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HYDROMECHANICAL CONTROL FOR A VARIABLE DELIVERY, POSITIVE DISPLACEMENT FUEL PUMP

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ABSTRACT

Fuel pumps for gas turbine engines have traditionally been fixed delivery, positive displacement type pumps. The critical pump sizing criteria are typically the fuel flow and pressure needed for engine lightoff at cranking speeds (approximately 10% of full speed). However, these pump sizing criteria result in excess fuel delivery at higher engine speeds and altitudes. This excess fuel is typically bypassed back to the pump inlet, resulting in significant fuel heating. In contrast, a variable delivery, positive displacement fuel pump has the ability to vary delivery flow, to thereby match engine demands for a wide range of engine speeds and altitudes. This eliminates the excessive fuel delivery and resulting heat generation inherent in fixed delivery pumps. An approach to controlling fuel flow delivered to the engine is presented in which the differential pressure across a fuel metering valve is regulated by simultaneously varying the pump displacement and a small amount of bypass flow. This approach results in improved transient response and steady state accuracy at all operating conditions, as compared with alternate methods.

NOMENCLATURE

A_pcv	Pump control valve spool end area
B	Fuel bulk modulus
D	Pump control valve damping constant
FADEC	Full authority digital engine control
Fnet_pcv	Net force on pump control valve spool
F_press	Net pressure force on pump control valve spool
F_spring	Pump control valve spring force
G_1	Integral path gain from pump control valve position to rate of change of pump flow
G_2	Proportional path gain from pump control valve position to bypass flow, at constant pump discharge pressure
G_3	Common gain from pump control valve position to metering valve differential pressure
m	Pump control valve mass
MA	Milliamps
MVPOS	Metering valve position
MATRIXx	Registered trademark of Integrated Systems, Inc.
OLTF	Open loop transfer function
Pb	Engine burner pressure
Pd	Pump interstage (drain) pressure
Pf1	Main pump discharge pressure
Pf2	Pressure downstream of metering valve
Pf3	Pressure upstream of engine nozzles
PRV	Pressure regulating valve
q_byp	Volumetric flow bypassed to pump interstage
q_mv	Metering valve volumetric flow
q_or	Pump control valve damping orifice volumetric flow
q_pump	Main pump volumetric flow
V	Main pump discharge volume
Wf	Fuel weight flow delivered to the engine
WFMD	Metering valve weight flow
X_act	Pump actuator position
Xdot_pcv	Pump control valve velocity
X_pcv	Pump control valve position
ΔP	Differential pressure
ΔP_mv	Fuel metering valve differential pressure
ΔP_or	Pump control valve damping orifice differential pressure
τ_d	Pump control valve mass time constant
τ_v	Pump discharge pressure volume time constant

CONTROL TECHNIQUES FOR POSITIVE DISPLACEMENT PUMPS

Positive displacement pumps have been overwhelmingly selected as the main fuel pump for gas turbine engine control systems, particularly when high flows and pressures are required over a wide range of drive speeds (Petro, 1972). Two of the most widely used types of positive displacement pumps are gear pumps and vane pumps.

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Historically, the control of desired engine flow from a positive displacement pump has been obtained by three techniques:

Variable pump displacement - Requires precise control of pump geometry.
Control of drive shaft speed - Requires precise control of accessory drive speed.
Bypassing of excess flow - Requires an oversized pump and additional valving.

FIXED DISPLACEMENT PUMP CONTROL DESIGN TRADITIONAL APPROACH - BYPASS OF EXCESS PUMP FLOW

Of the three control techniques, the overwhelming choice has been to bypass excess flow from a fixed displacement gear pump. This approach reduces technical risk and capitalizes on components with very high mechanical reliability (Petro, 1972). A block diagram representation of this approach is illustrated in Figure 1.

Fuel flow output from the pump enters a fuel metering unit. A full authority digital engine control (FADEC) determines the desired engine flow, and controls the position of a fuel metering valve with a varying flow window using a feedback signal from a position transducer on the metering valve. The heart of the fuel metering system is the pressure regulating valve (PRV). The PRV senses the differential pressure (ΔP) across the metering valve, and bypasses flow back to the inlet of the pump as required to maintain a constant, regulated pressure drop across the metering valve. In this way, desired fuel flow is directly proportional to metering valve position.

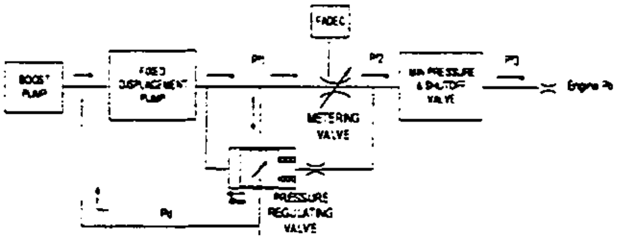

FIGURE 1. FIXED DISPLACEMENT PUMP WITH BYPASS REGULATOR

In this approach, the critical pump sizing criteria are typically the fuel flow and pressure required for engine lightoff at cranking speeds, which is approximately 10% of full speed. However, these pump sizing criteria result in excessive fuel delivery at higher engine speeds and altitudes, which represents the majority of normal engine operation. As a result, actual pump delivery flow in excess of required engine flow is bypassed back to the pump inlet. The ratio of bypass flow to engine flow can be 20 or higher for aircraft gas turbine engines (Gibson and Fox, 1970). A major problem with this result is that bypass and recirculation of fuel results in significant fuel heating due to the pressurizing of the fuel by the pump and the subsequent pressure drop of the fuel in the bypass line upstream of the pump. The high fuel temperature is a problem since the fuel is typically used as a heat exchanger medium in the engine.

With the latest fuel efficient engine designs, excessive fuel heating becomes a serious problem. Reduced engine fuel consumption is accompanied by increased engine and oil lubrication system temperatures. Excess oil lubrication system heat is normally managed with a combination of fuel/oil and air/oil heat exchangers.

However, heat exchangers are undesirable in this situation because of their associated size, weight, and cost. Air/oil coolers are problematic because of the drag penalty they incur on the aircraft. Yet, the cooling burden on an air/oil cooler is decreased with lower fuel temperatures. This is because the lower fuel temperatures permit more lubrication system heat to be directed to the fuel system through the fuel/oil heat exchanger. This can result in a significant reduction in heat exchanger system size, weight, and cost, as well as a reduction in the drag penalties associated with air/oil coolers (Reuter and Gaudet, 1998).

