FLUIDIC CONTROL OF HYDROSTATIC TRACTOR TRANSMISSIONS
TO ACHIEVE MAXIMUM ENGINE POWER OUTPUT

by

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INTRODUCTION

The hydrostatic transmission tractor, a recent innovation in the agricultural industry, provides versatility unknown in the past. The operator is presented an infinite selection of transmission ratios; therefore, he may choose the speed and power best suited to the field load. Power-take-off operations such as mowing, raking, baling, and forage chopping fully utilize the infinite speed range. Precise speed control is advantageous in planting, cultivating, and field spraying. Ease of reversing the hydrostatic drive tractor is a favorable feature for loading operations.

The versatility of the hydrostatic transmission is not gained without cost. Its efficiency is presently lower than that of a gear type transmission. While rapid improvement of the efficiency has been made in the past several years, it appears that the hydrostatic transmission will never excell on the basis of efficiency alone. However, an opportunity for increasing overall operation efficiency by utilizing the characteristics of the hydrostatic transmission does appear to exist.

Wheel and crawler-type tractors perform most efficiently when their engines are operated at rated load. Varying field load conditions have, in the past, made continual operation at rated engine load impossible. The hydrostatic transmission presents an infinite selection of transmission ratios to the operator; therefore, it allows the possibility of maintaining rated load at all times.

The previous argument is true only for loads demanding full engine power; however, heavy tillage work does meet this condition. The widescale use of heavy tillage and the increasing application of the hydrostatic transmission justify further investigation of methods for maintaining rated power output.

An automatic control system controlling the transmission ratio could provide these advantages:

1. Decrease fuel consumption per acre.

2. Relieve operator of adjusting transmission ratio.

3. Increase work per hour output.

4. Increase tractor life by preventing overload.

Relieving the operator from the tedious task of continually adjusting the transmission ratio to keep the engine at rated power would be one of the more important features of the automatic control system. Many operators would probably tire of this manual chore and set the transmission ratio at a point allowing operation without change.

The investigation was limited to fluidic control devices. Although it was not intended to imply that fluidic control would provide the best solution to this problem their use does seem especially attractive for the following reasons:

1. Fluidic elements are rugged and insensitive to vibration.

2. A fluid, hydraulic system oil, is readily available as a source of energy for the fluidic device.

3. Fluidic elements can perform both digital and
   analog functions including proportional ampli-
   fication, logic decision making, measurement,
   comparison, and counting.

4. Possible simplicity of controls.

5. Cost could be quite low for mass production.

It was intended that this investigation also suggest other
fluidic control applications in the agricultural industry.

OBJECTIVES

Design, construction, and test of an automatic control
system for maintaining full load on a hydrostatic drive
tractor engine were the objectives of this research. The trans-
mission ratio was to be controlled to compensate for changing
field loads, thus allowing the engine to continually operate
at maximum power.

Fluidic devices were to be used in the control system
where applicable.

Proportional rate of change of the transmission ratio was
desired. Thus, a large error would result in an initially rapid
rate of correction, and a smaller error would initiate a less
rapid rate of correction.

The system should have as small a steady state error as
possible to demonstrate the feasibility of the control. Steady
state error could easily be increased in the design if desirable
from a stability standpoint.

Safety features were not considered in this investigation, but would obviously need to be incorporated in a practical working model. The control system would automatically position the transmission ratio for maximum output speed if sufficient load is not available to prevent operation at full speed. Raising a tillage tool from the ground would cause the control system to rapidly increase the tractor speed in attempting to keep the engine at rated load. A safety device to disengage the automatic control for this and other unsafe conditions would be necessary.

The hydraulic supply of a tractor would be a convenient source of energy to operate the fluidic control system. However, fluidic devices do not presently operate as predictably on oil as on air. This study was restricted to operation on air, but with the expectation that the control system developed will become more practical from an economic standpoint when fluidic devices operating on oil are developed.

The relationship of design parameters was not initially obvious. Therefore, the design included provisions for varying the parameters during tests to establish as near optimum control as possible.

Simplicity appeared to be an important factor in the ultimate feasibility of the system; therefore, complex designs were avoided.

REVIEW OF LITERATURE

Research reports on gear transmission tractors published
before the hydrostatic drive tractor was developed gave strong
arguments supporting the hydrostatic transmission. Weber (1951)
stated that a gear transmission may have very good mechanical
efficiency, but be ineffective in providing the proper ground
speed for the most efficient use of fuel, labor, and equipment.
These factors may have a greater effect on overall operating
efficiency than the mechanical efficiency of the transmission.
Various ground speeds are desired for different operations
and conditions; therefore, a transmission should provide the
desired ground speed with the required drawbar pull to utilize
the full horsepower of the engine. A graphical method was
developed to show the range of speeds and drawbar pulls which
a given gear transmission tractor could provide.

Weber (1951) also reported that as the choice of operating
speeds is enlarged, significant fuel savings can be achieved
by operating at lower engine speed during part load operations.
He concluded that a tremendous amount of study would be
necessary to determine the most efficient speed for each oper-
ation, but that investigation would undoubtedly prove an in-
finite selection of ground speed desirable. Thus, a strong
argument was established for the infinite speed capability of
the hydrostatic transmission.

Ricketts and Weber (1961) sought to provide information on the percentage of time for which maximum engine power was used by a farm tractor. Farm records and one test tractor were used to establish the operations on central Illinois farms over a period of one year. The summary covered 8,500 hours of tractor operation.

It was concluded that 54.7 percent of maximum tractor horsepower was the average load; however, 600 of 1000 gallons of fuel was consumed at the higher loads. The study does not have direct application for determining the percentage of time at which full load field operations occur in central Illinois. The inability to select a gear ratio to load the engine to maximum power resulted in a lower percent of full power readings for operations obviously capable of providing full load. For example, during a plowing operation in which the engine was overloaded (developing less than maximum horsepower but more torque than at maximum horsepower) the percentage of full power was recorded as 81.5. To conclude that the available load was only 81.5 percent of the maximum which the tractor was capable of pulling is obviously incorrect. Proper matching of implement to tractor would also increase the percentage of time at which the tractor could be operated at maximum power.

If correctly controlled the hydrostatic transmission offers the possibility of increasing overall fuel efficiency by allowing part throttle operation at any ground speed with a less than maximum load. Huber and Lamp (1960) reported information concerning

tractor engine efficiencies at various engine speeds and loads.
They discovered that best efficiency at full throttle governed
speed occured at 90 to 100 percent of maximum power. However,
for part load operations it was possible to obtain efficiencies
equal to that obtained at maximum power for loads down to one-
half of maximum power by reducing engine speed.

