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ABSTRACT

An event-based transient fuel compensator (TFC) algorithm was developed for production application on SPFI gasoline engines.

The independent parameters of the TFC were related to fundamental mass-transfer principles from the research of Gilliland and Sherwood [1] to simplify cold-driveability and emissions calibration activities. A compact intake valve temperature model was developed to further simplify calibration. Digital Control Theory was applied to the calibration structure of the algorithm to clarify the relationship between compensator stability and parameter settings.

In its final production algorithm form on a 2.4L DOHC engine application, the TFC met the required subjective cold-driveability requirements and emission standards with a significant reduction in transient fuel calibration complexity.

ALGORITHM PURPOSE

The TFC algorithm was designed to provide compensation for fuel film dynamics in a SPFI gasoline engine. The TFC problem has been presented previously in Aquino [2] and many others.

As shown below in Figure 1, the TFC insures that a desired mass of fuel (Input 1) enters the engine cylinder by producing a commanded fuel mass output to account for fuel film and fuel injection dynamics. The TFC uses the independent engine operating states (Inputs 2-6) to model the dynamics of the fuel film and fuel injection.

The purpose of the TFC was to significantly reduce the number of calibratable parameters needed to relate the dynamic characteristics of the fuel film to the independent engine operating parameters, using fundamental Chemical Engineering research as a guide.

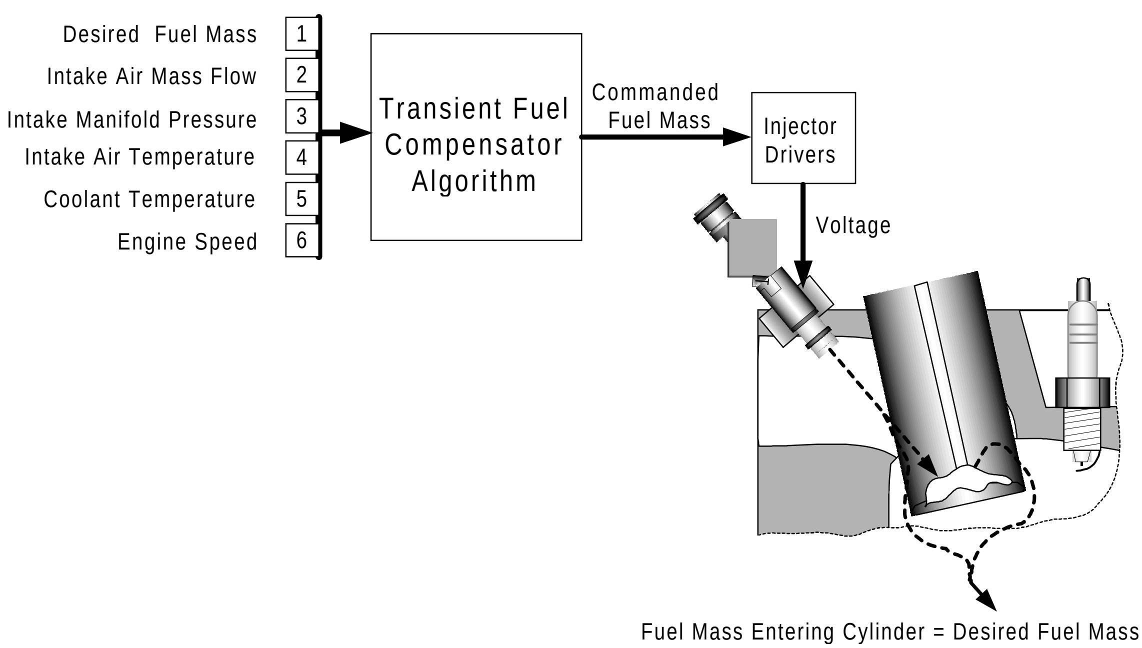

Figure 1. Transient Fuel Compensator and Plant

RESULTS SUMMARY

In addition to meeting the minimum requirement of production emissions levels, the physically based TFC algorithm was validated relative to a conventional TFC algorithm on the basis of subjective cold driveability testing. Design-intent validation was based on the reduction of adjustable calibration parameters and computational requirements of the new TFC algorithm.

COLD DRIVEABILITY PERFORMANCE – Cold driveability was tested on a standardized drive-procedure and subjective evaluation in an environmental chamber with controlled temperature and wind-speed.

A test vehicle equipped with a 2.4L 4 cylinder DOHC engine and manual transmission was filled with 6 gallons of 1200 Driveability Index (DI) fuel and soaked in an environmental chamber until the engine oil temperature stabilized at –30C.

Figures 2a and 2b below show results from one of the most difficult tests. Figure 2a shows an overlay of the commanded and measured equivalence ratios. Test points where the fuel was cut-off on decelerations was are marked with an “x” to differentiate between desired and undesired lean excursions. Commanded equivalence ratio is shown as a dashed line.

The top three graphs of Figure 2b show the throttle position, engine speed, and coolant temperature states which occurred over the driveability schedule. The fourth graph of Figure 2b shows the fuel mass output/input gain of the TFC.

5 seconds after engine run, the throttle pedal was “stomped” to wide-open to test the performance of the TFC under worst-case conditions. The extreme conditions of the “stomp” maneuver required the transient fuel algorithm to briefly saturate the fuel injector pulse-widths, meaning that a brief lean excursion was unavoidable with the fuel system under test. After the “stomp”, a subjective driveability evaluation was made using an established drive procedure.

