APTD-1558

TRANSMISSION STUDY FOR TURBINE AND RANKINE CYCLE ENGINES

ENVIRONMENTAL PROTECTION AGENCY
Office of Air and Water Programs
Office of Mobile Source Air Pollution Control
Advanced Automotive Power Systems Development Division
Ann Arbor, Michigan 48105

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Contract No. 68-04-0034

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Prepared For

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December 1972

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ACKNOWLEDGEMENT

The EPA Project Officer was James C. Wood of the NASA-Lewis Research Center. Mr. Wood worked for EPA under a special technical assistance agreement between NASA and EPA.

I. INTRODUCTION – ABSTRACT

A. Introduction

During the study effort, "Hybrid Prpulsion System Transmission Evaluation – Phase I, "sponsored by EPA under Contract 68-04-0034 and performed by Sundstrand Aviation, a transmission configuration without a flywheel was investigated. Although the effort was minimal, it did indicate that a transmission with an infinitely variable ratio offered many advantages for a conventional automotive propulsion system, particularly if the engine had a limited speed range. These conclusions, coupled with the efforts and interest by the Environmental Protection Agency Division of Advanced Automotive Power Systems (EPA/AAPS) in the gas turbine and Rankine cycle engines, made a more detailed evaluation of such a transmission desirable.

A study was, therefore, initiated by EPA/AAPS to quantitatively assess the technical and economic feasibility of existing and potential types of transmissions most suitable for the gas turbine and Rankine cycle engines. Such a study was to be aimed specifically at the AiResearch single shaft gas turbine and the Aerojet Rankine cycle engine with a turbine expander. Both of these engines operate over limited speed ranges, although the approach was applicable and advantageous to other engines including the conventional spark ignition type. The Aerojet engine is now in the development stage while the AiResearch engine is a conceptual design.

The study consisted of analytically evaluating the performance, physical characteristics and cost of candidate transmissions. Although the study was analytical, design criteria, test data, and experience from over 25 years involvement in the design, test and production of transmissions was used in the analysis by Sundstrand.

Sundstrand's Aviation Division provided the program management, design, and analysis effort. Detailed cost estimates of the transmissions were aided by personnel from Sundstrand's other operating groups.

Requirements, scope of work, and other data utilized in and pertinent to the study are included in Appendices I-1 through I-8.

B. Abstract

The study was carried out under contract to the Environmental Protection Agency, Office of Air Programs. The object of the study was to determine the technical and economic feasibility of a transmission to be utilized with gas turbine or Rankine cycle engines. Application of the engine/transmission was to a full size "family car." Since the Rankine cycle prototype engine hardware will be available before single shaft gas turbine hardware, priority was given to the Rankine cycle engine.

The study was accomplished through a two-phase, multi-task program which included:

1) Evaluation of transmission types through a feasibility study and ultimate selection of a transmission type.

2) Evaluation of the selected transmission type through design calculations and layouts, performance analysis, control system analysis, and cost analysis. A number of different types of

transmission were initially evaluated including conventional multi-shift, hydrostatic, hydrokinetic, electric, belt/chain, hydromechanical, and traction types. They were assessed in terms of efficiency, technology and production status, cost, controllability, size, weight, and noise. The hydromechanical and traction transmissions were eventually selected for more detailed evaluation.

Sketch layouts were made of both types and extensive system efficiency data was generated by the Sundstrand vehicle system computer model. System efficiency was the actual fuel consumption of the vehicle, transmission, and engine system over the Federal Driving Cycle, Simplified Urban Driving Cycle, and Simplified Country Driving Cycle.

At the conclusion of this more "in-depth" evaluation, it was not evident that either candidate should be discarded. It was, therefore, decided to continue through the "Selected Transmission Evaluation" phase with both the hydromechanical and traction transmissions. The hydromechanical transmission selected was a multi-mode type. It has three modes of operation – one hydrostatic and two hydromechanical (Tri-Mode). The traction transmission configuration selected was a toroidal type with 2 rows of toroids, and output torque converter, and a forward/reverse gearbox.

Detailed sizing of the two transmissions was done eventually resulting in detailed layouts of both. In parallel with the mechanical layout effort, the controls were defined and evaluated. An analog study of the control system and a failure analysis were also completed.

Emmissions were a major evaluation criteria. However since little data was available relative to various engine operating conditions, it was assumed in conjunction with EPA that minimum emmissions occurred at maximum fuel economy. System efficiency, therefore became a major evaluation criteria with considerable effort being expended in this area. Detailed system fuel consumption values were determined utilizing a vehicle system computer simulation in which vehicle, engine, transmission, and duty cycle characteristics were programmed. Both transmissions were evaluated with the two specified engines – Aerojet Rankine cycle with a turbine expander and the AiResearch regenerated single shaft gas turbine. The engine fuel consumption data for the engine was supplied by EPA/AAPS at the initiation of the study and is reflected in the values shown in the report. Modifications to this data have occurred particularly relative to the Aerojet engine, which would change the absolute values in the report.

Cost estimates were made for both transmissions using comparative data, vendor quotations, and in-house estimates. The resulting costs were also compared with a conventional three speed automatic transmission. The study resulted in the determination that both the hydromechanical and traction transmissions were technically and economically feasible for application to limited speed range Rankine or gas turbine engines. Each had certain features better than its competitor and, in some instances, they were equivalent. The hydromechanical transmission has slightly better efficiency, is shorter, and represents less technical and program risk to develop a pre-prototype unit. The traction transmission is estimated to be lower in cost, is lighter, and is inherently quieter.

A specific selection of the best type will have to be made by weighing the various characteristics, in terms of their overall importance; a task not undertaken in this study.

II. RESULTS AND CONCLUSIONS

1) Infinitely variable ration transmissions are the most flexible and best type for application to engines required to run over a narrow speed range. They are also applicable to all other types of engines, including the conventional spark ignition type, providing the ability with all engines to optimize a selected characteristics such as fuel consumption, emissions or performance.

2) The hydromechanical and traction transmissions are the most feasible and desirable types to provide an infinitely variable ratio.

3) Based on the given engine data, the hydromechanical or traction infinitely variable ratio transmission becomes more attractive when utilized with the AiResearch single shaft gas turbine engine than with the Aerojet Rankine cycle engine due to the shape of the specific fuel consumption curves.

4) A multi-mode (3 mode) hydromechanical transmission and a two-element traction ratio changer combined with an output torque converter and forward / reverse gearbox are the two configurations selected as the best candidates for use with the AiResearch gas turbine and the Aerojet Rankine cycle engine. The use of a slipping clutch rather than the torque converter with the traction ratio changer may be acceptable although further study is required.

5) Overall feasibility assessment of the hydromechanical and traction / torque converter transmissions indicates that their specific features are similar with the overall rating dependent upon the relative importance of the various criteria. Transmission comparison utilizing two important criteria, efficiency and technology status, indicates that the hydromechanical transmission has a higher overall efficiency and presents a lower overall risk to produce a pre-prototype by early 1974. Efficiency of the traction/slipping clutch transmission is equal to that of the hydromechanical transmission.

6) Both the hydromechanical and traction transmission compare favorable with a typical three speed automatic transmission in terms of performance, weight, and size. The cost increase must be viewed in terms of the increased capability and flexibility of this type transmission. A comparative summary is as follows:

Type	Actual Weight, Pounds	Relative Weight	Actual Cost, $	Relative Cost	Volume In^3
Hydromechanical	92	0.80	122	1.37	1390
Traction	77	0.67	105	1.18	1275
3 Speed Automatic	150	1.00	89	1.00	2500

7) The hydromechanical or traction transmission can be utilized with either the single shaft gas turbine or the Rankine cycle engine with minor modification to the control system and a change in torque converter diameter.

8) The computer simulated performance of the full size automobile, Rankine cycle engine, and either the hydromechanical or traction transmission met or exceeded all start-up, acceleration, gradeability, and maximum speed requirements of "Prototype Vehicle Performance Specification" dated January 3, 1972.

9) The computer simulated performance of the full size automobile, Brayton cycle engine, and either the hydromechanical or traction transmission met or exceeded all start-up, acceleration, gradeability, and maximum speed requirements of "Prototype Vehicle Performance Specification" dated January 3, 1972. However, to meet the 0-10 second acceleration requirement, the Brayton cycle engine power must be increased to 155 HP (12% above specified) when coupled to the traction transmission and to 145 HP (5% above specified) when coupled to the hydromechanical transmission.

10) The hydromechanical transmission is inherently noiser than the traction / torque converter transmission. However, utilizing presently known and demonstrated noise reduction techniques, it is anticipated that the vehicle noise requirements of "Prototype Vehicle Performance Specification" dated January 3, 1972, can be met.

11) The control of either the hydromechanical or traction transmission is essentially the same. It is compatible with the specified engines, is stable, is simple, and provides a "driver-feel" comparable to existing automative transmissions.

12) Installation of the hydromechanical or traction transmission is compatible with the specified engines and when installed in a full size automobile as specified, requires no modification to the structure.

III. RECOMMENDATIONS

Initiate a hardware development program for an infinitely variable ratio hydromechanical or traction transmission. Such a program should include the design, manufacture, dynamometer test and vehicle test of the selected transmission for a specific engine. Analysis of the transmissions with the two specified engines would seem to indicate that the single shaft gas turbine provides a better application.

While selection of the specific transmission type is dependent upon the relative importance given to the various transmission parameters, the hydromechanical type is the better candidate based on higher efficiency and lower development risk.

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IV. FEASIBILITY STUDY

A. INTRODUCTION

1) Methodology

Identification of a large number of possible concepts indicated that within the short study time available, a successive screening process would have to be utilized. Inital screening was a "gross" process in that basic, known characteristics were evaluated and those concepts which were promising were considered further.

As each screening process was undertaken more detailed evaluation was made. System characteristics of efficiency cost, controllability, size, weight, noise, and technical and production status were considered in varying degrees depending on the particular screening level. For the most promising candidates sketch layouts were made to determine hardware feasibility and complexity. Also a vehicle system computer model was utilized to determine vehicle efficency and performance.

In most instances, information for evaluation was from "in-house" sources including test data, studies, patent files, and literature searchs.

2) Requirements

Comparison of the technical and economic feasibility of the various transmission types for use with the two specified engines — the Aerojet Rankine Cycle and the AiResearch Brayton cycle engines required consideration of the following parameters.

a) Vehicle Parameters

Detailed vehicle description and performance requirements are given in Appendix I-2 "Prototype Vehicle Performance Specification." A summary of the more important parameters that affect the transmission are:

- Test Weight — Wt. = 4600 pounds. This weight to be used for all fuel economy and acceleration calculations.
- Gross Weight — Wg. = 5300 pounds. This weight to be used for sustained velocity calculations at 5 and 30% grades.
- Maximum Vehicle Speed — 85 mph.

b) Engine Parameters

Brayton Cycle	Rankine Cycle	Rankine Cycle
47,460	Minimum Turbine Output Speed - RPM	16,000
83,055	Maximum Turbine Output Speed - RPM	22,000
145 (Tri-Mode) 155 (Traction)	Maximum Power - HP	163

Engine speed to be controlled by the transmission to give minimum specific fuel consumption over the operating load range.

c) Other

Assumed maximum tractive effort – 2500 lb.

B. ANALYSIS OF TRANSMISSION TYPES

1) Discussion of Types

a) Multi-Shift Fixed Ratio – Power Shifting Gearbox with Start Clutch for Fluid Coupling/ Converter

This type of transmission would need an energy dissipating device for starting the vehicle from rest (friction clutch or fluid coupling) and would need 6 to 8 fixed ratios to keep the engine speed within limits over the vehicle speed range. The primary advantages are good efficiency and existing high production technology. The disadvantages are: inability to control engine speed to any required operating curve; discontinuous power to the wheels; and power shifting life limiting clutch packs. The cost of this type of transmission would be comparable with a traction or hydromechanical type infinitely variable ratio transmission but without their advantage of smooth, precise engine speed control.

b) Electric – Engine Driven Generator, D.C. or A.C. Motor Driven Wheels and Electronic Solid State Controls

Where good efficiency, light weight, and low cost are requirements, electric transmissions are invariably rejected. Efficiency of this transmission system is normally in the 60 to 70% range. A study report (Reference 1, Section IX) shows in Table 4, page 25, that a family car system weight exluding the A.C. generator power source is 348 pounds. Adding 100 pounds for the generator, rectifier, speed increaser, and voltage regulator totals 448 pounds. The components considered were all lightweight aircraft types. This total of 448 pounds is at least twice that of the conventional 3 speed automatic transmission plus drive shaft and rear axle.

The cost would be at least twice that of a conventional torque converter automatic transmission. Cost and weight, therefore, eliminated the electric transmission from further consideration.

c) Hydrostatic – Engine Drive Variable Displacement Hydraulic Pump with Close Coupled Fixed or Variable Displacement Hydraulic Motor

Pure hydrostatic transmission are invariably rejected for high speed vehicles where high efficiency, light weight, and small size are important. Hydraulic unit displacement required to transmit full rated HP at top vehicle speed can be as much as 18 times that required in a 3 mode hydromechanical transmission for the same vehicle. It is readily apparent that a pure hydrostatic transmission is unacceptable for a high speed vehicle. Overall efficiency of the pure hydrostatic transmission is considerably lower than the hydromechanical transmission type.

d) Hydrokinetic – Torque Converter of Fluid Coupling

Hydrokinetic and aerodynamic torque converter / coupling devices can be used as the principle means of varying the speed ratio across a transmission, such as a converter in series with a one, two or three speed gearbox. (This category differs from the 6–8 speed multi-shift fixed ratio devices discussed in (1) where the principle means of varying the speed ratio is by changing gear ratios, and the clutch or converter is basically only a device for starting from rest.

This method of speed control is considered unsuitable because –

The converter is an energy dissipating device, and therefore inefficient unless operating close to the coupling, or lock up point. Wide vehicle operating speed ratios require converter operation considerably removed from the coupling point, in the very inefficient range.

The speed ratio for fixed blading type converters cannot be controlled at will, but is a function of the instantaneous load and speed condition. Variable blading does allow some measure of independent speed control, but at the expense of further operating efficiencies.

Converters, however, do work well and have their advantages when combined with some other means of ratio control, such as the traction transmission.

e) Belt-Chain – Variable Sheaves either Belt or Chain Driven

This type transmission must also be provided with an energy dissipating clutch or hydraulic coupling for vehicle start up as the output of the variable ratio belt cannot be brought down to zero speed.

Many belt or chain variable speed transmissions have been developed and used successfully in the machine tool and stationary construction or industrial machinery where a stepless variable output speed is advantageous. For automobile transmission, several have been built and tested in low power vehicles, but data is not available. Chains and belts are life limited items and are also speed and power limited. Because belts are a high maintenance item in even normal accessory drives, great efforts are being expended to replace them with more reliable drives such as hydrostatic. Therefore, further investigation of this type of transmission was dropped.

f) Hydromechanical – Engine power is transmitted through both mechanical and hydraulic paths to obtain infinetly variable ratios.

The hydromechanical transmission is an infinitely variable ratio transmissions and therefore offers maximum flexibility. Considerable development work has been accomplished on this type of transmission and hardware can be developed with a minimum amount of risk. Size, weight, durability, controllability are proven. Efficiency, although probably lower than some other alternatives, does provide an equivalent or better system efficiency due to its ability to operate the engine at its optimum fuel consumption.

Noise represents a potential problem although noise reduction techniques developed over the last few years should provide acceptable noise levels.

g) Traction – Transfer of torque and speed through friction contact.

The traction transmission is also an infinetly variable ratio device. The traction drive has potential benefits of high efficiency, low vibration, and low noise. Development of a new traction fluid by Monsanto has also improved force and stress levels such that it looks more attractive.

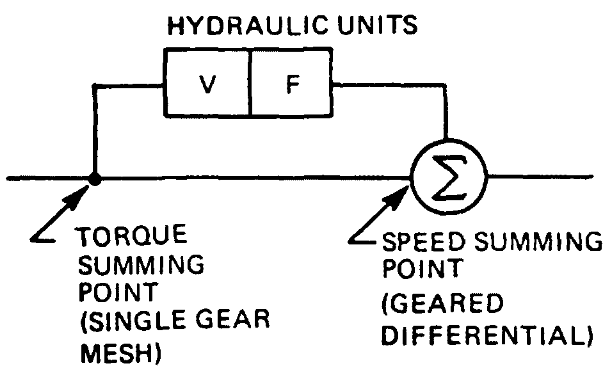
2) Conclusions

Based on the analysis of the various types of transmissions, it was concluded that an infinetely variable ratio transmission was the best type of transmission for the limited speed range engines being considered. The hydromechanical and traction transmissions are the best infinetely variable ratio candidates. It was therefore decided to continue with a more detailed optimization of both of these candidates with one to be selected at the conclusion of that phase.

C. OPTIMIZATION OF SELECTED CANDIDATES

1) HYDROMECHANICAL

The typical hydromechanical circuit schematic is shown in the following sketch.



The "V" unit is the variable displacement axial piston pump/motor type and the "F" unit is the fixed displacement axial piston pump/motor type. The two units are hydraulically ported to each other so that when one is a pump the other is a motor, and vice versa. The "F" unit swashplate (or wobbler) is at a fixed angle of 14^{o} and the "V" unit can be varied to any desired angle from 0^{o} to $\pm$ 15-1/2^{o}. Both swashplates are non-rotating.