Huber and Lamp (1960) concluded that increasing horsepower
at constant engine speed results in increasing torque. Mechan-
ical efficiency is improved because a smaller percentage of
the total power is lost through friction. Efficiency may be
gained for part load operation of all engines by reducing
engine speed because engine friction is decreased. Reduced
intake work and higher compression due to the opening of the
throttle valve as speed is reduced at constant horsepower im-
prove gasoline engine efficiencies. The speed regulation of
present governors is not satisfactory to allow field operation
at reduced engine speeds.

Barger et. al. (1963, p. 180) stated that close regulation
with a single spring flyweight governor is possible only over
a limited speed range. The situation arises because the forces
of the governor spring and governor weights do not increase at
the same rate, and governor equilibrium is reached when the
spring tension is balanced by the centrifugal force of the
rotating weights.

Maximum power was defined by Barger et. al. (1963, p. 402)
as the power developed by an engine at rated speed with the
throttle fully open.

Typical engine performance curves by Barger et. al. (1963, p. 432-434) showed that specific fuel consumption of a Caterpillar D-4 diesel engine decreased as power output was increased. Testing of six tractors, three gasoline and three diesel, in the Nebraska Tractor Tests, 1959-1960, showed that all six had lowest specific fuel consumption at maximum power torque. Some engines showed greater rates of decrease in specific fuel consumption as full load was approached than did others.

Performance curves provided by Cupit (1967) for one gasoline and one diesel engine varied slightly from the general trend. A 135 in.^{3} displacement, four cylinder gasoline engine rated at 3000 RPM showed a very slight increase in specific fuel consumption as engine speed increased beyond 2500 RPM at full load. Part load tests for the same engine operated at full throttle did show a very significant decrease in specific fuel consumption as maximum power operation was approached. A naturally asperated 239 in.^{3} displacement, four cylinder diesel engine rated at 2200 RPM showed nearly constant specific fuel consumption between 1200 and 2200 RPM at full load. However, part load tests of the diesel operated at full throttle also showed a marked decrease in specific fuel consumption as maximum power operation was approached.

Barger et. al. (1963, p. 388) presented a typical performance curve for a variable displacement pump-fixed displacement motor hydrostatic transmission. Overall efficiency was

approximately 80 percent at maximum pump displacement, 70
percent at one-half maximum displacement, and 55 percent at
one-fourth maximum displacement. Operation at maximum dis-
placement gave the best overall efficiency. Output torque
remained nearly constant for the system; therefore, output
power increased almost linearly with pump displacement.

A typical piston pump overall efficiency curve for various
operating speeds was shown by Lewis and Stern (1962, p. 129).
Pump efficiency remained nearly constant from 50 to 100 percent
of rated maximum speed, but decreased rapidly below 30 percent
of rated maximum speed. Efficiency was reduced by increasing
the output pressure for speeds less than 50 percent of rated
maximum, but was not significantly affected for greater speeds.

International Harvester (1967) developed the first com-
mercially available hydrostatic drive agricultural tractor in
the United States. A variable displacement pump-variable dis-
placement motor transmission is used. A two speed gear trans-
mission in the final drive allows the pump and the motor to
both operate at near maximum efficiency under field load and
road speed conditions.

The pump and motor displacements are controlled by separate
hydraulic servo cylinders which vary the swashplate angles.
Variable orifice openings in the cylinders controlled by the
transmission ratio level determine the swashplate positions.
The transmission ratio level is manually controlled.

Fluidonics (1966) stated that during the extremely short history of fluidic technology a wide variety of fluidic devices and circuit components have been developed. The technology, it is generally agreed, dates from public announcements made by Harry Diamond Laboratories in 1960. Two types of fluidic devices, digital and analog (or proportional), were discussed. It was concluded that applications of the proportional amplifiers were not as widespread as those of digital fluidic amplifiers. However, it was recognized that potential applications for the proportional devices were numerous and increasing.

Letham (1966) outlined the application areas for various types of fluidic elements. The proportional, stream-interaction amplifier was deemed suitable for control or power applications. The exact level at which the device becomes a power element was somewhat arbitrary. Graphic symbols for fluidic devices were shown in the article.

Staging proportional fluidic amplifiers to achieve high gains was discussed by Gesell (1966), a Senior Project Engineer for the Corning Glass Works. Many problems were encountered when more than three units were staged. A great deal of care was necessary to eliminate noise from the circuit. Using tubing connections and more than three stages of amplification caused the output to have a high amount of noise. Integrated circuits may improve the signal-to-noise ratio.

General Electric (1968) produces a five stage integrated circuit proportional amplifier which has a signal-to-noise ratio specification of 200:1 for 0 to 25 Hz. Forward gain of 1500 resulting in an output pressure differential of $\pm$ 15 PSI with 30 PSIG supply is reported.

Johnson (1967) described a Honeywell system for controlling jet engines. Proportional speed control of the engine was obtained by measuring engine speed with a boundary layer device. Low pressure air was supplied to a chamber closely surrounding a smooth disk which rotated at engine speed. The pressure differential between two probes correctly placed in the chamber was very nearly proportional to engine speed from 0 to 16,000 RPM. Proportional fluidic devices amplified the signal.

A hybrid fluidic system, one containing moving parts, was described by Letham (1967a). A vibrating reed frequency sensor which produced an output pressure only when driven at near resonant frequency was described. A wobble-plate signal generator produced the driving force. When the wobble plate was driven at the resonant frequency of the reed the driving force excited the reed and caused a pressure output. A variation of the same system using two reeds tuned to slightly different frequencies was also described. Accurate speed control using this system was possible for a limited range. However, if speed dropped below this range only a small error output signal was produced.

Letham (1967b) described a speed control system in which Helmholtz resonators replaced the vibrating reed frequency sensors. The Helmholtz resonator consisted of a fluid inductance in series with a fluid capacitance. Performance of the system was similar to that of the vibrating reed circuit. One disadvantage noted was the temperature dependency of the Helmholtz resonator.

Gesell (1966) stated that proportional devices were operated on gases from 0.1 PSIG up to 200 PSIG with only minor performance differences. Operation on liquids was somewhat different and venting became a problem. However, devices were used with water or similar liquids as the operating fluid.

An interesting application of a fluidic device operating on liquid was described by Binder (1967). A fluidic carburetor was tested which used air to control the fuel flow, thus resulting in a mixed gas and liquid application.

THE DESIGN

The system criteria outlined in the objectives greatly narrowed the types of controls applicable. The proportional control requirement eliminated consideration of digital control systems. The desire to utilize fluidic controls operating on air necessitated the use of some engine power-to-air pressure transducer.

A U-4 International engine operating on propane was available to drive the hydrostatic transmission. The framework designed to hold the engine and transmission is shown in Plate I.

