pressure of 120psi (8.27 bar above tank pressure) [TI pump's activation pressure is 112psi]. The dimensions of the Bosch low-side pump are 10 mm outlet barb fitting, 46 mm OD, and 138 mm overall length. Secondly, the PRV , which was placed in the injector rails during non-firing bench testing, was eliminated and not used during the firing phase. **Even by utilizing just the regulation valve, the high-side fuel pump was deemed capable enough to build and maintain the desired rail pressure for the scope of the durability testing considered in this program**. **Essentially, the regulation valve at the high-side pump inlet was sufficient to maintain greater than 43 bar (critical pressure of LPG), which guaranteed liquid state in the high-pressure system if below 97 ° C (critical temp of LPG) and maintains liquid fuel density if system was above 97° C.** If in the future, the fuel vaporizes in the injector rails or requires purging after losing pressure, then it is advisable to add the PRV valve in the injection rail. Since the team has tested the control & operation of the PRV, it is easy to implement this change in the future without any major implications.

After the above-mentioned modifications, the final fuel system was assembled on the engine test platform. Later, the team performed the initial engine calibration and verified the DI engine operation of LPG. The initial calibration included the engine operation through different fuel mapping sweeps, ignition sweeps, speed, and load sweeps. After this testing, the DI fuel system met the operating and performance requirements even under extreme conditions (i.e., repeatable hot start & hot soak operation) with the calibrated fuel maps. After engine mapping, 250 hrs. of durability test simulating on-road, low flow, idle, heat soak, refueling, and restarting conditions, through a 30-minute cycle (repeated 500 times, i.e., 500 x 0.5 hr. =250 hrs.) was performed on the fuel system components on a firing engine. In conclusion, the pump and injectors were tested for >50,220,000 and >16,740,000 cycles respectively. The durability cycling tested the robustness of the fuel system and the engine controls developed in this phase.

4. Test Facility, Setup, & Instrumentation

4.1 Test Location & Facility

All the tests were conducted at Katech Engineering LLC, a 36,000 square feet facility, located in Clinton Township, Michigan. This facility houses all Katech's precision CNC manufacturing machines, engine dynamometers, calibration cells, engine component manufacturing, and engine building stations.

4.2 Non-Firing Testbench

This section provides all the details regarding the test schematic, instrumentation, and equipment used during the non-firing phase for determining feasibility and performance characteristics of both individual fuel system components and the overall fuel system.

4.2.1 Experimental setup

For the feasibility and performance testing of the fuel system in a non-firing phase, a non-firing rotating engine assembly, initially designed for gasoline, was modified for propane fuel testing. The test rig was capable of handling individual pump testing, individual injector testing, pump and injector combination testing, and component durability testing with minimal modifications. An image of the test rig is shown in Figure 1.

The test rig was completely designed and modified in-house, and it operates the necessary fuel system components without the use of a firing engine. Necessary instrumentation and sensors were added between the fuel tank and the fuel injection rail to measure relevant fuel data (temperature, pressure, flow rate, etc.) during testing. As shown in Figure 1, the test rig consists of the following components in a given order:

1. Fuel Tank.
2. Fuel Delivery Pressure/Temperature (low-side outlet/high-side inlet conditions).
3. Fuel flow meters.
4. High-pressure DI pump.
5. Variable Pressure Regulation Valve.
6. DI fuel injectors.
7. Recirculation relief valve – mounted at the end of the fuel rail.
8. Fuel Return Pressure/Temperature.
9. Fuel return lines to the tank.
10. Fuel tank fill valve.
11. Fuel evacuation system (red) – during downtime/component modifications.

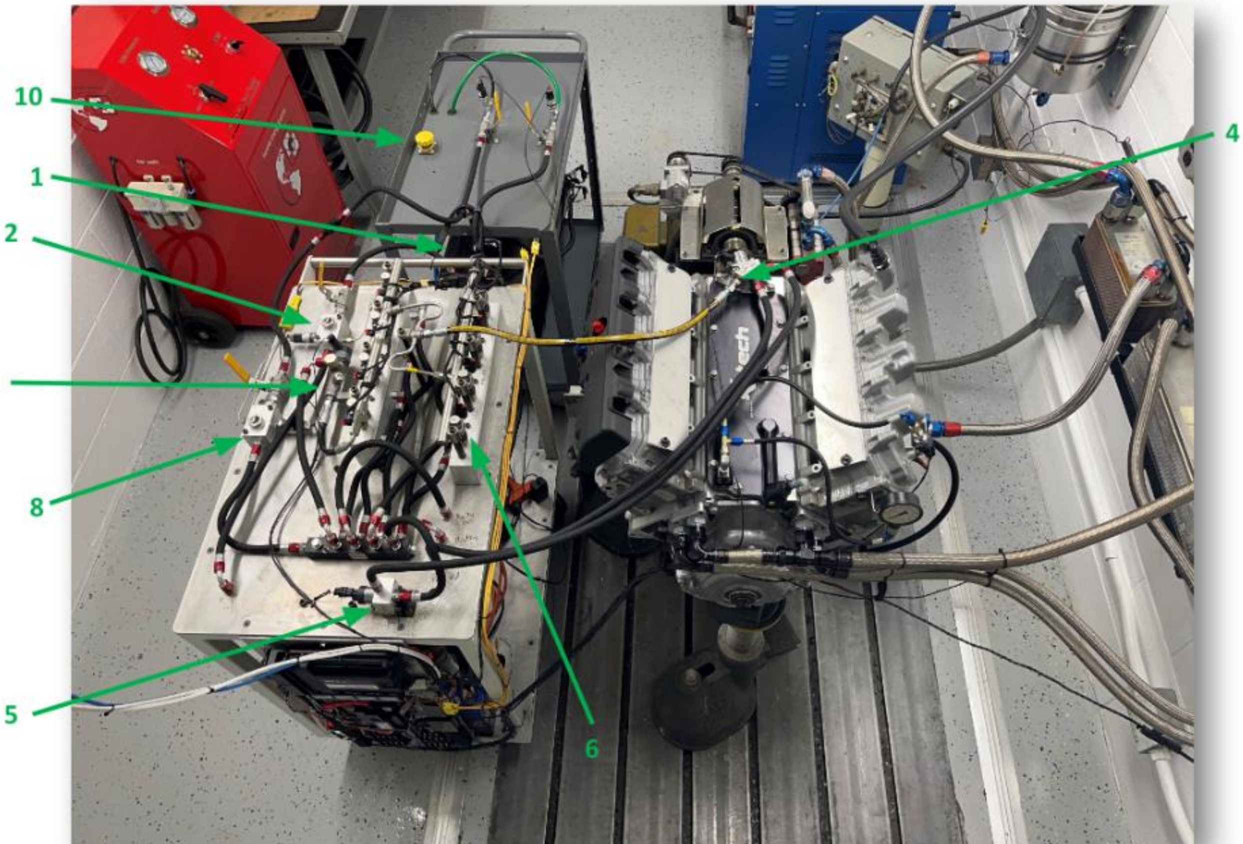

Figure 1: Photograph of the non-firing test rig developed for this project

Schematic representations of the pump and injector test rigs are shown in Figures 2 and 3. Upon careful observation, both the schematics are almost the same except for the location of the secondary Coriolis meter. In the pump test rig, the secondary Coriolis meter was placed in between the low-side pump outlet and the variable fuel regulation valve. In the injector test rig, it was placed in the fuel line connecting the injector outlet to the variable fuel regulation valve. During the injector testing process, one injector was disconnected from the injector block to measure single injector fuel flow rate. The secondary Coriolis meter, in this case, provided the amount of fuel delivered by the injector. During the pump testing process, the difference in the Coriolis meters provided the amount of fuel delivered by the high-pressure DI pump to the injector rails.

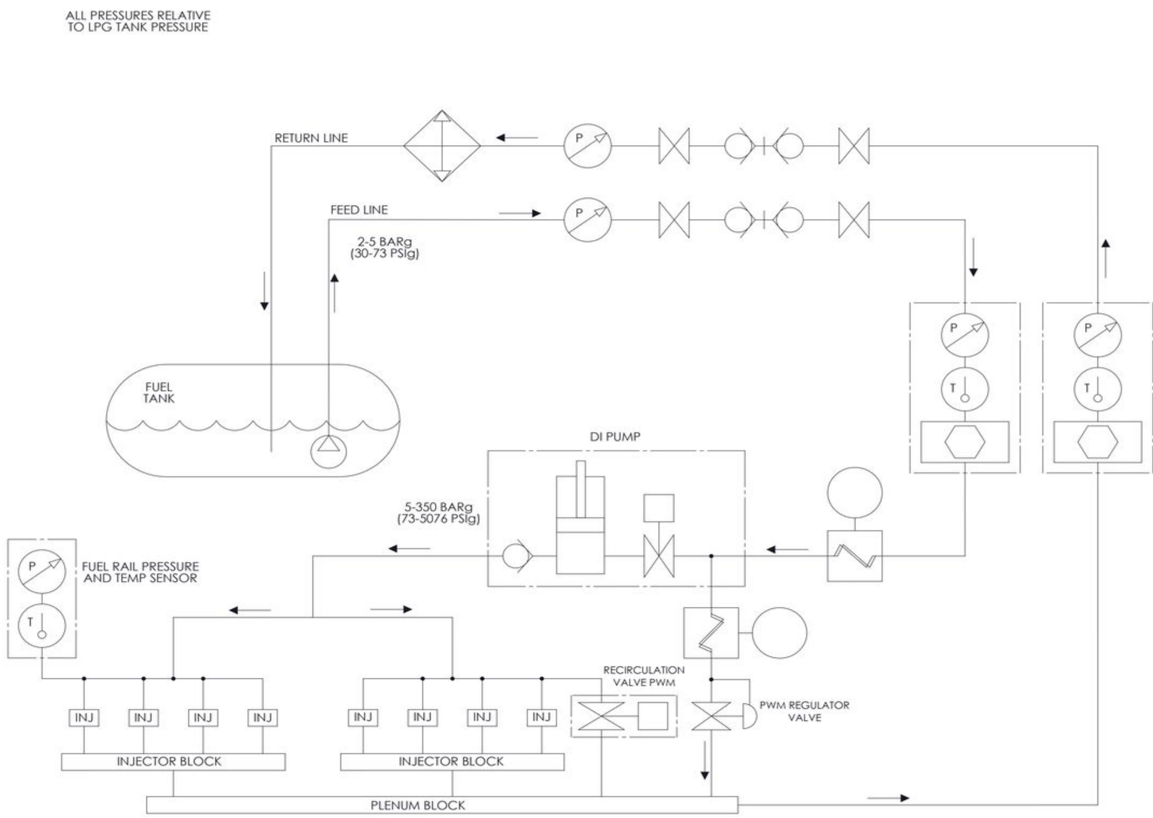

Figure 2: Schematic drawing of the high-pressure DI pump test rig

For example, for the high-pressure DI pump testing, one mass flow meter was placed prior to the inlet of the pump, and the other was placed in between the modified fuel pump port and PWM-controlled regulation valve. Modification in the fuel pump is necessary for adding a regulating valve in the system. This regulating valve is needed to overcome the vapor lock/cavitation issues in high-side DI pumps at low flow conditions. The outlet of the PWM regulation valve is connected to the return line that connects to the fuel tank.

The propane fuel tank, shown in Figures 2 and 3, is a twenty-five-gallon steel unit with access ports for mechanical, fuel pump, and valve servicing. The fuel, obtained from Corrigan Oil, complied with the HD5 grade standards. The fuel tank interface has ports to accommodate the fuel supply line, fuel return lines, and fill port. A low-side fuel pump, TI Automotive F90000285, submerged in the fuel tank, delivered propane to the high-pressure DI pump through a series of instrumentation. All the details regarding the instrumentation are provided in Section 4.2.2.

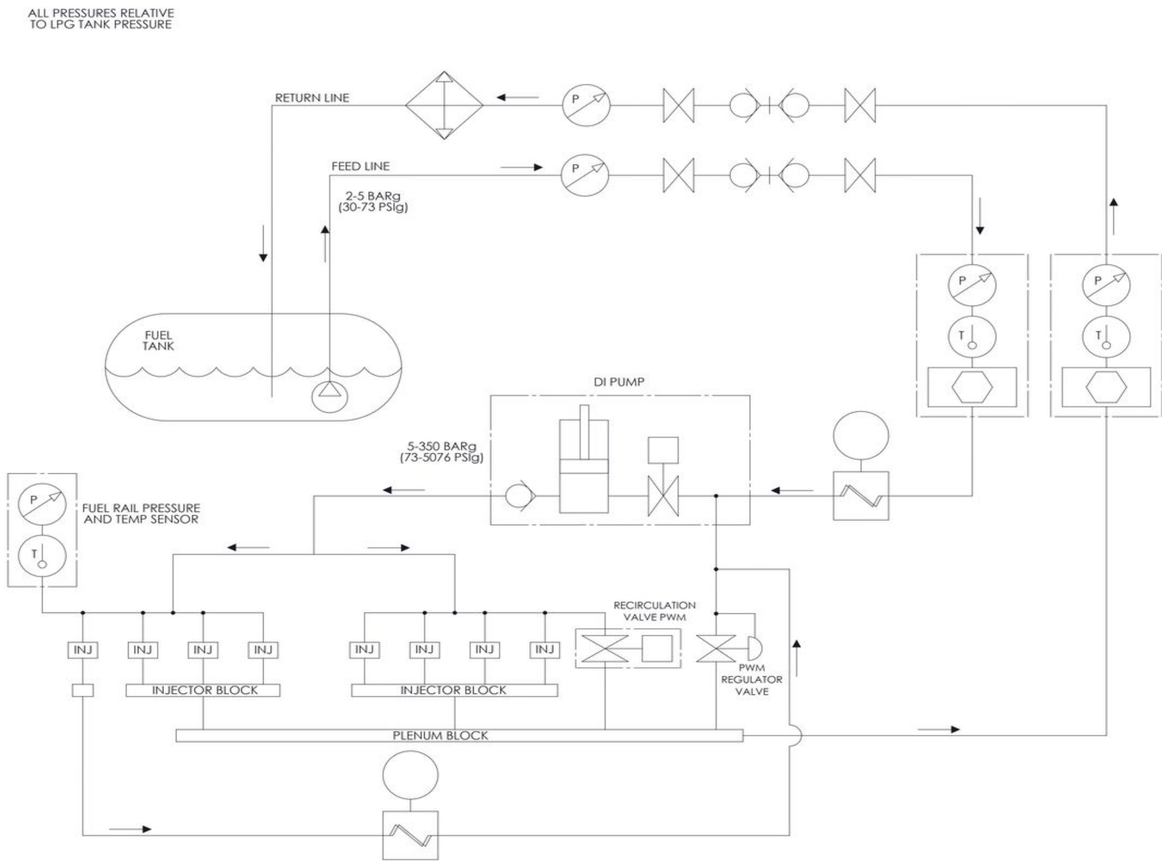

Figure 3: Schematic drawing of the high-pressure DI pump & injector combination test rig

The fuel flow from the DI pump outlet was directed to the injector rails and then to the injectors. Also, as shown in Figures 2 and 3, a PWM-controlled recirculation valve was fitted to the fuel rail in order to avoid the vapor lock issues in the injector rail. Furthermore, it also offered the capability to purge the fuel rail during extreme vaporization conditions. All the fuel from the injectors was collected into an injector block. The injector block, as shown in Figure, was connected to a plenum block which acted as a primary fuel collection chamber after fuel discharge. In other words, all the fuel from the recirculation valve, regulation valve, pumps, and injectors were collected into a plenum block before returning it to the fuel tank through a return line.

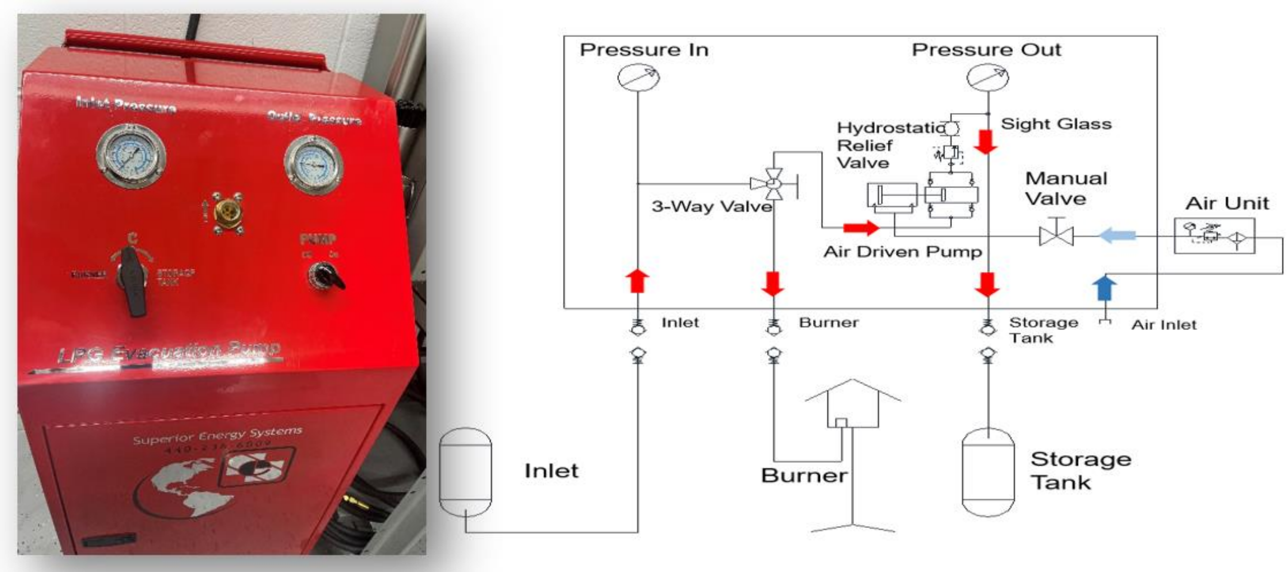

Figure 4: LPG evacuation system & its operating schematic.

An LPG evacuation system, Superior Energy Systems make, was used to evacuate the pressurized propane in the fuel lines, injectors, and pumps. The evacuation system was added to effectively and safely service and change the components if needed. The operating schematics and the actual evacuation system are shown in Figure 4. A 12kW heating unit with self-contained fluid was used for fuel temperature sweep testing. The heating unit, in the test cell, exchanges the heat with the engine oil and engine coolant from other test cells and it comes with a closed-loop control for regulating the temperature. The image of the heating unit used in the test rig is shown in Figure 5.

To minimize the possible fuel system component failure being tested, care was taken to choose components & materials that are compatible with propane. Furthermore, materials made with stainless steel were used for fuel tank, injector rail, fittings, and other supports. Eaton Aeroquip hose & fittings were used for the low-pressure fuel supply and return lines.

4.2.2 Instrumentation

Various sensors and instrumentation were installed in the non-firing testbench in order to understand the flow characteristics, performance, and stability of both individual components and the overall fuel system. For example, prior to the high-pressure DI pump, the pressure, temperature, and fuel mass flow rates were measured by using the respective sensors. For this project, the team used low-pressure sensors obtained from S&S with a measurement range is 0-35 bar(g) with 1.0% accuracy over the full scale and a response time of 1 ms. Proof Pressure for these low-pressure sensors is 2x the full scale, and burst pressure is 5x the full scale. For high-pressure measurement in the fuel rail, a Bosch PSS-420 high-pressure sensor was used. This sensor has an operating range of 0-420 bar(g), with a max measurement of 560 bar(g). The sensor maintains an accuracy of 1% for the full range. A simple K-type thermocouple was used to measure the fuel temperature at desired locations to decide the fuel state and also to calculate the fuel density correction factor.

Figure 5: 12 kW heating unit for regulating fuel temperature.

The mass flow rate was measured by Endress + Hauser Proline CubeMass C100 mass flow meter. The flow meter is capable of measuring mass flow rates of 0-100 kg/hr with liquid flow rate accuracy of +/- 0.1% and vapor flow rate accuracy of +/- 0.5%. Two mass flow meters were used and placed at different locations depending on the test requirements. For example, for the high-pressure DI pump testing, one mass flow meter was placed prior to the inlet of the pump, and the other was placed in between the modified fuel pump port and PWM-controlled regulation valve. A GCM48 Rapid Development control unit with MATLAB® Simulink-based software toolset is used to control the fuel pump output, regulation valve, and recirculation valve. The Proline C100 mass flow meters and GCM48 control units are shown in Figure 6.

A Bosch MS6.4 DI Engine Control Unit (ECU) with Motorsport Calibration Software (INCA based) was used for calibration and control of the fuel system assembly in the non-firing test bench. The ECU can handle eight DI injectors and uses internal data logging capabilities, either time-based (i.e., for every 1 ms) or speed based. A race grade thermocouple to CAN controller with 16 T-Type thermocouples is used for temperature data logging. An image of the ECU with the power distribution relays and thermocouple to the CAN adapter is shown in Figure 7.

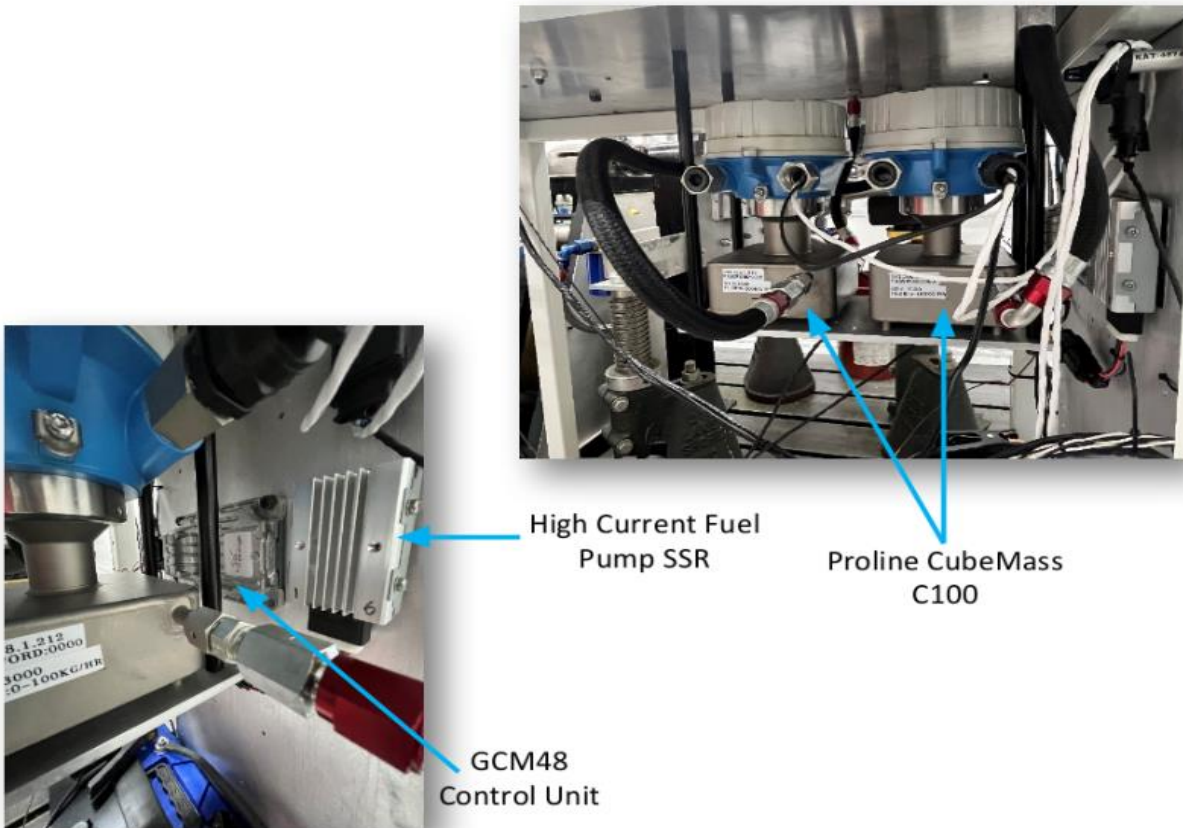

Figure 6: Mass flow meters and control unit used in the proposed test rig

4.2.3 Standard Conditions & Deviations

Unless otherwise mentioned in the text/tables/images, the following test conditions were maintained as standard test conditions for all the non-firing phase results presented in the document.

Ambient Conditions: The fuel system and the components were tested at an ambient temperature of 20°C +/- 2°C and an ambient pressure of 100 kPa +/- 5 kPa.

Fuel Composition: Propane adhered to HD5 grade standards.

DI Pump Inlet Fuel Temperature: For standard & baseline tests, the fuel temperature in the lines prior to the high-pressure DI pump inlet was maintained at 20°C +/- 2°C. For temperature sweep testing, the temperature was specified separately in the results. However, the accuracy was still maintained within +/- 2°C.

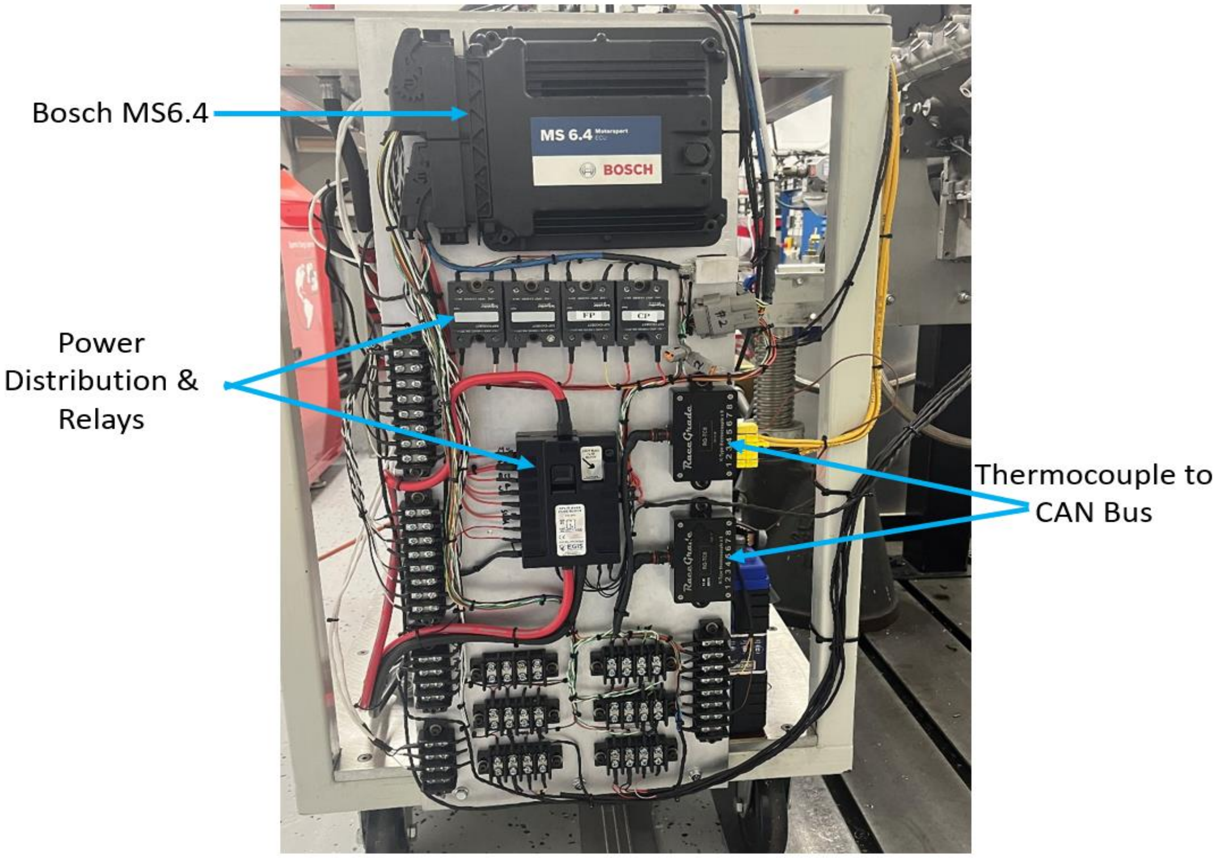

Figure 7: Image of ECU, power distribution relays, and data loggers

DI Pump Inlet Pressure: This is the average gauge pressure measured prior to the inlet of the DI pump. The DI pump inlet pressure was maintained 3-bar above the propane tank pressure for all the standard test cases. The tank pressure depends upon the amount of fuel, state of fuel, temperature, and other external factors. The fluctuations in DI pump inlet pressure were maintained to +/- 2% of the setpoint value throughout the entire test.

DI Pump Outlet Pressure: This is the average gauge pressure measured at the outlet of the DI pump. Similar to the pump inlet pressure, the fluctuations in outlet pressure were also maintained to +/- 2% of the setpoint value throughout the entire test.

Engine Speed: For most of the steady-state performance and baseline testing, the engine speed was maintained at 3600 RPM. For durability, pressure, and temperature sweep testing, the engine speed was varied, and it was specified separately wherever possible.

In both these cases, the engine speed was maintained within a tight narrowband range of +/- 25 RPM.

Injector Body Temperature: The temperature of the injector body at the beginning of the testing was close to the ambient temperature of 20°C +/- 2°C. While the injector body experienced temperature swings during testing, these temperature swings were out of the scope of the initial baseline and individual component performance testing in the non-firing phase. These temperature swings were monitored during the 250-hr. durability test, where the entire fuel system was tested on an actual engine in the firing phase.

Injector Pressure: Unless otherwise mentioned in the results, the injector pressure was maintained at 100 bar for baseline and performance testing. The fluctuations were also maintained within ±0.5% throughout the test. When a particular injector pressure was tested, the variations, as well as the actual pressure, were mentioned in the data-reporting sheet or in tables.

Injection Pulse Width (IPW): For standard, baseline, and durability tests, the injection pulse width was maintained at 4 ms. with a deviation of +/- 0.001 ms. When checking for linearity during performance testing, the injection pulse width was changed from 0.2 ms to 6 ms. These conditions encompass the entire operating conditions seen in a normal engine operation. For the tests requiring a specific IPW valve, the IPW was stated in the corresponding test sheet, table, or graph.

Injection Period (IP): The injection period is a test-dependent parameter, and it is specified separately for each and every test. The fluctuations in the injector period were maintained within +/- 0.005 ms and were determined by the engine operating rpm.

Injection Waveform & Drivers: The injector driver type varied from injector to injector. For this particular project, the injectors were energized by the internal injector driver in the Bosch MS6.4 Control Unit. The current profile used for controlling the injector is shown below in Figure 8.

INJ_PREMAG_CUR_H = 1.00 A (amps)
INJ_PREMAG_CUR_L = 0.75 A
INJ_PREMAG_LENGTH = 0.000 ms
INJ_BOOST_CUR_H = 17.00 A
INJ_BOOST_CUR_L = 16.00 A
INJ_BOOST_CYCLE = 1
INJ_BOOST_LENGTH = 0.800 ms
INJ_PICKUP_CUR_H = 8.00 A

INJ_PICKUP_CUR_H = 7.00 A
INJ_PICKUP_CUR_LENGTH = 1.000 ms
INJ_HOLD_CUR_H = 3.00 A
INJ_HOLD_CUR_L = 2.50 A
INJ_HOLD_LENGTH = Injector Pulse width – (boost length + pickup length)

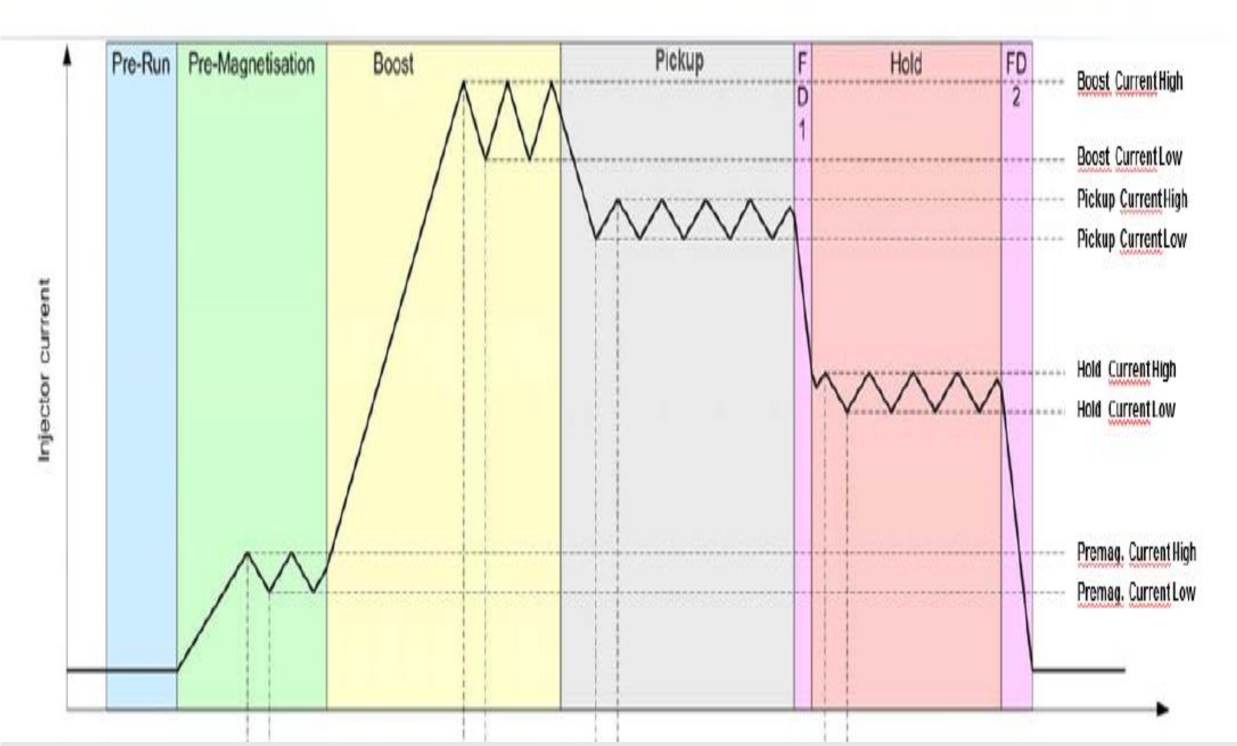

Figure 8: Injector current profile used in the testing process [4]

Flow measurement: The fuel flow rate can be measured by either mass or volume flow, with the former being the most preferable. The data presented in the results section was reported in mass flow units unless otherwise specified. For example, the flow rate was specified in g/s for static tests and g/s or mg/pulse for dynamic testing conditions.

