[redacted]

NASA CR-165296
DDA EDR 10696

[redacted]

Components for Digitally Controlled Aircraft Engines

August 1981

Prepared for
National Aeronautics and Space Administration
NASA-Lewis Research Center
Cleveland, Ohio 44135
Contract NAS 3-22046

Submitted By
Detroit Diesel Allison
Division of General Motors Corporation
P.O. Box 894
Indianapolis, Indiana 46206

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
CR-165296		

4. Title and Subtitle	5. Report Date
COMPONENTS FOR DIGITALLY CONTROLLED AIRCRAFT ENGINES	
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No.
Jay D. Meador	EDR 10696
	10. Work Unit No.
9. Performing Organization Name and Address	
Detroit Diesel Allison Division General Motors Corporation Box 894 Indianapolis, IN 46206	11. Contract or Grant No. NAS3-22046
	13. Type of Report and Period Covered Contractor Report
12. Sponsoring Agency Name and Address	14. Sponsoring Agency Code
National Aeronautics and Space Administration Lewis Research Center Cleveland, OH 44135	

15. Supplementary Notes

Project Manager, Robert Baumbick,
NASA Lewis Research Center, Cleveland, OH

16. Abstract

Studies were conducted to define control system components suitable for use in advanced technology full authority digital electronic control systems. These studies covered compressor geometry actuation concepts and fuel handling system concepts suitable for use in large high performance turbofan/turbojet engines. Eight conceptual system designs were formulated for the actuation of the compressor geometry. Six conceptual system designs were formulated for the engine fuel handling system. Assessment criteria and weighting factors were established and trade studies performed on their candidate systems to establish the relative merits of the various concepts.

Other studies were performed on fuel pumping and metering systems for small turboshaft engines. Seven conceptual designs were formulated, and trade studies performed. A simplified bypassing fuel metering scheme was selected and preliminary design defined for further consideration.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Fuel Control Actuation Systems Engine Control	

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified		

* For sale by the National Technical Information Service, Springfield, Virginia 22161

CONTENTS

1.0 INTRODUCTION

2.0 SUMMARY

3.0 CONTROL COMPONENT STUDIES
- 3.1 GMA400 Compressor Geometry Actuation Studies
- 3.2 GMA400 Fuel System
  - 3.2.1 Demand Fuel Pump Concepts
  - 3.2.2 Fuel Systems Descriptions
  - 3.2.3 Fuel System Trade Studies
- 3.3 Fuel Pumping and Metering for Small Turboshaft Engines
  - 3.3.1 System Requirements
  - 3.3.2 Fuel Pump Considerations
  - 3.3.3 Fuel Pumping and Metering
  - 3.3.4 Fuel System Trade Studies

4.0 FUEL METERING SYSTEM PRELIMINARY DESIGN
- 4.1 Baseline System Description
- 4.2 Bypass System Design Features
- 4.3 Preliminary Design Description
- 4.4 Conclusions

5.0 CONCLUSIONS AND RECOMMENDATIONS

6.0 APPENDIX A

7.0 APPENDIX B

LIST OF ILLUSTRATIONS

Figure	Title	Page
1	Two stage fuel system	12
2	Candidate fuel system schematics	17
3	GMA500 fuel control - baseline system	32
4	Torque motor operated hydraulic servoed spool type metering valve	34
5	Solenoid operatd hydraulic multiplied spool type metering valve	35
6	Variable speed hysteresis clutch metering gear pump	37
7	Variable speed D.C. motor controlled metering gear pump	39
8	Stepper motor drivey bypass valve	40
9	Stepper motor driven bypass valve with solenoid valve speed and temperature limiting	42
10	Bypass fuel metering system schematic	47
11	Fuel pump flow comparisons	49
12	Bypass fuel metering system functional schematic	53
13	Bypass fuel metering system preliminary design layout	54

LIST OF TABLES

Table	Title	Page
1	Actuator Trade Study Summary Chart	11
2	Pump Trade Study	16
3	GMA400 Fuel System Comparison Study	21
4	GMA400 Fuel System Comparison Study (Cost Factors)	23
5	GMA400 Fuel System Comparison Study (Reliability and Maintainability Factors)	24
6	GMA400 Fuel System Comparison Study (Emergency Operation Factor)	25
7	Fuel Metering and Pumping Concepts for Small Turboshaft Engines	30
8	Fuel Systems Assessment Criteria	41
9	Fuel Handling Systems Trade-Off	43

LIST OF SYMBOLS AND ABBREVIATIONS

EHSV	Electrohydraulic servo valve
GPM	Gallon per minute
H.P. or h.p.	Horsepower
LVDT	Linear variable differential transformer
M.V.	Metering valve
Ng, N_H	Gas generator speed
N_M	Mach number
N_P	Power turbine speed
$\frac{d\ Ng}{dt}$	Gas generator acceleration
P_{CD}, CDP	Compressor discharge pressure
ΔP_n	Fuel nozzle pressure drop
PLA	Power lever angle
pph	Pounds per hour
psi	Pounds per square inch
RPM	Revolutions per minute
S.L.	Sea level
S.O.	Shut off valve
TBO	Time between overhaul
$T_{4.5}$	Gas stream interstage turbine temperature
V/L	Vapor to liquid ratio
W_f	Fuel flow, mass
#/h	Pounds per hour

1.0 INTRODUCTION

The objective of this program was to define new, innovative control components for use in a digital control system on an advanced aircraft engine which improve the reliability and maintainability of the propulsion control system. Definition of these components included conceptual design of the new configurations, trade studies of comparative approaches, and preliminary design effort on the selected component. The initial study effort was directed toward the definition of suitable control components for the Detroit Diesel Allison GMA400 variable geometry engine which utilizes a full authority digital control system. This work covered component studies on fuel system components and on variable geometry actuation concepts. Following these study efforts, the program was changed to perform design studies and preliminary design on advanced technology fuel pumping and metering system concepts for future rotorcraft engines. Subsequent efforts were devoted to conceptual designs and trades studies to define reduced cost, highly reliable and maintainable fuel systems for small turboshaft engines such as the DDA GMA500 and 250-C30 engines. Other current development programs at DDA are addressing the digital controller and advanced sensors plus the definition of control logic for this class of engines. Thus, this effort becomes an integral part of an advanced technology digital control system program for future rotorcraft engines.

2.0 SUMMARY

Studies were conducted for defining advanced technology control component concepts suitable for use with full authority digital electronic control systems for future aircraft engines. This effort was initially directed at the DDA GMA400 Variable Cycle Engine (VCE) requirements for the compressor variable geometry actuation concepts and for the fuel system concepts. Eight conceptual system designs were formulated for the actuation of the compressor vanes and comparative trade studies performed to obtain a relative assessment of the various system approaches. The actuation system concepts considered were:

- o Hydraulic cylinder actuator (oil)
- o Hydraulic motor operated screw jacks
- o Air motor operated planocentric actuators
- o Air motor operated screw jacks
- o Electric motor operated planocentric actuators
- o Electric motor operated screw jacks
- o Hysteresis clutch operated screw jacks
- o Fuel cylinder actuators

Each system was rated with respect to the following criteria:

- o Power consumption
- o Weight
- o Cost
- o Maintainability
- o Reliability
- o Vulnerability
- o Failure mode

The study indicated that a specific system selection is not clear-cut; but instead is dependent upon which criteria are most important for specific

engine applications and other engine system decisions. Fuel cylinder actuators are a very good approach if the resultant fuel temperature rise is acceptable and sufficient fuel pressure/flow capacity is available. Electric motor operated screw jack actuators show promise in applications with sufficient electrical power compatible with other engine requirements.

For the GMA400 engine fuel system, six candidate fuel system designs were formulated, assessment criteria established and trade studies performed to establish a basis of overall system design. The designs considered generally utilized a throttled discharge centrifugal pump with an integral retracting vane starting pump to provide variable delivery pump flow for minimum temprature rise. The six system concepts incorporated various arrangements for metering and staging the fuel flow for the primary and secondary nozzle systems of the combustion system. The fuel system assessment factors were defined as:

- o Weight
- o Temperature rise
- o Cost
- o Reliability
- o Time between overhaul
- o Maintainability
- o Performance
- o Emergency operation

The trade study did not indicate one system design to be best. Instead it identified the effects of the relative weighting of assessment factors upon the selection of the system. The system selection depends upon the relative importance of the various factors in meeting specific objectives of the engine application. However, the trade studies indicate the discharge throttled centrifugal pump with an integral retracting vane starting pump to be a good candidate for an advanced technology aircraft engine of the GMA400 type.

The study program was redirected to conduct trade studies on various approaches of fuel pumping and metering systems suitable for turboshaft engines for rotorcraft powerplant applications up to 1000 hp. Seven conceptual system designs were formulated, assessment criteria established and trade studies performed to define a system concept suitable for use in a full authority digital electronic control system on advanced technology engines.

The candidate systems studies were:

- o GMA500 fuel system - baseline
- o Torquemotor operated spool metering valve
- o Proportional solenoid operated spool metering valve
- o Hystersis clutch metering gear pump
- o Controlled D.C. motor driven metering gear pump
- o Stepper motor driven bypass metering valve
- o Stepper motor driven bypass metering valve with redundant solenoid valve

These studies resulted in the selection of a stepper motor driven bypassing fuel metering concept, along with a high pressure gear pump, to achieve a simplified fuel handling system for reduced cost and improved reliability.

A preliminary design layout was accomplished on this simplified fuel metering concept which is compatible with existing fuel pumps and electronic controllers. Design analysis and tests were performed to verify basic metering valve functional capability in the bypassing mode of operation.

This simplified system design shows great promise of improved reliability and reduced cost as compared to existing fuel handling systems for small turboshaft engines used on rotorcraft.

Follow on development work is recommended to further explore the performance capabilities of the simplified bypassing fuel metering concept. This work should include the detail design of the metering system including control mode definitions for total closed loop operation over the entire range of engine operation. The system should then be fabricated and software developed to implement the control modes. System evaluations should be performed on closed loop simulated engine/control bench tests.

3.0 CONTROL COMPONENT STUDIES

The initial studies undertaken to identify innovative control components for use with advanced digital control systems were directed toward application on the DDA GMA400 engine. This engine incorporates fully variable geometry in all stages of the axial compressor, variable turbine geometry and exhaust nozzle area control as well as the fuel pumping and metering system control.

A survey was made of these control components requirements of the DDA GMA400 variable geometry engine. Assessments were made of the control requirements for the following functions of the engine: Primary engine fuel pumping and metering system, compressor variable geometry actuation, H.P. turbine vane actuation, exhaust nozzle area control, and modulation of turbine blade cooling. From these assessments, it was concluded that the most productive areas of study involved the fuel pumping and metering and the compressor variable geometry actuation system. Other advanced technology development programs at DDA have addressed the H.P. turbine vane actuation system for this class of engine. The requirements for exhaust nozzle area control and turbine blade cooling modulation were not adequately defined to enable a meaningful system design study.

Following these preliminary assessments, the effort was directed toward the fuel pumping and metering requirements for advanced small turboshaft engines for rotorcraft applications. The primary objective of this work was to identify fuel pumping and metering system concepts having potential for improving the reliability and maintainability and reducing the cost of the system as compared to existing technology components used in full authority digital electronic control systems.

3.1 Compressor Variable Geometry Actuation Studies

The GMA400 axial compressor incorporates variable geometry on all six stages of vanes for flexible aerodynamic control. The vanes at each stage of the compressor are interconnected to a synchronization ring which surrounds the engine at that location. The synchronization rings are interconnected to a camplate on either side of the compressor which imparts a few degrees of rotation to the compressor vanes via tracked cam slots as the camplates are given a fore and aft movement by the actuation system. The actuation system requirements were established as a maximum axial load on each camplate of 1500 lbs. This load occurs at a sea level Mach number of 1.2 and a compressor discharge pressure equal to 370 psia. Maximum travel of the camplates required to move the vanes through their full travel at slew of one inch per second for a total travel of two inches. The system must be capable of position modulation within the full travel range. Vane position control will be closed loop through the engine as a function of corrected engine speed as calculated by the electronic controller. Accuracy of position will be dependent upon the accuracy of the position transducer system selected (resolver or LVDT).

Eight actuation system conceptual designs were formulated and defined in sufficient detail to enable assessments for trade studies. The systems studied are shown schematically in Table 1 and are described as follows:

1. Hydraulic cylinder actuators - This system is a 3000 psi oil system supplied by an engine driven hydraulic pump using engine lube oil from a common supply. An electro hydraulic servo valve (EHSV) controls the oil pump flow rate to the actuators as a function of an electrical signal from the engine electronic control. An engine accessories drive pad is required for the hydraulic pump and a larger capacity fuel-oil heat exchanger.

2. Hydraulic motor operated screw jacks - This system uses a hydraulic motor to drive a screw jack attached to each camplate. Mechanical rotation is transmitted from the motor to each screw jack via flexible drive cables. A synchronizing cable between the screw jacks provides redundancy in case any one of the drive cables fails. The motor operates on engine fuel (JP-4, JP-5, etc.) which is supplied by the engine fuel pump and metered by an EHSV which is controlled by a signal from the engine electronic control.

3. Air motor operated planocentric actuator - This system uses an air motor to drive a planocentric actuator (high ratio gear reduction device) attached to each camplate. Compressor discharge air, metered by an air servo valve, is used to drive the motor. Motor torque is transmitted to each of the planocentric actuators via a flexible drive cable. A synchronizing cable between the actuators is provided for redundancy.

4. Air motor operated screw jack actuators - This system is the same as Configuration 3 except that screw jacks are used instead of planocentric actuators.

5. Electric motor operated planocentric actuators - This system uses two D.C. electric motors with samarium cobalt magnets to drive two planocentric actuators, each one attached to a camplate. Torque is transmitted from the motor to the primary actuator and between the two actuators via flexible drive cables. The samarium cobalt magnets have eight to ten times the magnetic pull of Alnico magnets of the same size. This also contributes to a lower inertia of the motor rotor and results in faster response during speed changes. Motor speed is controlled from the engine electronic control output driver section.

6. Electric motor operated screw jacks - This system is the same as Configuration 5 except that screw jacks are used instead of planocentric actuators.

7. Hysteresis clutch operated screw jack actuators - This sytem uses hysteresis clutches to drive screw jacks attached to the two camplates. The hysteresis clutch is composed of an input and an output rotor. The input rotor is driven off the engine accessories gear case and the output rotor is connected to the screw jack actuator via a flexible drive cable. An electrical current is used to energize a field coil which results in the output shaft coupling to some degree with the rotating input shaft. Modulation of actuator RPM and camplate movement

is provided by varying the level of electrical current from the driver output section of the engine electronic control. The electrical power required to couple the input and output rotors is a fraction of the power required to drive the load which is received as a shaft horsepower from the engine. An interconnecting drive cable between the two screw jacks is necessary for rotation in either direction.

8. Fuel cylinder actuators - This system is similar to Configuration 1 except that engine fuel pump flow is used to operate the actuators. Since fuel system pressure is lower than hydraulic system pressure in Configuration 1 actuator diameters are larger. The engine fuel pump flow rate must be increased by the amount required for the actuators. However, an extra pump drive pad is not required and the engine fuel/oil heat exchanger size does not have to be increased.

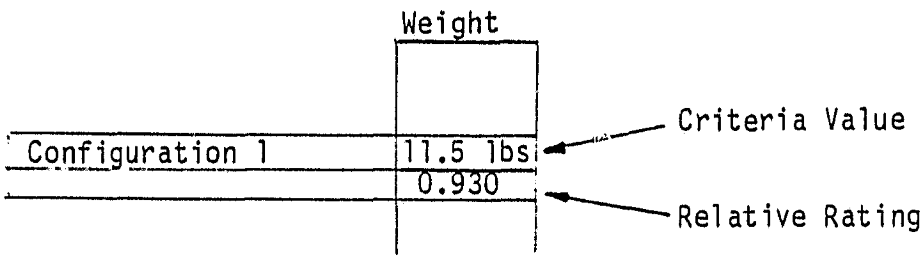
Appendix A contains design considerations for these eight systems along with discussion on the assessment criteria.

Compressor Geometry Actuation System Trade Studies

Each system configuration was rated with respect to the following criteria:

Engine cycle effects	Reliability
Weight	Vulnerability
Cost	Failure Mode
Maintainability	Dynamic Performance

The methods used for determining the specific criteria values for the comparative evaluation are described in Appendix A of this report. The normalizing technique for establishing the relative rating in each criteria category was based upon assigning a factor of unity (1.0) to the configuration having the best criteria value, (i.e. minimum weight, minimum cost, maximum reliability, best failure mode, etc). The relative ratings for the other system configurations were then derived as a ratio of each individual criteria value to the criterial value for the best configuration. Thus the relative evaluations are based upon "highest wins" for the composite score. For example, (refering to Table 1 Trade Study Summary Chart) the relative ratings for the criteria of weight are derived as follows. As seen in Table 1, configuration 8 is the lightest at 10.7 lbs. Configuration 1 at 11.5 lbs has a lower relative rating, determined to be...10.7/11.5 = 0.93. In Table 1 the specific criteria value (weight in this example) is presented, with the computed Relative Rating presented immediately below.



The ratings in all categories are tabulated in Table 1, individual scores above, relative scores below, and total scores are shown for the eight configurations. On the basis of "highest wins" the best overall actuation system for the GMA400 compressor variable geometry is the electric motor operated screw jack, Configuration 6, closely followed by the fuel pressure cylinder actuators, Configuration 8. This is based upon all factors being weighted equally. In an actual installation, certain factors such as weight, failure mode (there are wide variations in these), etc., may be more important and by weighting these more than others, a differnt conclusion may be reached.

3.2 GMA400 Fuel System Studies

The GMA200 and GMA400 series of demonstrator variable cycle engines employ staged combustion fuel systems for greater burner efficiency. The fuel system must supply fuel to the primary nozzles initially until a specified fuel/air ratio is reached. As the fuel requirements increase beyond this point, the specified fuel/air ratio is maintained in the primary nozzles while the excess fuel is directed to secondary nozzles. Once the secondary nozzles reach the same fuel/air ratio, the additional fuel is divided between the primary and

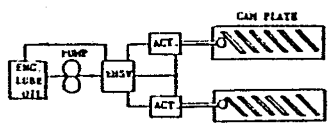
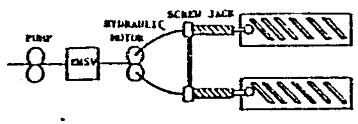
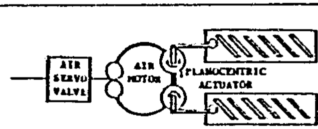
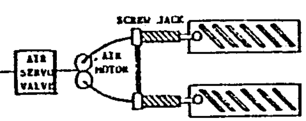
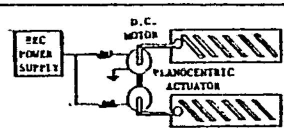
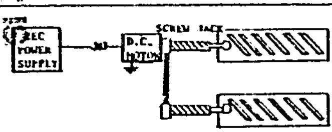
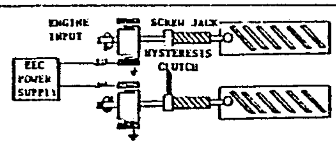
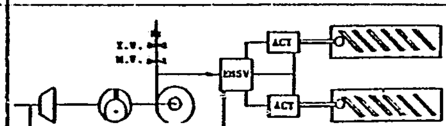
Configuration	Factor	Type of Power	Power Source: Additional System Requirements	Power Source: Engine Cycle Effects	Weight	Cost	Maintainability	Reliability	Vulnerability	Failure Mode	Dynamics	Total Score
1		Hydro Mechanical	Large Fuel-oil Heat Exchanger	0.893 Shaft H.P.	11.5#	12.1K	0.617	0.815	–	0.342		
				0.705	0.930	0.599	0.776	0.951	0.6	0.600	–	5.16
2		Hydro Mechanical	Large Fuel-oil Heat Exchanger	0.71 Shaft H.P.	11.2#	8.85K	0.571	0.689	–	0.342		
				0.887	0.955	0.819	0.718	0.816	0.6	0.600	–	5.39
3		Pneumatic	–	3.0 Air H.P.	58.0	16.3K	0.652	0.615	–	0.342		
				0.21	.184	0.445	0.820	0.722	0.9	0.600	–	3.04
4		Pneumatic	–	0.82 Air H.P.	15.7	12.3K	0.652	0.734	–	0.342		
				0.768	.681	0.589	0.820	0.856	0.9	0.600	–	5.22
5		Electro-Mechanical	Extra Capacity Alternator	.958 KW (1.28 HP)	53	11.25K	0.795	0.534	–	0.216		
				0.492	.20	0.644	1.0	0.623	1.0	0.379	–	4.34
6		Electro-Mechanical	Extra Capacity Alternator	0.47 KW (0.63 HP)	13.5#	7.25K	0.795	0.634	–	0.360		
				1.0	0.781	1.0	1.0	0.738	1.0	0.632	–	6.15
7		Electro-Mechanical	Engine Drive Pad	0.758 Shaft H.P. +16W	26.9#	7.65K	0.715	0.628	–	0.480		
				0.829	0.397	0.948K	0.899	0.733	1.0	0.842	–	5.65
8		Hydro Mechanical	Extra Capacity Fuel Pump	0.891 Shaft HP	10.7	7.6K	0.680	0.857	–	0.570		
				0.707	1.0	0.954	0.855	1.0	0.6	1.0	–	6.12

Table 1. Actuator Trade Study Summary Chart

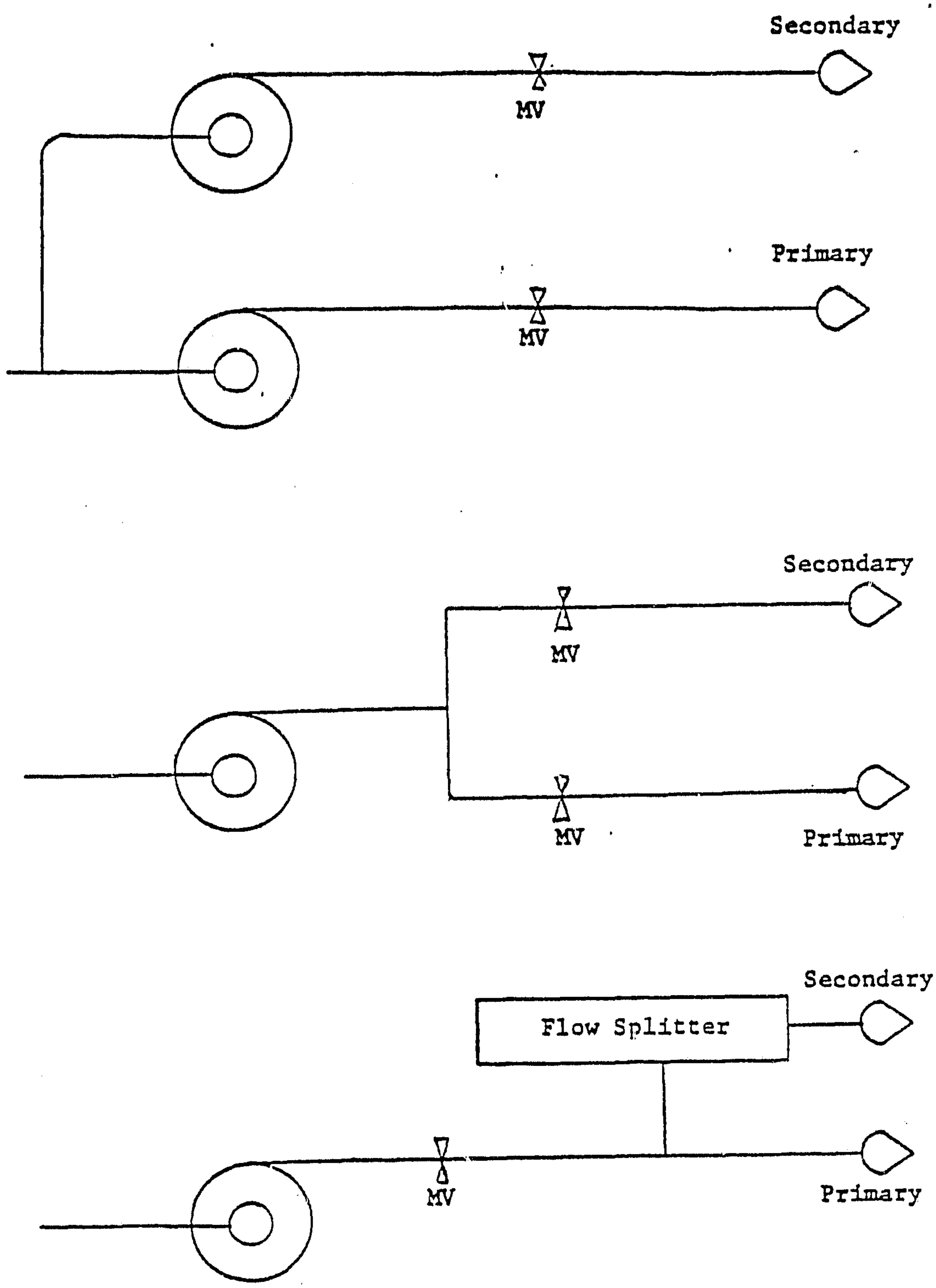


Figure 1. Two Stage Fuel Systems

secondary nozzles to hold the same fuel/air ratio for both systems. Control modes implemented in the digital controller will determine the flow staging between the primary and secondary nozzles to satisfy the above requirements. An optimum fuel handling arrangement must be defined for this class of engine.