For best utilization of the hydraulic unit to cover the complete operating range of the vehicle, it is desirable to run the "V" unit at its rated speed, and run the "F" unit through its full operating speed range (plus to minus rated speed).

Hydraulic unit efficiency is a function of speed, pressure and wobbler angle.

Studies conducted in 1966 for TACOM R & E Directorate under contracts DA-11-022-AMC-695 (T) and 2269 (T) as well as in 1971 under Phase I of EPA Contract 68-04-0034 covered many different types of hydromechanical transmission schemes. Some of those eliminated from consideration then, and are eliminated now for similar reasons, were the simple output differential, input differential, multiple hydraulic units such as the David Brown, Stratos, or Ebert designs, and the Borg-Warner multiple path arrangement which utilizes four variable displacement hydraulic units.

As stated previously, some hydromechanical schemes were eliminated through analysis of:

Hydraulic unit size, number of hydraulic units, quantity of power transmitted hydraulically, number of clutches, amount of gearing, and overall complexity.

The remaining schemes for final analysis were the Dual Mode (DMT), the Tri-Mode (TMT), and the Quadri-Mode (QMT).

Hydromechanical transmissions can be designed to operate in more than one mode in order to reduce the maximum HP that is transmitted hydraulically, which reduces the hydraulic unit size, and increases efficiency. This may be achieved by having a straight hydrostatic mode for start-up (whence all the engine power is transmistted hydraulically), and then one or more hydromechanical modes of operation. In a single mode of hydromechanical operation, the fixed displacement hydraulic unit is typically operated over its full rated speed range; that is, from plus its maximum rated speed down through zero speed and up to minus its rated speed. Two modes of hydromechanical operation then would typically consist of a system of gears and clutches that would operate the fixed displacement unit over its full plus to minus rated speed, two times, and three hydromechanical modes would do this three times, etc. The gears and clutches are arranged in such a manner that there is no speed discontinuity at each clutch shift point when changing from one mode of operation to another. Thus, all clutch shifting is done at synchronous speeds and no power is absorbed by the clutches. The more modes that are added, the more efficient the transmission can become, but at the cost of increased complexity of gears and clutches.

Sundstrand has considered the following hydromechanical transmission types:

a) The Dual Mode Transmission (DMT)

This transmission has one hydrostatic mode and one hydromechanical mode, schematically represented in Figure IV-1. Sundstrand has developed a transmission of this type for the heavy duty truck market and will be in production in 1973.

b) The Tri-Mode Transmission (TMT)

This transmission consists of one hydrostatic mode, and two hydromechanical modes. There are many ways of achieving this schematically; one way is shown in Figure IV-2.

c) The Quadri-Mode Transmission (QMT). This transmission consists of one hydrostatic mode and three hydromechanical modes. One way of achieving this is shown by the schematic in Figure IV-3. It should be noted that this schematic is similar to the Orshansky Transmission Corporation "Three Range Transmission" which was also evaluated. The Orshansky transmission requires, however, one additional friction element.

In evaluating the optimum number of modes for this application, Sundstrand chose the Tri-Mode Transmission as being the best compromise between efficiency and complexity. As more clutches

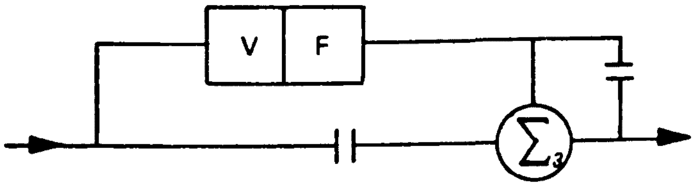


Figure IV-1 Dual Mode Transmission

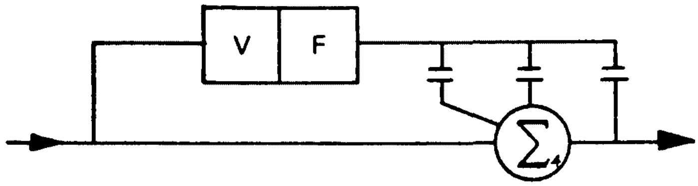


Figure IV-2 Tri Mode Transmission

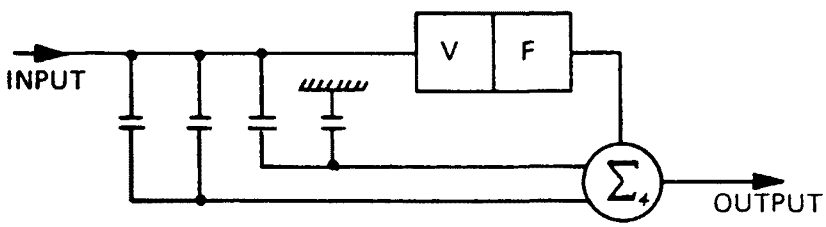


Figure IV-3 Quadri-Mode Transmission

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are added to obtain increased number of modes, the spin losses from these clutches become substantial, and reduce the gains in efficiency from this larger number of modes. As an example, following is a tabulation including some relative cost and efficiency data for Dual Mode, Tri-Mode, and Quadri-Mode Transmissions.

Sundstrand Transmission	Hyd. Unit Displacement	No. Clutches	No. of Gears	Cost	Fuel Consumption (over Veh. Life)
DMT	100%	2	11	100%	100%
TMT	42%	3	15	114%	94%
QMT	22%	4	21	125%	92%

2) TRACTION

The potential benefits of high efficiency, low vibration and low noise of traction transmission are well known. With the availability of a suitable fluid and a design to reduce forces and stresses, the traction drive becomes competitive enough to warrant a more thorough evaluation.

Variable ratio traction transmissions of various types have been studied over the years. The toroidal type has emerged as the best design for highest power density, reasonable life, and good efficiency. Sundstrand, Lycoming, Rotax, General Motors, English Electric, and others have built and tested the toroidal types; Tractor also is developing a modified torodal type. Lycoming has been the only company to market a toroidal traction transmission but other companies have successfully tested prototype designs.

There are many options open in the design of a toroidal traction variable speed drive. One of the basic design considerations is the use of a two row (dual toroid) design or a single row (single toroid) design.

A single row toroidal drive derives its name from the fact that there is only one set of traction rollers, typically three rollers per set, that transmit power between the input toroidal disk and the output toroidal disk.

In a two row device, the power flow is split from the center, or input toroid, through two sets of traction rollers. The two sets of rollers, with one set on either side of the input toroid, transmit power to the output toroids which are located at each end of the traction drive.

Large thrust loads are required at the toroid/roller interface to provide the normal force necessary to prevent gross slip between the rollers and the toroids. The benefit from the use of a two row device is that since it is symmetrical, the thrust loads from the two halves of the unit cancel each other and do not have to be taken thru thrust bearings. On the other hand, a single row traction drive requires very large thrust bearings to react to the large axial thrust loads, and also, since there is load sharing between the two halves of a two row device, the required toroid diameter for a two row unit is considerably smaller than for a single row unit.

One of the most important considerations throughout this study has been system energy efficiency. The relatively large power loss that would be encountered in the large high speed thrust bearings of a single row unit make the two row design much more attractive from an efficiency point of view.

Although a single row traction drive would cost less to produce, it is felt that the advantages to be realized from using a two row unit more than offset the cost.

The basic scheme of the toroidal type is shown in Figure IV-4. Principal components are the input toroid, rollers, and output toroid. The rollers are steerable and are "steered" to the necessary angle to provide the input to output speed ratio desired. When the input toroid is rotated, the rollers turn and exert a traction force on the output toroid. The toroids must be held together to insure sufficient traction exists with the rollers to transmit the desired power. The rollers are steered or tilted to the angle which produces the desired output speed. With the rollers angled as shown in the figure, output speed is lower than input speed. At the opposite angle, output speed is higher than input. Power capacity for a given size and given number of rollers is a function of the clamping force between input and output toroids across the rollers, and the traction coefficient of the fluid being used. The torque producing force at the point of roller contact is the product of the clamping force and the traction coefficient. Life of the unit is a function of this force and devices have been developed to vary this force in proportion to the load with a resultant increase in life.

The traction transmission design for this application must have a disengaging device to permit zero output speed when the engine is running. Three ways of achieving this were considered, and they are listed below along with the effects of each type on the traction drive unit.

a) Slipping Clutch

- The traction drive unit must be sized to take the maximum slipping torque of the clutch which must be greater by some 10-30% than the maximum torque generated by the engine. For narrow speed range engines, where minimum speed is close to maximum speed, the power which must be dissipated in the clutch is high.

- The slipping clutch does not readily offer any load "shock absorbing" protection to the traction unit. This is an important consideration with variable thrust types of traction drive units in that thrust must always be maintained sufficient to prevent gross slip between roller and toroid. Slips in excess of approximately 3% result in decreased torque ability and damaging spin out to even greater slips until the load is removed. These shock loads could be seen in the drive train under such conditions as accelerating on an ice-patched surface or sudden wheel-lock when braking.

- Once locked up, the slipping clutch does have the advantage of virtually zero power loss, giving the most efficient traction drive system of those being considered.

b) Split-Path with Planetary Summer

This consists of a power splitting circuit, similar to the hydromechanical concept, but replacing the hydraulic unit with a traction unit.

- The traction drive unit would have to be upsized to take the high power which can recirculate through this type of power circuit at start-up. The hydraulic ratio control system

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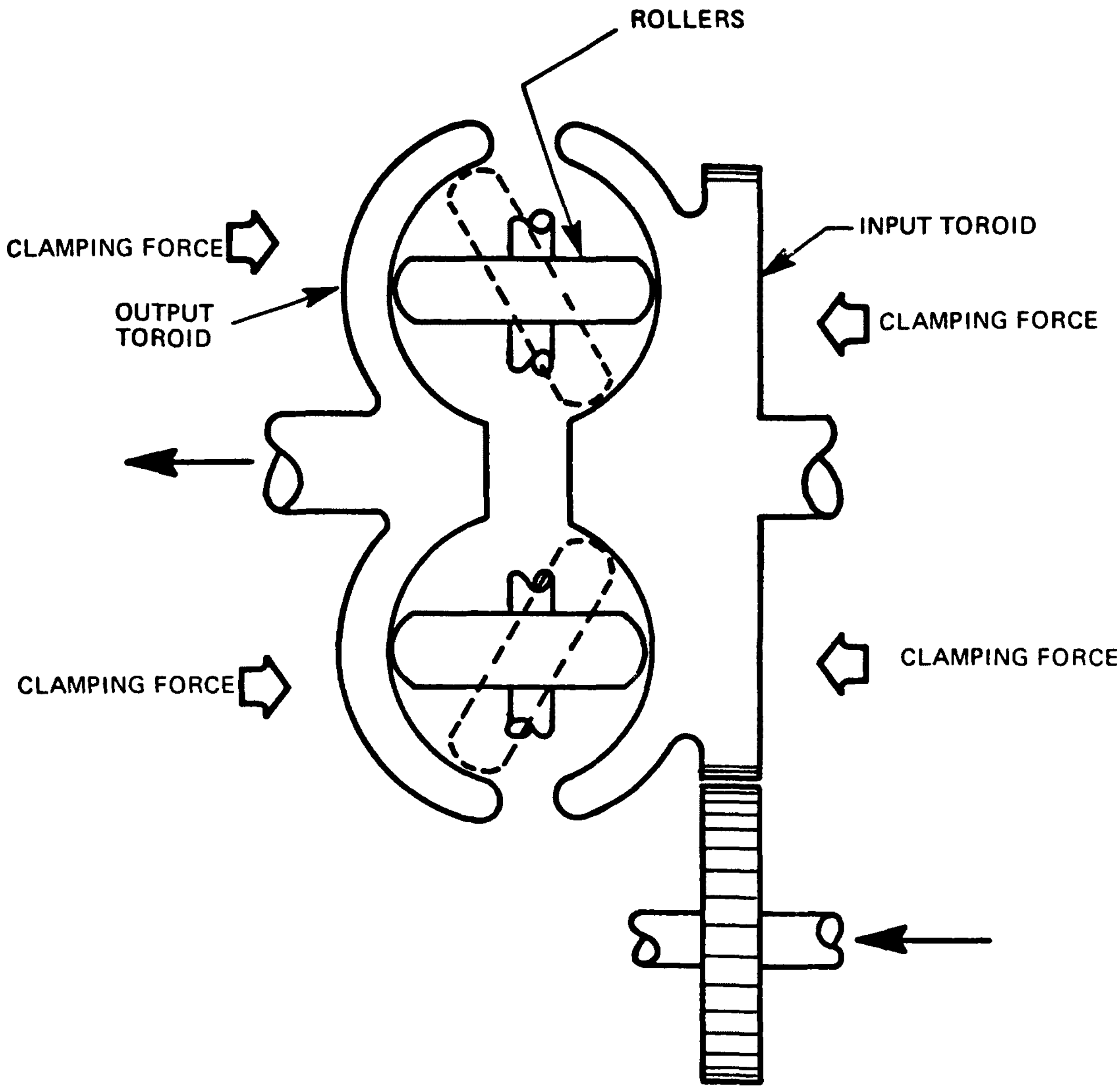


Figure IV-4 Traction Drive

could be used to act as a torque limiter. Without torque limiting, the torque would theoretically go to infinity under conditions of holding the output stalled at wide open throttle. The traction drive unit would have to be sized to take this limiting torque value, which could be several times greater than maximum engine torque.

- This system does not offer any load "shock absorbing" protection for the traction drive unit.

- The split-path system is non-dissipative and would be efficient. It could also be arranged to give a "built-in" reverse ratio capability, obviating the need for a reverse gearbox.

c) Torque Converter, or Fluid Coupling

The torque converter can be placed on either the input or output of the traction drive unit.

- The speed versus horsepower absorption characteristics (torque at a given speed) of an output torque converter can be utilized to down-size the traction drive unit by reducing the maximum torque that can be seen by the traction drive output. This is illustrated graphically in Figure IV-5. An input mounted torque converter will up-size the traction drive.

- The output mounted torque converter offers a high degree of load "shock absorbing" for protection of the traction unit. The input mounted converter offers lesser protection.

- The torque converter system would be the simplest and most reliable although less efficient than the other two schemes described previously.

From these considerations it was decided to use an output mounted torque converter as it is the only device which gives any degree of shock load protection to the traction drive unit, and it allows the use of the smallest possible traction unit. The torque converter has the additional advantages of low cost, excellent reliability and virtually zero maintenance. The efficiency penalty, in terms of MPG over the EPA Combined Driving Cycle, is about 5-8% relative to that which a slipping clutch start-up system could achieve.

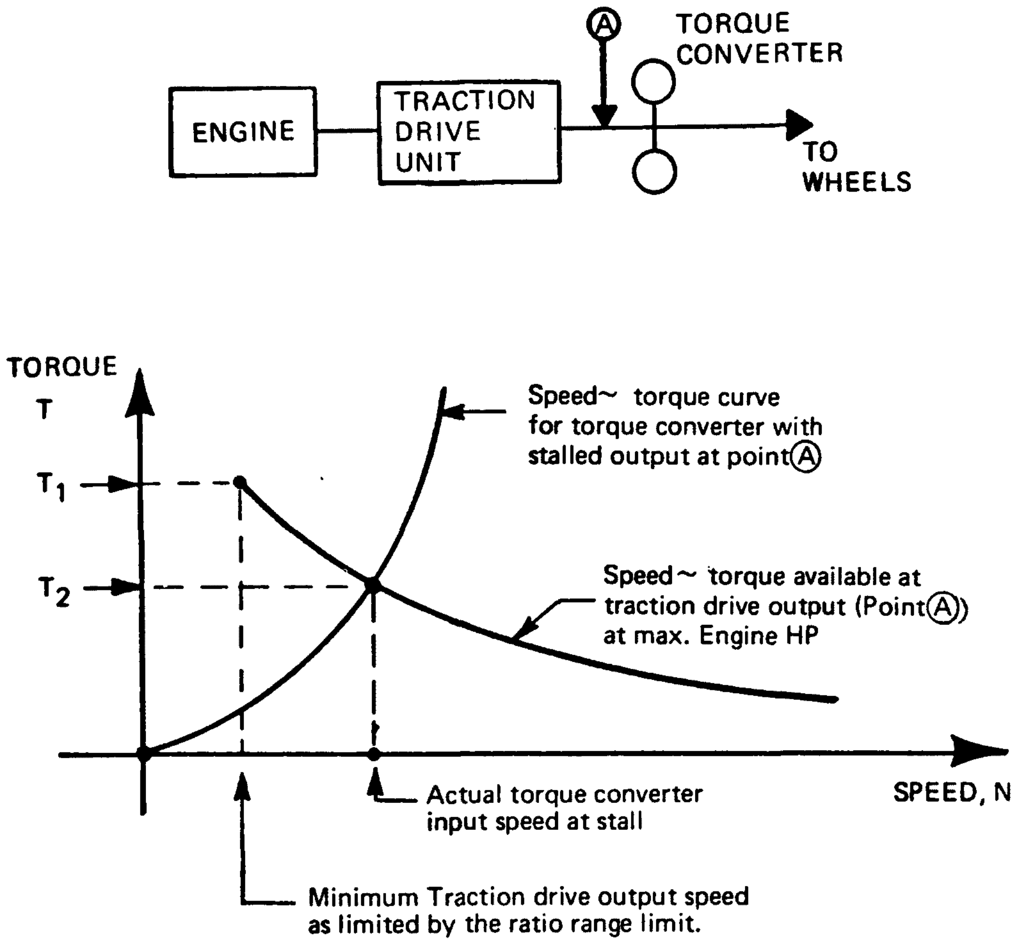
3) Comparison of Final Hydromechanical and Traction Design Choices

a) Size and Weight

When the two finalist transmission designs were decided upon, preliminary hand sketched layouts were made of each. These layouts indicated that both transmission schemes had the potential to become practical automotive transmissions from a size and weight perspective.

b) Efficiency and Fuel Consumption

An exhaustive computer analysis was completed on each of the two finalist transmission candidates using two full scale computer programs. Results of the computer analysis indicated that both transmission types offer the capability to program engine speed to minimize fuel consumption and / or emissions at a high level of energy efficiency. The computer efficiency and consequently the fuel consumption of the two systems were very close.



