An Indirect Sensing Technique for Closed-Loop Diesel Fuel Quantity Control

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ABSTRACT

Despite significant advances in electronic control technology applied to diesel engines, commercially available injection systems for automotive diesel engines remain limited by the open-loop mapping of the injection pump. An initial calibration is relied upon to translate a fuel delivery command to an actual fuel quantity. In practice, however, these two variables may be substantially different due to the effects of mechanical wear, repair, and the wide range of operating conditions. Possible ramifications of this discrepancy are excessive exhaust smoke due to overfueling, increased fuel consumption, degraded driveability, and poor idle characteristics. The major obstacle to closing the fuel control loop is the lack of a suitable sensor for instantaneous fuel delivery from the injector.

An indirect fuel delivery sensing mechanism based upon the use of the injector needle lift in conjunction with the fuel temperature is evaluated. An estimation of the injection rate characteristic is determined from real-time analysis of the needle lift signal using a high-speed sensor processor. Integration of the rate characteristic and temperature correction yields a total mass delivery estimate for use as a feedback quantity for closed-loop fuel control. Signal processing algorithms are derived from computer modeling of the injector and verified experimentally. Possible long-term decalibration due to nozzle coking is studied. Advantages and limitations of the technique are identified.

BACKGROUND AND PROBLEM DESCRIPTION

Although diesel engines and injection systems represent a mature technology, it is only in recent years that electronic controls have been successfully applied. The majority of diesel applications are still mechanically controlled, with no electronics involved other than the fuel shutoff solenoid valve control.

The potential benefits of microprocessor-based control applied to diesel engines have been well established [Reams82, Kihara83, Martinsons82, Trenne82, Kawai84]. However, many of the improvements made possible by advanced electronic control

are dependent upon exact knowledge by the controller of the hydraulic characteristics of the injection system components.

This dependency is particularly important in the case of small-displacement automotive diesel engines, which use low-cost distributor pumps which must accurately meter very small (less than 50 mm³ per injection) fuel quantities at high speeds.

Major incentives for more accurate fuel control have appeared in the form of recent regulatory pressures for "cleaner" diesels along with the demands of the automotive market for driving characteristics more like those of gasoline fueled engines. Gradually increasing concerns about rising gasoline costs may be expected to produce a renewed interest in automotive diesels.

Available and currently envisioned electronically controlled distributor-type fuel injection pumps for automotive diesel engines generally operate with a map-based translation between commanded fuel quantity and actual fuel delivery. A ROM-stored multi-dimensional map is typically accessed with inputs of commanded fuel volume and pump speed. A third parameter, fuel or pump housing temperature, may be additionally used to modify the table output to yield a corrected fuel control position corresponding to a given commanded fuel mass. The map and correction factor(s) are generated experimentally from pump test data, typically using a reference pump. A fully specified map of adequate resolution and valid correction factors may require the acquisition of a large number of data points on a pump test stand.

There are several limitations of this open loop fuel control method. The generation of individual calibration maps for each pump, and subsequent storage of individual maps in ROM, is impractical in production. At best, a linear correction for pump misalignment is performed during final checkout of individual pumps, using either an external resistor network [Stumpp83] or final PROM programming procedure. However, minor machining differences, even within manufacturing tolerances, can cause noticeable differences between the calibration maps of individual production pumps and the test pump. This difference is compounded by the synergistic relationship between the injectors and the pump. The injectors fitted to a particular production pump may differ slightly in their flow characteristics from those used with the test pump during the master calibration, thus changing the overall calibration. One or more injectors might also be replaced or readjusted at a later date.

Possibly more important is the problem that the injection system often operates under conditions much different than those that existed during the master calibration tests. The calibration also changes over the course of time due to normal wear and corrosion of pump and injector internal components, and the accumulation of carbon deposits in the injector nozzle (nozzle coking).

In actual service, the delivered fuel quantity may differ substantially from the mapped quantity. Possible effects of this discrepancy are excessive exhaust smoke due to overfueling, inaccurate torque limiting, increased fuel consumption, degraded driveability, and unstable or noisy idle characteristics.

