Piezoelectric valve actuator for flexible diesel operation

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ABSTRACT

Midé Technology Corporation, under the supervision of the U.S. Army, is developing fast hydraulic valve technologies for fuel injection systems. Midé aims to address the Army's 21st Century Vehicle programs by providing the flexibility to achieve economic, environmentally friendly, and power dense diesel operation from a single platform. The technology couples a highly efficient gained piezoelectric actuator to a diesel unit injector's control valve spool. Piezoelectric actuation enables proportional authority over the injector's control valve, as opposed to traditional digital (on/off) operation. This authority allows the integrated device to provide electronically controlled fuel injection rate shaping capability. Each injection event profile may be independently shaped to govern diesel engine operation in one of three selectable modes: Lean, for fuel efficiency; Clean, for reduced emissions; Mean, for improved battlefield performance. To date, Midé has shown injection rate shaping capability in the laboratory using the industry standard "rate tube test" to measure injection profiles. Future development will focus on an engine demonstration of Lean, Clean, and Mean™ operating mode flexibility using rate shaping technology.

Keywords: Piezoelectric, actuator, valve, diesel, injector, rate shaping, flexibility, lean, clean, mean

1. INTRODUCTION

The Army's 21st Century Vehicles need the flexibility to operate in Lean, Clean, or Mean mode. The Army alone has a fleet of over 246,000 tactical wheeled vehicles that drive more than 8,000,000 miles combined annually. In peacetime, it is important that this fleet of vehicles is as environmentally friendly as possible: **Clean** (also referred to as Green). Since fuel constitutes 70% of the bulk tonnage needed to sustain a military force on the battlefield, high fuel economy is critical to survival with a weak supply line: **Lean**. However, there is no substitute for high power density to be effective and elusive in an ever-changing battle theatre: **Mean**.

In a diesel engine, combustion within the cylinder is closely related to the injection process. By controlling injection timing and duration, fuel quantity, and rate shape (flowrate profile as a function of time), it is possible to effectively control engine performance[1,2]. Unfortunately, a single injection rate profile will not simultaneously provide the lowest emissions, best fuel economy, and highest torque performance. However, if an engine's fuel delivery system can be commanded to yield a Lean, Clean, or Mean engine, depending on operational needs, the Army can meet all three requirements. For example, the engine controller would set Clean mode to minimize emissions during peacetime. In wartime, the engine controller could switch to either Lean or Mean mode, depending on operational needs. When hauling cargo or transporting troops, the Lean mode would minimize fuel use. Mean mode would be engaged when the vehicle is under fire or when its load capacity must be maximized.

Several diesel engine manufacturers introduced electronic fuel injection in the 1980's to address tighter emissions regulations but maintain fuel economy and performance. In electronic fuel injection systems, a solenoid valve replaced the traditional mechanical fuel metering method. This change increased the accuracy and individual control over fuel delivery as a function of load, speed, and ambient conditions (air and fuel temperatures and pressures). However, two-way solenoid valves are typically limited to digital operation: they are either fully open or fully closed. This characteristic is beneficial for controlling fuel quantity, and in some cases the injection timing, but is generally poor for shaping the flowrate profile.

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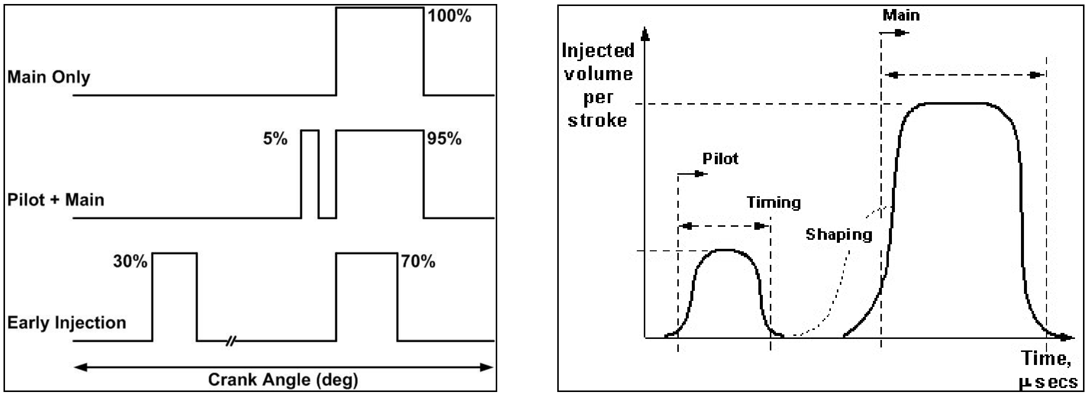


Fig. 1. Traditional injection profile control consisted of timing and fuel quantity only (left)[3]. Tomorrow's control will provide injection rate shaping capability (right).

An electronic, high speed, *proportional* control valve could add rate-shaping capability to simple injection timing and fuel quantity control, as shown in Fig. 1 above. This authority would produce large improvements in the performance and flexibility of electronic fuel injection systems in the Army's fleet of diesel-powered vehicles (see Fig. 2 below). To accomplish this, Midé has developed a high-speed valve for diesel fuel injection using a mechanically gained piezoelectric actuator.

The evidence for the move to active materials over traditional electromagnetics is strong. Piezoelectric (or piezo, for short) actuators have the bandwidth needed for extremely fast switching. Typical switching times can be less than 100 μs with no delays, while solenoid valves are up to ten times slower and have substantial lag due to magnetic reluctance. Active materials are capable of delivering much higher actuation forces. This characteristic lends itself to opening larger valve flow sections than comparable solenoid actuators to enable faster needle velocities. Electrical energy conservation is also an advantage, since energy can be regained from a piezoelectric load due to its capacitive nature. Electromagnets waste away most input power through resistive heating losses. Industrial development has already thrust towards piezo-actuated valves for diesel fuel injection, especially in European markets. Siemens, Bosch, Delphi, and others have produced piezo-driven injectors for common rail diesel fuel systems[4,5,6].