The hydrostatic transmission pump was a variable displacement Sundstrand axial piston unit with 3.2 in.^{3}/rev. maximum displacement. A mechanical linkage varied the swashplate angle. Displacement increased from zero to maximum as the swashplate angle was varied from zero to 16°. The pump was an experimental model rated at 2500 PSIG and 2500 RPM.

The 3.2 in.^{3}/rev. fixed displacement piston-type motor was also manufactured by Sundstrand. It was rated at 1150 in.-lb. torque for 2500 PSIG operating pressure. Since the pump and motor were both experimental units[1], they had no model numbers.

Five control components connected in series: the air pump, the fluidic amplifier, the air cylinder, the hydraulic valve, and the hydraulic cylinder comprised the control system. The components are shown in Fig. 1 and Plate II.

The Air Pump

With the engine hand throttle set in the fully open position a given engine speed corresponded to a certain load. Measurement of the engine speed gave a representation of the

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[1]John W. Pinkerton. Sept. 13, 1967. Personal communication. Hydro-Transmission Division of Sundstrand Corp. LaSalle, Illinois.

EXPLANATION OF PLATE I

Overall view of the power train, control system, and instrumentation.
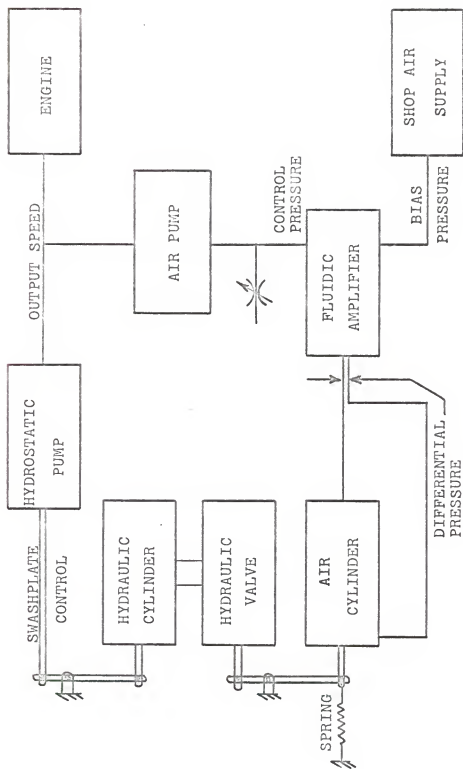
PLATE I

FIG. 1. A schematic diagram of the control system.

EXPLANATION OF PLATE II

The control components:

1. Air pump

2. Fluidic amplifier

3. Air cylinder

4. Hydraulic valve

5. Hydraulic cylinder

6. Pump swashplate linkage

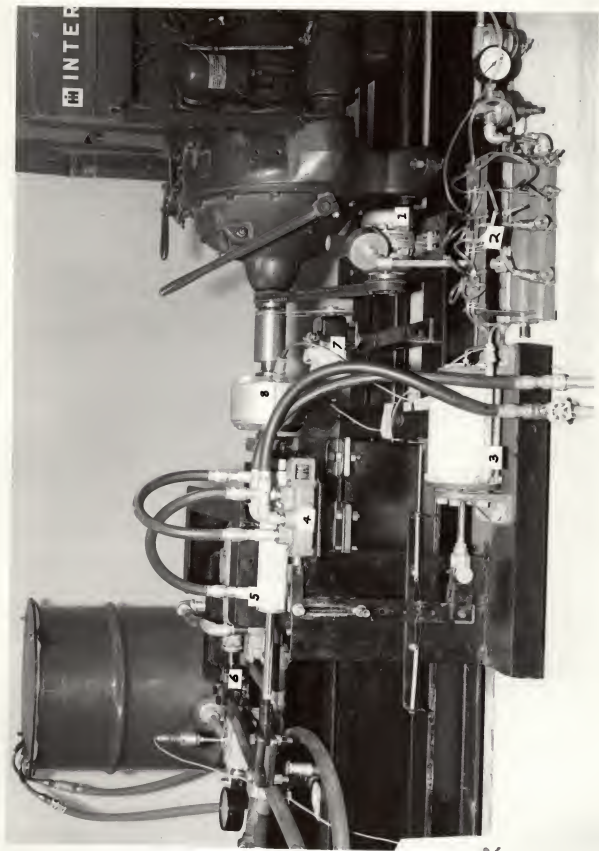7. DC tachometer

8. Torquemeter

PLATE II

load. An air pump driven at engine speed was selected as the engine power-to-air pressure transducer.

A model 0240 Gast, vane-type air compressor rated at 1.9 CFM @ 0.0 PSIG discharge pressure and 1725 RPM was specified as the transducer. Its output was piped, through a paper element filter to remove dust from the carbon pump vanes, into a 21 in.^{3} reservoir. A variable orifice in the reservoir allowed the air to escape to the atmosphere. The control pressure generated in the reservoir was transmitted to the fluidic amplifier.

For low output pressures in the operating speed range it was assumed that the air pump output flow was proportional to engine speed. The reservoir pressure which resulted from this flow passing through the reservoir and the fluidic amplifier orifice was given by the orifice equation:

$$q = C_d \, A \sqrt{\frac{2P}{\rho}}$$

$$P = \frac{\rho}{2 \, C_d^2 \, A^2} \, q^2$$

Where q – rate of flow

C_d – orifice discharge coefficient

A – area of orifice

P – reservoir pressure, (PSIG)

ρ – density of air

Considering C_d, A, and ρ constants, the reservoir pressure varied with the square of the flow. Thus, the air pump reservoir pressure also varies with the square of the engine speed.

The Fluidic Amplifier

Three Corning, load insensitive, proportional fluidic devices (model 190879) were cascaded as shown in Fig. 2 to achieve the necessary amplification of the air signal generated by the air pump. A shop air supply filtered through a 20 micron filter was used to operate the amplifier. Needle valves on the shop air manifold were used to control the supply pressures at various points in the fluidic amplifier. Pressure in the shop air manifold was maintained constant by a pressure regulator.