In addition to these parameters, all pertinent information (number of pulses, period, speed, etc.) were recorded to make relevant conclusions. All the injectors and pumps were preconditioned and flushed prior to the static flow measurements. The dynamic flow measurements were obtained after the static flow measurements.

4.3 Firing Testbench

This section provides the details regarding the integration of the fuel system, test schematic, sensor and instrumentation used in the firing phase for engine performance characteristics and durability of fuel system, valve seats, and the engine itself. The schematic of the test cell with the engine installed on the dynamometer is provided in Figure 9 below.

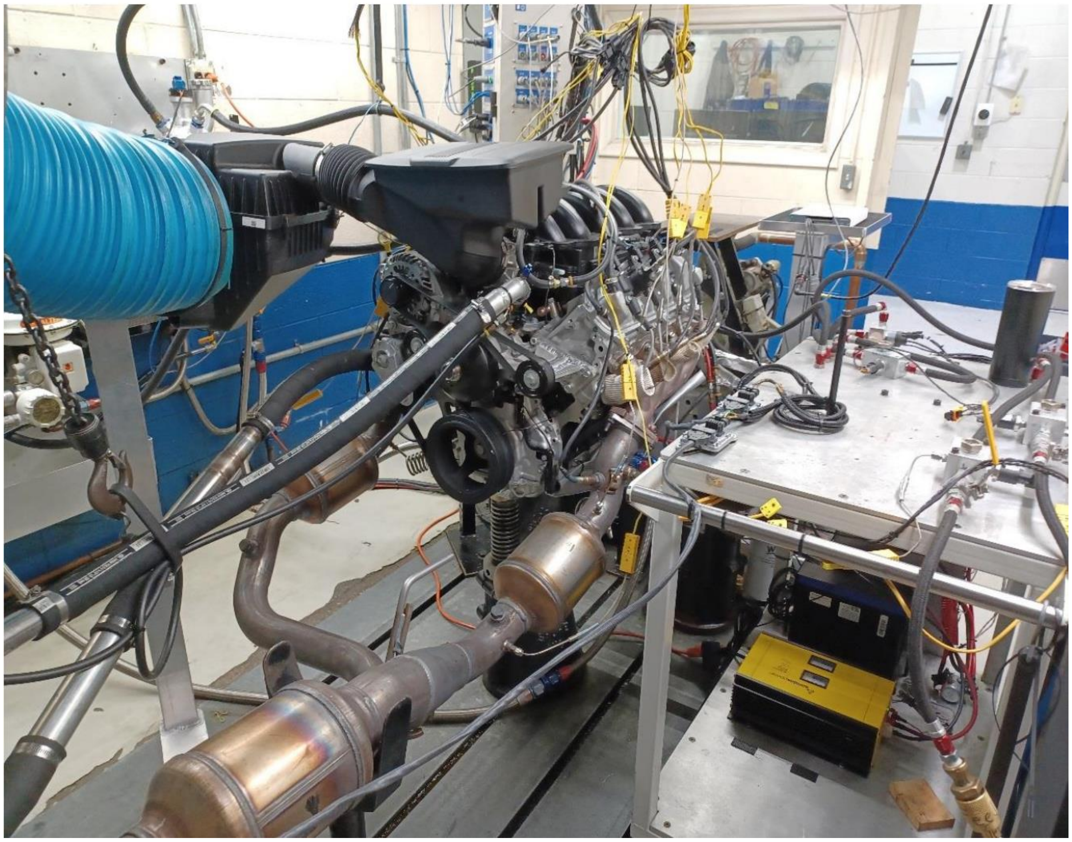

Figure 9: Experimental setup of GM L8T Engine on Superflow dynamometer

4.3.1 Engine Unit-Under-Test (UUT)

The engine chosen for the testing purpose is a General Motors L8T, 6.6-liter, direct injected gasoline V8 engine. Katech initially conducted the gasoline baseline testing with the Original Equipment Manufacturer (or OEM) engine design. The specifications of the engine are provided in Table 2.

Table 2: Specifications of GM L8T 6.6-liter DI engine

Engine Parameter	Description
Engine Code	L8T
Engine Configuration	V8 Naturally Aspirated
Engine Displacement	6564 cc
Engine Bore	103.25 mm
Engine Stroke	98 mm
Compression Ratio	10.8:1

Valvetrain Configuration	Pushrod OHV with Variable Valve Timing
Valves Per Cylinder	2
Rated Horsepower –2500 GM *Isuzu Medium Duty N-Series*	401 *hp* @ 5200 *rpm* *350 hp @ 4500 rpm*
Rated Torque – 2500 GM *Isuzu Medium Duty N-Series*	464 *lb-ft* @ 4000 *rpm* *425 lb-ft @ 3800 rpm*

4.3.2 Pre-testing Engine Inspection & Documentation

The engine procured from the OEM was baseline tested with the Gasoline DI fuel injection system. This baseline test established the gasoline benchmark for the LPG system. Since the gasoline DI fuel injection system was replaced with the propane DI system, it is of utmost importance to inspect and document all the modifications done to the original UUT. Prior to engine testing, the modified UUT was prepped & evaluated by Katech's Engine Build & Testing Department. The engine build team evaluated the condition of each base engine component to make sure the modified UUT was eligible for dynamometer testing. In addition, not all the OEM components/sensors were required for dynamometer testing, or some were incompatible with the data acquisition or control/calibration system. Depending upon the requirements, the Engine build team either replaced or modified the incompatible components/sensors.

4.3.3 Experiment Platform

During this phase, Katech retrofitted one of its pre-existing engine test platforms to accommodate the proposed liquid propone DI fuel system & the required test engine to develop and validate the functionality and performance.

For this testing phase, Katech used a Schenck DS750 water-brake dynamometer operated with a SuperFlow data acquisition and control unit. This test cell is capable of 1000hp and 15000rpm, allowing for the complete control of engine speed and load with the use of automated dynamometer servo inlet and outlet valves.

4.3.4 Instrumentation, Data Acquisition, and Control

The engine test cell used for the duration of this project was equipped with multiple sensors, heat exchangers, and other components needed for engine performance mapping. The sensors/instrumentation used in the firing phase, along with different data acquisition systems, are provided in Table 3 below. The majority of the instrumentation/sensors were off-the-shelf automotive sensors available in the market. Whenever a special sensor was used, the make and model details were provided in the table for future reference (e.g., Optrand H322J6-SP Spark plug cylinder transducer). As shown in the table below, the team used three different data acquisition systems. Time-averaged data series like intake manifold pressure, exhaust pressures and temperatures, engine oil temperatures, engine power, torque, etc. were measured using a low-speed data acquisition system that came with the engine dynamometer (i.e., SuperFlow software). The DI fuel system parameters like pump temperature, pump localized temperature (i.e., engine block), fuel pressure and temperatures, fuel flow, etc. were captured in the Bosch Engine Control unit. Finally, the time-dependent (i.e., crank angle related data) like injection and ignition timings, engine cylinder pressures, etc. were captured by National Instruments (NI) data acquisition system. Prior to any data acquisition & reporting, system calibration checks were performed multiple times (i.e., before the test, at random intervals throughout tests, and after test completion) to ensure accurate data.

Table 3: Instrumentation and Data Acquisition Systems used in Firing Phase

Dynamometer DAQ List	Bosch ECU DAQ	National Instruments DAQ
Manifold Absolute Pressure (kPa)	Dual Bank Lambda	CaTool RT Combustion Analysis
Individual Cylinder Lambda	Dual Bank and tailpipe NOx	Actual Injection Timing/Duration
Exhaust Gas Temperature (°F)	DI Pump temperature (°C)	Actual Ignition Timing/Duration
3Pre/Post Catalytic Converter Temperature (°F)	Engine block temperature (local to DI Pump) (°C)	Cylinder Pressure (kPa)
Exhaust Backpressure (both cylinder banks) (kPa) gauge	Fuel temperature (inlet and outlet) (°C)	User Defined Calculated Fields
Inlet Air Temperature (°F)	Fuel Delivery Pressure (Inlet/Outlet) (Bar)	Optrand H322J6-SP Spark plug cylinder transducer LTR7IX-11 Spark plug
Engine Coolant In/Out Temperatures (°F)	DI Pump Fuel Flow (g/min)	60-2 Crankshaft Encoder Wheel (factory crank position sensor/wheel)
Engine Coolant Flow (GPM)	DI Pump Pressure (Bar)	
Engine System Coolant Pressure (psig)	Ignition Timing	
Oil Pressure (psig)	Fuel Pulse-width	
Engine Oil In/Out Temperature (°F)	DI Fuel Pump Delivery Angle	
Engine Oil Flow (GPM)	Throttle Position	
Engine Crankcase Pressure (inHg) gauge	Engine Speed	
Engine Blowby (acfm)	Vapor Lock Fuel Module Data	
Engine Torque (lb-ft)		

Apart from the data acquisition, these three systems (Superflow, Bosch, and NI) in the test cell allowed the team to develop control algorithms needed for automated load testing. The SuperFlow dynamometer controller handled automatic throttle position and engine speed changes using a closed-loop algorithm developed by the Katech team. While there are multiple ways/strategies for automated load control, the throttle position was used for load control, and the dynamometer servo valves were used for engine speed control. These two modes were used simultaneously to simulate driving cycles.

- *Load Control Throttle Position:* The dynamometer software was programmed with a defined desired throttle position value. The desired throttle position valve was derived from the load point requested in the test design. When a target load point was set, the dynamometer throttle actuator, connected to an accelerator pedal position sensor, advanced the pedal position until it reached the programmed throttle opening value. Unlike the torque-based control strategy, the load control throttle strategy ensured that the engine performance loss was not concealed by the automated increase in demanded throttle for throttle positions below 100%.

- *Engine Speed Control:* The dynamometer was also equipped with servo inlet and outlet valves to control the water flow through the brake. The test cell equipment controller utilized these valves to maintain the engine speed very close to the target setpoint.

4.3.5 Standard Conditions & Deviations

The standard test conditions and their deviations for the firing tests are very similar to the non-firing standard test conditions and deviations unless otherwise mentioned in the graphs/figures/tables. Apart from those conditions, the following are the additional conditions that were considered for the firing phase testing.

Engine Oil Temperature: Unless otherwise mentioned in the results, the engine oil temperature was maintained at 200 F for gasoline baseline, LPG performance, and durability testing. The fluctuations were also maintained within ±5 F throughout the course of the test.

Engine Oil Type: The team used Mobil1 5W30 Dexos I Gen oil throughout the course of the test.

Engine Coolant Temperature: Like the engine oil temperature, the coolant temperature was maintained at 200 F with ±2 F throughout the course of the test.

Engine Coolant Composition: The team used ACDelco Dexcool 50/50 mix throughout the course of the test.

Fuel: The team used 87AKI Gasoline, and propane adhering to HD-5 standards.

Other conditions/components: The team used L8T OEM Cast manifolds from GM, and one OEM three-way catalyst for each cylinder bank, and one OEM downstream tailpipe catalyst. An OEM water pump, and OEM alternator (not operated for charging) were used during the testing process. Other auxiliaries like the power steering pump and AC compressor were not considered/used during the testing.

4.4 Testbench Differences

The following are the differences in DI fuel system components and other instrumentation used between the firing and non-firing test phases.

- The DI injectors (initially Delphi in non-firing) were replaced with Stanadyne 350bar Development injectors that were designed specifically for propane in the firing phase.
- The high-side DI pump (initially Stanadyne 200 in non-firing) was replaced with a Stanadyne 350bar development pump that was designed and engineered for propane for the firing phase.
- The low-side fuel pump (initially TI automotive pump) was replaced with Bosch submerged pump for increased capacity.
- The pressure recirculation valve in the injector rail was eliminated as the high-pressure DI pump with the proposed vapor lock inhibitor was sufficient to prevent the vaporization of fuel in the fuel lines.
- Added external high-side DI pump temperature sensor to understand the pump failures if any.
- Added engine block temperature (localized high-side DI pump temperature) to understand the pump failures if any.
- Added high-side DI pump mass flow rate sensor which was not available in gasoline baseline testing.

4.5 Vapor Lock Inhibitor Control

To combat the vaporization of fuel in the fuel lines connecting different components in the fuel system, certain vapor lock inhibitor technologies were developed and implemented both at hardware and software levels. These technologies assisted the team in the testing process and also offered the refinement of the fuel system without the risk of damage. In this section, the team presented software logic, internally developed, for stable fuel system operation.

The software, written and designed in a GCM-048 Fuel Pump Module, was used to control various actuators to avoid the vapor lock situation and to feed adequate low-pressure propane to the high-pressure pump. These actuators include the fuel pump voltage (regulated PWM DC %), the regulation valve voltage (regulated PWM DC %), the recirculation valve voltage (regulated PWM DC %), the fuel fill solenoid valve (digital output), and the fuel run solenoid valve (digital output). All of these are driven by low-side outputs in the GCM-048 unless stated otherwise.

To control these actuators, data from various sensors and ECU data was utilized through CAN-bus communication and direct sensor measurement. These sensors include engine speed, fuel flow rate calculated, tank pressure, fuel feed pressure, rail pressure, high-pressure fuel pump delivery percentage, and fuel temperature. Based on the inputs from various sensors, the team calculated the required outputs, one of which is the identification of operating mode. The control logic behind the operating mode identification was provided in the “State machine” Simulink block shown in Figure 10. This block does the selection process for what mode the fuel system is to operate in.

The modes, enumeration and criteria are listed below:

- Mode 0: Off
- Mode 1: Ignition On (not running)
- Mode 2: Engine Cranking
- Mode 3: Running
- Mode 4: Low Flow Running / Deceleration Fuel Cut
- Mode 5: Error/Fault

Based on the software, the priority of the system modes is 5, 2 ,4, 3, 0, 1. The criterion for each of these modes based on priority is provided below.

Mode 5 Criteria:

- Ignition On.
- Any critical sensors have failed or are outside of acceptable range.
- Tank pressure or temperature has exceeded acceptable level.
- Loss of communication with ECU.

Mode 2 Criteria:

- System Does not meet criteria for Mode 5.
- Ignition On.
- Engine Speed > 50rpm (programmable value).
- Engine Speed < 350rpm (programmable value).

Mode 4 Criteria:

- System Does not meet criteria for Mode 2 or 5.
- Ignition On.
- Engine RPM > 350rpm (programmable value).

- Engine LPG_LowFlow_Limit = 1 (map that is programmed for low flow).
- Engine RL_LowFlow_Limit = 1 (map that is programmed for low flow).
    - RL (relative engine load).

<u>Mode 3 Criteria:</u>

- System Does not meet criteria for Mode 4, 2 or 5.
- Ignition On.
- Engine RPM > 350rpm (programmable value).

<u>Mode 0 Criteria:</u>

- System Does not meet criteria for Mode 3, 4, 2 or 5.
- Start Delay has not been met or Ignition is off.

<u>Mode 1 Criteria:</u>

- System Does not meet criteria for Mode 0, 3, 4, 2 or 5.
- Ignition On.

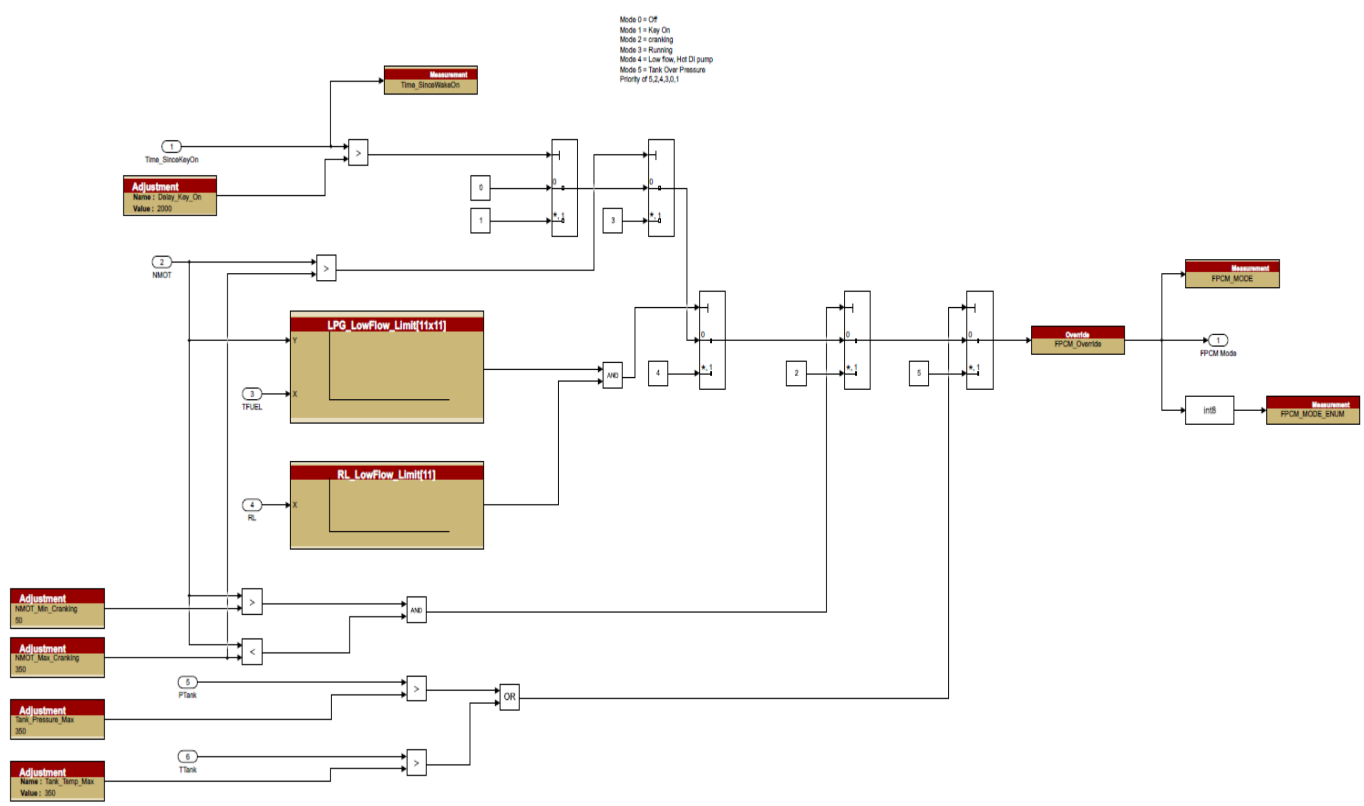

Figure 10: Simulink Mode Selection Block

The actuator output based on the modes can be seen in table 4 below. The project team will provide the images of the annotated Simulink® block design with high-level quality if needed.

Table 4: Mode Enumeration and Actuators Criterion identified for firing phase testing.

MODE		Actuator			
#	**Enumeration**	**Fuel Pump**	**Regulation Valve**	**Recirculation Valve**	**Fuel Run Solenoid**
0	Off	0% Off	0% (Open)	0% (Open)	0% Off
1	Ignition On	Priming Pulse	0% (Open)	0% (Open)	On if Fuel Pump > 0%
2	Engine Starting	100%	PID controlled output	100% (Closed)	On if Fuel Pump > 0%
3	Running	PID Controlled Output Target Fuel Pressure	100% (Closed)	100% (Closed)	On if Fuel Pump > 0%
4	Running Low Flow	Look Up table (25%-100% DC)	PID Controlled Output	100% (Closed)	On if Fuel Pump > 0%
5	Fault/Error	0% Off	0% (Open)	0% (Open)	0% Off

4.6 Pump Recirculation Valve Control

During the high-pressure DI pump temperature and performance sweep testing in the non-firing phase, the recirculation and regulation valves (one of the anti-vapor lock solutions developed by Katech) were non-operational. In other words, the regulation and recirculation valves were closed during those tests. Due to this, the Stanadyne pump failed to achieve a steady flow rate at idle and low throttle conditions below 1000 RPM. However, during the firing phase, the regulation valve was fully operational, and as a result of this, the fuel system achieved a steady state even at the low throttle conditions. The steady state of the fuel system is evident from the steady engine power and torque characteristics achieved during the durability testing in the firing phase. The recirculation valve, mounted in the injector rail, was eliminated in the firing phase testing.

5. Testing Procedures

This section provides detailed information on testing and benchmarking procedures considered in the non-firing and firing test phases, along with the propane fuel composition grade.

5.1 Propane Grade

Propane or LPG, refined or processed in the US, is sold in three different grades, namely HD5, HD10, and commercial grade [7]. All these grades differ from each other in terms of consistency even though they are processed from the same raw materials (crude oil and natural gas). In addition, all three grades are used for different purposes. For injector, pump, and engine testing, the Katech team used propane complying with the HD5 grade standards. The number “5” in the HD5 refers to the maximum percentage of propylene used in the fuel blend along with neat propane. For example, HD5 propane can be either 100% neat propane or 95% propane & 5% propylene or 95% propane & 2% propylene & 3% of other constituents (butane, iso-butane, ethane, etc.).

The propane fuel composition was tested by Paragon Laboratories Inc. in Livonia, Michigan according to the following standards:

- ASTM D2163-14 Standard Test Method for Determination of Hydrocarbons in Liquefied Petroleum (LP) Gases and Propane/Propene Mixtures by Gas Chromatography.
- ASTM D6667-21 Standard Test Method for Determination of Total Volatile Sulfur in Gaseous Hydrocarbons and Liquefied Petroleum Gases by Ultraviolet Fluorescence
- ASTM D2158-21 Standard Test Method for Residues in Liquefied Petroleum (LP) Gases
- ASTM D2598 Standard Test Method for Liquefied Petroleum Physical Properties.

5.2 Non-firing Phase Procedure

This section provides detailed information on testing and benchmarking standards considered for independent fuel system components as well as for the entire fuel system assembly.

5.2.1 High-pressure DI Pump Characterization

To verify the suitability of the high-side DI fuel pump, the team followed the SAE J2714 standards. All the instrumentation and test cell conditions were considered according to these SAE standards. During the non-firing, the team performed the following tests:

- Static measurements
- Dynamic flow test
- Temperature & pressure sweeps
- Durability testing
- Visual wear

In addition to the tests specified in SAE J2714, additional tests relevant to propane were also conducted on the pump. The tests shown above are only functional and durability tests. Integrity tests like sea salt spray, environmental tests, mechanical shock, vibration, proof, and burst pressure tests are out of the scope of the study. Each of these tests and their procedures are explained in detail below.

Static measurements: In this test, all the geometrical and dimensional properties of the pump were measured. The geometric and dimensional properties include control resistance, overall length, width, height, mass, pump bore diameter, bore surface finish, pump piston diameter, piston surface finish, piston stroke, pump displacement, spring rates, etc. These measurements were used as a basis for obtaining the results related to the dynamic flow tests.

Dynamic flow tests: In this test, the pump's flow rate and volumetric efficiencies were quantified at different engine speeds and for a given inlet and outlet condition. The inlet pressure to the DI pump was maintained at 10 bar +/-1 bar (depending upon tank vapor pressure), whereas the outlet pressure of the DI pump was maintained at 100 bar. The fluid temperature (i.e., propane) was maintained between 20°C to 23°C on the suction side. The dynamic tests helped the team to understand the pump's performance at different speeds as well as the flow characteristics deviation when compared to gasoline. In addition, the dynamic tests also helped in comparing the performance deviation between different fuel pumps as well as flow deviations during durability tests.

Temperature & Pressure Sweeps: Depending upon propane's pressure and temperature in the fuel lines, it exists as a two-phase mixture (both liquid and vapor). Especially at low loads and idle conditions, propane evaporates prior to the high-pressure fuel pump due to low injection pressures. In addition, propane may change its state to supercritical (especially during hot soak and hot start conditions) due to low critical temperature. Since the high-pressure DI pumps were designed to circulate liquid, they fail when a fluid is either two-phase or vapor. Considering this issue, temperature and pressure sweep tests were carried out to understand the flow rate, deviations, and performance of the pump at different fuel conditions. Furthermore, these tests will also help the team understand the impact of cavitation on the pump operation.

Durability tests: This durability test evaluated the variation in performance parameters of the pump with an increasing number of operating cycles. The durability test interval can be defined as either one billion pump cycles, years, hours, or a number of miles/kilometers. In this project, each pump was tested for 25 hrs., at 3600 RPM or 8.1 million pump cycles (3 pump cycles per 2 engine revolutions) in the non-firing phase. The performance and flow deviations of the pump for three different speeds were obtained for every 5hours of testing. The flow deviation or shift is obtained using the following formula:

(New flow rate - Baseline Flow rate) / (Baseline Flow rate)$Flow\ Shift\ (\%)\ = \left(\frac{End\ of\ Test\ Q_p -\ Start\ of\ Test\ Q_p}{Start\ of\ Test\ Q_p}\right)\ X\ 100$

GDI pumps were designed for wet operation. They depend upon the fuel for cooling and lubrication. Also, the pumps were not intended for two-phase mixtures or gaseous fuel. However, either during the assembly, and build phase, to simulate a start-up test immediately after component modification/assembly, or due to fuel properties of propane, the DI pumps were operated with either a two-phase mixture or gaseous fluid. Furthermore, the cavitation (presence of air/vapor in the liquid line) impacts the performance and longevity of the pump. Dry run tests are usually performed on the DI pumps to understand these deviations. The dry run refers to the test condition where the fluid environment has residual fluid from the previous run but does not receive incoming fuel flow at the inlet from the tank-submerged low-side fuel pump. Dry run tests were not performed on the pumps, and they were out of scope for this project.

Visual Wear: The premature wear and tear of fuel system components is also an issue due to the poor lubrication characteristics of propane. The detailed wear and tear of the high-pressure DI pump are out of the scope of the project. However, photographs of different DI fuel pump components were taken during the durability testing procedure to identify & analyze the wear and tear of seals and other subcomponents.

5.2.2 DI Injector Characterization

Similar to the pump characterization tests, the SAE standards (SAE J2713) were used in this project to verify the suitability of the down-selected DI injectors. The following tests were performed on the injectors:

- Static measurements
- Pulse width variation & linearity check
- Temperature & pressure sweeps
- Durability testing
- Visual wear

In addition to the tests specified in SAE J2713, additional tests relevant to propane were also conducted on the injectors. Furthermore, similar to the pumps, only functional and durability tests were performed. Integrity tests like sea salt spray, environmental tests, mechanical shock, vibration, proof, and burst pressure tests are out of the scope of the study.

Static measurements: Similar to the pump static measurements, all the geometrical and dimensional properties related to the injectors were measured in this case. The geometric and dimensional properties include control resistance, overall length, width, height, mass, tip diameter, number of seals, connector style, number of injection holes, injection angle and type, and rated leakage.

Pulse width variation & linearity check: In this phase, the flow rate of the propane from the injector tip was quantified for different injection pulse widths and for a given inlet pressure and period. The pressure in the fuel rail was maintained at 100 bar, whereas the period for the injector was maintained at 33.3 ms (3600 RPM, Injector firing once per 2 engine revolutions). The fluid temperature (i.e., propane) was maintained at 30° C (+/- 1° C). These dynamic tests helped the team to understand the deviation in the fuel mass flow rates at different injection pulse widths. In addition, these dynamic tests also helped in comparing the injector flow deviation from the linear regression fit. Finally, the data from this test was used in understanding the linear flow range and working flow range in both static and dynamic conditions.

Temperature & Pressure Sweeps: Due to high temperatures and pressures of the injector rails & injector bodies, propane may change its state to a supercritical state (or two-phase mixture) under normal engine operating conditions. During this time, the injectors may fail to deliver the required amount of fuel to the engine. Considering that, the temperature and pressure sweep tests were performed on the injectors to understand the flow rate deviations and performance at different pulse widths and different fuel conditions.

Durability tests: This durability test evaluated the variation in the fuel flow rate of an injector at a given pulse width and period for an increasing number of operating cycles. The durability test interval can be defined as either one billion injector cycles, years, hours, or number of miles/kilometers. In this project, each injector was tested for 25 hrs., at 3600RPM crank speed or 2.7 million injector cycles in the non-firing phase. The performance and flow deviations of the injectors were obtained for every 5 hours of testing. The same formula mentioned in the High-pressure DI Pump characterization section was used for calculating the flow shift.

Visual Wear: Similar to the pump, the injector subcomponents and seals experience potential premature wear and tear due to poor lubrication characteristics of propane. So, the photographs of injector subcomponents were taken during the durability testing procedure to identify & analyze the wear and tear. Again, any results/conclusions from the visual-based studies are at the basic level and the detailed investigation is out of this project's scope.

5.3 Firing Phase Test Procedures

This section provides detailed information on testing, benchmarking, and performance standards considered for the integrated DI fuel system, as well as for the engine.

5.3.1 Engine Startup & Control System Calibration Sequence

This section describes the engine control system calibration and test sequence right from the moment the engine is installed on the engine dynamometer. Following is the step-by-step procedure for control calibration and test sequence:

- Initial Control Setup Evaluation & Startup.
- Engine Break-in Procedure.
- Post Engine Break-in Calibration Sequence
    - Engine Start Calibration & Testing
    - Steady State Calibration & Testing
    - Transient Calibration & Testing
    - Acceleration Calibration & Testing

All these procedures are explained in sufficient detail below.

1. ***Initial Control Setup Evaluation & Startup:*** Following the physical installation of the engine onto the test platform, a control setup evaluation was performed to ensure the accuracy of all engine control inputs. This procedure involved an inspection of all sensor scaling, potentiometer sweeps, servo configuration, engine reference, sync correlation, etc. This was done to ensure the validity of all engine sensors/actuators.

2. ***Engine Break-in:*** Immediately after the control setup evaluation, a base calibration was generated, and the engine break-in process began. The primary objective during initial startup and engine break-in was to place the modified UUT under load to aid in piston ring sealing. Calibration refinement was completed following the successful completion of this stage. The engine shake-down or break-in procedure consisted of a two-stage, in-house developed process. It is a mandatory process for all new engines developed and tested at Katech. The two stages of the break-in process are provided below:

    - *Stage I:* The objective of this stage is to monitor engine vitals. The modified *UUT undergoes* test cycles with varying loads (in the range of 10% - 40% of peak load) and engine speed (in-between 2500 RPM – 4500 RPM). The load and engine speed were varied proportionally with respect to oil temperature to eliminate potential wear pattern generation. Once the engine reached the desired operating temperature, data was recorded at 3 different engine speeds/load points. This data was used as a baseline for the second stage process.

- *Stage II:* This stage is performed to identify any loss of cylinder pressure. It also evaluates real-time engine performance. The loading process in this stage was similar to the first stage of engine break-in. During this stage, the load applied to the engine was varied proportionally to the 3 engine speed intervals as shown below. The engine was cycled between these points at the programmed rate and repeated twice to evaluate/monitor any degradation in engine performance.