VARIABLE DISPLACEMENT PUMP

In contrast to fixed delivery pumps, variable delivery, positive displacement fuel pumps have the ability to vary delivery flow, to thereby match engine demands for a wide range of engine speeds and altitudes. Thus, the variable delivery pump eliminates the excessive fuel delivery and resulting heat generation inherent in fixed delivery pumps. An actuator is typically used to alter the pump displacement. In the following discussion, three approaches to control flow by modulating a variable displacement pump actuator are presented.

VARIABLE DISPLACEMENT PUMP CONTROL DESIGN APPROACH #1 - REGULATE METERING WINDOW DIFFERENTIAL PRESSURE BY VARYING PUMP DISPLACEMENT

Fundamentally, this approach is similar to the traditional control method that has been used with fixed displacement pumps. A FADEC controls the position of a variable metering window in the pump flow path, with hydromechanical regulation of the ΔP across the metering window. The difference here is that instead of bypassing flow to regulate ΔP, pump displacement is varied.

In this type of control scheme, illustrated in Figure 2, pump displacement is typically altered by an actuator driven by a pilot valve. The spring biased pilot valve senses pressure both upstream and downstream of the metering window. As pump flow conditions change, a different metering valve pressure drop is sensed by the pilot valve. In response, the pilot valve translates and moves the pilot valve windows from their null position. This causes the pump actuator to stroke, thereby varying pump displacement until the desired and constant metering valve ΔP is restored.

However, a problem with this control scheme is its inability to quickly and adequately respond to sudden external disturbances in flow. These disturbances occur if the fuel pump also provides flow to slew an actuator which is used to position engine components such as a stator vane or bleed valve. The bandwidth of this pressure drop control is limited by the integrating nature of the pump actuator servo system. If the servo system response could be improved, the control system bandwidth could increase. However, the increase in control system bandwidth is limited by the requirements for control stability at all operating conditions.

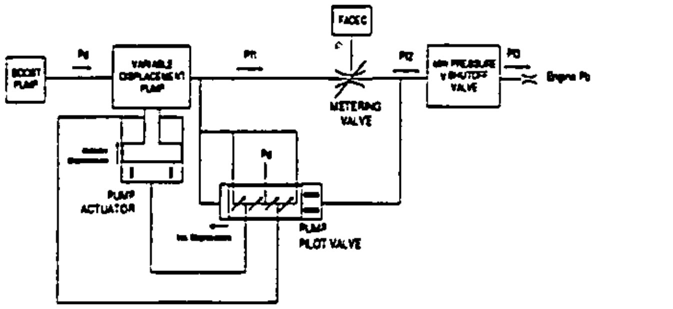

FIGURE 2. REGULATE METERING WINDOW ΔP BY VARYING PUMP DISPLACEMENT

A second inherent drawback in the pressure drop control scheme is its sensitivity to pump servo friction. A pump control actuator inherently has a large amount of friction, which causes a flow scheduling deadband, which can lead to inaccuracies and instability.

VARIABLE DISPLACEMENT PUMP CONTROL DESIGN APPROACH #2 - DIGITAL CLOSED LOOP CONTROL USING FLOW SENSOR FEEDBACK

A flow control for a variable displacement, fixed delivery pump may be carried out with a loop closure through the FADEC. The FADEC detects pump fuel flow to the engine via a flow sensor located in the engine fuel flow delivery line. The FADEC software compares actual flow with desired flow, and based on the difference, varies pump displacement by positioning a pump servo system via an electromechanical interface device, such as an electrohydraulic servovalve, until commanded flow matches delivered flow. A block diagram representation of this approach is illustrated in Figure 3.

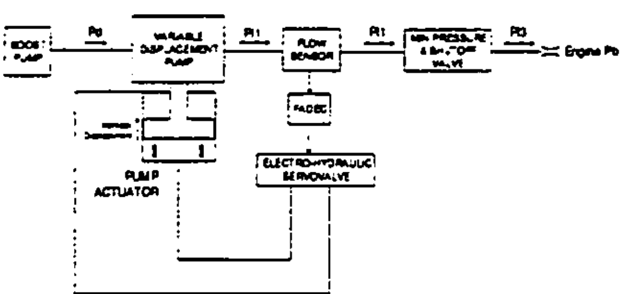

FIGURE 3. CLOSED LOOP ON FLOW BY VARYING PUMP DISPLACEMENT

A benefit of the flow sensor and FADEC approach is that a variable metering window with a fixed regulated pressure drop is not required. Elimination of this pressure drop across the metering window reduces the total system pressure drop. This reduces the maximum working pressures at maximum fuel flow conditions, as well as reduces the required pump head at starting conditions. In addition, control gains as well as engine flow schedules can easily be altered electronically through software, rather by hardware implementation.

However, meeting control system dynamic response requirements can be a problem with the closed loop on flow approach. The bandwidth of the flow control loop is limited by both flow sensor dynamic response and FADEC digital sampling delays. To achieve the necessary flow control loop bandwidth, digital sampling rates much higher than 100 Hz are required, which results in increased FADEC cost and complexity. The selection of a flow sensor requires a trade-off between sensor dynamic response and accuracy. In addition, the electronic control scheme exhibits the same sensitivity to pump servo friction as the pressure drop regulation scheme.

VARIABLE DISPLACEMENT PUMP CONTROL DESIGN APPROACH #3 - REGULATE METERING VALVE DIFFERENTIAL PRESSURE BY SIMULTANEOUS VARIATION OF PUMP DISPLACEMENT AND BYPASS FLOW

A preferred method of controlling flow using a variable displacement pump is to regulate the ΔP across a metering window using a pump control valve which combines two previously discussed techniques. The pump control valve varies pump displacement as in approach #1. In addition, the pump control valve simultaneously varies the amount of bypass flow, in a manner similar to the traditional fixed displacement pump approach. One major difference of this approach is that only a small amount of flow is bypassed, and thus the resultant fuel heating is insignificant. The addition of this bypass flow path results in improvements in dynamic response, stability, and steady state accuracy.

A schematic of this approach is illustrated in figure 4. Fuel flow output from the pump enters a fuel metering unit and initially encounters a metering valve with a variable flow window. The metering valve position is controlled by the FADEC as in other approaches, using LVDT position feedback.