The three different arrangements, shown in Figure 1, were considered for the fuel system. The first system consists of separate fuel systems for the primary and secondary nozzles. The second system has a single pump and separate metering valves for the primary and secondary nozzles. The third system is a variation on the second in that one fuel metering valve controls the total fuel flow from the single pump and a flow splitter controls the flow to the secondary nozzles. Preliminary studies were made of these basic system concepts and it was concluded that the primary system consideration was the selection of an optimum variable delivery fuel pump to achieve minimum fuel temperature rise. Following the selection of the type of pump, the overall fuel handling system study was then undertaken.

3.2.1 Demand Fuel Pump Concepts

Five fuel pump design concepts that provide variable fuel flow based upon engine requirements were analyzed. These are

1) Discharge flow throtled centrifugal pump with retracting vane starting pump.

2) Inlet flow throttled centrifugal pump, otherwise known as a vapor core pump, with retracting vane starting pump.

3) Positive displacement gear pump with variable speed drive which changes speed to match engine fuel requirements.

4) Positive displacement gear pump with DC electric motor drive which changes speed to match engine fuel requirements.

5) Variable displacement vane pump.

All of these pumps are engine driven with the exception of 4). A pressure drop type fuel metering control used in conjunction with each pump was assumed and was included in the weight figure. By this means, engine fuel demand which originates in the engine electronic digital control, is converted into an electrical signal, either analog or digital, which enters the electrical interface of the metering control and results in a change in metering valve area. This change in area upsets the constant pressure drop across the valve. By means peculiar to each pump, i.e., by discharge throttling, inlet throttling, speed change, or swept volume change, the output flow is increased or decreased until the referenced pressure drop is restored across the metering valve. The response of the metering valve was assumed to be optimum for each pump. The mechanism used by each pump to vary the output flow was evaluated in each case as part of its performance and reliability rating. All five configurations were evaluated with respect to:

	Weighting Factor
Weight	1
Temperature Rise °F	2
Cost/1500 units	1
Reliability - technical risk, frequency of repair	2
Time Between Overhaul	1
Maintainability - Removal time, ease of repair	2
Performance - Response	2
Installation - Volume	1

(The weighting factors shown were selected based upon certain engine/aircraft missions for the GMA400 class engine. The relative weighting of the various assessment criteria will depend upon the final engine design requirements and will therefore effect the results of the trade study.

Pump configurations (1) and (2) were previously evaluated as candidates for the DDA GMA200 Joint Technology Demonstrator engine. At that time configuration 1) was selected because of lower fuel temperature rise, and lower technology risk. The later reason was due to lack of demonstrated main engine experience with the inlet throttle concept of the vapor core pump.

Weight, volume, cost, temperature rise and detailed layout features were available from this earlier study and in the case of the vapor core pump (2) were extrapolated to meet the GMA400 requirements.

Weight and temperature rise figures for configurations (3), (4), and (5) were obtained from vendor supplied information of similar units. Where numerical information was not available, such as for reliability, TBO, maintainability etc., relative figures of merit were used based upon expert engineering judgement from qualified vendors from experience with similar components. The figure of merit is based upon unity as the highest score in each category.

A trade study was conducted on the five variable delivery pump configurations and the factors tabulated as shown in Table 2. On the basis of this evaluation, the discharge throttled centrifugal pump (Configuration 1) was selected for further evaluation in the various fuel system configurations.

3.2.2 Fuel Systems Descriptions

Six different candidate fuel systems were conceptually defined for comparative trade studies. These systems generally all incorporate the configuration (1) centrifugal pump described above. Consideration was also given to variable drive speed gear pump schemes for comparative purposes. The following is a short description of each fuel system configuration as depicted schematically in illustrations in Figure 2.

Configuration 1 - Composed of 1) a fuel pump which includes a fuel inlet inducer which keeps the main centrifugal element filled with fuel at all inlet pressure conditions, a retracting vane starting element which furnishes fuel from light-off to idle RPM, a centrifugal element which furnishes fuel from idle RPM to max. thrust conditions, 2) a metering valve control assembly which includes a pressure drop control assembly and throttle valve to match fuel pump output to engine total fuel requirements, and 3) a flow divider and drain valve assembly. The flow divider meters the amount of fuel to the main fuel nozzles (with the remainder of the total fuel flow directed to the pilot fuel

	Performance	Temp.Rise ΔT°F	Maintainability	Reliability	Cost	Weight	TBO	Installation	Total
Discharge Flow Throttle Centrifugal Pump	2.0	.68	2.0	2.0	1.0	.95	.75	1.0	10.38
Vapor Core Centrifugal	1.5	.68	2.0	1.5	.85	1.0	1.0	1.0	9.53
Positive Displacement Pump With Variable Speed Drive	1.4	2.0	1.6	1.0	.5	.47	.5	.6	8.07
Electric Motor Driven Positive Displacement Pump	1.2	2.0	1.6	1.0	.5	.20	.5	.5	7.5
Variable Displacement Vane Pump	1.5	1.5	1.0	1.0	.4	.59	.5	.8	7.29
Ideal Score	2.0	2.0	2.0	2.0	1.0	1.0	1.0	1.0	12.0

Table 2 – Pump Trade Study

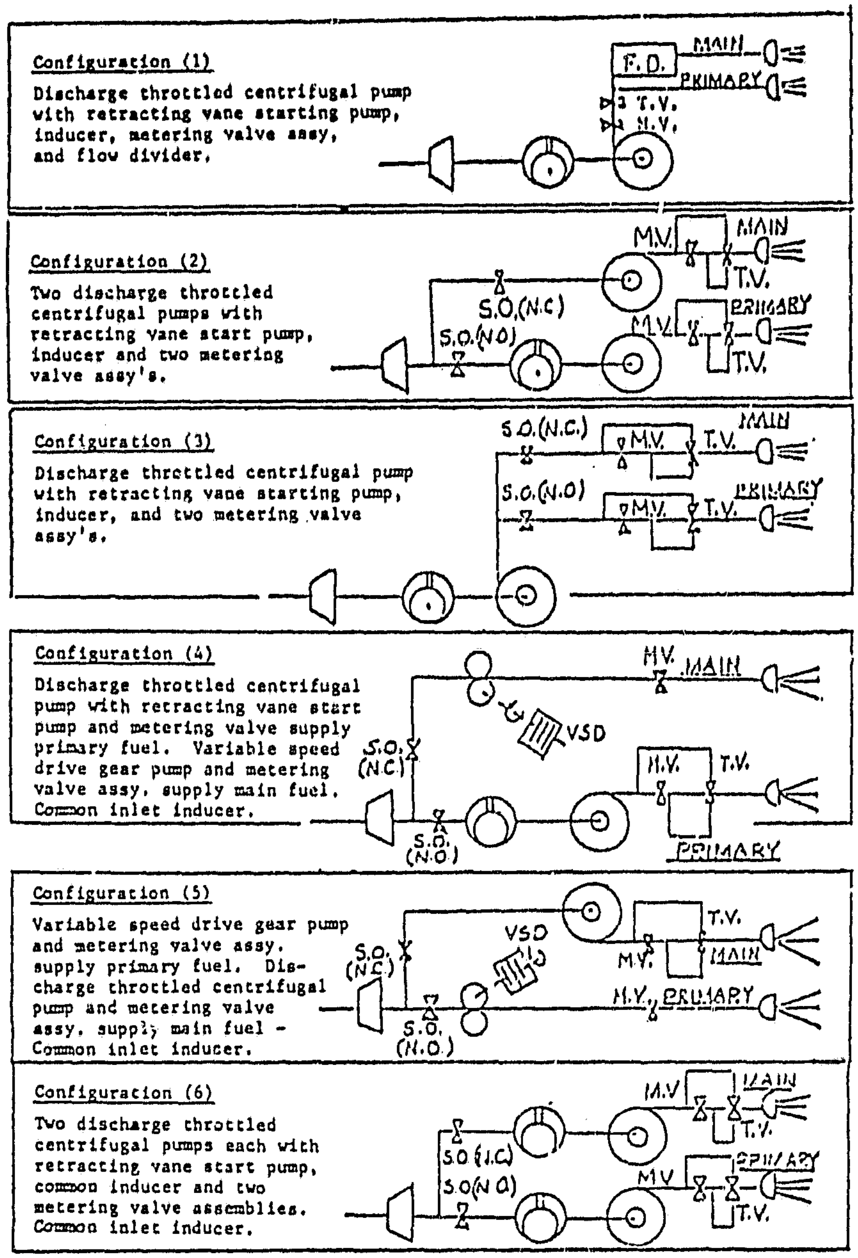


Figure 2. Candidate Fuel System Schematics

nozzles) to maintain the desired fuel-air ratio for the main and pilot nozzles. It also incorporates a shut-off valve which prevents fuel from the pump/control assembly from passing into the combustion chamber, collecting there and causing a hot start when the control metering valve is at the minimum flow setting. Its location near the fuel nozzles also minimizes manifold fill time during starts. When the shut-off valve is closed, the primary and main fuel nozzles are vented overboard via the nozzle manifolds and flow divider. The shut-off valve is activated open (manifold drains closed0 at light-off during a start and activated closed (manifold drains open) during shutdown.

Configuration 2 - Composed of 1) two centrifugal fuel pumps operating in parallel with a common inducer and a starting pump, 2) a separate metering valve control assembly for each pump, and 3) two shut-off valves which isolate the pumps from each other. The primary and main fuel nozzles are each supplied by a separate fuel pump and control assembly. The main fuel nozzle system is operated in conjunction with the primary system to achieve the desired total fuel flow while maintaining the required fuel air ratio at each set of nozzles. When the main system is not operating at low fuel demand conditions, its shut-off valve is closed. Likewise, if a failure should occur in the primary system, it can be isolated from the main system by closing its shut-off valve.

Configuration 3 - Composed of 1) a single pump including an inducer, starting element, and centrifugal element, 2) two metering valve control assemblies, and 3) two shut-off valves. The main system operates in conjunction with the primary system to achieve the desired total fuel flow while maintaining the proper fuel-air ratio at each nozzle. Its shut-off valve is activated open when the main fuel nozzles are flowing at the higher total fuel flows. The primary shut-off valve is used to isolate the primary metering valve control in case of failure.

Configuration 4 - Composed of 1) a primary system centrifugal pump and starting pump operating in parallel with a main system variable speed drive gear pump and a common inducer, 2) a separate metering valve control assembly for each pump, and 3) two shut-off valves which isolate the pumps from each other. The shut-off valves and metering valves are operated in the same manner as Configuration 2.

Configuration 5 - Composed of 1) a primary system variable speed drive gear pump operating in parallel with a main system centrifugal pump (no starting pump) and common inducer, 2) separate metering valves for each pump, and 3) two shut-off valves which isolate the pumps from each other.

Configuration 6 - Essentially the same as Configuration 2 except a retracting vane starting pump is added to the main fuel nozzles pumping and metering system. The configuration has complete pump/control back-up capability and means for making an air start in either primary or main systems with a nominal weight and cost increase over Configuration 2. This configuration evolved during the investigation of the "emergency operation" where it was observed that a slight modification to Configuration (2) would permit the aircraft to complete a mission after incurring any single fuel sysytem failure (excluding gear box failures). Configuration (2) has individual pumps, metering valves, and shut-off valves for the primary and main nozzles. However, only the primary nozzle system has the starting pump since the main nozzles are turned on when a substantial flow already exist with the primary nozzles. Also, the two pumps share a common inducer. By putting a retracting vane starting pump in each nozzle fuel handling system, a single failure cannot make both nozzle systems inoperative at any condition. The burner can operate on a single set of nozzles (either set) up to 50% of the maximum total fuel flow with an undefined loss in efficiency.

3.2.3 GMA400 Fuel System Trade Studies

The metering valves which determine the amount of fuel going to the engine are controlled by a electrical input from the engine digital electronic control. A constant pressure drop is maintained across the metering valves by a throttling valve in systems where there is a centrifugal pump. Where a variable speed drive gear pump is used as in Configurations 4 and 5, the pressure drop across the metering valve is maintained by controlling the speed of the pump with an electrical signal from the engine electronic control.

The pumps in all system configurations are engine driven and are part of an integrated package along with the metering valve control assemblies. The flow divider in Configuration 1 is located separately from the pump/control assembly. The metering valve in the flow divider is controlled by an electrical signal from the electronic control to satisfy the required main nozzle system fuel-air ratio.

Since fuel temperature rise due to pumping losses is one of the prime considerations in selecting a fuel system for high performance aircraft, this study was completed first. If system fuel temperature rise could not be brought into reasonable limits, the system was eliminated from further consideration. In this particular study the variable delivery feature of the five pump configurations, as compared to fixed delivery bypassing systems, automatically assured their consideration, from a temperature rise standpoint, for further overall trade studies. Some assumptions and design modifications were necessary in order to achieve this such as determining the reasonableness of engine operation and its duration at high fuel temperature conditions, and using a gear ratio change on the variable speed drive to reduce slip horsepower losses.

Table 3 shows those parameters which are considered essential in determining whether or not a system is suitable for service use. The reasoning and factual data upon which the ratings are based are described in the backup data which is included in Appendix B. Assumptions made are as follows:

Weight - Weight figures are based upon GMA200 and GMA400 proposals previously received from TRW Inc. for the pump and metering valve control assemblies. Estimates for the gear pump and variable speed drive are based also upon TRW information on similar units.

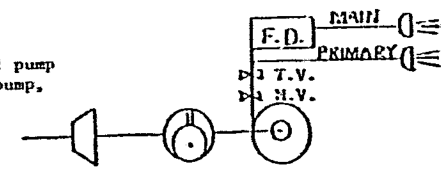
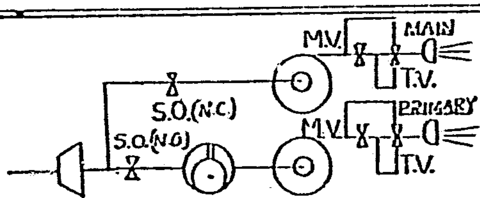
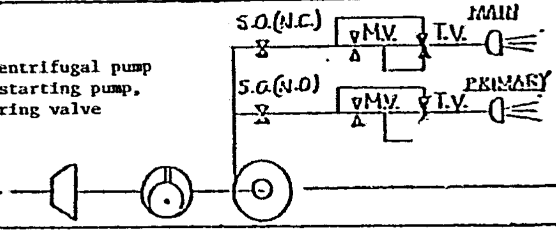
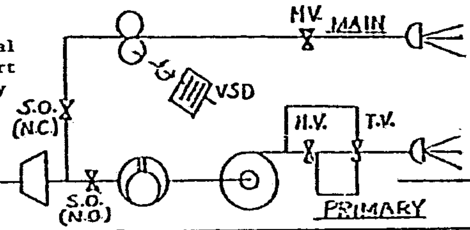
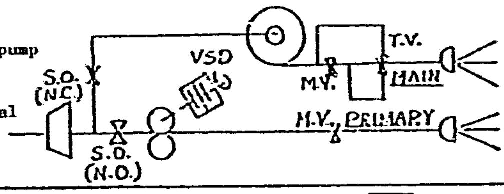
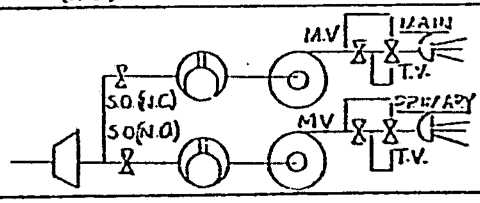
Configuration	Schematic		Weight	ΔT	Cost	Reliability	Maintainability	Performance	TBO	Emergency Operation	Total
Configuration (1) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, metering valve assy, and flow divider.		Raw	44#	65°F	1[illegible].5K	0.65	0.89	0.92	0.84	1.2	Highest Wins
		Factored	0.84	1.0	1.0	1.0	1.0	0.97	0.97	0.6	7.38
Configuration (2) Two discharge throttled centrifugal pumps with retracting vane start pump, inducer and two metering valve assy's.		Raw	37#	88°F	15.5K	0.59	0.82	0.95	0.85	1.8	—
		Factored	1.0	0.74	0.73	0.91	0.92	1.0	0.98	0.9	7.12
Configuration (3) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, and two metering valve assy's.		Raw	37#	65°F	15.5K	0.59	0.82	0.86	0.87	1.6	—
		Factored	1.0	1.0	.87	0.91	0.92	0.90	1.0	0.8	7.40
Configuration (4) Discharge throttled centrifugal pump with retracting vane start pump and metering valve supply primary fuel. Variable speed drive gear pump and metering valve assy. supply main fuel. Common inlet inducer.		Raw	46#	78°F	21.5K	0.54	0.76	0.81	0.79	2.0	—
		Factored	0.80	0.84	0.63	0.83	0.85	0.85	0.91	1.0	6.71
Configuration (5) Variable speed drive gear pump and metering valve assy. supply primary fuel. Discharge throttled centrifugal pump and metering valve assy. supply main fuel – Common inlet inducer.		Raw	46#	88°F	18.5K	0.54	0.76	0.86	0.78	1.8	—
		Factored	0.80	0.74	0.73	0.83	0.85	0.90	0.90	0.9	6.65
Configuration (6) Two discharge throttled centrifugal pumps each with retracting vane start pump, common inducer and two metering valve assemblies. Common inlet inducer.		Raw	42#	88°F	19.5K	0.57	0.80	0.95	0.85	2.0	—
		Factored	0.88	0.74	0.69	0.88	0.90	1.0	0.98	1.0	7.07

Table 3. GMA 400 Fuel System Comparison Study

Temperature Rise - Based upon estimates as described in Appendix B.

Cost - Costs are based upon 1978 dollars used in calculating the cost in GMA200 and GMA400 proposals (see Appendix B).

Reliability -	
Maintainability -	Figures of merit are used based upon engineering judgement from production and operational experience on similar components
Performance -	
TBO -	
Installation -	
Engineering Operation -	

The evaluation of system reliability as shown in Table 3 consider the possibility of failure when based upon such things as the number of parts, service experience on similar designs and design simplicity. Emergency operation capability considers that certain items will malfunction or fail and evaluates the probability of continued operation of the engine to complete a mission or get back to base at reduced power. Both views of reliability were included in the trade study. The emergency operation capability was not weighted as heavily as the other since built-in design realiability is considered more important.

The effects of weighting factors on the fuel handling system study results are summarized in Tables 3-6. The final results are sensitive to the weighting factors assigned to each assessment factor. If all factors are considered equally, the Configurations (1) and (3) are nearly equal as seen in Table 3. If the traditional factors of cost and weight are emphasized (Table 4) then Configuration (3) is still the best by a small margin over Configuration (1). If realiability, maintainability and time between overhaul are stressed as in Table 5, then Configuration (1) is then the best system. However, if more importance is given to the emergency capability, then Configuration (6) is the best (Table 6). It should be noted that the single pump configuration is the best in all cases except the last.