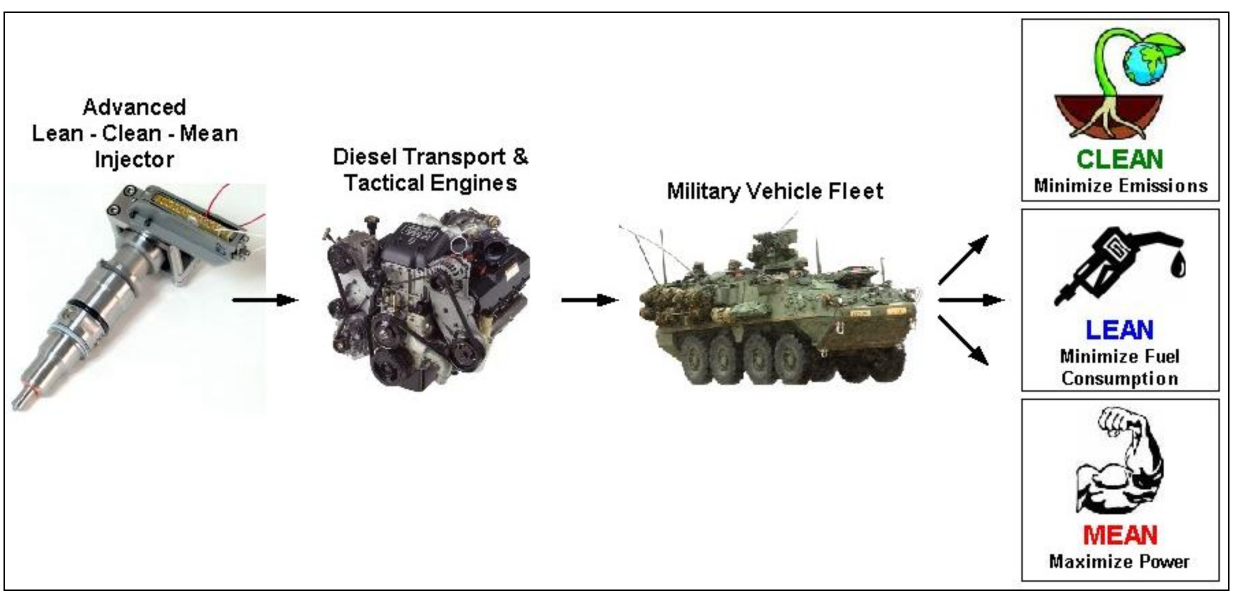


Fig. 2. By integrating an advanced, high speed injector into military diesel engines, the fleet will have operational flexibility to run in Lean, Clean, or Mean mode depending on the situation[7].

2. APPROACH

An off-the-shelf, hydraulic unit injector (HUI) was selected as the platform for rate shaping demonstrations. Along with the HUI, there is a technology push toward two other types of diesel injector for modern fuel delivery systems, including the *mechanical* unit injector and the high-pressure common rail injector. The HUI's design and operation are different from both mechanical unit injectors and common rail injectors, but also have some similarities to each (see Fig. 3 below). Like mechanical unit injectors, the HUI acts as a local pump on low-pressure fuel supplied throughout the system by a small, centralized pump. An intensifier piston pushes a plunger, which compresses the available quantity of fuel in a small chamber that extends downward and around small lobes on the injector needle itself. When fuel pressure in the chamber reaches a critical level, the upward hydraulic pressure on the needle lobes overcomes the downward seating force (see Fig. 4 below). At this point, the needle lifts from its seat to expose the nozzle, and high-pressure fuel is injected into the engine cylinder.

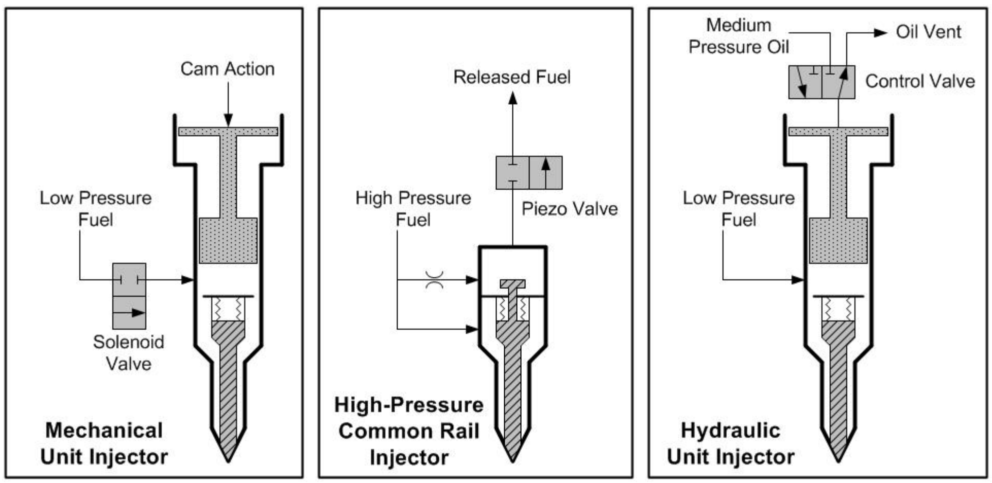


Fig. 3. Simplified comparison among three modern diesel injector designs.

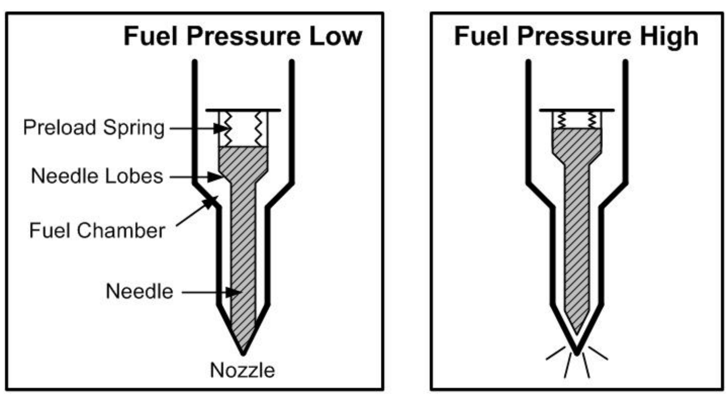


Fig. 4. Needle valve mechanics in a unit injector. Non-injecting (left) and injecting (right) states are shown for cases of low and high-pressure fuel in the chamber, respectively.