The torque converter limits the maximum output torque of the traction drive unit to T^2 rather than T^1 without a converter. The torque converter then, allows the traction drive unit to be sized to torque T^2.

Figure IV-5 Effect of Torque Converter Power Absorption Characteristics on Traction Drive Unit Torque

c) Cost

A preliminary total manufacturing cost comparison based upon a rate of 1 million per year indicated that the hydromechanical transmission would probably cost slightly more than the traction drive transmission.

d) Noise

The traction drive torque converter type transmission is inherently quiet. Attention must be given to the gearing to ensure that noise is reduced to a minimum. Sundstrand is committed to meeting acceptable noise levels with the present production Dual Mode hydromechanical transmission (DMT). Experience gained from testing the DMT has been applied in the proposed Tri-Mode hydromechanical transmission design. It appears at this time that the noise from either transmission can be brought within acceptable limits.

e) Producibility

There are no totally new unobtainable materials or processes involved in the manufacture of either transmission type. Tooling requirements will be similar.

f) Technology Status

The hydromechanical transmission represents a more highly developed device than the traction type. More companies are involved in actual testing and evaluation of hydromechanical transmissions than with traction transmissions. Production hydromechanical vehicle transmissions are being offered for sale, while production traction transmissions have been produced only for aircraft constant speed drive applications. Design and development of a pre-prototype transmission by late 1973 or early 1974 can be accomplished for either transmission type, although the traction type represents a somewhat greater risk. Either type could be ready for production by 1980.

The hydromechanical transmission primary development task will be the integration of its controls with the engine. Although the basic control scheme has been mechanized and demonstrated, operation with the specified engines will require additional effort.

The major development task for the traction type transmission is assurance and demonstration of the required life. Since the life capability is highly dependent upon the vehicle load (toroid and roller stress), vehicle and engine speed (traction ratio) and time at each condition, determination of the actual vehicle duty cycle is very important. Since little experience has been obtained with traction transmissions in vehicles this definition of the "real" operational requirements and the mechanical design reflecting these parameters, becomes the major development item.

g) Conclusion

After consideration of this comparative evaluation, Sundstrand felt they could not justify dropping either transmission from further consideration. As a result, it was decided to continue the detailed evaluation of both transmissions through to the completion of the study.

V. TRANSMISSION DESCRIPTION

A. HYDROMECHANICAL

1. Mechanical Operation

The following is a discussion of the mechanical operation of the tri-mode hydromechanical transmission with regard to the direction of power flow, component speed and torque relationships, and variable unit displacement. The transmission is shown in simplified schematic form on Figure V-1.

The transmission has three distinct modes of operation in forward. At 100% engine speed, the shift between Mode 1 and Mode 2 occurs at 12.3 MPH, and the Mode 2 to Mode 3 shift occurs at 40.8 MPH. At lower engine speeds the shift points occur at proportionately lower vehicle speeds. During Mode 1, the output from the fixed displacement hydraulic unit is geared directly to the output. In Mode 2 and Mode 3 operations, the fixed unit is geared into the plantetary. Reverse is the same as Mode 1, but in opposite direction and is obtained by stroking the variable displacement hydraulic unit in the reverse direction.

Figure V-2 shows schematically the geartrain arrangement.

a) Component Speeds:

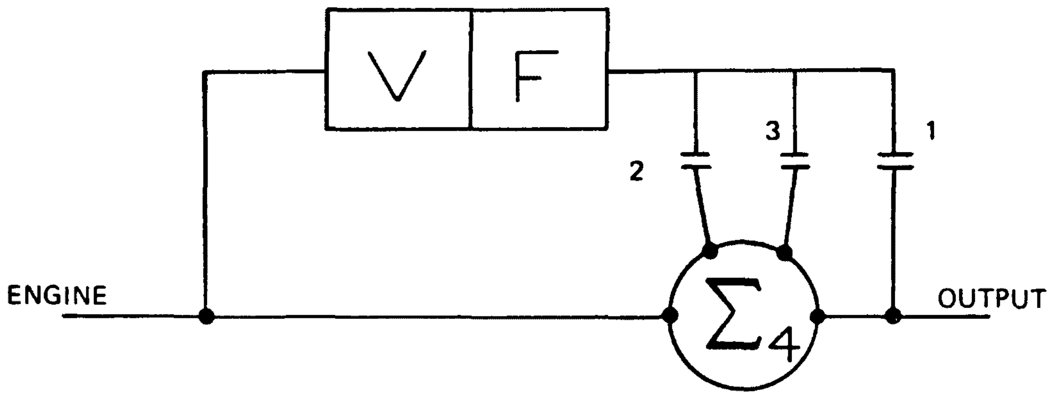
The variable displacement hydraulic unit is geared directly to the engine, therefore, its speed will always be directly proportional to engine speed.

In Mode 1 the fixed displacement hydraulic unit is geared directly to the output planetary link, so in Mode 1 its speed will be directly proportional to output speed, hence, vehicle speed. When the fixed displacement hydraulic unit speed increases to the point where it is equal to variable displacement hydraulic unit speed, a mode shift from Mode 1 to Mode 2 occurs.

In Mode 2, the fixed displacement hydraulic unit is geared to a leg of the planetary which causes power to be transmitted both hydraulically through the hydraulic units and mechanically through the planetary. The fixed unit speed decreases with increasing vehicle speed until it passes through zero speed and then increases in the opposite direction. When the fixed displacement hydraulic unit speed increases to minus one times the variable displacement hydraulic unit speed, a second mode shift from Mode 2 to Mode 3 occurs. Both Mode 1 and Mode 2 shifts are accomplished when the driving and driven clutch discs are at essentially equal speeds.

In Mode 3, the fixed displacement hydraulic unit is geared to another leg of the planetary, different from that of Mode 2, which again causes power to be transmitted both hydraulically and mechanically. The characteristics of Mode 2 and Mode 3 are very closely related, the only difference being the speed and torque ratios between the various elements. Increasing vehicle speed further after the Mode 2 to Mode 3 shift results in decreasing fixed displacement hydraulic unit speed (from its negative maximum) until it passes through zero, and then increases to its positive maximum speed (one times the variable displacement hydraulic unit speed) at maximum vehicle speed. Figure V-3 shows the hydraulic unit speeds schematically.

The speeds of the various links of the compound summer (in this case a four element planetary) can also be represented on a nomograph, shown on Figure V-4. A straight line



V = Variable Displacement Hydraulic Unit

F = Fixed Displacement Hydraulic Unit

Σ_4 = Four Element Differential

1 = Mode 1 Clutch

2 = Mode 2 Clutch

3 = Mode 3 Clutch

Figure V-1 Simplified Schematic

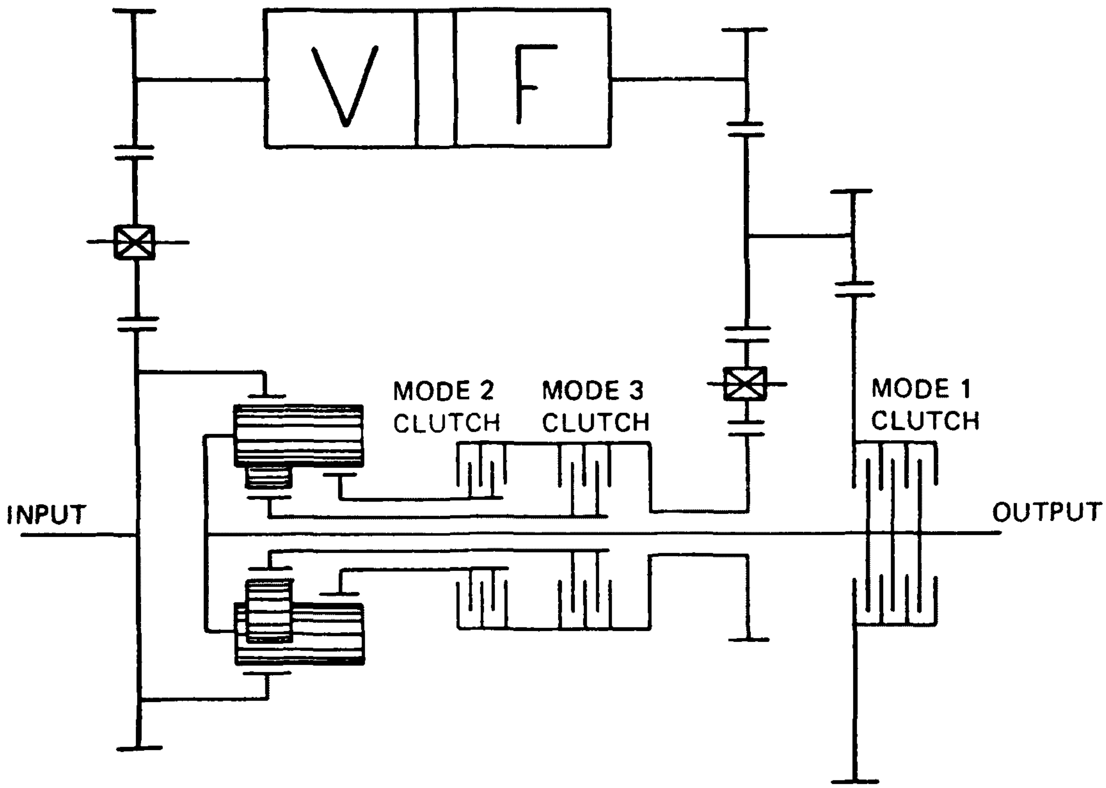


Figure V-2 Geartrain Schematic

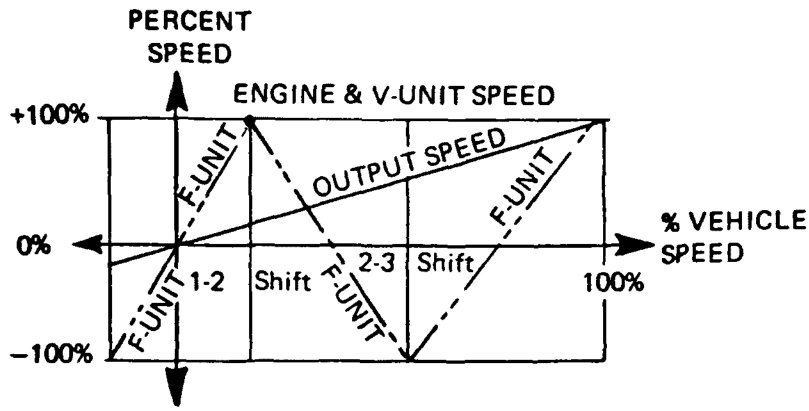


Figure V-3   Hydraulic Unit Speeds

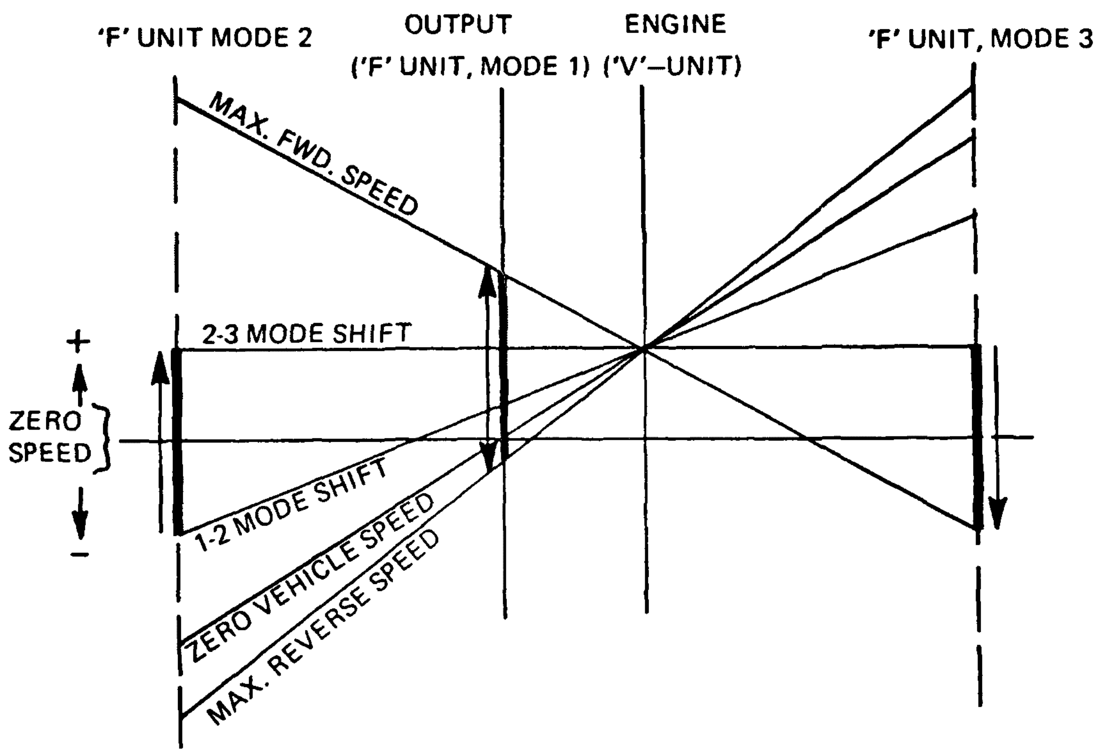


Figure V-4   Speeds of the Various Links of the Compound Summer

passing through any two link speeds defines the speed of the other two links. Thus, when output speed and engine speed are known, fixed and variable displacement hydraulic unit link speeds can be found. Since the fixed and variable displacement hydraulic units are related directly to their respective planetary links by gear ratios, all the system speeds can be calculated.

b) Hydraulic Unit Displacement:

The displacement of the variable displacement hydraulic unit can be calculated from the flow continuity equation. This equation is shown below in its simplified form (neglecting volumetric efficiencies):

Where:

$$Q = D_F N_F = D_V N_V$$

Q = Flow (in^3/min)

D = Displacement (in^3/rev)

F = Fixed Unit

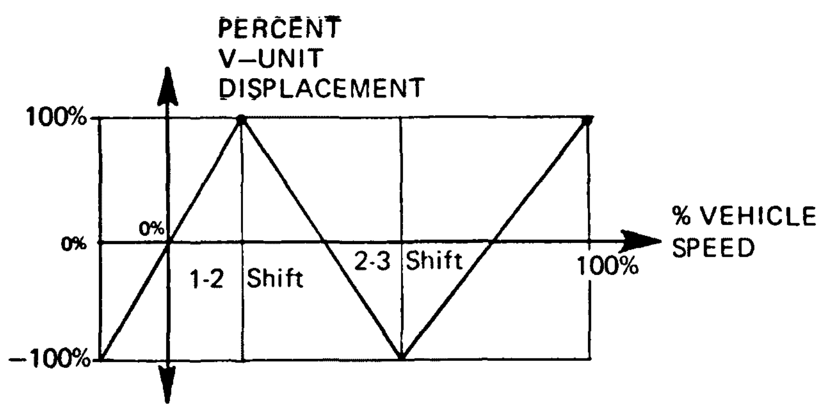
V = Variable Unit

N = Unit Speed (RPM)

Thus:

$$D_V = \frac{N_F}{N_V} \times D_F$$

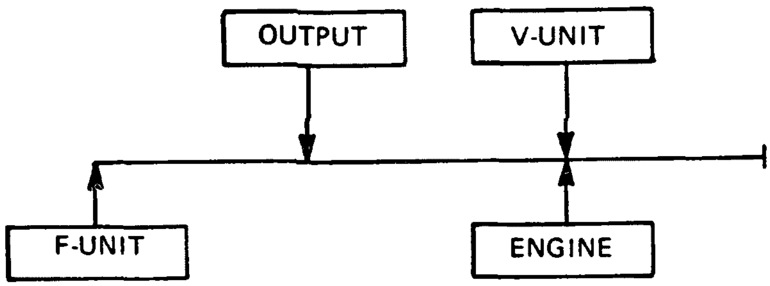
Therefore, for any given engine speed, the displacement of the variable displacement hydraulic unit will vary directly proportionally to fixed displacement hydraulic unit speed (see the following sketch).



c) Torque:

The reaction torques in a compound summer may be represented as vectors acting on a beam at positions that correspond to the link locations on the speed nomograph (Figure V-4). Unknown torques may be found by applying the equations of statics to the torque vector-beam analogy of the planetary (see the following sketch).