ORIGINAL PAGE IS
OF POOR QUALITY

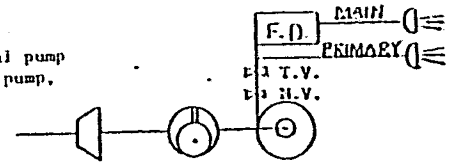
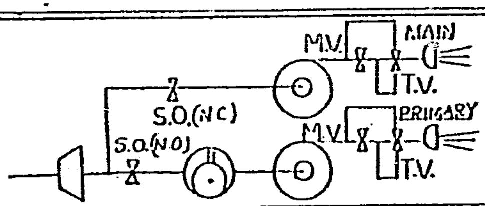
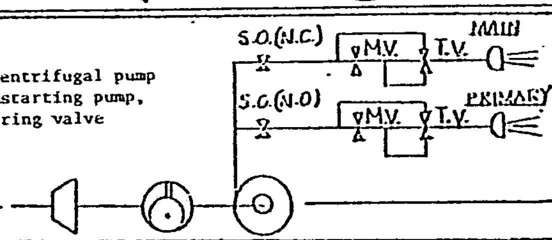
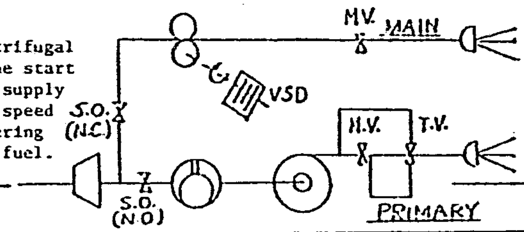
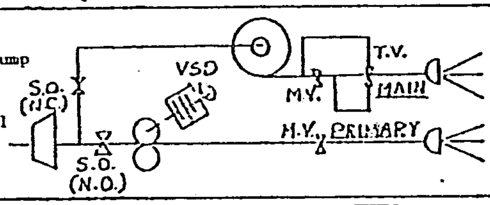
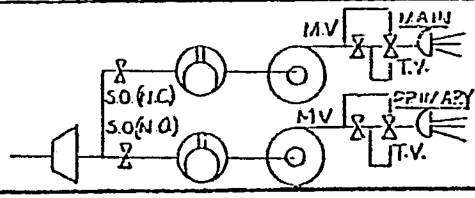
Configuration		Weight	ΔT	Cost	Reliability	Maintainability	Performance	TBO	Emergency Operation	Total
Configuration (1) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, metering valve assy. and flow divider. 	Raw	44#	65°F	13.5K	0.65	0.89	0.92	0.84	1.2	Highest Wins
	Factored	1.68	1.0	2.0	1.0	1.0	0.97	0.97	0.6	9.22
Configuration (2) Two discharge throttled centrifugal pumps with retracting vane start pump, inducer and two metering valve assy's. 	Raw	37#	88°F	15.5K	0.59	0.82	0.95	0.85	1.8	---
	Factored	2.0	0.74	1.46	0.91	0.92	1.0	0.98	0.9	8.91
Configuration (3) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, and two metering valve assy's. 	Raw	[illegible]	65°F	15.5K	0.59	0.82	0.86	0.87	1.6	---
	Factored	2.0	1.0	1.75	0.91	0.92	0.90	1.0	0.8	9.28
Configuration (4) Discharge throttled centrifugal pump with retracting vane start pump and metering valve supply primary fuel. Variable speed drive gear pump and metering valve assy. supply main fuel. Common inlet inducer. 	Raw	46#	78°F	21.5K	0.54	0.76	0.81	0.79	2.0	---
	Factored	1.60	0.84	1.26	0.83	0.85	0.85	0.91	1.0	8.14
Configuration (5) Variable speed drive gear pump and metering valve assy. supply primary fuel. Discharge throttled centrifugal pump and metering valve assy. supply main fuel - Common inlet inducer. 	Raw	46#	88°F	18.5K	0.54	0.76	0.86	0.78	1.8	---
	Factored	1.60	0.74	1.46	0.83	0.85	0.90	0.90	0.9	8.18
Configuration (6) Two discharge throttled centrifugal pumps each with retracting vane start pump, common inducer and two metering valve assemblies. Common inlet inducer. 	Raw	42#	88°F	19.5K	0.57	0.30	0.95	0.85	2.0	---
	Factored	1.76	0.74	1.38	0.88	0.90	1.0	0.98	1.0	8.64

Table 4. GMA 400 Fuel System Comparison Study

ORIGINAL PAGE IS
OF POOR QUALITY

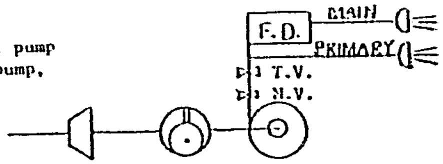
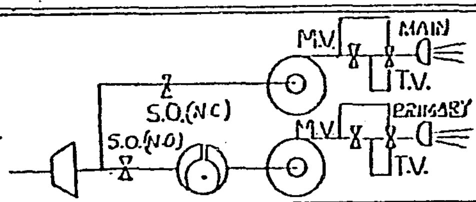
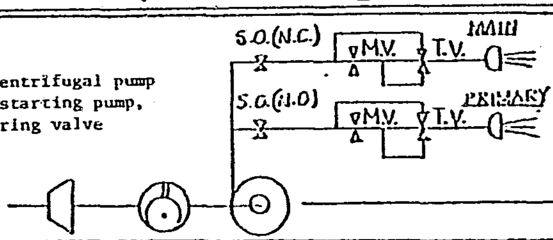
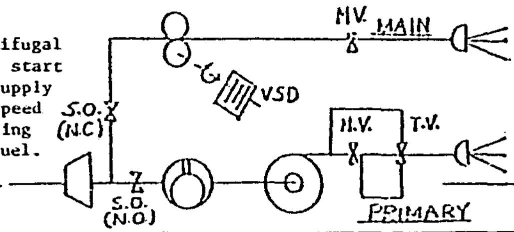
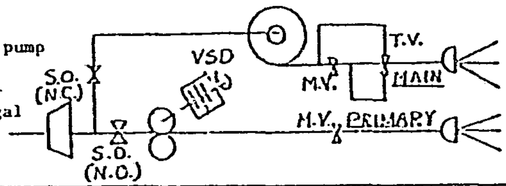
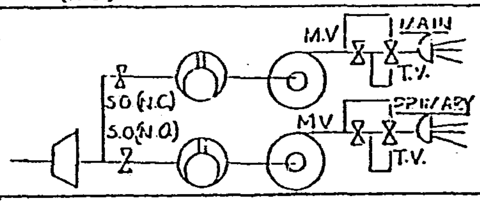
Configuration	Row	Weight	ΔT	Cost	Reli-ability	Maintain-ability	Per-formance	TBO	Emergency Operation	Total
Factor		1	1	1	2	2	1	2	1	
Configuration (1) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, metering valve assy, and flow divider. 	Raw	44#	65°F	13.5K	0.65	0.89	0.92	0.84	1.2	Highest Wins
	Factored	0.84	1.0	1.0	2.0	2.0	0.97	1.94	0.6	10.35
Configuration (2) Two discharge throttled centrifugal pumps with retracting vane start pump, inducer and two metering valve assy's. 	Raw	37#	88°F	15.5K	0.59	0.82	0.95	0.85	1.8	---
	Factored	1.0	0.74	0,73	1.82	1.84	1.0	1.96	0.9	9.39
Configuration (3) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, and two metering valve assy's. 	Raw	37#	65°F	15.5K	0.59	0.82	0.86	0.87	1.6	---
	Factored	1.0	1.0	.87	1.82	1.84	0.90	2.0	0.8	10.23
Configuration (4) Discharge throttled centrifugal pump with retracting vane start pump and metering valve supply primary fuel. Variable speed drive gear pump and metering valve assy. supply main fuel. Common inlet inducer. 	Raw	46#	78°F	21.5K	0.54	0.76	0.81	0.79	2.0	---
	Factored	0.80	0.84	0.63	1.66	1.71	0.85	1.92	1.0	9.41
Configuration (5) Variable speed drive gear pump and metering valve assy. supply primary fuel. Discharge throttled centrifugal pump and metering valve assy. supply main fuel - Common inlet inducer. 	Raw	46#	88°F	18.5K	0.54	0.76	0.86	0.78	1.8	---
	Factored	0.80	0.74	0.73	1.66	1.71	0.90	1.80	0.9	9.24
Configuration (6) Two discharge throttled centrifugal pumps each with retracting vane start pump, common inducer and two metering valve assemblies. Common inlet inducer. 	Raw	42#	88°F	19.5K	0.57	0.80	0.95	0.85	2.0	---
	Factored	0.88	0.74	0.69	1.75	1.80	1.0	1.96	1.0	9.82

Table 5. GMA 400 Fuel System Comparison Study

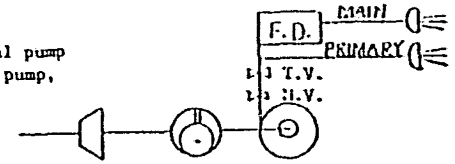
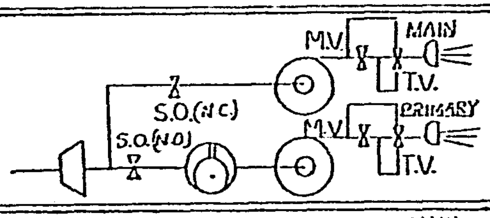
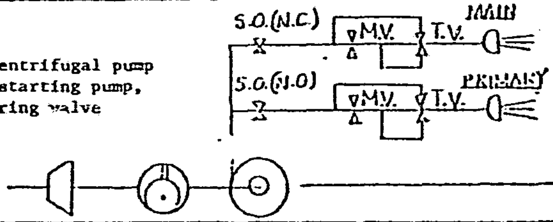
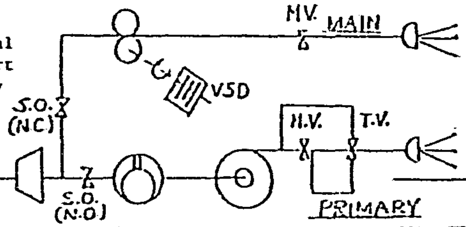
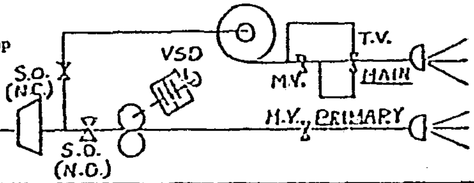
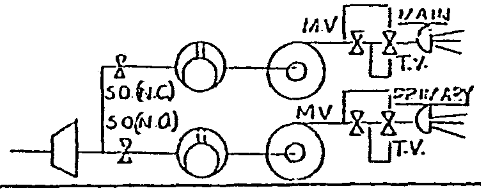
Configuration		Weight	ΔT	Cost	Reliability	Maintainability	Performance	TBO	Emergency Operation	Total
Configuration (1) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, metering valve assy, and flow divider. 	Raw	44#	65°F	13.5K	0.65	0.89	0.92	0.84	1.2	Highest Wins
	Factored	0.84	1.0	1.0	1.0	1.0	0.97	0.97	2.4	9.18
Configuration (2) Two discharge throttled centrifugal pumps with retracting vane start pump, inducer and two metering valve assy's. 	Raw	37#	88°F	15.5K	0.59	0.82	0.95	0.85	1.8	---
	Factored	1.0	0.74	0.73	0.91	0.92	1.0	0.98	3.6	9.88
Configuration (3) Discharge throttled centrifugal pump with retracting vane starting pump, inducer, and two metering valve assy's. 	Raw	37#	65°F	15.5K	0.59	0.82	0.86	0.87	1.6	---
	Factored	1.0	1.0	.87	0.91	0.92	0.90	1.0	3.2	9.80
Configuration (4) Discharge throttled centrifugal pump with retracting vane start pump and metering valve supply primary fuel. Variable speed drive gear pump and metering valve assy. supply main fuel. Common inlet inducer. 	Raw	46#	78°F	21.5K	0.54	0.76	0.81	0.79	2.0	---
	Factored	0.80	0.84	0.63	0.83	0.85	0.85	0.91	4.0	9.71
Configuration (5) Variable speed drive gear pump and metering valve assy. supply primary fuel. Discharge throttled centrifugal pump and metering valve assy. supply main fuel - Common inlet inducer. 	Raw	46#	88°F	18.5K	0.54	0.76	0.86	0.78	1.8	---
	Factored	0.80	0.74	0.73	0.83	0.85	0.90	0.90	3.6	9.35
Configuration (6) Two discharge throttled centrifugal pumps each with retracting vane start pump, common inducer and two metering valve assemblies. Common inlet inducer. 	Raw	42#	88°F	19.5K	0.57	0.80	0.95	0.85	2.0	---
	Factored	0.88	0.74	0.69	0.88	0.90	1.0	0.98	4.0	10.07

Table 6. GMA 400 Fuel System Comparison Study

Conclusions

These trade studies on fuel handling systems for the fully variable geometry GMA400 advanced technology engine identify, in a systematic manner, the effects of the assessment factors in the selection of system configuration.

The optimum system depends upon relative importance of the factors to meet the specific design objectives of the engine application. However, the study indicates that a discharge throttled centrifugal pump with a retracting vane starting element and sized to provide total engine fuel requirements is the preferable pump configuration.

3.3 Fuel Pumping and Metering Systems for Small Turboshaft Engines

The study program for the identification of components for digitally controlled engines was directed to consider various approaches of advanced technology fuel pumping and metering concepts suitable for turboshaft engines for helicopter powerplant applications in the general power class up to 1000 horsepower.

A program was formulated to accomplish these conceptual component designs and comparative trade studies through a joint effort with Chandler Evans Division of Colt Industries. Chandler Evans (CECO) has an extensive background in the design and development of advanced technology fuel pumping and metering components for this class of engines. This involvement of a control manufacturer provided realistic design background data to these design studies.

The DDA GMA500 engine fuel pump and metering system was selected as the baseline for design of the candidate advanced systems. Preliminary specifications were prepared as a basis of design for the fuel pumping and metering concepts and include the following:

3.3.1 System Requirements

Fuel Pump Performance Requirements:

o Fuel Flow: (Engine Requirement)	100% Ng = 600 pph, maximum power	
	10% Ng = 40 pph	
o Pressure:	100% Ng = 1000 psi	
	10% Ng = 100 psi	
o Drive Speed:	Optional	
o Fuel Type:	JP4, JP5, Commercial Grade Jet A and "hydrafine" fuels	

- o Fuel Contamination per MIL-STD-8593A with use of a barrier type filter element.
- o Starting Requirement: To 20,000 ft. altitude
- o Pump Inlet Conditions: The pump shall supply fuel at the required pressures for operation of the engine with the following conditions at the inlet:

  Fuel Temperature - From a minimum of -54°C (-65°F) (JP4) or the temperature corresponding to a fuel viscosity of 12 centistokes (JP5) to a maximum of 57°C (135°F).

  Fuel Pressure - From a minimum of 1 psi above true vapor pressure of the fuel to a maximum 50 psi with a vapor to liquid ratio (V/L) up to 0.45.

- o Dry Lift - A minimum of 60 inches dry lift shall be provided by the pump.

Fuel Metering Requirements

The fuel metering section shall provide for direct interface with a full authority electronic controller and have provisions for metering fuel flow from 20 pph starting to 600 pph max. flow. A mechanical input shall provide for power available functions of fuel on/off, ground idle and max. power setting. The metering system shall have dynamic characteristics compatible with engine/helicopter control requirements. The failure mode of the metering system shall be compatible with both single and twin engine helicopter operation to provide acceptable safety of flight without unusual pilot requirements. A redundant/back-up concept shall be incorporated to provide opeation in event of primary electronic controller system failure. Transfer to the redundant/back-up function shall be accomplished without undesirable fuel flow transient and shall be consistent with acceptable pilot work loads.

The fuel pump and metering functions shall be implemented in such a manner to achieve modular construction for ease of maintenance and be of minimum size and weight consistent with high reliability and low cost.

3.3.2 Fuel Pump Considerations

Because of the basic role the pumping element plays in the conceptual system design, assessments were made of possible pumping approaches to select one concept for all system considerations.

The positive displacement gear pump was selected as the primary pumping means in all of the systems being evaluated based on specification reuirements as well as reliability and cost considerations. A comparison of the cost and reliability ratings of the various pumps considered, using the centrifugal pump as the baseline, is tabulated below.

	Centrifugal	Gear	Vane
Cost	1	3	4
Reliability	1	.5	.3

This table reflects production and field experience which shows that the centrifugal pump is the most reliable and lowest cost pumping element. In addition, it is also less susceptible to contaminated fuel. However, it cannot meet the pumping requirements of a helicopter gas turbine over its full range of operation from startup to maximum power. Therefore, the gear pump was selected as the next best choice on the basis of cost and reliability.

The pressure rise of a centrifugal pump is proportional to speed squared. Thus, it is unfeasible to design a centrifugal pump to meet engine starting pressure (10% speed, 100 psi) requirements, and then run that design at 100% speed - discharge pressure would be too high. To get around this problem, a separate starting pump (gear or vane) is needed. The centrifugal pump can the be designed only for operation between idle and maximum speed. The resulting design would be complicated and expensive. However, by utilizing a variable speed drive, such as an electric motor or hysteresis clutch, the centrifugal pump speed/pressure characteristics could be utilized in an engine system. If the pump was located in the fuel tank, the V/L and dry lift problems would be eliminated to make the centrifugal pump a possible candidate. This arrangement of having the fuel pump remote from the engine would be unconventional and also result in high pressure fuel line from the tank to the engine causing a potential safety hazard.

Designing a pump to meet dry lift and V/L = .45 operation are additional considerations in selecting a pump design since with these capabilities, a tank mounted boost pump is not required. Sixty inches of dry lift and .45 V/L are typical requirements for a helicopter applications.

V/L operation to handle inlet line losses can be provided with a jet pump which also provides the gear pump charging pressure. However, the primary flow for the jet pump must be provided by the gear stage, and this can require oversizing of the gear pump and poor efficiency. A further consideration is filtering contaminated fuel. It is desirable to protect the gear pump from contaminated fuel, but locating the filter at the pump inlet requires a higher pressure jet pump to handle the filter pressure drop. This further aggravates efficiency. To deal with this problem, a centrifugal boost element can be added to the system as was done in the GMA500 Advanced Technology Demonstrator Engine pump to achieve a capability of 1.0 V/L operation. With this pumping stage, it may not be necessary to also have a jet pump to achieve a 0.45 V/L.

3.3.3 Fuel Pumping and Metering System Concepts

Various fuel metering and pumping schemes with possible electronic interfaces and back-up fuel control concepts are defined in Table 7. The schemes were devised for operation with digital electronic fuel controls on turboshaft powered rotorcraft. Although specific electronic interfaces have been identified for each metering scheme, it is expected that other combinations could be configured. For example, a stepper motor can be used to drive a rotary spool valve as well as a rotary flat plate valve.

Functional back-up concepts which provide the pilot means to operate the engine in the event of electronic control failure were also defined. With the objectives of low cost, simplicity and high reliability, simple manual fuel metering or manual W_f/p back-up systems were considered. However, Systems 4, 5 and 6 use a pump to meter fuel and, therefore, these schemes require an electronic back-up control since it is necessary for engine operation to keep the pump running.

Schematic implementations of seven candidate fuel pumping and metering concepts were developed. These schematics are shown in Figures 3 through 9 and are described as follows:

GMA500 Fuel Pumping and Metering - Baseline System

This system, shown in Figure 3, comprises a jet induced centrifugal pump, main stage gear pump, and an interstage barrier filter. The metering system includes a bypass head regulator, stepper motor operated flat plate metering valve and feedback potentiometer, and a pressurizing valve. The back-up system provides for manual fuel metering by connecting PLA to the metering valve via a solenoid. Deenergizing the solenoid allows the spring loaded plunger to push a slotted disc against the metering valve. Rotating PLA will result in the alignment of the slot with a pin sticking out of the metering valve. Thereby, PLA and the metering valve are engaged and indexed.

ORIGINAL PAGE IS
OF POOR QUALITY

Table 7. Fuel Metering and Pumping Concepts for Small Turboshaft Engines

Fuel Metering Device	Method for Actuating Metering Device	Fuel Flow Feedback Device	Metering ΔP Regulator	Backup Concept	Comments
1) Spool Valve	Torque motor operated hydraulic servo	Potentiometer, LVDT, or Resolver to sense metering valve position	Bypass type	Manual W_f or W_f/P using hydraulic P multiplier	Hydraulic multiplication can be implemented by transducing P_3 or P_1 to a proportional fuel pressure.
2) Poppet Valve	Solenoid operated hydraulic servo	Same as above	Bypass type	Same as above	A proportional or on-off solenoid can be used. By varying on and off time, proportional operation can be obtained with a two position solenoid.
3) Sliding Flat Plate Valve	Stepper motor	Same as above	Bypass Type	Manual W_f or W_f/P using mechanical P multiplier	Primary advantage of stepper motor is that it will fail fixed.
4) Positive Displacement Pump	DC motor or hysteresis clutch	Pump speed	Not required with closed loop control mode	Independent electronic analog control	The effect of a failure of DC motor or clutch is critical in this system.
5) Variable Displacement Pump	Torque motor hydraulic servo	Pump speed	Same as above	Same as above	Variable displacement pumps are expensive.
6) Variable Speed Centrifugal Pump	DC motor	Not required-operation based on pump map stored in computer memory	Same as above	Same as above	DC motor may be too large because of low pump efficiency.

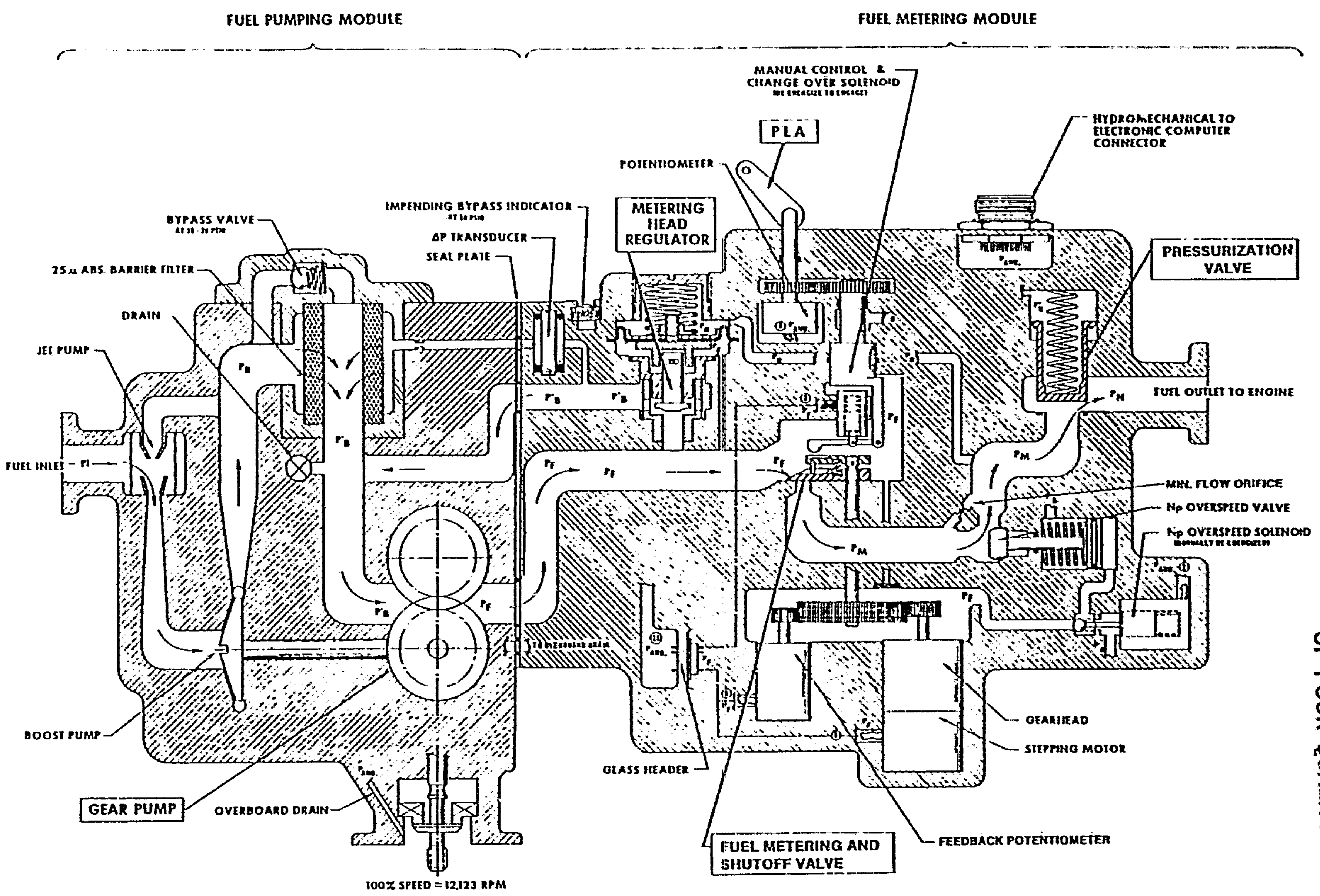


Figure 3 GMA 500 FUEL CONTROL - BASELINE SYSTEM

The stepper motor is designed to fail fixed. The possibility of a runaway is remote since it would require a failure resulting in the precise sequencing of the motor's four phases which is highly unlikely.