Figure 2a shows that a significant amount of delivered fuel (up to 70%) is unburned at the beginning of the test under steady conditions. Under worst-case transient conditions, the TFC algorithm amplified the Desired Fuel Mass input shown previously in Figure 1 by a factor of 7 to produce the Commanded Fuel Mass output.

The subjective driveability results related to Figures 2a and 2b were considered acceptable relative to results from conventional TFC algorithm tests. The envelope of the measured equivalence ratio shown in Figure 2a is typical of a well-calibrated transient fuel algorithm.

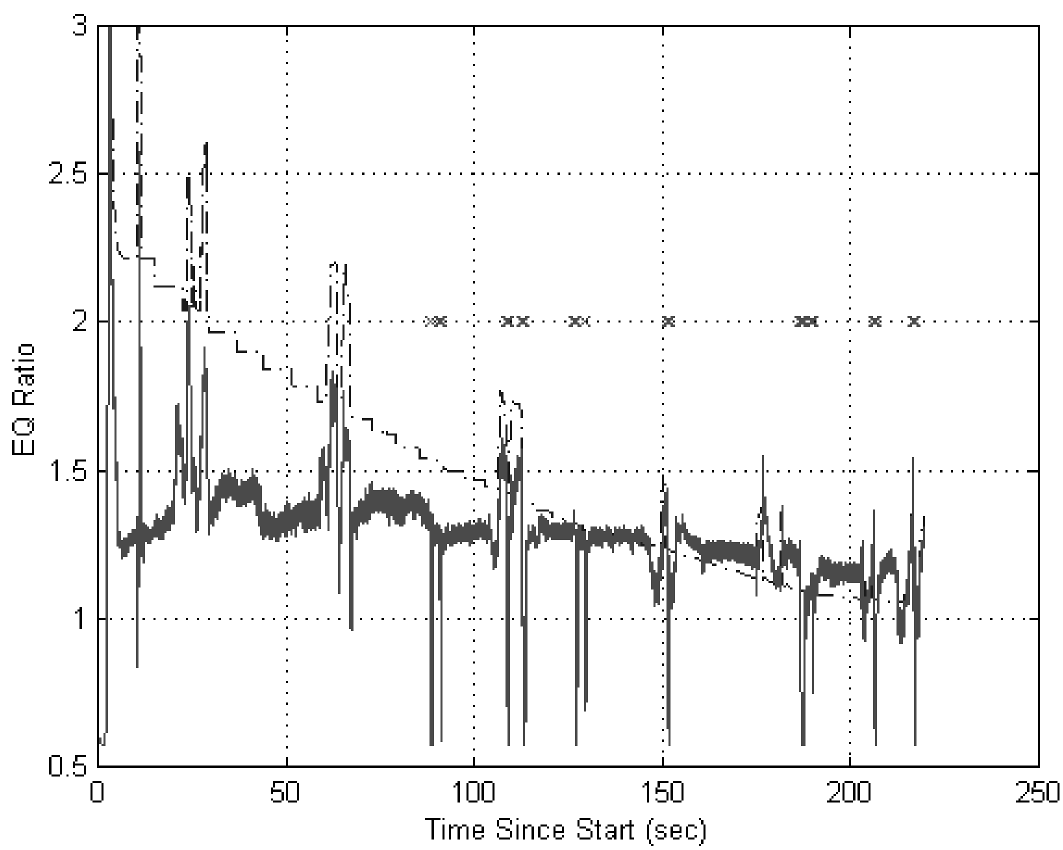

Figure 2a. Cold Driveability Test Results – Equivalence Ratio

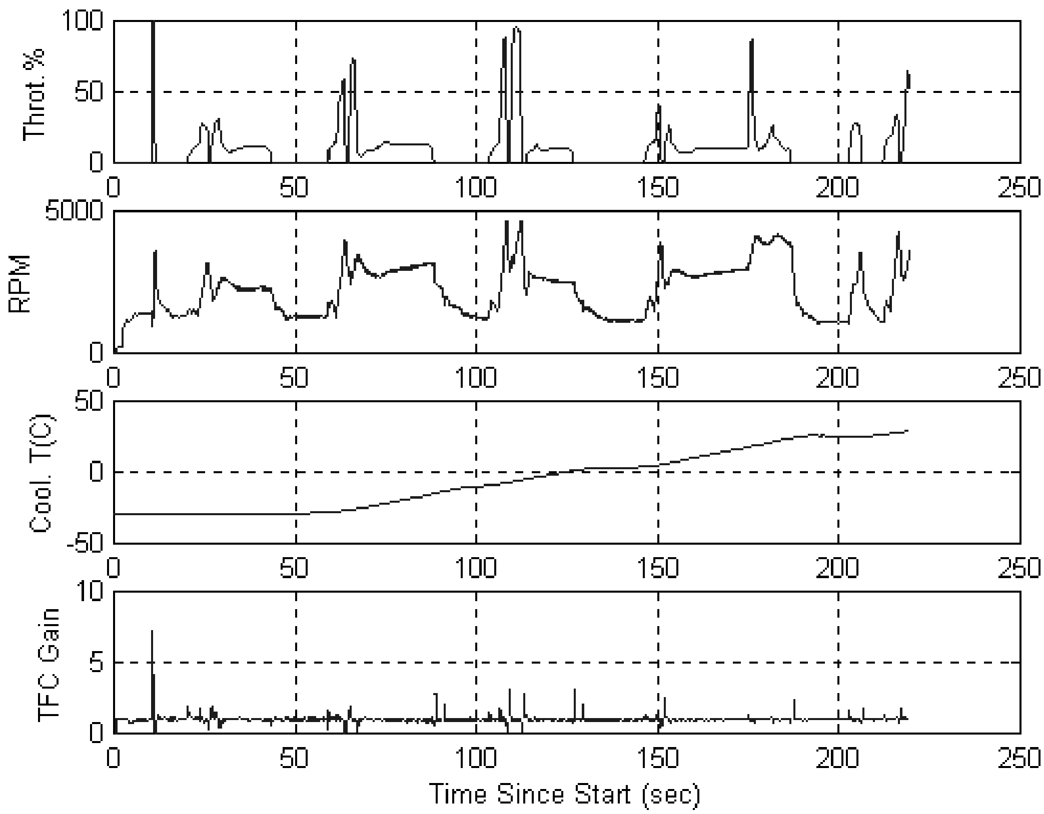

Figure 2b. Cold Driveability Test Results – Operating States and Compensator Gain

ALGORITHM COMPUTATIONAL REQUIREMENTS – The computational requirements of the physically based TFC algorithm were acceptable for production application based on a memory and throughput comparison to its conventional TFC counterpart algorithm. Table 1 shows comparable memory requirements between the new and conventional TFC.

Table 1. Comparison of Memory Requirements

Memory Type	New Physically Based TFC	Conventional TFC
RAM	0.11K	0.09K
ROM	4.4K	4.5K

Table 2 contains a comparison of throughput requirements for the new and conventional TFC algorithms. Three software execution-rate loops were used in the new algorithm. The Engine Event Loop is executed 4 times per engine cycle (2 crankshaft revolutions). As shown in Table 2, The new TFC algorithm had comparable throughput demand compared to its conventional counterpart.

Table 2. Comparison of Throughput Requirements

Execution Loop	New Physically Based TFC	Conventional TFC
Engine Event	176 µs	185 µs
7.81 ms	59 µs	48 µs
1 second	32 µs	N/A

CALIBRATION PARAMETER REDUCTION – A significant reduction in the number of adjustable calibration parameters associated with transient fuel calibration was achieved through the use of compact physical equations in place of large lookup tables. The complexity reduction was achieved primarily through the application of theory from the Chemical Engineering field to remove much of the ambiguity associated with the conventional set of calibration parameters.