Hydraulic unit injectors and high-pressure common rail injectors both have electronically controlled timing and duration. Rather than relying on camshaft rotation for fixed injection timing, the HUI design includes solenoid actuators. The coils are used to translate a spool within the control valve to regulate a medium pressure hydraulic oil supply. When opened, the control valve allows the medium pressure oil to enter a chamber above the intensifier piston. As the pressure builds on the piston face, it travels downward, compressing the low-pressure fuel as mentioned earlier. When closed, pressurized oil in the chamber above the piston is vented through the control valve to its surroundings.

Because of its venting design characteristic, HUI's resides completely under the cam cover to allow recovery of used oil after each injection event. Electrical connections to the control valve are made through the cam cover itself. Fig. 5 below illustrates the two flow states of a typical hydraulic unit injector control valve.

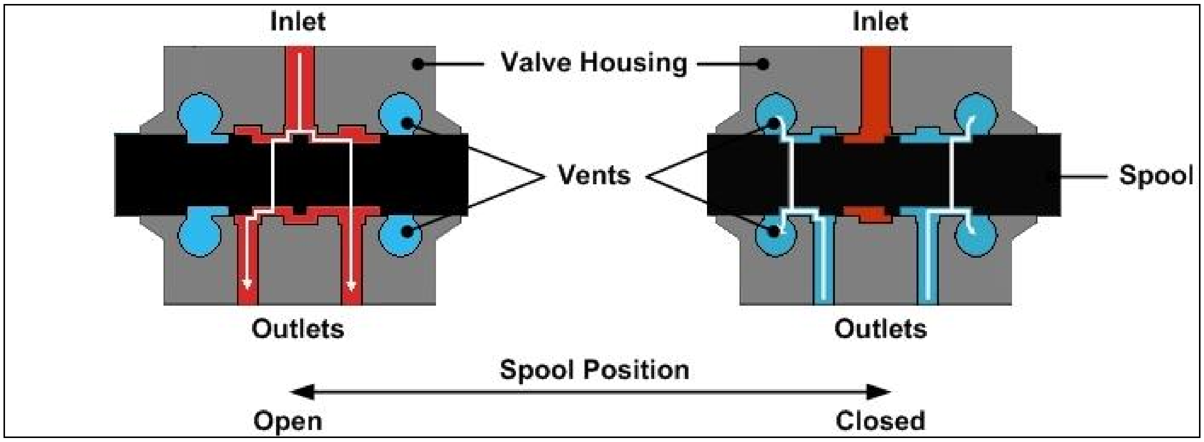


Fig. 5. Example of hydraulic unit injector control valve states with spool in the open position (left) and closed position (right). Piezo actuation would enable an infinite number of spool positions between fully open and fully closed.

This fuel delivery design eliminates the sluggish injections common to traditionally mechanical systems at low engine speeds. It also allows for electronically variable injection timing and duration, not possible with simple cam action alone. Piezo actuation of the spool within this control valve was the focus of Midé's development. Where HUI solenoids limit the control valve to digital states only (open-close), a proportionally actuated spool adds an infinite number of flow characteristics to the valve. This authority would increase rate-shaping capability of the HUI.

Looking downstream from the control valve, one similarity among all three injector designs is the fuel injection needle valve in the nozzle itself. It is primarily this needle valve that determines the injection profile under given conditions. Typically, the needle must overcome fairly large seating forces before finally lifting to allow injection – usually hundreds of Newtons in a unit injector design. At the start of this development effort, it was unknown whether or not rate shaping would be feasible using a traditional needle valve and nozzle. Proportional authority over the timing and flowrate of working hydraulic fluid through the injector's control valve may not translate to shaped injection profiles further downstream at the nozzle outlet.

3. PROGRAM TARGETS

An industry standard testing methodology was followed – referred to in literature as the "two-microphone test" or "rate tube test" – to demonstrate Lean, Clean, and Mean shaped fuel injection profiles in the laboratory. The rate tube test is a method of measuring the rate-of-injection (ROI) by monitoring the output pressure fluctuations of a fuel injector coupled to a known hydraulic load.

First, a literature survey was conducted to identify the best target injection profile shapes for Lean, Clean, and Mean engine operation. Recent models of injection rate shaping have confirmed that NO_x emissions, fuel economy, and output torque can be significantly altered with various injection profiles[1,2,8,9]. Based on the survey results, Fig. 6 (below) shows generalized injection rate profiles that can provide for Lean, Clean, and Mean engine operation. Mean, which corresponds to maximum torque output, is achieved by creating the highest possible cylinder temperatures for the longest period of time throughout the injection cycle. The Mean profile is characterized by a steep rising edge, followed by a sustained 'dwell time' and sharp falling edge. In direct contrast, Clean mode corresponds to lower cylinder temperatures. Emissions are minimized by achieving both a low initial injection rate for NO_x reduction, combined with an abrupt injection termination for soot reduction. For Lean performance, it is desirable to achieve combustion, or maximum cylinder temperature, at top dead center (TDC). The best shape for this combustion timing includes gradual rising and falling flowrates. Analytically, functions for the Lean, Clean, and Mean shapes are shown in Table 1 below. Note that the *A* coefficients represent the relative amplitude of the function, *B* coefficients relate to the width (duration) of the injection function, and *t* is the time to TDC.