If the actual fuel delivery per injection were directly sensible, closed-loop control of the fuel quantity would be possible as a means for improving the fuel control accuracy without the need for further mechanical refinements and tighter tolerances in the pump. The major obstacle to closing the fuel control loop appears to be the lack of a suitable sensor for fuel quantity.

Real-time monitoring of the actual fuel delivery per individual injection stroke is difficult. A number of factors can be cited in relation to this technical obstacle. For a typical small displacement (i.e., under two liter) diesel engine, the fuel delivery volume is very small (in the range of from 5 to 50 mm³ per stroke), and the duration very brief (on the order of 1.0 ms). The repetition rate per cylinder is typically from 5 to 50 injections per second, with line pressure fluctuating from the delivery valve opening pressure to the injection peak pressure (as high as 100 MPa) at this repetition rate. The static volume in each fuel injection line and the secondary passages of the pump may exceed the fuel delivery per injection by a large factor. Fluid compressibility, inertial effects, tubing strain, and internal leakage in the pump make the process non-ideal, so that the actual fuel delivery often differs significantly from the metered plunger displacement in the pump. Mass transport in this medium is characterized by propagation of a pressure wave between the pumping chamber and the injector nozzle. Computer simulation of the injection pump hydraulics using finite difference methods is often relied upon to predict the delivered quantity and injection characteristics [Oren83, Kumar83, Sharma83].

While suitable sensors for the rate characteristic have been suggested [Bosch66, Komaroff66, Thoma74] for use in test bench calibration of pumps, a practical sensor suitable for use during

actual engine operation as a real-time feedback control device is not, to the best of our knowledge, currently available. Methods utilizing injection pulse duration in conjunction with engine speed to estimate fuel usage have been investigated [Wolff86]. A thermal convection based fuel flow sensor has also been studied [Challen88].

The advantages of closed loop control in general are well established. Efforts to close the control loop on engine torque [Ribbens81, Fleming82, Sood84], combustion luminescence [Bunting84], and cylinder pressure [Challen88] have been important recent contributions to diesel control technology. All of these techniques may be considered as indirect indicators of the injected fuel quantity, which for some performance metrics (i.e., emissions) is the target variable of primary importance.

EXPERIMENTAL METHOD

The reported work investigates the feasibility of using information contained in the motion of the injector needle in combination with other sensor signals, to infer the delivered fuel quantity, rate characteristic, and timing of critical portions of the delivery schedule on an individual injection basis. It is hypothesized that the availability of these metrics in real time could facilitate improvements in engine control, with resultant improvements possible in engine emissions, efficiency, driveability and noise.

Needle valve position is often used as an approximate qualitative indicator of the injection rate characteristic [Burman62(1), Hiroyasu80, Obert68(1)]. The beginning and end of needle lift are recognized as the respective start and end of the injection pulse. Start of needle lift is usually used as the injection timing reference event. The use of a needle lift sensor for closed loop injection timing control is common practice [Stumpp83, Ives84, Wolff82]. Pre- and post- injections are also identified by the needle lift. The duration of the injection period, measured using the needle lift signal, has been used as an approximation for the fuel quantity [Stumpp83], and in conjunction with a engine speed has been used to map fuel consumption for a specific engine and pump [Wolff86]. However, the displacement of the needle is, at best, a very nonlinear indicator of actual fuel flow rate through

Figure 1 Cross-sectional View, Optimized Injector/Flow Transducer

The on-engine testing of engine controls based on this method is currently in progress, although results were not yet available at the time of submission of this paper. Of particular interest is the long-term reliability of this sensing method, in view of possible degradation of the injector/sensor due to nozzle coking and internal wear. Cylinder-to-cylinder variations in the injection characteristics and the relative benefit of instrumenting more than one cylinder for fuel flow are also being assessed.

wave is attenuated as it is reflected back and forth along the length. Fluid injected into the measurement tube is relieved through a needle valve and pressure relief valve at the end of the tube. The relationship between the pressure of the wave and the injector flowrate is:

$$\dot{q} = \frac{f_t P}{10 a \rho} \left(\frac{\text{cm}^3}{\text{sec}} \right) \quad (1)$$

where

f_t = flow area of measurement tube (cm²)
 a = acoustic velocity in diesel fuel (m/sec)
 ρ = density of diesel fuel (gm/cm³)
 P = differential pressure (Pa)