One control nozzle in the first stage fluidic device was connected to the air pump reservoir, and the other control nozzle was connected to a needle valve on the shop air manifold. Opening the needle valve produced a bias pressure. Adjusting the bias pressure to equal the air pump reservoir pressure with the engine operating at rated speed produced equal pressures at the first stage output ports. Changing the engine speed altered the air pump reservoir pressure, and a pressure differential across the control nozzles of the first stage was produced. The first stage amplification increased the differential, and the output was used as the control for the second
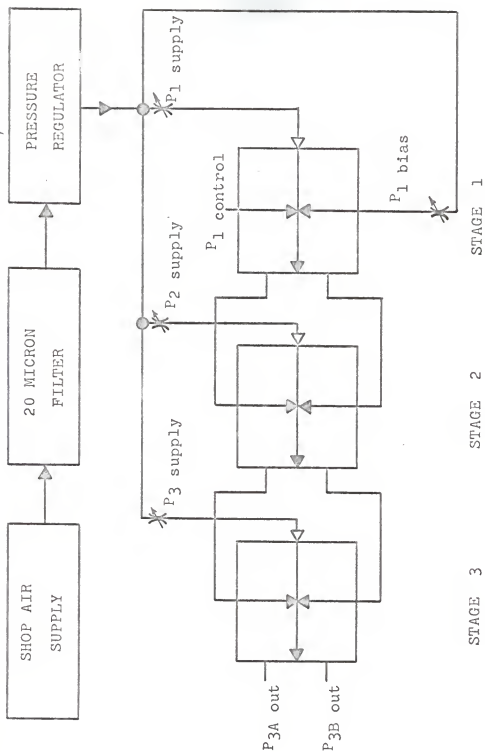
FIG. 2. Schematic diagram of the fluidic amplifier.

stage of the amplifier. The final output was achieved by
using the output of the second stage as the control for the
third and final stage.

The amplifier output was initially predicted using Corning's
operation specifications[1] for the individual fluidic devices.
Pressure recovery was assumed to be 55 percent of supply pres-
sure. Maximum control pressure was assumed as 10 percent of
supply pressure. The third stage supply pressure was arbitrari-
ly assumed to be 40 PSIG.

Assuming 40 PSIG as the supply pressure of the third stage,
required the output of the second stage to be 4.0 PSIG. The
second stage supply was 7.27 PSIG since the maximum output was
55 percent of the input. To provide a control pressure 10
percent of the second stage supply, a 0.73 PSIG output was
necessary from the first stage. At 55 percent recovery a 1.32
PSIG first stage supply pressure was required. The air pump
reservoir pressure at rated engine speed was 0.132 PSIG to
produce the desired control pressure in the first stage.

Each amplifier stage produced a gain of 6.0 at the bias and
control levels assumed. Total amplification was 216 for the
three stages. Assuming an operation point of 1800 RPM the am-
plifier output change for a 100 RPM increase was given by:

$(0.132)(\frac{1900}{1800})^2 = 0.147$ PSIG(Pump reservoir pressure @ 1900 RPM)

$0.147 - 0.132 = 0.015$ PSI(First stage control differential)

$(0.015)(216) = 3.26$ PSI(Third stage output differential)

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[1] Fluidics Department of Corning Glass Works. Center Dump Pro-
portional Device (catalog no. 190879) specification sheet.Jan. 1967.

Thus a signal of 3.26 PSI was available to activate the controls necessary to return the engine to the operating point. Gain reductions due to the loading effects of one stage upon another were not considered.

The Air Cylinder

The pressure differential generated at the output of the third stage of the fluidic amplifier was applied across the piston of an air cylinder. A double-acting, rolling-diaphragm cylinder (Bellowfram model D-16-F-BP-UM) was selected for the application because of its low static and dynamic friction coefficients. The effective piston area of the cylinder was 16 in.^{2}, and the stroke was 3.7 inches.

The cylinder rod was externally spring loaded to position the piston in the center of its stroke when no pressure differential existed across the piston. A pressure differential caused the piston to move to a position where the spring force was equal and opposite to the force which the pressure differential exerted on the piston. Thus, a displacement proportional to the differential pressure was obtained.

The Hydraulic Valve

A four-way, three-position control valve (Cessna model 30501-2C) was connected to the air cylinder by a lever. The valve was in the neutral position (work ports blocked) when

the air cylinder piston was centered. The lever controlling the valve had a moveable pivot point to allow the valve spool position for a given air cylinder piston displacement to be changed.

The valve was rated for 0-15 GPM flow at 2500 PSIG. The flow required from the valve in this application was small in comparison to the rated flow. The valve was designed for manual operation on industrial and agricultural machinery.

A pressure-compensated, variable-volume Continental power unit supplied oil to the valve. A needle valve was installed in the supply line to allow the rate of oil flow through the valve for a certain spool setting to be controlled. Thus, the velocity of the swashplate servo cylinder which the valve controlled was variable.

The valve spool was overlapped .091 inch. A movement of .091 inch either direction from the centered position was therefore necessary to open the work ports and allow flow to the cylinder. The area open to return flow through the valve increased slowly as the valve spool was displaced from .091 inch to .172 inch from the neutral position due to a metering notch in the spool housing. An annular area in the return passage began to open for spool displacements greater than .172 inch. The supply passage had no metering notches; therefore, an annular area opened at .091 inch spool displacement. Restriction of the return flow was to allow the operator a means of 'feathering' the valve. The valve was symmetrical about the neutral position.

The Hydraulic Cylinder

A double-acting, single-ended hydraulic cylinder (Cessna model MH14245) controlled the pump swashplate position. The cylinder piston diameter was 2.0 inches, and the rod diameter was .684 inch. The cylinder stroke was seven inches.

The lever attaching the cylinder to the pump control was designed to give zero swashplate angle when the cylinder was fully extended and maximum swashplate angle when the cylinder was fully retracted. Reverse flow was not allowed because instrumentation and loading devices functioned for flow in only one direction. Allowing the cylinder to be fully extended or retracted also prevented large forces from occuring in the swashplate linkage.

LABORATORY INVESTIGATION

Field Load Simulation

The pump output pressure in a variable displacement pump-fixed displacement motor hydrostatic transmission is proportional to the torque load on the motor output shaft. For a given tractive efficiency the drawbar pull of a tractor is proportional to the torque on the output shaft of the power train; therefore, the drawbar pull is proportional to the pump output pressure for the above situation.

A dynamometer absorbing power at the motor output shaft could have been used to simulate field loads, but rapidly changing the load on the motor would have been impossible with the equipment available. Since pump output pressure was proportional to drawbar load in the system it was decided to load the pump hydraulically using a Double A (model BT-06-3M-10A2) pilot operated relief valve. Rapid increase of the pump output pressure was possible with the valve. The load valve is shown in Plate III.

Use of a relief valve to load the system caused a large amount of energy to be converted into heat. The small charge pump in the variable displacement Sundstrand pump did not add enough oil to the closed loop circuit to prevent overheating. An external charge pump, shown in Plate IV, with a 5.5 GPM capacity was placed in the system to pump cooler reservoir oil into the closed loop. The warmer oil was returned to the reservoir where cool water circulating through 50 feet of 3/8 inch copper tubing cooled the oil. A needle valve was installed in the return line to regulate charge pressure. Although continuous full load operation did overheat the reservoir oil after the external charge pump was added, five minute, full load tests starting with 15 gallons of room temperature oil in the reservoir were possible.