The main metering valve is also used as the shutoff valve. This feature eliminates the need for a pressure relief valve because the metering head regulator bypasses pump output flow when fuel is shut off.

Torque Motor Operated Metering Valve

This metering system shown in Figure 4 has a torque motor operated hydraulic servo which positions a spool type fuel metering valve. A linear variable differential transformer (LVDT) is used for feedback. The system includes bypass-type metering head regulator and a combination pressurizing and shut off valve. The shutoff valve can be operated via the shutoff solenoid or by PLA. The manual backup system is activated by deenergizing the changeover solenoid which transfers control of the metering valve servo from the torque motor via electric feedback to PLA via force feedback.

The servo system will be high gain so failing the torque motor to null (zero voltage) would result in only a small variation in fuel flow. However, hardover failure can occur.

The manual fuel metering backup system requires that the pilot reposition PLA to the approximate failed power level (PLA is normally at 100% in helicopter operation) to avoid a transient when changeover is initiated.

This system requires a pressure relief valve since shutdown deadheads the pump.

Solenoid Operated Spool Metering Valve

This system shown in Figure 5 uses a solenoid operated hydraulic multiplier (LVDT feedback) to position the spool-type metering valve. The hydraulic multiplier is implemented by transducing compressor discharge pressure P_{CD}

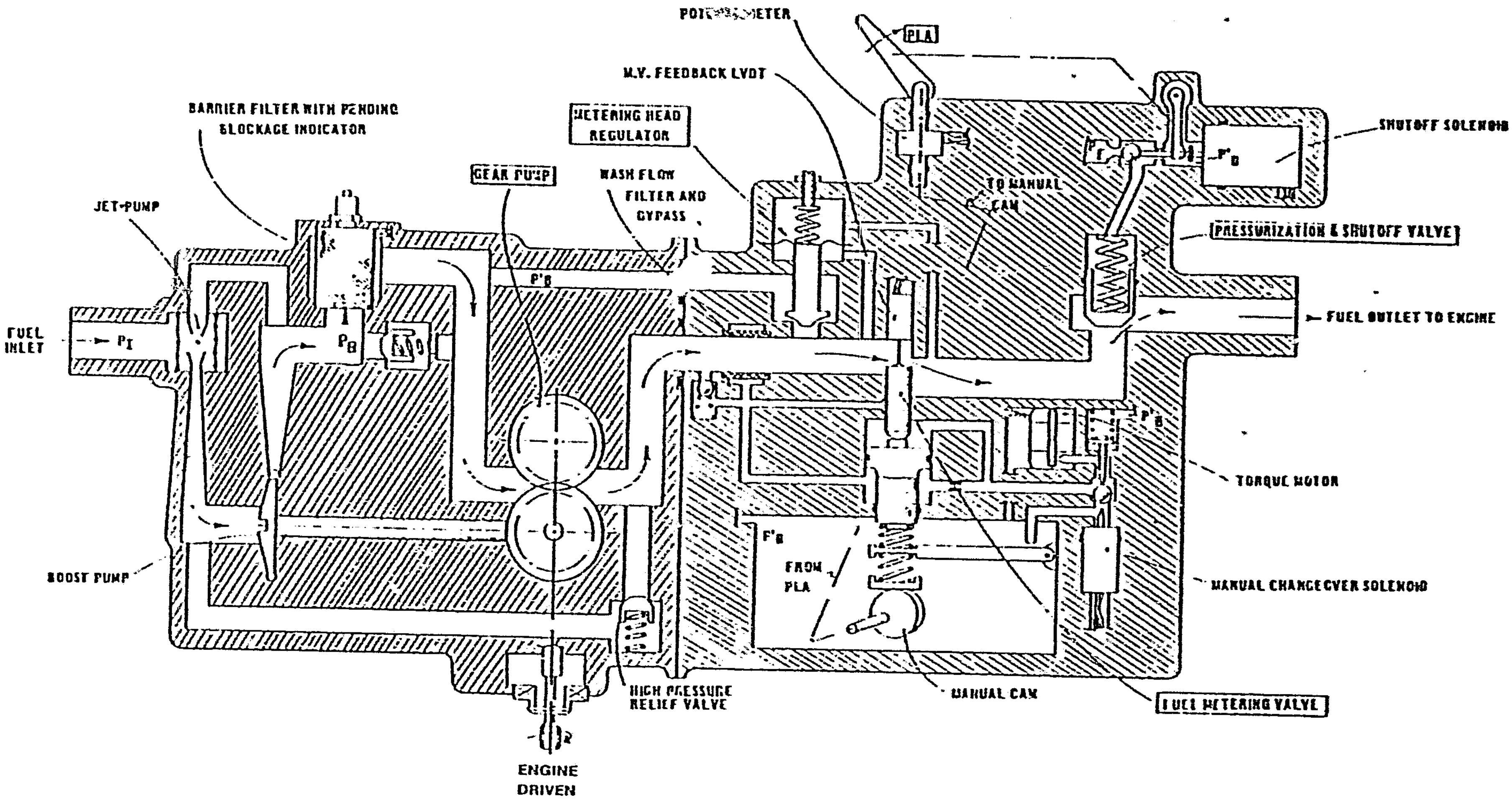


Figure 4 TORQUE MOTOR OPERATED HYDRAULIC SERVOED SPOOL TYPE METERING VALVE

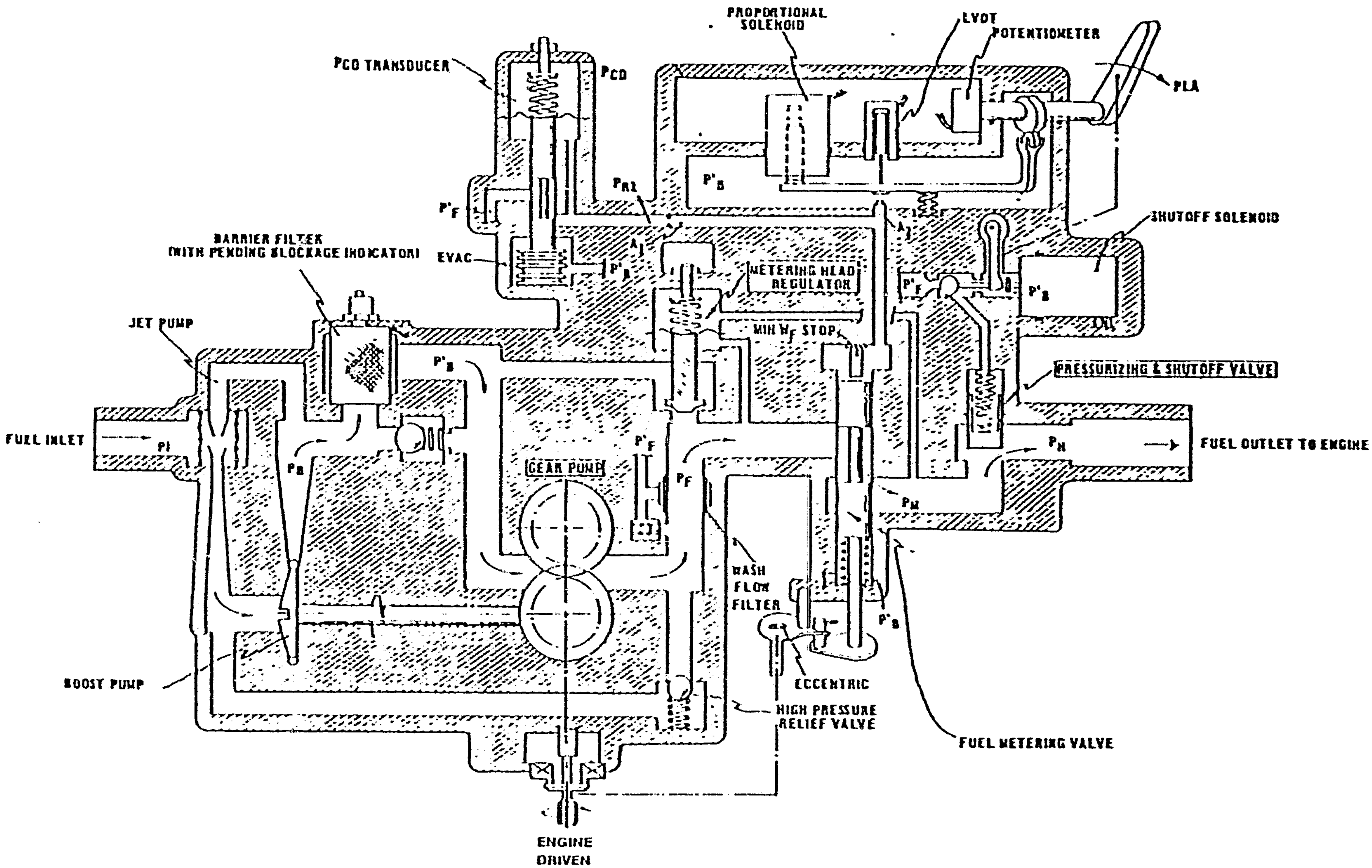


Figure 5 SOLENOID OPERATED, HYDRAULIC MULTIPLIED SPOOL TYPE METERING VALVE

to a fuel pressure P_{R3}. Varying A_2 which is in series with A_1, creates pressure P_C. P_C positions the main meter valve (spring force feedback) and thereby fuel flow is proportional to P_C since the metering head regulator holds the metering valve pressure droop (P_F-P_M) constant. The metering valve is jiggled via an eccentric drive to eliminate friction problems since metering valve P_C force levels are low. The system includes a pressurizing and shutoff valve like the one in System 2.

Manual backup is provided via PLA varying A_2. There is no changeover required. The solenoid fails to a fixed reference position and since PLA is normally at 100%, a transient to full power results.

The hydraulic multiplier results in $P_C = (W_f/P_{R3})P_{R3} = (W_f/P_{CD})P_{CD}$, thereby varying A_2, is in effect, varying W_f/P_{CD}. This gives the advantage that the backup system via PLA controls W_f/P_{CD} instead of W_f as in Systems 1 and 2. This offers altitude compensation and on some engines, it could protect the engine from surge and overtemperature.

Hysteresis Clutch Controlled Metering Gear Pump

This system shown in Figure 6 provides fuel metering by controlling the speed of the gear pump using a hysteresis clutch drive arrangement. Energizing the clutch coil sets up a magneticflux between the engine driven shaft and the output shaft via the cup configured arrangement. Engine speed and gear pump speed are sensed using magnetic speed pickups. A solenoid operated shutoff valve is located at the pump inlet.

A backup control is difficult to implement in this system. Consideration was given to providing dual stator windings for redundancy, but this does not deal with loss of electric power. The size and weight of the dual type clutch was excessive because of the design complexity.

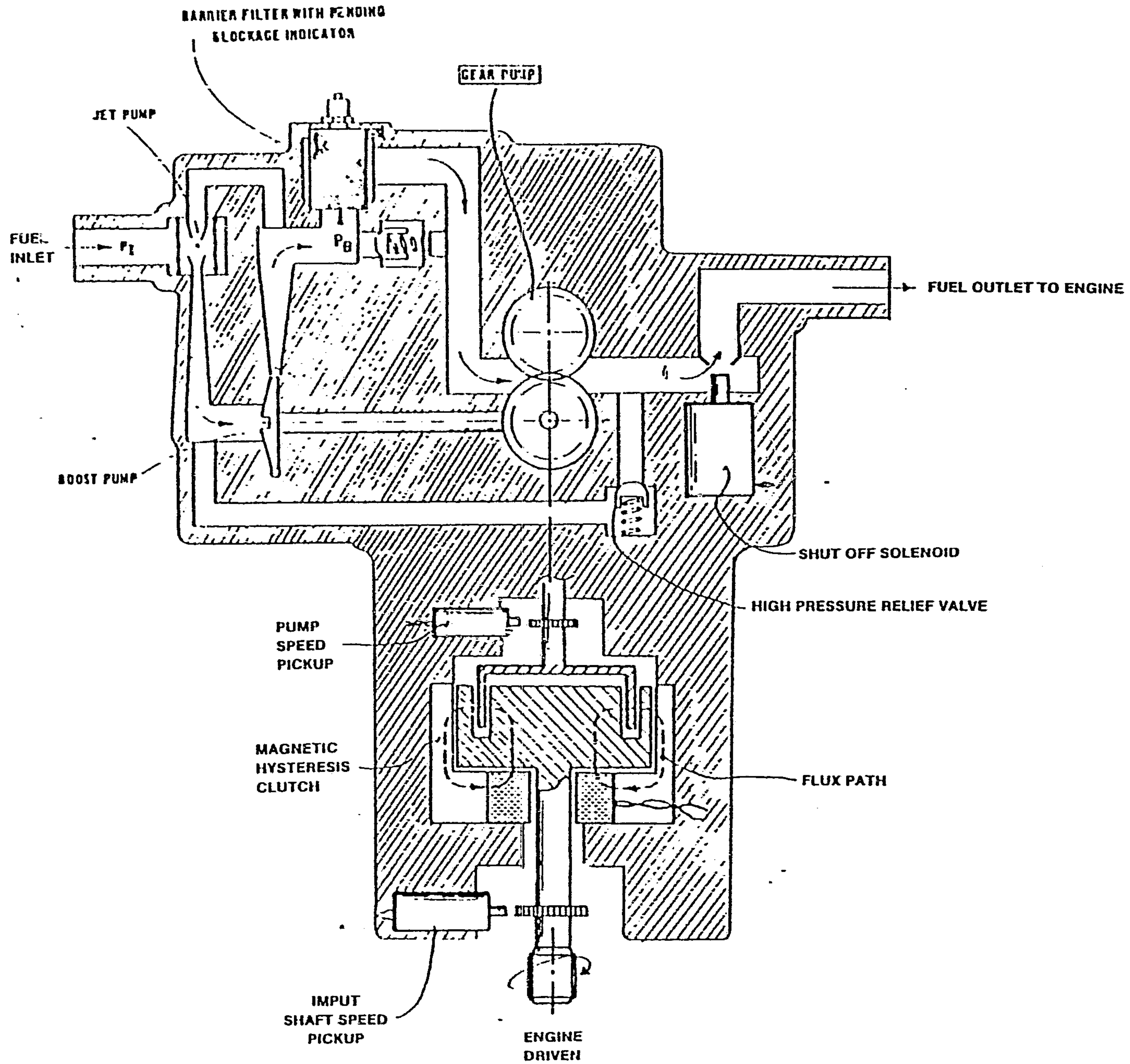


Figure 6 . **VARIABLE SPEED HYSTERESIS CLUTCH CONTROLLED METERING GEAR PUMP**

Controlled D.C. Motor Driven Metering Gear Pump

This system shown in Figure 7 uses a DC motor driven gear pump for fuel metering. A magnetic speed pickup on the pump provides feedback for sensing fuel flow. Although pump fuel flow will also vary with backpressure, it is anticipated that accurate fuel flow scheduling will not be required with closed loop full authority electronic control. The requirements for accurate flow setting for light-off and for an overspeed dump to a minimum flow are formidable problems in this scheme.

Backup control is also a problem with this system. Consideration was given to dual winding designs using samarium cobalt motors, however, weight, size and cost were significant problems in design approaches.

It is anticipated that this pumping unit could be mounted in the fuel tank, thereby eliminating the V/L and dry lift requiremets. Also, the fuel tank provides a needed heat sink for the motor.

Stepper Motor Driven Bypass Metering Valve

This metering system shown in Figure 8 utilizes a stepper motor driven metering valve as in baseline system. Fuel is directly metered to the engine by bypassing more or less pump flow back to the pump inlet as a function of electronic control demand. This scheme requires total closed loop control of fuel flow by the electronic control, based upon an appropriate engine parameter over the entire rage of operating including light off, starting, acceleration, governing and deceleration. Fuel shut-off to the engine is accomplished by fully opening the by-pass metering valve. Thus, control discharge pressure drops and allows the spring loaded shut-off valve to close.

This system provides manual control provision of direct metering valve actuation. The sensitivity of this operation may not be acceptable and an alternate back up control concept may be required to achieve a smooth transition into the back up mode to provide acceptable control characteristics.

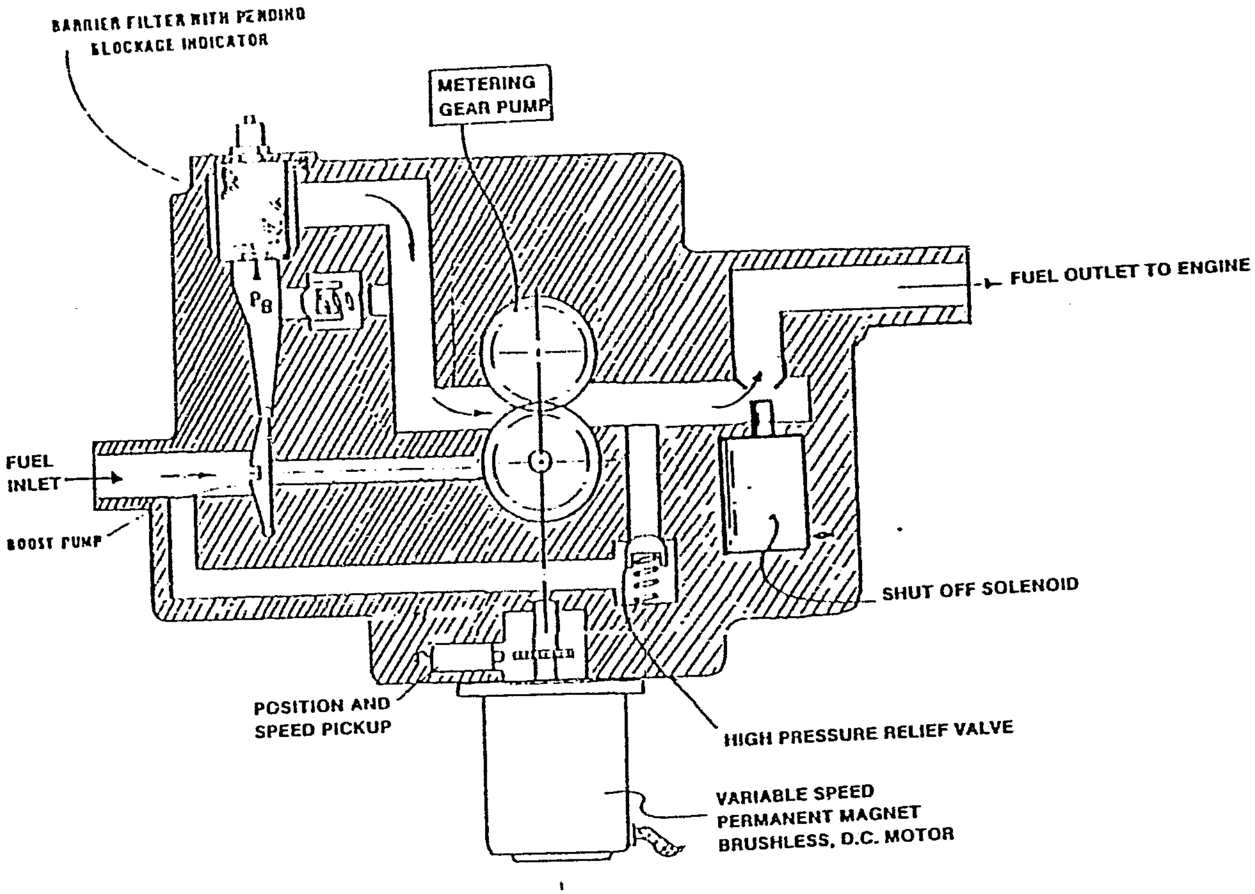


Figure 7 VARIABLE SPEED D.C. MOTOR CONTROLLED METERING GEAR PUMP (TANK MOUNTED)

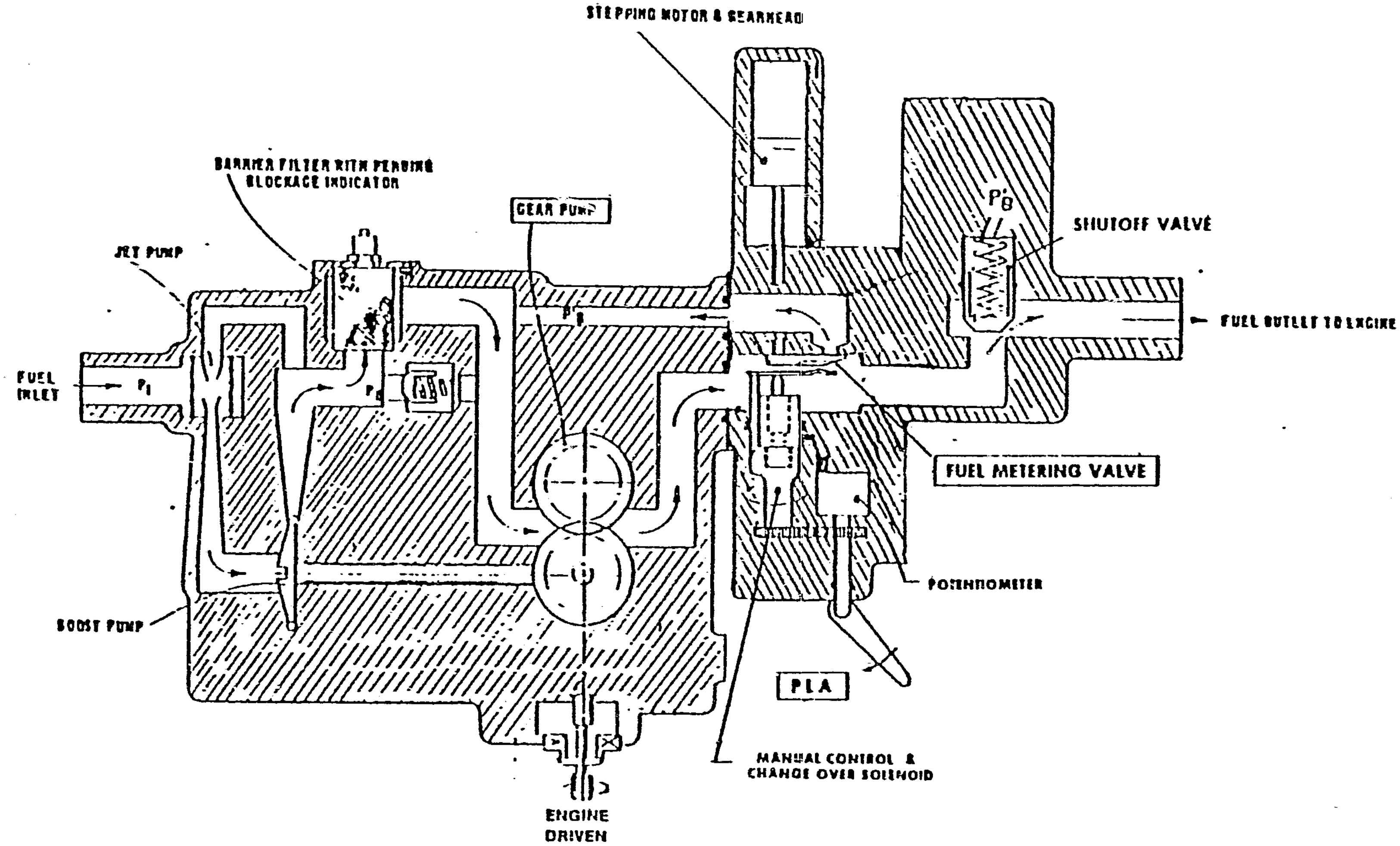


Figure 8 . STEPPER MOTOR DRIVEN BYPASS VALVE

Stepper Motor Driven Bypass Valve with Redundant Solenoid Valve

An added feature was incorporated in the concept described in Figure 8 to improve the back-up mode of operation is shown in Figure 9. A solenoid actuated valve for electronic overspeed and overtemperature protection is located in parallel with the primary sliding plate metering valve. By duty cycle modulating the relatively high frequency solenoid drive signal as a function of speed and/or temperature error from a back-up electronic control, the limiter valve proportionately opens to bypass more fuel and compensate for excessive manual inputs. Another approach to back up operation on the solenoid valve is for automatic reversion in the event of primary control failure and operation on the back-up electronic system. This arrangement effectively takes the pilot out of the control loop.