The product $a\rho$ is referred to as the acoustic impedance. For diesel fuel at the experimental conditions we used $a = 1250$ m/sec and $\rho = 0.84$ gm/cm³. The inner flow area f_t of the measurement tube was 0.317 cm². The pressure transducer must not cause any reflections, so pressure is measured by a strain gauge attached to the outside of a thin-wall section of the measurement tube, referred to as the pressure transducer section. Hoop strain in the tube serves as the mechanism for measurement of the passing wave-front pressure. The quiescent pressure in the measurement tube is maintained by the pressure relief valve at approximately peak cylinder compression pressure to simulate the actual backpressure encountered in an engine installation of the injector.

The length of the measurement tube is critical, since reflected pulses can interfere with the primary wavefront. Interference is selective with pump speed. Bosch prescribed the use of two different length measurement tubes to handle the complete RPM range, 4.67 and 9.34 meters. In order to provide sufficient attenuation of the primary reflex pulse to avoid interference at all speeds, we found it necessary to use a tube 80.16 meters long.

Our apparatus was fitted to accept two types of nozzle holders: Robert Bosch KCA series (RB) and Diesel-Kiki Type 71-1280 (DK). The nozzle holders were fitted internally with Hall-effect needle position sensors manufactured by Wolff Controls. Both nozzle holders work with pintle-type nozzles intended for indirect injection applications. All tests were performed using a Diesel-Kiki (Robert Bosch licensed) NP-VE4 injection pump mounted on a Bacharach Type YYQ pump test stand.

Total fuel delivery per injection may be calculated by integration of the rate characteristic. By integrating and comparing with the volume delivery measured by the test stand burette, Bosch reported an error of 3.3% for a selected rate characteristic sensed by the apparatus. Using this same method, we observed an average absolute error of 4.82% (RB injector) or 6.52% (DK injector) for the Bosch rate apparatus over the operational range of the pump/injector, with a peak error of 14.6% (RB) or 15.1% (DK) at a high delivery conditions. Figure 4 is a matrix indicating the accuracy of the Bosch-type reference rate apparatus with the RB injector over the operational speed and throttle angle range of the pump.

STATIC NONLINEAR MODEL

A simplified injector flow model was previously proposed [MacCarley87] which essentially ignores dynamic effects and assigns a one-to-one mapping between needle position and instantaneous flow rate, modified only by a density factor dependent on fuel temperature. A basic requirement of this model is that the

RPM	THROTTLE (degrees)							
	22	45	67	90	112	135	157	180
125	-0.429	-4.881	3.501	-2.599	-5.039	-1.472	-0.703	-3.173
250	-4.473	-6.128	-4.528	-4.781	-2.913	-5.160	-5.693	-6.222
375	3.129	-5.78	1.406	-1.759	-1.765	-3.606	-2.099	1.263
500	**	**	-1.705	-2.297	-5.326	-6.039	-5.410	-2.505
625	**	**	-0.882	-1.293	-4.824	-1.137	-2.784	-4.739
750	**	**	3.363	1.635	-2.991	-5.850	.358	-2.714
875	**	**	3.267	-7.65	-4.902	-5.832	-4.233	-2.470
1000	**	**	3.751	-3.508	-10.767	-9.421	.848	-6.480
1125	**	**	-3.510	-.910	2.134	-10.161	2.757	-4.809
1250	**	**	-4.699	.843	-6.354	-5.544	2.274	-5.379
1375	**	**	-3.582	-11.832	-1.817	-2.043	-3.736	-8.463
1500	**	**	2.454	-6.394	-9.696	-11.306	-12.024	-9.840
1625	**	**	-.795	-.337	-11.372	-1.487	-6.604	-.194
1750	**	**	3.526	-1.544	-12.634	-7.794	-11.211	-11.167
1875	**	**	3.977	-8.375	-11.804	-3.426	-10.942	-9.636
2000	**	**	1.738	-6.876	-8.705	-10.319	-13.004	-14.611

** Indicates that buret volume was too small to accurately measure

Average magnitude of % difference = 4.816

Figure 4 Percentage Difference Between Volume Calculated by Integration of Reference Rate Data and Actual Burette Volume (RB Injector).

needle (pintle, poppet) be unrestricted by a physical stop in its fully open position. Both types of IDI injectors evaluated met this requirement.