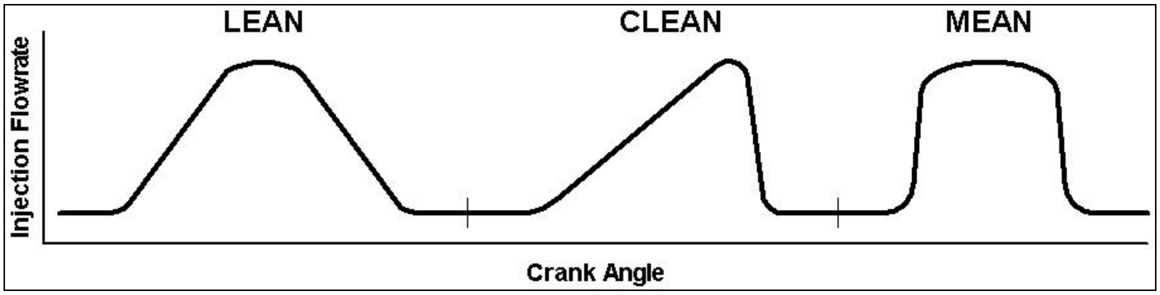


Fig. 6. Lean, Clean, and Mean target injection profiles for the same equivalent total injection volume.

Table 1. Analytical functions for Lean, Clean, and Mean target injection profile shapes.

Mode	Lean	Clean	Mean
Function	$y_L = A_L\, e^{\frac{-(x-t)^2}{B_L}}$	$y_C = A_C\, e^{\frac{-(x-t_S)^8}{B_C}}\, e^{\frac{-(x-t_F)^4}{B_C}}$	$y_M = A_M\, e^{\frac{-(x-t)^6}{B_M^6}}$

The injection profile shapes shown above in Fig. 6, and analytically represented in Table 1, served as the target shapes for the advanced high-speed injector development. Although it was not reasonable to expect injection traces to follow the mathematical functions exactly, they did provide a frame of reference and preliminary spool trajectory command signals. In general, the goal was to demonstrate each of the injection shapes in the rate tube test, on pressure and temporal scales appropriate for diesel operation. Typical dynamic pressures measured with the rate tube test setup are on the order of 2000 – 6000 psi, while relevant durations range between 1 and 3 ms.

After characterizing the target injection rate shapes, an assessment of the piezo actuator's performance requirements was made prior to design (see Table 2 below). Without a valid model of the injector's internal hydraulics and mechanics, the necessary speed or bandwidth of the actuator was speculative to some degree. From limited injector data, the OEM-configuration (solenoid-actuated) control valve could theoretically operate up to about 1200 Hz. Adding a 25% margin to this range resulted in a target actuation bandwidth of 1500 Hz. The original control valve spool weighed over four grams, which meant the actuator would be required to drive at least five grams once interfacing or coupling hardware was added between the actuator and the spool. Finally, spool stroke was known to be a minimum of 400 μm from the fully open to the fully closed positions.

Table 2. Summary of actuator and injection targets.

Actuation Stroke	400 μm
Usable Actuation Bandwidth	1500 Hz
Driven Mass	5 g
Injection Rate Shapes	Lean/Clean/Mean
Injection Durations	1-3 ms
Measured ROI Pressures	2000–6000 psi

4. ACTUATOR DESIGN & STATIC TESTING

The piezoelectric actuator and spool coupling mechanism were the two main areas of development under this program. Unfortunately, the target spool stroke of 400 μm would require a prohibitively long single piezo stack. Generously assuming 0.15% reversible strain, this corresponds to a 267 mm stack length, which is not practical for most applications. On the other hand, the 90 N force required to drive a 5 g spool mass at 1500 Hz was well below the force capabilities of most commonly manufactured piezoelectric stacks. Considering the force and displacement targets listed above, the actuator should be designed for a stiffness of 200-300 N/mm, about two orders of magnitude less than a typical piezo stack alone. These mismatched force and displacement requirements, or stiffnesses, steered development toward *gained* piezo actuators, which would multiply a given piezo stack's stroke at the expense of driving force, ultimately resulting in high stroke actuator with lower overall stiffness.

Several different mechanical gain concepts were investigated, the most important of which was a proprietary dual stack design. In this design, discussed in greater detail below, two piezo stacks were arranged physically in parallel but mechanically in series to produce a compact, large stroke device. Prior research using this type of actuator meant its maturity level was higher than other concepts at the start of the project[7,10]. Eventually, it would become the only actuator integrated onto the unit injector control valve for rate tube testing.

Detailed actuator design centered on meeting two primary objectives. The first objective was to maximize anticipated actuator stiffness. Not only would a high stiffness actuator provide large force output, but it would also be advantageous for maximizing bandwidth. In the design and analysis iteration phase, predicted first natural frequency was maximized, since most piezo actuators exhibit a second-order roll off and limited bandwidth beyond the first resonance. Second, the stroke requirement had to be met, with an additional goal of optimizing the device efficiency. To maximize the actuator's work efficiency, it should act against a load of equal stiffness. In this configuration, the free displacement of the actuator would be equal to twice the target stroke, or 800 μm. Prior experience in gained piezo actuator design indicated that achieving 1.5 kHz of bandwidth and 400 μm displacement from a gained stroke device would be very difficult, thus the goal to at least maximize bandwidth. Another technique used to achieve a high first mode frequency was to minimize the amount of moving mass in the actuator's gaining mechanism. Of course, secondary design considerations included packaging, system integration, and manufacturability.

An exaggerated representation of the actuator design's kinematics is shown in Fig. 7 below. Besides two parallel piezo stacks, the device consists of a moving diagonal frame and flexure mounted to a rigid housing. As electric field is applied to the stacks, they both extend longitudinally. Fig. 7 illustrates how the left-hand stack pushes the cantilevered flexure and lower end of the moving frame downward, since the stack is constrained by the housing at the top. Simultaneously, the right-hand stack pivots with the moving frame and extends, forcing the free end of the frame leftward even more. The resulting stroke by the free end of the frame is predominantly translation. However, a small amount of rotation also exists due to arcing and foreshortening of the moving frame. This design effectively added the displacement of the two piezo stacks and produced a gain of 5.3 times that combined stroke.