3.3.4 Fuel System Trade Studies

The seven candidate fuel pumping and metering system concepts were evaluated on a comparative basis. Assessment criteria and assigned weighting factors were established based upon engineering design, manufacturing and field service experience judgements. These criteria and weighting factors are shown in Table 8.

Table 8

Fuel Systems Assessment criteria

Factor	% Weighting
Reliability	33
Cost	25
Weight and Size	16
Maintainability	13
Back-up Operation	13

An assessment of all seven systems on weighted factors for reliability, cost, weight, and volume, maintainability, and manual backup performance is summarized in Table 9. Out of a maximum possible points of 100, the fuel systems were rated in the order shown in Table 10.

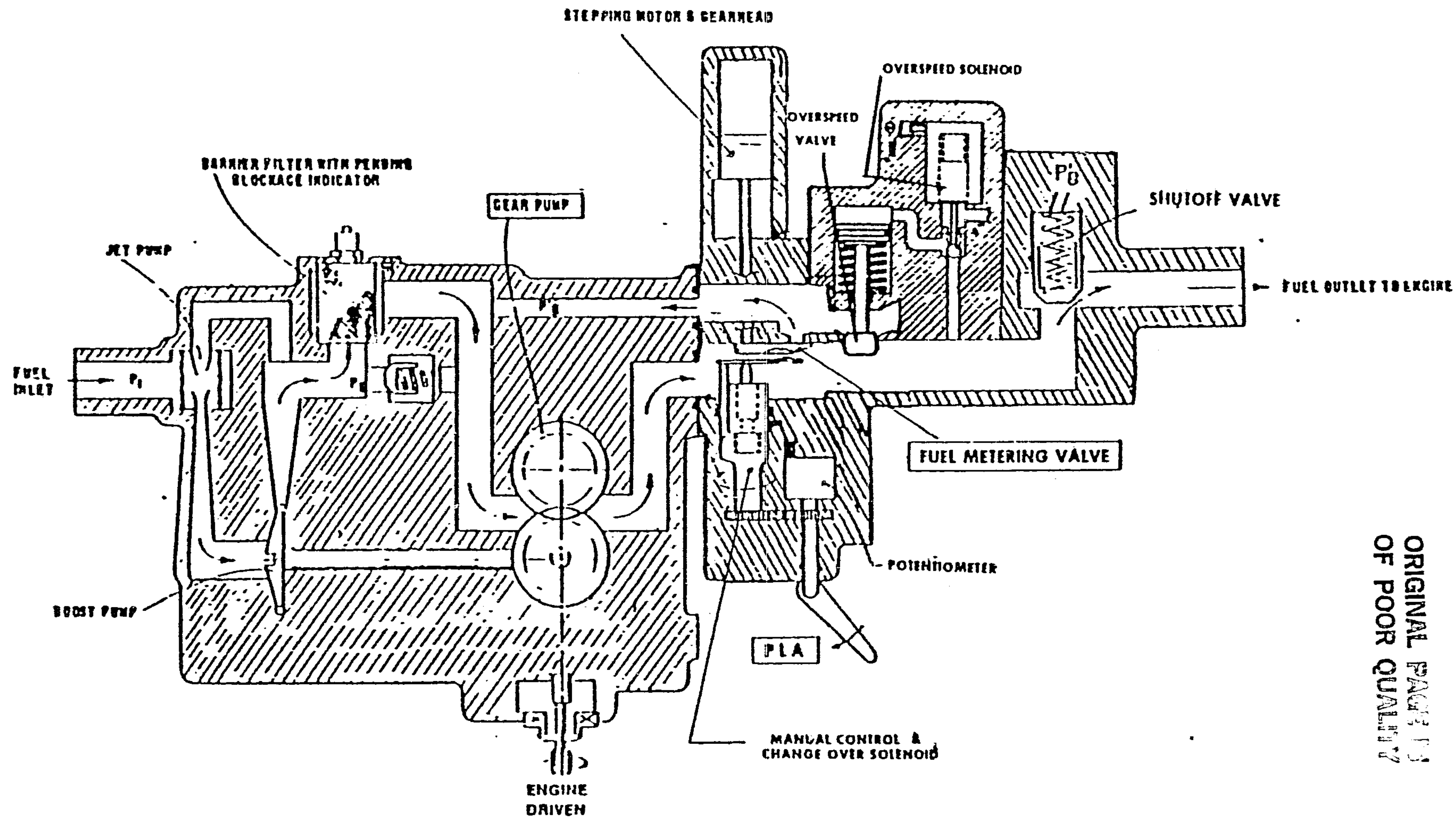


Figure 9 . STEPPER MOTOR DRIVEN BYPASS VALVE WITH SOLENOID VALVE SPEED AND TEMPERATURE LIMITING

Table 9. Fuel Handling Systems Trade-off Table

	MAX ASSESSMENT	STEPPER MOTOR	TORQUE MOTOR	PROPORTIONAL SOLENOID	HYSTERESIS CLUTCH	PERMANENT MAGNET BRUSHLESS D.C. MOTOR	STEPPER BYPASS	STEPPER & SOLENOID BYPASS
RELIABILITY	33	31	26	22	29	19	33	31
COST	25	23	20	22	25	18	25	23
WEIGHT & VOLUME	16	15	11	8	9	6	16	15
MAINTAINABILITY	13	13	13	13	13	10	13	13
BACKUP SYSTEM	13	5	4	6	0	0	2	10
TOTAL	100	87	74	71	76	53	89	92

Table 10

Fuel metering systems assessment summary.

		Points
Stepper Motor and Solenoid Bypass	(Figure 9)	92
Stepper Motor Bypass	(Figure 8)	89
Baseline System	(Figure 3)	87
Hysteresis Clutch	(Figure 6)	76
Torque Motor	(Figure 4)	74
Proportional Solenoid	(Figure 5)	71
D.C. Motor	(Figure 7)	53

The stepper motor systems were top rated primarily because of their reliability and weight advantages. With the metering valve in the bypass loop, metering head pressure regulation is not required, thus improving the overall rating of the system because of the simplification of the system. The overall rating of the stepper bypass system was high in spite of a downgraded manual backup mode. In manual, the pilot must compensate for metered flow changes as the engine and pump slow down or speed up. The stepper motor and duty cycle modulated solenoid bypass system allevite the sensitivity of the manual mode by providing engine overspeed and temperature limiting. Therefore, this system shown in Figure 9, by virture of its improved backup mode, emerged as the top rated system.

The decision was made, therefore, to proceed with the stepper motor driven bypassing metering valve system concept through a preliminary design description.

4.0 FUEL METERING SYSTEM PRELIMINARY DESIGN

This report section describes the preliminary design work conducted as Task II of the contracted effort for defining the bypassing fuel metering system selected as the best of the seven candidate systems studied in Task I of the program. A brief description of the baseline system (the GMA500 fuel pumping and metering module) is included to show by comparison design improvements made possible in the selected system. This baseline control incorporates the latest technology in advanced fuel pumping and metering components for small helicopter engines utilizing full authority digital electronic control systems. Therefore, improvements beyond this system meet the goals of this program which are to evolve advanced concepts that offer design features for improved reliability and reduced cost and weight.

4.1 Baseline System Description

A schematic of the hydromechanical section of the GMA500 engine fuel control is shown in Figure 3. The system is composed of the fuel pumping and fuel metering modules. These units are joined together at one split line by a molded elastomer seal plate. Thus, the pump and fuel metering modules can be easily separated, tested and individually maintained or overhauled.

The fuel pump is a three-stage device consisting of a jet inducer and centrifugal boost element priming a positive displacement gear pump. Return flow from the fuel metering module is routed to the inlet of the gear pump, thus minimizing the flow handling requirements of the boost pump and fuel filter.

The fuel metering module is composed of a stepper motor driven sliding plate metering valve which is in series with a solenoid-activated overspeed valve. A pressure regulator maintains a relatively constant head across the two valves by returning excess fuel flow back to the inlet of the gear pump. Thus, fuel delivered to the engine is proportional to the effective opening of the primary metering and overspeed valves.

During normal operation, the overspeed valve is wide open and engine flow is determined by the stepper motor driven metering valve. However, should an overspeed occur, the overspeed valve rapidly reduces metered fuel flow to contain the speed of the free turbine below safe limits.

A pressurization valve is located at the discharge of the control to maintain a sufficient muscle pressure (P_F-P_B) in the control. Thus, accurate metering head regulation can be achieved and actuator force levels maintained. In addition, the pressurization valve bottoms on a soft seat and provides drop tight fuel shut off to the engine when the primary valve is closed off.

A manual backup system is provided in case of a total loss of electrical power. When deenergized, the changeover solenoid allows a mechanical link to the spring loaded into place to couple the metering valve directio to PLA. Thus, the pilot can manually vary fuel flow to the engine via the gas generator power select lever in the cockpit.

4.2 Bypass System Design Features

A schematic of the simplified stepper motor driven bypass fuel metering system is shown in Figure 10. In this system the stepper motor operates the flat plate valve to meter fuel to the engine by bypassing excess pump flow. The stepper motor actuator was selected because it provides a fail-fixed control mode, thereby preventing engine runaways or flame out in the event of electronic control failure. The stepper motor is highly reliable, competitively priced and light weight.

Simple and Reliable

The bypass valve and the pressurizing/shutoff valve comprise the primary fuel metering means in this system. A metering head regulating valve which is used in all current known conventional metering systems is not required. It is not known how a simpler fuel metering system could be implemented and the evaluation study in Phase I showed this system to be the best system when measured in terms of cost, weight, and reliability.

BYPASS FUEL METERING SYSTEM

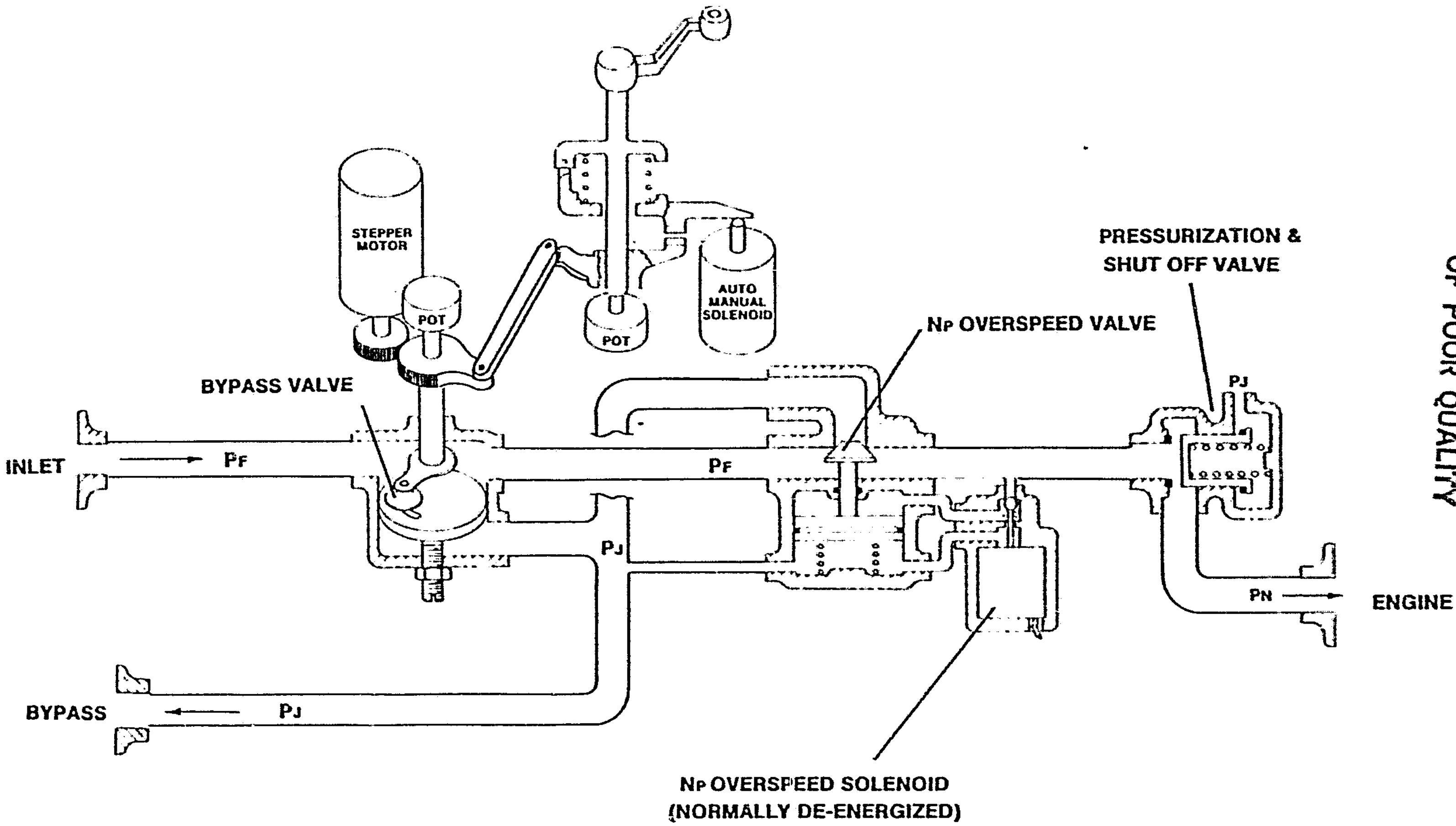


Figure 10 Bypass Fuel Metering System Schematic

Pump Considerations

The system will operate with any pumping system, but future work will concentrate on operation with a gear pump. This is because the results of the Phase I study indicated that the gear pump will likely dominate future pumping systems for helicopters. This was concluded because a positive displacement pump is needed in the helicopter to meet the dry lift requirements, and the gear pump wins because it is more reliable and lower in cost than a vane or piston pump. The fuel flow requirements for the GMA500 or the 250-C30 are very similar and the fuel pump characteristics are very similar as shown in Figure 11.

Backup Control

A backup control of fuel flow is also provided in the system through operation of the changeover solenoid. This arrangement provides manual fuel flow control by connecting the pilot's power lever to the bypass valve. This backup control was conceived in consideration of a twin engine installation and envisioned to be operated in the following procedure. If a failure should occur, the stepper fails fixed and the pilot would engage the manual mode, set a nominal power level and fly the aircraft using the other engine.

Overspeed Governor

Redundant power turbine overspeed protection, which is a requirement imposed on free turbine engines, is provided by the solenoid-operated overspeed valve hardware used in the baseline system. In the bypass system, the overspeed valve bypasses metered flow instead of throttling flow as in the baseline system. The overspeed governor can be operated either on-off or in a proportional mode via duty cycle moudlating (varying on/off time) the solenoid.

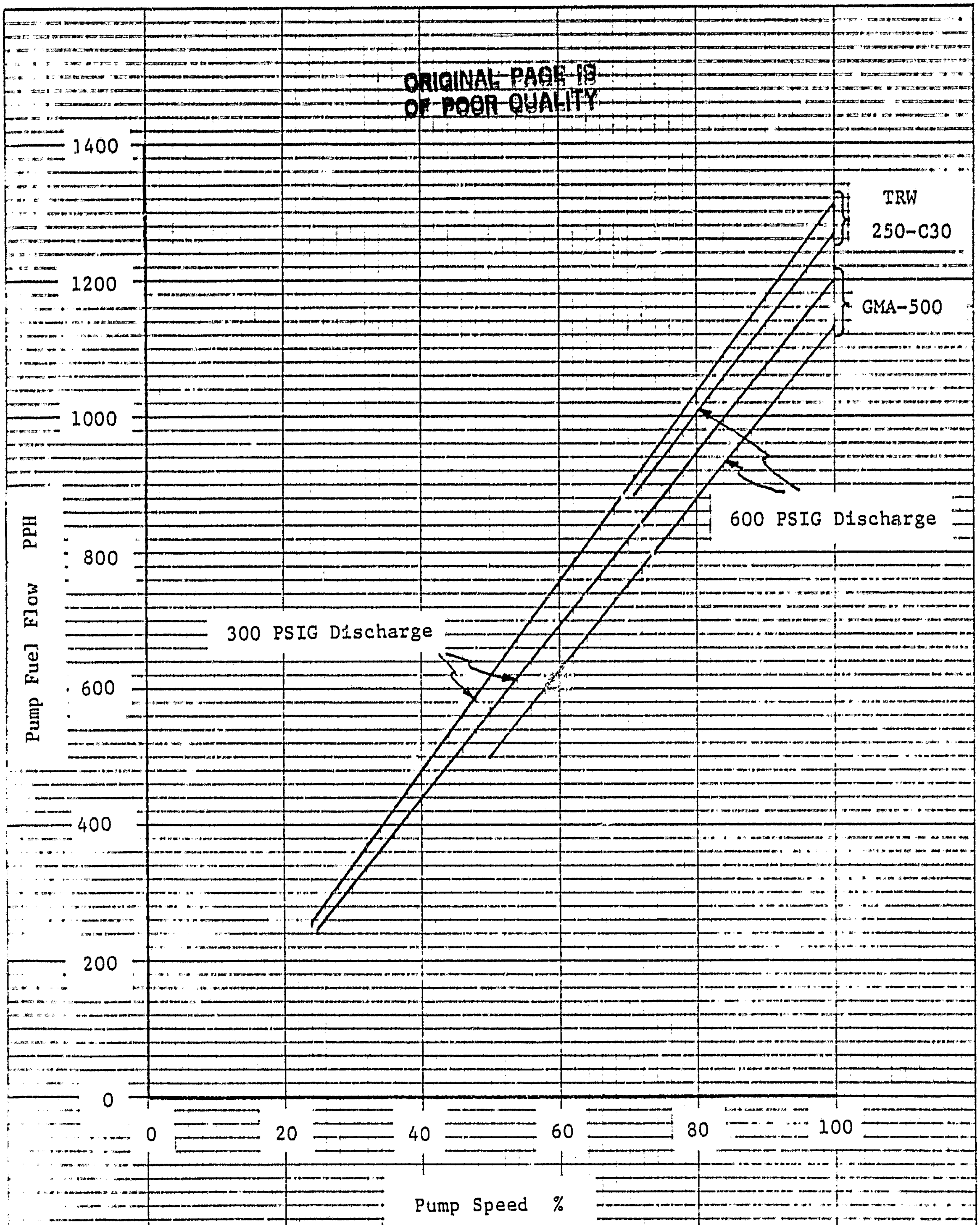


Figure 11 . Fuel Pump Flow Comparisons

Control Laws

The bypass fuel metering concept is untried and represents an advanced concept made possible only by the computational power of the digital electronic fuel control. The concept relies on the digital computer to implement continuous closed loop engine control and from available engine sensors to compensate for the fuel pump and engine fuel nozzle pressure/flow characteristics. It is planned that the operating characteristic of the pump and fuel nozzle would be stored in computer memory. This information, along with sensed speed, temperature and compressor discharge pressure should provide sufficient computer data to maintain stability and provide responsive engine closed loop control.

The bypass fuel metering system will require fuel flow control based upon closed-loop control of sensed engine parameters, as opposed to open-loop fuel scheduling. The logic to be programmed into the digital electronic controller will be based principally upon concepts which have been used in the GMA500 full authority and the 250-C30 electronic supervisory control program developments. These electronic control concepts have been extensively evaluated an simulated bench tests, engine test stand operation and in helicopter flight demonstrations. These control mode concepts, and their implementation in digital control systems, have been well validated and form a solid basis for this control mode development.

The fuel control mode can be defined as:

- o Gas producer acceleration rate $\frac{dN_g}{dt}$ control during starting and accelerations.
- o Power turbine inlet gas stream temperature ($T_{4.5}$) limiting during starting and accelerations.
- o Power turbine speed (N_p) governing, isochronously, during steady state operation.
- o Gas producer speed (N_g) maximum and minimum limiting.

A new requirement with the simplified bypassing system is that the control mode for deceleration transients must also be closed-loop on an engine parameter. In the GMA500 and 250-C30 control systems, with their more complex fuel metering systems, the decelerations are controlled at a minimum fuel flow. The simplicity of the proposed system does not enable the scheduling accuracy required to provide acceptable deceleration rate and freedom from burner blowout. Thus the requirement to employ closed-loop deceleration control. Some candiate control modes to be considered are negative $\frac{d\,N_1}{d\,t}$, rate of change of compressor discharge pressure $\frac{d\,P_{CD}}{d\,t}$ and fuel nozzle pressure drop ΔP_n control.

Another area of control requiring special investigation deals with the control of the fuel flow during starting, prior to burner light-off.

Closed-loop control on $\frac{d\,N_1}{d\,t}$ or $T_{4.5}$ is not possible prior to lightoff; and, open-loop scheduling with the simplified valve will not be accurate. Two concepts have been identified for evaluation. One involves step-ramp programming of fuel flow for light-off. Another is based upon fuel nozzle ΔP programming and closed loop control.

Control Algorithm

The digital electronic control is a "powerful" unit which brings forth the possibility of simplified fuel metering systems. In addition to the algorithms/logic for the principal control functions of governing, limiting, and mode transition programming the control program will include algorithms/logic to address metering fuel in the bypass loop. The reason for the additional logic is to provide "tight" control of the closed-loop engine parameters, which are responsive to fuel flow to the engine (combustor), while performing the fuel regulation in the bypass loop.

The basic control algorithms/logic will essentially compute how much fuel is required to the combustor. The additional algorithms/logic are to "predict" the total fuel available from the fuel pump, with the difference (total minus combustor) being the amount to be bypassed. This type of approach is required to provide compensation to the basic closed-loop control functions, and enable responsive fuel control with overshoots and undershoots minimized. In addition, system pressure sensing and valve position sensing may be useful compensating parameters to improve the dynamics of closed-loop control with this type of fuel system. Figure 12 shows schematically the functional features of this bypassing fuel metering concept.

4.3 Preliminary Design Description

Figure 13 shows a partial layout of the bypass system taken from the complete layout design. The system has been packaged to mount on an available pump adapter fixture that will allow installing the unit on the GMA-500 engine pump or on the C30 engine pump. Thereby, if hardware is fabricated for a future engine demonstration test, the test could be run on either of these engines. Following is a discussion of some of the pertinent design considerations that were implemented to reduce production manufacturing, assembly and test costs.