3200 rpm	4000 rpm	4800 rpm
30% of estimated peak torque	40% of estimated peak torque	50 % of estimated peak torque

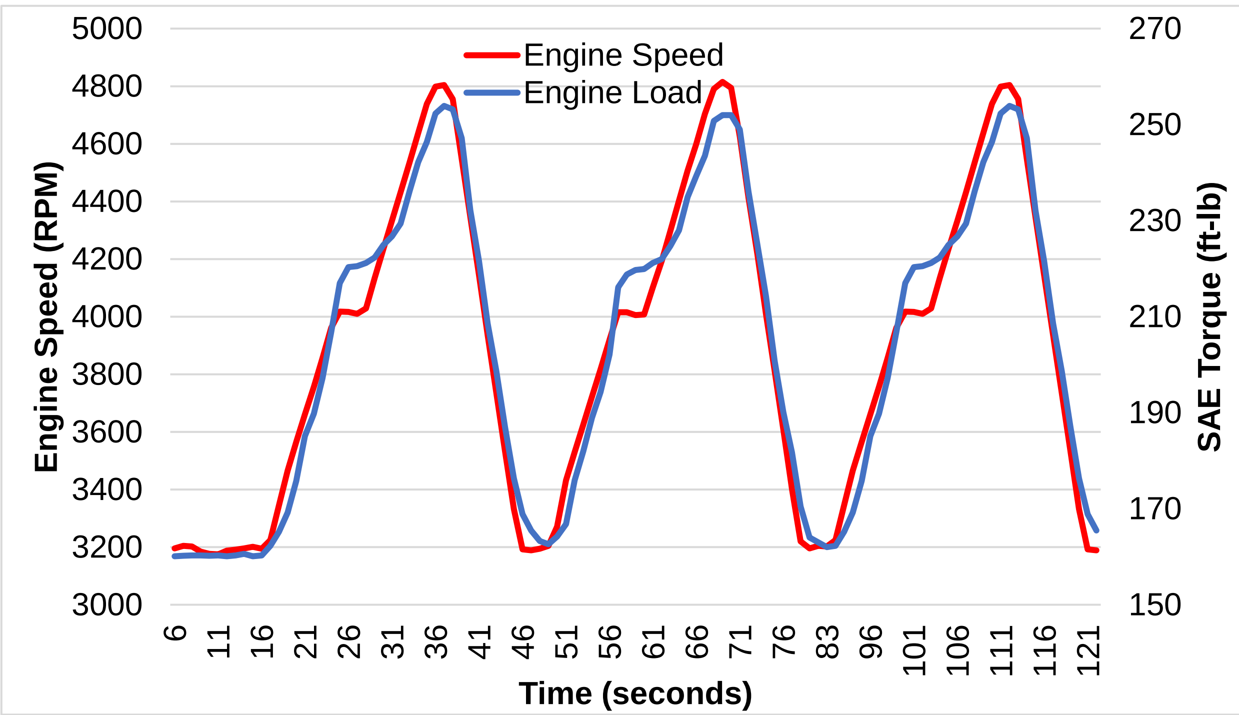

Figure 11: Stage II Engine Break-In procedure during engine shake down process.

3. ***Post Engine Break-In Calibration Sequence:***
   Following the completion of the engine break-in process, further engine calibration was performed to allow for proper engine output and health assessment. The standard calibration process has four stages. Each of these stages is explained below.

- ***Engine Start Calibration & Testing:*** The previously generated engine start parameters were refined and validated to ensure consistent start characteristics. The modified UUT was repeatedly subjected to hot/cold start situations, with the team executing calibration adjustments throughout the process to ensure repeatable results.

- ***Steady State Calibration & Testing:*** This calibration method was performed to reduce errors induced by the engine operating at different conditions. During this process, the modified UUT was held at the designed operating temperature and constant engine speed. UUT load was varied to achieve breakpoints assigned in the engine control unit calibration structure. This process assisted the team in obtaining accurate base fuel and spark ignition advance adjustment, as well as refining the variable valve timing, injector, and fuel pump calibration tables. After finishing the calibration for one breakpoint, the process was repeated for all the achievable engine running conditions. Following the completion of the base engine control calibration, the operating temperature swings were induced into the modified UUT, within the limits of the test cell equipment, and the calibration procedure was repeated and validated for all the breakpoints.

- ***Transient Calibration & Testing:*** Following the completion of steady-state calibration, further calibration was performed on the modified UUT for transient engine conditions. This was done to mitigate errors during engine transitions from different operation speeds and throttle positions. This process allowed the team to evaluate the engine operation through a series of varying throttle opening rate conditions at both steady state and acceleration conditions. Successful completion of this calibration resulted in stable fuel delivery & power even in extreme conditions. However, it should be noted that the transient calibration in a controlled test cell environment might differ from the results obtained through real-world application testing.

- ***Acceleration Calibration & Testing:*** Finally, the last step in the calibration process was acceleration testing. This testing was done to evaluate the torque and power output of the engine through a controlled engine speed acceleration test. The acceleration test was performed between engine speeds of 2000RPM to 5000RPM at an acceleration and deceleration rate of 150RPM/sec.

For all the engine power and torque calculations, SAE correction factors mentioned in SAE J1349 standards were applied. Unless otherwise mentioned, the power reported in any figures/tables/graphs is the SAE corrected power adjusted for ambient conditions and engine friction relative to engine displacement.

5.3.2 Baseline testing with Gasoline

In this phase, initially, the team followed the calibration process mentioned in the previous section with 87AKI gasoline to establish the baseline power and torque characteristics of the GM L8T engine with the OEM DI fuel system components. These baseline power and torque characteristics will help the team in choosing the right DI components for liquid propane in order to maintain the same performance. Furthermore, during the gasoline baseline testing phase, the engine control unit calibration was only populated for the speeds/conditions required for the assessment of reliable health and power output.

5.3.3 Performance mapping with Propane

After the baseline establishment, the final DI fuel system selected based on the decision matrix was installed on the GM L8T engine. Unlike the gasoline testing phase, the team followed the detailed calibration process for all the speeds/loads/throttle positions both at steady state and transient conditions. Upon detailed calibration, the torque and power trends were obtained and then compared with the gasoline operation. Also, prior to the liquid propane DI operation, the team implemented vapor lock inhibitor technologies mentioned in the previous sections both at hardware and software levels. As mentioned before, the PRV mounted in the injector rail was eliminated during the firing phase as the high-side DI pump with a regulating valve was capable enough to deliver the propane in the liquid state for the injector rail even at low loads.

5.3.4 Durability Cycle Creation & Testing

To test the feasibility of the proposed vapor lock inhibitor technologies and the suitability of the proposed DI fuel system components., Katech team designed a 30-minute durability cycle encompassing all the possible steady-state and transient running conditions. This 30-minute cycle was repeated 500 times in order to accomplish 250 hrs. of durability testing. In the 30-minute cycle, there were 8 distinct stages with each representing a different form of engine operation. There was a scheduled heat soak period in between every 3 tests (i.e., after 1.5 hrs. of continuous operation) to mimic engine shutoff conditions. During this time, the engine remained off for approximately 5 to 10 minutes as the propane fuel tank was refilled.

To encompass all the operating conditions, the team researched and referenced existing test cycles for on-road vehicle applications. These cycles include the US EPA's Federal Test Procedure's (FTP) Urban Dynamometer Driving Schedule (UDDS), the US EPA's Ramped Modal Cycle (RMC), CARB's Low Load Cycle (LLC), the EPA's US06, and the EPA's 55mph and 65mph cruise cycles [8, 9, 10, 11]. Although these existing test cycles provided helpful direction in the scripting of the LPG durability cycle, the actual 30-minute durability cycle, used in this project, was varied from those existing protocols. The reasoning behind these differences was due to limited access to OEM engine speed and torque data. These test cycles referenced vehicle speed and time as the common metrics. Table 5 and Figure 12 below show the 8 stages of the 30-minute durability cycle used in this project.

Table 5: Durability Test Cycle Description and Similarities

	Description	Test Similarity
1	Start & End of Test SS Health Check	Health Check
2	Stop/Start	FTP NYNF
3	Speed/Load Changes Few Stops	FTP LANF/US06
4	Load Changes Near Constant Engine Speed	FTP LAFY/US06
5	55 MPH	55 MPH Cruise
6	65 MPH	65 MPH Cruise
7	Steady State Modes	RMC/Health Check
8	Low Load	LLC

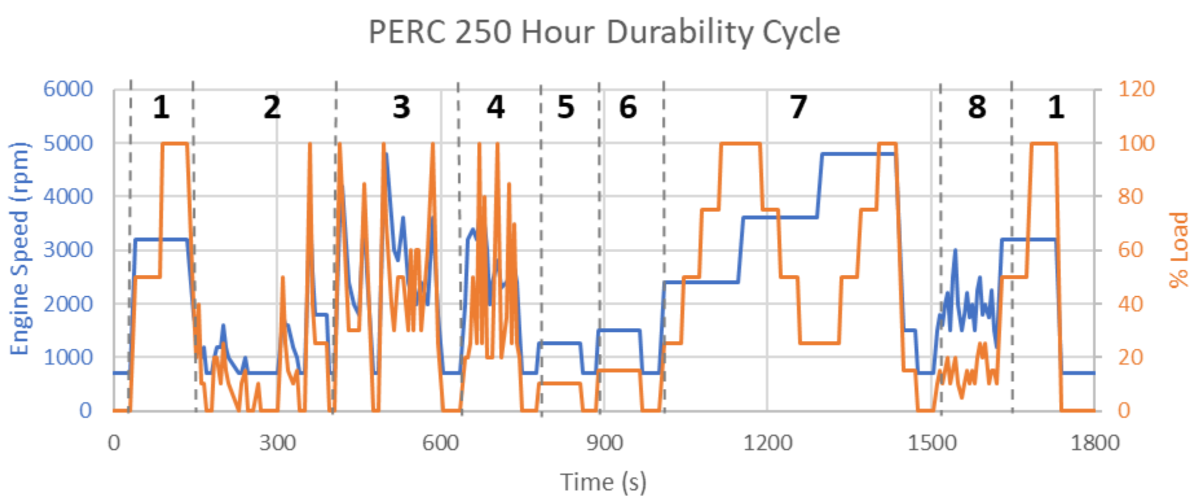

Figure 12: Modes in 30-min Durability cycle designed for 250 hrs. of durability testing.

The main requirement of the durability testing is to understand the steady state operation of the fuel system when subjected to transient conditions in between. The transient stages programmed in the durability cycle introduced the wear by mimicking on-road driving operation and conditions, multiple start stops, etc.

6. Results & Discussion

6.1 Propane Grade

The propane fuel used for testing was tested at Paragon Laboratories Inc. in Livonia, Michigan according to ASTM standards. The results obtained from these tests are presented below in Table 6.

Table 6: Laboratory Analysis of Commercial Propane tested according to ASTM Standards

Parameter	Result	Unit
Calculated Physical Properties by ASTM D2598		
LPG Vapor Pressure at 37.8 °C	1281	kPa (g)
LPG Vapor Pressure at 100 °F	186	psi (g)
Rel. Density at 15.6 °C	0.504	
Motor Octane Number	97.0	
Elemental Analysis by ASTM D6667		
Sulfur	23.0	Ppm
Individual Parameters by ASTM D2158		
Residue on Evaporation	<0.05	mL
Oil Stain Observation	0.1	mL
Extraneous Matter	No	
LPG Hydrocarbons (HCs) by ASTM D2163		
Ethane	2.86	%v/v
Propane	95.26	%v/v
Propene	0.51	%v/v
Propyne	<0.01	%v/v
Butanes	1.34	%v/v
Other	<0.03	%v/v

6.2 Non-Firing Phase Results

In the non-firing phase, three high-side fuel pumps and three fuel injectors were chosen for the testing process. Out of three high-side fuel pumps, two of them are off-the-shelf, commercially available components (Stanadyne 1250 - 200, Bosch HDP6 - 350), while the other is a prototype development pump for propane (Stanadyne 350) from Stanadyne Inc. Similarly, out of three DI fuel injectors, two of them are off-the-shelf commercially available components (Delphi 200, Bosch 350), while the other is a custom development injector for propane (Stanadyne 350). During the non-firing phase, the team performed the feasibility and performance tests individually on three high-side DI pumps and three fuel injectors. Based on the independent testing, a total of 9 combinations were available. All these pumps/injectors were rated against different metrics using a decision matrix and the pump and injector with the highest rating were chosen for engine testing in the firing phase.

6.2.1 High-side DI pump testing

In this section, the results from the individual pump testing were compiled & compared together in order to draw comparisons. This section is organized into four subsections:

1. Static Measurements – Compares the geometrical, surface finish, and design parameters of the pump.
2. Dynamic testing – Compares the volumetric efficiency and mass flow rate variations at a given temperature, pressure, and engine speed.
3. Temperature & Pressure Sweeps – Compares the volumetric and mass flow rate variations at different fuel temperatures and pressures.
4. Durability testing – Compares the mass flow rate deviations at a given temperature, pressure, and engine speed over a period of time.

Static measurement results: The results pertaining to the static measurements of three DI fuel pumps are provided in Table 7 below. As shown below, the team captured all the static measurements (resistance, temperature, geometrical dimensions, mass, surface finish, bore & piston diameters, spring rate, etc.) using different instruments available at Katech's test facility. Of major importance was the modification of the off-the-shelf Stanadyne SP1250 and Bosch HDP6 pumps for initial testing. As mentioned in the additional information tab, the current design eliminated the flow-dampening valves in the off-the-shelf pumps to offer the integration of an SAE ORB - 4 fitting for the fuel inlet and an SAE ORB – 4 fitting for the low-pressure outlet/recirculation flow path. However, in the custom development pump designed for LPG by Stanadyne, the flow dampening valve was available along with propane-rated fuel interfaces (i.e., JIC/SAE fittings).

Table 7: Static measurements for all High-pressure DI pumps

Data Reporting Sheet for LPG-DI Pump Static Measurements			
Part 1: General Test Logistics			
Test Name or Log	Stanadyne 200 Baseline	Bosch 350 Baseline	Stanadyne 350 Baseline*
Operator Name & location	Eric S., & Derek P., Katech Engineering, Clinton Twp, MI		
File Name or Data Archive	SP1250-200 Benchmark	0261520587 Benchmark	SP Custom – 350 Benchmark
Additional Information	*Custom Development pump from Stanadyne Inc. (Tier – I OEM for DI fuel systems).		
Part 2: Information on Pump			
Manufacturer	Stanadyne	Bosch	Stanadyne
Description	Stanadyne SP1250-200	Bosch 0261520587	350 Bar LPG Prototype Fuel Pump
Part Number	GMP-12697966	GMP-12668802	14-2281

Serial Number	14-9309 L0102J1	11480116426	30402A1
Displacement V_D	0.494cc /stroke (1.481cc/cam rev)	0.447cc /stroke (1.342cc/cam rev)	0.494cc /stroke (1.481cc/cam rev)
Part 3: Measurements Electrical			
Control Valve Resistance (Ohms)	0.55	0.8	0.2
Ambient Temperature (°C)	21.5	21.5	21.5
Part 4: Measurements Mechanical			
Overall Length (mm)	58.90	97.20	106.00
Overall Width (mm)	111.70	120.00	112.00
Overall Height (mm)	155.40	155.40	114.00
Mass of Pump Assembly (grams)	1520	826	1260
Bore Diameter (mm, 4 measurements)	10.508mm, 10.508mm, 10.508mm, 10.508mm 0.4137in, 0.4137in, 0.4137in, 0.4137in	9.9992mm, 9.9992mm, 9.9992mm, 9.9992mm 0.3937in, 0.3937in, 0.3937in, 0.3937in	N/A Stanadyne Confidential
Bore Surface Finish (Ra mm) Avg. of 5 measurements	0.0395 Ra mm	0.0744 Ra mm	N/A Stanadyne Confidential
Bore Surface Finish (Rz mm) Avg. of 5 measurements	0.3230 Rz mm	0.7390 Rz mm	N/A Stanadyne Confidential
Piston Diameter (mm, 4 measurements)	10.500mm, 10.500mm, 10.500mm, 10.500mm 0.4134in, 0.4134in, 0.4134in,0.4134in	9.987mm, 9.987mm, 9.990mm, 9.987mm 0.3932in, 0.3932in, 0.3933in,0.3932in	N/A Stanadyne Confidential
Piston Diameter Min Clearance	0.0076mm, 0.0003in	0.0092mm, 0.000362in	N/A Stanadyne Confidential
Piston Diameter Max Clearance	0.0076mm, 0.0003in	0.0122mm, 0.000480in	N/A Stanadyne Confidential
Piston Surface Finish (RA mm) Avg. of 5 measurements	0.0480 Ra mm	0.0920 Ra mm	N/A Stanadyne Confidential
Piston Surface Finish (Rz mm) Avg. of 5 measurements	0.3810 Rz mm	0.8372 Rz mm	N/A Stanadyne Confidential
Piston Mass (grams)	32	20	N/A Stanadyne Confidential
Designed Pump Stroke (mm)	5.7	5.7	6.0
Max Stroke (mm)	8.9	8.9	N/A Stanadyne Confidential
Spring Rate Inner (Kg/mm)	1.25	2.857	N/A Stanadyne Confidential
Spring Rate Outer (Kg/mm)	5.715	NA	N/A Stanadyne Confidential

Additional Information	Pump modified for flow through design. The current design eliminates flow dampening valves. Options to correct this are being explored.	Pump modified for flow through design. The current design eliminates flow dampening valves. Options to correct this are being explored.	Pump specifically developed for Propane by the OEM Supplier according to Katech team's requirements
This is the **Basic Pump Static Test Results**. There Are No Test Deviations.			

The reason for installing the SAE ORB fittings is two-fold. First, the factory 3/8 SAE fuel quick connect is rated for gasoline fuel at 0-5 bar (g). With propane operating upward of 20 bar (g), it was not feasible to test with the factory 3/8 quick connect fitting. The second reason is that the pumps only have one inlet and one high-pressure outlet port, in their original configuration. However, testing the pumps requires three ports. In other words, the proposed fuel pump design requires one inlet, one low-pressure outlet for the regulation valve, and one high-pressure outlet. The images of modified GM's Stanadyne SP1250, modified Bosch HDP6, and Stanadyne custom development pump are shown in Figure 13.

Based on the static measurements presented in Table 7, the following conclusions can be made regarding the geometrical, surface finish, and design parameters:

- Bosch HDP6 and Stanadyne custom development pumps were designed for 350 bar whereas the Stanadyne pump was rated for 200 bar.

- The Bosch pump weighs 46% lower than the Stanadyne SP1250 pump and 35% lower than the Stanadyne custom development pump (826g for Bosch & 1520g, 1260gm for Stanadyne SP1250 and custom pumps respectively).

- Both the off-the-shelf pumps needed plumbing modifications to add a regulation valve port. The off-the-shelf pumps came with only two ports (one for low-pressure inlet and one for high-pressure outlet) whereas the fuel system design necessitated the need for three ports (one for low-pressure inlet, one for high-pressure outlet, and one for the regulation valve outlet). Of the two pumps, it was hard to modify the Bosch pump when compared to the Stanadyne SP1250 pump. In addition, due to laser welding and machining, the Bosch pump was more prone to contaminants when compared to the Stanadyne SP1250 pump. The Stanadyne's custom development pump had the three ports designed into the body by the OEM based on Katech's requirements.

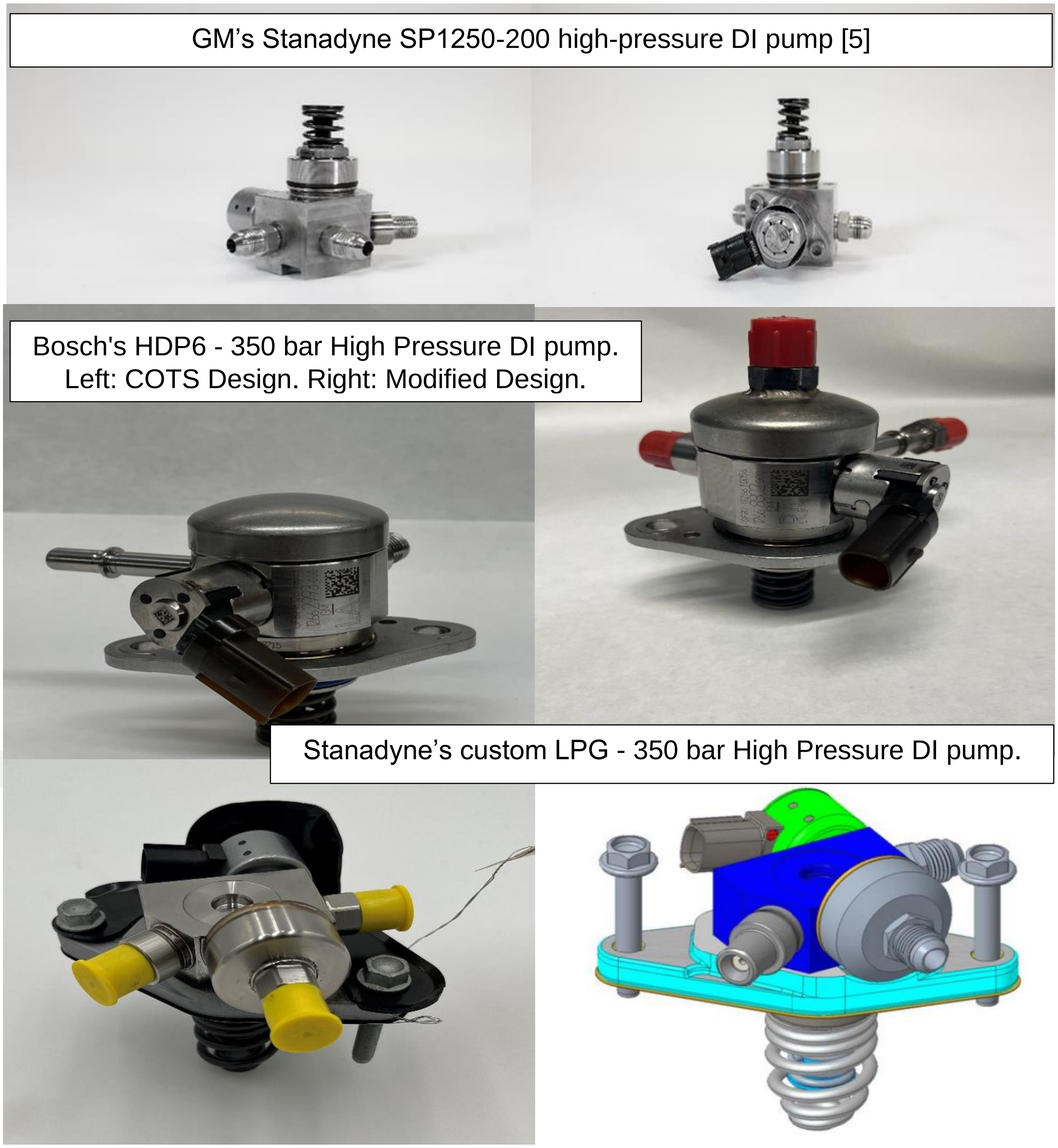

Fig 13: Images of High-side DI pumps selected for this program. Flow design was modified for Stanadyne development pump.

- The displacement volume of the Bosch pump is 10.1% (0.447 cc/stroke vs 0.494 cc/stroke for both Stanadyne pumps) smaller than the Stanadyne pump for the same stroke. The reduction in the displacement volume was mainly due to the reduction in the cylinder bore diameter (10.0 mm for Bosch Vs 10.5 mm for Stanadyne pump). Stanadyne also offers a high-pressure DI pump with an 11.5 mm bore and Bosch also offers a pump with an 8 mm bore, but these alternatives were not tested during this program. As mentioned previously, the off-the-shelf pumps, when operated with propane, experienced localized cavitation due to heat in the cylinder chamber. This problem of localized cavitation can be minimized by optimizing the bore-to-stroke ratio of the pump in the future depending on the results.

- The average mean roughness and the mean roughness depth of the Bosch bore were almost twice the average mean roughness and mean roughness depth of the Stanadyne SP 1250 pump. It is unknown currently what effect surface finish and ratio or rpk and rvk have on piston lubrication and maximum operating pressure. Details of the average mean and mean roughness depth of the custom development pump were not provided to Katech due to confidentiality.

- Bosch runs a single square spring whereas the Stanadyne SP1250 pump runs a dual spring (one conical & 1 straight). The spring stiffness of the single Bosch spring is 2.857 Kg/mm whereas, for Stanadyne SP1250 pump, the spring stiffness is 1.25 Kg/mm for the inner spring and 5.715 Kg/mm for the outer spring.

Apart from the above-mentioned points, the piston clearances, the maximum stroke, the designed pump stroke, and the bounding box dimensions remained almost the same for all the pumps.

Dynamic flow testing results: This section presents the results about the dynamic flow & performance testing of the DI pumps using a series of figures below. The tests were performed at an ambient temperature of 20°C +/- 2°C and at an ambient pressure of 100 kPa +/- 5kPa. The inlet pressure to the high-pressure DI pump (i.e., Tank pressure + In-line pump pressure addition) was maintained at 11 bar +/- 0.2 bar throughout the testing process. The fuel temperature was maintained at 30°C +/- 2°C. Tests were performed at different engine speeds for a constant high-pressure DI pump outlet pressure of 100 bar to understand the variations in pump volumetric efficiency and fuel mass flow rates. These fuel mass flow rates and pump volumetric efficiency variation for the DI pumps are shown in Figures 14 and 15 below.

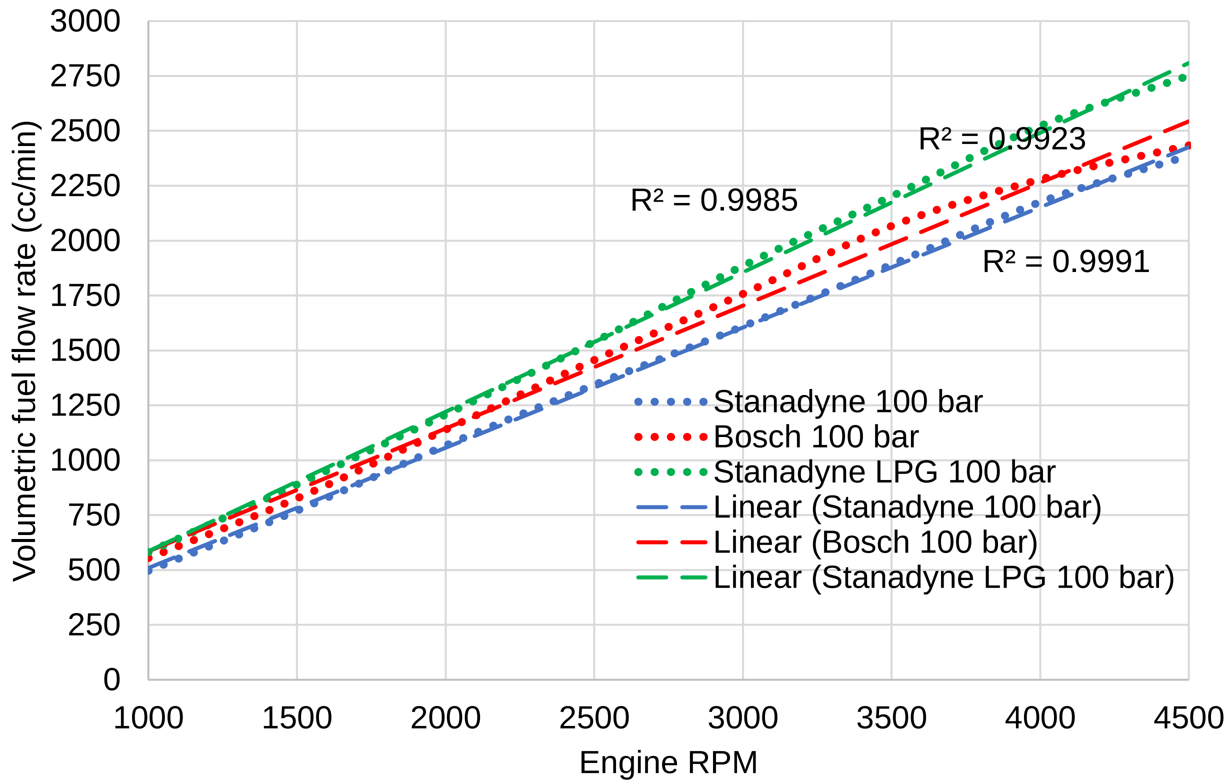

Figure 14: Volumetric fuel flowrate of three DI pumps @100 bar at 30°C fuel temperature.

As shown in Figure 14, the volumetric flow rate (in cc/min) increased almost linearly with the engine. The linear curve fitted to the volumetric flow rate with respect to engine speed has an R^2 value greater than 0.99 for all the pumps considered in this project. Of all the pumps, at the same engine speed, pressure, and temperature, the Stanadyne development pump - 350 bar, has a higher flow rate when compared to the other two pumps. The flow rate of the Bosch falls in between the two Stanadyne pumps. As mentioned in the static measurement results section, the displacement volume of the Bosch pump is 10.1% (0.447 cc/stroke vs 0.494 cc/stroke for both Stanadyne pumps) smaller than the two Stanadyne pumps for the same stroke. When the two 350 bar pumps are considered, Stanadyne has a higher flow rate than Bosch due to higher displacement volume. On the other hand, when Bosch 350 pump is compared with Stanadyne 200 pump, the Bosch 350 bar pump has a higher flow rate than the Stanadyne 200 bar pump due to MSV design and VE differences (see Figure 15).

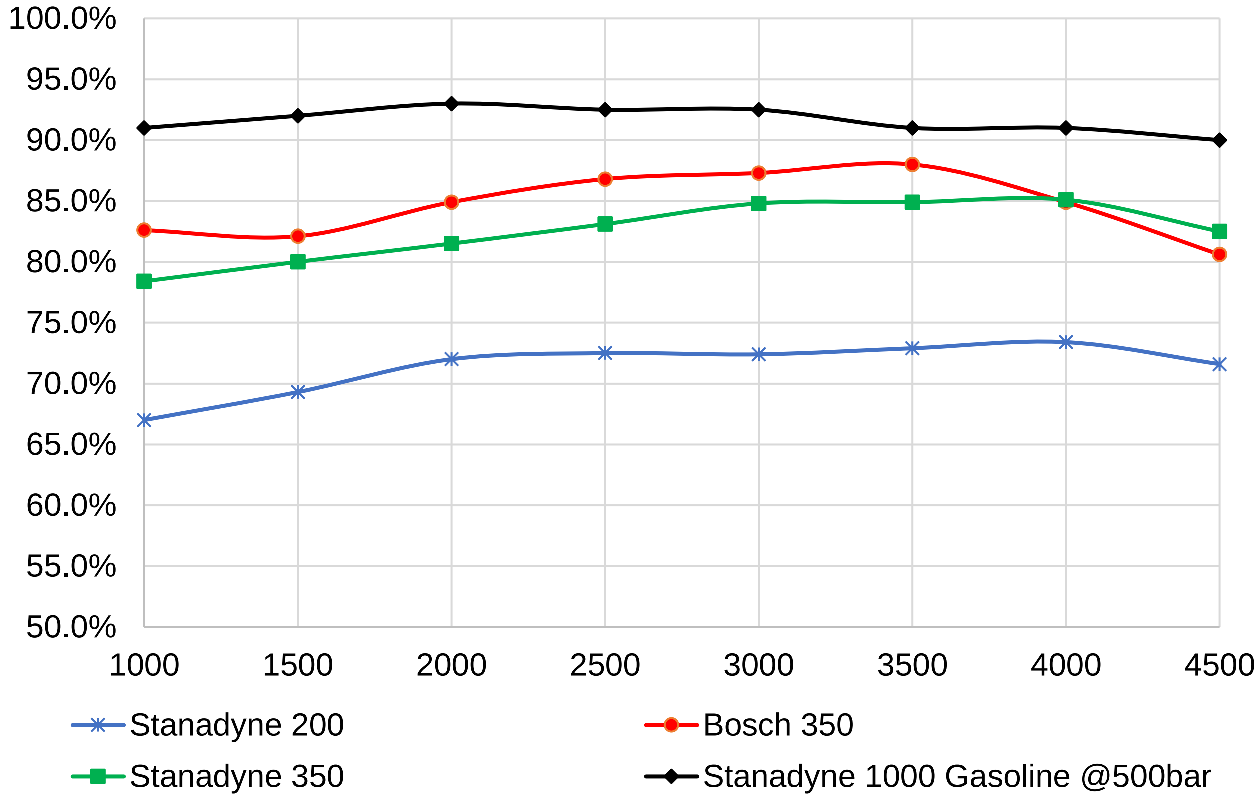

Figure 15: Volumetric efficiency of three DI pumps @100 bar at 30°C fuel temperature compared against volumetric efficiency of gasoline at 500 bar.