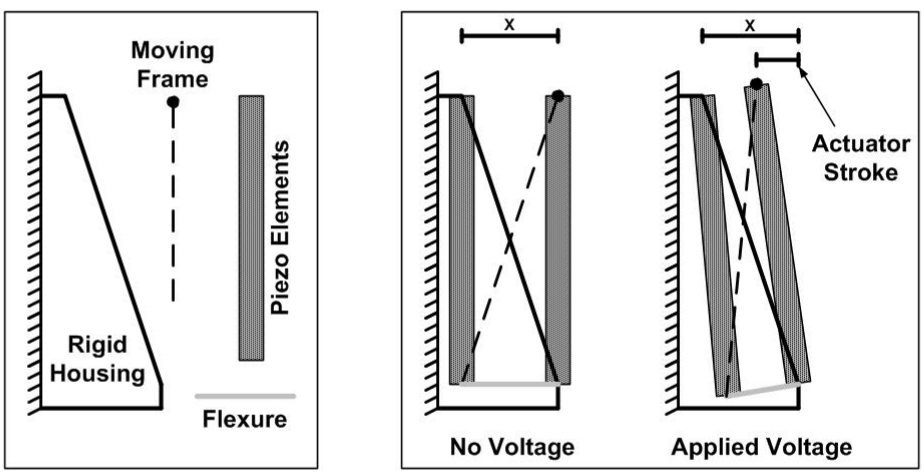


Fig. 7. Representation of actuator's basic components (left), and illustration of actuator design's kinematics (right).

Quasi-static (1 Hz) operation of the actuator working against loads of various stiffnesses produced its characteristic load line shown below in Fig. 8, which corresponded to a measured actuator stiffness of about 82 N/mm. The assembly included two 5 x 5 x 50 mm piezoelectric stacks (Noliac, Inc.) which each produced in excess of 1500 με.

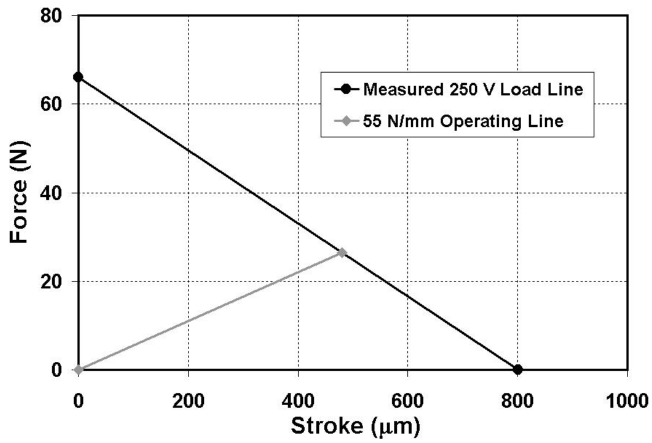


Fig. 8. Measured actuator load line corresponding to a stiffness of 82 N/mm. In order to achieve the 400 μm stroke target (with 20% margin), a 55 N/mm preload spring was selected for operation.

Results of the load line measurements indicated a poor matching of the actuator stiffness (82 N/mm) to the NE/NASTRAN predicted stiffness of 225 N/mm. This was attributed to unwanted compliance in the system due to assembly and inconsistencies in actuator modeling prior to fabrication. In order to meet the maximum stroke requirement of 400 μm with some margin, a 55 N/mm return spring was implemented into the system. Working against this load stiffness, the new actuator provided 480 μm of stroke, but only 26 N of driving force with an applied voltage of 250 V. It was subsequently integrated onto the unit injector control valve for further testing.

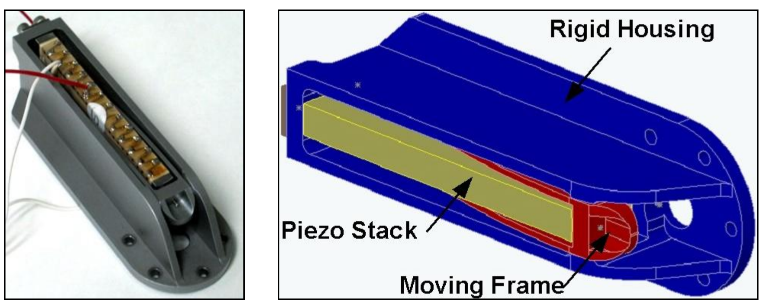


Fig. 9. Photo (left) and rendering (right) of Midé's gained piezoelectric actuator for high speed diesel fuel injection.

5. ACTUATOR INTEGRATION & DYNAMIC TESTING

After the basic actuator design had been established, provisions were made for interfacing with the translating spool, preloading the actuator, and mounting the assembly onto the injector control valve. Keeping the actuator and spool in close proximity to each other was critical in minimizing the moving mass of the system, thus maximizing actuation bandwidth. In the final design, the actuator pushed one end of the spool to open the valve, while a low profile 55 N/mm spring (Fig. 10 below right) returned the spool to its nominally closed, or vented, position. No rigid mechanical connection was made between the moving frame and the spool in an effort to maintain purely axial loads through the plant and avoid concentricity alignment issues. The spool design remained substantially unchanged, but was merely extended in one direction to reach the actuator's moving frame, and hollowed out to reduce mass. The new lengthened spool was of approximately the same mass (5 grams) as the original one. Finally, an inductive displacement sensor was mounted to the opposite (and open) side of the control valve, for measuring spool position. Fig. 10, below left, shows a cross-sectional rendering of the complete assembly.