Stepper Motor/Bypass Valve Assembly

For proper operation to prevent valve wear or leakage, the flat plate valve must be free to float on the valve plate. This was accomplished in the baseline system via a spherical joint on the metering valve drive shaft. A separate plate pinned to the top of the drive shaft drives the metering valve via a pin closely fitted to a hole in the top of the metering valve.

In the bypass system a simplified mechanical arrangement was devised for reduced cost and simplicity. The metering valve is driven via a leaf spring that is riveted to the drive shaft. A cone shape slot in the leaf spring fits over a cone shape pin on the metering valve allowing the metering valve to float. This arrangement eliminates close machining tolerances and reduces assembly time.

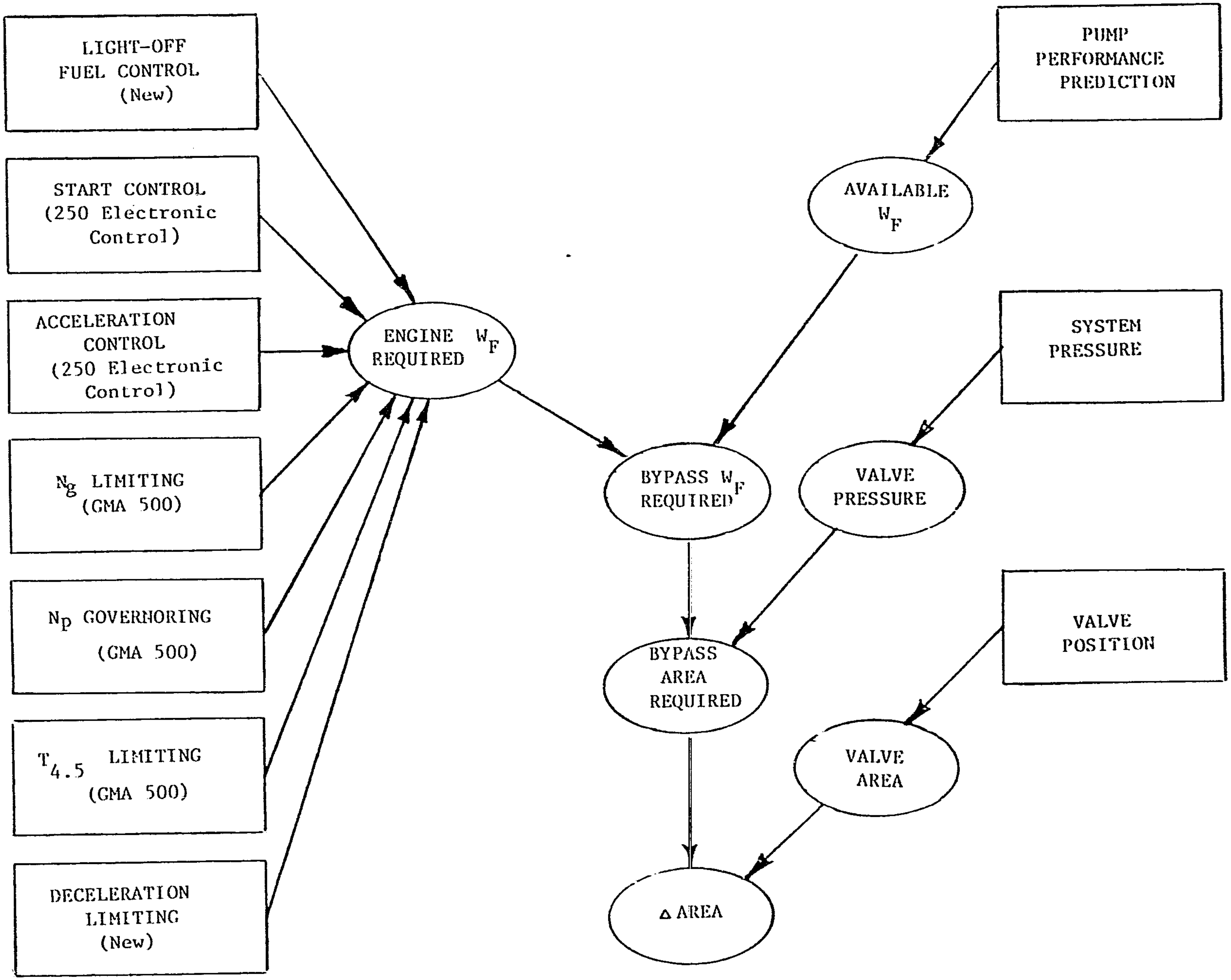


Figure 12. Bypass fuel metering system functional schematic.

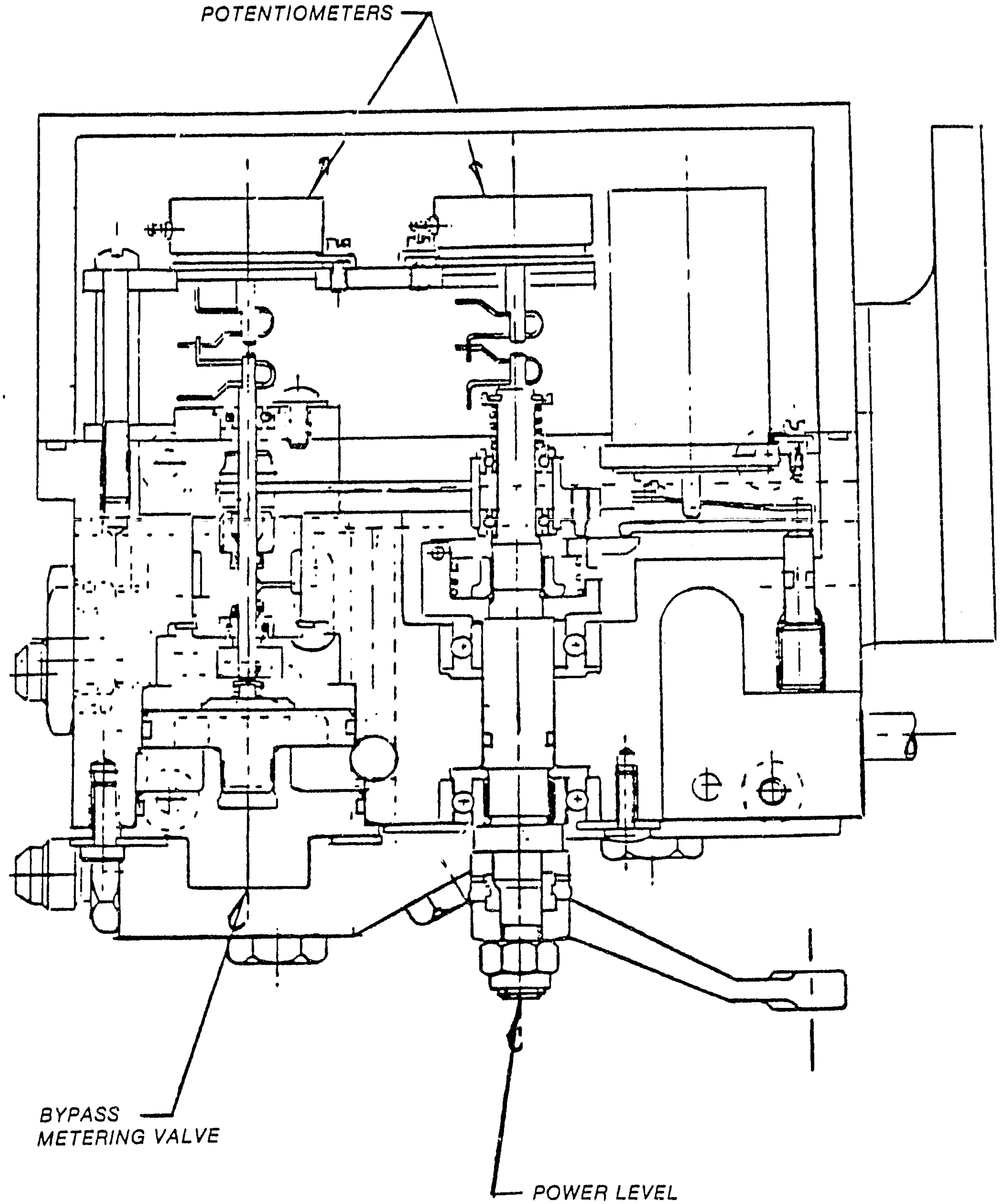


Figure 13 Bypass Fuel Metering System Preliminary Design Layout

Additional simplification in this assembly is made in the method used to drive the feedback potentiometer. The baseline system uses a pinion gear which requires accurate alignment. The bypass system uses interlocking spring clips which allow considerable misalighment, both axially and concentrically, and this lower cost arrangement minimizes side loads on support bearings.

Manual Backup

The manual backup design in the baseline system requires a precise shimming operation to insure proper parallelism and clearance between the manual engagement link and pin and the slotted metering valve drive plate. The amount of clearance required is determined by the displacement of the solenoid plunger and in this design shimming to a tight clearance of about 0.008 in. is required.

The simplified bypass system uses a larger solenoid which provides more displacement to allow larger clearances. The design provides two screw adjustments to set the proper parallelism and clearance between the drive pin and the slotted plate. This arrangement saves assembly costs associated with the shimming operation used in the baseline system. Moreover, the adjustments are made with the unit operational, thereby eliminating reassembly work when a shimming error is made.

4.4 Conclusions

The simplified bypassing system includes only two flow handling valves which is a uniquely simple concept for metering fuel made possible by the computational power of the digital computer. Its simplicity will offer improved reliability and lower cost and weight as compared with current systems.

Preliminary value engineering design work has addressed production costs considering fabrication, assembly and test requirements in view of our experience with the baseline system.

To establish the capability of the bypass system will require computer-aided control mode studies to develop the closed loop engine control laws. Current plans for a follow-on program to conduct this work and then run a closed loop bench test using a real-time engine simulator will provide a conclusive evaluation as to the merits of the bypass system.

5.0 CONCLUSIONS AND RECOMMENDATIONS

Conceptual design and trade studies were performed in three areas of control component technologies for digitally controlled aircraft engines. Two areas of study were directed at the DDA GMA400 variable geometry advanced technology engine for compressor geometry actuation devices and the fuel pumping and metering functions. The third study considered simplified fuel pumping and metering systems for small turboshaft engines for advanced technology rotorcraft.

Eight conceptual system designs were formulated for the actuation of the variable compressor vanes of the GMA400 and comparative trade studies performed to obtain a relative assessment of the various candidate systems. These studies indicate that an electric motor driven screw jack system or a hydraulic cylinder fuel pressure operated system are the more suitable systems for the GMA400 type engine. The final selection is dependent upon factors such as allowable fuel temperature rise and overall considerations on the selections of electric motor actuators for other engine control functions.

The conceptual design and trade studies for the fuel pumping and metering system for the GMA400 engine considered six system concepts. From these studies it was concluded that the preferable pump configuration was a throttled discharge centrifugal pump having an integral retracting vane starting stage. A single pump suitably sized to supply total engine flow, for both primary and secondary nozzle systems, is recommended. While the trade studies did not indicate the selection of any specific system configuration, the study results do show, in a systematic manner, the factors upon which a decision can be made. The final system configuration is dependent upon relative weighting of the various assessment factors which, in turn, are dependent upon engine application and mission requirements.

The third area of study was directed at defining a simplified fuel pumping metering concept suitable for full authority digital electronic control systems for small advanced technology rotorcraft engines such as the DDA GMA500 advanced technology demonstrator engine. Trade studies were accomplished on seven conceptual system designs and comparative evaluations made on the basis of reliability, cost, weight and volume, maintainability, and backup system. These studies resulted in the selection of a simple bypassing fuel metering system. A preliminary design layout was completed on this novel fuel metering concept which is compatible with existing fuel pumps and digital electronic controllers. This simplified system design shows great promise of improved reliability and reduced cost as compared to existing fuel handling systems.

Follow on development work is recommended to further explore the performance capabilities of the bypassing fuel metering concept. This work should include the detail design of the metering system including control mode definitions for total closed loop operation over the entire range of engine operation. The system should then be fabricated and software developed to implement the control modes. System evaluations should be performed on closed loop simulated engine/control bench tests.

APPENDIX A

Variable Geometry Actuation Systems Trade Study Calculations

1. <u>Hydraulic Cylinder Actuators</u> - Compressor Variable Geometry

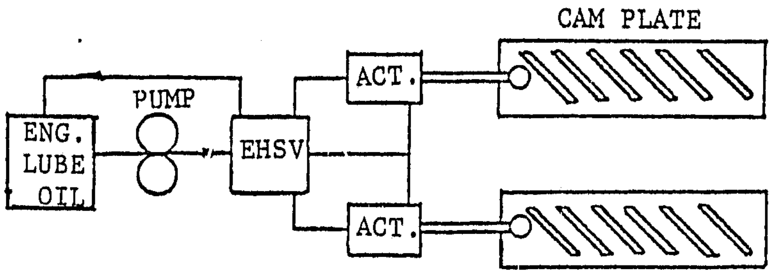
<u>Requirements</u>

Load - 1500# (axial) per side - max load @ S.L.

Travel - 2 inches

Slew Time - 2 seconds

<u>Hyraulic Oil Operation</u> - MIL-L-7808



<u>Use 600 psig oil pressure at slew</u>

Hydraulic cylinder size

Piston area required = $\frac{1500 \text{ lb. load}}{600 \text{ psig}} = 2.5 \text{ in.}^2$

Diameter = $\sqrt{\frac{4 \times 2.5}{\pi}} = 1.784$ in.

Assume piston rod diameter of 5/8" (0.675 in.)

Cross sectional aea of rod = $\frac{\pi(.675)^2}{4} = 0.358 \text{ in.}^2$

Total cylinder area = $2.5 + 0.358 = 2.898 \text{ in.}^2$

Cylinder I.D. = $\sqrt{\frac{4(2.898)}{\pi}} = 1.92$ in.

Hydraulic cylinder volume = $2.898 \text{ in.}^2 \times 2$ in. (travel) = 5.796 in.^3

<u>Max flow requirement</u> = $\frac{5.796 \text{ in.}^3}{2 \text{ sec.}} \times 2 \text{ actuators} \times \frac{60 \text{ sec}}{\text{min}} \times \frac{\text{gal}}{231 \text{ in.}^3}$

= <u>1.51 G.P.M.</u>

<u>Use 2400 psig at slew</u> (3000 psig system)

Piston Area = $\frac{1500}{2400} = 0.625 \text{ in.}^2$

Cross sectional area of 0.675 in. piston rod = 0.358 in.^{2}

Total cylinder area = 0.625 + 0.358 = 0.983 in.^{2}

Cylinder I.D. = $\sqrt{\frac{4(0.983)}{\pi}}$ = 1.12 in.

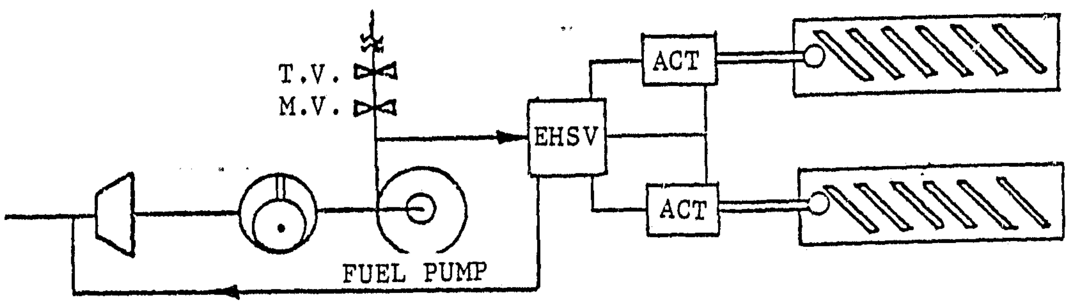
Cylinder volume = 0.983 in.^{2} x 2 in. = 1.966 in.^{3}

Max flow requirement= $\frac{1.966 \text{ in.}^3}{2 \text{ sec}}$ x 2 actuators x $\frac{60 \text{ sec}}{\text{min}}$ x $\frac{\text{gal}}{231 \text{ in.}^3}$

= 0.51 GPM

Fuel Operation

Cylinders must be sized for maximum load and also for minimum supply pressure. Actuator loads are comprised almost entirely of vane pivot bearing loads which are proportional to compressor discharge pressure (CDP). This relationship, however, does not apply as we approach CDP equal to zero and a static friction factor of 20% of max load (300 lbs) is considered to be the minimum load condition. Aerodynamic loads are balanced such that no appreciable vane torque exists that must be overcome by the actuators. The following calculations determine the size of actuator cylinder required.



	Altitude	N_H Mech	Mach No.	Air Flow	CDP	Fuel Flow	Fuel Nozzle P	Pump Out Pr.
	S.L.	100%	0.8	213.2 #/sec.	379.8 PSIA	33196 #/Hr.	385 PSI	800 PSIG
)	S.L.	100%	1.2	277.7	370.0	18056	255	745
)	S.L.	100%	1.2	281.3	370	40996	520	1065
)	36K'	100	2.0	179.1	252.68	15531	250	620
)	70K'	73.85	0.8	7.6	8.05	337	100	207
)	70K'	77.93	2.5	33.3	40.97	1545	125	270

n the basis of CDP values, the max load occurs at S.L. N_M = 1.2, and min oad occurs at 70,000' at N_M = 0.8. All six operating points are within the 0-100% N_H corrected speed range where compressor vane operation occurs. ctuator diameter is determined by fuel pump discharge pressure available and he maximum load requirements within this range.

Calculation of Cylinder Diameter.

	Condition	Wf	Pump Disch.Pr.	CDP	Load/ Actuator	Cyl. Area	Cyl. Dia.
)	S.L. N_M = 0.8	33196 #/Hr	800 psig	279.8 PSIA	1134	1.42 in.^{2}	1.35 in.
)	S.L. N_M = 1.2	18056	745	370	1500	2.01	1.6
)	S.L. N_M = 1.2	40996	1065	370.0	1500	1.41	1.34
)	36K' N_M = 2.0	15531	620	252.68	1024	1.65	1.45
)	70K' N_M = 0.8	337	207	8.05	300	1.45	1.36
)	70K' N_M = 2.5	1545	270	40.97	300	1.11	1.18

ondition 2 determined actuator cylinder diameter. Actual I.D. must include liameter of piston rod. Assume rod O.D. of 5/8 (0.675) in. Cross sectional rea of rod = $\frac{\pi(.675)^2}{4}$ = .358 in.^{2}.

otal cylinder area = 2.01 + .358 = 2.37 in. Cylinder I.D. = $\sqrt{\frac{4(2.37}{\pi}}$ = 1.74 in.

Cylinder volume and travel are close to dimensions of Arkwin Industries P/N 1211127 hydraulic cylinder.

Actuator volume required = 2.37 in.^{2} x 2 in. x 2 act = 9.48 in.^{3}

$$\text{Max. flow requirement} = \frac{9.48}{2\ \text{sec}} \times \frac{60\ \text{sec}}{\text{min}} \times \frac{1\ \text{gal}}{231\ \text{in.}^3} = \underline{\underline{1.23\ \text{GPM}}}$$

Fuel pump capacity must be increased by <u>1.23 GPM</u>

<u>Weight Estimate</u>

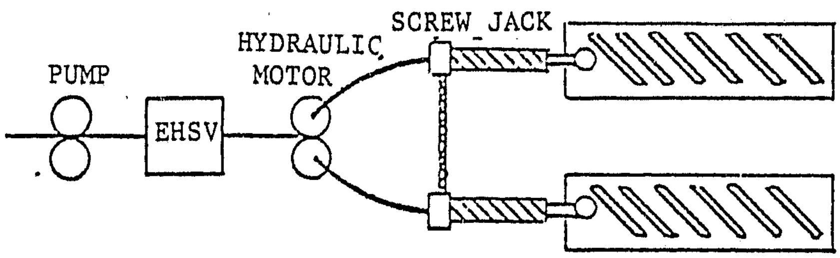
<u>Fuel Actuators</u>

2 fuel actuator cylinders Arkwin No. 1211127 - 1.5# ea.		3.0#
1 hydraulic servo (EHSV) Moog model 62-102		2.7#
Fuel lines, fittings, brackets, LVDT		5.0#
	Total	10.7#

<u>Oil Actuators</u>

2 hydraulic actuator cylinders (3000 psi) - 1# ea		2.0#
1 hydraulic servo (EHSV) - Abex model 410		1.0#
1 variable delivery hydraulic pump		[illegible].5#
Oil lines, fittings, brackets, LVDT		5.0#
	Total	11.5#

2. <u>Hydraulic Motor Operated Screw Jacks</u>



Calculation of Motor Size

Condition	Cam Plate Load	Motor Torque Reqd.	Motor HP	Pump Disch. Pr.	Pump Cap. Reqd.
1) S.L. N_M = 0.8	1134	112.8 in.#	.536	800 PSIG	0.920 GPM
2) S.L. N_M = 1.2	1500	149.2	.710	745	1.310
3) S.L. N_M = 1.2	1500	149.2	.710	1065	.914
4) 36K' N_M = 2.0	1024	101.86	.485	620	1.070
5) 70K' N_M = 0.8	300	14.9	.071	207	.470
6) 70K' N_M = 2.5	300	14.9	.071	270	.360

Sample Calculation -

Assume 0.25" lead screw jacks,

$$\text{Ball screw RPM} = \frac{4 \text{ rev}}{1'' \text{ travel}} \times \frac{2'' \text{ travel}}{2 \text{ sec}} \times \frac{60 \text{ sec}}{\text{min}} = 240 \text{ RPM}$$

$$\text{Torque input} = \frac{\text{load} \times \text{lead}}{2\pi \times \text{eff.}} \times \frac{1500 \times .25}{2\pi \times 0.8} = 74.6\#$$

Motor torque - 2 x actuator torque = 149.2 in.#

$$\text{Motor hp} = \frac{\text{Torque} \times \text{RPM}}{63025 \times \text{Motor eff.}} = \frac{149.2 \times 240}{63025 \times .8} = \underline{\underline{.71 \text{ HP}}}$$

$$\text{Pump Capacity Reqd} = \frac{1714 \text{ HP} \times \text{eff.}}{\text{Press.}} = \frac{1714 \times .71 \times .8}{745}$$

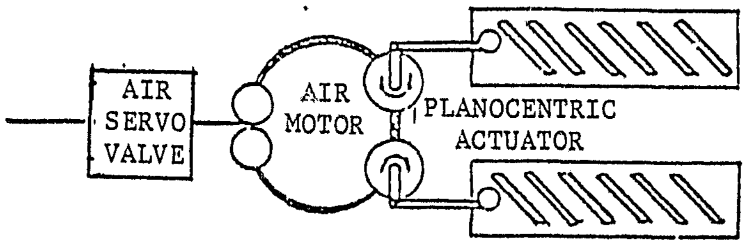
Condition 2 determines delta capacity = 1.31 GPM

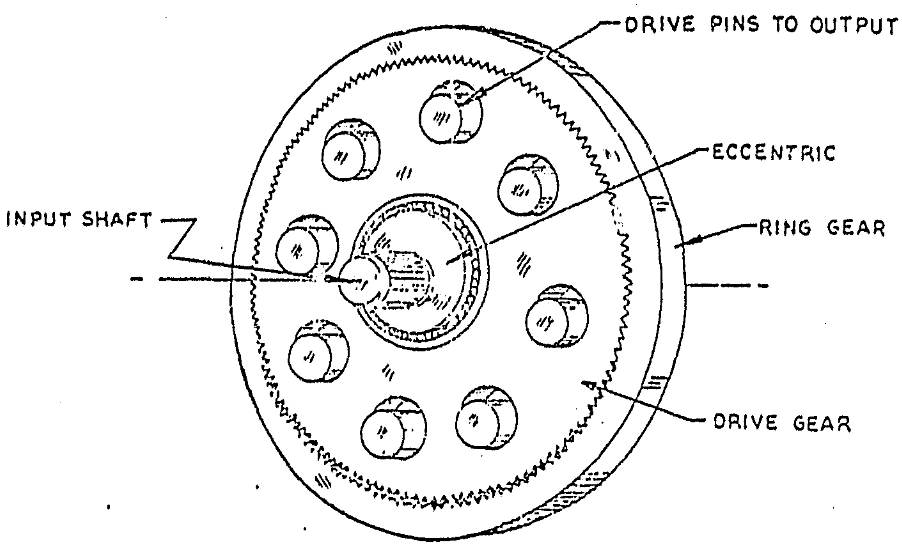
Weight Estimate

2 screw jack type actuators 1.25# ea	2.5#
1 hydraulic motor similar to Sundstrand Model 012782	1.0#
1 electrohydraulic servo valve - Moog Model 62-102	2.7#
Lines, fittings, brackets, LVDT	5.0#
Total	11.2#

ORIGINAL PAGE IS
OF POOR QUALITY

3. Air Motor Operated Planocentric Actuator





PLANOCENTRIC SPEED REDUCER

The planocentric actuator is a high gear ratio speed reduction device which converts low torque at high speed at the inlet to high torque at low speed at the outlet. Its strong points are its compact size and good position resolution of the output.