A basic nozzle flow relationship is commonly used to describe the instantaneous mass fuel flow rate through an injector nozzle, as illustrated by the diagram of Figure 5. [Streeter71, Oren83, Obert68(2), Kumar83, Burman62(2), Sharma83]:

$$\dot{m} = 10^3 A_{no}(x) C_d(x, \dot{m}) \sqrt{2(p_1 - p_2) \rho(T)} \quad (2)$$

$\dot{m}$ = mass flow rate [mg/sec]
 A_{no} = nozzle area [mm²]
 C_d = nozzle discharge coefficient [unitless]
 p_1 = injection pressure, immediately upstream of the orifice [kPa]
 p_2 = downstream cylinder pressure [kPa]
 ρ = local fuel density [mg/mm³]
 x = needle displacement from closed position, [mm]
 T = fuel temperature [°C]

Figure 5 Simplified Flow Model for Injector Nozzle

A_{no} is a unique function of the needle position x . C_d is both a function of x and of the flow velocity, so that normally an iterative solution is necessary for the flow problem. It is reasonable to assume p to be dependent upon temperature T only, since fluid compressibility affects it only slightly, even at the high pressures involved [Bosch66]. It is impossible to determine the time history of p_2 without the use of a cylinder pressure transducer. If a pressure transducer is available to the feedback control system, exact knowledge of p_2 can be used in this calculation. Lacking such a transducer, a constant average value $\bar{p}_2$ based upon test measurements may be used [Obert68(2)], with some loss of accuracy.

The linear movement of the needle may be modeled by:

$$p_1 A_1 - (p_1 - p_2) A_2(x) = 10^{-3} M \ddot{x} + 10^3 k_f \dot{x} + 10^3 (k_{spr} x + f_0) \quad (3)$$

M = combined mass of the needle, spindle, and part of the spring mass [gm].

k_f = combined linear coefficient of friction [Newtons-sec/mm]

k_{spr} = differential linear spring coefficient for small displacements [Newtons/mm]

f_0 = nozzle opening force [Newtons]

A_1 = upper valve piston area [mm²]

$A_2(x)$ = effective lower (counteracting) valve piston area [mm²]

x = needle position, $x > 0$ [mm]

$\dot{x}$ = needle velocity [mm/sec]

$\ddot{x}$ = needle acceleration [mm/sec²]

A dynamic model for the nozzle flow must take into account the inertial and frictional effects on the moving parts, as well as the fact that the flowrate is rapidly changing.

However, the use of (3) to calculate these effects requires generation of the first and second derivative from samples of the needle position. It was observed experimentally that noise amplification obscured the true dynamics, requiring filtering techniques to yield even an approximation to $\dot{x}$ or $\ddot{x}$.

A considerable simplification is possible by neglecting the inertial and frictional effects, so that pressure p_1 is characterized by x alone, and none of its derivatives. This is not unreasonable considering the extremely high force to mass ratio acting on the needle assembly. Errors introduced by this assumption tend to be self-canceling when the rate curve is integrated over the entire injection interval, especially if the curve is close to symmetric. The error is also reduced by minimization of the needle assembly mass and frictional contact area.

The final assumption that

$$A_1 \gg A_2(x) \quad (4)$$

allows the slight effect of the cylinder pressure to be ignored in the needle position equation. (Assumes fluid pressure acting on upper piston area A_1 only.) This is actually valid only after the nozzle is flowing, so that some ambiguity occurs during initial and final flow, reducing accuracy particularly at small deliveries.

The preceding assumptions allow p_1 to be expressed as a unique function of the needle position x :

$$p_1 = p_1(x) = \frac{10^3}{A_1} k_{spr} x + p_{no}, \quad x > 0, \quad p > p_{no} \quad (5)$$

where p_{no} = nozzle opening pressure [kPa].