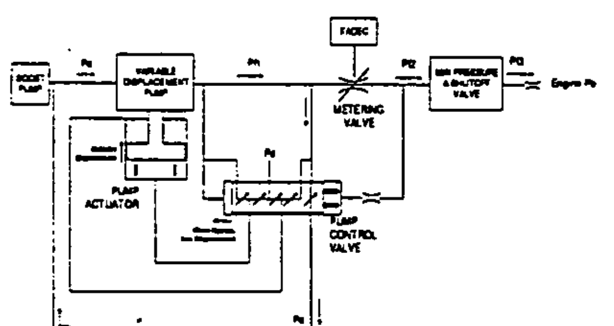

FIGURE 4. REGULATE METERING VALVE ΔP BY VARYING BOTH PUMP DISPLACEMENT AND BYPASS FLOW

The pressures at two different positions across the metering window are transmitted to the corresponding ends of a spool of the pump control valve. The spring load on the pump control valve spool equals the difference of the spool pressures multiplied by the piston area. The pump control valve spool has 4-way pilot valve lands to provide flow to/from both sides of the pump control actuator. An additional land is used to bypass flow from pump discharge back to pump inlet.

As the metering valve window area changes, due to external changes in pump flow demand, the pump control valve spool senses the changed ΔP across the metering valve window, and moves in a direction to restore ΔP back to its regulated value. For example, an increase in flow demand results in an initial drop in metering valve ΔP. In response, the pump control valve spool moves to command an increase in pump actuator displacement, while simultaneously bypassing less pump output flow. As metering valve ΔP returns to its regulated value, the pump control valve spool returns to its null position. At this null position, which is always the same, the pump actuator velocity is zero, the pump actuator is in steady-state, and a small amount of flow is bypassed.

Since the pump control valve null position does not vary, the steady-state bypass flow area is constant, and the amount of steady-state bypass flow varies as a function of the square root of pump ΔP. Pump ΔP typically varies by a factor of 4:1, resulting in a 2:1 variation in bypass flow from min-to-max engine flow.

DYNAMIC ANALYSIS OF PUMP CONTROL VALVE ΔP REGULATOR

The dynamic analysis of the pump control valve ΔP regulator includes a linearized representation of the ΔP control loop, shown in Figure 5.

The state variables associated with the ΔP control loop are pump control valve position (X_pcv) and velocity (Xdot_pcv), pump actuator position (X_act), and pump discharge pressure (Pf1). Starting at the left side of the block diagram, the summation of forces on the pump control valve (Fnet_pcv) is calculated based on the pressure force (F_press) and the spring force (F_spring). The pressure force consists of the metering valve ΔP plus the ΔP across the pump control valve damping orifice. Pump control valve acceleration is the net force divided by the mass; acceleration is integrated to calculate velocity (Xdot_pcv), which is integrated to calculate position (X_pcv).

There are two parallel paths downstream of pump control valve position. One path represents the effect of pump control valve position on pump flow (q_pump), while the other path represents the effect on bypass flow (q_byp). It should be noted that the pump flow path is an integral path, while the bypass flow path is proportional. The ratio of these integral and proportional gains are critical to ΔP control loop stability.

The net flow at pump discharge is represented by the summing junction which adds pump flow (q_pump) coming into the volume, and subtracts bypass flow (q_byp) and metering valve flow (q_mv) which exit the volume. Rate of change of pressure is calculated by multiplying the net flow by the fuel bulk modulus (B), and dividing by the volume (V). This rate of change of pressure is integrated to calculate pump discharge pressure (Pf1). Feedback paths represent the effect of Pf1 on bypass flow and metering valve flow. A change in Pf1 affects the ΔP across the metering valve (ΔP_mv).

Insight into the effect of individual terms on the ΔP control loop stability can be gained through evaluation of the open loop transfer function using block diagram reduction. The result of this reduction is illustrated in Figure 6

This block diagram reduction is accomplished by combining terms as follows.

$$D = A\_pcv^2 \frac{d(\Delta P\_or)}{d(q\_or)} \text{ (pump control valve damping)}$$

$$\tau_d = \frac{m}{d} \qquad \text{(pump control valve mass time constant)}$$

$$G_1 = \frac{d(xdot\_act)}{d(x\_pcv)} \frac{d(q\_pump)}{d(x\_act)} \text{ (integral path gain)}$$

$$G_2 = \frac{d(q\_byp)}{d(x\_pcv)} \qquad \text{(proportional path gain)}$$

$$G_3 = \frac{A\_pcv * \dfrac{d(\Delta P\_mv)}{d(Pf_1)}}{\left[\dfrac{\partial(q\_byp)}{\partial(Pf_1)} + \dfrac{d(q\_mv)}{\partial(Pf_1)}\right]} \text{ (common path gain)}$$

$$\tau_v = \frac{V/B}{\left[\dfrac{\partial(q\_byp)}{\partial(Pf_1)} + \dfrac{d(q\_mv)}{\partial(Pf_1)}\right]} \text{ (volume time const.)}$$

There are 2 paths from pump control valve position to the pump control valve net force, a mechanical path via the spring, and a fluid path via pump control valve ΔP. The mechanical path is small relative to the fluid path, and will be ignored in the subsequent analysis. The open ΔP control loop transfer function can be expressed as:

$$OLTF = \frac{(G_1 + G_2 s)G_3/D}{s^2(\tau_d s + 1)(\tau_v s + 1)}$$

Using this open loop transfer function, the stability and response of the ΔP control loop can be evaluated. Typical fuel control ΔP regulators have a bandwidth in the range of 50 rad/sec (0.020 sec equivalent time constant), which corresponds to an open loop crossover frequency of approximately 50 rad/sec. For the pump control valve ΔP regulator to achieve this response while maintaining adequate stability margin, the secondary time constants τ_d and τ_v must be substantially smaller than 0.020 sec, and the ratio of integral gain to proportional gain, G_1 / G_2, must be less than 50 rad/sec.

For the traditional bypass type ΔP regulator with a fixed displacement pump, $G_1 = 0$, and stability margin is acceptable as long as G_2G_3/D is not excessively large. Gain G_2 is a function of the bypass window gain, and D is determined by the damping orifice diameter. These parameters can be sized to yield good stability and response.

For variable displacement pump control design approach #1, which has no bypass flow, $G_2 = 0$, resulting in an unstable ΔP control loop if the pump pilot valve spring rate is ignored. However, this configuration can be made to be stable by incorporating a spring with a rate much higher than that used on conventional bypass regulating valves.

Based on this analysis, stability margin is improved as the proportional path gain G_2, from pump control valve stroke to bypass flow, is increased. A decrease in the integral path gain G_1, from pump control valve stroke to rate of change of pump flow, also improves stability. Hence, the implementation of the bypass path in approach #3 results in improved ΔP control loop stability, relative to approach #1, with acceptable dynamic response. At a low engine flow condition, the ΔP regulator was calculated to have an equivalent time constant of 0.003 sec. This increases to 0.012 sec at a high engine flow condition, primarily due to the fact that gain G_3 is lower at higher engine flows.