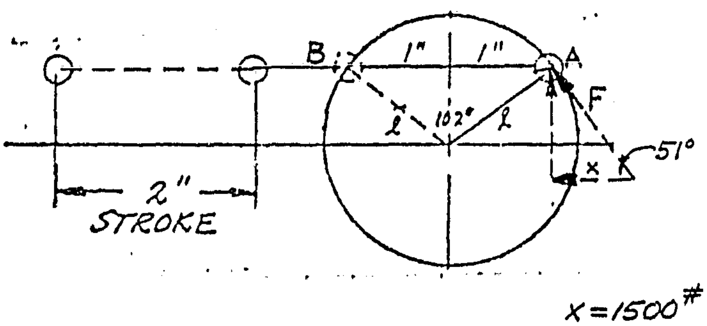
The actuator is composed of an external drive gear which rotates inside a stationary ring gear having more teeth than the drive gear. The drive gear is driven by an eccentric cam on the input shaft which causes it to walk around the surface of the ring gear. Since there are fewer teeth on the drive gear than on the ring gear, the drive gear precesses in the opposite direction.

The output shaft which is connected in one manner or another to the drive gear turns at a slower speed and in the opposite direction to the input shaft. Speed reduction,

$$S_R \quad \frac{\text{No. teeth on ring gear}}{\text{No. teeth on ring gear - No. teeth on drive gear}}$$

The planocentric actuator for this application has a power requirement approximately four times that of the GMA 200 H.P. turbine actuator. The stroke (2 inches) is an order of magnitude greater than the GMA 200 (0.225 inches). Output torque is also an order of magnitude greater. Weight of the existing actuator for lower temperature environment (less than 500°F) is two pounds. Physical characteristics of the GMA 200 actuator are used to size the larger torque actuator.

Planocentric lever length (1) for 2" stroke



Use 102° of planocentric output shaft travel to obtain linearity.

Lever length 1 = 1" x cosecant 51°

= 1.29"

Speed ratio = 81.74:1

$$F = \frac{x}{\text{Cos } 51^{\circ}} = \frac{1500}{.629} = 2383.5\#$$

Max. output torque (1 at Point A) = 1.29 x 2383.5 = 307475 in.#

Max. output torque of GMA 200 actuator = 187.8 in.# (KNOWN)

Actuator input torque = 3074.75 - (81.74 x 3 x 0.44 Eff.) = 31.35 in.#

$$\text{Motor rotor torque} = \frac{2(\text{Actuators}) \times 31.35 \text{ in.}\#}{5(\text{Red. Ratio}) \times 0.69 \text{ (Motor Eff)}} = 18.17 \text{ in.}\#$$

Actuator weight for higher torque = Act. wt, $(\text{torque}_2/\text{torque}_1)^{0.8}$

$= 2\ (3074.75/187.8)^{0.8} = 18.72$

Actuator input speed = 102°/2 sec x 60/360 x *(81.74 x 3) = 2084 RPM, Motor RPM = 2084 x 2.5**= 5210 RPM *Total planocentric reduction ratio

**Air motor reduction ratio

ORIGINAL PAGE IS OF POOR QUALITY

Since output torque of these actuators is approx. 16x greater than GMA 200 actuators, air motor torque required is 2(actuators) x (16x)/4(actuators-GMA 200) = 8 x air motor torque for GMA 200.

If air motor reduction ratio is increased from 2.5 to 5.0, then air motor slew speed under load is same as GMA 200 air motor (10400 RPM) and torque required will be halved, from 8x to 4x.

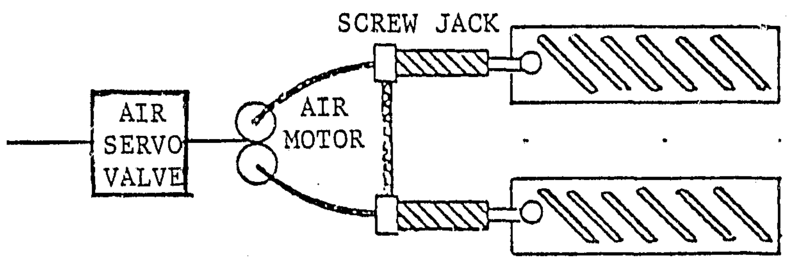
Wt of increased torque = GMA 200 motor wt $(\text{torque ratio})^{0.8}$

= $7.5 \times (4)^{0.8}$

Weight Estimate = 22.74#

2 increased torque planocentric actuators @ 18.74# each	37.44#
1 increased torque air motor	22.7
2 flexible torque air motor	2.0
Inlet and exhaust air piping	2.0
Air motor bracket	1.0
Total	58.04

4. Air Motor Operated Screw Jack Actuators



ORIGINAL PAGE IS OF POOR QUALITY

Calculation of Motor Size

Assume screw jack has 0.25" lead/revolution

Actuator torque = load x lead/2π x eff = 1500 x 0.25/2π x 0.8

Actuator eff = 0.8

= 74.6 in.#

Air Motor Output torque reqd = 149.2# (2 actuators)

From configuration 2), Ball screw RPM = 240 RPM

This RPM is too low for efficient operation of air motor

Use screw jack lead = 0.1"/rev

Actuator torque input = 1500 x 0.1/2 x 0.8 = 29.84 in.#

Ball screw RPM = 2 in. travel/0.1"/rev/2 sec = 10 rev/sec = 600 RPM

Air motor with 17.33/1.0 reduction will run at 10400 which is slew speed of GMA 200 air motor under load. Larger motor can now be sized on basis of relative torques.

Motor torque @ 10,400 RPM = 29.84 x 2/17.33 x .69 (Motor Eff.) = 4.99 in.#

Torque of GMA 200 motor rotor = 187.8 x 4/*(81.74 x 3 x 2.5) x .69 (Motor Eff) = 4.44 in.#

*Reduction ratios from reduction output to motor gears.

Ratio of torques = 4.99/4.44 = 1.12

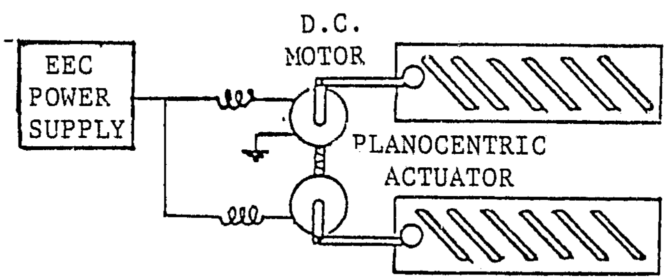
Weight of higher torque motor = $(1.12)^{0.8}$ x 7.5

= 1.09 x 7.5 = 8.23

Weight Estimate

2 screw jack actuators @ 1.25# each	2.50#
1 air motor	8.23
2 flexible drive cables @ 1# each	2.00
Air piping, bracketry, electrical harness	3.00
Total	15.73#

5. Electric Motor Operated Planocentric Actuators



From configuraton 3 planocentric actuator input torque = 3074.75 in.#/*(81.74 x 3)(0.40)* *40% efficiency

*Reduction ratio across planocentric actuator

Electric motor torque reqd = 31.35 in.# x 0.8 (eff.) = 39.19 in.#

Proposal from clifton Precision Div. of Litton Industries defines a 148 VDC motor with samarium cobalt magnets operating at 9600 RPM and a torque load of 9.75# (156 oz in.)

At 9600 RPM motor RPM, planocentric actuator is rotating at 9600/81.74 x 3 = 39.15 RPM (0.653 RPS).

.653 RPS = 234.89^{0}/second

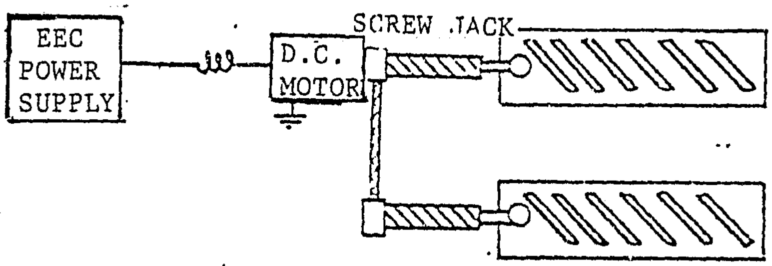
Planocentric used 102^{0} travel, therefore full travel will occur in 102^{0}/234.89^{0}/sec = 0.43 sec.

Full travel requirement is 2 sec. With a 4.19:1 reduction between motor and actuator the time for full travel is still 2 sec., including 0.2 sec to bring motor up to speed and to decelerate, and actuator torque is 40.81 in.# whereas required actuator torque is 39.19 in.#. Actuation time, i.e. motor speed, can be controlled by modulating armature field current or by pulse width modulation of input voltage.

Weight estimate

2 increased torque planocentric actuators # 18.72# ea (ref. configuration 3)		37.44#
2 Clifton Precision electric motors @ 2.3# ea		4.6
2 Motor speed controllers (same wt as motors)		4.6
1 Spur gear reduction unit - 4.65:1		1.5
3 Actuator & motor brackets @ 1.0# ea		3.0
2 Flexible drive cables @ 1.0# ea		2.0
	Total	53.14

6. Electric Motor Operated Screw Jacks



Assume screw jack with 0.10" load/revolution

Actuator torque (from configuration 4) = 29.84in.#/actuator

Use 10/1 gear ratio between motor and actuator

Motor torque = 2.98 in.#

Operate two actuators with one motor - reqd torque = 5.96 in.#

Ball screw RPM @ slew (from config. 4) = 600 RPM

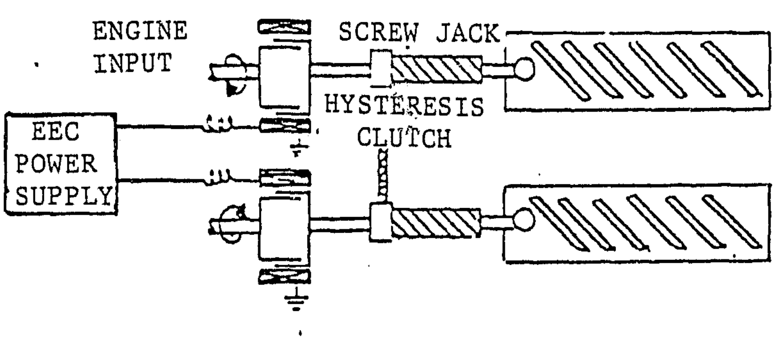
Motor RPM @ slew = 600 x 10 (gear ratio) = 6000 RPM

As in configuration 5 speed regulation from 9600 rated RPM to 6000 RPM at slew can be obtained by modulating armature field currnt or by pulse width modulation of input voltage. One Clifton Precision motor with lower torque rating similar to one used in configuration 5 is also used in this configuration.

Weight Estimate

2 screw jack type actuators @ 1.25# ea	2.5#
2 Flexible drive cables @ 1.0# ea	2.0
1 Clifton Precision electric motor @ 2.0# ea	2.0
1 Motor speed controller (same wt as motor)	2.0
2 Spur gear reduction units - 10:1 @ 1.5# ea	3.0
2 Actuator and motor brackets @ 1.0# ea	2.0
Total	13.5#

7. Hysteresis Clutch Operated Screw Jack Actuators



Assume screw jack with 0.25" load/revolution

Actuator torque reqd = load x screw jack lead/2 x eff   eff = 80%

= 1500 x 0.25/2 x 0.8 = 74.6 in.# average torque - does not include excess torque required for acceleration.

Use 1.5 sec slew time instead of 2.0 sec to allow extra time due to inertia effect of hysteresis clutch and gear reduction during acceleration.
Ball screw RPM = 2"/1.5 sec x Rev/0.25" x 60 sec/min = 320 RPM
Assume engine pad speed RPM (input to clutch) = 13000 RPM
Gear ratio required between clutch and ball screw = 13000/320 = 40.625
Clutch torque requird/actuator = 74.6 in.#/40.625 x 16 oz/# = 29.38 in. oz.

Input drive from the engine is one direction only (CCW or CW). Therefore one clutch must drive two screw jacks in one direction. The other clutch must drive the two screw jacks in the reverse direction. This is accomplished by driving one screw jack directly and back driving the other screw jack via an interconnecting loop cable. Clutches must be sized to deliver two times torque of each screw jack, i.e. 2 x 29.38 = 58.76 in. oz.

There is additional torque required to accelerate clutch and gear reducer up to slew speed. Assume time required to get actuators up to slew speed is .050 sec (50 ms). Based upon similar application, acceleration torque required to overcome system inertia is approximately equal to steady state torque. Therefore required clutch torque = 2 x 58.76 = 117.52 oz. in. This requirement can be met by Delevan size 45 clutch which has a rating of 120 oz. in. Weight of speed reducer (assume planocentric type) = weight of planocentric config. 3 x (torque of config. 7)/(torque of config. 3)$^{0.8}$ = 18.72 (74.6 in.#/3074.75 in.#)$^{0.8}$ = 18.72 x .051 = 0.955# (use 1.0#)

Weight Estimate

2 size 45 Delevan hysteresis clutches @ 8.7# ea		17.4#
2 41:1 speed reducers @ 1.0# ea		2.0
2 1" O.D. x 0.25" lead screw jacks @ 1.25# ea		2.5#
2 flexible drive cables @ 1.0# ea		2.0
1 interconnecting loop cable @ 1.0# ea		1.0#
2 actuator mounting brackets @ 1.0# ea		2.0
	Total	26.9#

Engine Cycle Effects

Configuration 1 Horsepower requirements @ 80% system efficiency.

$$HP = \frac{GPM \times \Delta P}{1714 \times Eff.}$$

$$= \frac{0.51 \times 2400}{1714 \times 0.8}$$

$$= 0.893 \text{ hp}$$

Configuration 2 Horsepower requirements @ 80% system efficiency.

$$HP = \frac{1.31 \times 745}{1714 \times 0.8}$$

$$= 0.71$$

Configuration 3 Air horsepower requirements @ 23% overall system efficiency.

Planocentric output torque = 3074.75 in #

" input torque = $\dfrac{\text{output torque}}{\text{actuator reduction ratio}}$

" " " = $\dfrac{3074.75}{81.74 \times 3} \times 0.4$ eff.

" " " = 31.35 in #

Motor output torque = 31.35 x 2 actuators

= 62.7 in #

Motor rotor torque = $\dfrac{\text{output torque}}{\text{motor gear reduction ratio x systems e}}$

Motor eff. = 69%

Motor rotor torque = $\dfrac{62.7}{5 \times 0.69}$ = 18.17 in #

Motor rotor speed = actuator input RPM x motor red. ratio.

= 2084 x 5

= 10420 RPM

Motor horsepower = $\dfrac{\text{Motor torque x Motor RPM}}{63025}$

$$= \frac{18.17 \times 10420}{63025}$$

= 3.00 hp

Motor Eff. = 69%, Planocentric Eff. + 40%, Overall Eff. = 28%

Configuration 4 Air horsepower required @ 55% overall system efficiency.

$$\text{Actuator input torque} = \frac{1500 \times 0.1}{2 \times 0.8} \quad @ \ 80\% \text{ eff.}$$

$$= 29.84 \text{ in \#} / \text{Actuator}$$

Actuator RPM (0.1 "/rev) @ Slew = 600 RPM

Use 17.33/1.0 reduction so that motor runs @ 10400 RPM.

$$\text{Air motor torque} = \frac{2 \times 29.84}{17.33 \times .69} \quad \text{Motor eff.} = 69\% \text{ same as GMA 200.}$$

$$= 4.99 \text{ in \#}$$

$$\text{Air motor hp} = \frac{4.99 \times 10400}{63025} = 0.82 \text{ hp}$$

Configuration 5

$$\text{Planocentric actuator input torque} = \frac{3074.75 \text{ in \#}}{(81.74\text{x}3)\text{x}0.40(\text{eff.})}$$

$$= 31.35 \text{ in \#}$$

$$\text{Electric motor torque required} = \frac{31.35 \text{ in \#}}{0.8\,(\text{motor eff.})\text{x}4.65 \text{ red. ratio}}$$

$$= 8.43 \text{ in \#}$$

$$\text{Motor hp} = \frac{\text{Torque x RPM}}{63025}$$

$$= \frac{8.43 \times 9600}{63025}$$

$$= 1.28 \text{ hp}$$

$$= 1.28 \times 0.746 = 0.958 \text{ kw}$$

Configuration 6 Horsepower requirements

Actuator torque (from configuration 4) = 29.84 in #/actuator

Use 10:1 gear ratio between motor and actuator

Motor torque = 2.98 in #/actuator

Use one motor to operate two ball screw actuators

Motor torque = 2 x 2.98 = 5.96 in #

Ball screw RPM (0.1"/rev) = 600 RPM

Motor RPM = 6000 RPM

$$\text{Motor H.P.} = \frac{\text{torque} \times \text{RPM}}{63025 \times \text{eff.}} \qquad \text{motor eff.} = 0.9$$

$$= \frac{5.96 \times 6000}{63025 \times .9} = 0.63 \text{ hp}$$

$$= 0.47 \text{ kw}$$

Configuration 7

Shaft horsepower required to drive hysteresis clutch.

Actuator torque = 74.6 in #

$$\text{Clutch hp} = \frac{\text{torque} \times \text{RPM}}{63025}$$

$$= \frac{\dfrac{74.6 \times 13000}{40.625 \text{ (red. ratio)}}}{63025}$$

$$= 0.3788$$

Total shaft hp required = 0.3788 x 2 actuators

= 0.758 hp

Max wattage required for zero slip = 8

Total electrical power required = 16 watts

Configuration 8

Horsepower Requirements at 60% system efficiency using JP-4 fuel from main engine pump

Fuel flow required for actuators = 1.23 GPM

Fuel pressure available at max. load condition = 745 psig

$$\text{HP} = \frac{\text{GPM} \times \text{Press.}}{1714 \times 0.6 \text{ (eff.)}}$$

$$= \frac{1.23 \times 745}{1714 \times 0.6}$$

$$= 0.891 \text{ hp}$$

Maintainability

Rationale

The ease with which a system can be maintained is based upon 1) accessibility for in site repair or for removal, 2) the number of non related components that must be removed, 3) the number of fuel, oil, electrical, and mechanical linkage connections that must be disconnected and connected, and 4) the time required for rework and re-installation. Weight of the individual components does not figure appreciably in this study since the largest single component is the air motor which weights 22.7#, (configuration 3), somewhat heavier than a man's bowling ball.

With the exception of item 3, a quantitative analysis depends on a final configuration of the engine or at least an engine mockup neither of which exist at this time. Therefore, relative maintainability ratings are based upon the number of corrections that must be made and broken while removing components for trouble shooting or replacement.

Relative Maintainability Numbers of Components

Fuel, oil, electrical, mechanical connections - subtract 0.02 for each connection.

Pneumatic connection - subtract 0.03 for each connection.

Pump mounting and motor mounting - subtract 0.05 for each.

Clutch and actuator mounting - subtract 0.05 for each.

System Scores

Configuration	Connections	Factor		Score
1	Oil (12)	0.76		Product of Factors
	Electrical (2)	0.96		
	Mechanical			
	Cam plate (2)	0.96		
	Pump Mounting (1)	0.95		
	Actuator Mounting (2)	0.90		
			Total	0.617
2	Oil (8)	0.84		
	Electrical (2)	0.96		
	Mechanical			
	Cam plate (2)	0.96		
	Flex cables (6)	0.88		
	Pump Mounting (1)	0.95		
	Motor Mounting (1)	0.98		
	Actuator Mounting (2)	0.90		
			Total	0.571
3	Electrical (2)	0.96		
	Pneumatic (2)	0.94		
	Mechanical			
	Cam plate (2)	0.96		
	Motor Mounting (1)	0.95		
	Flex cables (6)	0.88		
	Planocentric Mounting (2)	0.90		
			Total	0.652

Configuration	Connections	Factor		Score
4	Electrical (2)	0.96		
	Pneumatic (2)	0.94		
	Mechanical			
	Cam plate (2)	0.96		
	Flex cables (6)	0.88		
	Motor Mounting (1)	0.95		
	Actuator Mounting (2)	0.90		
			Total	0.652
5	Electrical (4)	0.92		
	Mechanical			
	Cam plate (2)	0.96		
	Motor Mounting (2)	0.90		
			Total	0.795
6	Electrical (4)	0.92		
	Mechanical			
	Cam plate (2)	0.96		
	Motor Mounting (2)	0.90		
			Total	0.795
7	Electrical (3)	0.94		
	Mechanical			
	Cam plate (2)	0.96		
	Flex cables (6)	0.88		
	Clutch Mounting (2)	0.90		
			Total	0.715
8	Fuel (9)	0.82		
	Electrical (2)	0.96		
	Mechanical			
	Cam plate (2)	0.96		
	Actuator Mounting (2)	0.90		
			Total	0.680

Reliability

Rationale

The reliability of these systems is based upon the confidence that one has that a component will perform when called upon to do so. This is based to a great extent upon design simplicity. The components which make up the complete systems include those which have been in use in aircraft power systems for some time and for which there is a relative feeling of confidence. Included also are those components which have little service experience such as the planimetric actuator and hysteresis clutch which offer potential advantages which were felt to be worth investigating. Therefore, the component in which there is the highest degree of confidence is given a rating of 1.0 and the one with the least experience is rated 0.90 with all others rated in between.