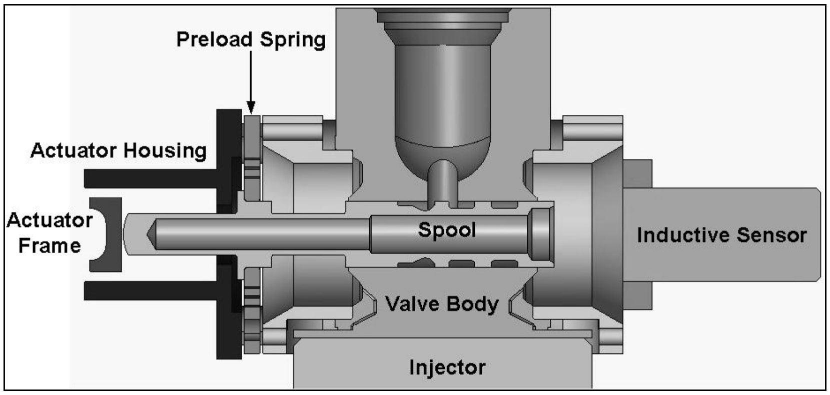
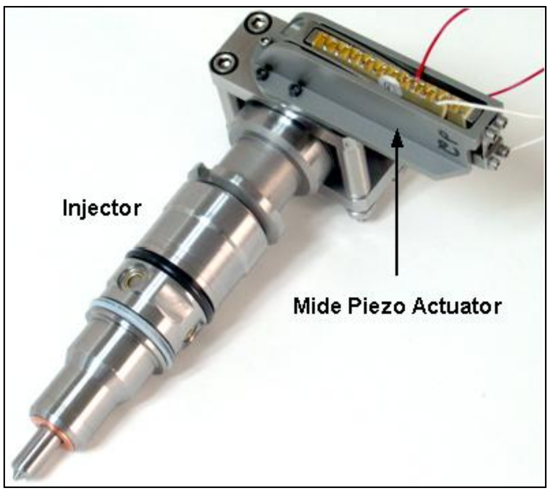


Fig. 10. Cross-sectional view of the control valve, spool, spring, sensor, and actuator assembly (left). Front view photo of the low profile, 55 N/mm return spring (right).



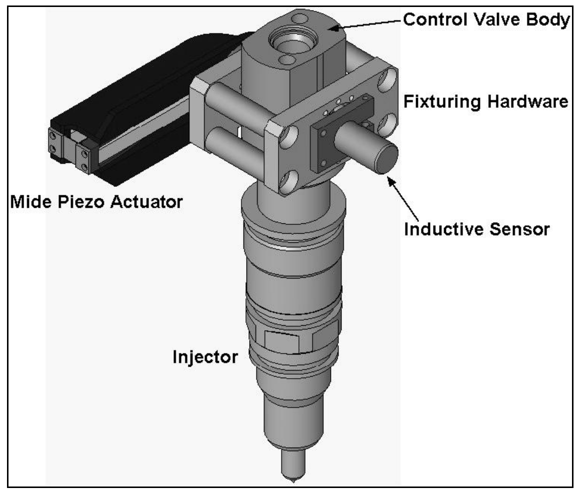


Fig. 11. Photo (left) and rendering (right) of the commercial unit injector retrofitted for spool valve control with Midé's gained piezoelectric actuator.

Dynamic testing of the integrated assembly showed actuation bandwidth to 1 kHz (see Fig. 12 below), even when fully mounted into the rate tube test setup with an 1800 psi oil rail input to the control valve. The first natural frequency of the system was around 625 Hz, with a smaller second resonance appearing around 800 Hz. A characteristic second-order roll off was also present beyond 800 Hz, but the –3 dB point on the magnitude plot was just over 1 kHz. Phase loss was evident even at the lower frequencies, most likely due to sliding friction between the spool and control valve bore, along with the medium pressure oil acting as a damper on the spool motion. In fact, it's theorized that the hydraulic fluid actually extended the system bandwidth by attenuating the higher frequency roll off.

Although sliding friction between the spool and valve is impossible to eliminate, the phase loss at lower frequencies was still higher than expected. The slight arcing and foreshortening of the moving frame created difficult problems transferring that motion to the spool in a truly linear fashion. This prevented a purely axial load on the spool, and the small radial loads most likely added friction to the system. That friction, and resulting phase loss, increased the complexities in closed-loop controller development. Instead, rate tube testing was carried forward using open-loop control only. This consisted of commanding and observing spool trajectory during an event, without closing the loop on spool position. Iterations on the spool trajectory were performed until the desired injection shape was produced. If successful, open loop spool position control would increase the commercialization potential of the injector by keeping controller development, implementation, and processing costs down.

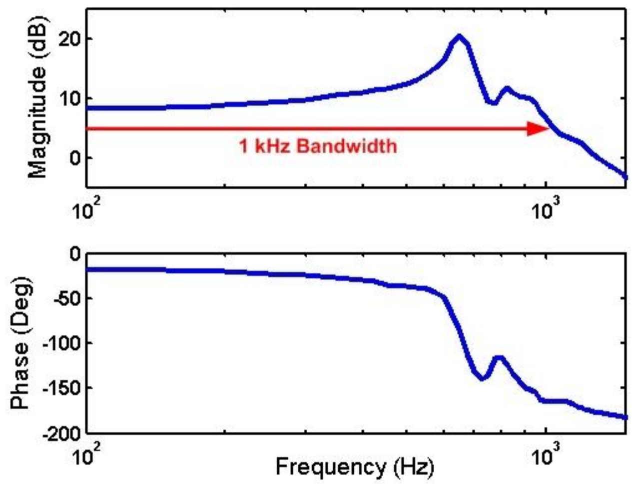


Fig. 12. Bode diagram for Midé's gained piezo actuator, driving the control valve spool, on the injector, in the rate tube test setup (1800 psi oil rail). The first resonance of the system is above 600 Hz, but bandwidth extends to 1 kHz. Note the phase loss even at lower frequencies.

Table 3: Summary of measured actuator characteristics.

Gain	5.3
Driven Mass	5 g
Free Displacement	800 μm
Blocked Force	66 N
Actuator Stiffness	82.4 N/mm
First Natural Frequency*	625 Hz
Bandwidth*	1 kHz
Maximum Actuation Stroke*	480 μm
Maximum Actuation Force*	26 N

*against 55 N/mm load stiffness

6. RATE TUBE TESTING & RESULTS

The rate-of-injection (ROI), or rate tube, test is an industry standard method of measuring the profile of injected fuel from an injector nozzle without the complications of air-fuel mixtures and combustion. Construction of this facility allowed experimental evaluation of rate shaping performance at Midé.