Unlike the mass flow rate, the volumetric efficiency, in Figure 15, followed a quadratic trend with respect to engine speed i.e., the VE initially increased with respect to speed and reached its peak valve and then reduced with a further increase in speed in both pumps. Especially for engine speeds between 2000 - 4000 RPM, the volumetric efficiency stayed relatively constant for all the pumps. At speeds below 1000 RPM (i.e., idle or low throttle conditions), the volumetric efficiency was reduced by 5% to 7% (absolute). This is mainly due to two reasons, 1. Low mass flow rate 2. Increased heat transfer due to prolonged duration of fuel in the pump cylinder (high residence times). These two reasons caused the localized cavitation inside the pump cylinder, as a result, a significant amount of fuel was vaporized at those speeds, thereby reducing the volumetric efficiency. On the higher RPM front, the rapid compression and expansion cycles of the pump piston reduced the time to dissipate the heat into ambient. This was very similar to the dominance of the piston velocity and temperature terms in Hohenberg's heat correlation used for estimating the convective heat transfer of an engine cylinder [6]. Due to this domination, the fuel began to evaporate due to localized cavitation, which explains the downward trend (reduction of 2% -3% absolute) in the volumetric efficiency curve.

However, the downward trend on the higher RPM side is much better when compared to the idle or low throttle conditions. In addition to these two reasons, the cavitation on the suction side of the high-pressure DI pump was also responsible for the low volumetric efficiencies encountered in these cases.

The volumetric efficiencies of all the pumps operated on LPG were compared with the volumetric efficiency of the Stanadyne 1000 pump operated with gasoline at 500 bar. The main reason for considering 500 bar gasoline operation with a different pump was due to lack of data. As shown in the figure, the VE of pumps operated on LPG at 100bar were below the VE of gasoline. With an increase in operating pressure, the VE is reduced (see next section for more details). By using these two conclusions, it is safe to assume that the VE of LPG is below the VE of gasoline for the same pump operated at the same pressure and engine speeds. This conclusion, again, explains our previous hypotheses of cavitation/vaporization of propane inside the pump cylinder due to its inherent properties. **To combat this issue, DI propane pumps need to be slightly oversized (or select pumps with high MSV duty cycle) in order to achieve the same fuel flow rate for a given engine operating condition.**

Of all the pumps, the volumetric efficiency of Bosch HDP6 - 350 and Stanadyne custom development pump – 350 were around 85% with Bosch being slightly higher by 2 – 3% when compared to Stanadyne 350 pump. For the Stanadyne SP1200 – 200, the volumetric efficiency was around 72%. One reason for the increase in the volumetric efficiency of the Bosch pump is because of the simple pump cylinder geometric relationships, i.e., the smaller bore-to-stroke ratio has a smaller surface area exposed when compared to the larger bore-to-stroke ratio. The smaller area resulted directly in reducing localized heat which further reduced cavitation. As a result, more liquid was pumped through the Bosch pump when compared to the Stanadyne SP1200 - 200 pump at a given engine speed. This was reflected in the mass flow rate curves of these two pumps.

One hypothesis that might contradict the reasoning above is the surface roughness of the Bosch cylinder. As mentioned in the previous section, the Bosch pump's surface was rougher when compared to the Stanadyne SP1200 – 200 pump. With increased surface roughness, one might presume increased frictional losses (increased heat) when the rest of the geometrical parameters and operating conditions remained constant. While the engine speed, stroke, and ambient temperatures remained the same in both cases, the bore, displacement volume per one cam revolution, and internal conditions (temperature, etc.) were not the same. Considering that, Katech thinks that the gains obtained from reducing the bore overpowered the losses from the surface roughness.

Secondly, the Magnetic Solenoid Valve (MSV) or fuel delivery solenoid of each of these pumps also influences the pump operation and subsequent changes in the VE of each pump. The Bosch HDP6 - 350 bar pump, being developed ~6 years later, likely has improved flow design and allows for a higher VE when compared to the Stanadyne SP1200 – 200 bar pump that was first released in 2014. The Bosch pump also operates at up to 350 bar, compared to Stanadyne SP1200 – 200 which operates at up to 200 bar. When looking at the VE over the percentage of operating range, the VE numbers of both these pumps match much more closely. This is the case with the volumetric efficiency of the Stanadyne custom development pump – 350 bar. The custom Stanadyne pump, being the latest pump design for LPG and having a maximum operating pressure of 350 bar, closely resembles the flow rate and volumetric trend curves of the Bosch HDP6 – 350 pump.

Overall, the Bosch HDP6, Stanadyne 350 pump performed much better at 100 bar for all the engine speeds when compared to the Stanadyne 200 pump. Of the two 350 bar pumps, Stanadyne 350 has a slight advantage due to minimal modifications, OEM technical support, etc.

<u>Temperature & pressure sweeps:</u> While the project team swept the variation in the fuel flow with respect to the engine speed, it is also important to understand the fuel flow variations with respect to different DI pump discharge pressures and different DI pump inlet fuel temperatures. Due to the two-phase mixture properties of the propane, this sweep is of utmost importance when compared to gasoline testing. To avoid redundancy, initially, parametric sweep testing was performed for different engine speeds at six different discharge pressures (50, 75, 100, 125, 150, 175 bar) and three different inlet temperatures (20°C, 30°C, 40°C) for Stanadyne SP1250 pump. Since the same conclusions can be drawn from temperature and pressure parametric sweeps for the other two high-side DI pumps, the parametric sweeps for the other two pumps were performed with the parameter that has the highest impact on the pump performance and volumetric efficiency. The variation in the fuel flow rate with varying outlet pressure and engine speed at a fuel inlet temperature of 20°C for the Stanadyne SP1200 – 200 bar pump is shown in Figure 16.

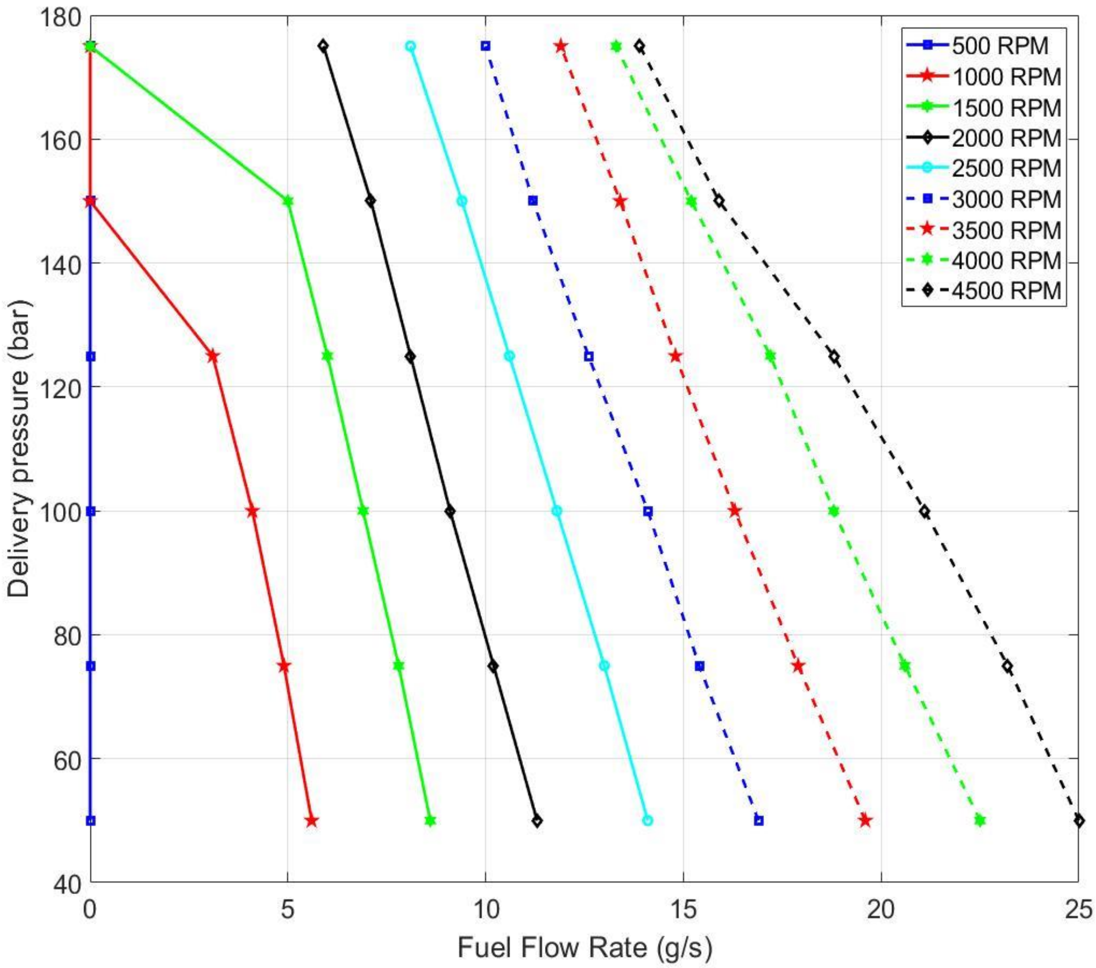

Figure 16: Mass flow rate variation with respect to engine speed and delivery pressure for Stanadyne SP1200 – 200 bar pump.

Similar to Figure 14, the mass flow rate increased with an increase in engine speed at a given delivery pressure. Moreover, as seen shown in Figure 16, the fuel flow rate followed a near-similar vertical trend for a given engine speed as the delivery pressure increased. In other words, the propane mass flow rate, similar to gasoline, has a very low dependency on the delivery pressure when compared to engine speed. Due to similar operation characteristics (not major deviations) as gasoline, the down-selected pumps can be used with propane as a fuel. At the low engine speeds of 500 RPM, the fuel flow rate was unsteady for all the delivery pressures tested. This is again due to propane fuel vaporization at low flow rates. A point to note here is that the recirculation and regulation valves were closed (i.e., the fuel was not flooded at the inlet), and any other vapor lock inhibition techniques, mentioned above, were not utilized for the non-firing testing process. As the engine speed is increased, a steady fuel mass flow rate is achieved at the low & intermediate delivery pressures.

However, the fuel flow rate is still unsteady at 175 bar delivery pressure at engine speeds below 1500 RPM. This is again due to fuel evaporation caused by rapid compression and expansion (increased temperature & heat) inside the pump chamber. The variation in the volumetric efficiency with respect to delivery pressure and engine speed is shown in Figure 17. Unlike the quadratic trend of VE with respect to engine speed, the volumetric efficiency reduced linearly with respect to delivery pressure for the engine speeds under normal operating conditions (1000 – 4500 RPM). The reduced volumetric efficiency at higher delivery pressures is because of three major reasons:

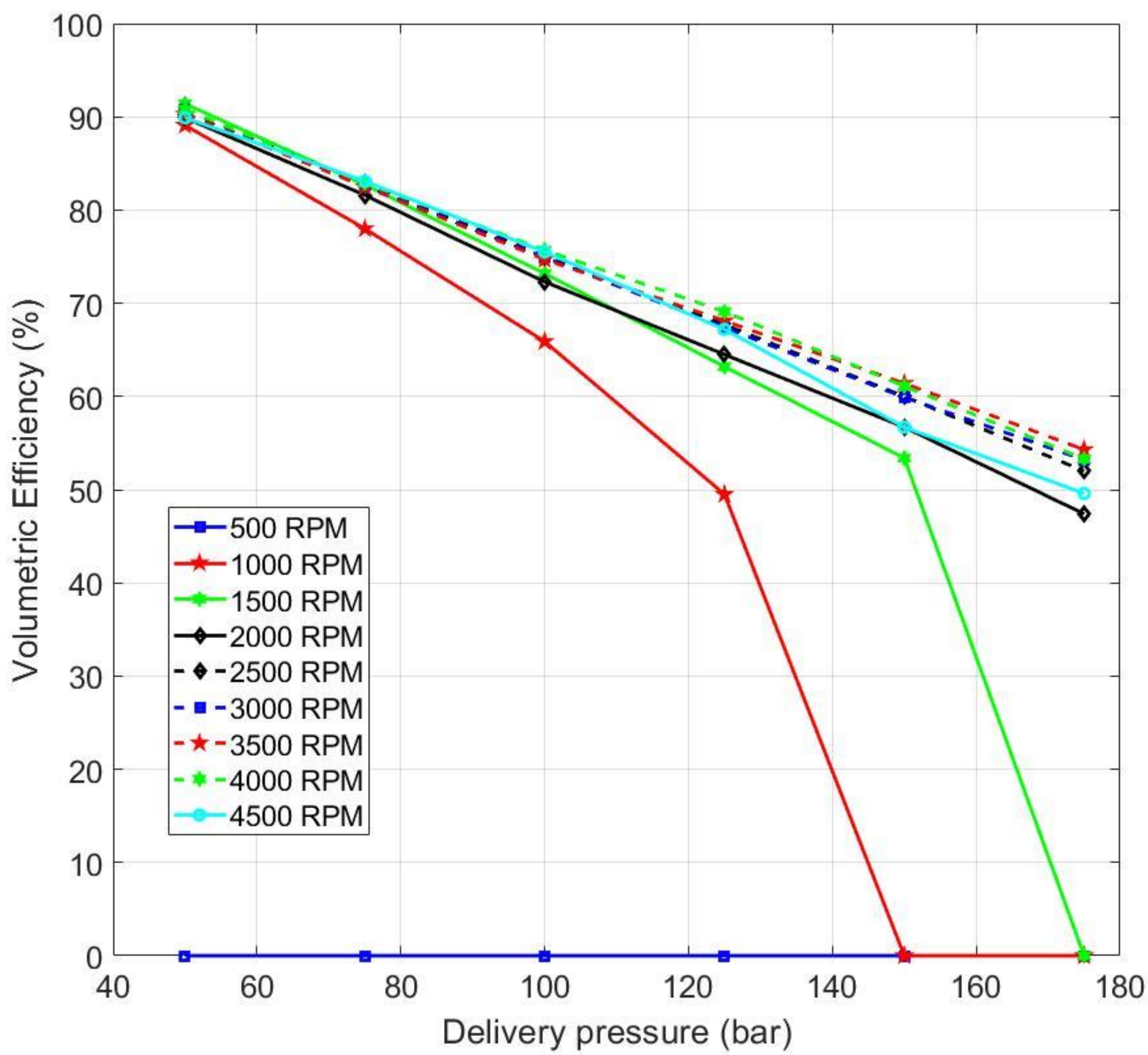

Figure 17: VE variations with respect to engine speed and delivery pressure.

1. Increase in pressure in the pump chamber increased the fuel losses through the pump piston and delivery solenoid.
2. Increase in re-expansion work of the fuel due to increased delivery pressure.
3. Localized cavitation and fuel evaporation due to minimal time for the heat to dissipate from the pump chamber.

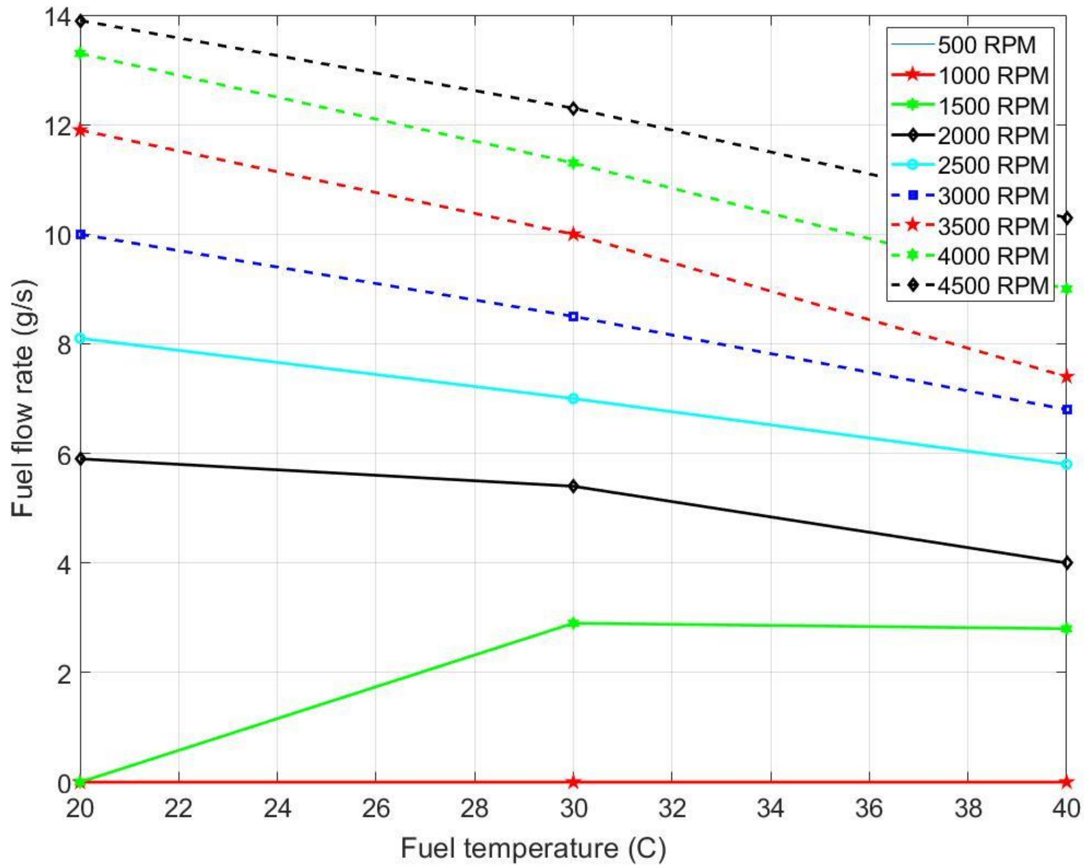

Figure 18: Mass flow rate variations with respect to engine speed and fuel temperature at 175 bar delivery pressure

Figure 18 shows the variations in the fuel mass flow rate with respect to engine speeds and fuel inlet temperatures at a constant delivery pressure of 175 bar. As shown in Figure 18, the fuel flow rate reduced with an increase in the fuel temperature. At engine speeds below 1000 RPM, the fuel flow rate was unsteady. This is mainly due to low fuel flow and evaporation of the liquid fuel into vapor. At an engine speed of 1500 RPM and fuel temperature of 20°C, the flow rate was still unsteady. However, the flow rate was around 2.9 g/s at fuel temperatures of 30°C, and 40°C. For all the engine speeds above 1500

RPM, the fuel flow rate was reduced with respect to the fuel inlet temperature. Fitting the data from different engine speeds (not shown in Figure) with a linear curve obtained almost the same slope of 0.12 - 0.22, whereas the intercept changed from 6 – 16 depending upon the engine speed.

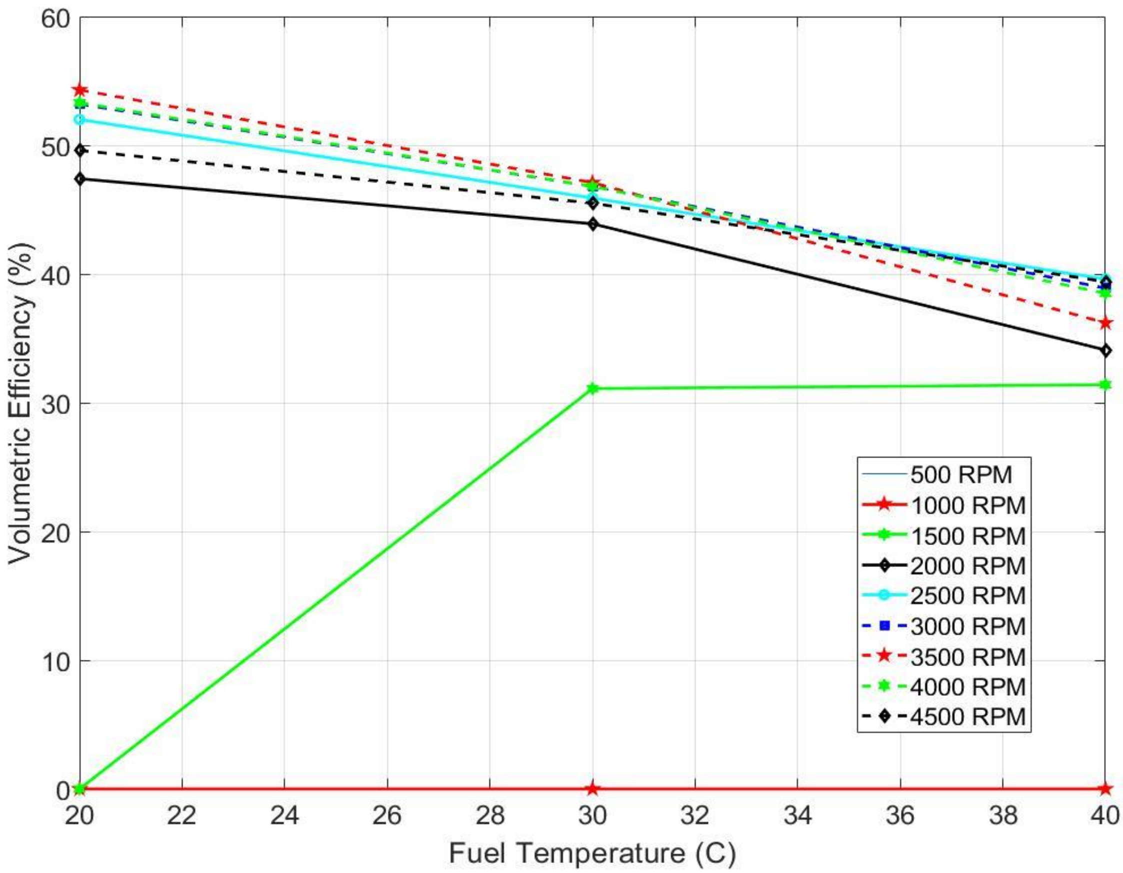

Figure 19: VE variations with respect to engine speed and fuel temperature at a delivery pressure of 175 bar

The variations in the volumetric efficiency with respect to engine speed and fuel temperature at a given delivery pressure of 175 bar are shown in Figure 19. Similar to the pressure sweep, the volumetric efficiency decreased with respect to fuel temperature. At a constant pressure of 175 bar, the VE reduced from around 55% to 40% for most of the engine speeds. However, an interesting trend was observed when the engine speed was maintained constant. The variations in the volumetric efficiency with respect to fuel temperature and delivery pressure at a constant engine speed of 3000 RPM are provided in Figure 20. As shown in Figure 20, at a delivery pressure of 50 bar, the volumetric efficiency is almost constant. In other words, fuel temperature had a very minimal effect on the flow rate & VE at very low delivery pressure. As the pressure is increased to 100 bar, a slight decline in the volumetric efficiency is observed. However, the variation in the volumetric efficiency is still within +/- 2.5%. Increasing the pressure to 175 bar, the variations in volumetric efficiencies increased to +/- 10%.

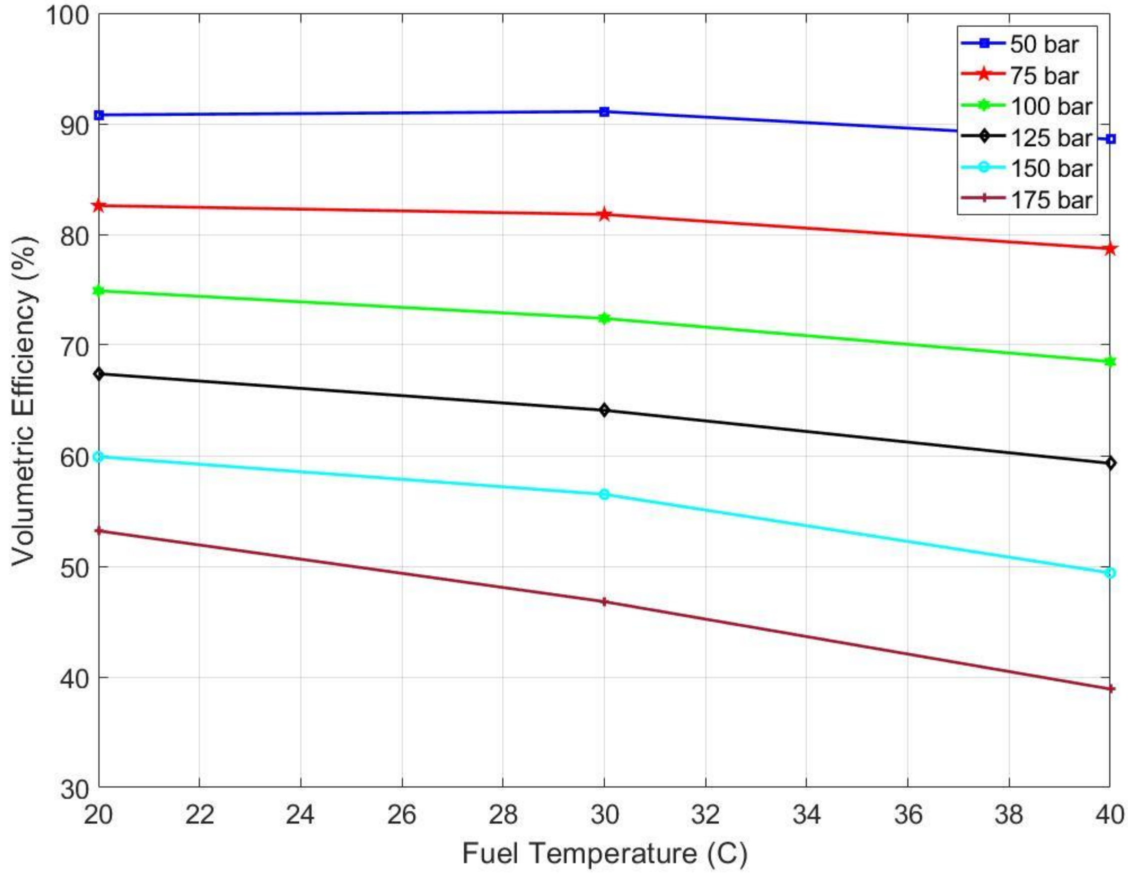

Figure 20: VE variations with respect to engine speed and delivery pressure.

Based on the results presented in this subsection, it is safe to draw the following conclusions from the pressure and temperature sweep testing:

- The delivery pressure had a dominant effect on the fuel flow rate and VE followed by engine speed and fuel temperature.
- The fuel temperature sweep was performed in the narrow range (i.e., 20 - 40°C). So, the results presented pertain to the range mentioned above.
- Increasing the fuel pressure from 175 bar to 250 or 350 bar will undoubtedly take a penalty hit on the volumetric efficiency and pump power.

- Pressure regulation valve operation is needed, especially at low engine speeds (idle or low throttle conditions) for stable fuel delivery or for meeting the engine torque demands.
- Apart from very low and very high speeds, the volumetric efficiency remained almost constant (+/- 2.5% variations) for the majority of the normal operating range (1500 - 3500 RPM).

Since the delivery pressure had a dominant effect on the pump flow rate and volumetric efficiency, the parametric sweeps with respect to delivery pressure were performed for the other two high-side DI pumps (Bosch HDP6 – 350 bar, and Stanadyne Development Pump – 350 bar). Stanadyne SP1200 – 200 bar pump was rated for 200 bar, so the tests were carried out up to 175 bar. The Bosch and Stanadyne development pumps were rated for 350 bar, so the tests were carried out up to 350 bar. Even though these pumps were tested up to 350 bar, the rail pressures were unsteady above 200 bar for most of the speeds with the regulation valve closed. At certain speeds (2000 - 3000 RPM), the pump was capable of maintaining constant rail pressure. However, below 2000 RPM and above 3000 RPM, the pump failed to maintain constant rail pressure at 200 bar due to low fuel flow rate and rapid expansion and compression. As mentioned before, this problem at low & high speeds can be resolved by operating the low-side fuel pump at near or maximum flow rate and controlling the regulation valve (i.e., implementing the vapor lock inhibitor technology designed by Katech). The variation in the volumetric fuel flow rate for all three pumps at three different delivery pressures (50 bar, 100 bar, and 150 bar) is shown in Figure 21.

Similar to Figure 14, the fuel flow rate increased almost linearly with the engine speed for all the pumps at a given delivery pressure. As shown in Figure 21, the mass flow rate reduced with an increase in delivery pressure at a given engine speed. At a low delivery pressure of 50 bar, the Stanadyne pumps (both 200 and 350 bar pumps) had a higher volumetric flow rate when compared to the Bosch HDP6 – 350 bar pumps due to higher displacement volume.

As the delivery pressure is increased from 50 bar to 150 bar, the VE (shown in Figure 22) is reduced at a higher pace for the Stanadyne pumps when compared to the Bosch pump. This faster rate is explained by the broad spacing in the fuel flow rates and volumetric efficiency lines as the rail pressure is increased. As a result, at a high delivery pressure of 150 bar, even with a high displacement volume, the volumetric flow rate of the Stanadyne 350 bar pump is the same as the volumetric flow rate curve of the Bosch 350 pump. The volumetric flow rate for the Stanadyne SP1200 – 200 bar pump is lower than the Bosch 350 bar pump and Stanadyne 350 bar pump. This is again due to poor volumetric efficiency and also due to operating the pump close to its maximum operating pressure of 200 bar. In addition to displacement volume and volumetric efficiency, the MSV design for all these pumps is different and it plays a major role in volumetric flow rate variations.

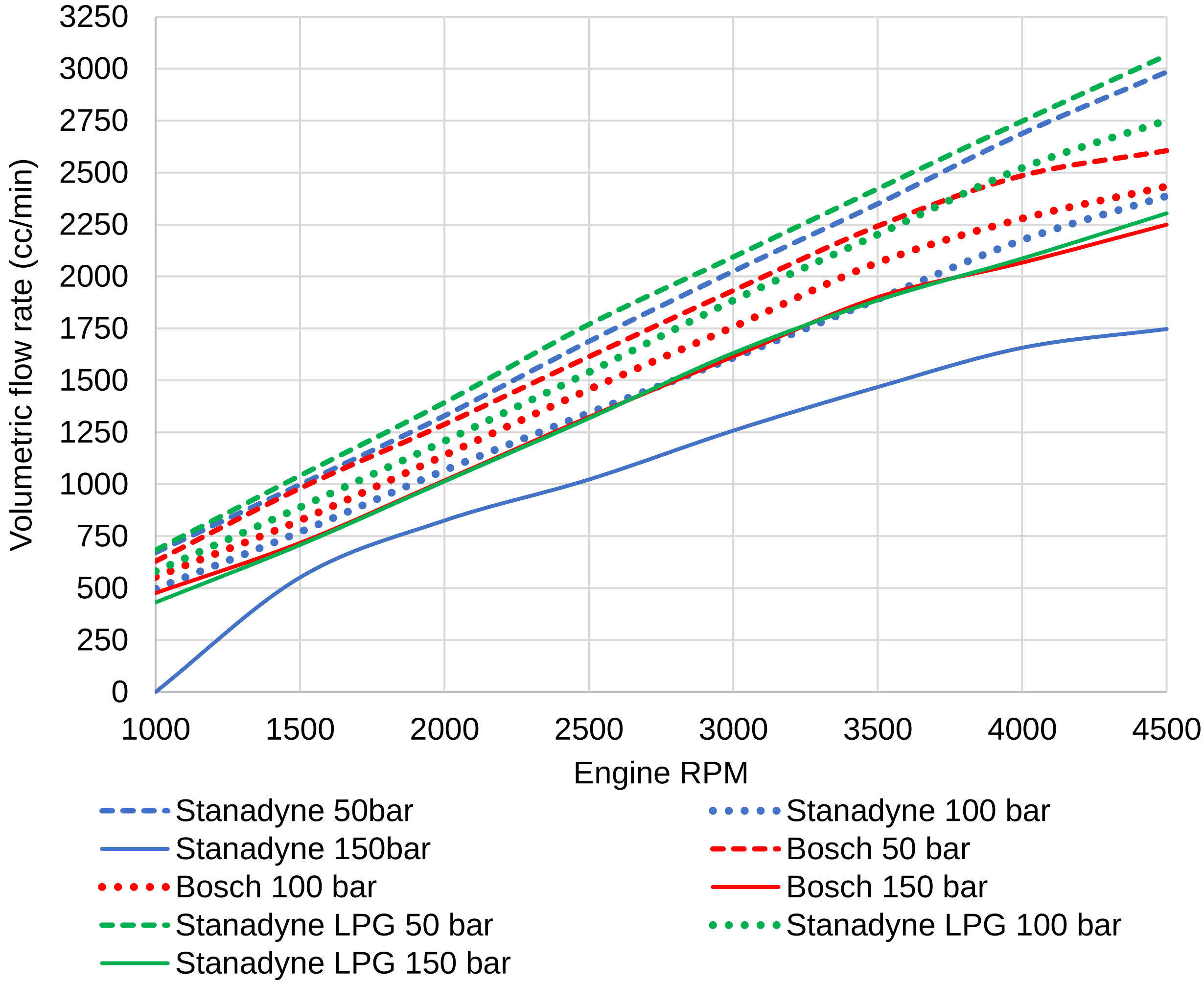

Figure 21: Volumetric flow rate of three DI pumps @ 50 bar, 100 bar, and 150 bars at 30°C fuel temperature.