Table 3 below summarizes the reduction in calibration parameters resulting from the application of the new TFC algorithm. The parameter reductions are broken down into mass transfer, injection dynamics, and intake valve temperature model categories.

The mass transfer category is for parameters related to the mass transfer mechanism from the fuel film to its surroundings, which was traditionally thought of as a boiling and distillation phenomenon [2]. The injection dynamics category is for parameters related to the assumed deposition rate of injected fuel on the fuel film, as defined in [2]. The intake valve temperature parameter category is for parameters related to a new model of intake valve temperature, which is included in the new TFC algorithm. Since the conventional TFC algorithm did not contain a valve temperature sub-model, valve temperature parameter comparisons are marked as "Not Applicable".

The design of the new TFC algorithm resulted in an overall decrease of 97.8% in calibratable TFC parameters.

Table 3. Reduction of Adjustable Calibration Parameters

Parameter Type	New TFC Parms	Conv. TFC Parms	Reduction
Film Mass Transfer	2	170	98.8%
Injection Dynamics	1	153	99.4%
Intake Valve Temperature	4	N/A	N/A
Totals	7	323	97.8%

ALGORITHM DESIGN

Figure 3 shows an inside-view of the TFC algorithm block from Figure 1. The TFC algorithm is composed of a Compensator Equation block, and three other blocks responsible for calculating optimal parameters for the compensator equation based on operating state inputs from the engine. The primary input and output of the TFC algorithm are the Desired Fuel Mass and Commanded Fuel Mass respectively.

The dynamic characteristics of the compensator are related solely to the Film Mass Remainder Fraction and the Gain Factor, which are respectively related to the fuel film time constant τ and impact factor X of [2].

As shown in Figure 3, an intake valve temperature model was used to provide thermal information to the Fuel Film Mass Transfer Calculation block. Both the Valve Temperature Model block and the Fuel Film Mass-Transfer Calculation block use the intake port airflow state, although for different purposes.

COMPENSATOR EQUATION-SET – The Compensator Equation block shown in Figure 3 contains the digital form of the well-known continuous transient fuel compensator equation-set from Aquino [2].