TYPICAL CASE: (MODE 2)



Although there is more involved when efficiency is taken into account, the V-unit and F-unit torques are related by the equations:

$$HP_{HYD} = T_V N_V = T_F N_F$$

$$T_V = T_F \times \frac{N_F}{N_V}$$

Where:

HP_{HYD} = Hydraulic horsepower

T = Torque

N = Speed

V = Variable unit

F = Fixed Unit

d) Hydraulic Unit Pressure:

When the torque balance is solved for any given set of external load and speed conditions, the working pressure can be calculated directly from the F-unit torque reaction. The basic formula relating F-unit torque and the working pressure is:

$$P = \frac{2\pi \times T_F}{D_F}$$

Where:

P = Working pressure (psi)

T_F = Fixed unit torque (in-lb)

D_F = Fixed unit displacement (in^3/rev)

e) Horsepower:

Horsepower is the product of torque times speed. The basic methods of solving for torque and speed in the transmission were defined previously. The magnitude of the horsepower in any link is the torque in that link times the speed of that link divided by the appropriate dimensional constant.

The direction of horsepower flow, on the other hand, must be determined from the direction of link rotation and the direction of applied torque. Sign conventions were established for the planetary speed nomograph (Figure V-4) such that any speed above the nomograph absissia is positive, and any speed below is negative. In the planetary torque balance beam (sketch), any vector pointing up is positive and any vector pointing down is negative.

2. Hardware Description

The following is a brief description of the various components which make up the tri-mode hydromechanical transmission. Reference should be made to the cross section drawing, 2724A-L5, shown in Appendix V-2 for indication of component arrangement and relative size.

a) Hydraulic Units:

The hydraulic units are the axial piston hydrostatically balanced configuration, typical of Sundstrand's standard line of hydraulic units for the aircraft, agricultural, and construction equipment market.

Figure V-5 shows a schematic cross section of a typical hydraulic unit of this configuration. While a variety of hydraulic pump/motor units could have conceivably been evaluated for this application, Sundstrand based hydraulic unit selection on our extensive experience in designing hydrostatic and hydromechanical transmissions for a variety of applications over the last 30 years.

The hydraulic units are identical in construction to hydraulic units presently being manufactured by Sundstrand for hydromechanical transmission applications where they have proven their reliability, low cost, and good efficiency.

Both hydraulic units have a displacement of 1.5 in^3/rev. One unit is variable displacement, the other is fixed displacement. The units are designed for 3000 psi nominal, 7500 psi overloads, and 9000 psi proof pressure.

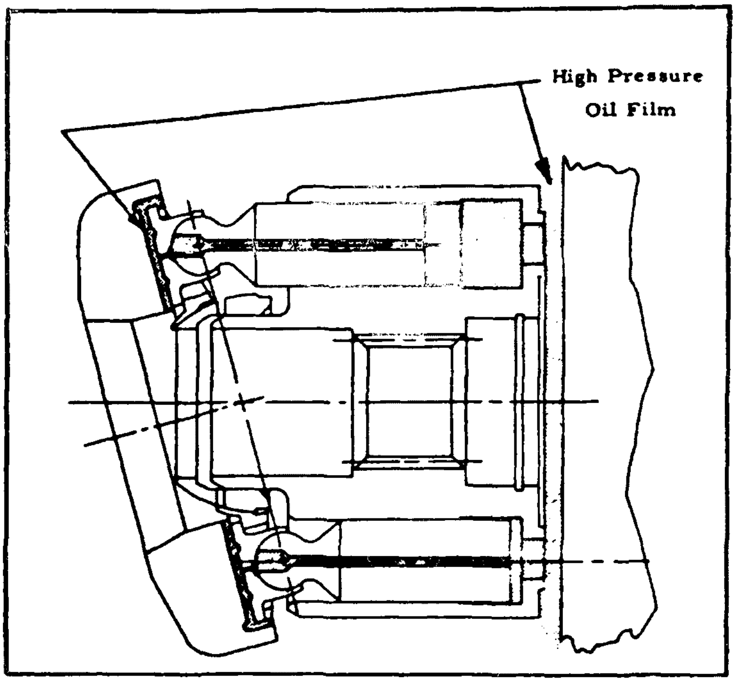


Figure V-5 Axial Piston, Slipper Type Hydraulic Unit

The units are mounted side by side with a common port plate manifold. Mounting the units in this manner provides for a shorter transmission length and allows for better noise reduction techniques to be utilized.

b) Clutches:

The clutches perform the shift function during the change from one mode to another. These clutches are of the conventional multiplate disc type common to automotive applications. They are simple to control, inexpensive, and have high torque capability. At the shift, the shaft speeds are essentially synchronized, thereby allowing the use of light duty clutches and are thus sized on torque capability and not energy dissipation.

Clutch design follows standard automotive practice. Steel separator plates are used with organic linings and the drums are ductile cast iron. The piston and the back-up ring are aluminum.

A centrifugal operated pressure sensitive check valve is incorporated within each clutch to preclude centrifugal pressure from actuating the clutch.

c) Seals:

Standard lip seals are used on the transmission input and output shafts.

Rotating seals between concentric shafts are of the cast iron piston ring type common with standard automotive practice.

d) Gears:

Helical gears have been assumed throughout the transmission, as in all automotive transmissions, to minimize noise. The gears are all designed to permit use of economical mass production techniques.

e) Charge Pump:

The charge pump is of the gerotor type common to automotive applications. It has been sized to provide for main hydraulic unit charging, control operation, clutch application and cooling, gear and bearing lubrication, and flow to the transmission cooler.

f) Bearings:

Extensive use has been made of radial and thrust load needle bearings. Bearings of this type are widely used in automotive applications as they are inexpensive, reliable, and have minimum lubrication requirements.

Tapered roller bearings are used in the hydraulic units as needle bearings are not suitable at these locations.

g) Component mounting:

The clutch pack assemblies and the planetary gear set are mounted on coaxial shafting which is supported by bearings at the input and output of the main housing and at the intermediate support plate. The hydraulic unit assembly and its drive gears, along with the charge pump, are mounted entirely on the intermediate plate which is mounted to the main housing. This type of construction allows for very easy assembly, maintenance, and gives the best possible noise isolation.

h) Controls:

The spool control valves are typical of those found in present automatic transmissions. The valve bodies are cast iron, the spools are hardened steel and, where applicable, steel sleeves are used.

The control linkages from the driver will be of similar type and construction to those presently used in automotive applications.

Speed sensing governors are of the rotating flyweight type and act directly on a valve stem.

i) Transmission Cooler:

The transmission cooler is not an integral part of the transmission and is listed here only as a reminder that it is required to dissipate the heat generated in the transmission. As this transmission does not vary speed ratio by dissipating energy, such as the torque converter, the cooling capacity would be less than required for a conventional automatic transmission while the transmission fluid flow rate will be about the same.

3. Size and Weight

The tri-mode hydromechanical transmission is designed to fit within the requirements stated in paragraph 6 of the "Prototype Vehicle Performance Specification" (see Appendix I-2). In brief, the transmission tunnel is not widened so as to decrease clearance between the accelerator pedal and the tunnel; the tunnel height does not affect full fore and aft movement of the front seat; it does not violate the ground clearance lines; it does not violate the space allocated for wheel jounce and steering clearances; and it does not degrade the handling characteristics of the vehicle.

The input or mounting flange is not a standard to fit the conventional internal combustion engine. However, as a reduction gearbox is required at the Rankine or Brayton cycle engine output, the mounting flange and output shaft location may be located to suit the proposed transmission.

The weight of the tri-mode transmission is 92 pounds dry. A weight breakdown is shown in Table V-1.

4. Design Analysis

By far, the majority of components in an automotive transmission are sized by considerations other than material stress such as economy of manufacture, or requirements of fitting over or around some other component. When weight is not a major consideration, components are often oversized to "keep out of trouble," and no heed is taken or calculations made of the exact margin of safety.

Obvious exceptions to this are gears, highly torqued small diameter or thin walled shafting, bearings that see predictable loads, and clutches (or other forms of friction elements). Appendix V-5 gives the summary of sizing this class of component along with a schematic which shows torques and speeds. The hydraulic units are sized by proprietary Sundstrand methods to meet their rated speeds and pressures.

In a study of this type where basic concept and feasibility are of prime importance, it is not appropriate to go into extensive sizing detail analysis. This is especially true when the design is being

TABLE V-1

HYDROMECHANICAL TRANSMISSION WEIGHT BREAKDOWN

Planetary Gearset	5.3
Transfer Gears	3.8
Idler Gears and Bearings	1.2
Shafting and Bearings	9.5
Clutches	13.7
Hydraulic Units (excluding shafts)	20.0
Housing and Port Plate	24.7
Control System and Charge Pump	6.2
Miscellaneous Hardware	7.5
Total (pounds)	91.9

made by personnel with many years of transmission experience. There are no areas in the transmission that are so critical that any increase in component size, that may be required after a detailed design study, would precipitate any significant cost performance or weight penalty.

B. TRACTION DRIVE

1. Mechanical Operation

This subsection is a discussion of the mechanical operation of the traction drive-torque converter transmission. Figure V-6 shows the general schematic.

Transmission input speed is proportional to engine speed. Therefore, the speed of the input toric disk is proportional to engine speed since it is driven by a gear on the transmission input shaft.

The speed of the output toric disks relative to the speed of the input toric disk is a function of the inclination of the traction rollers. The speed ratio across the traction drive is the same as the ratio of the radius of rolling contact on the output toric disk to the radius of rolling contact on the input toric disk with respect to the axis of the traction drive.

Transmission ratio changes are effected by changing the "tilt angle" of the roller axis which varies the radius of the two points of contact with the toroids. The "tilt angle" of the rollers can be changed by either of 2 methods:

a) Application of an external force to the roller mounting yoke and physically forcing the axis of rotation of the roller to the required angle.

b) "Steering forces" can be generated at the point of roller – toroid contact that will cause the roller "tilt angle" to change by translating the axis of rotation of roller relation to the toroid center. An explanation of how this is achieved is as follows:

Figure V-7a shows a cross-section of the traction unit with the roller in the 1:1 ratio position.

Figure V-7b shows the roller in an end view of the traction unit. The velocity vector of the roller at point of contact with the toroid is shown by vector "V". With the roller positioned as shown with zero slip between the roller and the toroid, vector "V" also represent the velocity vector of the toroid.

Figure V-7c shows the axis of rotation of the roller displaced an amount "X" to the left of a parallel center line ggoing through the axis of rotation of the toroid. The velocity vectors at the point of contact between the roller (V_R) and the toroid (B_T) bonger coincide. This difference causes a relative slip between the two members, represented by "V_S". This slip vector will be "down" at the point of contact between the roller and the toroid shown, and "up" as the other point of contact because the other toroid is rotating in the opposite direction. These two equal and opposite speed vectors produce equal and opposite forces on the roller which, if unrestrained at the roller bearing support will cause a turning movement on the roller, at right angles to its axis of rotation, which will "steer" the roller to some new angle to achieve equilibrium. The angle to which the roller axis will tilt to achieve equilibrium is a function of the distance "X".

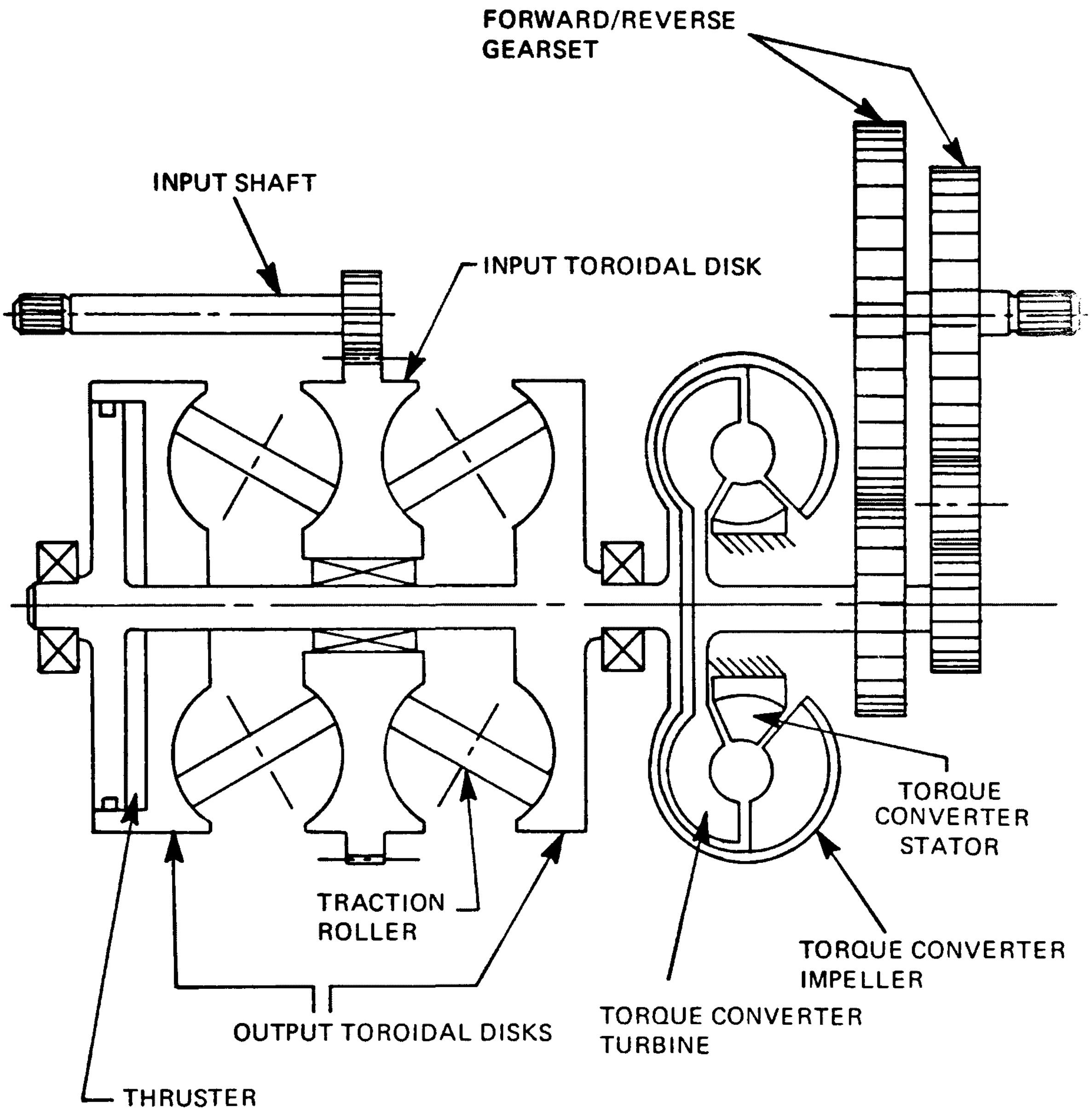


Figure V-6 Traction Drive - Torque Converter Transmission Schematic

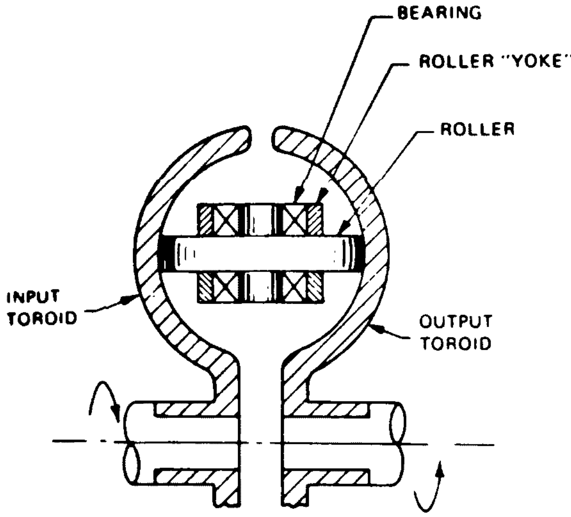


Fig. V-7a

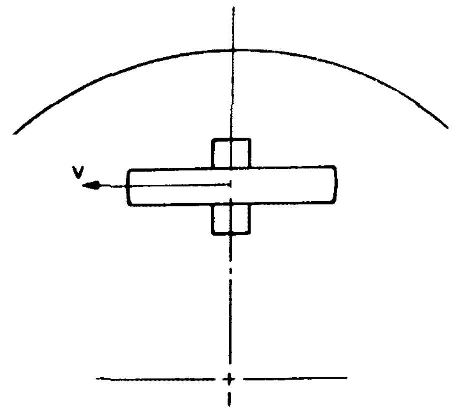


Fig. V-7b

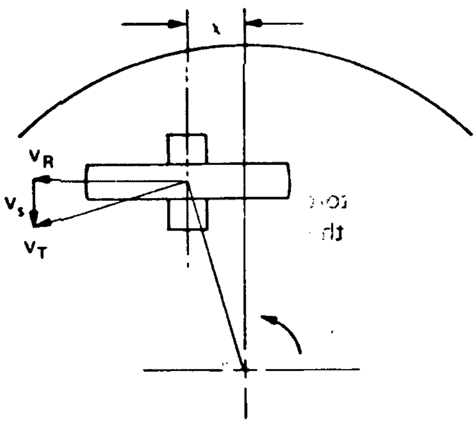


Fig. V-7c

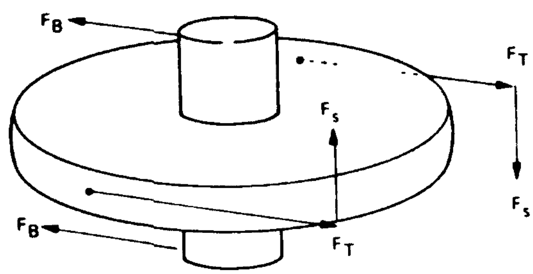


Fig V 7d

FIGURE V-7 TRACTION ROLLER STEERING MECHANISM

Figure V-7d shows a three dimensional free-body diagram of the roller, illustrating the two forces "F_S", which produce the couple that "steers" the roller. "F_T" is the tangential force on the roller at the contact points with the toroids and "F_B" the bearing reaction forces.