This permits the discharge coefficient to also be expressed in terms of x alone:

$$C_d(x, \dot{m}) = C_d(x) \quad (7)$$

so that (2) may be simplified to:

$$\begin{aligned} \dot{m} &= 10^3 A_{no}(x) C_d(x) \left[2(p_1(x) - \bar{p}_2) \rho(T) \right]^{1/2} \\ &= \left[10^3 A_{no}(x) C_d(x) \left[2(p_1(x) - \bar{p}_2) \rho(T_0) \right]^{1/2} \right] \left[\frac{\rho(T)}{\rho(T_0)} \right]^{1/2} \quad (8) \\ &= f_1(x) f_2(T) \end{aligned}$$

where T_0 = fuel temperature at which f_1 is measured.

Total fuel delivery is determined by time integration of $\dot{m}$ over the duration of the injection period. Since T is slowly varying, it can be considered constant over the integration period.

$$\begin{aligned} m &= \int \dot{m} dt = \int f_1(x) f_2(T) dt \\ &= f_2(T) \int f_1(x) dt \quad (9) \end{aligned}$$

In practice $f_1(x)$ is determined experimentally for a particular injector,

$$f_1(x) = \dot{m}(x) \Big|_{T=T_0} \quad (10)$$

while $f_2(T)$ is a known function for diesel fuel. This simple flow model is not valid in the case of a chattering or oscillating needle, known to occur under certain conditions in some injectors [Burman62(3)].

The reference rate measurement apparatus was used to generate the nonlinear map of f_1 . The composite f_1 function is generated by averaging measured flow vs needle position data taken over the speed and throttle angle range of the pump. At each condition, 256 samples of the rate characteristic and needle lift curve are acquired and digitized. Eight throttle positions are tested at each of sixteen pump speeds, from 125 to 2000 RPM. Figure 6 illustrates the variation of the f_1 function with pump speed for a

FLOW RATE (microliters/ms)

Figure 6 Nozzle Flow vs Needle Position, Parametric with Pump Speed

Diesel-Kiki 71-1280 injector. (The governor was operative, so that the fuel sleeve position was also changing with speed in Figure 6.) The rather minor variation in the f_1 manifold along the RPM axis is noted.

Figure 7 shows the overall average f_1 function for a Robert Bosch KCA30SD27/4 injector. Figure 8 shows the average f_1 for the Kiki injector.

This model requires only the nonlinear mapping of needle position into injector flowrate, temperature-density correction, and integration to yield total mass delivery per injection. The effectiveness of this approach is illustrated by Figure 9, which shows a typical needle lift curve (1500 pump RPM, 180 deg. throttle, Kiki injector) with corresponding rate curves derived from (1) the reference rate measurement apparatus, (2) the nonlinear mapping of the simplified static model, and (3) a dynamic model to be

described later. Comparison of the rate curve generated by the static model with that obtained from the reference rate apparatus indicates that the flow estimate lags slightly the movement of the needle, underestimating flow while the nozzle is opening, and overestimating while it is closing.

Figure 10 is an error matrix similar to that of Figure 4, comparing the integrated fuel quantity calculated by the static model from the needle lift signal, with the burette volume from the test stand. Over the operational range, an average absolute error of 4.46% (RB) 4.72% (DK) is observed, with a peak error of 18.0% (RB) or 14.6% (DK) at medium-flow conditions in both cases. It may be concluded by comparison of this matrix with that of Figure 4 that the simplified static model provides a fuel quantity estimate of similar accuracy to that obtained from the reference rate apparatus (that was used to generate the f_1 function for the static model).

Figure 7 $f_1(x)$ Flow Function, RB Injector

Figure 8 $f_1(x)$ Flow Function, DK Injector

Figure 9 Rate Characteristics Derived from Needle Lift by Static and Dynamic Models