Relative Reliability Numbers of Components

Linear hydraulic actuators - fuel and oil	1.00
Hydraulic oil pump	0.95
Electrohydraulic servo valve (EHSV)	0.95
Hydraulic motor	0.95
Screw jack actuator	0.98
Air motor and servo valve control	0.90
Planocentric actuator	0.90
D.C. motor	0.90
Hysterisis clutch	0.90
Electrical connectors	0.95
Synchronizing cable between actuators	0.98
Hydraulic piping	1.0
Flexible drive cables	0.98

System Scores

Configuration	Components	Factor		Score
1	Hydraulic Actuator (2)	1.0 each		Product of factors
	Hydraulic oil pump	0.95		
	EHSV	0.95		
	Electrical Conn. (2)			
	EHSV	0.95		
	LVDT	0.95		
			Total	0.815
2	Screw jack actuator (2)	0.98 each		
	Synchronizing cable	0.98		
	Hydraulic motor	0.95		
	EHSV	0.95		
	Hydraulic pump	0.95		
	Electrical Conn. (2)			
	EHSV	0.95		
	LVDT	0.95		
	Flexible drive cables (2)	0.98		
			Total	0.699
3	Planocentric Actuator (2)	0.90		
	Flexible drive cable (3)	0.98		
	Air motor and servo	0.90		
	Electrical conn. (2)			
	Air servo	0.95		
	LVDT	0.95		
			Total	0.619

Configuration	Components	Factor	Score
4	Screw jack actuator (2)	0.98	
	Flexible drive cable (3)	0.98	
	Air motor and servo	0.90	
	Electrical conn (2)		
	Air servo	0.95	
	LVDT	0.95	
		Total	0.734
5	Planocentric Actuator (2)	0.90	
	D.C. motor drive (2)	0.90	
	Electrical connector		
	D. C. motor (2)	0.95	
	LVDT (2)	0.95	
		Total	0.534
6	Screw jack actuator (2)	0.98 each	
	D.C. motor (2)	0.90 each	
	Electrical conn.		
	D. C. motor (2)	0.95 each	
	LVDT (2)	0.95 each	
		Total	0.634
7	Screw jack actuator (2)	0.98 each	
	Hysteresis clutch (2)	0.90 each	
	Flexible drive cable (3)	0.98 each	
	Electrical conn.		
	Hysteresis clutch (2)	0.95 each	
	LVDT (1)	0.95 each	
		Total	0.628

Configuration	Components	Factor		Score
8	Hydraulic actuator (2)	1.0		
	Hydraulic piping	1.0		
	EHSV	0.95		
	Electrical Conn. (2)			
	EHSV	0.95		
	LVDT	0.95		
			Total	0.857

COMPONENT	COST - $	CONFIGURATION 1	2	3	4	5	6	7	
Fluid Actuator With EHSV									
a) Fuel	3500								
b) Oil	3500	7000							
Air Motor With Servo Valve	7500			7500	7500				
Planocentric Actuator	3500			7000		7000			
Flexible Cable	150		450	450	450	150	150	450	
D. C. Motor (Includes Speed Re- ducer)	2000					2000	2000		
Ball Screw Jack	1500		3000		3000		3000	3000	
Hysteresis Clutch	1500							3000	
Brackets	300	600	900	900	900	600	600	1200	
Hydraulic Motor With EHSV	4500		4500						
Variable Delivery Hydraulic Pump	4500	4500							
Air Supply and Exhaust Ducts	450/Set			450	450				
D.C. Motor Controller	1500					1500	1500		
Speed Reducer						With Motor	With Motor	With Clutch	
Total		12100	8850	16300	12300	11250	7250	7650	76

FAILURE ANALYSIS

COMPONENT	RESULT	RATING
HSV	Can move to any desired position - once	0.95
crew Jack	Geometry remains in last position	0.60
.ir Motor	Geometry remains in last position	0.60
.ir Servo	Can move to any desired position - once	0.95
.ydraulic Motor	Geometry remains in last position	0.60
ydraulic Pump	Geometry remains in last position	0.60
lexible Cable	System operation is unaffected with one failure	1.00
lanocentric ctuator	Geometry remains in last position	0.60
ysteresis lutch	<u>Electrical</u> - Geometry can move to either extreme position	0.80
	<u>Mechanical</u> - Geometry remains in last position	0.60
ydraulic ylinder	<u>Seizure</u> - Geometry remains in last position	0.60
.C. Motor	Geometry remains in last position	0.60
.C. Motor ontroller	Geometry remains in last position	0.60

Rating 1.00 - System is unaffected
0.95 - Can move to any position once
0.80 - Can move to either extreme position
0.60 - Geometry remains in last position

ORIGINAL [illegible]
OF POOR QUALITY

SYSTEM FAILURE RATINGS

CONFIGURATION	COMPONENTS	PRODUCT SCC
1	1, 6, 10	0.342
2	1, 2, 5, 7	0.342
3	3, 4, 7, 8	0.342
4	2, 3, 4, 7	0.342
5	7, 8, 11, 12	0.216
6	2, 7, 12	0.360
7	2, 7, 9	0.480
8	1, 10	0.570

VULNERABILITY - Effects of Minimal Battle Damage

1) 0.6 Eventual loss of power source - loss of system - fire hazard

2) 0.9 Power source damaged - low supply pressure

3) 1.0 Power source damaged - alternate source available

All configurations are equally vulnerable to damage to mechanical linkage between the power source and the cam plates. This includes screw jacks, hydraulic cylinders, and direct mechanical links all of which can be twisted, bent or otherwise distorted and prevent them from being moved. This analysis looks at the effect of minimal bottle damage which leaves the mechanical parts of the system undamaged but affects the system power source. In the most extreme situation 1) the hydraulic power source is hit and the fluid leaks away until none is left and the system is inoperative. A fire hazard exists if the surrounding area is hot enough to cause combustion. A less serious situation occurs 2) in a pneumatic system because the compressor continually supplies air to the actuation system and the system continues to operate at a lower pressure level and a lesser level of performance, unless, of course, an air supply line is completely severed.

The least severe situation 3) occurs in an electrical system. The system can continue operation by providing a redundant electrical supply harness and means for operating the system off the airplane batteries if the alternator is knocked out.

APPENDIX B

Justification of Ratings Assigned in GMA 400 Fuel System Trade Study

	Cost	1978 $
Configuration (1)	Pump Assy (inducer, start pump, cent. element)	8,500
	Metering Valve Control Assy.	3,000
	Flow Divider and Drain Valve	2,000
		$13,500
Configuration (2)	Pump and Metering Valve Control Assy.	10,000
	Centrifugal Pump (no start pump-no inducer)	4,000
	Metering Valve Control Assy.	2,500
	Shut-off Valve (2) at $1000 each	2,000
		$18,500
Configuration (3)	Pump and Metering Valve Control Assy.	11,000
	Metering Valve Control Assy.	2,500
	Shut-off Valve (2) at $1000 each	2,000
		$14,500
Configuration (4)	Pump and Metering Valve Control Assy.	10,000
	Gear Pump	5,000
	Variable Speed Drive and Control[a]	3,000
	Metering Valve Assy.[b]	1,500
	Shut-off Valve (2) at $1000 each	2,000
		$21,500
Configuration (5)	Centrifugal Pump and Inducer	4,500
	Metering Valve Control Assy.	2,500
	Gear Pump	5,000
	Variable Speed Drive and Control	3,000
	Metering Valve Assy.[b]	1,500
	Shut-off Valve (2) at $1000 each	2,000
		$18,500
Configuration (6)	Pump and Metering Valve Control Assy.	10,000
	Pump (2100 PPH) and Retracting Vane Pump	5,000
	Metering Valve Control	2,500
	Shutoff Valve (2) at $1,000 each	2,000
		$19,500

(a) Variable Speed Drive $1500
Servo Valve $1500

(b) Metering Valve Control Assy. less throttle valve and pressurizing valve - contains delta P sensor.

Weight Comparisons

Configuration (1)	Pump Assy (42,000 PPH)	27#
	1) Retracting Vane Start Pump	
	2) Inducer	
	3) Centrifugal element	
	Metering Valve Control	
	1) Metering valve	
	2) Throttling Valve	
	3) Pressurizing Valve	
	4) Torquemotor Servo	
	5) Delta P Sensor	5
	Flow Divider & Drain Valve	12
	TOTAL	44
Configuration (2)	Pump Assy (21,000 PPH)	19
	Metering Valve Control (2) 3#/ea	6
	Centrifugal Pump (21,000 PPH)	8
	Shut-off Valve (2) 2#/ea.	4
	TOTAL	37
Configuration (3)	Pump Assy (42,000 PPH)	27
	Metering Valve Control (2) 3#/ea.)	6
	Shut-off Valve (2) 2#/ea.	4
	TOTAL	37
Configuration (4)	Pump Assy (21,000 PPH)	19
	Metering Valve Control - Woodward Gov. (21,000 PPH)	3
	Gear Pump - includes 2 gears, sleeve bearings and castings	10
	Variable Speed Drive	7.5
	Metering Valve - includes valve and EHSV	2.5
	Shut-off Valve (2) 2#/ea.	4
	TOTAL	46
Configuration (5)	Same weight as Configuration (4) - different arrangement	
	TOTAL	46
Configuration (6)	Pump Assy (21000 PPH) includes 4200 PPH inducer	20
	Pump (2100 PPH) and Retracting Vane Pump	12
	Metering Valve Control (2) 3#/ea.	6
	Shutoff Valve (2) 2#/ea.	4
	TOTAL	42

Reliability

Rationale:

The reliability of these systems is based upon the confidence that one has that a component will perform when called upon to do so. This in turn is based upon design simplicity. The components include those which have been in use in aircraft fuel systems for some time and for which there is a relative feeling of confidence. Also included are those which have no service experience (VSD and retracting vane starting pump). Therefore, the component in which there is the highest degree of confidence is given a rating of 1.0 and the one with the least experience is rated 0.90. All others are rated in between. The exception is the retracting vane starting pump which operates for a minute or less during each engine start and is subject to very little wear and fuel contamination. For these reasons, it is rated 0.95.

Relative Reliability Factors for Individual Components

Centrifugal pump (42000 pph)	1.0
Centrifugal pump (21000 pph)	1.0
Retracting Vane Starting pump	0.95
Metering Valve Control (42000 pph)	0.95
Metering Valve Control (21000 pph)	0.95
Flow Divider	0.95
Fuel Inlet Inducer (42000 pph)	0.98
Shut-off Valve	0.98
Metering Valve (Pressure drop control only - no throttle valve)	0.96
Gear type fuel pump	0.98
Variable speed drive unit	0.90
Electrical connectors	0.95

Reliability

Configuration	Components	Factor	Score
1	Electrical connectors (5)	0.95 ea.	(Product of factors)
	Pump (42000 pph)	1.0	
	Flow divider	0.95	
	Metering Valve Control	0.95	
	Retracting Vane Pump	0.95	
	Inducer	0.98	
			Total 0.65
2	Electrical connectors (6)	0.95 ea.	
	Pump (21000 pph) - primary	1.0	
	Pump (21000 pph) - main	1.0	
	Retracting vane pump	0.95	
	Inducer	0.98	
	Metering valve control-primary	0.95	
	Metering valve control-main	0.95	
	Shut-off valve-primary	0.98	
	Shut-off valve-main	0.98	
			Total 0.59
3	Electrical connectors (6)	0.95 ea.	
	Pump (42000 pph)	1.0	
	Retracting vane pump	0.95	
	Inducer	0.98	
	Metering Valve control-primary	0.95	
	Metering valve control-main	0.95	
	Shut-off valve-primary	0.98	
	Shut-off valve-Main	0.98	
			Total 0.59
4	Electrical connectors (6)	0.95 ea	
	Pump (21000 pph)	1.0	
	Retracting vane pump	0.95	
	Inducer	0.98	
	Metering valve control	0.95	
	Gear pump	0.98	
	VSD	0.90	
	Metering Valve-main	0.98	
	Shut-off valve-primary	0.98	
	Shut-off valve-main	0.98	
			Total 0.54

Reliability

Configuration	Components	Factor	Score
5	Electrical connectors (6)	0.95 ea.	(Product of factors)
	Pump (21000 pph)	2.0	
	Retracting vane pump	0.95	
	Inducer	0.98	
	Metering valve control	0.95	
	Gear pump	0.98	
	VSD.	0.90	
	Metering Valve-main	0.98	
	Shut-off valve-primary	0.98	
	Shut-off valve-main	0.98	
			Total 0.54
6	Electrical connectors (6)	0.95 ea.	
	Pump (21000 pph)-primary	1.0	
	Pump (21000 pph)-main	1.0	
	Retracting vane pum-primary	0.95	
	Retracting vane pump-main	0.95	
	Metering valve control-primary	0.95	
	Metering valve control-main	0.95	
	Shut-off valve-primary	0.98	
	Shut-off valve-main	0.98	
			Total 0.57

Maintainability

Rationale:

The ease with which a system can be maintained is based upon 1) accessibility for in situ repair or for removal, 2) the number of fuel, oil, electrical connections that must be reached for disconnecting and connecting, 3) the time and accessibility required to remove the pump from its mounting flange, and 4) the weight of the component which determines whether one mechanic can remove it or whether two mechanics or a sling are required. Two component packages reduce the number of times each component is removed for maintenance whereas one component package must be removed every time something inside malfunctions. Fault isolation makes certain that in case of two packages (Configuration 1) the right one is removed. However, this requires a condition monitoring system which the other systems do not.

Relative Maintainability Factors for Individual Components

Attach - disconnect mounting flange		1.00
Single package		0.95
Multiple packages		1.00
Fuel and oil line connection		subtract 0.02 for each connection 3
Electrical harness connections		subtract 0.02 for each connection 1
Weight	Over 30#	subtract .005/#
Fault isolation required	Yes	.95
	No	1.00

Configuration		Factors	Score
1	QAD	1.0	(Product of factors)
	Two packages	1.0	
	Fuel line connections		
	Flow divider-3	1.0	
	Pump Assy-2	1.0	
	Electrical Connections		
	Flow divider-3	0.96	
	Pump Assy-2	0.98	
	Weight		
	Flow divider 12#	1.0	
	Pump 32#	.99	
	Fault isolation - yes	0.95	
	Total		0.89

Maintainability

		Factors	Score (Product of factors)
2	QAD	1.0	
	Single package	0.95	
	Fuel line connections-3	1.0	
	Electrical connections		
	2 M.V. controls	.90	
	2 shut-off valves		
	2 RVDT feedbacks		
	Weight - 37#	.965	
	Fault isolation - No	1.0	
	Total		0.82
3	QAD	1.0	
	Single package	0.95	
	Fuel line connections-3	1.0	
	Electrical connections		
	2 M.V.controls		
	2 RVDT feedbacks		
	2 shut-off valves	.90	
	Weight - 37#	.965	
	Fault isolation - No	1.0	
	Total		0.82
4	QAD	1.0	
	Single package	.95	
	Fuel and oil line connections-5	.96	
	Electrical harness connections		
	M.V. control - 1	.90	
	RVDT feedback - 1		
	M.V. Assembly - 1		
	VSD - 1		
	S. O. valves - 2		
	Weight - 46#	.92	
	Fault isolation	1.0	
	Total		0.76
5	QAD	1.0	
	Single package	.95	
	Fuel and oil line connections-5	.96	
	Electrical harness connections		
	M.V. control - 1		
	RVDT feedback - 1	.90	
	M.V. Assy - 1		
	VSD - 1		
	S.O. valves - 2		
	Weight - 46#	.92	
	Fault isolation	1.0	
	Total		0.76

Maintainability

		Factors	Score
6	QAD	1.0	
	Single package	.95	
	Fuel line connections-3	1.0	
	Electrical harness connections	.90	
	M.V. controls - 2		
	RVDT feedbacks - 2		
	S.O. Valves - 2		
	Weight - 42#	.94	
	Fault Isolation	1.0	
	Total		0.80

Time Between Overhaul (TBO)

Rationale:

The rationale for TBO is based upon simplicity of design and operational time, For example, the centrifugal pump is a simple design and in the case of the 42000 pph pump, it will be operating most of its life at less than design flow and pressure. This also applies to the main system pump. Both of these are rated 1.0. The primary system pump is operating all the time, in contrast to the main system pump and is subject to greater wear and is rated 0.98. The retracting vane starting pump in the main system is rated higher than in the reliability analysis because it only operates in case of a failure in the primary system and if an air start is to be made. This reasoning applies to all the components in the primary and main systems.

Relative TBO Factors for Individual Components - Time before wear out

Centrifugal pump (42000 pph) 24000 RPM	1.0
Centrifugal pump (21000 pph) 28000 RPM-Primary	0.98
Centrifugal pump (21000 pph) 28000 RPM-Main	1.0
Retracting vane starting pump-primary	0.95
Retracting vane starting pump-main	1.0

TBO

Metering valve control 42000 pph	0.95
Metering valve control 21000 pph - Primary	0.95
Metering valve control 21000 pph - Main	0.98
Metering valve assy 21000 pph - Primary	0.95
Metering valve assy 21000 pph - Main	0.98
Flow divider	0.95
Fuel inlet inducer	0.98
Shut-off valve (both primary and main)	1.0
Gear type fuel pump - primary	0.95
Gear type fuel pump - main	0.98
Variable speed drive unit - primary	0.90
Variable speed drive unit - main	0.95

Configuration	Components	Factor		Score Product of factors
1	Centrifugal pump (42000 pph)	1.0		
	Retracting vane pump	0.95		
	Inducer	0.98		
	Flow divider	0.95		
	Metering Valve Control	0.95		
			Total	0.840
2	Centrifugal pump-primary	0.98		
	Centrifugal pump-main	1.00		
	Metering valve control-primary	0.95		
	Metering valve control-main	0.98		
	Retracting vane pump-primary	0.95		
	Inducer	0.98		
	Shut-off valve-primary	1.00		
	Shut-off valve-main	1.00		
			Total	0.849

TBO (Continued)

3	Centrifugal pump (42000 pph)	1.00	
	Retracting Vane pump-primary	.95	
	Inducer	.98	
	Metering valve control-primary	.95	
	Metering valve control-main	.98	
	Shut-off valves	1.00	
		Total	0.871
4	Centrifugal pump (21000 pph)	0.98	
	Retracting vane pump-primary	0.95	
	Inducer	0.98	
	Metering valve control-primary	0.95	
	Gear pump-main	0.98	
	VSD-main	0.95	
	Metering valve assy-main	0.98	
		Total	0.791
5	Gear pump-primary	0.95	
	VSD-primary	0.90	
	Inducer	0.98	
	Centrifugal pump-main	1.00	
	Metering valve control-main	0.98	
	Metering valve assy-primary	0.95	
	Shut-off valves	1.00	
		Total	0.780
6	Centrifugal pump - primary	0.98	
	Centrifugal pump - main	1.00	
	Metering valve control - primary	0.95	
	Metering valve control - main	0.98	
	Retracting vane pump-primary	0.95	
	Retracting vane pump-main	1.00	
	Inducer	0.98	
	Shut-off valves	1.00	
		Total	0.849

Performance - Response, Accuracy, and Stability

Since performance is generally a system level attribute rather than component oriented, the previous techniques of rating components is not directly applicable. The approach used here provides relative ratings for each area of performance according to different system "characteristics" deemed critical in determining the performance level.

In the case of accuracy, the metering valves will probably have the same percent of point accuracy over the desired ranges. Therefore, the two metering valve configurations can be no worse than the single metering valve and can achieve near zero error for total fuel flow when the errors cancel. This is best characterized by the root sum squared errors.

Stability is primarily a function of the interaction of the two pressure regulating controls operating the same throttling valve. This stability problem is only a potential problem in Configuration (3).

The response time is a function of pump response time, metering valve response time, and manifold fill times. Because of the remove location of the flow divider, the manifold fill time will be considerably less, especially when the main fuel nozzles come on. The throttled pump will respond to demand changes much faster than the variable speed pumps. The metering valve and flow divider responses will be approximately the same with a small advantage to those with the lower flow capacity.

In summary, the performance factors used are (1.0 best 0.9 worst).

Accuracy		
total flow metered (5% max. error)	.95	(normalized to .97)
individual flows metered (root sum square)	.975	(normalized to 1.0)
Stability		
no interaction	1.00	
possible interaction	.90	
Metering response time		
1/2 flow metered	1.0	
total flow metered	.95	
variable speed drive	.90	
Manifold fill time		
flow divider	1.0	
single package	.95	
variable speed drive	.90	

System Ratings on Performance

Configuration	1	2	3	4	5	6
Accuracy	.97	1.0	1.0	1.0	1.0	1.0
Stability	1.0	1.0	.9	1.0	1.0	1.0
Response Time	.95	1.0	1.0	.9	.9	1.0
Manifold Fill	1.0	.95	.95	.9	.95	.95
Raw Score	.92	.95	.86	.81	.86	.95
Normalized Score	.97	1.0	.90	.85	.90	1.0

Emergency Capability -- Operational State for a Single Failure

An additional capability exists with the dual pumping (metering concept that should be considered in the study. It was shown in Attachment I that a 50% reduction in the maximum fuel flow does not affect the capability of the 680-B1 to produce full thrust throughout the mission profile (except at take-off). Therefore, the engine can satisfactorily complete a mission on one nozzle set (primary or secondary). The effect of different failures were assessed as shown in Table II. Failures such as electrical connectors, inducer, and shut-off valves were not considered since they have the same effect on all the systems. That leaves the following two categories (1) pumping system, (2) metering system. The following ratings were used:

Rating	Pumping System	Metering System
1.0	Can pump 50% flow for any single failure	Can meter 50% flow for any failure
.9	Same as above except for air starts	Same as above except cannot work if failure in primary while main was off
.6	Cannot pump 50% flow	Cannot meter 50% flow

Applying the above factors to the 6 configurations yields:

Configuration	Pumping	Metering	Total
1	.6	.6	1.2
2	.9	.9	1.8
3	.6	1.0	1.6
4	1.0	1.0	2.0
5	.9	.9	1.8
6	1.0	1.0	2.0

The above totals were normalized by dividing by two to yield the factors found in Table II.

FAILURE ANALYSIS CHART

Configuration	Fuel Pump Failure	Metering Valve Failure
1	Cannot operate	Cannot operate if total flow metering valve fails. Can operate up to 100% power if flow divider metering valve fails
2	Can operate up to 50% power on primary system if main system pump fails.	Can operate up to 50% power on primary system if main system metering valve fails.
	Can operate 50% power on main system if primary system pump fails while both systems are operating.	Can operate up to 50% power on main system if primary metering valve fails while both are operating.
3	Cannot Operate	Can operate up to 50% power on either primary or main fuel system.
4	Can operate up to 50% power on either primary or main system.	Can operate up to 50% power on either primary or main fuel system.
5	Can operate up to 50% power on primary system if main system pump fails.	Can operate up to 50% power on primary system if main system metering valve fails.
	Can operate up to 50% power on main system if primary system pump fails while both systems are operating.	Can operate up to 50% power on main system if primary system metering valve fails while both are operating.
6	Can operate up to 50% power on either primary or main system.	Can operate up to 50% power on either primary or main system.