The proposed design utilizes the "steering force" approach. Translation of the roller axis of rotation is accomplished by applying a force to them through the hydraulic suspension and control cylinders. When the desired change in ratio is achieved, the steering forces will be cancelled and the unit will operate at the new ratio until the next ratio change is requested by the control system. The traction rollers are hydraulically interconnected in such a way that their tangential loads, rather than their absolute positions, must correspond. Therefore, load sharing is positively assured. Mechanization of this approach utilized in the design is defined by an English patent by McGill.

The required torque at the traction drive creates a tangential load at the traction roller/toric disk interface. This tangential load is sensed by the hydraulic control-suspension system, and the hydraulic pressure thus generated is applied to the hydraulic thruster which produces the axial clamping force across the rollers. Thus, the normal force necessary to allow a torque producing tangential force to develop at the roller contacting points is directly proportional to the torque being transmitted. The normal force then is only as large as it must be to prevent traction roller skidding, and unit life (which is inversely proportional to the cube of the normal force) is greatly extended. Initial pre-load is provided by a belville washer which develops sufficient initial force to allow charge pressure build-up. This force is negated when charge pressure is applied moving the piston out of contact with the toroid (See Appendix V-4).

The one-way clutch is provided between the output of the traction drive unit and the transmission housing. This clutch prevents the output of the traction drive unit from rotating backwards (such as would happen if the vehicle were allowed to roll backwards while engaged in forward drive). This reverse rotation of the traction drive could cause the traction rollers to "steer" themselves out of position.

The output of the traction drive is connected directly to the torque converter input member, the impeller. The speed of the torque converter output member, the turbine, is a function of vehicle speed and the ratio of the transmission output gears.

The function of the forward/reverse gearset at the transmission output is twofold. It serves to bring the relatively high torque converter speed down to a more favorable transmission output speed, and it also provides the capability for reverse vehicle operation.

2. Hardware Description

The following is a brief description of the principal components of the traction drive torque converter transmission. Reference may be made to the cross section drawing, 2724A-L4, shown in Appendix V-4 for indication of component arrangement and relative size.

a) Forward-Reverse Gearing

This gearing is all helical type constant mesh and is designed to conform to the conventional automotive type manual synchromesh gearbox.

b) Torque Converter

The torque converter is a single stage three element converter with a 3.0 to 1 stall torque ratio. The elements are typical automotive pressed steel construction scaled down in size from a standard automotive converter. The maximum diameter of the oil path is 6.12 inches and the maximum speed of the input impeller is 13,800 RPM at maximum engine speed.

c) Toroids and Traction Rollers:

Both the toroids and the traction rollers are form ground from M50 or M1 steel forgings. There are three 2.5 inch diameter rollers in each of the two toroids. The rollers rotate at a radius of 1.66 inches from the axial centerline of the toroids. At the forward side of the first toroid disk is the variable thrust device. A Belleville type spring imposes a 1700 pound thrust preload on the toroids and rollers. This preload is held constant as the control pressure builds up sufficiently to overcome the constant spring force. From then on the clamping force is directly proportional to the control pressure. Maximum control pressure is 400 psi.

The traction roller steering and suspension mechanism is all hydraulic and is based on an existing design which ensures the accurate load sharing described earlier.

d) Seals:

Standard lip seals are used on input and output shafts and rotating seals of the cast iron piston type. All seals are typical of those found in standard automatic transmissions.

e) Bearings:

Standard anti-friction ball and roller bearings are used throughout the transmission.

f) Control and Lube Pump:

The control and lube pump is a proven single lobe vane unit driven at input speed. Maximum flow is approximately 6 GPM at maximum engine speed.

g) Gears:

All gears except the input mesh are constant mesh of the helical type. The input mesh are the spur type, 26 diametral pitch, 20° pressure angle with a modified involute profile.

h) Controls:

The control valve block is cast iron with hardened steel spool valves or sleeves where applicable. The speed sensing governor is the rotating flyweight type acting directly on a valve stem.

Control linkages may be of similar type as presently used with automatic transmissions.

i) Transmission Cooler:

The transmission cooler is a separate item and is noted here as a reminder. The capacity required is equivalent to the present automotive automatic type transmission cooler.

j) Lubricating Oil:

The design of the traction transmission is based on the use of Monsanta Santotrak as the lubricating and cooling fluid.

3. Size and Weight

The traction transmission is designed to fit within the requirements stated in paragraph 6 of the "Prototype Vehicle Performance Specification" (see Appendix I-2). In brief, the transmission tunnel is not widened so as to decrease clearance between the accelerator pedal and the tunnel; the tunnel height does not affect full fore and aft movement of the front seat; it does not violate the ground clearance lines; it does not violate the space allocated for wheel jounce and steering clearances; and it does not degrade the handling characteristics of the vehicle.

The input or mounting flange is not a standard to fit the conventional internal combustion engine. However, as a reduction gearbox is required at the Rankine or Brayton cycle engine output, the mounting flange and output shaft location may be made suitable for the proposed transmission.

The weight of the traction transmission is 77 pounds dry. A weight breakdown is shown in Table V-2.

By far, the majority of components in an automotive transmission are sized by considerations other than material stress such as economy of manufacture, or requirements of fitting over or around

TABLE V-2

TRACTION DRIVE TRANSMISSION WEIGHT BREAKDOWN

Transfer Gears	3.7
Idler Gear and Gearing	0.5
Shafting and Bearings	9.3
Converter	4.4
Traction Drive	20.4
Housings	25.2
Control System and Charge Pump	5.6
Miscellaneous Hardware	7.5
Total (pounds)	76.6

some other component. When weight is not a major consideration, components are often oversized to "keep out of trouble," and no heed is taken or calculations made of the exact margin of safety.

4. Design Analysis

The double row traction transmission is basically designed from a stress-cycle curve and previous experience in designing and testing traction drives. The mean transmission input power requirement was calculated for the Federal Driving Cycle, the Simplified Suburban Route, and the Simplified Country Route. The maximum input power requirement was calculated from the acceleration and grade velocity requirements. These input powers, the specified life of 3500 hours, a transmission speed range of five to one, and appropriate toroid geometry ratios with a particular traction coefficient provide the basis for the traction transmission design.

The toroid geometry ratios involved in the design are (1) the toroid pitch diameter to roller diameter ratio, and (2) the conformity ratio, which is defined as the ratio of roller crown radius to roller pitch radius. The first ratio (1) defines the size of the machine and the amount of rolling to twisting contact that the rollers experience with the toroids. Since large toroid pitch radius to roller radius ratios approach more nearly pure rolling, the traction coefficient increases and the speed range decreases with this ratio. Therefore, in order to accommodate a 5 to 1 speed range and stay within package size limits, a ratio of 1.33 was chosen.

The second ratio (2) affects the shape and size of the footprint as well as the normal stresses. Higher conformity ratios for the same load result in higher stresses. Experience dictates a circular footprint or one that has its major axis in the direction of rolling. As a result a conformity ratio of 50% was chosen.

The traction coefficient also affects the overall transmission size and decreases as rolling contact velocity increases. For this design, at an input speed of 8000 RPM, the rolling contact velocity is 1400 inches per second. A traction coefficient of 0.04 is reasonably attainable for traction fluids at this velocity.

The maximum stresses calculated for this design are 448 ksi at maximum input power of 140 HP and 234 ksi at the mean input power. Mean input power is weighted average power over the EPA combined driving cycle as defined by the duty cycle. Using an assumed stress cycle curve and scaling from 1,000,000 cycles at 700 ksi for M50 tool steel, it was determined that the 3500 hour life requirement was satisfied.

Appendix V-6 shows a schematic of the transmission with individual component speeds and torques. The actual component sizing summary is the same as for the hydromechanical sizing shown in Appendix V-5.

In a study of this type where basic concept and feasibility are of prime importance, it is not appropriate to go into extensive sizing detail analysis. This is especially true when the design is being made by personnel with many years of transmission experience. There are no areas in the transmission that are so critical that any increase in component size, that may be required after a detailed design study, would precipitate any significant cost, performance, or weight penalty.

5. Traction ratio and torque converter optimization

An optimization study was carried out to determine the required system gear ratios, traction drive ratio range, and torque converter type and size. These parameters had to be determined within the system requirements of:

- Maximum vehicle 'creep' speed
- Maximum vehicle operating speed
- Engine speed range
- Reasonable torque converter idle HP absorption

These studies were made using the Sundstrand vehicle performance computer program for a traction drive transmission. Fuel consumption transmission energy efficiency, and other important criteria, were recorded for simulated runs over the Combined Driving Cycle while varying the transmission parameters – some, one at a time, and some in combination.

Many different torque converters were simulated and studied to gain a better understanding of the effects of converter characteristics on vehicle performance. For example, the study showed that a high stall torque ratio converter gave better fuel consumption than a low torque ratio converter, due to its more favorable efficiency curve at lower converter output/input speed ratios. Another important factor was torque converter diameter. Making the converter diameter larger makes it "tighter;" that is, it slips less, and is therefore more efficient. However the power absorbed at engine idle by a torque converter also increases with diameter, and must be considered.

A study was also made replacing the torque converter with a friction clutch. Total energy efficiency increased from 74% to 80%, and fuel consumption improved from 10.0 MPG to 10.8 MPG for the Aerojet Rankine engine, and from 15.0 to 15.7 MPG for the AiResearch Brayton engine. It should be noted that in realizing these gains, the advantages of having a torque converter as discussed in Section IV are lost. It would appear that these advantages outweigh the efficiency disadvantage. However, it is not completely evident that a clutch could not be used. A more detailed study of this would be made prior to a hardware design commitment.

Studies were also made using a torque converter lock-up clutch, and an input clutch in the system. The result of these studies, and some of the other optimization studies, are summarized in Table V-3. Figures are for the Aerojet Rankine engine.

The parameters chosen for the final transmission design were a torque converter ratio of 3 to 1 and a transmission speed ratio of 5 to 1. These result in only 3 HP absorbed at idle speed and 10.01 miles per gallon fuel consumption for the Combined Driving Cycle. A trade-off study of all of the studies and computer runs involving complexity, cost, and overall economics resulted in the choice of the above parameters.

C. Maintainability

It is expected that either the tri-mode or the traction transmission should provide no greater maintainability problems than present automotive automatic transmissions.

TABLE V-3
CLUTCH/TORQUE CONVERTER PARAMETER TRADE-OFF SUMMARY

Single Parameter Being Varied	Torq. Converter: Stall Torq. Ratio	Torq. Converter: HP Absorbed at Idle	Traction Drive Ratio Range	MPG Combined Driving Cycle
<u>Converter Lock-Up Clutch</u>				
Without	2.23:1	7	6:1	9.57
With: (Locks up at 0.9 Spd. Ratio)	2.23:1	7	6:1	9.65
<u>Converter Stall Torq. Ratio</u>				
	1.82:1	7	6:1	9.34
	3.00:1	7	6:1	9.70
<u>Converter Idle HP Absorption Rate</u>				
	2.23:1	7	6:1	9.57
	2.23:1	5	6:1	9.88
(an input clutch req'd	2.23:1	3	6:1	10.04
to give 0 HP loss)	2.23:1	0	6:1	10.39
<u>Traction Drive Ratio Range</u>				
	2.23:1	7	4:1	9.49
	2.23:1	7	5:1	9.55
	2.23:1	7	6:1	9.57
	2.23:1	7	8:1	9.58
Finally Chosen Combination	3.0:1	3	5:1	10.01

C.) NOISE

The transmission noise whether air-borne or structure borne is an important consideration for any automotive transmission. It is of particuliar concern because the vehicle levels required are relatively low. The hydromechanical transmission is inherently a higher noise generation source than the traction transmission.

1. Hydromechanical

The primary potential noise source is the hydraulic units and the secondary source is the gears. Solution to the latter is represented by fairly well known techniques utilized and demonstrated in millions of automotive type transmission. Such techniques will be utilized in the recommended configuration to minimize noise. Of primary importance will be the gear tooth profile and speeds which will be similiar to present automotive transmissions.

Considerable effort has been expended in the last few years to understand and reduce hydraulic unit noise. The cause is fairly well known and techniques have been developed to minimize it. However it must be recognized that because of the large number of variables involved, the only positive assurance of meeting the required noise levels comes through actual hardware demonstrations.

The basic approach to the hydraulic noise reduction is to minimize the noise or energy level at the source, isolate or attenuate the conduction of the noise energy to the housing, and if necessary, attenuate the energy at the housing through isolation blankets before it can be conducted or radiated to the air and/or surrounding surfaces.

The hydraulic units represent the major noise source. This source is primarily related to the rate of generation of high pressure from low pressure and vice versa, the level of maximum pressure and the porting rate rotational speed. This process is accomplished within the hydraulic unit itself – pistons, cylinder block, and port plate. Considerable experience has been gained in the last few years in minimizing porting noise. This is accomplished by modifying the ports between the cylinder block and port plate to prevent large, abrupt pressure transients. Another means of minimizing the noise is to limit the maximum working pressure within the unit. In the recommended configuration, the working pressure is limited to 4500 psi, which would only occur with "floored accelerator" below about 20 MPH. Hydraulic unit operational speeds can be selected to insure the best noise characteristics. Therefore the variables involved are pressure level, rate of pressure increase or decrease in the individual pistons during parting, porting modifications, hydraulic unit speed and to some degree the stroke or displacement of the hydraulic units. Optimization of these parameters without degradation of hydraulic unit efficiency can only be accomplished through extensive testing.

Attenuation of the generated noise to the outside of the transmission is very important. The attenuation itself is very important but it is also important to insure that component natural frequencies are such that no resonants occur. Minimizing resonances will simplify energy attenuation techniques. Also noise frequencies should be kept as high as possible as attenuation is much easier at higher frequencies.

Air-borne noise within the transmission to the main housing has been suppressed by a deep-drawn sound shield made from a special laminated sandwich around the hydraulic unit rotating components. The oil pan is formed from the same material to prevent the air to fluid-borne noise from being transferred outside.

Structural-borne noise isolation is achieved by using a similar special laminated sandwich between the hydraulic porting plate, the intermediate support plate, and between the support plate and the main housing. The laminated sandwich is a composite of two metal plates with 2 layers of viscoelastic material between them seperated by a steel screen. This isolation material has a high crush force and good attenuation above 100 Hz. This double barrier should be very effective in minimizing noise propogation. In addition to this, a similar type of isolation is provided between the main input and output transmission bearings and the main housing, thus eliminating any "hard path" between the noise producing dynamic components and the main housing.

As indicated previously, noise tends to be in the category of "black art". Extensive testing and evaluation has defined design techniques which will minimizing noise. Although it is impossible to know at this time what the final noise level will be, it is anticipated that the noise requirements will be met.

2. Traction

The primary source of noise in the traction transmission are the gears.

The output gears are constant mesh and of the helical type similar to present automotive practice.

The input gear mesh to the input toroid is constant mesh and is shown as a spur gear. The input toroid cannot tolerate any external thrust. To ensure lowest possible noise generation, these input gears will be fine pitch, low pressure angle, and with a modified involute profile.

Should it become necessary to further reduce the noise from the input mesh, helical gears with thrust runners directly between the gears to cancel the resultant thrust will be used.

In addition to reducing gear noise to a minimum at its source, laminated sound insulating bearing sleeves are used to isolate the noise from the main housing.

D. MAINTAINABILITY

It is expected that either the tri-mode or the traction transmission should provide no greater maintainability problems than present automotive automatic transmissions. The only normal maintenance required will be to check the transmission oil level as is now done. Repair or overhaul of the transmission should not require any additional complication. The only "new to the business" component would be the hydraulic units or the toroids and rollers. It would be expected that these assemblies would be provided to the garage or overhaul shop as reworked assemblies similar to present torque converter assemblies.

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VI. PERFORMANCE

A. Introduction

The basic performance relating to the tri-mode hydromechanical and traction/torque converter transmission is given in this section. Transmission efficiency is shown as a function of road load at various vehicle speeds and also as a function of variable output loads from maximum to 10% of maximum.

It is important to recognize that although transmission efficiency is interesting the real measure of efficiency is that of the complete vehicle system – vehicle, engine, transmission.

This efficiency is reflected as fuel consumption over the defined driving cycle.

Acceleration data is also presented in several forms. This also represents total vehicle system performance.

B. Ground Rules and Transmission Parameter Summary

The following ground rules for all performance calculations were either specified or mutually agreed to by the Environmental Protection Agency.