Figure 22 shows the volumetric efficiency variation of all three DI pumps at three different delivery pressures (50 bar, 100 bar, and 150 bar). As the rail pressure is increased from 50 bar to 150 bar, the volumetric efficiency reduced from 94.5% to 76.2% for the Bosch pump, whereas it was reduced from 90.4% to 54.5% for the Stanadyne SP1200 - 200 and it was reduced from 92% to 69.1% for the Stanadyne development 350 pump. This reduction in volumetric efficiency explains the increased localized cavitation in the Stanadyne pumps when compared to the Bosch pump. The variations in fuel flow rate and volumetric efficiency with respect to fuel temperature were not presented in the section for a couple of reasons. Firstly, the Bosch pump was only tested at two temperature points, whereas the Stanadyne SP1200 – 200 pump was tested at three temperature points. Secondly, from the Stanadyne SP 1200 – 200 pump temperature sweep testing, the volumetric efficiency changed minimally with temperature. Furthermore, the minimal change in the volumetric efficiency is attributed to the changes in the fuel density and slight improvement in the rail stability at some operating points when the pump is tested at high fuel temperatures.

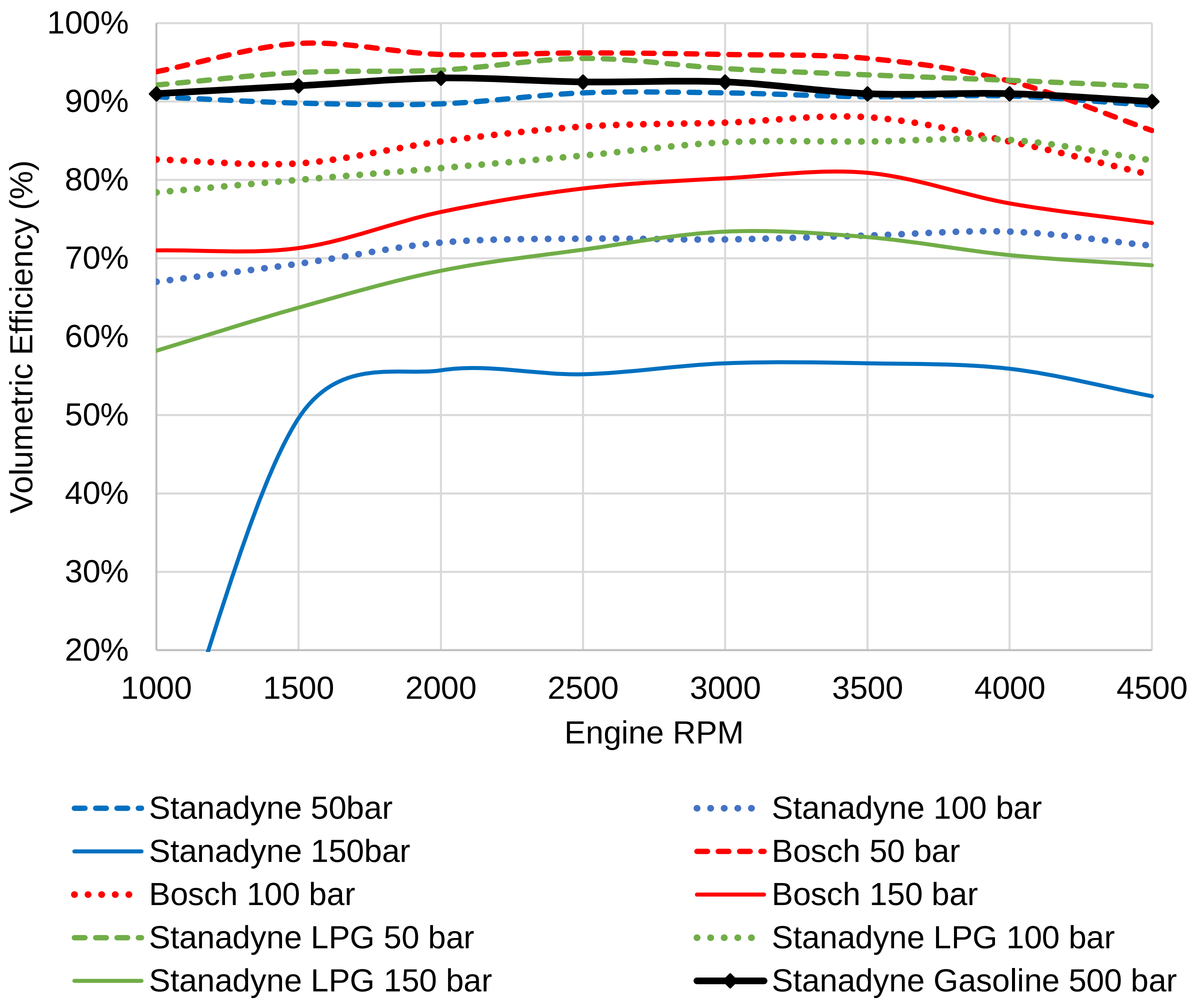

Figure 22: Volumetric efficiency of three DI pumps @50 bar, 100 bar, and 150 bars at 30°C fuel temperature.

Overall, while all the pumps met the flow metrics, the Stanadyne Development 350 pump has slight advantage in the operating pressure and temperature ranges considered in this study.

Durability testing: Limited time (25 hr.) durability tests were performed on the Stanadyne SP1250-200 & Bosch HDP6 - 300 pump to test for degradation over limited run time. The flow shift variations at three different engine speeds were observed at 5 hour increments (1000 RPM, 2000 RPM, 3600 RPM). The tests were performed at a fuel inlet temperature of 30°C +/- 2°C and a pump outlet pressure of 100 bar. The results were reported with respect to the number of pump cycles. The Stanadyne development pump was not available at that time due to supply chain and program timing constraints. Since the Stanadyne 350 pump has a similar build as the Stanadyne 200 pump, the 25 hrs. durability testing for this pump was passed after discussing with the PERC team.

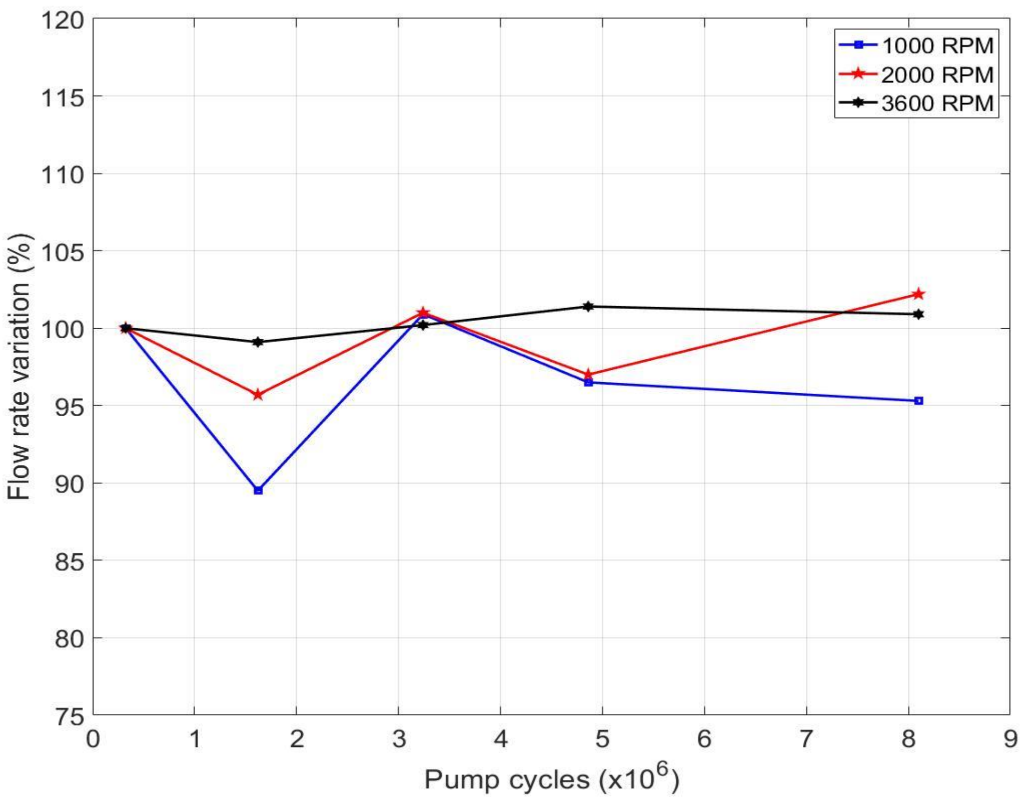

Figure 23: Flow rate variations for Stanadyne 200 pump at three different speeds during the 25-hr. pump reliability testing

The variation in the fuel flow rate with respect to pump cycles at three different engine speeds for the Stanadyne SP1200 - 200 is shown in Figure 23. As shown in Figure 23, the flow rate deviations were within +/- 5% for most of the time except for one point (outlier) where it was 10%. In addition, the tests were performed under conservative conditions, unlike the extreme test conditions mentioned in the temperature and pressure sweep section. The flow rate variation might vary significantly at higher pressures (~175 bar) and temperatures (40°C) and alternatively at speeds below 1000 RPM due to increased localized vaporization/cavitation of fuel. The Bosch pump also has flow rate deviations within +/-5%. To avoid redundancy, the data of the Bosch pump with respect to engine speed was not presented.

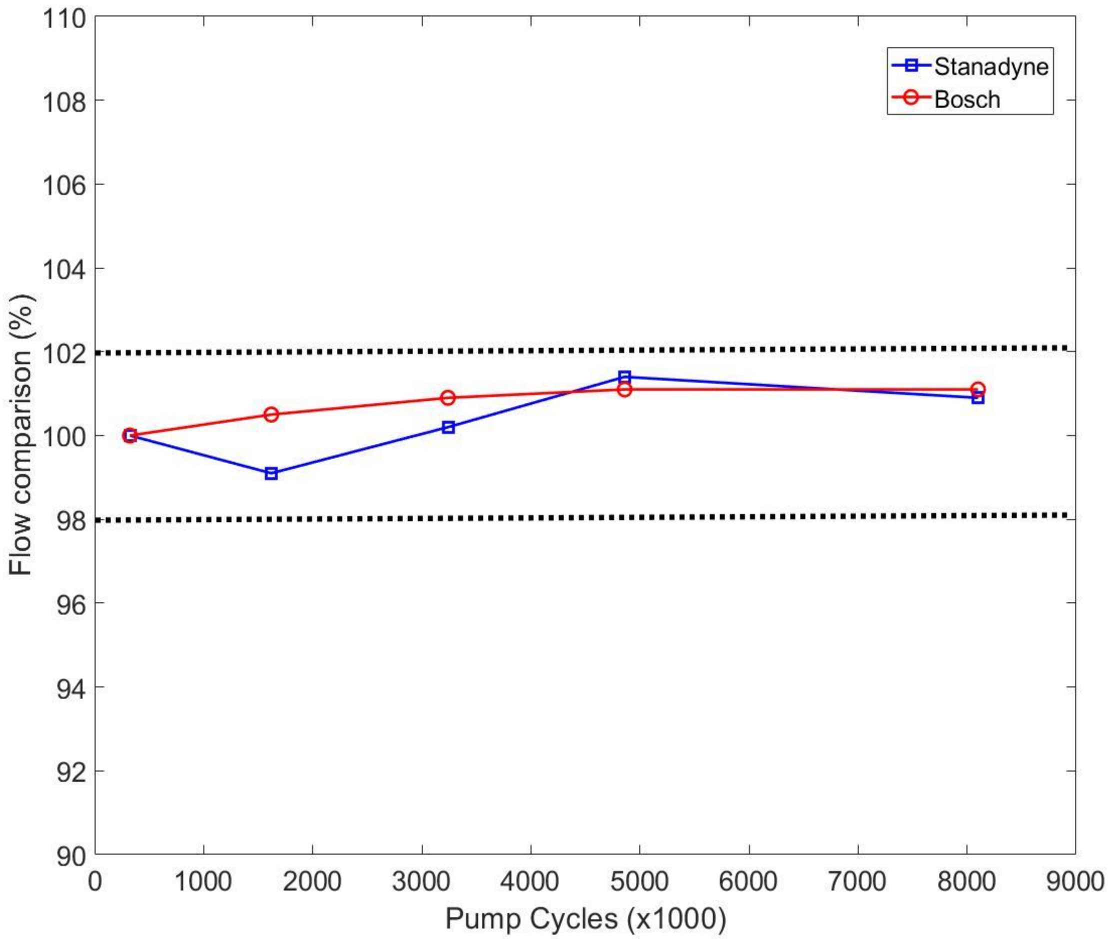

Figure 24: Flow shift variation comparison of Bosch & Stanadyne pumps during limited time durability testing

Figure 24 shows the deviations in the flow rate for Stanadyne SP1200 – 200 and Bosch 350 pump at 3600 RPM and 100 bar delivery pressure. The flow rate deviations, as shown in Figure 24, were within +/- 2% (dotted black lines) for all the pump cycles. Under the tested conditions, both pumps passed the limited durability tests with no signs of wear or degradation.

Visual Wear: After the 25-hr. durability testing, the Stanadyne SP1200 – 200 and Bosch 350 pumps were disassembled to understand the wear caused by the poor lubrication characteristics of propane when compared to gasoline. Images of different subcomponents within the pump were captured to understand the wear properties and provided. After the 25-hr. durability testing, no signs of degradation or wear were observed on the pump piston, bore, DLC coating, and seals.

Figure 25: Image of pump pistons after durability testing

Figure 26: Image of lower pump piston seal of Stanadyne SP1200 – 200 pump

6.2.2 DI Fuel Injectors Testing

Similar to the individual pump testing, all three injectors were individually tested & baselined. The three injectors are the Delphi injector (200 bar), Bosch HDEV6 (350 bar), and Stanadyne Development (350 bar) injector. In this section, the results from the individual injector testing were compiled & compared together in order to draw comparisons. This section is organized into three main subsections:

1. Static Measurements – Compares the resistance, temperature, geometrical dimensions, mass, tip diameter, seals, etc. for the injectors.
2. Injector pulse width variation and linearity checks – Compares the fuel flow rates with respect to injector pulse width at a given temperature, pressure, and engine speed.
3. Durability testing – Compares the mass flow rate deviations at a given temperature, pressure, and engine speed over a period of time.

Static measurements: The results pertaining to the injector static measurements are provided in Table 8 below. As shown below, the team captured all the static measurements (resistance, temperature, geometrical dimensions, mass, tip diameter, injector seal type, injector style etc.) using different instruments available at Katech's test facility.

Table 8: Static baseline benchmark measurements for all DI Injectors

Data Reporting Sheet for LPG-DI Injector Static Measurements			
Part 1: General Test Logistics			
Test Name or Log	Delphi Baseline Benchmark	Bosch Baseline Benchmark	Stanadyne 350bar Benchmark
Operator Name & location	Eric S., Katech Engineering, Clinton Twp, Michigan		
File Name or Data Archive	N/A	N/A	N/A
Additional Information	N/A	N/A	N/A
Part 2: Information of Injector			
Manufacturer	Delphi	Bosch	Stanadyne
Description	LT1/L86 Injector GM	4.2L LTA	LPG Injector Prototype
Part Number	GMP-19420316	GMP-12693823	607716
Serial Number	1006101908	202105041300	KB089 3074
Additional Information	NA	N/A	NA
Part 3: Measurements Electrical			
Control Value Resistance (Ohms)	1.3	2.1	1.7
Ambient Temperature (°C)	21.5	21.5	21.5
Part 4: Measurements Mechanical			
Overall Length (mm)	108.9	96.6	95.8
Overall Width (mm)	50.0	20.6	20.9

Overall Height (mm)	29.5	41.6	20.9
Tip Install Diameter (mm)	7.45	6.00	6.00
Mass of Injector (gm)	84	56	66
Dual or Single Teflon Seal	Dual	Dual	Dual
Connector Style	USCAR Connector	USCAR Connector	Flying Lead – Testing
Injector Holes	6	6	6
Injector Style & Angle	Concentric Cone, Compound Offset	Concentric Cone, Compound Offset	Concentric Cone, Compound Offset, Bore to Length Adjusted for LPG penetration
Rated Leakage	Reference Bosch Standard <2.5mm3/min	Reference Bosch Standard <2.5mm3/min	Reference Bosch Standard <2.5mm3/min
These are the **Basic Injector Static Test Results**. There Are No Test Deviations.			

Based on the static measurements presented in Table 8, the following conclusions can be made:

- Delphi injector was rated for 200 bar, whereas, the Bosch and Stanadyne injector were rated for 350 bar.
- All the injectors have 6 holes with a concentric cone design.
- The Bosch injector (56 g.) weighs lighter when compared to the Delphi injector (84 g.), and Stanadyne injector (66 g.).
- The control resistance of the Bosch injector is 2.1 Ohms whereas, for the Delphi and Stanadyne injector, it is 1.3 Ohms, and 1.7 Ohms respectively.
- The Bosch and Stanadyne injector mounts utilize 6.0mm OD, and the Delphi utilizes 7.5mm. This makes the injectors switch incompatible between Bosch/Stanadyne and Delphi injectors without modifying the cylinder head. Stanadyne will be able to support 7.5mm tip size on future programs.
- Of major importance was the modification in the bore-to-length ratio of the Stanadyne injector for LPG fuel penetration. The Stanadyne maintained the angle of the reference Delphi Injector but adjusted the length vs bore ratio. The Delphi and Stanadyne has near identical hole size due as seen in flow data in Figure 27. The Bosch injector had proportional smaller nozzle hole diameter due to its base platform being on a smaller output engine.

Apart from the above-mentioned differences, the rest of the details remained the same for both injectors.

Pulse width variation & linearity check: Dynamic flow verification tests that indicate the fuel flow rate at a specified pulse width were performed on all the injectors. These tests are needed for defining the suitability of the down-selected injector for specific engine applications. Tests were performed at different rail pressures and different pulse widths at an ambient temperature of 20°C +/- 2°C and at an ambient pressure of 100 +/- 5 kPa. The fuel in the rail is at a temperature of 30°C +/- 2°C. The results were reported as milligrams per injection or injector stroke. The injector period is maintained at 33.3 ms for all the test conditions (i.e., 3600 RPM engine speed).

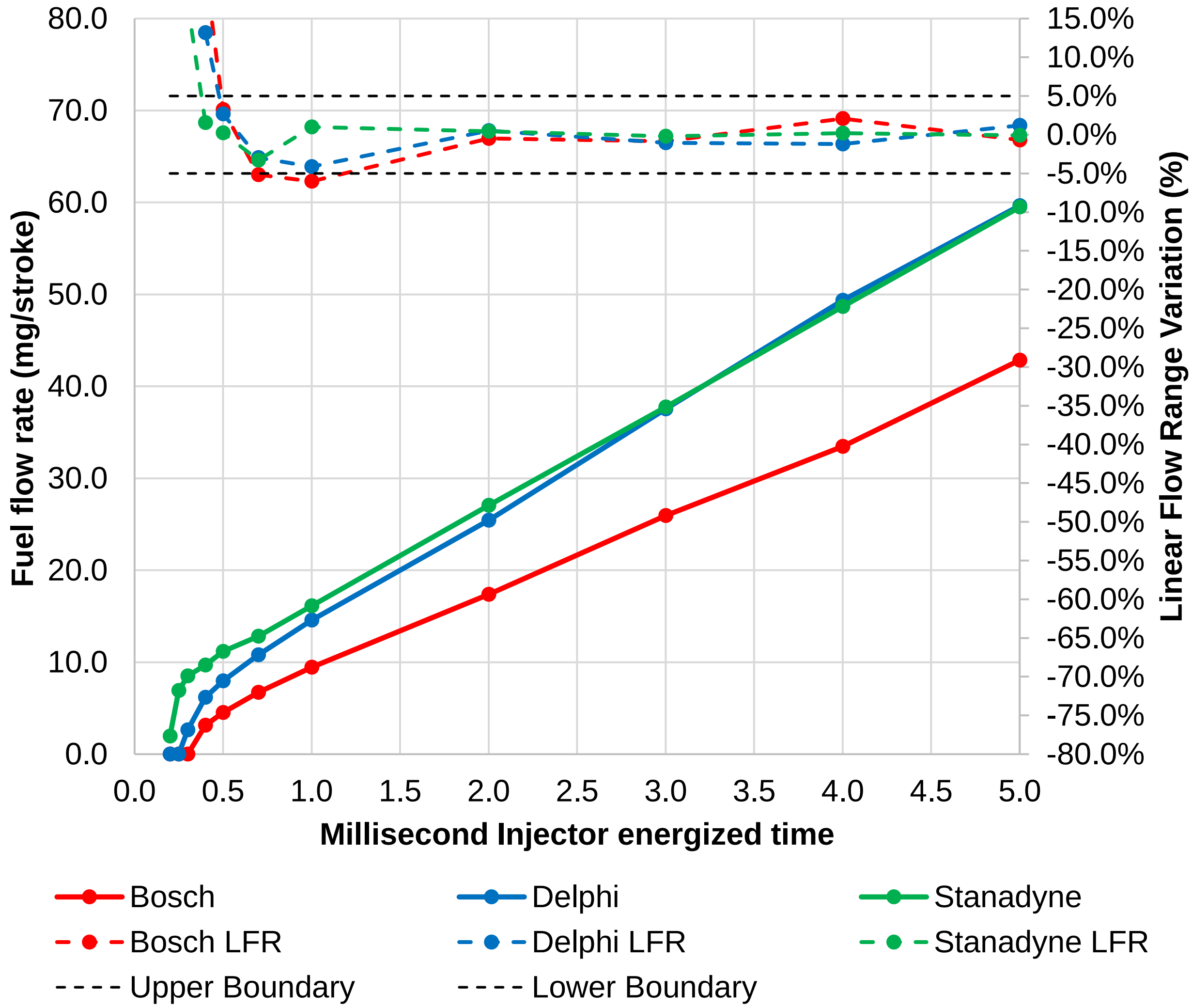

Figure 27: Variation in injection flow rate with respect to pulse width at 100 bar rail pressure

The reason for limiting the rail pressure to 100 bar is two-fold: 1. The Delphi injector operated with Stanadyne SP1200 – 200 pump has a delivery pressure limitation of 200 bar. However, the Bosch and Stanadyne injectors were operated to 350 bar with the

Bosch HDP6 pump (350 bar) and Stanadyne development pump (350 bar). 2. The majority of the GDI engines operated at delivery pressures close to 100 bar under steady state conditions.

The variation in the injector flow rate with respect to pulse width at 100 bar is shown in Figure 27. As shown in Figure 27, the Bosch injector has a lower injection fuel flow rate for all the injector pulse widths when compared to the Delphi and Stanadyne injector at a rail pressure of 100 bar. The main reason is in the selection process of the injector for the proposed application. When choosing the injector, the Katech team considered fuel mass flow rate as a major factor. In other words, both Delphi and Bosch injectors will attain almost the same fuel flow rate for all the pulse widths at their respective peak operating pressures. This was confirmed by plotting the Delphi and Bosch injectors' fuel flow rate curves at 175 bar and 250 bar respectively.

The flow rate with respect to the injector pulse width was fitted to a linear regression curve to obtain the linear flow range of the injectors. The R^2 valve for all the linear regression curves were maintained above 0.985 for all the injectors at 100 bar as well as other

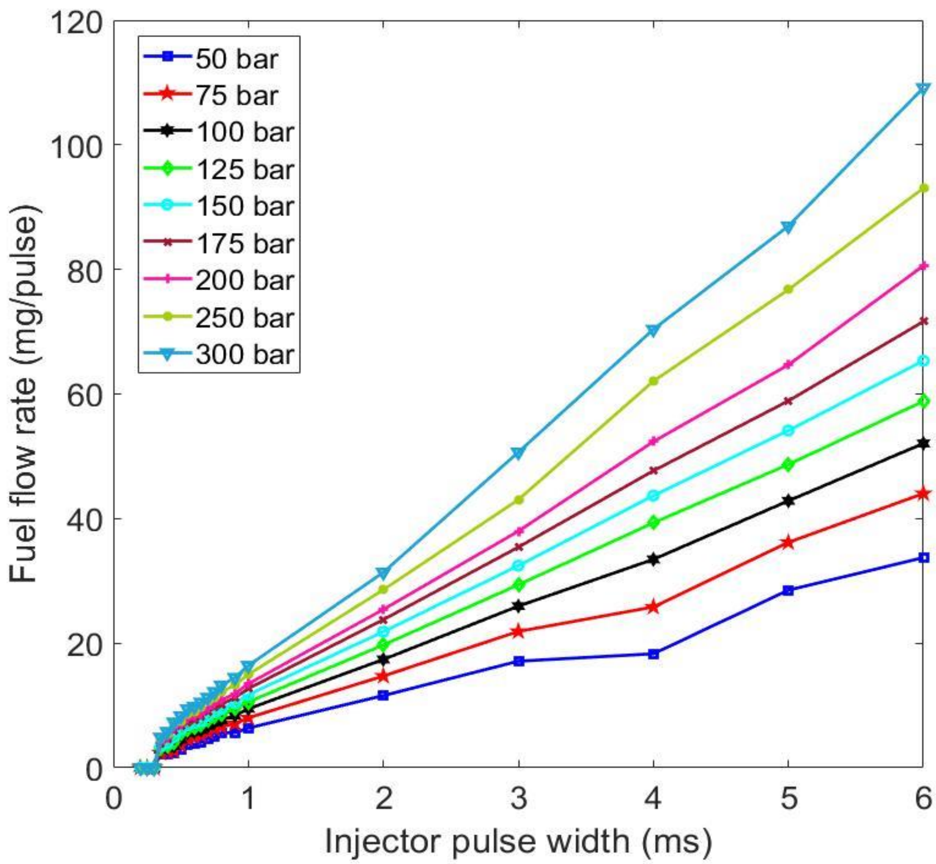

Figure 28: Injector flow rate variation with respect to pulse width (Bosch)

delivery pressures. For accurate determination of linearity, sensitivity in test points was changed according to the pulse width. For example, in the low range below 1ms, a total of 15 test points were chosen, and in the long range above 1ms, a total of 5 test points were chosen. As shown in Figure 27, the flow variations are within +/- 5% for all the injectors with injection pulse widths above 0.5 milli sec.

The variation in the fuel flow rate with respect to different delivery pressures for the Bosch injector was shown in Figure 28. The fuel flow comparison for the injector pulse widths below 1 ms is shown in Figure 29. To avoid redundancy, the variation in the flow rates with respect to injection pulse width was not shown for the other two injectors (i.e., Delphi and Stanadyne). The sensitivity in test points with respect to the injector pulse width is evident from these figures. As shown in these Figures, the flow rate is linear at above pulse widths of 0.7 milli sec, whereas in-between 0.25 to 0.75 milli se, it is non-linear. Below, 0.5 milli sec, the predicted flow rate is within +/- 15%, whereas the acceptable

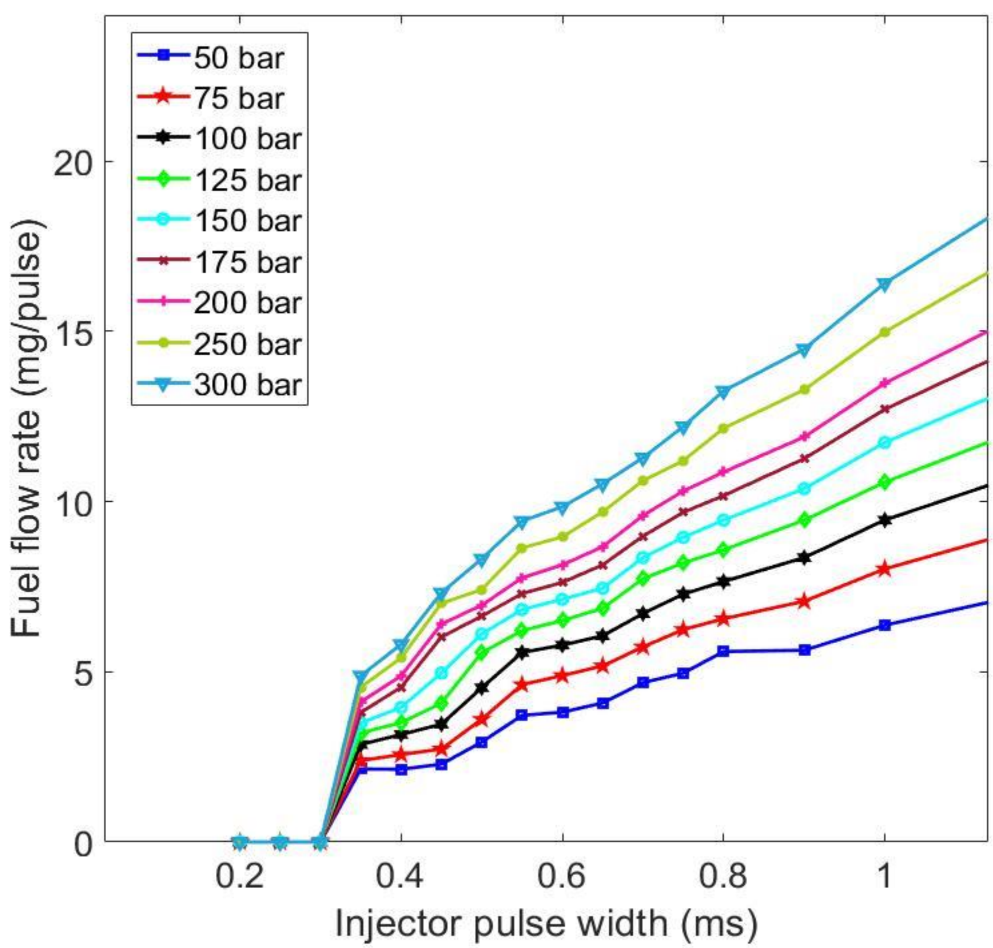

Figure 29: Injector flow rate variation with respect to pulse width

deviation is +/- 5% according to SAE J2713 Injector standards. Overall, based on the flow testing, all the injectors are capable enough to meet the metrics needed by PERC.

__Durability testing:__ Similar to the short high-pressure DI pump durability testing, 25 hr. durability tests were performed on Delphi and Bosch injectors in order to understand the flow shift variations. The Stanadyne development injector was received after the durability testing of these injectors due to supply chain constraints. Furthermore, the learning from the testing of these two injectors was implemented in the Stanadyne development injector. As a result, durability testing of Stanadyne injector was not carried out at this time. The tests were performed at a fuel inlet temperature of 35°C +/- 2°C and an injector rail pressure of 100 bar. The injector period is maintained at 33.3 ms or 3600 RPM engine speed, and the pulse width is maintained at 4 ms. The results were reported with respect to the number of injector cycles. The variation in the injection rate with respect to injector cycles is shown in Figure 30. As shown in Figure 30, the flow rate deviations are within +/- 1%.