The relief valve was used to test the response of the system to a change in load. The pressure was increased as rapidly

EXPLANATION OF PLATE III

The hydrostatic pump and motor in the closed loop
hydraulic system. The Double A pilot operated load valve
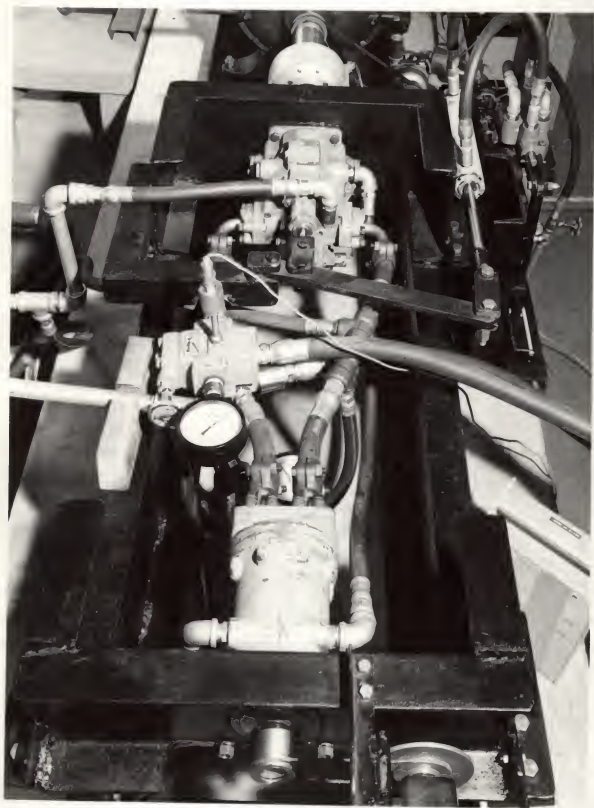is shown with the pressure transducer protruding from it.

PLATE III

EXPLANATION OF PLATE IV

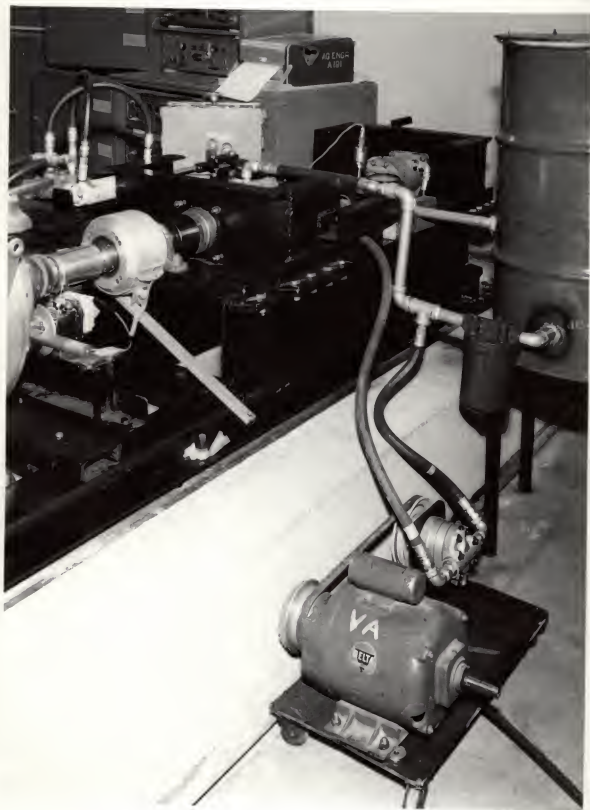The external charge pump and hydraulic fluid reservoir.

PLATE IV

as possible from a steady state value to determine the response
of the system to a sudden increase in drawbar load.

A variable displacement motor-variable displacement pump
hydrostatic system could not be loaded as easily in the above
manner because drawbar load is not proportional to pump output
pressure as the motor displacement is changed.

Instrumentation

The simultaneous recording of four variables: engine speed,
engine torque, pump output pressure, and pump output flow gave
detailed knowledge of the system performance. The variables
were recorded using four single-channel Brush amplifiers and
a four-channel Brush oscillograph. The recording equipment
is shown in Plate I.

A small Wesston dc generator, number 7 in Plate II, was
used to record the engine speed. The output voltage of the
generator was proportional to the engine speed. The Brush
amplifier connected to the output of the generator amplified
the dc signal, and the oscillograph recorded the speed output
on the first channel. The amplifier was calibrated so that
the oscillograph pen was centered on the first chart strip when
the engine speed was 1800 RPM, and each small line of deflection
corresponded to 25 RPM.

Considerable noise from the generator brushes was initially
recorded by the oscillograph. An R-C filtering network with a
69KΩ resistor in series and a 4.0 MFD capacitor in parallel

eliminated the noise without slowing the response to RPM changes appreciably. The output of the oscillograph may be seen on Plate V.

A strain gage torquemeter[1] was incorporated into the power train to measure engine output torque. The torquemeter is number 8 in Plate II. The torquemeter and its Brush carrier amplifier were calibrated using a Tinius Olson torque machine. The pen deflection was calibrated to be one small line for 4.0 ft.-lb. of torque. The torque machine also verified that the Brush amplifier could be recalibrated quite accurately using the internal calibration resistance.

The torquemeter heated more than would be expected in normal operation. It was suspected that the bearings were not operating as freely as would be desirable. The heat differentials produced in the torquemeter caused zero shift in the output. For this reason the torque values shown on the second channel in Plate V are not correct in magnitude; however, a qualitative analysis of the torque is possible.

The pump output pressure was measured using a Stratham (model MC3) universal transducing cell with a 2000 PSIG diaphragm accessory. A Stratham analog readout (model MR5) provided the excitation for the strain gage transducer, and gave

[1] Carroll K. Reece. Investigation and Design of a Torquemeter Using SR-4 Strain Gages. Unpublished M.S.Thesis, Kansas State University Library. Manhattan, Kansas. 1957.

EXPLANATION OF PLATE V

System response parameters recorded by the four
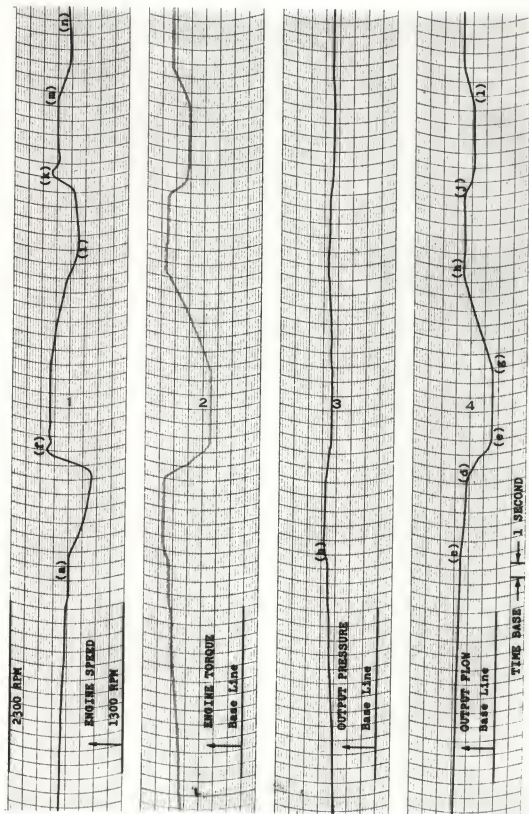channel Brush oscillograph.