1. Test vehicle weight = 4600 pounds (Prototype Vehicle Performance Specification).

2. Gross vehicle weight = 5300 pounds (Prototype Vehicle Performance Specification).

3. Vehicle road drag and air resistance (Prototype Vehicle Performance Specification) Frontal area = 20 sq. ft.; Coefficient of drag = .6

4. Rolling radius of wheels = 1.10 feet (assumed by Sundstrand).

5. Axle efficiency = .95 (assumed by Sundstrand).

6. Total rotating inertia of tires, wheels, and brakes for all four wheels = 11.2 ft-lb-sec^2 (assumed by Sundstrand).

7. Ambient air temperature was assumed by mutual agreement with EPA to be 85°F throughout the study. Although differences in air temperature do make a difference in air drag forces, their inclusion is somewhat meaningless without corresponding data on variation in engine performance with temperature, which was unavailable.

8. Accessory power requirements (Prototype Vehicle Performance Specification).
NOTE: Performance specification accessory losses representative of 6:1 engine speed range. For narrower speed range engines, the idle accessory power requirements were assumed unchanged, but the accessory power requirements at maximum engine speed were reduced proportionately.

9. Engine Speed – Power – Fuel Consumption data supplied by EPA. (See Appendices I-4 and I-5.)

10. Density of fuel used in fuel consumption calculations = 6.28 lb/gallon (Prototype Vehicle Performance Specification).

11. The driving cycle used to calculate fuel consumption was the Combined Duty Cycle. The Combined Duty Cycle consists of: a) the Federal Driving Cycle (see Appendix I-3); b) Simplified Suburban Route; and c) Simplified Country Route – (Prototype Vehicle Performance Specification).

12. Acceleration and fuel economy performance for the referenced typical 3-speed automatic transmission (as specified by EPA) is summarized in Appendix VI-3.

PARAMETER SUMMARY – TRI-MODE HYDROMECHANICAL TRANSMISSION

Transmission Input Speed (at 100% Engine Speed) . . . . 2336 RPM
Direction of Input Rotation (looking at Pad) . . . . Clockwise
Maximum Input Torque . . . . 315 ft-lb
Transmission Output Speed (89.2 MPH and 100% Engine Speed) . . . . 5114 RPM
Direction of Output Rotation (looking at Pad) . . . . Counterclockwise
Maximum Output Torque . . . . 592 ft-lb

Assumed Drive Axle Ratio 4.50:1

Maximum Vehicle Creep Speed at Engine Idle . . . . 0 MPH
Maximum Vehicle Reverse Speed . . . . 12.5 MPH

Hydraulic Unit:
- Displacement . . . . $1.50\ in^3/rev.$
- Rated Speed . . . . 5172 RPM
- Maximum Pressure . . . . 4500 psi

Clutch Type . . . . Multi-Plate, Flat Disk
Axial Piston, Hydraulic

Planetary Type . . . . Four Element Ravigneaux

Lubricating Fluid . . . . .Type F Automatic
Transmission Fluid
Cooler Size and Flow . . . . .Typical of Existing
Automatic Transmission Coolers

Transmission Weight (Dry) . . . . 92 pounds

PARAMETER SUMMARY – TRACTION DRIVE TRANSMISSION

Transmission Input Speed (at 100% Engine Speed)	14,000 RPM
Direction of Input Rotation (looking at Pad)	Clockwise
Maximum Input Torque	52.5 ft-lb
Transmission Output Speed (89.2 MPH and 100% Engine Speed)	5411 RPM
Direction of Output Rotation (looking at Pad)	Counterclockwise
Maximum Output Torque	550 ft-lb
Assumed Drive Axle Ratio	5.00:1
Maximum Vehicle Creep Speed at Engine Idle	13.0 MPH
Maximum Vehicle Reverse Speed	20.0 MPH (Arbitrary Limit)
Torque Converters:	
Diameter	6.12 in. (Rankine) 7.06 in. (Brayton)
Speed at 85 MPH	14,140 RPM
Power Absorbed at Engine Idle	3.0 HP
Stall Torque Ratio	3.00:1
Traction Drive:	
Maximum Input Speed	8000 RPM
Ratio Range	5.00:1
Lubricating Fluid	Sanotrack 40
Cooler Size and Flow	Typical of Existing Automatic Transmission Coolers
Transmission Weight (Dry)	77 pounds

C. Transmission Efficiency

Transmission efficiency has been calculated for both the tri-mode hydromechanical transmission and the traction drive transmission for both the Aerojet Rankine engine and the AiResearch Brayton engine. Conditions of output speed and load for which transmission efficiency tabulations and graphs have been calculated include: 1) The Federal Driving Cycle, 2) The Simplified Suburban Route, 3) The Simplified Country Route, 4) The Combined Driving Cycle (a combination of 1, 2, and 3), 5) Constant Vehicle Speed (cruise), and 6) Part Load (tractive effort at 100, 75, 50, 25, and 10 percent of maximum acceleration tractive effort).

The instantaneous transmission efficiency for each point in the driving cycle was calculated. Also, an accumulative efficiency, that is, an average efficiency, for the driving cycles was calculated and is presented as part of this report (see Table VI-1 and Figure VI-1 through VI-8). This average efficiency represents the quiotent of the accumulative power utilized over the given driving cycle and the accumulative power supplied.

Two computer programs, one for systems using hydromechanical transmissions and the other for systems using traction drive transmissions were used to simulate the vehicle, the engine, the transmissions, and the required duty cycles to generate the efficiency data. In the two programs, every effort was made to simulate the system's realistically. Therefore, the absolute values of efficiency presented in this report should be representative of actual hardware. It should also be emphasized that since the two programs were developed together, the relative efficiencies of the systems considered are also quite meaningful.

Transmission efficiency as used in this report is defined as the total power out of the transmission output divided by the total power into the transmission input. The primary or engine gear reduction has been assumed by Sundstrand to be part of the engine gearbox and is therefore not reflected in the transmission efficiency data presented here.

Calculations for the power losses contributed by gears and bearings, planetaries, open clutch spinning, charge pumps, and torque converters are well known and accepted.

The following paragraphs describe the background used in calculating hydraulic unit and traction unit efficiencies (or losses).

1. Hydraulic Unit Efficiency – Hydromechanical Transmission

The efficiency of each of the hydraulic units for any given working fluid viscosity is a function of the hydraulic working pressure, speed, and, in the case of the variable unit, actual displacement (or wobbler angle). This efficiency is markedly reduced below certain levels of working pressure and displacement (or wobbler angle). For example, at full stroke and 3000 psi working pressure, hydraulic pump efficiency is in the 88-94% range depending on speed, while at 500 psi and 1/4 stroke, the corresponding efficiency range is 25-45%.

The hydraulic unit design, and the predicted operating efficiencies used in this study are based on the testing and field experience of the past 30 years. The present axial piston-hydrostatic bearing design has evolved from past experience with many hydraulic unit configurations including radial piston units and anti-friction thrust bearing units, and has proved to be the best design in terms of cost, size, efficiency, and reliability.

TABLE VI-1
COMBINED DRIVING CYCLE TRANSMISSION EFFICIENCY

Cycle	Average Transmission Efficiency Over The specified Driving Cycles			
	Rankine Engine		Brayton Engine	
	HMT	TDT	HMT	TDT
Federal Driving Cycle	78%	67%	81%	71%
Simplified Suburban Route	72%	68%	80%	71%
Simplified Country Route	90%	85%	87%	86%
Combined Driving Cycle	81%	74%	83%	76%

HMT – Hydromechanical Transmission (Tri-Mode)

TDT – Traction Drive Transmission (with Torque Converter)

Vehicle Weight – 4600 lb. (Test Vehicle Weight)

Accessory Power – Air Conditioner On. Total Vehicle Accessory Power; 4 HP at Engine Idle, Linear to 4.83 HP (Rankine Engine) or 5.65 HP (Brayton Engine) at maximum Engine Speed. See Appendix I.

Atmospheric Conditions – 85^{o}F, 14.7 PSIA

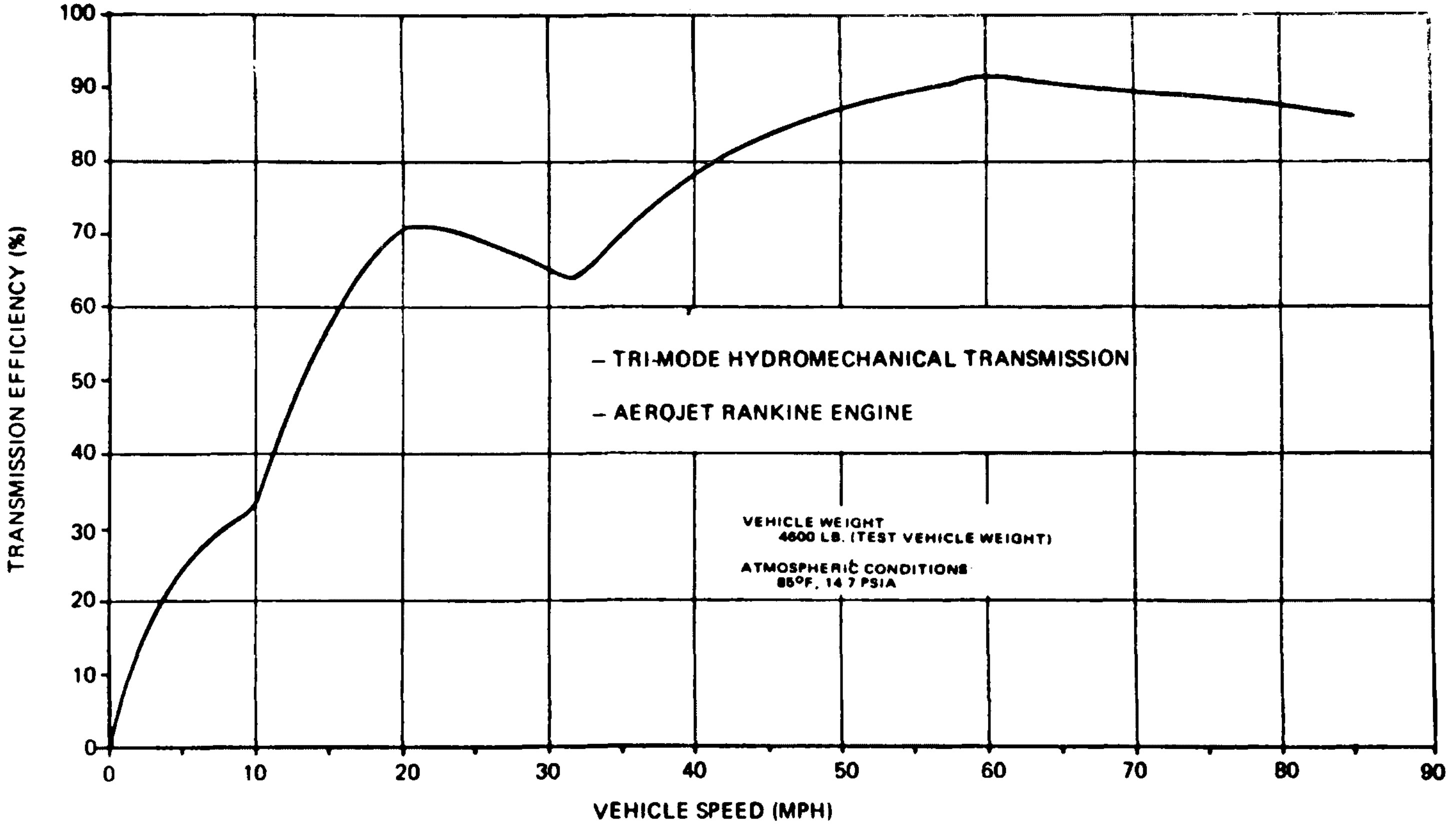

Figure VI-1 Tri-Mode Hydromechanical Transmission Efficiency at Constant Speed - Rankine Engine

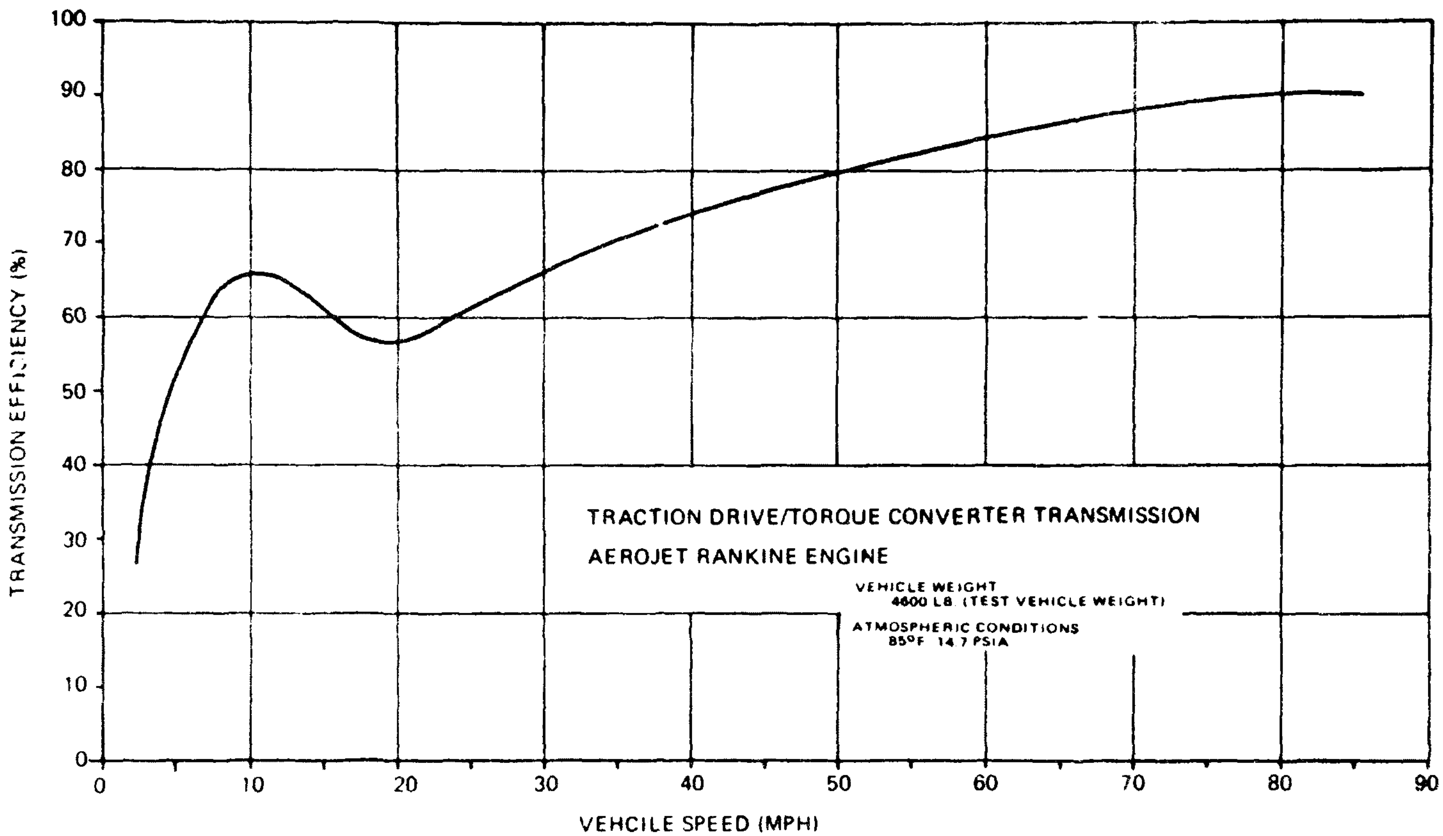

Figure VI-2 Traction Drive Transmission Efficiency at Constant Speed - Rankine Engine