SIMULATION OF PUMP CONTROL VALVE ΔP REGULATOR

A detailed, non-linear MATRIXx simulation was built to demonstrate the predicted performance of the variable displacement vane pump with the pump control valve based ΔP regulator. In addition to the pump and fuel metering system, the simulation also includes a FADEC which incorporates a metering valve position control loop with an equivalent time constant of 0.04 sec.

Figures 7 and 8 illustrate the predicted response to a 100 PPH step change in Wf demand at a low and high engine flow condition. At the 300 PPH low flow condition, the dynamics of the ΔP regulator are fast relative to the dynamics of the metering valve position control loop.

Hence, the plot of Wf engine is dynamically similar to the plot of metering valve position. At this condition, the steady state bypass, or spill, flow is approximately 1000 PPH, much less than that associated with a conventional bypass regulator and fixed displacement pump. At the 24,000 PPH high flow condition, the Wf engine response illustrates a slight overshoot, which is related to the effect of pump actuator friction. The bypass flow at this condition is approximately 1800 PPH. Figure 9 compares the step response at 24,000 PPH with and without friction, indicating that the response without friction does not exhibit overshoot.

An advantage of the hydromechanical ΔP regulator approach (#3) over the closed loop on flow approach (#2) is improved rejection of flow disturbances. These typically occur when pump discharge flow is suddenly diverted to move actuators which are used to position external components such as engine stator vanes and bleed valves. With the closed loop on flow approach, the response to these disturbances is limited by the FADEC sampling time, which is typically 0.012 sec. More frequent sampling results in increased FADEC cost and complexity.

Figure 10 illustrates the response of the hydromechanical ΔP regulator to a 4500 PPH flow disturbance. The resultant Wf engine transient is settled out within 0.10 sec, which would have a negligible impact on thrust.

The hydromechanical ΔP regulator approach (#3) compares favorably against the closed loop on flow approach (#2) when the effects of pump actuator friction are considered. With the ΔP regulator approach, friction has a minor effect on steady state accuracy. This is because the pump actuator is only one of two paths used to regulate ΔP. When the pump actuator is in its friction band, the bypass regulation path is still active.

For the closed loop on flow approach, a software integrator in the FADEC logic is required to correct for steady-state errors which would occur as the result of actuator friction. This software integrator degrades Wf loop stability. In order to achieve both stability and dynamic response requirements, a pump actuator position control loop, which is nested within the Wf closed loop, will most likely be necessary. A MATRIXx simulation of control design approach #2 with this FADEC control loop architecture was created.

Figures 11 and 12 illustrate the response of the closed loop on Wf approach, with and without pump actuator friction. In this case, friction results in a hysteresis from commanded to actual pump actuator position, which has a de-stabilizing effect on the Wf control loop.

FUEL LAB BENCH TEST RESULTS FROM ΔP REGULATOR

A fuel control unit incorporating the ΔP regulator approach was built and tested both in the fuel lab and on an engine. Frequency response testing of the first unit indicated that pump control valve damping was greater than analytically predicted, which necessitated an iteration in the pump control valve windows. With the modified windows, the unit exhibited stable dynamic response, as illustrated in Figure 13 which was recorded in the Hamilton Standard hot fuel lab.

SUMMARY

Three different approaches to control engine fuel flow, using a variable delivery, positive displacement vane pump, were considered. Approach #1, which controls metering valve ΔP by varying pump displacement only, is bandwidth limited in order to achieve acceptable stability. Approach #2, which is a closed loop on flow approach, has disturbance rejection limitations, and is susceptible to limit cycling due to pump actuator friction. Approach #3, which controls metering valve ΔP by simultaneously varying pump displacement and bypass flow, has superior dynamic response and stability when compared with the other approaches. The amount of fuel temperature increase due to the bypass flow associated with this approach is insignificant.

ACKNOWLEDGEMENT

I would like to acknowledge Charles Reuter, a mechanical systems design engineer for the Hamilton Standard Division of United Technologies. His original ideas initiated the development of the pump control valve concept presented in this paper.

REFERENCES

Gibson, W. H. and Fox, L., *A New Development in High Speed Variable Delivery Vane Pumps*, SAE Paper No. 700792 presented at the National Aeronautic and Space Engineering and Manufacturing Meeting, Los Angeles, CA, October 5-9, 1970

Petro, David, *Fuel Pump Design Considerations for Aircraft Gas Turbine Engines*, presented at the Joint Gas Turbine Division / Fluids Engineering Division Conference of the ASME, San Francisco, CA, March 26-27, 1972

Reuter, Charles and Gaudet, Timothy, *Hydromechanical Control for a Variable Delivery, Positive Displacement Fuel Pump*, United States Patent No. 5,715,674, February 10, 1998