RPM	THROTTLE (degrees)							
	22	45	67	90	112	135	157	180
125	-4.72	-3.78	-7.98	1.39	-4.64	-3.98	4.24	1.96
250	2.68	-1.01	-4.32	1.72	-2.28	-3.90	-0.34	2.52
375	0.91	-2.35	-4.32	0.41	-4.93	-4.95	-3.03	-0.84
500	**	**	1.16	0.77	-5.38	-6.23	1.68	-1.87
625	**	**	8.86	1.53	-2.80	-4.14	-1.18	-2.35
750	**	**	5.75	3.92	-1.85	-3.74	-1.72	-1.41
875	**	**	6.56	2.65	-2.90	-3.67	-1.70	-1.16
1000	**	**	8.40	4.89	1.12	-1.91	3.57	0.15
1125	**	**	0.99	14.32	2.13	-3.71	6.63	1.48
1250	**	**	2.18	14.81	3.10	-1.35	7.39	6.66
1375	**	**	4.79	10.35	2.05	-3.67	5.28	-0.71
1500	**	**	17.36	10.94	2.35	0.69	4.20	3.29
1625	**	**	17.99	11.68	4.09	5.63	1.97	2.21
1750	**	**	10.54	11.07	4.07	4.98	4.97	1.50
1875	**	**	14.47	6.36	9.21	3.63	3.42	1.14
2000	**	**	12.32	8.92	4.04	7.93	3.25	-0.78

** Indicates that buret volume was too small to accurately measure

Average magnitude of % difference = 4.464

Figure 10 Percentage Difference Between Volume Calculated by Static Nonlinear Model and Actual Burette Volume

DYNAMIC MODELS

Several closed-form models were studied which incorporate inertial effects acting on the needle and fluid. The best results were obtained with a model developed as an modification of the simple static model.

From (3) and (4), p_1 may be written as the sum of static and dynamic contributions:

$$p_1 = \left[\frac{10^3}{A_1} k_{spr} x + p_{no} \right] + \left[\frac{10^{-3} M}{A_1} \ddot{x} + \frac{10^3 k_f}{A_1} \dot{x} \right] \\ = p_s(x) + p_d(\dot{x}, \ddot{x}) \quad (11)$$

From (8),

$$\dot{m} = 10^3 A_{no}(x) C_d(x) \left[2\rho(T_0) \left[p_s(x) + p_d(\dot{x}, \ddot{x}) - \bar{p}_2 \right] \right]^{1/4} f_2(T) \quad (12)$$

$$= \left[\left[10^3 A_{no}(x) C_d(x) \right]^2 2\rho(T_0) \left[p_s(x) - \bar{p}_2 \right] \right. \\ \left. + \left[10^3 A_{no}(x) C_d(x) \right]^2 2\rho(T_0) p_d(\dot{x}, \ddot{x}) \right]^{1/4} f_2(T) \\ = \left[f_1^2(x) + f_2^2(\dot{x}, \ddot{x}) \right]^{1/4} f_2(T) \quad (13)$$

where

$$f_d(\dot{x}, \ddot{x}) = 10^3 A_{no}(x) C_d(x) \left[\frac{2\rho(T_0)}{A_1} \left[10^{-3} M \ddot{x} + 10^3 k_f \dot{x} \right] \right]^{1/4} \quad (14)$$

The product $A_{no}(x) C_d(x)$ describes the nozzle as a function of the needle position. It is known to be a monotonic function that saturates at some maximum value of x . Experimental determination of $A_{no}(x) C_d(x)$ by continuous flow testing of the injector was unsuccessful due to the high flow at high pressure required for needle lift values above 0.5 mm (10 ml/sec., 20.7 MPa at 0.5 mm lift). An approximate function was used:

$$A_{no}(x) C_d(x) = \alpha x^\gamma \quad [\text{mm}^2] \quad (15)$$

where α and γ are model parameters ($0 < \gamma < 1$).

With this assumption, it is possible to extend the static model previously derived to include some dynamic (velocity and acceleration) effects.

$$f_d(\dot{x}, \ddot{x}) = x^\gamma \left[a\dot{x} + b\ddot{x} \right]^{1/4} \quad (16)$$

where

$$a = \frac{2 \times 10^9 \alpha^2 \rho(T_0) k_f}{A_1} \left[\frac{\text{mg}^2}{\text{mm} \cdot \text{sec}} \right]$$

$$b = \frac{2 \times 10^3 \alpha^2 \rho(T_0) M}{A_1} \left[\frac{\text{mg}^2}{\text{mm}} \right]$$

The mass flow relationship is therefore

$$\dot{m} = \left[f_1^2(x) + x^w \left[a\dot{x} + b\ddot{x} \right] \right]^{1/4} f_2(T) \quad (17)$$

where $w = 2\gamma$, $0 < w < 2$.