In the rate tube setup, a fuel injector's nozzle is mounted into a sensor block, which also houses a dynamic pressure transducer (Kistler Instrument Corp.), the primary performance-measuring sensor. Exiting the sensor block is a long, fluid-filled tube, which simulates the back pressure conditions associated with an engine cylinder. At the end of the tube, a relief valve is used to control the back pressure level seen by the injector nozzle further upstream. As fluid is injected into the sensor block, subsequently raising the back pressure in the tube, a volume of fluid equal to the injection event is pushed through the relief valve and collected for mass or volume measurement. Monitoring every fluctuation near the nozzle, the sensor outputs a pressure signal, which is proportional to the time dependent rate-of-injection delivered by the injector. It's important to note that the distance between sensor and nozzle does not facilitate measurement of the actual injection pressure (usually greater than 10 ksi), but does provide relative rate shaping information. In this manner, injection flowrate profiles can be recorded through pressure measurements near the nozzle.

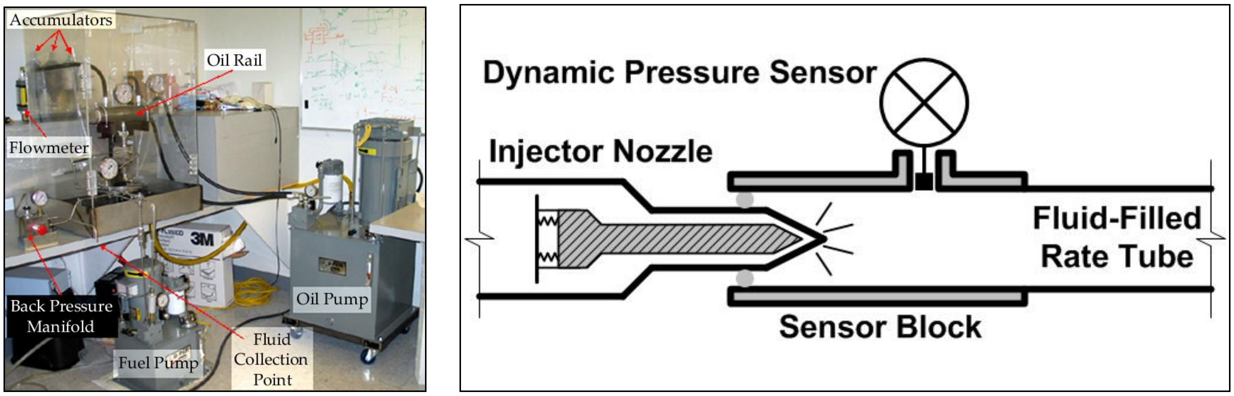


Fig. 13. Photo of the unit injector rate tube test setup with supporting hydraulic systems (left). Detailed illustration of injector nozzle, sensor, sensor block, and rate tube configuration (right).

In general, Lean, Clean, and Mean injection profiles were demonstrated on the rate tube test setup for appropriate injection durations on the order of 1-3 ms. Fig. 14, below, shows one result for each shape (taken from separate test cycles) for comparison to the analytical targets listed in Fig. 6 above. Although the measured profiles did not exactly replicate the target shapes, there is a clear resemblance between the theoretical and laboratory traces.

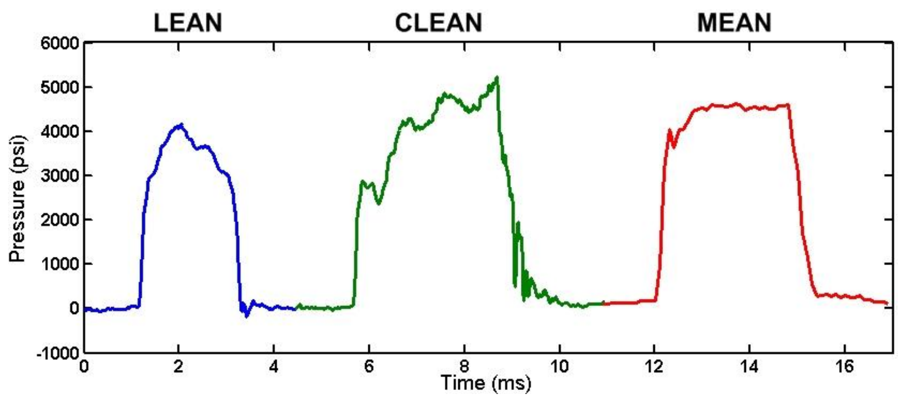


Fig. 14. Rate tube test pressure measurements demonstrating Lean, Clean, and Mean rate shaping capability. Each profile was taken from a separate test cycle and superimposed for illustration purposes.

One characteristic of the injection flowrate profiles that could not be modified was the sharp initial pressure rise at the start of each event. For example, the Lean and Clean target shapes each have leading edge slopes much less than 90°. However, this behavior could not be duplicated using the advanced, high-speed injector. Most likely, the steep rising edge slope in pressure (proportional to flowrate) evident on each injection event is a product of the rapid unseating of the needle valve, which seals the injector nozzle. Precision control over the initial needle lift was not possible through proportional spool authority on the control valve upstream. This relationship between control valve and needle valve was an unknown at the start of the development effort, and the limits of influence are apparent in the uncontrolled initial pressure rise.

Short-term shot-to-shot repeatability was fair, although it could not be quantified with the equipment on-hand. This result was promising for maintaining an open loop control system in the future, but not definitive. Using a slightly lower resolution data acquisition system for longer-term data collection, several consecutive injection events were captured to observe repeatability qualitatively. The results are shown in the ten consecutive injection events overlaid in Fig. 15 below. However, longer-term event repeatability (hundreds or thousands of injections) was not as high. Friction in the spool and valve sliding interface created a system hysteresis, which prevented the spool from returning to the nominal starting point after each injection event. Eventually, the drift was evident as the rate shape deformed over time. Closed-loop control on commanded spool position would eliminate this hysteresis, increasing the repeatability and precision of the injector.