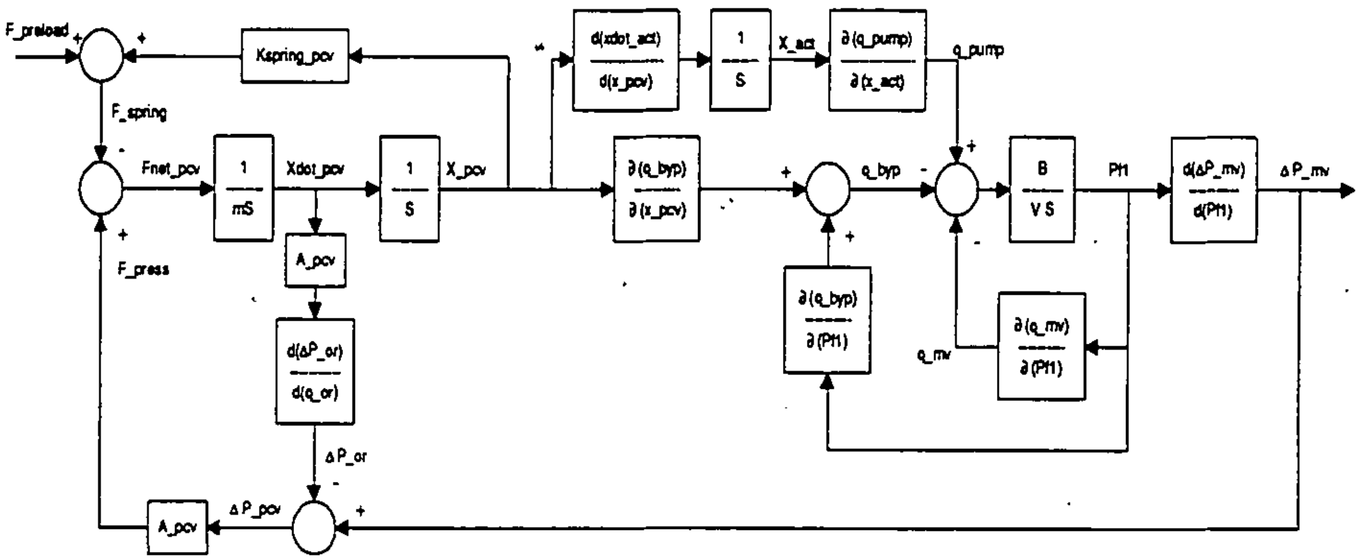

FIGURE 5. LINEARIZED BLOCK DIAGRAM OF ΔP REGULATOR

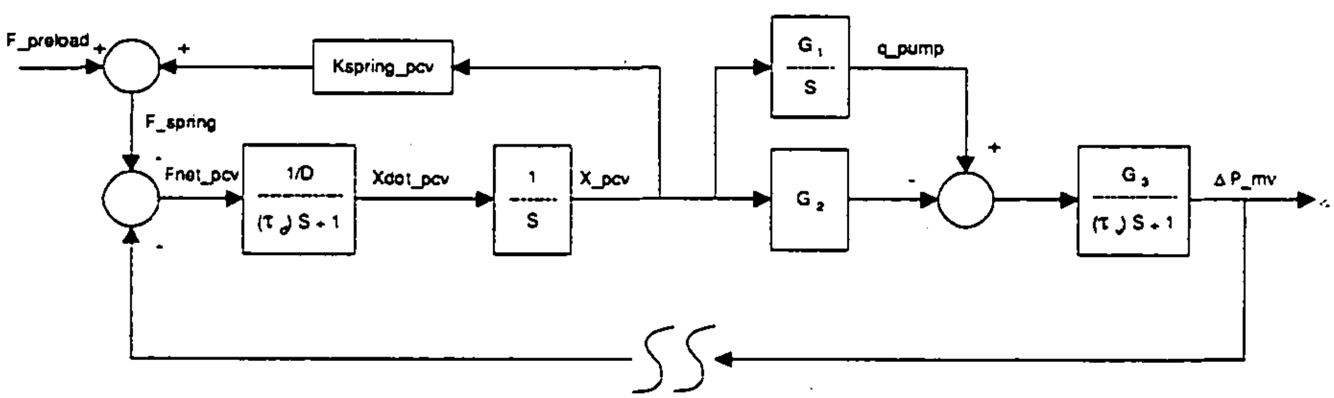

FIGURE 6. REDUCED LINEARIZED BLOCK DIAGRAM OF ΔP REGULATOR

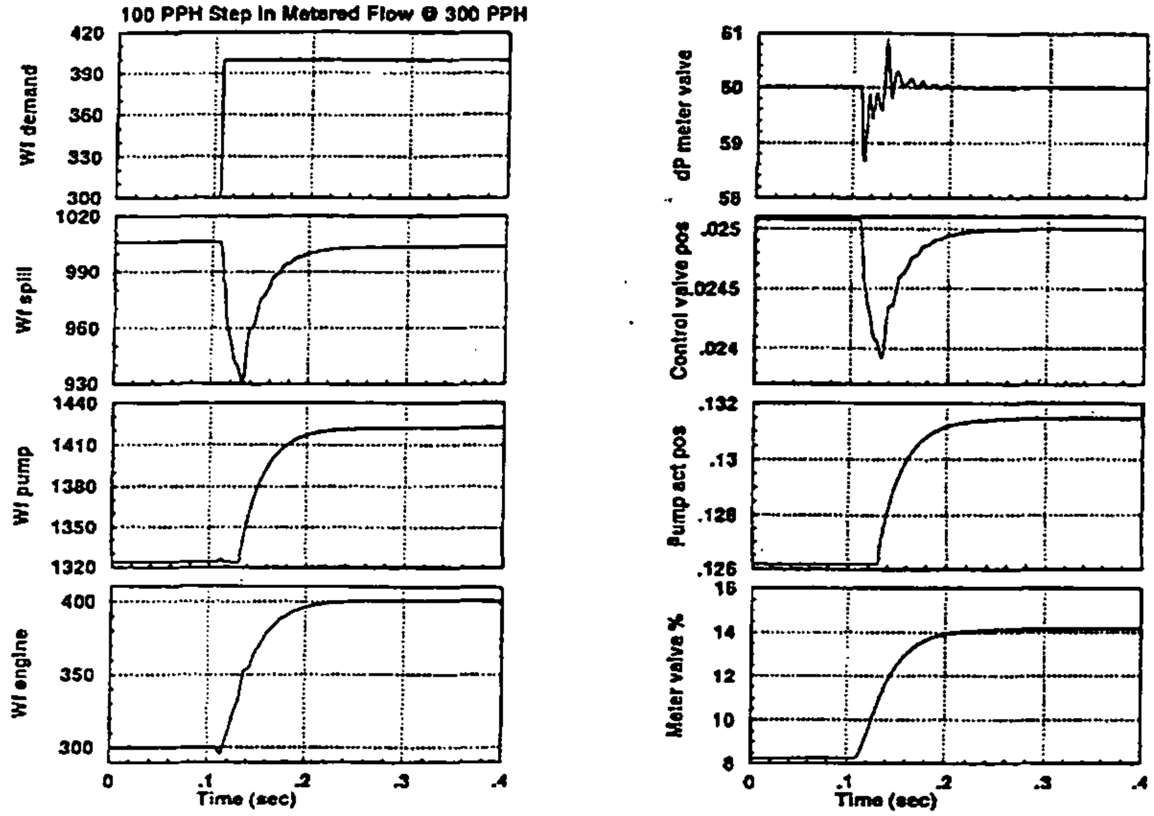