The compensator equation set consists of a mass-conservation equation for the fuel film, and an inverse of the mass-conservation equation for the fuel entering an engine cylinder. Equation set 1 below shows the continuous form of the compensator equations found in [2]. The first equation in the set relates the commanded (compensated) fuel flow rate to the desired fuel flow rate, and represents the inverse of the cylinder conservation equation. The second equation in the set represents the mass conservation of the fuel film.

$$\dot{M}_{cmd} = \frac{\dot{M}_{des} - M_f / \tau}{1 - X}$$

$$\dot{M}_f = X\dot{M}_{cmd} - M_f / \tau \qquad (1)$$

where

$\dot{M}_f$ is the net fuel film mass flow rate

$\dot{M}_{cmd}$ is the commanded fuel flow rate (output from TFC)

$\dot{M}_{des}$ is the desired fuel flow rate (input to TFC)

τ is the continuous time constant of fuel film mass-transfer

X is the fraction of injected fuel that impacts the fuel film

Since the TFC algorithm is event-based (1 update per cylinder, per cycle), the Compensator Equation block in Figure 3 contains the discrete form of Equation Set 1, produced by z-transform:

$$M_{cmd}(z) = \left(M_{des}(z) - (1-\alpha)\, M_f(z) z^{-1}\right) / \left(1-\beta\right)$$

$$M_f(z) = \alpha M_f(z) z^{-1} + \beta M_{cmd}(z) \qquad (2)$$

where

M_f is the fuel film mass

M_{cmd} is the commanded fuel mass (output from TFC)

M_{des} is the desired fuel mass (input to TFC)

α is the fraction of film mass remaining from previous cycle.

Note: $\alpha = \exp(-120 / (N\tau))$, N = engine RPM

β = compensator gain factor.

Note: $\beta = X$ where stability of Eqn. Set 2 permits

The stability aspects of the event-based compensator Equation-Set 2 were presented previously in [3]. Since the TFC algorithm is responsible for inverting the dynamics of the fuel film and injection, it was important to determine the values of α and β for which perfect compensation of fuel film dynamics is possible.

Equation 3 below represents a single transfer function form of Equation Set 2, which was necessary for stability analysis:

$$\frac{M_{cmd}(z)}{M_{des}(z)} = \frac{z - \alpha}{(1-\beta)z + \beta - \alpha} \qquad (3)$$

Figure 4 shows a z-plane representation of the pole and zero of Equation 3. For Equations 2 and 3 to be stable, the pole of Equation 3 must lie inside the unit circle. In physical terms, as the fuel film dynamics become faster $(\alpha \to 0)$, and/or the fraction of injected fuel impacting the film increases $(\beta \to 1)$, the compensator becomes less stable.