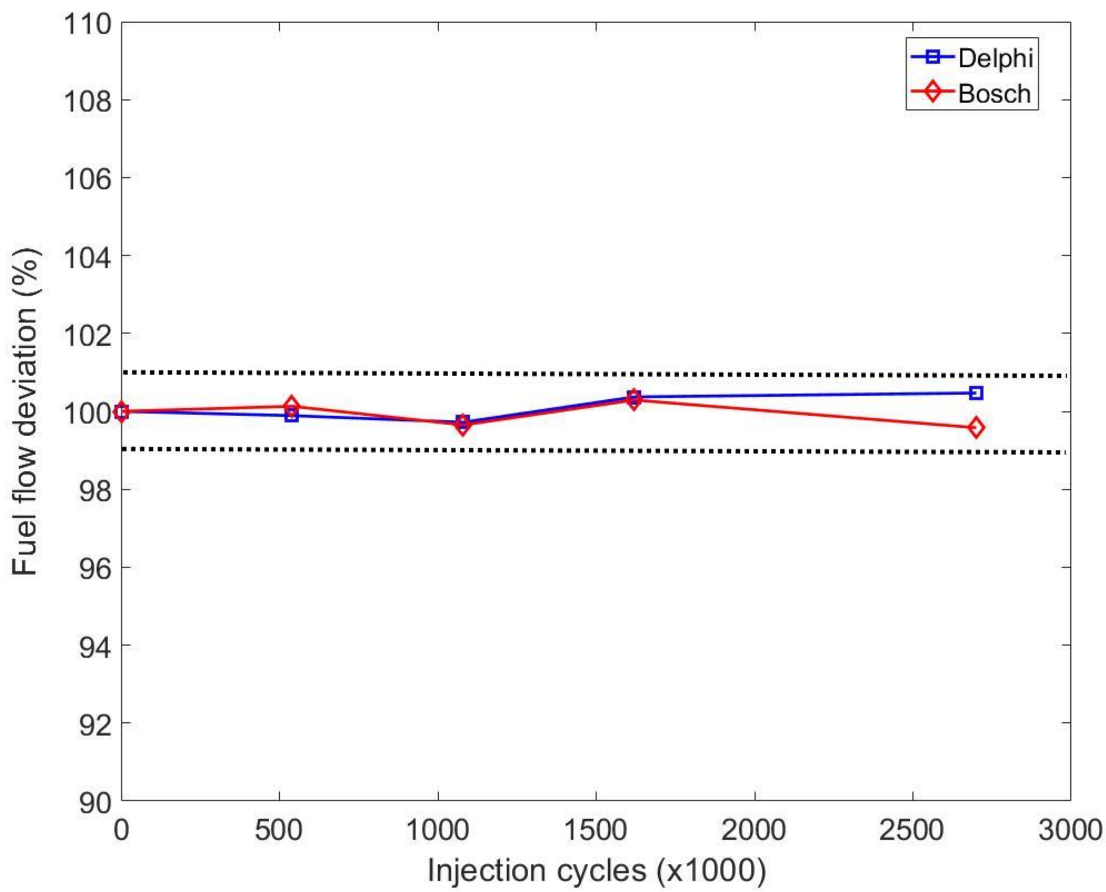

Figure 30: Flow rate deviations of Bosch & Delphi injectors during the 25-hr. injector reliability testing

Visual Wear: After the 25-hr. durability testing, the injector pump was photographed to understand the visual wear on the injector subcomponents. Images of injector needle and seat provided below have no signs of degradation or wear. In addition, the injectors also passed the post durability leakage process. The images of the post durability leakage test process is shown in Figures 31, 32, and 33.

Figure 31: Image of injector needle after 25 hr. injector reliability testing

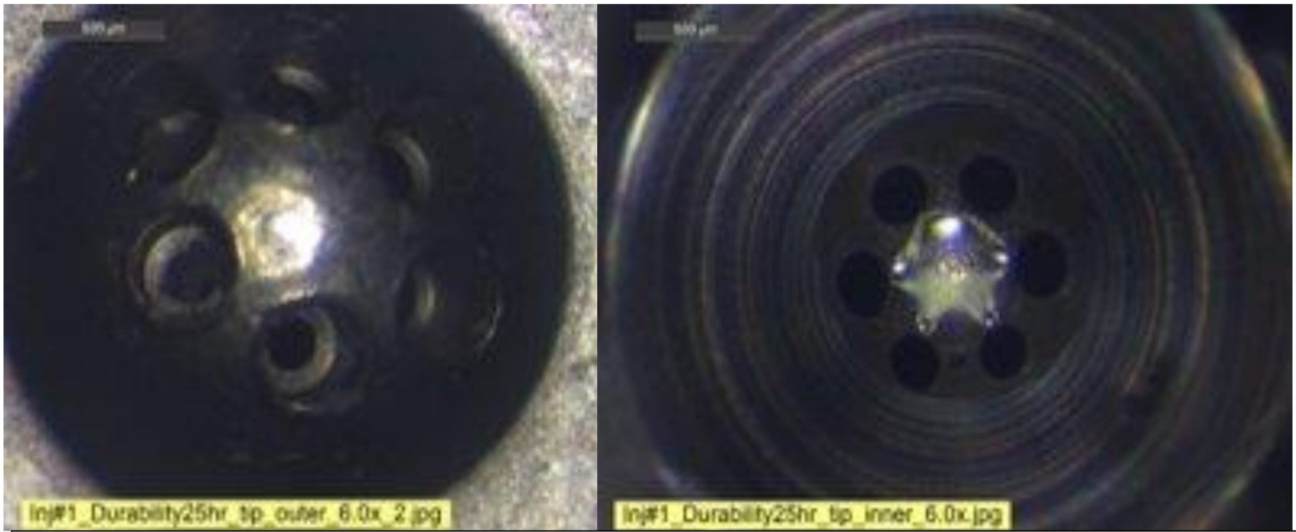

Figure 32: Image of outer and inner injector tips after 25 hr. injector reliability testing

6.2.3

Figure 33: Image of the injector during the post durability leakage test process

Decision Matrix Scores

In this section, the scores for the decision matrix based on the individual DI fuel system component testing were provided to down select the best possible High-side DI pump and fuel injector combination for firing phase testing.

Tables 9 and 10 show the scores of the decision matrix for high-side DI pumps and DI fuel injectors respectively. As shown in Figure, the Stanadyne development pump achieved a higher score followed by the Stanadyne SP1200 - 200 pump and the Bosch HDP pump. The Bosch HDP6 pump obtained low scores mainly due to its bad mounting ability and also due to hard internal modification. When compared to the development pump, the Stanadyne SP1200 - 200 had poor operating pressure and volumetric efficiency characteristics. One concern with the Stanadyne prototype pump is its cost when compared to the other COTs components, but with Stanadyne's ability to scale its manufacturing of the LPG specific pump it would very quickly become cost competitive if selected for a low or medium volume production application. Overall, the Stanadyne LPG 350 bar was rated the highest based on metrics.

Table 9: Decision matrix scores for the high-side DI fuel pumps.

Metric	Weight	Stanadyne SP1250-200	Bosch HDP6	Stanadyne LPG 350
Priming	Go/No-Go	Pass	Pass	Pass
Pump Flow Rate	10%	5	5	5
Operating Pressure	5%	3	4	4
Volumetric Efficiency	10%	3	5	5
Pump Stress Test	10%	5	5	5
Pump Endurance	15%	4	4	5
Mounting Ability	10%	5	1	5
Modifications	10%	3	1	5
Cost	10%	4	4	3
Scalability	20%	3	2	5
Total	**100%**	3.85	3.3	4.5

Table 10: Decision matrix scores for the DI fuel injectors.

Metric	Weight	Delphi 200 Bar	Bosch HDEV6	Stanadyne LPG 350
Injector Repeatability	10%	4	4	5
Static Flow Rate	15%	4	3	5
Injector Stress Testing	10%	5	5	5
Injector Endurance	15%	4	4	5
Mounting Ability	10%	5	4	5
Modifications	10%	5	1	5
Cost	10%	5	4	3
Scalability	20%	5	1	5
Total	**100%**	4.6	3.05	4.8

From Table 10, the Stanadyne development injector has a higher score followed by the Delphi and Bosch injector. In the injector case, the Bosch injector is hard to modify and scale when compared to the other injectors. On the other hand, the Delphi injector was rated slightly lower in terms of injector endurance and injector static flow rate when compared to the Stanadyne development injector. The Fuel injector is a closer comparison between the OE Delphi 200 bar injector and Stanadyne LPG 350. Both injectors are quite comparable for the L8T application, future testing will determine if a significant improvement in emissions is offered with specific nozzle / flow rates as this is the main benefit of the Stanadyne injector over an OE injector. For the purpose of this testing and metrics the Stanadyne LPG 350 was selected for engine firing phase.

6.3 Firing Phase Testing Results

The results from the engine testing phase are provided in this section. Different graphs ranging from DI fuel pump performance, and engine health were used as a means to prove the operating concept and robustness of the proposed liquid propane DI fuel system.

6.3.1 Baseline Power & Torque Comparison

As mentioned before, initially, the team operated the engine with 87AKI gasoline to establish the baseline power and torque characteristics of the GM L8T engine with the OEM DI fuel system components. Following the gasoline operation, the same engine was operated on propane with a down-selected Stanadyne LPG DI fuel system to compare the deviation in the power and torque characteristics. The engine lambda was maintained in between 0.95 – 1.0, as shown in Figure 34, for all the test conditions including gasoline and propane operation. These variations in the engine power and torque characteristics are shown in Figure 35.

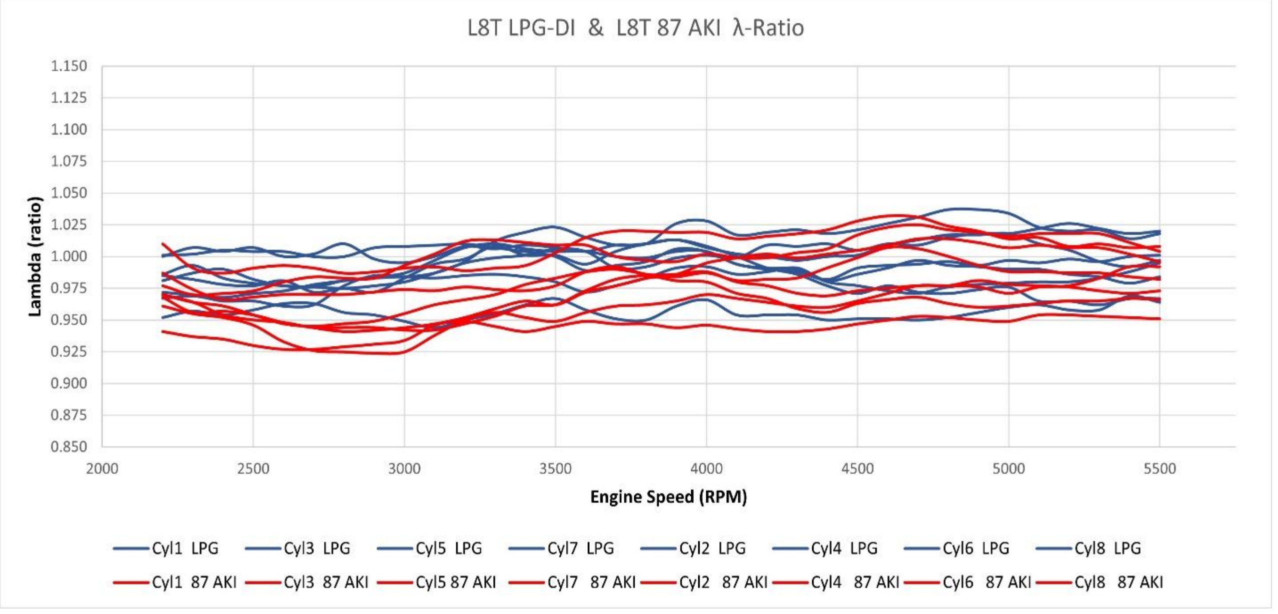

Figure 34: Individual cylinder lambda with respect to speed for both gasoline and LPG baseline testing.

As shown in Figure 35, the torque and power characteristics of the Stanadyne LPG-DI system was able to meet or exceed the 87 AKI baseline. The variations in exhaust gas temperatures for individual cylinders for engine speed is shown in Figure 36. As shown in Figure 36, the EGTs of propane are lower by gasoline by 25°F to 75°F. This is again due to ignition timing advance for propane when compared to gasoline as shown in Table 11. Since propane has much higher knock resistance when compared to gasoline, the ignition timing was advanced by 2 to 3 degrees, hence the EGTs were lower for LPG operation.

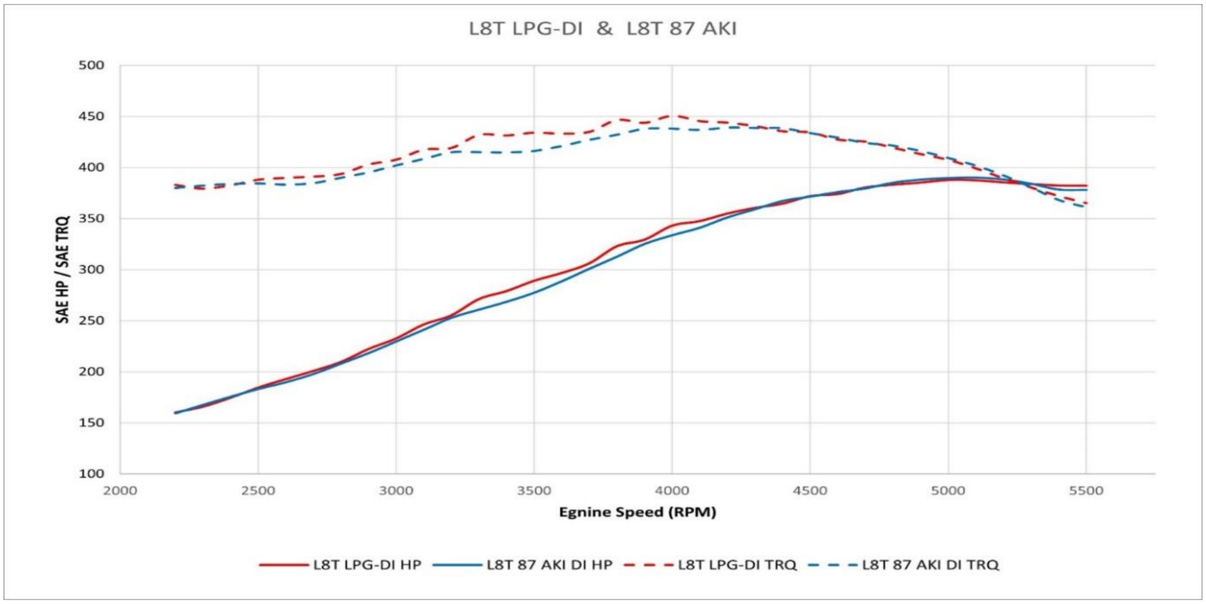

Figure 35: Torque and power characteristics comparison of L8T engine with 87AKI gasoline and LPG DI operation

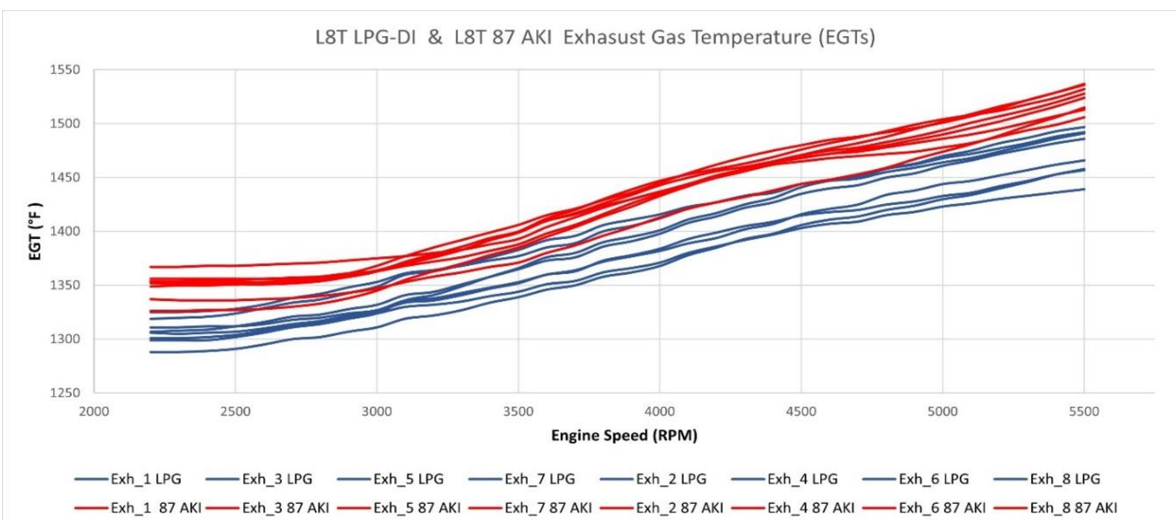

Figure 36: Exhaust gas temperatures of individual cylinders at different speeds with 87AKI gasoline and LPG DI operation

Table 11: Ignition timing advance for propane during baseline testing for full load conditions.

	Ignition Advance (LPG minus 87 AKI Map)								
Speed	**1500**	**2000**	**2500**	**3000**	**3500**	**4000**	**4500**	**5000**	**5500**
Degrees	3	3	3	2.875	2.75	2.65	2.5	2.5	2.5

Overall, the proposed liquid propane DI fuel system was correctly sized for the proposed engine in order to meet the performance requirements of the baselined system.

6.3.2 Ambient & Engine Steady State Conditions

Figure 37 below barometric and inlet temperature measurements at the end of every 50 hrs. interval. The blue and orange lines represent barometric pressure, and inlet air temperatures respectively. As shown in Figure 37, the barometric pressure was maintained at 99.5 kPa with a deviation of +/- 1 kPa, whereas the air inlet temperature ranged from 22°C and 30°C with an average of 26°C. Even though the team mentioned about +/- 1°C deviation in air inlet temperature, the air temperature did exceed this limitation during the testing process. On the other end, these significant deviations are beneficial to understand how the fuel system responds under these conditions/outliers.

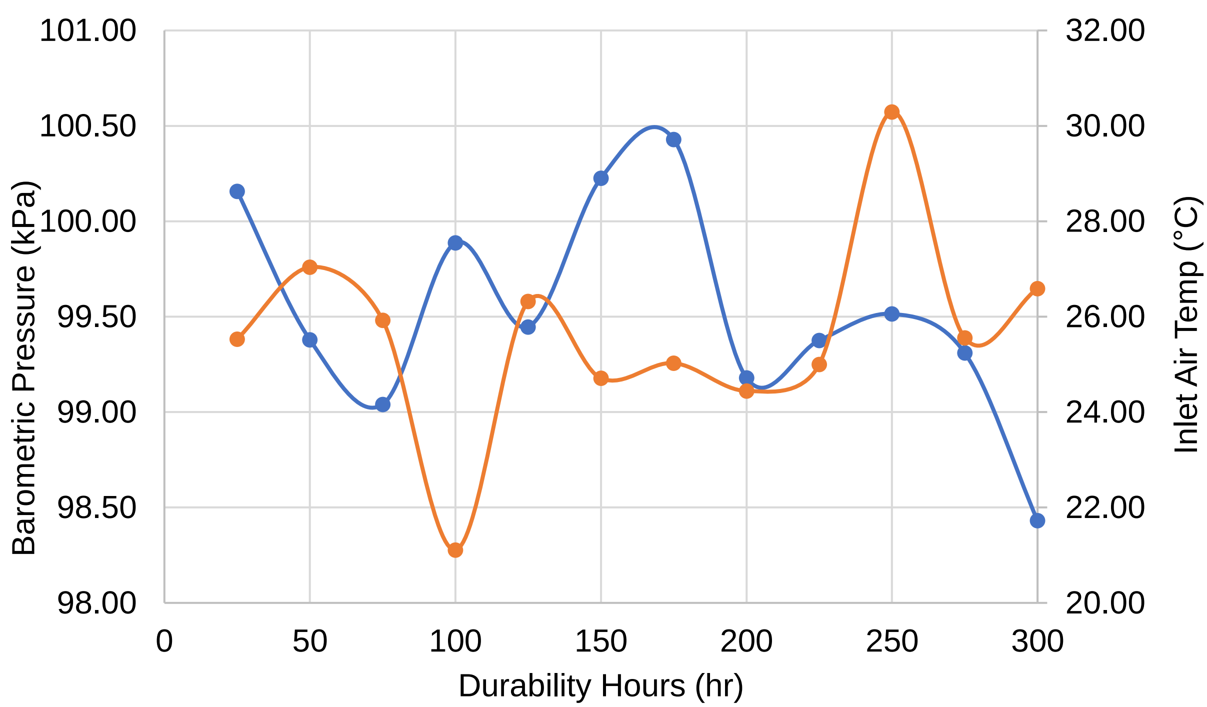

Figure 37: Barometric pressure and air inlet pressure variations during 250 hrs. durability testing.

Figure 38 below shows the engine speed and torque variations at different points during the 250 hour durability testing. The engine speed relatively stayed the same at 3200 RPM, wide open throttle condition, with +/- 10RPM at specified points in the 30-minute durability cycle that was repeated 500 times. The variation in the SAE torque is due to variations in the ambient conditions. Initially, the torque variations were around +/ - 2%. Upon detailed investigation, it was found that absorber shaft bearings showed signs of wear/damage. At test cycle 200, the dynamometer absorber was changed to prevent a negative impact on the data. After this modification, SAE torque was within an accuracy of +/- 0.6% for the remaining testing period.

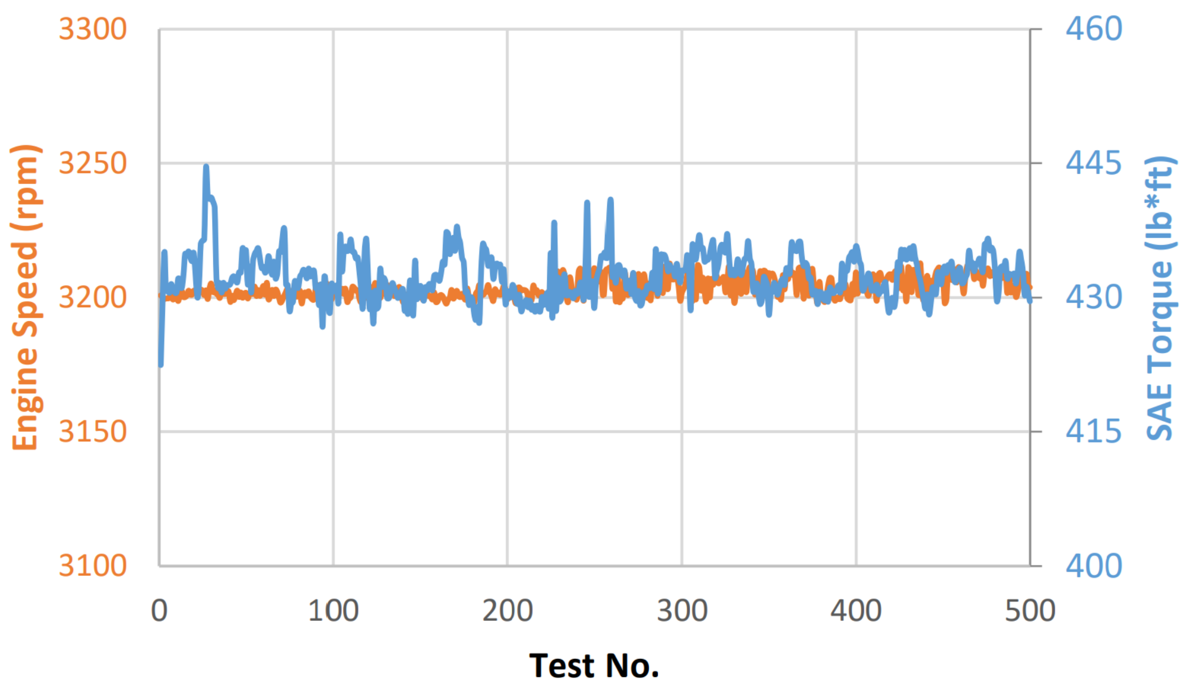

Figure 38: Variation in the engine speed and SAE torque at 3200 RPM WOT during the 250 hrs. durability testing.

6.3.3 Low-side DI Pump Health

This section provides the results related to the low-side DI pump performance from engine testing. For the engine testing, the team used a Bosch pump with PRV limited to 8.27 bar due to high fuel output when compared to the TI automotive pump used in the non-firing phase. The low-side pump performance was determined based on its ability to hold the pressure of 4.0 bar above the fuel tank pressure in the fuel lines connected to the high-side pump inlet at 3200 RPM wide open throttle condition. At other speeds & conditions, the pressure was targeted between 3.0 bar to 4.5 bar. The fuel delta pressure and fuel tank pressure at 3200 RPM wide open throttle during 250 hrs. of durability testing is shown in Figure 39.

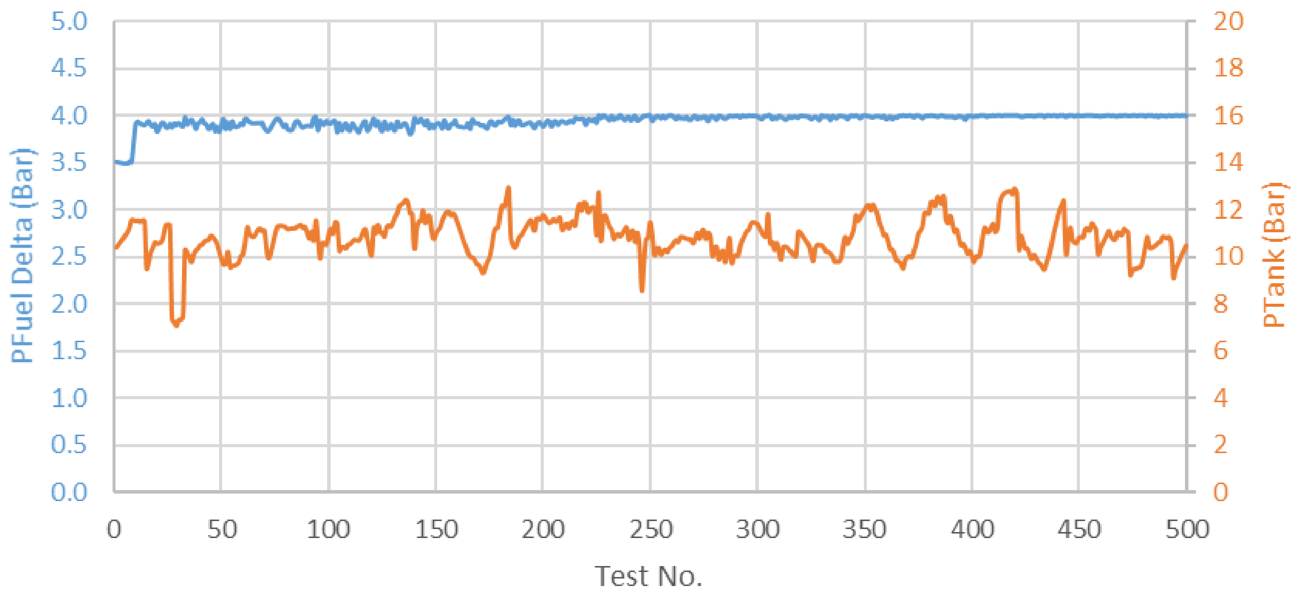

Figure 39: Fuel Delta pressure and Fuel Tank pressure at 3200RPM WOT during 250 hrs. durability testing.

As shown in Figure 39, the delta fuel pressure was maintained at 4 bar irrespective of fuel tank pressure due to control system algorithms such as mode identification and selection and recirculation valve operation developed by the Katech team. In addition, the tank pressure varied from 8 bar to 12 bar depending on the fuel temperature in the tank. Overall, the low-side DI pump was sufficient to deliver the fuel to the high-side fuel pump at the right pressure irrespective of the conditions in the fuel tank.

6.3.4 High-side DI Pump Health

This section provides the results related to the high-side DI pump performance from engine testing. For the engine testing, the team used the Stanadyne Development Pump with a maximum operating pressure of 350 bar. The high-side pump performance was determined based on its ability to hold the rail pressure at the set point, Magnetic Solenoid Valve (MSV) delivery angle, volumetric efficiency, and fuel flow rate from the pump. Figure 40 below shows the nominal (commanded) and actual pressures in the injection rail (outlet of the high-side pump). As shown in the figure, the nominal and actual pressures are almost the same.

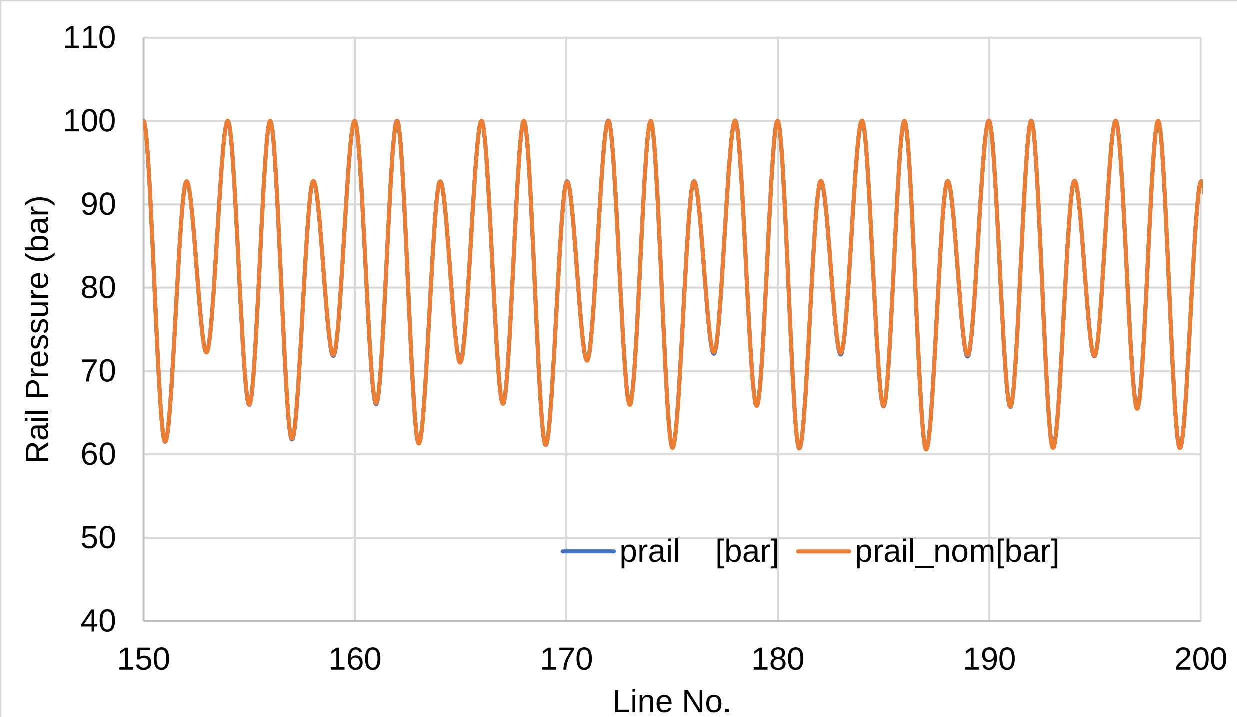

Figure 40: Nominal and actual fuel pressures proving the reliable performance of high-pressure DI pump.

Figure 41 shows the deviation in the MSV delivery angle and engine fuel mass flow rate during the course of durability testing. As shown in Figure 41, at 3200RPM WOT, the MSV delivery angle remained almost constant in-between 75° to 80° except for the few outliers at test points 265 and 350. These outliers were due to wrong engine calibration, and they were corrected immediately after identification, during refueling. These outliers were not observed again during the remainder of the test. On the engine fuel mass flow rate front, the same conclusion can be drawn. In other words, similar to the MSV delivery angle, the engine fuel mass flow rate remained at a constant value of around 720 gm/min with +/- 10 gm/min deviation.

MSV angle is the fuel pump delivery angle where the solenoid valve in the fuel pumps opens the inlet into the piston assembly for the compression of fuel. The MSV, being PWM activated, has its own operating frequency and pulse width. For this pump, the 100% duty cycle of the MSV refers to 120° crank angle degrees and 0% refers to 0° crank angle degrees. Since the pump that was chosen for testing had 3 lobes, and three pump cycles per cam revolution, the pump is open for 120*3 i.e., 360° at 100% duty cycle.