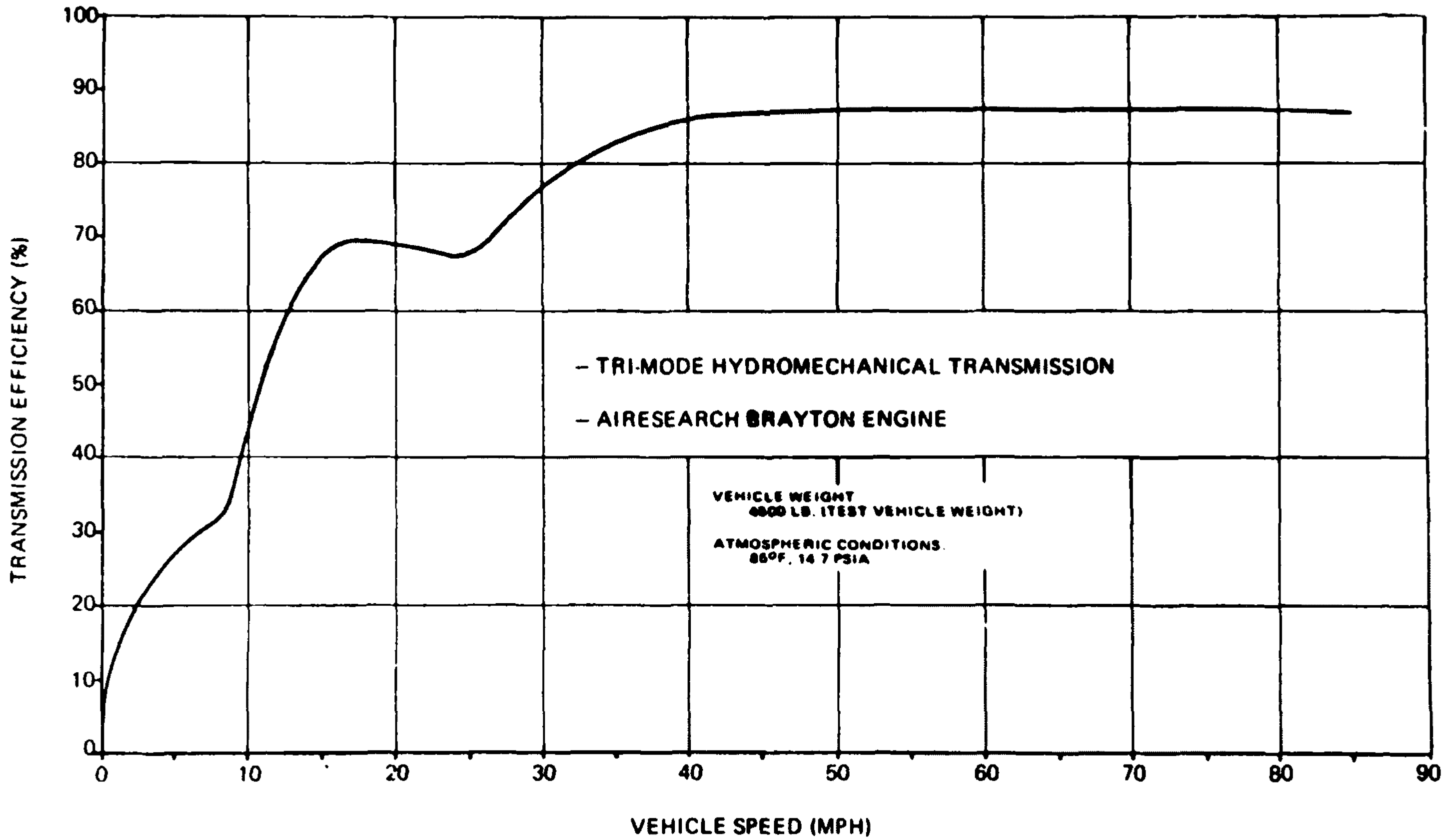

Figure VI-3 Tri-Mode Hydromechanical Transmission Efficiency at Constant Speed – Brayton Engine

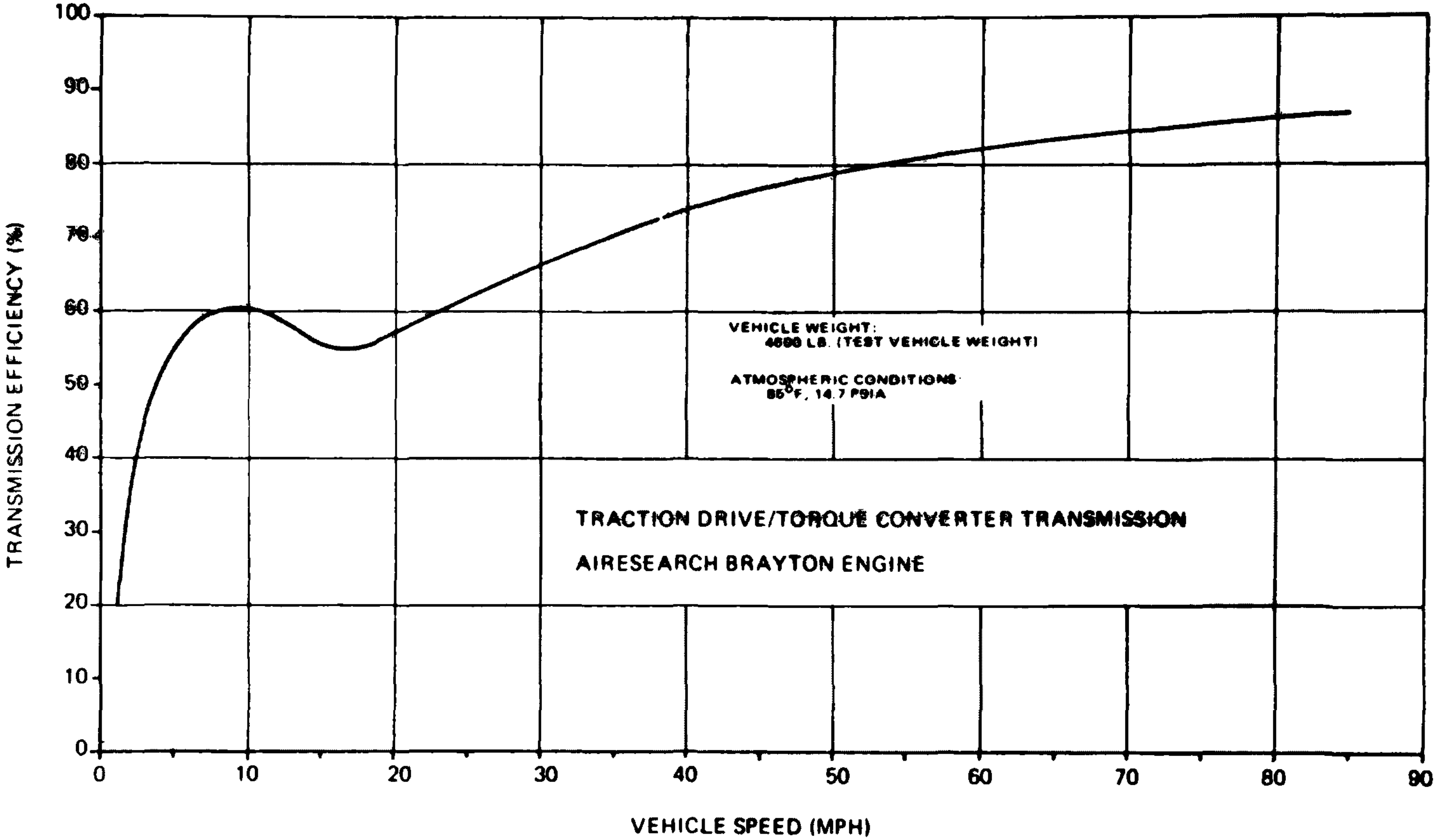

Figure VI-4 Traction Drive Transmission Efficiency at Constant Speed - Brayton Engine

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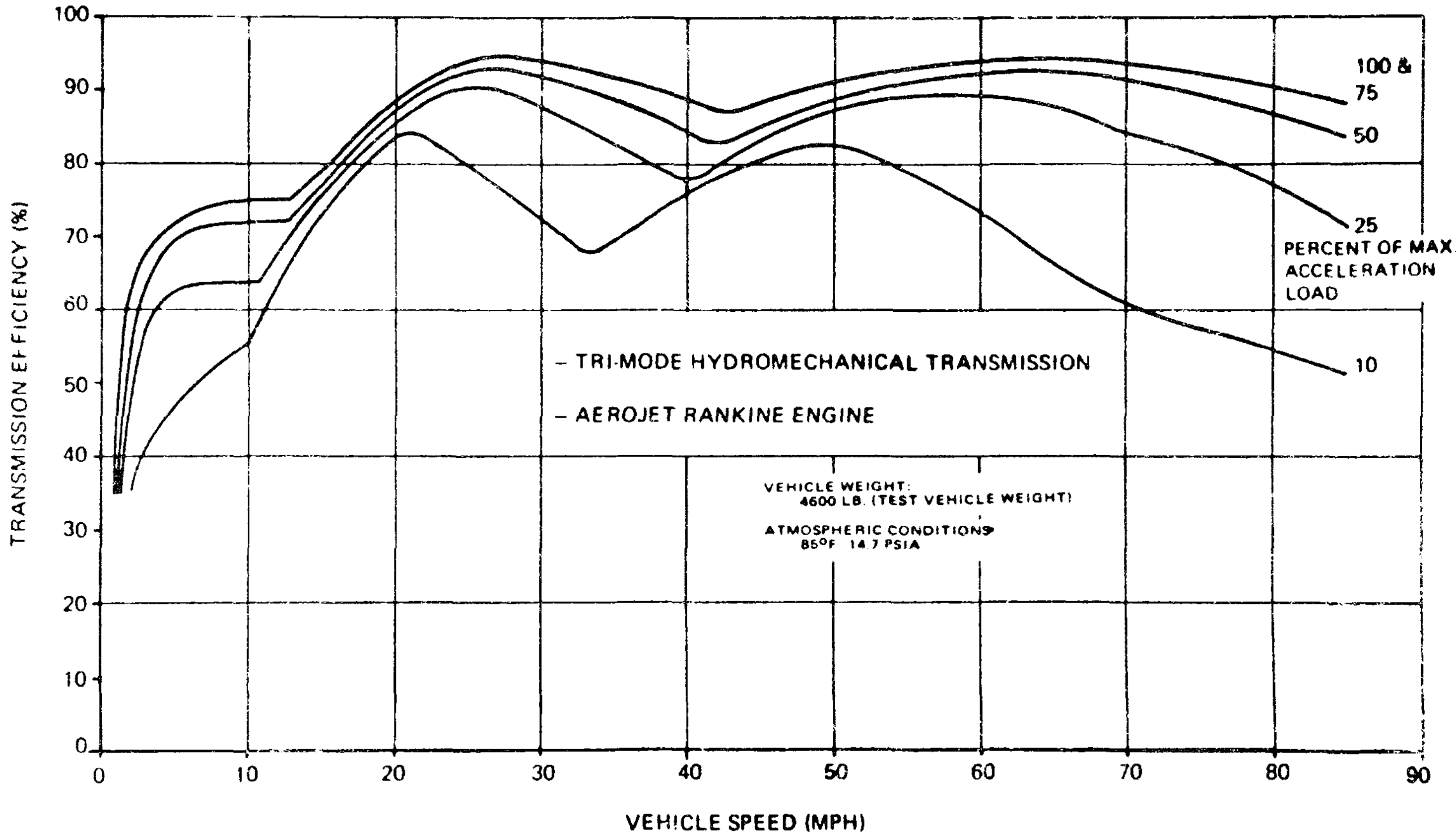

Figure VI-5 Hydromechanical Transmission Efficiency at Full and Part Loads – Rankine Engine

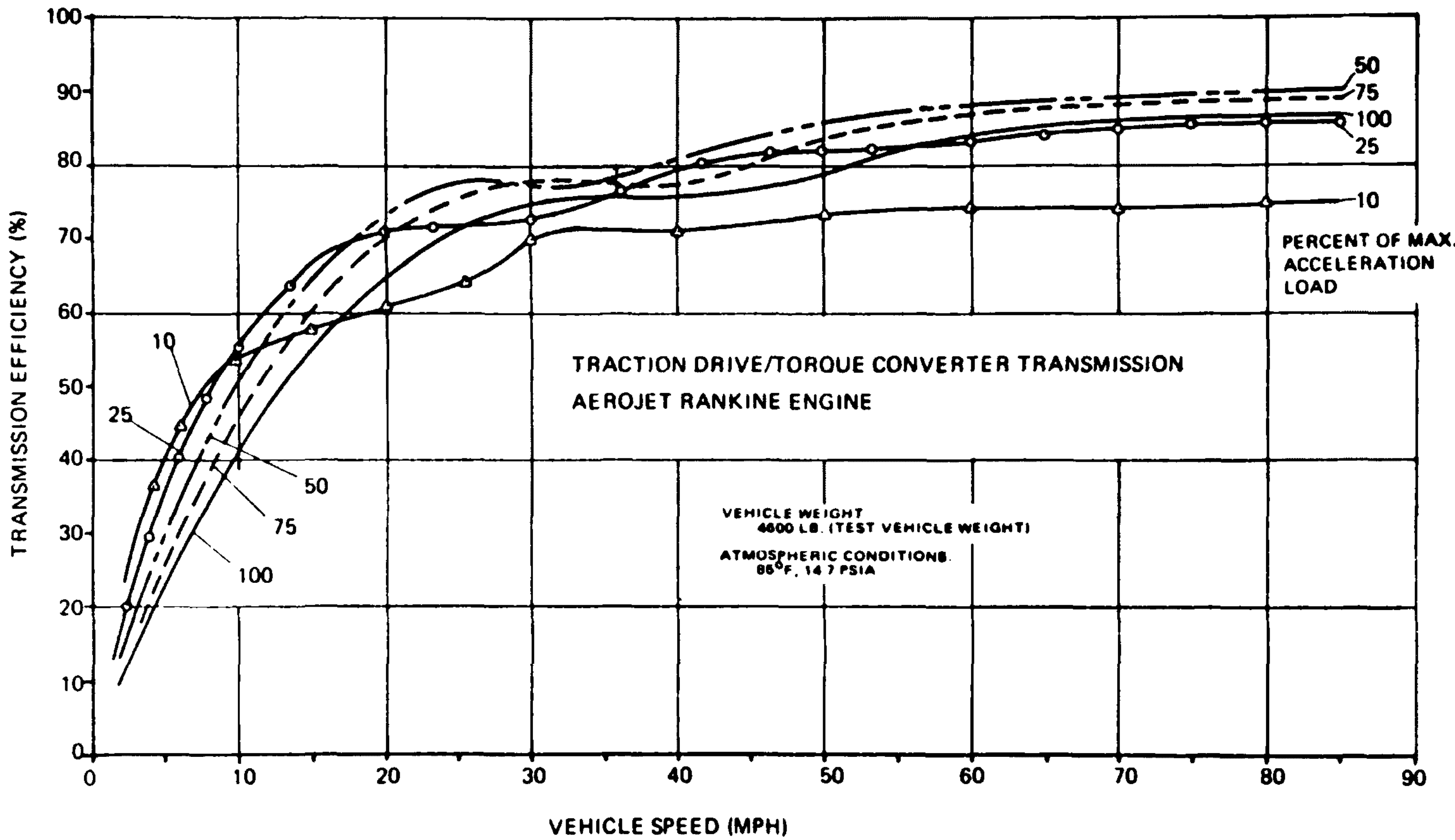

Figure VI-6 Traction Drive Transmission Efficiency at Full and Part Loads - Rankine Engine

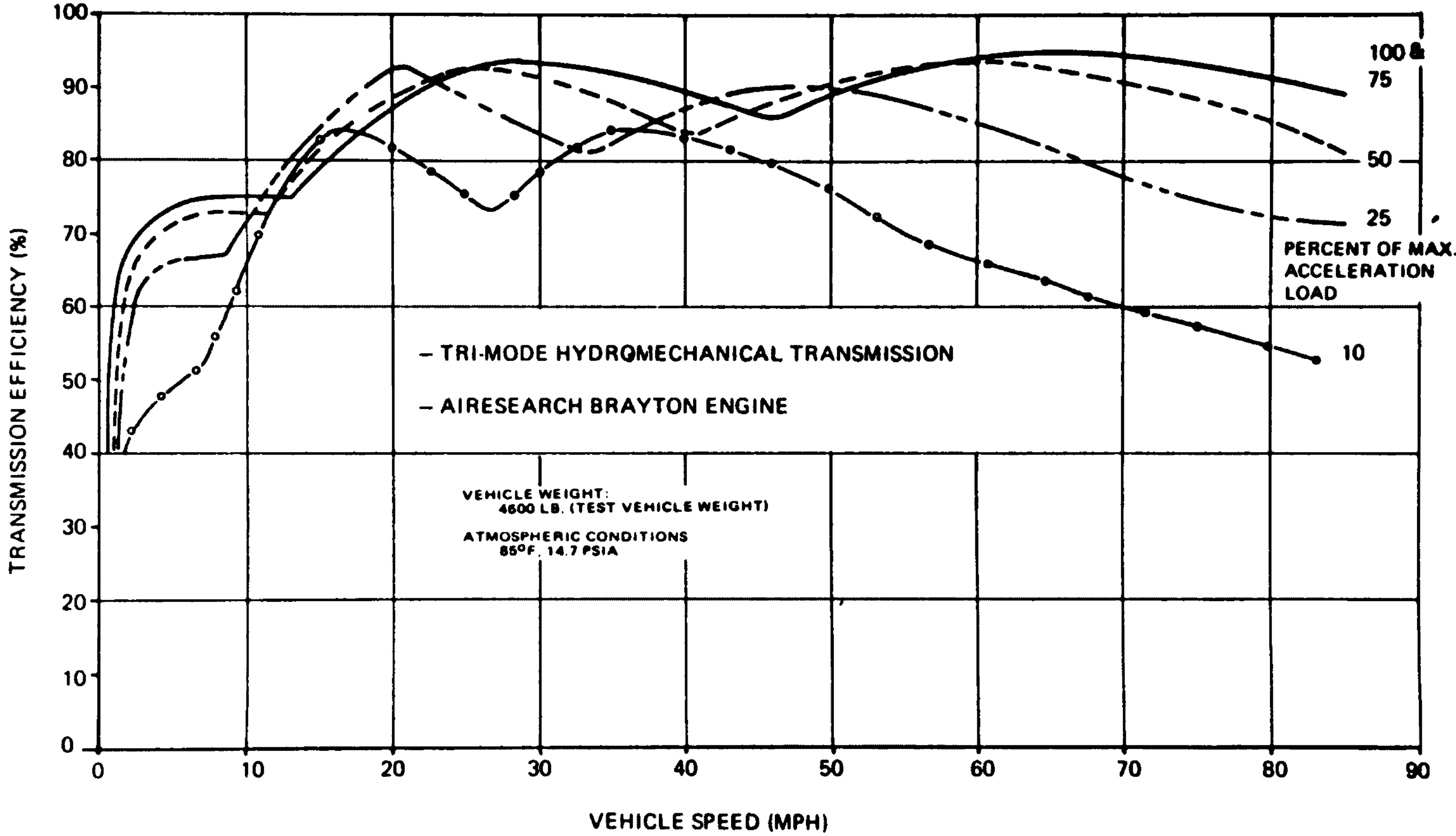

Figure VI-7 Hydromechanical Transmission Efficiency at Full and Part Load - Brayton Engine