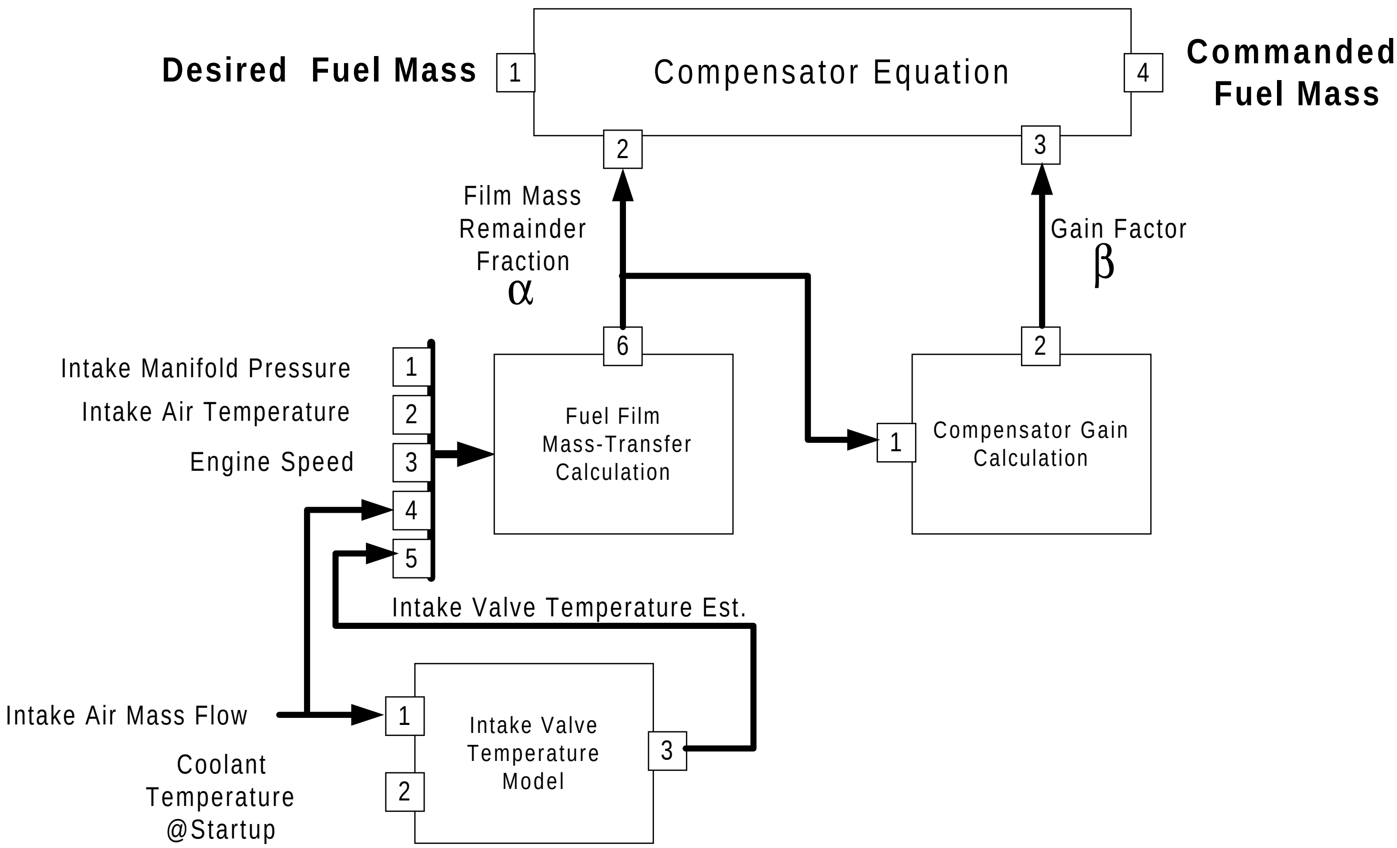

Figure 3. Transient Fuel Compensator Algorithm Components

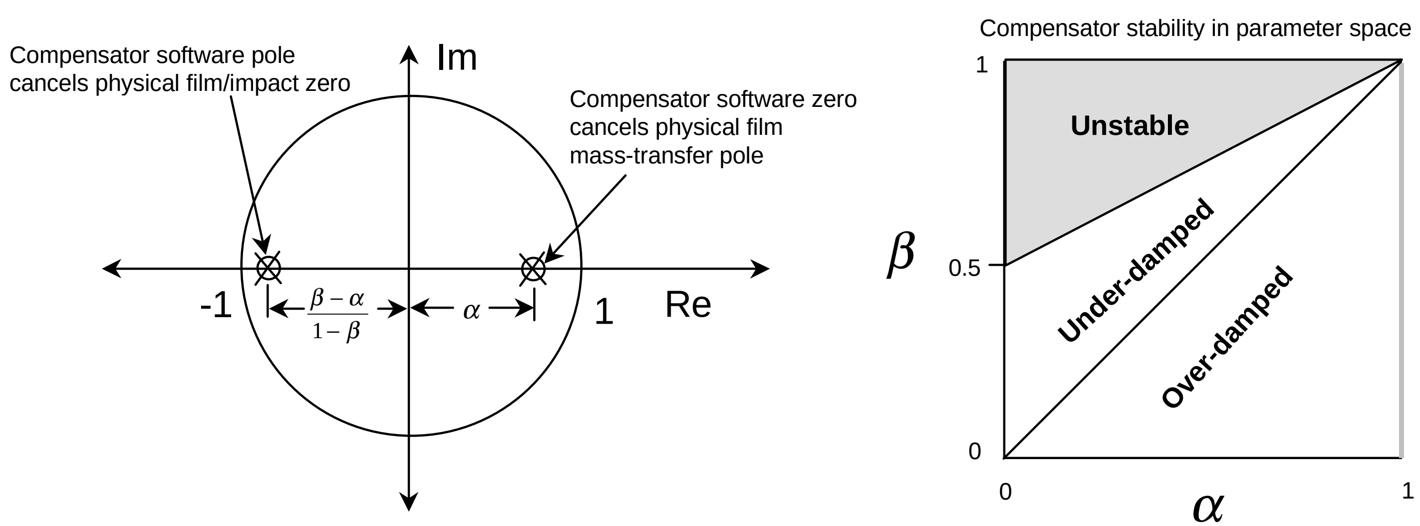

Figure 4. Relationship Between Compensator Stability and Physical Fuel Film Characteristics

The right portion of Figure 4 expresses the stability of Equations 2 and 3 in terms of the parameters α and β. Regions of unstable, under-damped, and over-damped compensator equation output behavior are shown.

As shown in Figure 4, the Compensator Equations 2 and 3 are capable of exhibiting oscillatory behavior. Oscillatory behavior can be necessary in application because the fuel injection system must increase fuel delivery on a positive transient to account for fuel which does not enter the cylinder on the first cycle, and then decrease fuel delivery on the next cycle to account for the fuel which impacted the film on the first cycle and left the film on the next cycle. It should be noted that the actual film mass and the model of the film mass in Equation 2 do not oscillate.

Under typical warm engine conditions, the actual fuel film characteristic parameters cannot always be used in compensator equation set 2, because the parameter coordinates can lie in the unstable region of the compensator equation shown Figure 4. In this paper, β is referred to in generic terms as a "gain factor" as opposed to the "impact factor" X because the use of the actual physical impact factor in the compensator would in some cases result in instability.

FUEL FILM MASS-TRANSFER CALCULATION – The Fuel Film Mass-Transfer Calculation block of Figure 3 is responsible for providing an estimate of the Fuel Remainder Fraction α to the Compensator Equation block. z-transform methods directly relate the event-based α the fuel film time constant τ of [2].

Conventional TFC algorithms typically determine τ as a calibratable function of engine states such as coolant temperature and intake manifold pressure through lookup tables containing hundreds of cells. Calibration of such a large number of parameters is difficult with little underlying theory of their relationship to the engine states.