FIGURE 7. SIMULATION OF ΔP REGULATOR BASED FUEL CONTROL STEP RESPONSE AT LOW WF

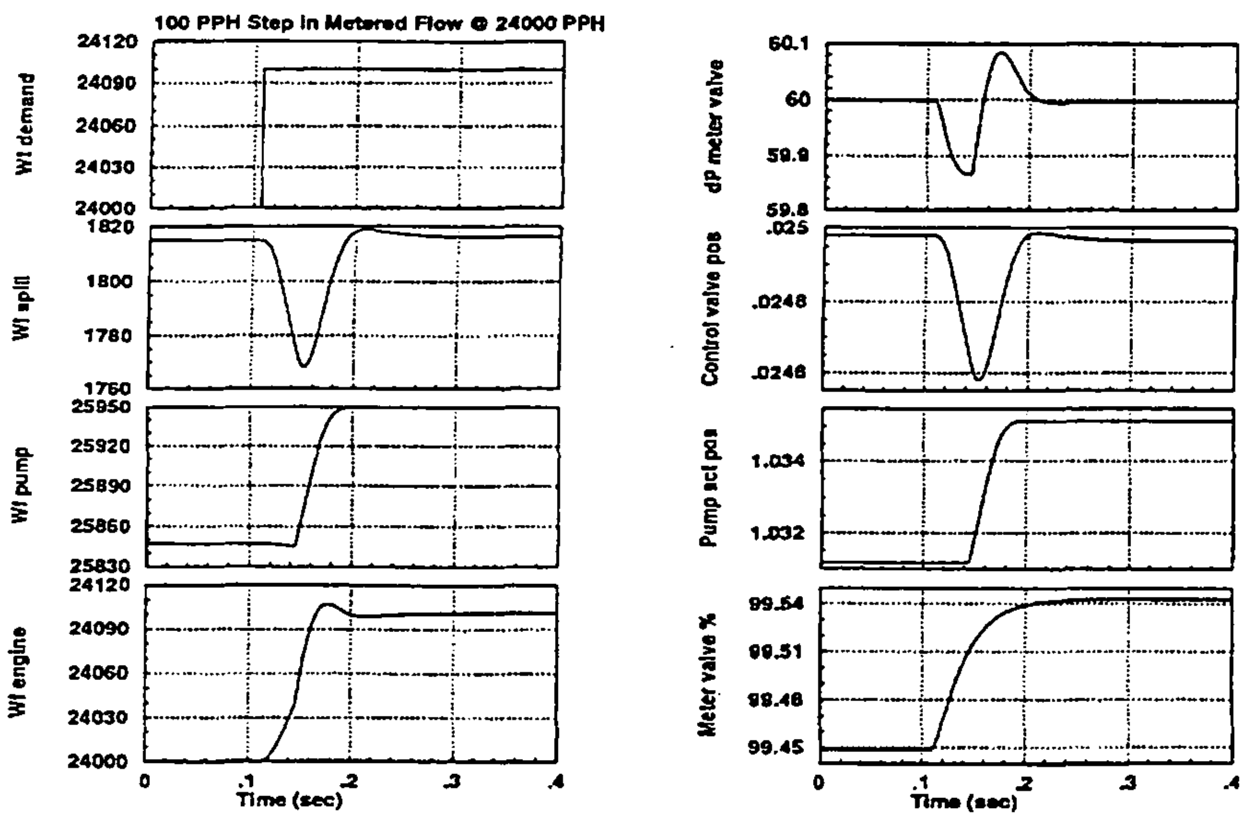

FIGURE 8. SIMULATION OF ΔP REGULATOR BASED FUEL CONTROL STEP RESPONSE AT HIGH WF

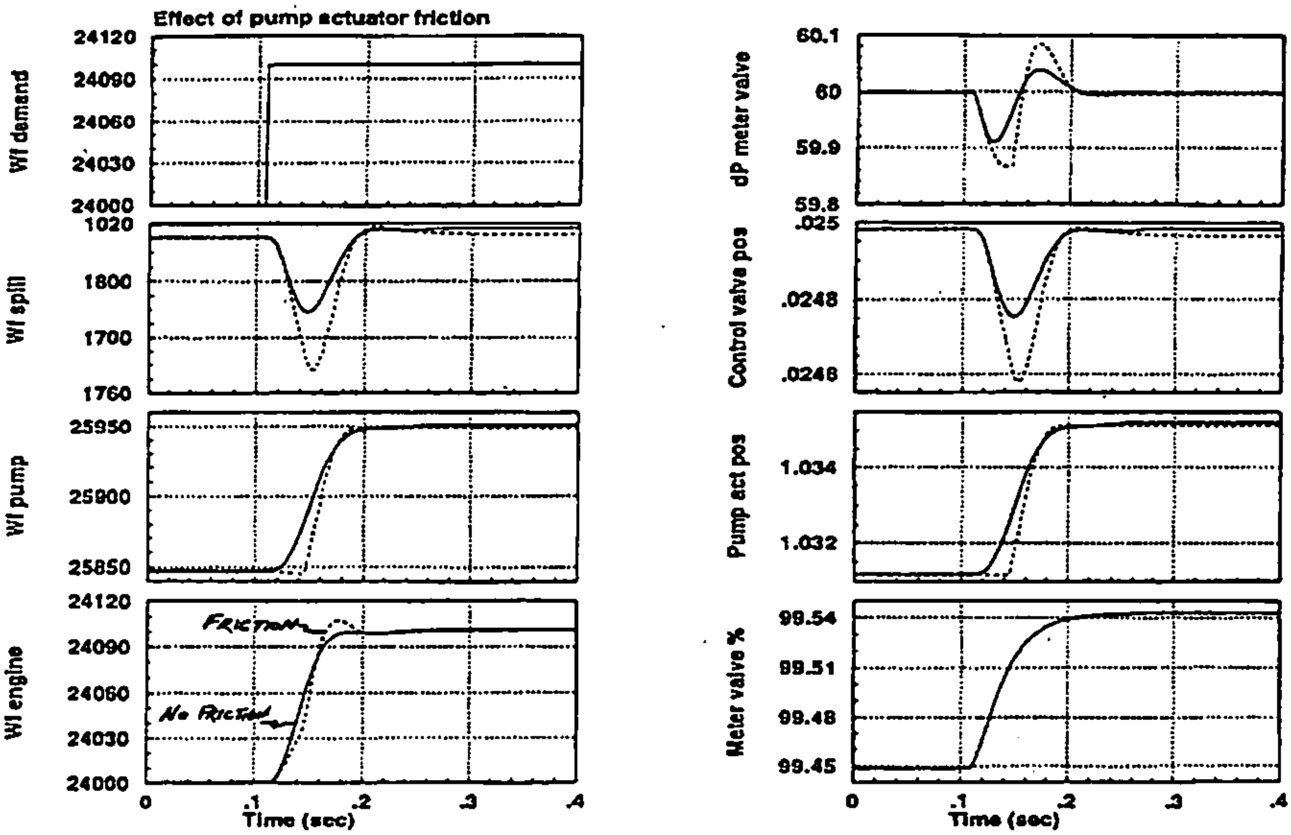

FIGURE 9. SIMULATION OF ΔP REG. BASED FUEL