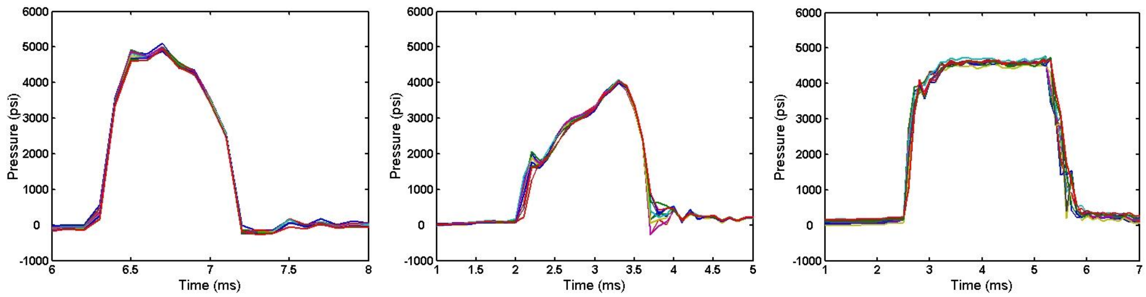


Fig. 15. Ten consecutive overlaid injection event pressure traces showing Lean, Clean, and Mean rate shape repeatability.

Additional rate shaping capability was also demonstrated, apart from the Lean, Clean, and Mean shapes shown above. Below are sample results from in-house rate tube tests that illustrate the injection profile control enabled by Midé's piezoelectric actuator. Short duration "pilot" injections were measured having durations as low as 300 μs (Fig. 16, below left). Rate shaping transients and polytonic shapes were produced even within the injection event itself (Fig. 16, below center). Rapid, extremely short injection bursts were also demonstrated, but were most likely due to unstable "chatter" as the needle would seat and unseat in an uncontrolled manner (Fig. 16, below right). Although not relevant for the target shapes established at the start of the program, injection profile *falling* edge control was established after an initial steep rise (Fig. 17, below left). At the limits of rising edge control, one or two incidents of needle chatter were observed, followed by a fairly steady rising slope (Fig. 17, below center). Finally, a unique boot-shaped profile was demonstrated, having a period of relatively constant flowrate between zero and the peak value achieved near the start of the injection event (Fig. 17, below right). This injection rate shaping control should be considered if the desired target profiles are modified in the future.

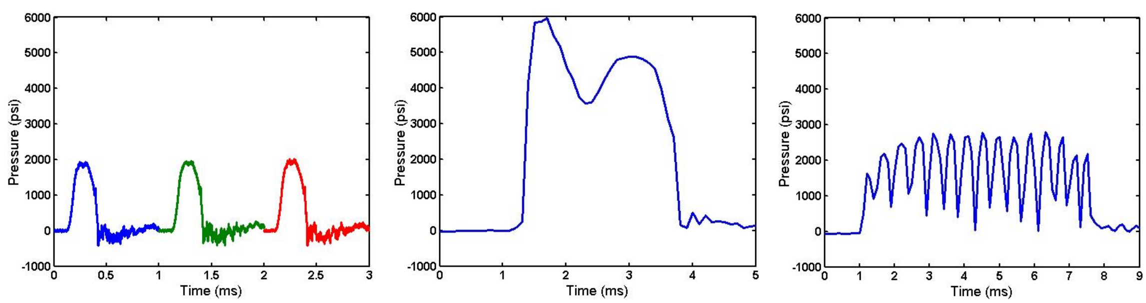


Fig. 16. 300 μs pilot injections from three separate tests (left), polytonic injection profile with two peaks (center), needle chatter effect drawn out over almost 7 ms (right).

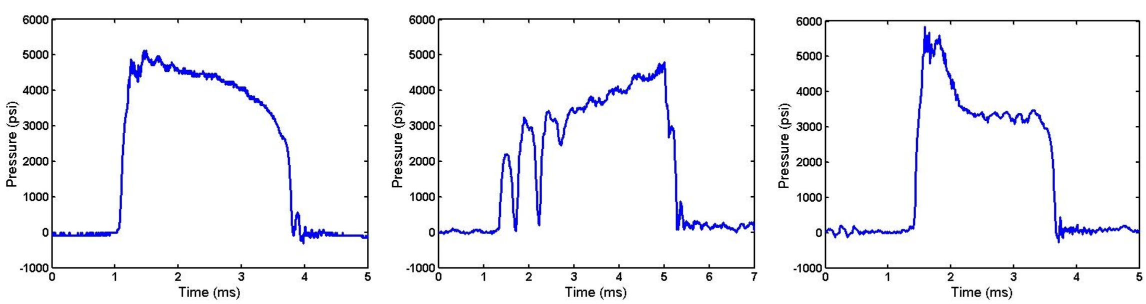


Fig. 17. Falling edge control (left), rising edge control after initial chatter (center), "boot" shaped injection profile (right).

7. CONCLUSIONS

Proportional authority over control valve spool position for Lean, Clean, and Mean injection rate shaping was demonstrated using open loop spool position control. This was accomplished despite the actuator's slightly deficient performance characteristics, when compared to the initial design requirements. Event durations between 0.3-4 ms, and rate tube peak pressures from 2000-6000 psi were recorded, which fall in line with appropriate performance characteristics. Further injection flowrate control was also proven through the variety of profiles and durations measured over the course of rate tube testing. These results indicated that the actuator dynamic target specifications, outlined early in the program, were well beyond the necessary performance metrics. The discrepancy could be attributed to poor knowledge of the commercial, off-the-shelf unit injector selected for technology demonstration.

In the future, a full-scale engine test demonstration would verify Lean, Clean, and Mean engine operation regime control through injection rate shaping. Several development improvements would lead to this objective. First, a higher bandwidth actuator would only enhance the rate shaping capability demonstrated already, possibly enabling precision micro injections. Second, an actuator with purely linear stroke would minimize the sliding friction between the spool and valve bore. Third, full closed loop control with respect to spool position during an injection event would eliminate hysteresis and drive repeatability up. And finally, an injector and spool control valve designed specifically for proportional authority (rather than digital) is desirable.

ACKNOLEDGEMENTS

The authors would like to extend their gratitude to the technical monitor Richard Siorek of the Army, Marco Serra formerly of Midé, and Chad Jansen of Sturman Industries for their contributions to this program. Research was conducted under contract number DAAE07-01-C-L097, through the U.S. Army RDECOM.

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