Equation 4 below was used to increase the understanding of τ by describing the mass of fuel vapor leaving the fuel film M_f / τ in terms of a mass-transfer process. For simplicity, the fuel was assumed to behave as an average single component.

$$w = h_D A(c_0 - c_\infty) = \frac{M_f}{\tau MW_{fuel}}$$

$$\Rightarrow \frac{1}{\tau} = \frac{A}{M_f} MW_{fuel} h_D (c_0 - c_\infty) \tag{4}$$

where

w is the molar flow rate of fuel from film surface to air stream
MW_{fuel} is the molecular weight of the fuel
h_D is the mass-transfer coefficient relating molar flow rate to chemical concentration
A is the area of the film exposed to the gas stream
c_0 is the concentration of the fuel film at its surface
c_∞ is the concentration of the fuel vapor in the film surroundings

For practical purposes, the film vapor concentration c_∞ at locations far away from the fuel film surface was assumed to be negligible. The concentration of the fuel vapor at the film surface was related solely to the film temperature by using the ideal gas law with an empirical exponential function for the saturated vapor pressure at the film/air interface:

$$c_0 = \frac{P_{sat0} e^{-svc/T_f}}{R_u T_f} \tag{5}$$

where

P_{sat0} is a fit coefficient to empirical saturation pressure data from fuel properties reference
svc is a constant exponential coefficient for the calculation of saturated vapor pressure of the fuel (from fuel properties reference)
R_u is the universal gas constant
T_f is the absolute temperature of the fuel film

It was desirable to relate the fuel film concentration to a measurable quantity, so it was assumed that the film temperature could be approximated to the intake valve temperature:

$$T_f \approx T_v \tag{6}$$

where

T_v is a measured or modeled value of intake valve temperature

For further simplification, the ratio of fuel film area to film mass was assumed to be constant:

$$\frac{A}{M_f} \approx K_{ma} \tag{7}$$

where K_{ma} is a film mass to area ratio constant

Equations 1-6 allow τ to be expressed entirely of known measurable or fixed constants, except for the mass-transfer coefficient h_D.

In 1934 researchers Gilliland and Sherwood published the results of an empirical study of the diffusional resistance of a moving airstream to liquid films of 9 organic liquids. One of the liquids tested was Toluene, which has properties similar to a mid-range gasoline component. The results of this work allow h_D to be expressed in terms of measurable engine states.

Figure 5 shows the experimental setup. A vertical pipe, 2.67cm i.d. and 117cm long, was coated with a falling liquid film via a gauze weir. The film fell into a sump at the bottom of the apparatus so that it could be re-circulated to the top. Air flow was passed through the pipe to vaporize the film into the airstream. The rate of film vaporization was determined by monitoring the liquid level in the sump. A variety of film temperatures, pressures, and air flow rates were used to determine the vaporization rates of the 9 film types.

The experimental apparatus in Figure 5 was considered to be at least roughly analogous to an engine intake port and fuel film, assuming air flow transients are small-signal disturbances and film fluid dynamics are not significant. Gilliland and Sherwood found a well-correlated dimensionless equation that related the mass-transfer coefficient h_D of the 9 films to the Reynolds and Schmidt numbers of the gas stream through the pipe, with both co-current and counter-current flow:

$$h_D = 0.023 \frac{D}{d} \mathrm{Re}^{0.83} Sc^{0.44} \tag{8}$$

where

D is the diffusivity of the organic liquid
d is the diameter of the pipe that the air flows through
Re is the Reynolds Number of the flow through the pipe
Sc is the Schmidt Number of the flow through the pipe

The relationship between h_D and measurable engine states was made with the following substitutions for the Reynolds number, Schmidt number, and diffusivity:

$$\mathrm{Re} \propto \dot{m}_{air} \quad Sc \propto \frac{T_{air}}{P_{air}}, D \approx Constant \tag{9}$$

where

$\dot{m}_{air}$ is the air mass flow rate over the fuel film in one intake port
T_{air} is the absolute intake port air temperature
P_{air} is the absolute intake port air pressure

Equations 4-9 were combined to form an equation suitable for calculating α and τ in the Fuel Film Mass-Transfer block of the production TFC algorithm. Equation 10 below relates α and τ directly to measurable engine states with only one calibratable parameter, assuming a model of intake valve temperature is available.

Contrary to the conventional viewpoint that the fuel film time-constant increases with increasing intake manifold pressure, Equation 10 implies that the fuel film time-constant decreases with pressure, because in an engine the manifold pressure is linked to the more dominant air flow rate effect.

$$\frac{1}{\tau} = K_1 \dot{m}_{air}{}^{0.83} T_{air}{}^{0.44} P_{air}{}^{-0.44} T_v{}^{-1} e^{-svc/T_v} \tag{10}$$

and

$\alpha = \exp(-120/(N\tau))$, $N =$ engine RPM

where

K_1 is a constant calibration coefficient primarily related to the film mass to area ratio, port diameter, and the average diffusivity of the fuel film