$\dot{x}$ and $\ddot{x}$ are found by numeric differentiation of the needle position. A four-period average derivative is used to suppress the noise amplification effects of the differentiation process. The parameters w , a and b were optimally determined by least squares fitting the model to the acquired rate vs needle lift histories over the operational speed and throttle range of the pump. For the DK

injector, optimum values were found to be $a = 3.99 \times 10^4$, $b = -4.80$, $w = 0.366$.

Figure 9 illustrates the performance of this dynamic model relative to the static model and the reference rate trace. A slightly better correlation with the reference trace is observed. Integration under the curve to calculate quantity yields 30.3 mm^3 , which improves slightly upon the static model estimate (31.5 mm^3) of the measured volume fuel delivery (29.8 mm^3). For comparison, integration of the reference rate curve yielded a volume of 31.1 mm^3 . Over the complete operational range for the DK injector, the dynamic model yielded an average absolute percentage error of 3.7%, an improvement over the 4.7% error of the static model. Peak error was reduced from 14.6% to 10.6%, both occurring at 2000 RPM, 135 deg throttle (measured volume 20.3 mm^3).

DEGRADATION DUE TO INJECTOR DECALIBRATION

The problem of nozzle coking, which occurs in all injectors to some degree, is expected to be a possible source of loss of calibration for this sensing mechanism. Long-term wear of the pintle and nozzle orifice may also contribute to a gradual loss of calibration of the injector. The result is a change in f_1 over time, which contributes to increased error of the sensing method.

Open nozzles, such as those commonly used in small displacement IDI automotive engines, are usually less prone to nozzle coking than closed, multi-hole nozzles found in larger, lower speed engines. Optimal injector/sensor design to minimize the influence of nozzle deposits on the flow calibration may reduce this concern to some degree. Adaptive signal processing techniques may also be applied to compensate for changes in the sensor calibration, if at least one other indirect indication of fuel delivery is available. One such comparison signal for long-term adaptation is driveshaft torque, sensed directly via magnetostrictive methods [Ribbens81, Fleming82] or indirectly via instantaneous crankshaft velocity as described by Sood [Sood84] and others. An adaptive corrective algorithm based on periodic comparison of the needle lift derived fuel quantity with a steady-state torque reference point is currently being studied.

The effects of nozzle coking on the long-term accuracy of methods utilizing needle lift to infer fuel delivery were studied by generating parallel bodies of data on the same injector, both clean and heavily coked. A Bosch KCA-series injector subjected to 188,000 km of continuous service was mapped over the previously defined operational range. The injector was then thoroughly cleaned and the mapping process repeated. Average f_1 functions for the nozzle in clean vs coked condition are overlaid in Figure 11. In clean condition, the static model produced an average absolute error of 6.86%. This poorer than usual accuracy may be partially attributed to the reference rate apparatus, which alone yielded an average absolute error of 7.02%. The "clean" f_1 function was then used to generate fuel delivery estimates from the "coked" injector needle lift data, simulating long-term degradation attributable to coking. An average absolute error of 6.05% was observed. The small difference (actually a slight improvement for this injector) is indicative of relatively minor calibration change attributable to nozzle deposits.

Figure 11 Degradation of $f_1(x)$ Calibration due to Nozzle Coking, RB (Volkswagen) Injector

CONCLUSIONS

The results of this work appear to confirm that for two specific types of IDI pintle nozzles, adequate information is contained in the needle lift signal to reasonably estimate the injection rate characteristic and total fuel delivery, independent of the pump or engine. The accuracy of the fuel calculation via this method is similar to the accuracy of the rate measurement apparatus used to calibrate the signal processing model.

A simple static nonlinear model appears adequate to achieve a sensing mechanism of $\pm 5\%$ average accuracy. Some further improvement is achievable by inclusion of dynamic effects in the model.

Nozzle coking and wear degrade the accuracy to a minor degree. Compensation for this degradation may be possible via adaptive control methods.

Closed-loop fuel control based upon this sensing mechanism appears to be feasible.

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