PLATE V

dc output which was used as an input to the third Brush
amplifier. The transducer may be seen protruding from the
top of the load valve in Plate III. The instrumentation
cable to the analog readout can be seen more clearly in Plate
I.

The Brush amplifier was calibrated to produce one line
deflection on the third channel of the oscillograph for 100
PSIG pump output. Plate V illustrates the output of the pres-
sure recording equipment.

The output RPM of a fixed displacement motor should be
very nearly proportional to its input flow. Volumetric ef-
ficiency should be high, and leakage should be low for low input
pressures. Therefore, the 3.2 in.^{3}/rev. fixed displacement
motor was used to measure the pump output flow in this appli-
cation.

The motor RPM was measured using a dc tachometer and
filter network identical to the engine speed recorder. The
fourth Brush amplifier was calibrated to produce one line de-
flection on the oscillograph for 100 RPM motor speed. The
RPM output is shown on the fourth channel in Plate V. Pump
output flow was calculated as follows:

$$\text{RPM} \times 3.2 \frac{\text{in.}^3}{\text{rev.}} \times \frac{1 \text{ gal.}}{231 \text{ in.}^3} = \text{Flow in GPM}$$

An Owatonna Flo-Rater Hydraulic Tester was also incor-
porated into the closed loop hydraulic circuit downstream from
the load value. Visual monitoring of oil flow and temperature
was possible using this device. The needle valve in the
tester was kept fully open during all tests. The tester was
located on a stand near the instrument cabinet as shown in
Plate I.

Component Performance

The performance characteristics of the individual control
components were examined before the overall system response
was evaluated. One poorly matched component could have given
the system undesirable response characteristics; therefore,
more emphasis was placed on component evaluation than on sys-
tem response.

Use of the air pump as an engine power-to-air pressure
transducer proved quite satisfactory. The output character-
istics of the pump as it was operated during the tests are
shown in Fig. 3. The characteristic curve obtained by closing
the orifice in the pump reservoir to give a higher output
pressure, 6.0 PSIG at 2000 RPM, was nearly identical in form
to the curve in Fig. 3. Therefore, it was evident that the
pump was satisfactory as a transducer for a wide range of out-
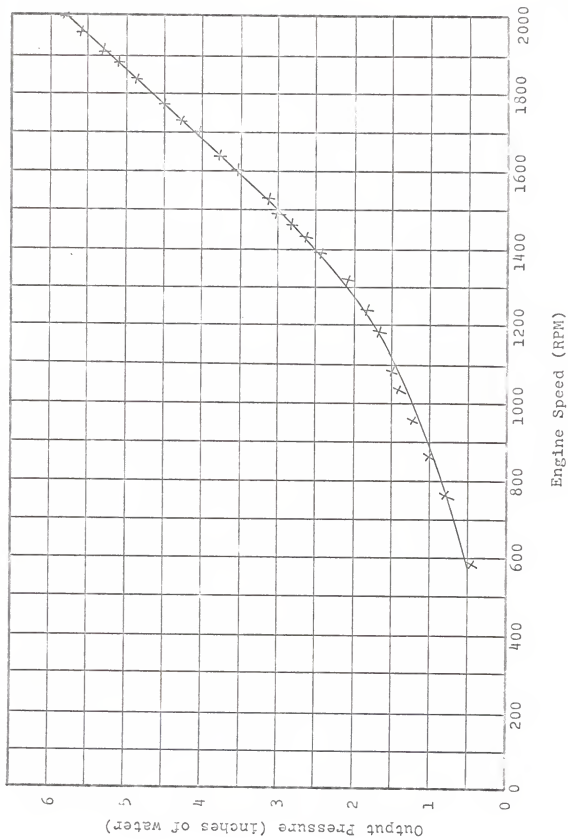put pressures.

FIG. 3. Output characteristics of the Gast air pump.

The air pump output pressure did not vary directly with the square of the engine speed as had been calculated. This was probably due to air compressibility and variations in the orifice discharge coefficient. The extent to which decreases in the volumetric efficiency of the pump resulting from increases in output pressure affected the output flow was not determined.

Characteristics of the fluidic amplifier as operated during the tests are shown in Fig. 4. The conditions were arrived at by initially setting $P_{3\ supply}$ at 40 PSIG; $P_{1\ control}$ at 0.0 PSIG; and adjusting $P_{1\ bias}$, $P_{1\ supply}$, and $P_{2\ supply}$ by trial and error to yield maximum deadhead pressure differential at the third stage output. Reference to Fig. 2 will clarify the location at which the pressures were measured.

The attempt to predict operation pressures during the design of the fluidic amplifier using Corning's specifications was not entirely successful because the loading of one stage upon another was not considered. Specifications allowing the load to be evaluated would be necessary to make accurate predictions.

After the amplifier operation conditions had been established experimentally the air pump reservoir pressure, $P_{1\ control}$, for rated engine speed of 1800 RPM was set equal to $P_{1\ bias}$ by varying the needle valve in the air pump reservoir. Thus, at 1800 RPM engine speed no input pressure differential existed at the first stage of the amplifier.
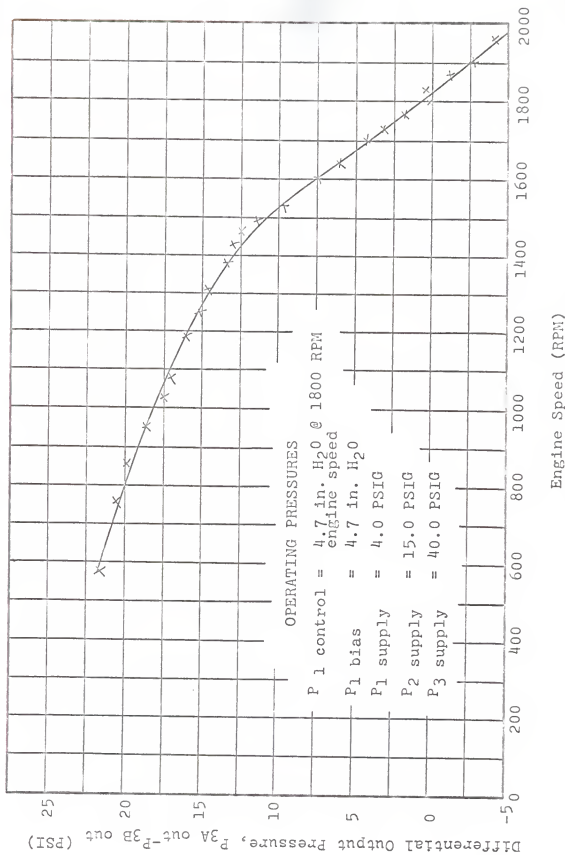
FIG. 4. Output characteristics of the fluidic amplifier. Supply and bias pressures were held constant; P_1 control varied with engine speed as shown in FIG. 3.