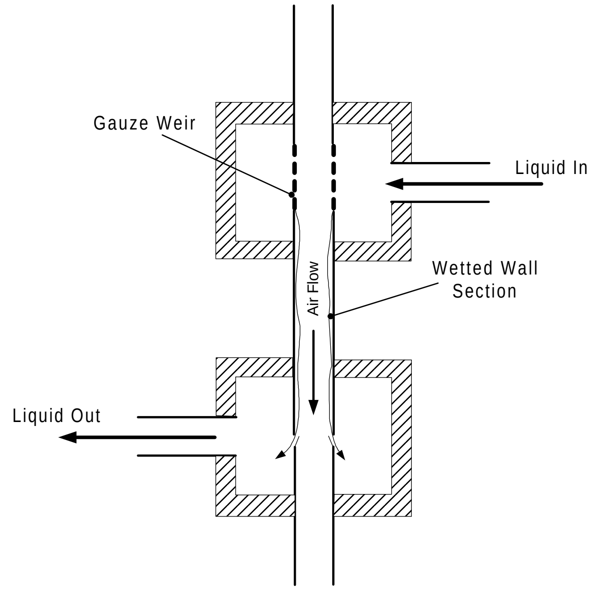

Figure 5. Experimental Setup of Gilliland and Sherwood (1934)

INTAKE VALVE TEMPERATURE MODEL – The calculation of the fuel film mass-transfer parameters α and τ from Equation 10 requires a modeled value of the intake valve temperature T_v . Figure 6 below shows the intake valve model used in the Intake Valve Temperature Model block of Figure 3.

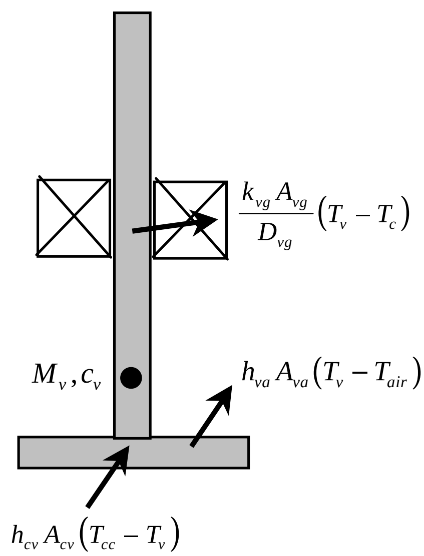

Figure 6. Intake Valve Temperature Model

Figure 6 shows an intake valve at temperature T_v heated by an average combustion chamber gas temperature T_{cc}, dissipating heat through convection to the surrounding intake air at temperature T_{air}, and dissipating heat through conduction via the valve guide and seat to coolant at temperature T_c.

Equation 11 below expresses the energy conservation of the valve mass M_v, treated as a thermal capacitance:

$$M_v c_v \dot{T}_v = h_{cv} A_{cv} \left(T_{cc} - T_v\right) - h_{va} A_{va} \left(T_v - T_{air}\right) - \frac{k_{vg} A_{vg}}{D_{vc}} \left(T_v - T_c\right) \quad (11)$$

where

- c_v is the specific heat of the valve material
- h_{cv} is the convection heat transfer coefficient from combustion chamber to valve
- A_{cv} is the valve area exposed to the combustion chamber
- h_{va} is the convection heat transfer coefficient from valve to intake air
- A_{va} is the valve area exposed to the intake air
- k_{vg} is the conduction heat transfer coefficient from valve to coolant
- A_{va} is the valve area for conduction heat transfer
- D_{vg} is the distance of the conduction path to the coolant

Equation 11 was expressed in terms of a valve temperature time-constant and forcing-function as follows:

$$\dot{T}_v = \underbrace{\frac{h_{cv} A_{cv}}{M_v c_v} T_{cc} + \frac{h_{va} A_{va}}{M_v c_v} T_{air} + \frac{k_{vg} A_{vg}}{M_v c_v D_{vc}} T_c}_{heat\_rate} - \underbrace{\left(\frac{h_{cv} A_{cv}}{M_v c_v} + \frac{h_{va} A_{va}}{M_v c_v} + \frac{k_{vg} A_{vg}}{M_v c_v D_{vc}} \right)}_{inverse\_time\_constant} T_v \quad (12)$$

The heat-rate term was assumed to be primarily a function of intake air mass flow rate, since it contains convection coefficients and combustion temperature. In like manner, the inverse time-constant term was assumed to be primarily a function of intake air mass flow since it also varies with convection coefficients.

Empirical work showed that the rate of change of valve temperature could be reasonably approximated by expressing the heat-rate and time-constant term as a linear function of air mass flow rate. The heat rate was expected to increase with air mass flow, and the time-constant was expected to decrease with increasing air mass flow, due to enhanced heat-transfer.

Equation 12 was cast in implementation form for the Intake Valve Temperature Model block as follows:

$$\dot{T}_v = K_2 + K_3 \dot{M}_{air} - \frac{T_v}{K_4 + K_5 \dot{M}_{air}} \quad (13)$$

where

- K_2 is an empirical coefficient for the heating rate offset
- K_3 is an empirical coefficient for the heating rate mass flow slope
- K_4 is an empirical coefficient for the valve temperature time-constant offset
- K_5 is an empirical coefficient for the valve temperature time-constant flow rate slope

COMPENSATOR GAIN CALCULATION – The Compensator Gain Calculation block in Figure 3 is responsible for calculating the compensator gain β. As was the case with τ, conventional TFC algorithms typically determine the "gain factor" β as a calibratable function of engine states such as coolant temperature and intake manifold pressure through lookup tables containing hundreds of cells.