CONTROL, WITH AND WITHOUT PUMP ACTUATOR FRICTION

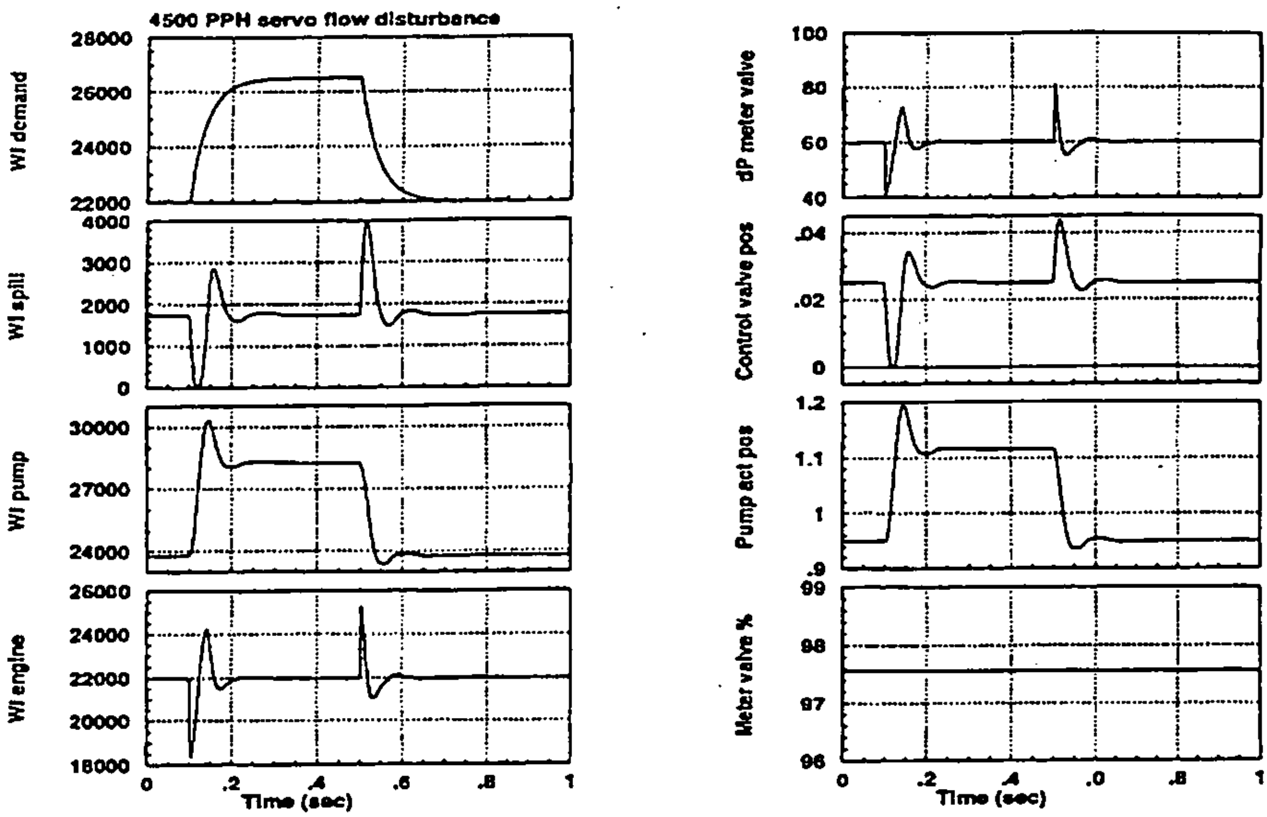

FIGURE 10. SIMULATION OF ΔP REGULATOR BASED FUEL CONTROL RESPONSE TO A 1000 PPH SERVO FLOW DISTURBANCE

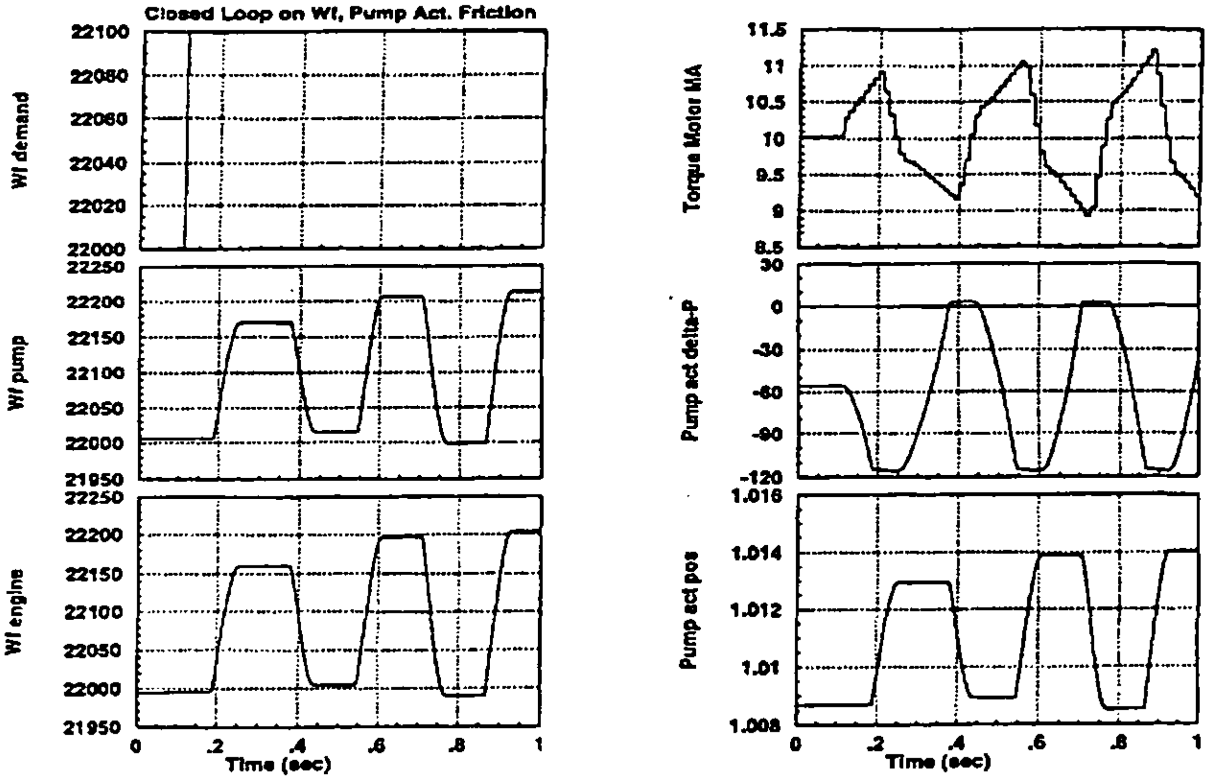