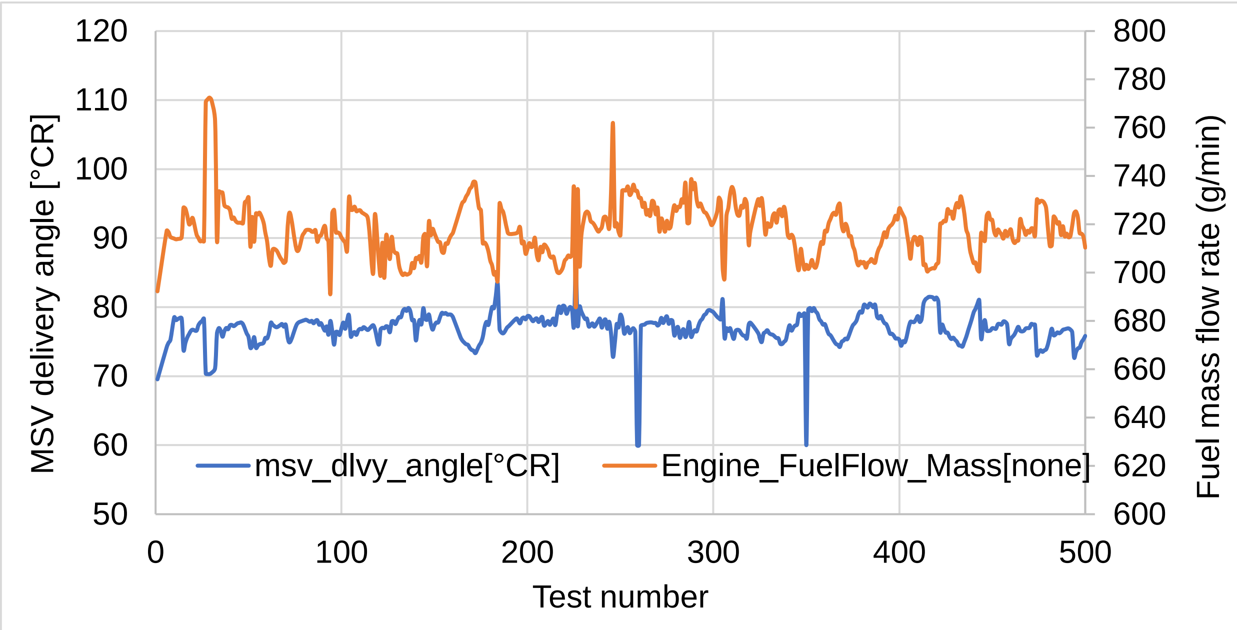

Figure 41: Stable MSV delivery angle and engine fuel mass flow rate of high-side fuel pump during durability testing, 3200 rpm and WOT

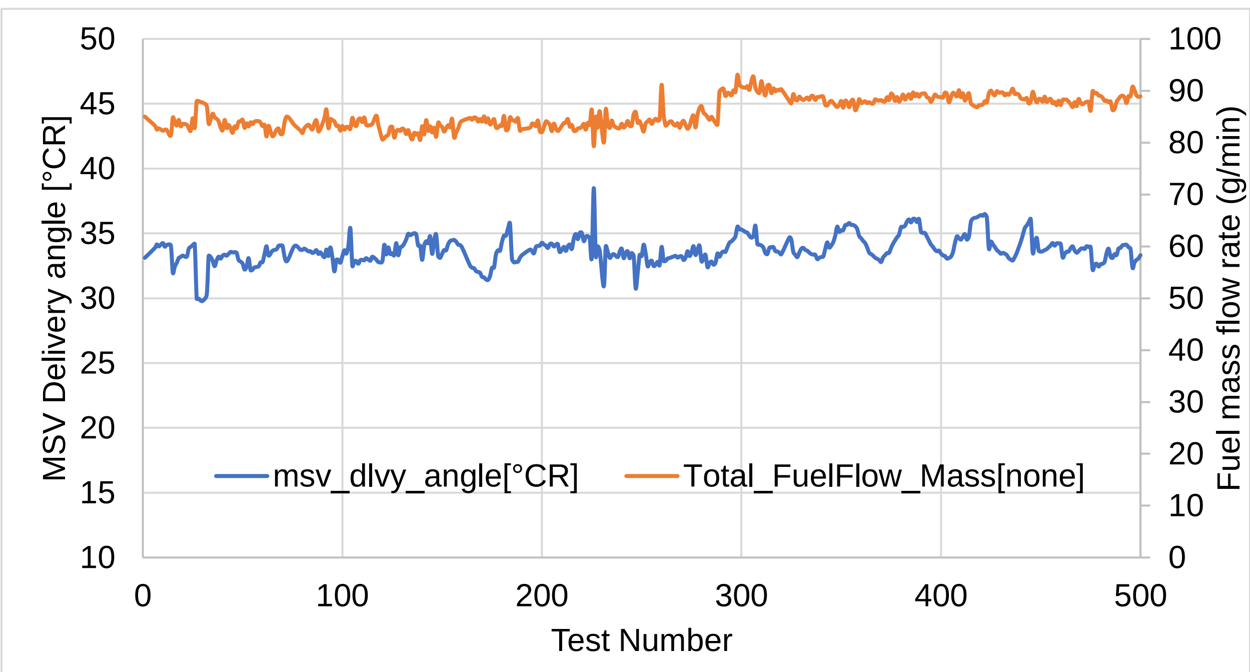

Figure 42: Stable MSV delivery angle and engine fuel mass flow rate of high-side fuel pump at 1250 RPM, 15% load (low throttle & low idle condition).

At higher duty cycles, since the MSV is open for a long time, the compression stroke is sufficient enough to deliver the right amount of fuel, however at lower duty cycles, if the MSV timing/ delivery angle is incorrect, the solenoid is open for a short time and pump compression was not sufficient at deliver the right amount of fuel i.e., the engine gets less fuel volume. This problem becomes worse with liquid propane when compared to gasoline. During the non-firing operation, this phenomenon was observed. Without the regulation valve operation, when the duty cycle was high, the fuel rail pressure was steady, however, when the duty cycle was modified to represent low throttle and idle conditions, the fuel rail pressure was unsteady. This was due to the limited fuel flow through the DI pump chamber, resulting in fuel vaporization before it was able to flow into the fuel rail at high pressure (over 43 bar) where vaporization was no longer a concern. This problem was resolved in the firing phase with the mode identification and the operation of the fuel regulating valve. This conclusion is made based on constant and steady MSV delivery angle and engine fuel mass flow rate during the course of durability testing at both high and low MSV duty cycles as shown in Figures 41 and 42.

Figure 43 below shows the variation in the fuel temperature in and out of the high-side DI pump during the engine testing phase. The blue line in the Figure below represents the fuel temperature in the rail, whereas the fuel temperature at the inlet of the high-side DI pump is represented by the orange line. As shown in the figure, the fuel rail temperature almost followed the same trend as the fuel inlet temperature except that the fuel temperature in the rails is 10° to 12° higher than the inlet temperature of fuel. This is due

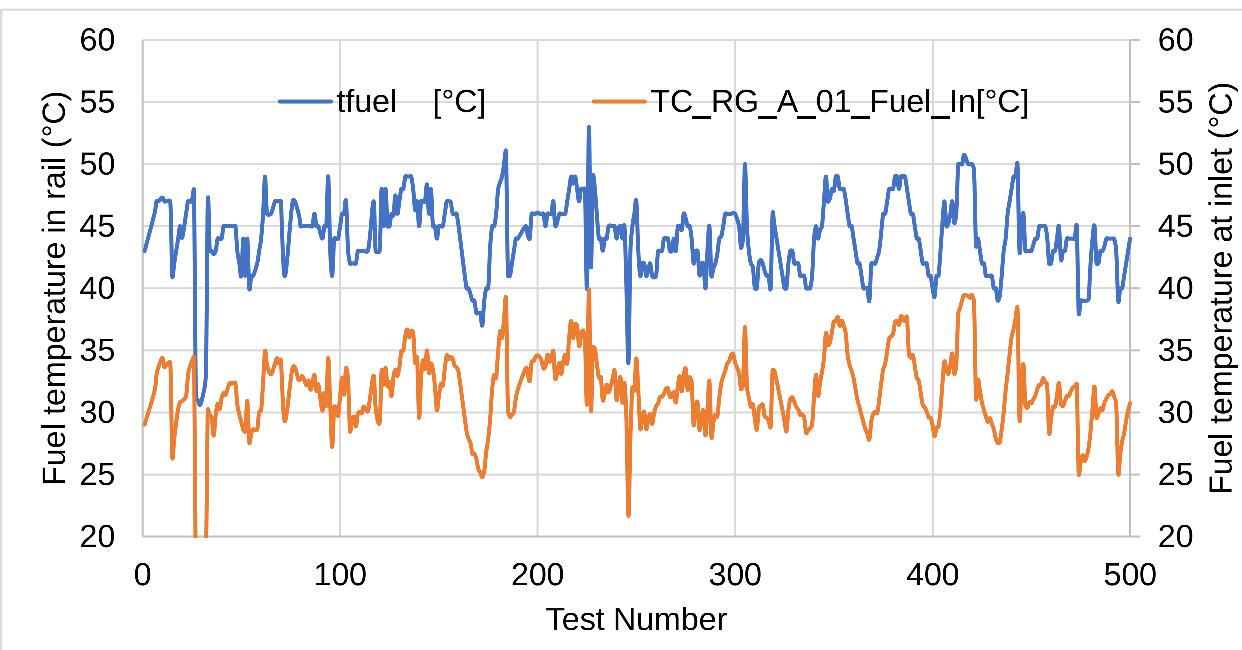

Figure 43: Fuel temperatures in injector rail and pump inlet showing the pump compression effect.

to the compression of fuel inside the pump chamber and also explains our previous conclusion of internal cavitation/vaporization of fuel in the pump chamber during the non-firing phase without the regulating valve operation.

Overall, based on the results provided, the DI fuel pumps are robust enough to handle the temperature, pressure, and other fluctuations encountered when installed on an engine. These results also provide the confidence to proceed forward with on-road or extreme durability testing with additional funding.

6.3.5 DI Fuel Injector Health

This section provides the results related to the DI fuel injectors based on different metrics from engine testing. For the engine testing, the team used the Stanadyne Development Injector rated for 350 bar. Since there are 8 injectors in the system, and these 8 injectors should function all the same time and that too steadily for stable operation of the engine. Hence, the health of DI fuel injectors was determined based on many metrics provided below:

- Lambda ratio per each cylinder
- Lambda Bank & Factor Lambda Control (FLC)
- Fuel Pulse Width vs test
- Exhaust Gas Temperature per each cylinder
- Fuel Flow Calculated & Fuel Flow Measured.

The above-mentioned metrics were used as a means to understand fuel injection parameters like mixing, spray, and flow deviations at a high level (i.e., using time-averaged parameters). Figure 44 shows the lambda8 ratio for each cylinder in an engine. The lambda was measured from the exhaust manifold connected to each cylinder. Cylinders 1, 3, 5, and 7 are on bank 1 whereas cylinders 2, 4, 6, and 8 are on bank 2. The firing order is 1-8-7-2-6-5-4-3.

As shown in the Figure, differences in lambda trends were observed between individual cylinders. Also from Figure 44, the lambda ratio of cylinders 4 and 7 has a consistent lambda from the beginning to the end. The lambda ratio of cylinders 1, 2, and 3 was reduced whereas the lambda ratio of cylinders 5, 6, and 8 was increased as the test progressed.

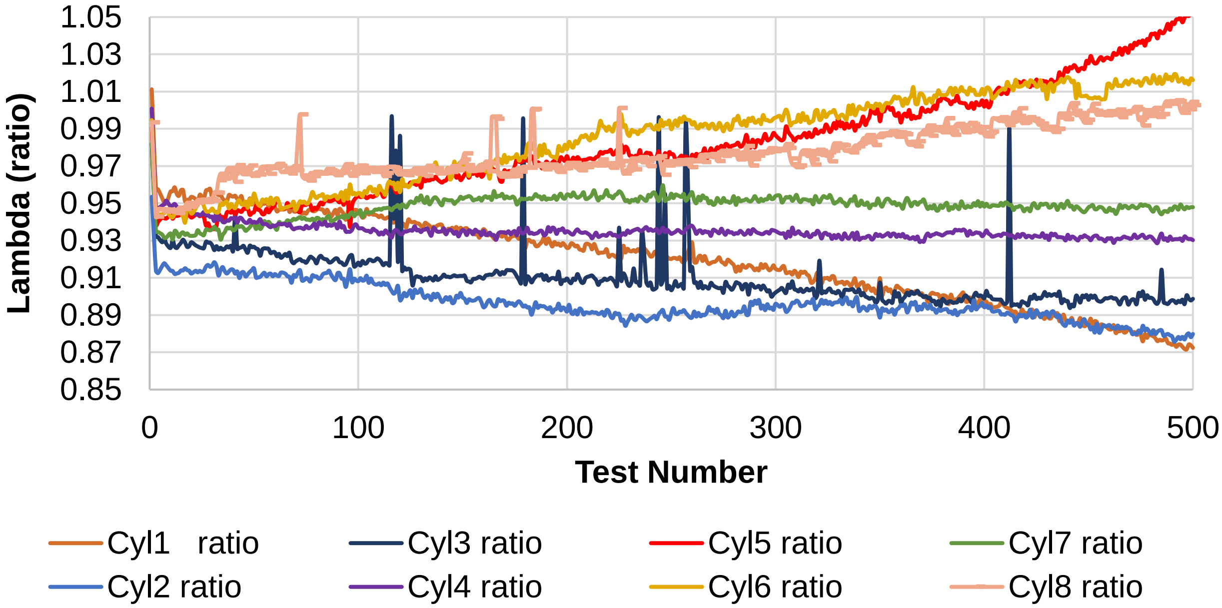

Figure 44: Lambda ratio and its deviations for each engine cylinder

The likely cause for the deviation of the lambda ratio was the degradation of injectors either because of fuel vaporization or because of other external conditions. To rule out external factors, the team performed exhaust gasket leak checks around the exhaust gaskets and also performed lambda/NOx sensor checks. The gaskets and lambda/NOx sensors were also replaced; however, these checks/replacements yielded no change or correlation to the observed trend. On the internal side, if the cylinder fueling variation is due to injector degradation, to be 100% sure whether it is injector wear and damage from propane or the fact that the Stanadyne development injectors used in the testing process are in the pre-production phase, the injectors were sent to Stanadyne for Root Cause Analysis (RCA) and to under other Failure Mode Effect Analysis (FMEA). Figure 45 shows the lambda ratio of each bank of cylinders compared with the nominal lambda value and the factor lambda control (FLC) for each cylinder.

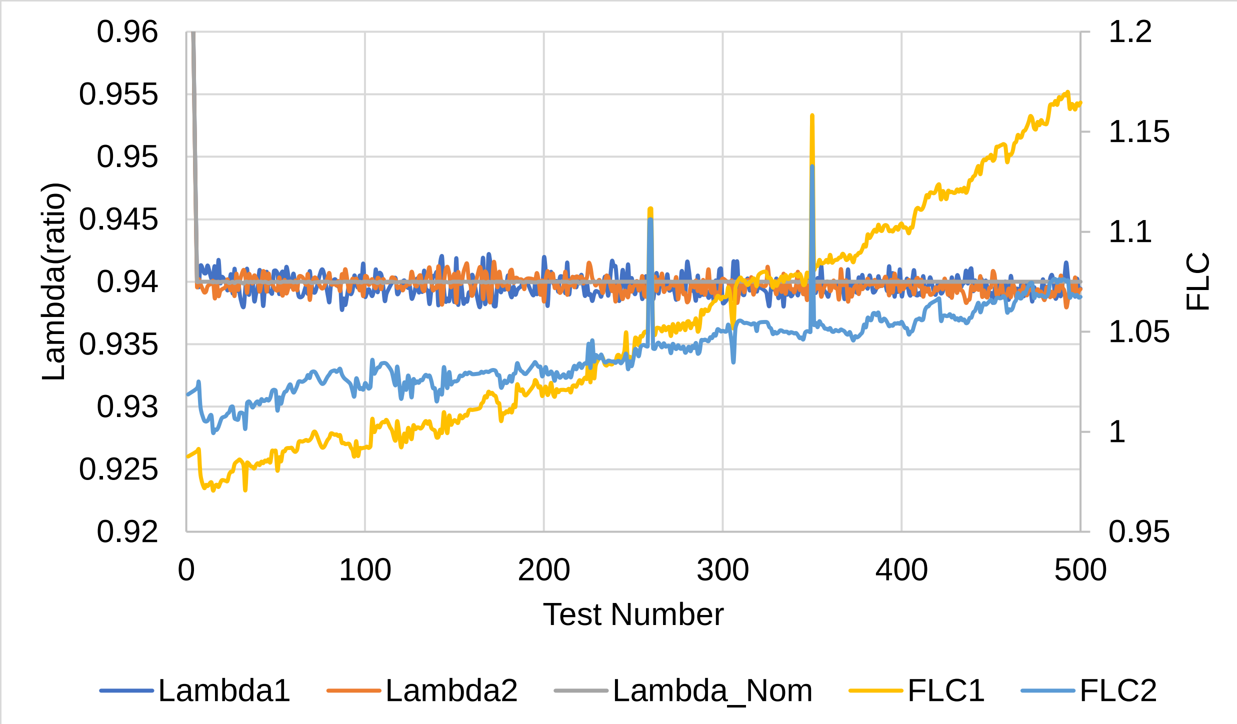

Figure 45: Lambda ratio and Factor Lambda Control (FLC) for each bank.

As depicted in Figure 45, the lambda ratio for each bank followed the nominal value however, the factor lambda control (a multiplier in the engine control unit that modifies the injection pulse width to reach the target lambda value) increased as the test progressed. The increase in FLC for bank 1 is much higher than the increase in FLC for cylinder bank 2. The increase in injection pulse width (or the Injector energized time) as the test progressed followed the same trend as FLC as shown in Figure 46. As mentioned before, it might be because of the degradation in the injectors either due to internal or external factors. On the other hand, even though the injector pulse width increased as the test progressed, the engine fuel mass flow rate from the high-side DI pump remained the same throughout the test. This represents the flow shift in the injectors with respect to injection pulse width from the linear flow range or maybe due to cavitation/vaporization of fuel in the injector when installed on an engine/or due to high temperatures experienced by the injector tip during operation. Figure 47 shows the variation in the injector fuel measured (orange line) versus the injector fuel calculated (blue line) from the linear regression model with a factor for lambda correction. As shown in Figure 47, the injector digressed from the linear regression model when installed on a firing engine.

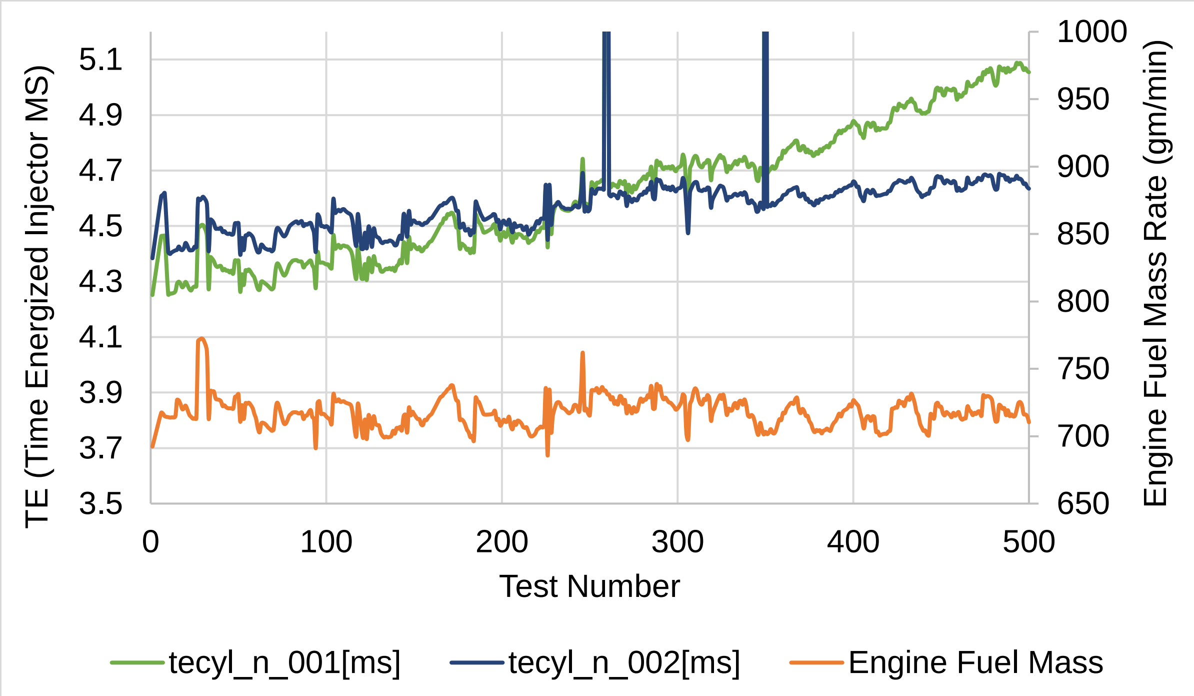

Figure 46: Variation in the injector energization time and engine fuel mass flow

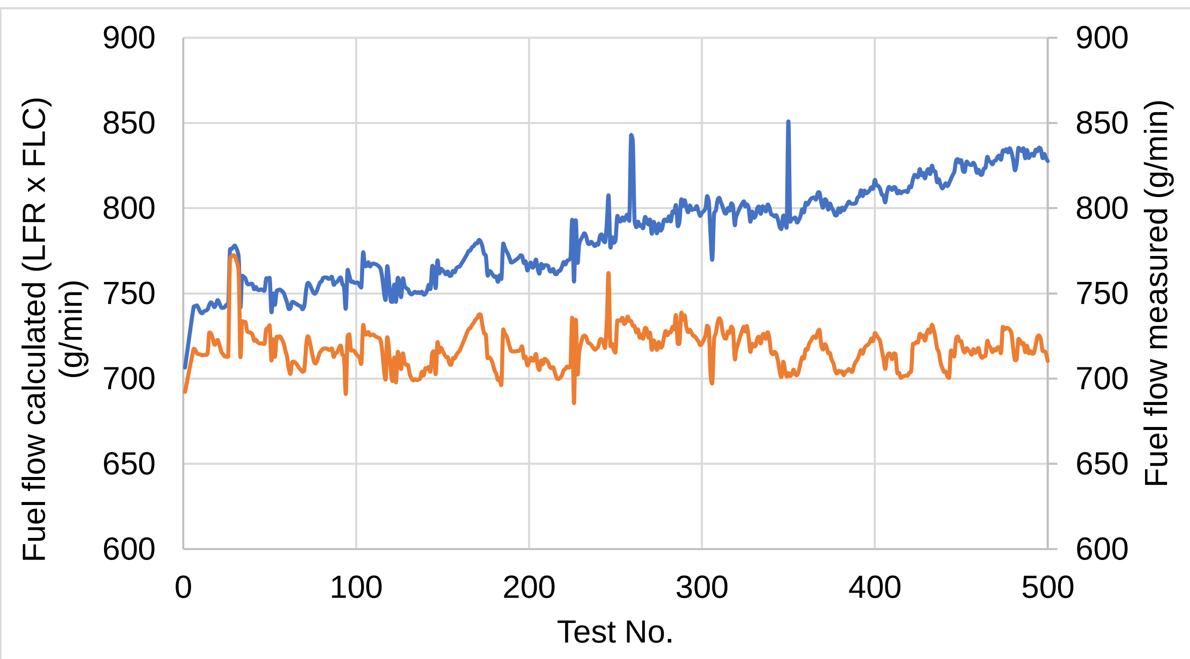

Figure 47: Variation in the fuel flow calculated vs fuel flow measured.

Figure 48 the variation in the exhaust gas temperature (EGT) for each cylinder in the engine. As shown in the figure, the exhaust gas manifold temperatures remained in-between 1350F to 1400F for all the cylinders except Cylinders 1 and 5. These cylinders are on the same bank and the lambda ratio of Cylinder 1 decreased whereas the lambda ratio of cylinder 4 increased as the test progressed. Again, the reduction in EGTs of cylinders 1 and 5 might be because of factors other than the lambda ratio. In addition, the steady & constant trend in EGT also proves the point that there were no exhaust gasket leaks during the 250 hrs. testing.

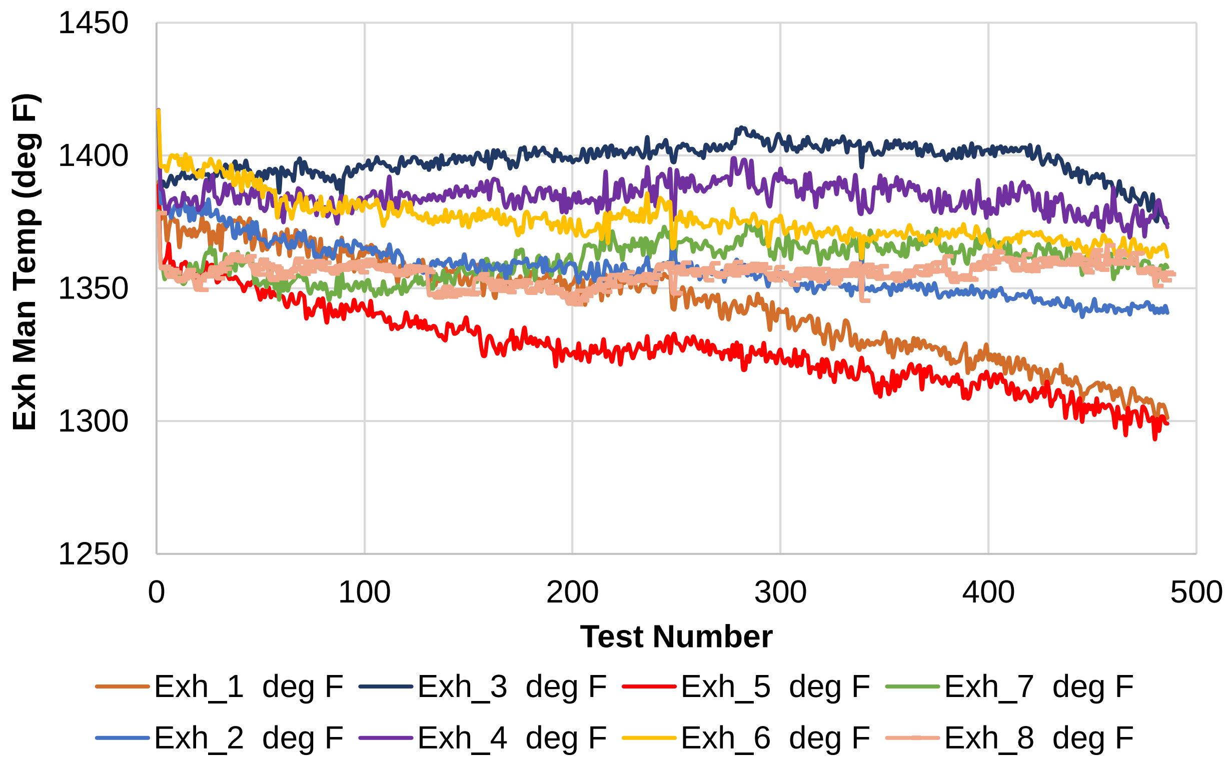

Figure 48: Variation in the cylinder EGTs during durability testing.

Overall, the Stanadyne development injectors were able to meet the flow and performance requirements, however, the drift in the injector fuel flow rate from the linear flow range was observed as the durability test progressed. Based on Stanadyne's root cause analysis, the team identified the decrease in lift due to wear at the retainer and armature interface. As a corrective action, the design was modified to prevent/reduce the wear at this location.

6.3.6 Engine Performance & Health

This section provides the results related to engine performance and health using various metrics with propane as a fuel. These metrics include power and torque characteristics, brake specific fuel consumption, and brake thermal efficiency to understand engine power and performance, aftertreatment temperatures, exhaust back pressure, and blowby to understand engine health.

Figure 49 shows the variation in the engine power and torque characteristics at 50% load and wide-open throttle conditions for a speed of 3200 RPM for 250 hrs. of durability testing. As shown in the figure, the engine power and torque characteristics stayed the same without any major deviations or outliers. Even though the injectors degraded as the tests progressed, the fuel flow from the injectors was sufficient enough to maintain the engine power and torque characteristics steady throughout the test.

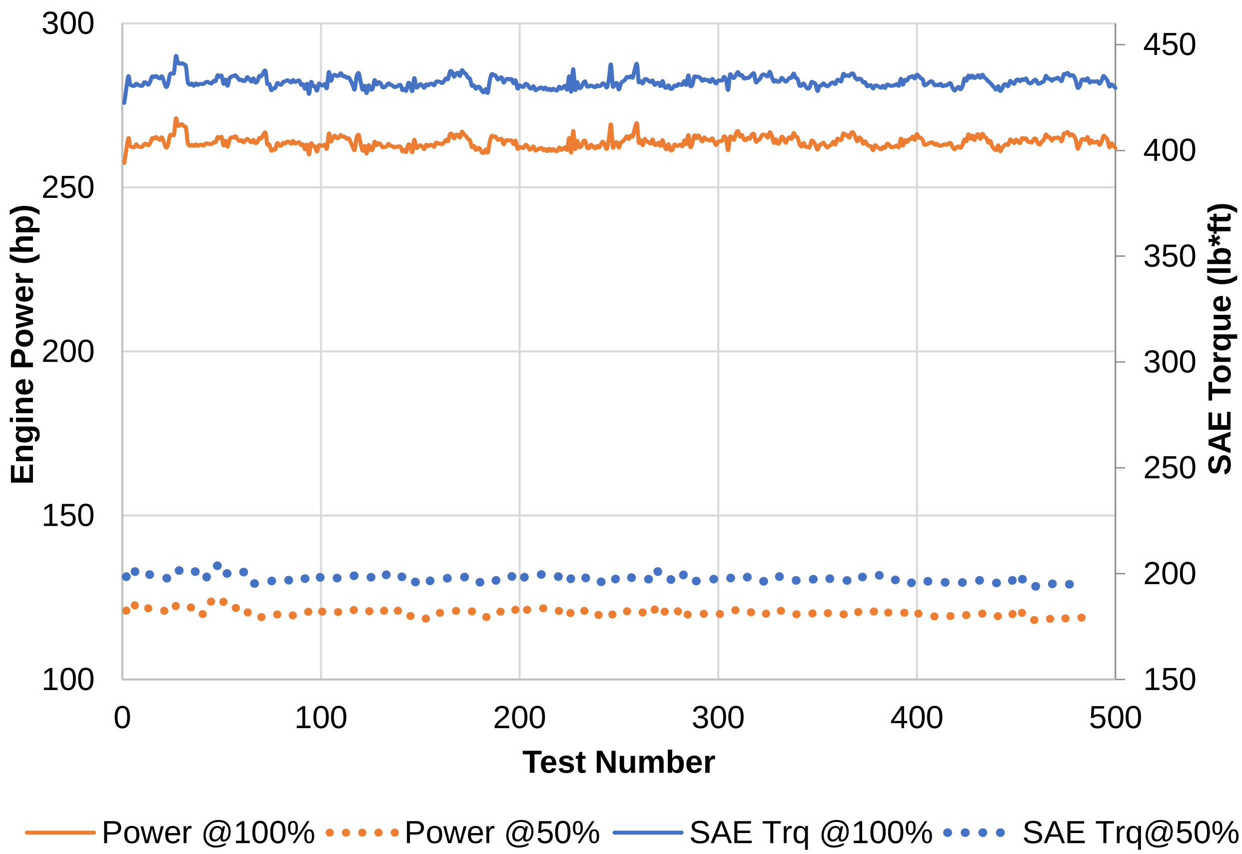

Figure 49: Variation in engine power and torque characteristics during durability testing.

The brake specific fuel consumption and brake thermal efficiency are listed for reference in Figure 50 and exhibited minimal change during the durability testing, even with the injector degradation. The BSFC and BTE data is from 3200 WOT conditions, with fuel enrichment (lambda of 0.94) for catalyst protection. Additional testing with emissions measurement would be required to understand peak efficiency capabilities of this engine on liquid propane while meeting current emission requirements. This testing is out of scope for this project.