Figures 3 and 4 are related by the conditions described above.
Figure 4 shows a slight pressure differential at the amplifier
output for 1800 RPM engine speed. Difficulty in adjusting the
needle valves to exactly the correct position caused this out-
put differential.

The gain of the fluidic amplifier was calculated using
values from Figures 3 and 4. Between 1600 and 1800 RPM the
gain was as follows:

$$\text{Gain} = \frac{\Delta P_o}{\Delta P_i} = \frac{\Delta \left(P_{3A \ out} - P_{3B \ out}\right)}{\Delta \left(P_{1 \ control} - P_{1 \ bias}\right)}$$

$$= \frac{6.8 \ PSI}{1.14 \ in.H_2O} \times \frac{27.7 \ in.H_2O}{1 \ PSI} = 165$$

The gain of 6.0 per stage or 216 overall obtained from Corning's
specifications proved to be somewhat high for this application.

The fluidic amplifier showed signs of instability when
operated either into the air cylinder or at deadload with
P_3 supply at 40 PSI. Decreasing P_3 supply increased stability,
but the output differential pressure available to operate the
air cylinder also decreased. No other major unfavorable ampli-
fier characteristic was discovered.

The air cylinder characteristics were adequate for low
pressure operation. The rolling diaphragm seal produced little
static and dynamic friction load. The piston did tend to have
a somewhat intermittant motion during operation. It was

suspected that the rod seal friction caused the rod to stick. The problem was not severe enough to affect operation of the control system; however, it could possibly have been reduced by installing a loose fit rod seal. A small air leakage around the rod would not adversely affect operation.

The hydraulic control valve produced a time delay in the control system, the time which was necessary for the valve spool to be displaced beyond the overlapped position. However, some time delay was desired to prevent response to short peak over-loads.

A drawback to the application of this valve in the con-trol system was that output flow did not increase in direct proportion to the spool displacement. The valve was designed for manual operation in which proportional flow was of small concern.

One major objection to the hydraulic cylinder was dis-covered. The effective piston area was greater on one side than on the other due to the single end piston rod. As a result the piston extended more rapidly than it retracted for given hydraulic valve spool settings equidistant from the centered spool position because a greater volume of oil was forced through the return passage restriction when the piston was retracted. A hydraulic cylinder with piston rods on each side of the piston would have eliminated this problem.

System Response

Several modifications of the system immediately appeared
to be desirable; therefore, extensive system response testing
was not undertaken. The initial tests are described to indi-
cate the weaknesses encountered and to indicate the performance
of the system. The test equipment was also evaluated.

The response of the control system to an increase in field
load is shown in Plate V. The pump output pressure was in-
creased gradually with the pump at maximum displacement.
Engine speed reached 1800 RPM, the desired operating speed at
point (a) on Plate V. A rapid increase in field load was then
simulated by increasing the pump output pressure (b) by
manually adjusting the pilot operated load valve.

The increased load immediately caused the engine speed
to decrease; therefore, the pump output flow also decreased
from (c) to (d). At point (d) the control system responded to
the drop in engine speed by reducing pump outflow. Note the
rapid decrease in flow between (d) and (e). The flow decrease
was more than sufficient to allow the engine speed to return
to 1800 RPM (f). The control system responded to the overspeed
condition by increasing pump output flow between (g) and (h).
Overshoot of the desired operating speed was again evident as
the engine speed fell to (i). The system response causing in-
creased flow at (j) resulted in engine speed overshoot at (k).

A final flow correction at (1) resulted in the engine speed being corrected from (m) to the desired operating speed at (n).

Repeated tests verified that the control system steady state error was approximately ± 40 RPM about the desired operating speed. The final correction on Plate V was more precise than average.

The system overshoot evident in Plate V could have been reduced by closing the orifice in the hydraulic valve supply line a small amount. The swashplate control cylinder rod would then have moved less rapidly, and the effect of the time delay in the control system response would have been reduced. The approach was impracticable because system response would have been slowed considerably. The difference in rate of change of reducing and increasing flow due to the single rod hydraulic cylinder compounded the problem.

Two problems arose from using the output of the fluidic amplifier directly to drive the air cylinder. A time delay, probably greater than desirable, resulted when the components were combined into a control system. Overlap in the hydraulic control valve necessitated a significant movement of the air cylinder to operate the hydraulic cylinder. Air displaced from the air cylinder was forced through a small orifice, the output nozzle of the third stage in the amplifier. Slow movement of the air cylinder rod resulted.

The second problem arose when attempts were made to compensate for the first problem. The response shown in Plate V

was with the air cylinder--hydraulic valve linkage adjusted to
give maximum valve spool displacement at full stroke of the air
cylinder. Moving the lever pivot point nearer the air cylinder
reduced the cylinder stroke necessary to open the valve to flow,
thus time delay in the response was seemingly reduced. However,
an instability in the fluidic amplifier caused the air cylinder
to oscillate about its neutral position and prevented success.
Apparently air returning to the amplifier from the air cylinder
interacted with the controlled air stream in the third stage
and caused an instability at low differential output pressures.
The oscillation magnitude was not great enough to interfere
significantly with the system's operation until the pivot was
moved nearer the air cylinder.

Although proportional response should have been possible
with this design the hydraulic control valve largely restricted
response to an on-off form. The response to a large speed
error was not greatly different than to a smaller error. The
change in valve spool displacement which allowed flow through
the valve to increase from zero to near maximum was small in
comparison with the overlap. A proportional valve with small
overlap would be necessary to correct these conditions.

The optimum spring rate for proportional operation was
not determined due to the system response taking an on-off form.
For a valve with small overlap the spring rate could be esti-
mated closely using the amplifier output curve, piston area,
and engine governor characteristics.

The Double A, pilot operated, load valve did not maintain the pump output pressure as constantly as expected as output flow varied. Plate V also shows that the pressures at point (b) and at the final operating condition, far right portion of the chart strip, varied for the same valve setting and flow.

DISCUSSION

The results were generalized to apply to other engines. Since the control system sensed engine speed rather than engine power output directly, examination of the full throttle speed regulation curve of any engine reveals how closely that engine could be controlled to maximum power output. The engine governor characteristics determine the steady state operation horsepower error of the control system. The overall closed loop system response would change somewhat if the system controlled another engine because of different engine dynamic response characteristics.