FIGURE 11. SIMULATION OF CLOSED LOOP ON WF APPROACH, WITH FRICTION

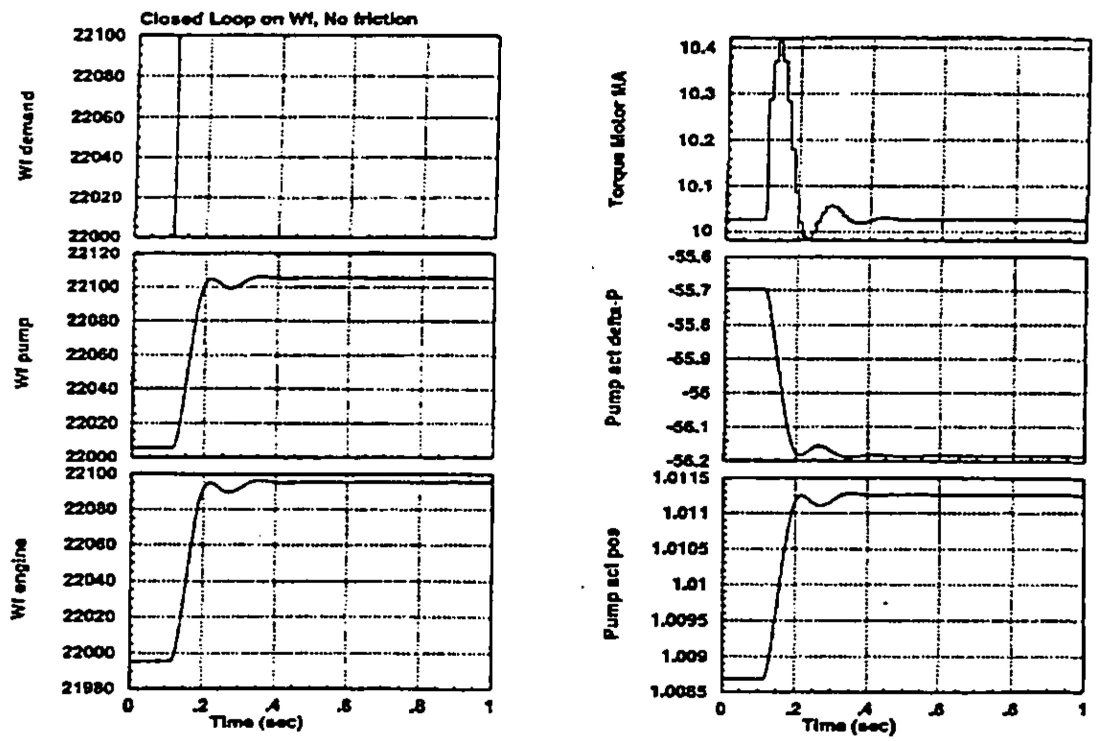

FIGURE 12. SIMULATION OF CLOSED LOOP ON WF APPROACH, WITHOUT FRICTION

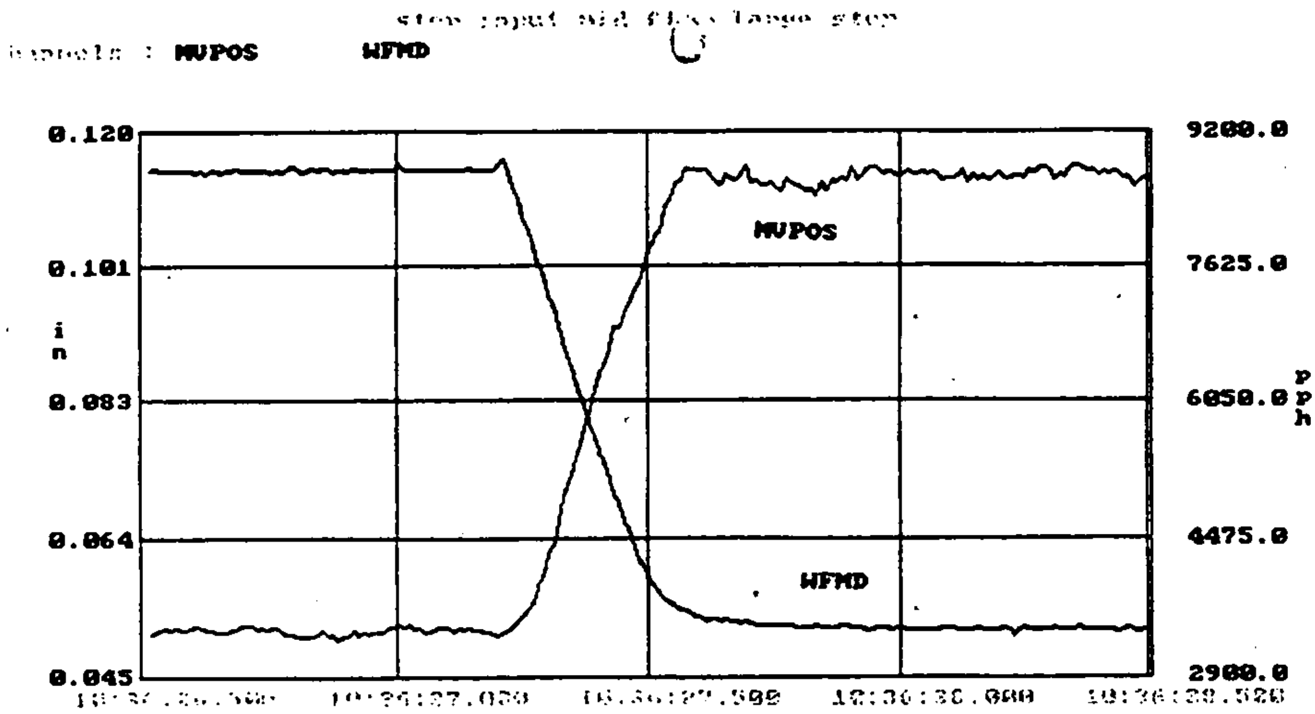

FIGURE 13. FUEL LAB TEST DATA OF ΔP REGULATOR BASED FUEL METERING UNIT