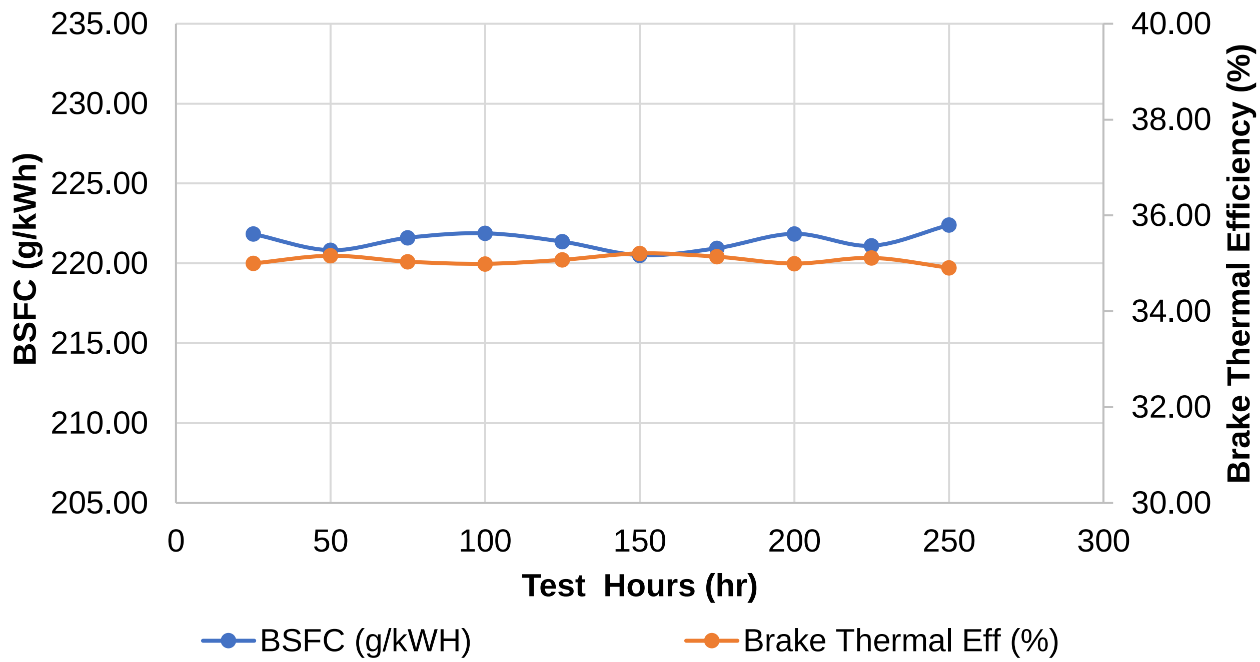

Figure 50: Variation in BSFC and BTE during 250 hrs. of durability testing.

Figure 51 shows the variation in the exhaust gas pressures and intake manifold pressure during the testing process. The exhaust back-pressure and intake manifold pressure did not have any significant changes/trends during durability testing and the trend did not correlate to trends exhibited by individual cylinder changes. The aftertreatment pre and post temperatures, shown in Figure 52, for cylinder bank 2 remained the same at 1500F whereas for cylinder bank 1, remained the same at 1475F until 300 test cycles after which they diverged significantly. At the end of 500 test cycles, there was a difference of about 100F between pre and post catalyst temperatures for cylinder bank 1. This difference is likely due to fuel injector delivery trends and lambda trends of that bank. Also, higher catalyst temperatures in the first ~5 to 10 cycles were due to the lambda target of 1.0, after that, the lambda target was reduced to 0.94 instead of 1.0 to lower exhaust gas and catalyst temperatures.

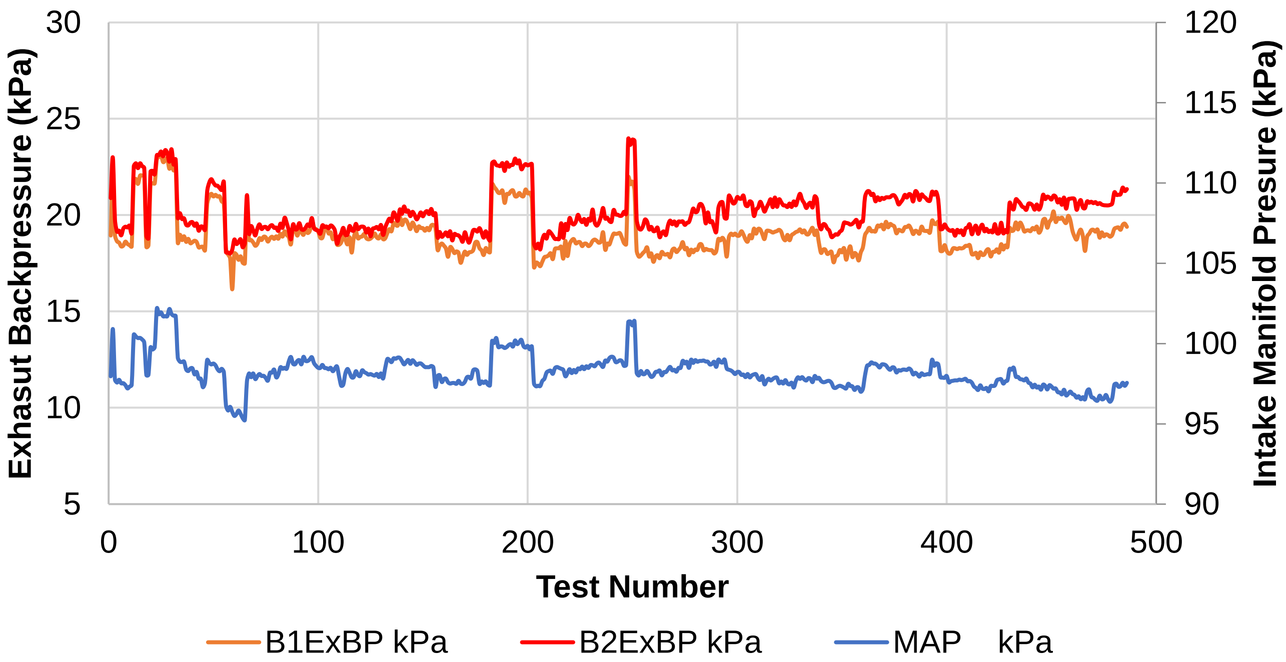

Figure 51: Variation in exhaust back pressures and intake manifold pressures during durability testing.

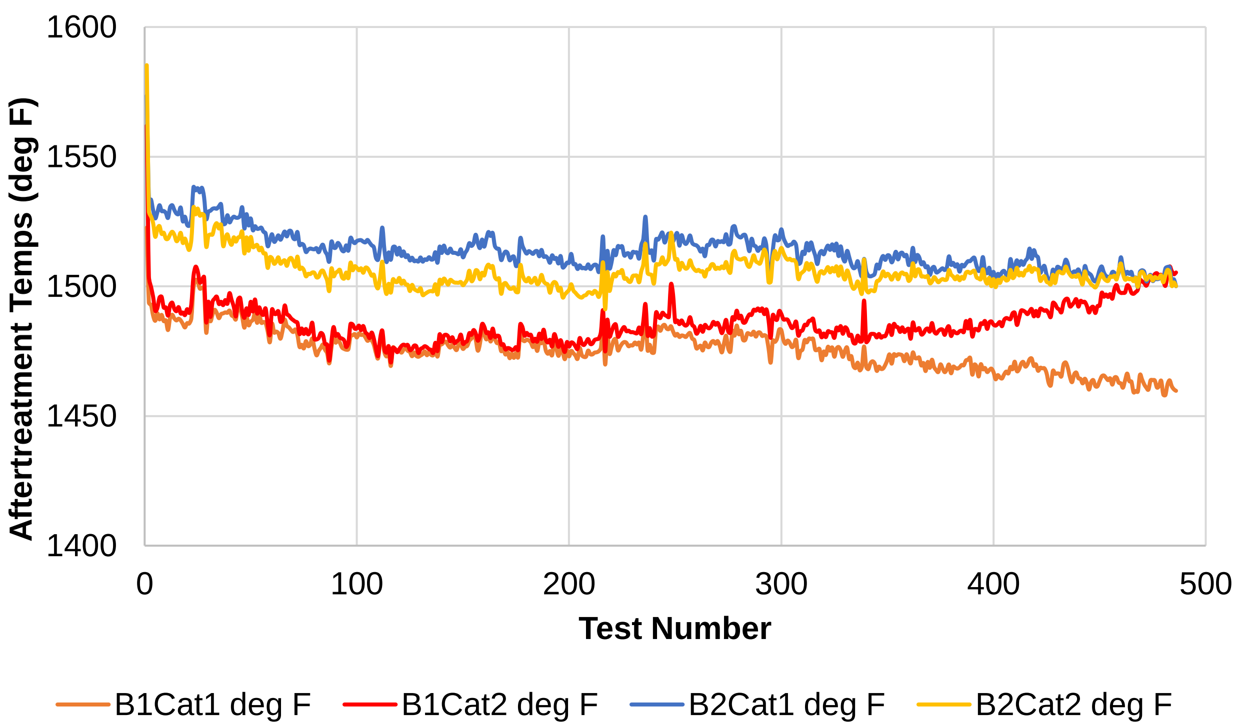

Figure 52: Variation in pre and post catalyst temperatures for cylinder banks 1 and 2.

Figure 53 shows the deviation in the engine blowby during the 500 durability test cycles. The engine blowby remained at minimum levels throughout the testing process. The value of 0 in between cycles 25 to 40 was due to a sensor malfunction and was fixed after identification.

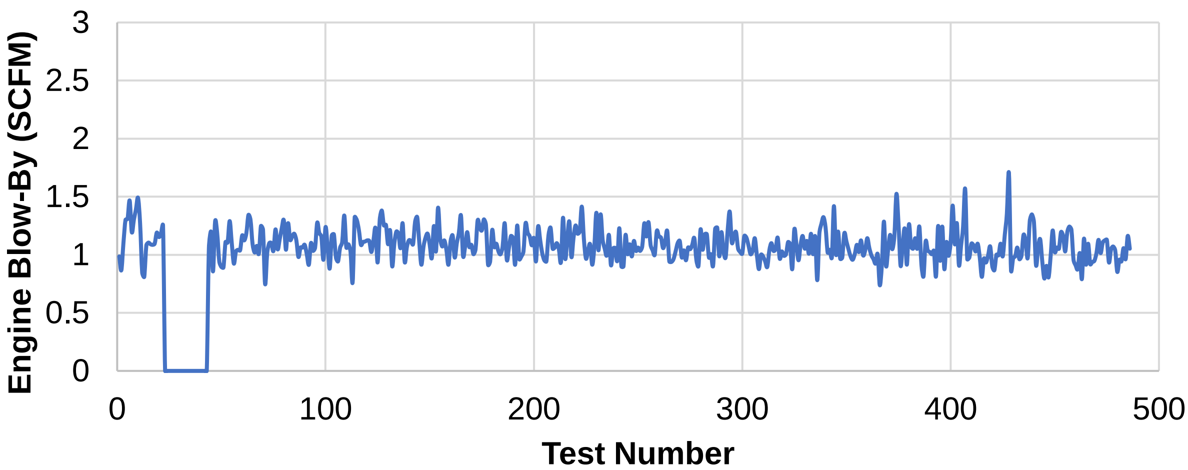

Figure 53: Engine blow-by trend during durability testing.

The blowby measurement method was very similar to that of the factory PCV (positive crankcase ventilation) routing method without any fresh air induction into the crankcase. In this method, initially, both the valve covers of the engine were vented to a remote mount plenum. The plenum, housing a filter/moisture separator, routed the gases to the blowby measurement sensor which were further routed through the intake tube, post-intake filter/pre-throttle body. The pressure differential between the crankcase and intake tube provided the flow (in this case blowby). Since the fresh air induction was closed, the only gases being measured were the gases generated by the engine blowby.

Overall, the engine performance and health remained stable and good during durability testing irrespective of injector degradation. The proposed DI system without injector degradation is robust enough to meet and exceed the DI fuel system metrics listed by PERC.

6.3.7 Hot/cold start and Heat Soak Operation

After every three 30-min durability cycles (or 1.5 hrs. of continuous testing) the engine fuel tank is refueled with propane. During the engine refueling period, the engine was heat-soaked for a period of ~8 to 10 minutes. The engine temperatures exceed 95° C at the beginning of the shutdown sequence. The engine after the refueling period entered the start procedure and went through the mode identification loop. The start procedure usually goes through modes 1, 2, 3, and 4 (Ignition – Prime, Start, Run, Low – Flow run respectively). While the engine fuel system functioned effectively for stable and transient conditions after the start procedure, it is too early to make strong conclusions about the maximum number of cycles the system is capable of. More analysis and testing need to be performed to understand the impact of short hot-soak, long heat-soak, dry running after long/short heat soak etc. on the performance of DI fuel system. In summary, more testing is required to continue to optimize and improve the control process encompassing all these abnormal operating modes. The team will closely work with Stanadyne on identifying and testing the abnormal operating modes.

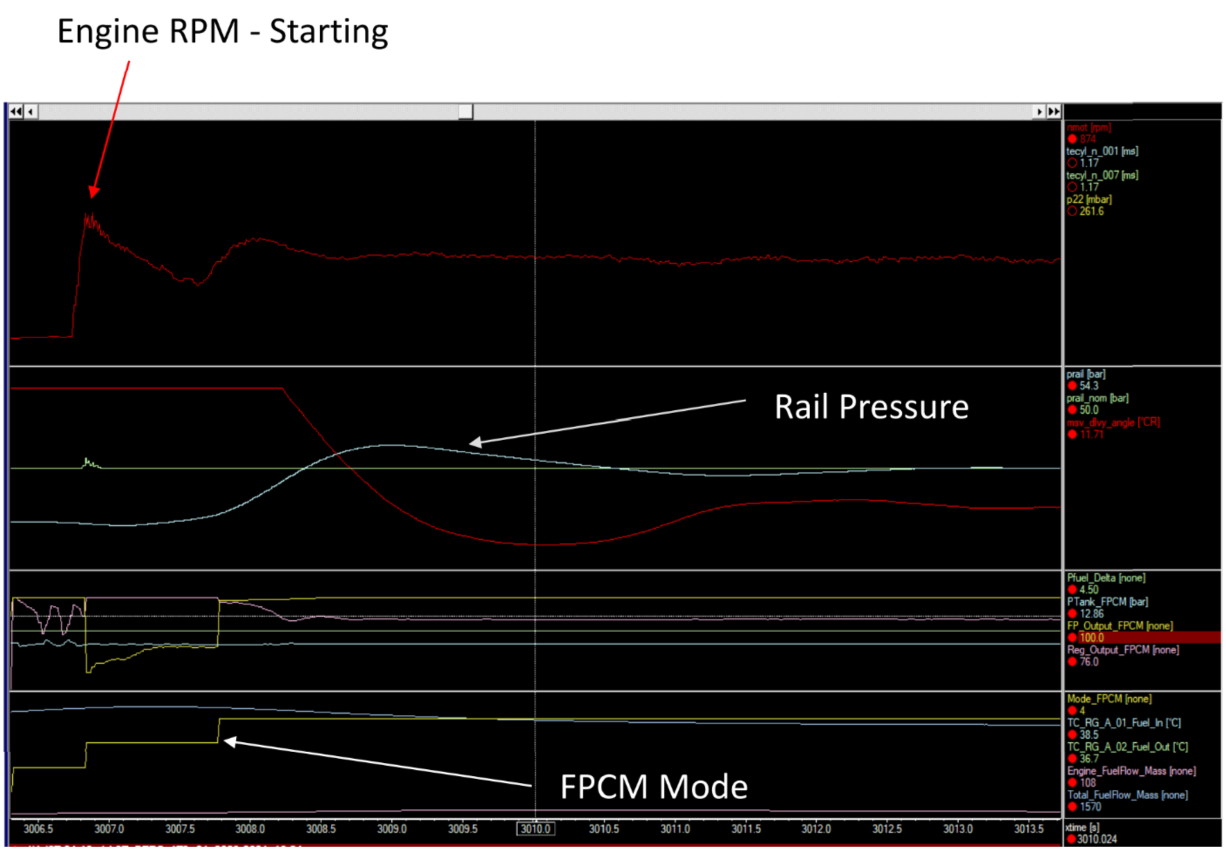

Figure 54: ECU screen depicting the start procedure & mode progression after heat soak.

6.3.8 Transient Rail Pressure Operation

This section provides details about the ability of the control system in maintaining the desired rail pressure when the engine is operating under transient conditions. Figure 55 below displays the trends of various parameters when the engine is subjected to transient operation. The first subplot shows how the speed of the engine was varied by the engine control system during the transient phase. The second subplot shows the actual (solid green line) and target (red dotted line) rail pressures for transient testing. These lines overlay very well making the red-line difficult to distinguish. The third subplot shows the low-side fuel system pressure (orange line), propane tank pressure (sky blue), and engine operating mode enumeration (light blue). Finally, the fourth subplot displays the variation in the engine fuel mass flow rate from the high-side fuel pump during the testing period.

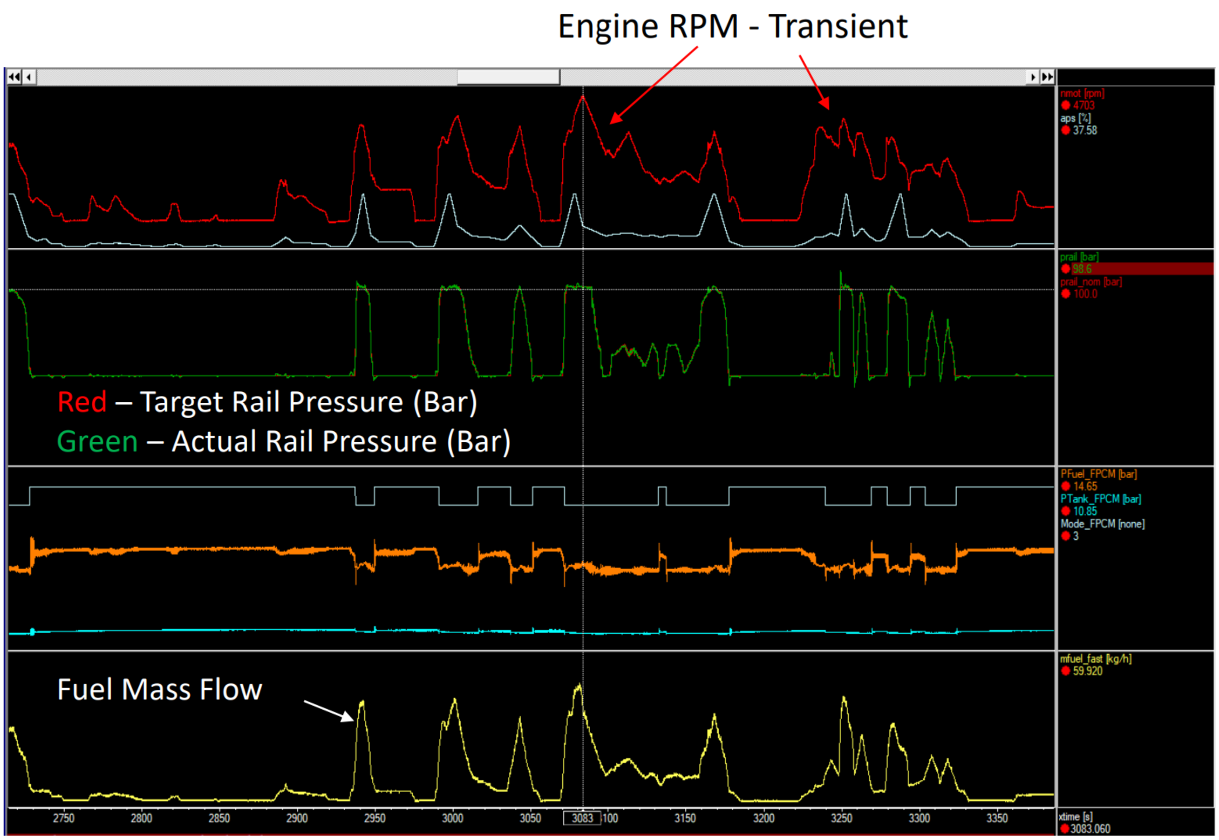

Figure 55: ECU screen depicting various parameters during transient operation.

As shown in the Figure, the actual/measured fuel rail pressure followed the target pressure defined by the engine control unit during transient and steady-state testing (similar to FTP). The fuel rail pressure has an average, absolute average, and maximum deviations of 0.007 bar, 0.417 bar, and 10.7 bar respectively. The low-pressure fuel feed variation may be one of the multiple factors that is responsible for the higher maximum deviation. Overall, these deviations are not out of the ordinary when compared to the gasoline DI system and the proposed propane DI fuel system met required operating metrics.

6.3.9 Post Durability Testing Inspection

This section describes the evaluations found from inspections made to the DI fuel system components and other engine components by the Engine Build & Test team after the 250 hrs. durability testing. The average leak down for an engine with this number of hours would be 5% to 15%. The leak down numbers, shown in Table 12 below, were measured from the durability test engine. Most cylinders have a leak down percentage below 5% except for cylinders 4 and 8. The increase in leak down for cylinders 4 and 8 is likely due to manufacturing tolerances and piston ring wear characteristics isolated to these cylinders. All observed leak down is within normal tolerance and exhibited pressure leakage past the ring sealing surfaces with no leakage found between valve sealing surfaces (i.e., valve or seat face). These values do not present a correlation to the cylinder-to-cylinder fueling imbalance that was observed throughout the course of durability testing.

Table 12: Post durability leak-down percentage for each cylinder

Post Durability Leak-down Percentage @ Ambient Temperature			
Cylinder (Bank 1)	Leak down %	Cylinder (Bank 2)	Leak down (%)
1	4%	2	4%
3	4%	4	8%
5	2%	6	4%
5	4%	8	10%

A cranking compression test was performed following the completion of durability and leak down testing. All cylinder cranking compression values were within 5% of each other, indicating proper cylinder operation with minimal to no degradation.

The cylinder heads were disassembled to measure valve recession, and to inspect valve sealing surfaces for wear/runout. The valves/seats showed no measurable amount of recession, and all valve seat characteristics were within OEM new service specifications.

In addition, oil samples were taken during maintenance intervals and these samples were sent out to Blackstone Laboratories for analysis. The results from the oil sample analysis are presented in Table 13 below. Apart from Magnesium, Phosphorus, and Zinc, all the elements were below the expected values. Magnesium, Phosphorous, and Zinc were found slightly above the expected values but still within acceptable limits.

Table 13: Oil Sample results from Blackstone Laboratories

Oil Sample - Blackstone Laboratories	
Lab Number	R28373
Code	63/1,430
Report Date	7/7/2023
Oil Type/Grade	5w30 Mobil1

UNIT ID	PERC 6.6 L8T - 250hr durability test			
	Element	Measured	Expected Value	Comment
Elements in PPM (parts per million)	Aluminum	5	6	Pass
	Chromium	0	1	Pass
	Iron	10	14	Pass
	Copper	9	10	Pass
	Lead	0	0	Pass
	Tin	1	1	Pass
	Molybdenum	120	142	Pass
	Nickle	0	0	Pass
	Manganese	1	1	Pass
	Silver	0	0	Pass
	Titanium	0	2	Pass
	Potassium	1	1	Pass
	Boron	40	43	Pass
	Silicon	6	14	Pass
	Sodium	2	5	Pass
	Calcium	755	1147	Pass
	Magnesium	713	670	Acceptable, Above Average
	Phosphorus	821	699	Acceptable, Above Average
	Zinc	926	825	Acceptable, Above Average
	Barium	0	0	Pass
	Measurement	**Values**	**Expected Value**	**Comment**
Properties	SUS Viscosity @ 210°F	56	n/a	Pass
	cST Viscosity @ 100°C	9.07	n/a	Pass
	Flashpoint in °F	420	>385	Pass
	Fuel %	<0.5	< 2.0	Pass
	Antifreeze %	0.0	0.0	Pass
	Water %	0.0	0.0	Pass
	Insoluble %	TR	<0.6	Pass
	TBN	4.9	>1.0	Pass

The Stanadyne injectors and pumps were also sent to Stanadyne for inspection and injector degradation root cause analysis. The results from the post-durability inspection analysis are shown in Figure 56. As shown in Figure 56, contact/sealing marks were observed for the high-side DI pump. Pre and post comparison of the mass flow rate of the pump showed no significant deviations for all the engine speeds at a given pressure.

On the other hand, the interface between the retainer and the armature in the DI injector showed significant wear. Pre and post comparison of the mass flow rate of the DI injector with respect to different injector pulse widths at a given rail pressure and fuel temperature showed a significant shift (greater than 10%). This explains the reason for the drift in the injector flow rate during the 250 hrs. durability testing. To mitigate this wear, Stanadyne modified the design of the retainer and the armature interface in the development injector.

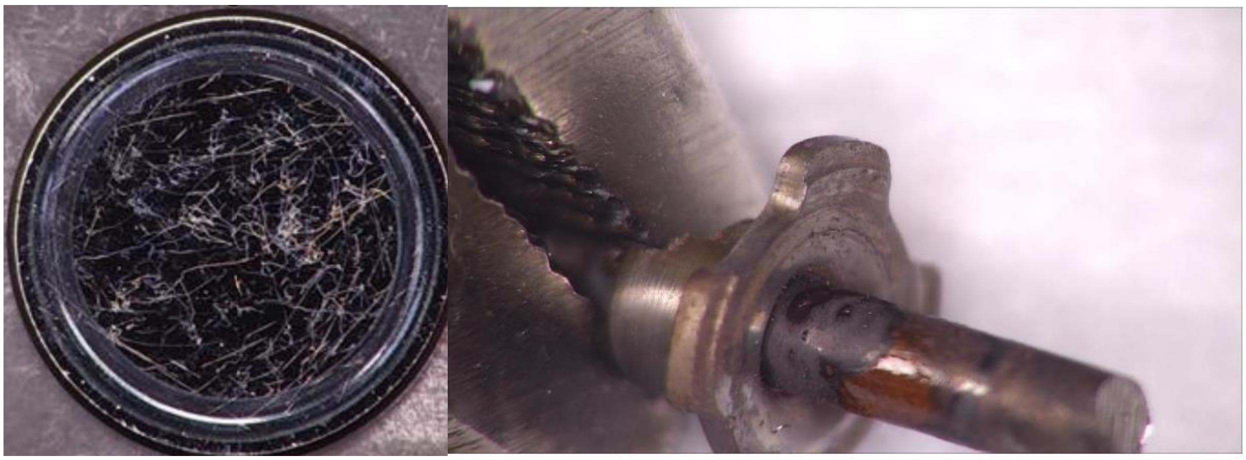

Figure 56: High-side DI pump outlet valve (left) and DI injector retainer and armature interface (right).

Following the completion of the firing engine durability test, the low side fuel delivery pump was inspected. Only one low-side Bosch fuel pump and Parker filter were used for the duration of firing engine testing. Throughout the durability test, fuel pump output was closely monitored for any drops in delivery pressure. The visual inspection process began with the post-pump fuel filter (Parker LPGD-201-05). The post-pump filter casing was cut to view the filter media pleating. The pleating was clean with no debris found that may have indicated pump wear or potential failure. Following the filter inspection, The Bosch low-side fuel pump was disassembled for inspection. No wear indicators outside of what's exhibited during normal operation was found. With no degradation of pump outlet pressure, and the acceptable condition of the internal components, the low-side pump proved to be an adequate design for use with LPG.

7. Conclusions

Overall, the proposed fuel system with in-house developed vapor lock inhibitor hardware and software addressed the design limitations faced by other LPG DI applicators till today, and also proved the efficacy of liquid propane as a fuel for direct injection. Since the project's inception, the project team mainly focused on understanding the operational, performance, and reliability characteristics of three high-pressure DI pumps (GM's Stanadyne pump, Bosch's HDP6 pump, and Stanadyne LPG Development pump) and three DI injectors (Delphi 250, Bosch 350, and Stanadyne LPG Development injector). Various steady, dynamic, performance, engine performance, component durability both short-term, and long-term, and visual tests were performed on all the designs at different engine speeds, delivery pressures, and fuel temperatures to encompass the entire engine operating range experienced by the proposed fuel system.

For the non-firing phase testing, the following conclusions can be made:

- All the fuel system components (high-side DI pumps & DI injectors) met the requirement metrics needed to operate the proposed GM L8T engine.
- The high-side fuel pumps operated steadily at medium loads for all the delivery pressures and fuel temperatures in the range 20° - 50° C. However, due to fuel vaporization, the high-side DI fuel pumps failed to achieve the steady-state at low and medium loads without the regulating valve and vapor lock control strategies.
- Of all the parameters, delivery pressure and engine speed had a dominant effect on the flow and performance characteristics of the pump.
- Due to fuel vaporization, the volumetric efficiency of the high-side DI pump was reduced anywhere in between 0% - 35% depending on the operating conditions when compared to gasoline operation.
- The two commercially available off-the-shelf pumps and injectors were durability tested for 8.1 million pump cycles and 2.7 million injector cycles respectively.

For the firing phase testing, the following conclusions can be made:

- The team worked with the Tier I fuel systems OEM (Stanadyne Inc.) to develop a high-side fuel pump and an injector specific for LPG based on the modifications and outcomes from the non-firing phase testing.
- The power and torque characteristics of the GM L8T engine, operated with the proposed Stanadyne LPG DI fuel system, were able to meet or exceed the output of its gasoline baseline.
- The exhaust gas cylinder temperatures for propane for all the cylinders were lower by 25F to 75F when compared to gasoline due to ignition timing advance.
- The proposed Stanadyne LPG DI fuel system was durability tested for 250 hrs. simulating on-road, low flow, idle, heat soak, refueling, and restarting conditions, through a 30-minute cycle that was repeated 500 times. During the durability

testing process, the pump and injectors were tested for 50.2 million pump cycles and 16.7 million injection cycles respectively.

- The low-side DI pump and the high-side pumps were able to maintain their respective delivery pressures for the entire durability tests without any adverse deviations.
- Drift in the injector flow rate from linear flow range was observed as they were durability tested for 250 hrs.
- Pre and post comparison of the mass flow rate of the DI injector with respect to different injector pulse widths at a given rail pressure and fuel temperature showed a significant shift (greater than 10%). After disassembly, the team identified the decrease in the injector lift due to wear at the interface between the retainer and the armature in the DI injector. The team modified the design of the interface after working with OEM (i.e., Stanadyne).
- Post durability wear and degradation assessment of the engine showed no sign of wear or blowby for valves, valve seats, cylinder rings etc.

Overall, the testing in both the phases proved the efficacy of LPG as a fuel for direct injection while addressing the previous design limitations. Furthermore, given that limited modifications were made to the proposed system and with Tier I OEM (Stanadyne Inc.) connection, the proposed technology has a high level of feasibility for cost-effectiveness and widespread commercialization.

8. Next Steps

1. Make Marketing & business advancements to promote results and evaluate the paths to utilize LPG-DI technology.
2. Commercialization opportunities of the LPG-DI fuel system components and vapor lock inhibitor system.
    a. New systems - OE level
    b. Alternative conversion systems to replace gas mixers on DI native engines.
    c. Evaluation of off-road and power generation applications.
3. The results from this program provide the confidence to proceed forward with on-road or off-road application / demonstrator.
4. Additional analysis and testing need to be performed to understand the impact of combustion, efficiency, and emissions benefits of LPG DI vs. competitive fuels and compared to port fuel LPG. This could be completed independently or collectively with point 3.

9. References

1. SAE J2714_201502 Standard for Gasoline Direct Injection Pump, SAE International, 2015. https://doi.org/10.4271/J2714_201502

2. Lacey, J., Harjon, A., Farzad, P., Michael, B., Robert, G., Charles, L., Brendan, B., and Steve, R., "Flash vaporization of propane in an optically accessible, directly injected engine.", International Journal of Engine Research, Vol 22, No. 2, 2021, pp: 685-696.

3. SAE J2713_201808 Standard for Direct Injection Gasoline Fuel Injector Characterization, SAE International, 2018. https://doi.org/10.4271/J2713_201808

4. Book: Bosch Motorsport MS6A_Base_900 firmware FDEV (Function development), Page 448.

5. Stanadyne, "Gasoline Direct Injection SP1250-200," *Stanadyne*. https://www.stanadyne.com/gasoline-products/sp1250-200/

6. Hohenberg, G. F., "Advanced approaches for heat transfer calculations," SAE Technical Papers, No. 790825, 1979.

7. Gas Encyclopedia Air Liquide, "Propane | Gas Encyclopedia Air Liquide," *encyclopedia.airliquide.com*, 2023. https://encyclopedia.airliquide.com/propane#properties

8. US EPA,OAR, "Dynamometer Drive Schedules | US EPA," *US EPA*, Sep. 16, 2015. https://www.epa.gov/vehicle-and-fuel-emissions-testing/dynamometer-drive-schedules
9. "Emission Test Cycles: Heavy-Duty Supplemental Emissions Test (SET)," *dieselnet.com*, 2004. https://dieselnet.com/standards/cycles/set.php

10. C. Sharp, "Emission Test Cycles: Low Load Cycle (LLC)," *dieselnet.com*. https://dieselnet.com/standards/cycles/llc.php (accessed Aug. 31, 2023).

11. Environmental Protection Agency, "Subpart F—Performing an Emission Test Over Specified Duty Cycles," *Code of Federal Regulations eCFR*, Jul. 13, 2005. https://www.ecfr.gov/current/title-40/part-1065/subpart-F