The new TFC algorithm calculates the β parameter based on the idea that the actual physical impact factor X does not vary much with engine conditions, but the compensator gain factor β must move with the film dynamics reflected by α and τ for the compensator to insure stability under all operating conditions.

Figure 7 shows how the gain factor is selected in the Compensator Gain Calculation block.

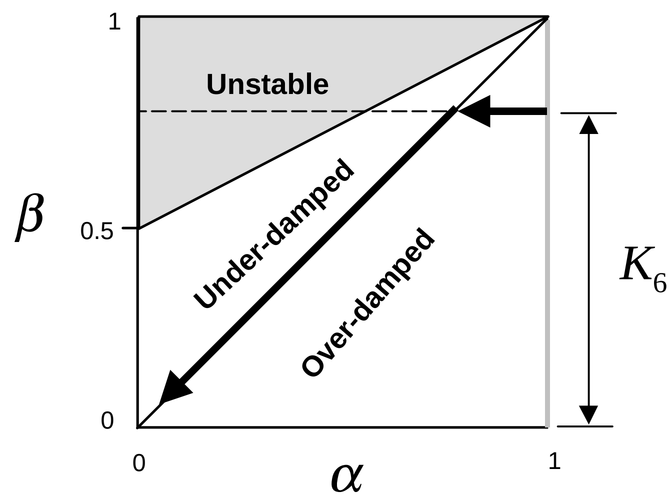

Figure 7. Stability-Constrained β Selection

The compensator gain-factor β is set equal to calibration constant K_6 where the fuel film dynamics are slow (e.g. cold conditions, low intake port air flow rates). K_6 represents the actual physical impact factor X of Equation 1 and [2] under very cold conditions.

As the intake port and valve warms up, α and τ decrease, but the actual physical impact factor X remains relatively constant. The compensator gain β is made to deviate from the actual physical impact factor X to preserve compensator stability where perfect compensation is no longer possible. β is made to follow the critical damping line (α = β) in Figure 7 for any values of α less than K_6 .

The Compensator Gain Calculation block relates β directly to fuel film dynamics with only on calibration constant K_6, and conservatively specifies critically damped compensator characteristics under warm conditions to avoid compensator instability.

ALGORITHM CALIBRATION

Calibration of the new TFC algorithm involved intake valve temperature model-fitting of 4 parameters, and activities for determining the 3 parameters related to α and β.

INTAKE VALVE TEMPERATURE MODEL FITTING – The intake valve temperature model coefficients K_2 to K_5 were calibrated by acquiring engine intake port flow rate and measured intake valve seat temperature as the engine warmed up under constant engine speed and flow conditions. Three warmup tests were done at low, middle, and high intake air flow rates starting from a 20C initial intake valve seat temperature. It was assumed that the intake valve seat temperature was a reasonable approximation of the intake valve poppet temperature for measurement convenience.

A least-squares fitting routine was used to post-process the data of the three warmup tests to determine the best-fit K_2 to K_5 coefficients. Figure 8 below shows the modeled and measured intake valve temperature under dynamic conditions during first part of a US 23 FTP test. The maximum errors in the valve temperature model were approximately 3%.

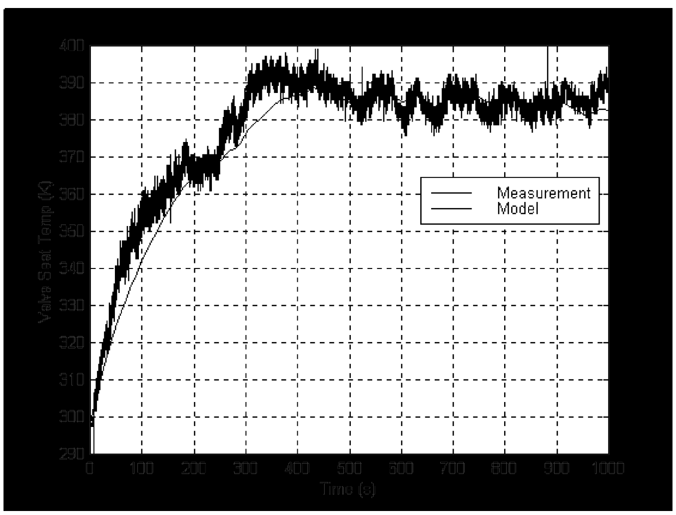

Figure 8. Intake Valve Temperature Model Quality

TRANSIENT FUEL PARAMETER CALIBRATION – Calibration coefficients K_1 and K_6 were determined from fuel perturbation tests at constant airflow, carried out from cold to hot intake valve temperature conditions (-30C to 117C) during engine warm-up at closed-throttle idle.

Calibration methods based on fuel-only perturbation as in [4] and combined fuel-air perturbation as in [5] were used to determine the true physical X and τ parameters as the engine warmed up from a –30C starting temperature. A least-squares fitting routine was used to fit K_1 from measured values of τ, $\dot{m}_{air}$ P_{air} , T_{air} , and T_v . K_6 was determined from the measured impact factor X at the beginning of the –30C test.

SUMMARY

An event-based transient fuel compensator (TFC) algorithm was developed to reduce production development complexity associated with transient fuel calibration. A physically based transient fuel compensator with clear stability constraints was developed to significantly decrease the number of adjustable calibration parameters required for successful cold driveability and emissions validation in a production application.

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