A rated engine speed of 1800 RPM was assumed for the test engine. The manufacturer's rated speed no longer applied due to modifications which had been made on the engine. As discussed above, the selection of rated speed was relatively unimportant since the results were generalized. Observing steady state speed error was a more general method of expressing performance than measuring how closely maximum power was maintained for a particular engine.

Examination of the full throttle speed regulation curve supplied by Cupit (1967) showed the steady state maximum power error for a four cylinder gasoline engine. The curve showed that the power output of that particular engine would have been controlled between 50 and 55 HP if the engine output speed at full throttle had been maintained within ± 40 RPM of rated speed.

Intake manifold vacuum pressure is often used as an indicator of gasoline engine output power. Two factors prevented use of this parameter in the control system. They are that most proportional fluidic devices are not designed to be controlled by vacuum pressures and that vacuum pressure would not have been a power indicator for diesel engines. For this reason the system would not have been as universal in application.

The fluidic amplifier output instability noted at dead-load may have been the result of either internally or externally generated noise. The air pump output pressure variations could have been a source of noise; however, noise inherently generated and amplified by the amplifier itself seemed to be the major factor. Observance of the output noise when only the bias pressure was present to control the first stage substantiated the conclusion.

Overall field efficiency may be improved in two ways: by reducing the operation time or by reducing the total fuel required for the operation. Maintaining maximum power output from the tractor engine will minimize the operation time. The

total fuel required will be very near minimum if the engine
is operated at maximum power during the operation. More op-
portunity exists for increasing field efficiency for full
load operations by reducing labor costs than by reducing fuel
costs.

Opportunity for reducing fuel costs during part load
operation does exist. A system to reduce engine speed while
maintaining constant ground speed during part load operations
would decrease specific fuel comsumption. Since more fuel is
consumed at higher loads, and the labor savings promise to be
greater for the system maintaining maximum engine power for
full load operation it was chosen for study. A single system
to maintain maximum efficiency operating conditions at both
full and part load would be desirable.

CONCLUSIONS

The following conclusions were drawn from analysis and
laboratory investigations:

1. A control system for automatically maintaining full
    load on a hydrostatic drive tractor engine is desirable
    from the standpoints of efficient operation and oper-
    ator convenience during full load field operations.

2. The system proposed showed potential for practical
    operation although certain modifications appear
    necessary.

3. Instability in the three stage fluidic amplifier was a significant problem.

4. Prediction of the gain of the fluidic amplifier using Corning's specifications was fairly successful. Calculation of the supply and bias pressures was less successful because the effect of loading of one stage upon another was not evaluated in the specifications.

5. More effective matching of the hydraulic valve and cylinder with other components of the control system would improve the system response.

6. Power requirements for the control system were relatively low.

7. The system could be used to control variable displacement pump-variable displacement motor hydrostatic transmissions by integrating it with controls presently used on that type of transmission.

8. Use of a pilot operated relief valve to load the hydrostatic pump was satisfactory; although a valve with better pressure regulation would have been desirable.

9. The external charge pump satisfactorily replaced hot oil from the closed loop circuit with cooler oil from the reservoir.

Recommendations for Further Study

Much work remains to be done in developing the control
system. Modifications of the present system are necessary to
determine if the time delay can be reduced and system stability
increased. The time delay could be reduced by installing a
hydraulic valve with less overlap and more proportional flow
characteristics, but the stability of the fluidic amplifier at
deadload would need to be increased correspondingly.

Looking at the hydrostatic transmission control problem
in a broader sense suggests the following related areas in
which additional knowledge is needed to outline criterion for
a practical control system:

1. Time and fuel savings possible through maintaining
   full engine load during a field operation.

2. Possibility of reducing fuel consumption by developing
   controls to reduce engine speed for part load operation.

3. Determination of relationships between engine speed
   and hydrostatic pump displacement giving maximum overall
   part load efficiency. Hydrostatic motor displacement
   would be an additional parameter in a variable dis-
   placement pump-variable displacement motor transmission.

4. Interaction of the engine governor and the hydrostatic
   control system for part and full engine loads at various
   hand throttle settings.

5. Use of oil in fluidic devices.

6. Investigation of controls other than fluidic.

As was anticipated, the study uncovered several ideas as to other effective applications of fluidic controls in the agricultural tractor. A system similar to the control developed for adjusting the hydrostatic pump displacement might also be used as an engine governor or draft control unit. Both applications would result in controlling engine speed, either directly or indirectly, as did the hydrostatic pump control system. Development of a fluidic carburetor would also be a challenging and worthwhile project.

ACKNOWLEDGMENTS

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FLUIDIC CONTROL OF HYDROSTATIC TRACTOR TRANSMISSIONS
TO ACHIEVE MAXIMUM ENGINE POWER OUTPUT

by

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AN ABSTRACT OF A MASTER'S THESIS

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ABSTRACT

An infinite range of ground speeds is available to the
operator of the recently introduced hydrostatic transmission
tractor. The transmission ratio which provides the proper
ground speed for the most efficient use of fuel, labor, and
equipment may be selected. Full utilization of the trans-
mission's desirable characteristics will help overcome its
primary weakness--an efficiency less than that of a gear trans-
mission.

The overall field efficiency of the hydrostatic tractor
may be improved by maintaining maximum engine power during
full load operations. Field time and specific fuel consump-
tion would each be minimized.

An automatic control system for maintaining full load on
a hydrostatic drive tractor engine was designed, constructed,
and laboratory tested during the study. Such a system could
relieve the operator of the task of selecting the proper trans-
mission ratio to maintain maximum engine power. Human error
would be eliminated.

Fluidic controls were used in the system where applicable.
The system design was kept as versatile as possible to facili-
tate universal application.

Engine speed at full hand throttle setting was the para-
meter sensed by the control system. An air pump driven at engine
speed produced the control pressure. As engine speed varied
from the rated speed the control pressure varied proportionally.

A three stage proportional fluidic device amplified the differential between the control pressure and a bias pressure to activate the control system. The bias pressure was equal to the control pressure at rated engine speed.

The amplified air signal activated an air cylinder which in turn controlled a hydraulic valve. The valve controlled a servo cylinder which positioned the hydrostatic pump swashplate to keep the engine at rated speed.

Field load was simulated in the laboratory by hydraulically loading the pump of the variable displacement pump-fixed displacement motor transmission. Steady state engine speed error between actual and rated engine speed indicated the accuracy of the system. Modifications to improve system response and stability were suggested.

The results from this investigation can be generalized by examining the full throttle speed regulation curves of other engines. The steady state speed error will indicate how closely other engines can be controlled to maximum power.

With the recommended improvements the system could be developed into a practical field unit.