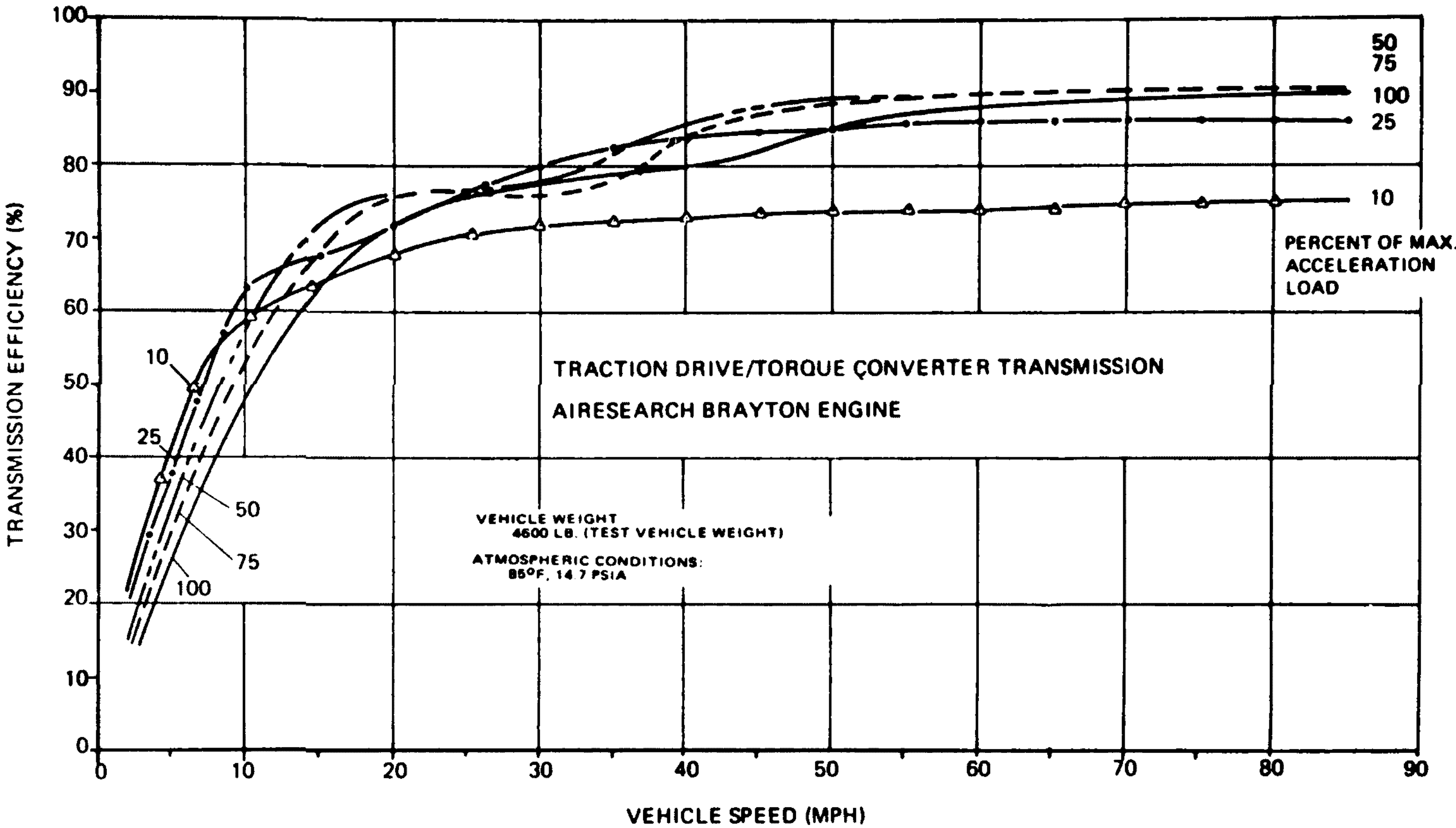

Figure VI-8 Traction Drive Transmission Efficiency at Full and Part Load - Brayton Engine

2. Traction Unit Efficiency – Traction Drive Transmission

There is always some degree of slip or creep at the roller/toroid interface. Since the contact area between the roller and the toroid has a finite area, the drive rollers tend to spin with respect to their toroidal races, so the motion of the rollers is rolling/spinning rather than pure rolling. Consequently, there is a power loss due to roller spin. Also rolling resistance is encountered between the toroids and the rollers, as well as in the rolling element bearings.

The overall efficiency of the traction unit is the product of the speed efficiency and the torque efficiency. The speed efficiency is a measure of slip. The torque efficiency, in general, is a measure of spin loss, rolling resistance, and windage.

The efficiency of the traction unit was calculated using experience gained from the development and testing of a Sundstrand toroidal type variable input speed constant output speed traction drive for aircraft applications. The Sundstrand efficiency data correlates well with data published by General Motors and Tracor on the efficiency of rolling contacts.

D. Grade and Acceleration Performance

Grade and acceleration performance was calculated for both transmissions and engines. The grade performance is a function of engine power and transmission efficiency. Acceleration performance is a function of engine power, transmission characteristics, drive line efficiency, tire adhesion, the engine time lag in going from idle to the maximum power condition, and the ratio of engine power going into accelerating the engine, to that which is accelerating the vehicle during the time lag period. This time lag itself is a function of the shape of the engine speed–torque curve, and the combined engine–transmission inertia. Because of the many variables involved several assumptions were made:

1) Maximum acceleration can be achieved by allowing the engine to accelerate to the maximum power condition unloaded and then applying maximum power to the wheels. For the required 0-60 mph acceleration time and the distance traveled in 10 seconds, an engine acceleration time of 0.5 seconds for the Rankine cycle engine and 1.0 seconds for the Brayton cycle engine was assumed based on discussion with the engine suppliers. For the 25-70 mph and 50-80 mph acceleration times, an engine acceleration time of 0.25 seconds for the Rankine cycle engine and 0.7 seconds for the Brayton engine was assumed. In practice, these time lags would probably be unacceptable from the "driver feel" point of view, and to overcome this, the engine power during this engine acceleration period would be split, some going to accelerate the engine, and some to accelerate the vehicle. The exact ratio of this power split would depend very much on "driver feel" and would be determined experimentally. Regardless of the split, it has been assumed that the 0-60 mph and 0-10 sec. acceleration performance would not be significantly different.

2) During the maximum acceleration from start conditions, the assumed vehicle weight and weight distribution shift combined with a reasonable tractive coefficient will allow a maximum tractive effort of 2500 lb. at the wheels without wheel slippage.

3) The reflected inertia of the transmission to the engine is very small and for the purpose of this study is ignored. The actual inertia is not only small but is reduced by the square of the gear ratio between the two. For example, this factor is 1/1295 for the hydromechanical transmission and 1/314 for the tractron drive transmission, when mated with the Brayton Cycle engine.

Calculation of the acceleration from start requirements when utilizing the Airesearch Brayton cycle engine indicated that the maximum power as defined in Appendix 1-5 was not sufficient. It was therefore necessary to assume a higher power to drive the vehicle and accessories – 145 HP for the hydromechanical transmission and 155 HP for the traction drive transmission. The power available from the Aerojet Rankine cycle engine, 148 HP at zero vehicle speed increasing to 163 HP at 85 mph, appears to be adequate to meet all performance requirement limits.

No problem was encountered in meeting the gradeability requirements. The maximum achievable vehicle speed along with the corresponding engine power requirements are tabulated in Table VI-2.

Table VI-2 also lists the actual acceleration performance of the various systems, taking into account engine lag. Also tabulated are the performance requirement limits.

Plots of vehicle speed and distance as a function of time during a maximum acceleration run are presented in Figure VI-9, VI-10, VI-11 and VI-12. The plots are based on a start from maximum power condition as can be achieved by locking the brakes.

Table VI-2 Idle Acceleration and Grade Performance

Performance Requirements		Weight	Rankine Engine HMT	Rankine Engine TDT	Brayton Engine HMT	Brayton Engine TDT
Idle						
Creep Speed	18 MPH (max)	1	0	13	0	13
Accel.						
Time to 60 MPH	13.5 sec (max)	1	11.1	13.2	12.6	12.6
Dist. to 10 sec.	440 ft. (min)	1	490	445	440	440
Time 25 → 70 MPH	15 sec (max)	1	12.3	14.4	15.0	14.6
Time 50 → 80 MPH	15 sec (max)	1	11.7	12.3	12.9	12.7
Dist. 50 → 80 MPH	1400 ft. (max)	1	1125	1175	1235	1230
Grade Velocity						
30%	5 MPH (min)	2	29	20	26	23
5%	65 MPH (min)	2	85	84	80	81
0%	85 MPH (min)	1	85 (91 HP)	85 (86 HP)	85 (93 HP)	85 (89 HP)

HMT – Hydromechanical Transmission (Tri-Mode)

TDT – Traction Drive Transmission (with torque converter)

1 at 4600 lb (test vehicle weight)

2 at 5300 lb (gross vehicle weight)

Accessory Power – Air Conditioning On. Total Vehicle Accessory Power; 4.0 HP at Engine Idle, Linear to 4.83 HP (Rankine Engine) or 5.65 HP (Brayton Engine) at Max Eng. Speed. See Appendix I

Atmospheric Conditions – 85°F, 14.7 PSIA

Engine Power: In accordance with Figure I-7 except as noted.

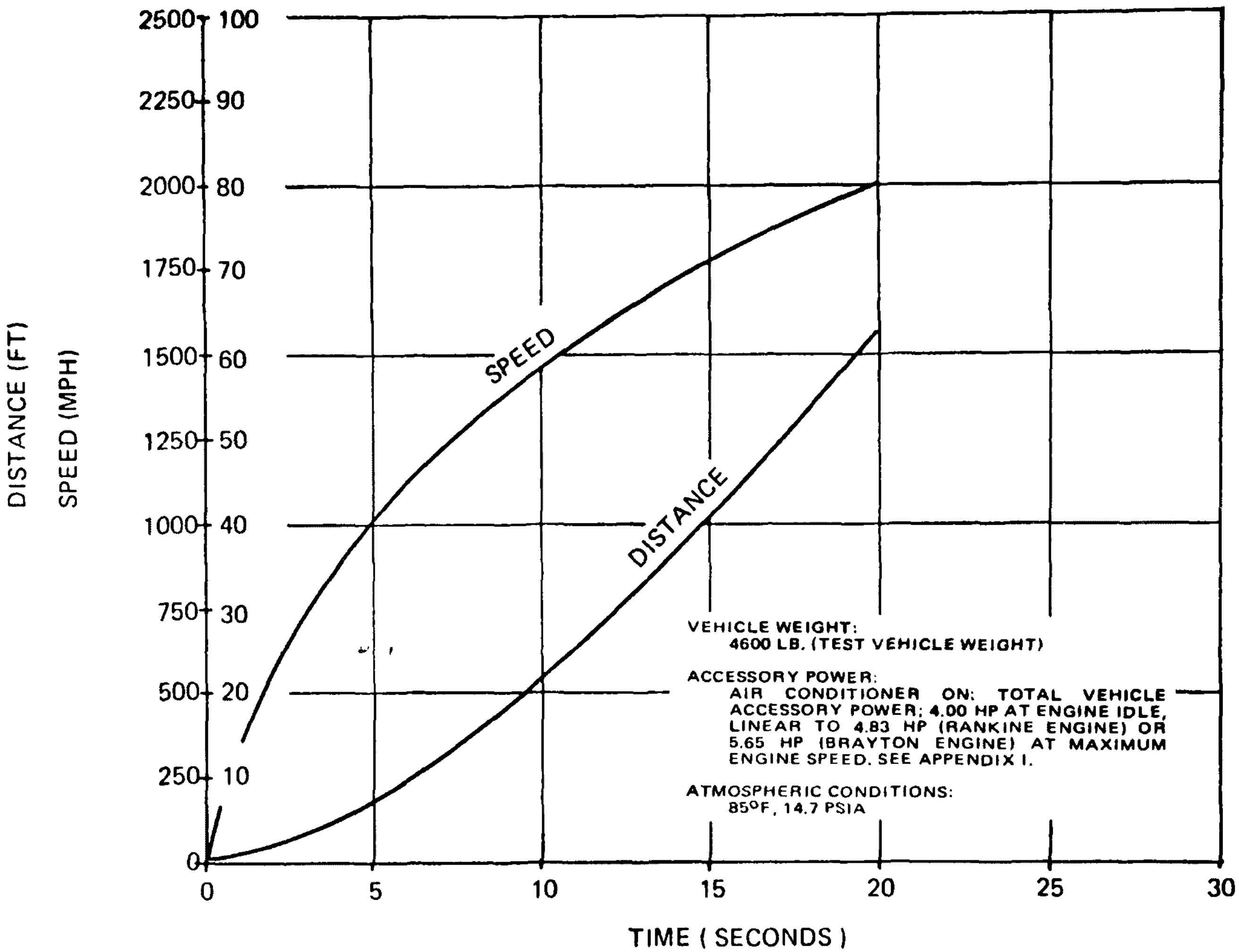

Figure VI-9 Hydromechanical Transmission Acceleration
Rankine Engine

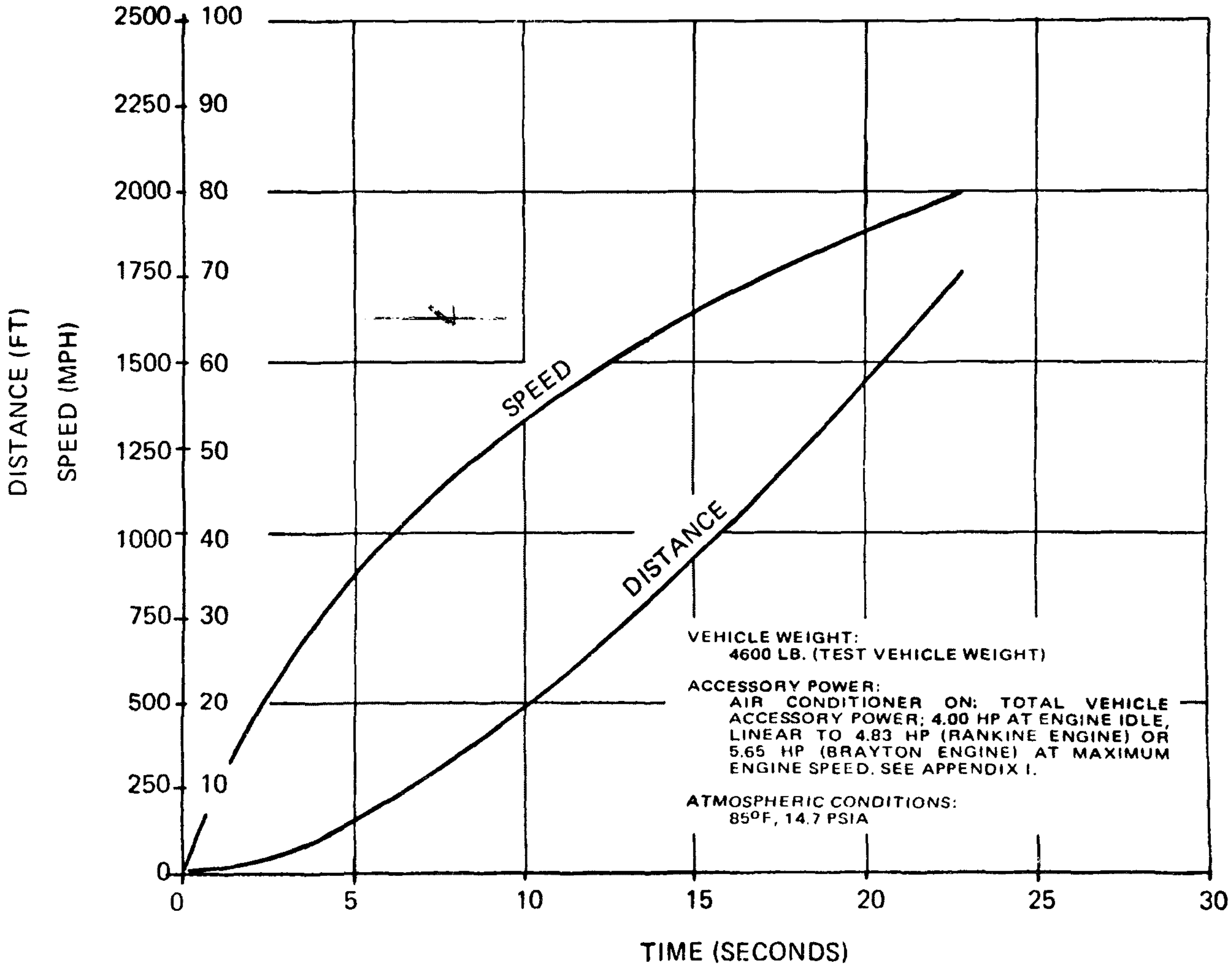

Figure VI-10 Traction Drive Transmission Acceleration Rankine Engine

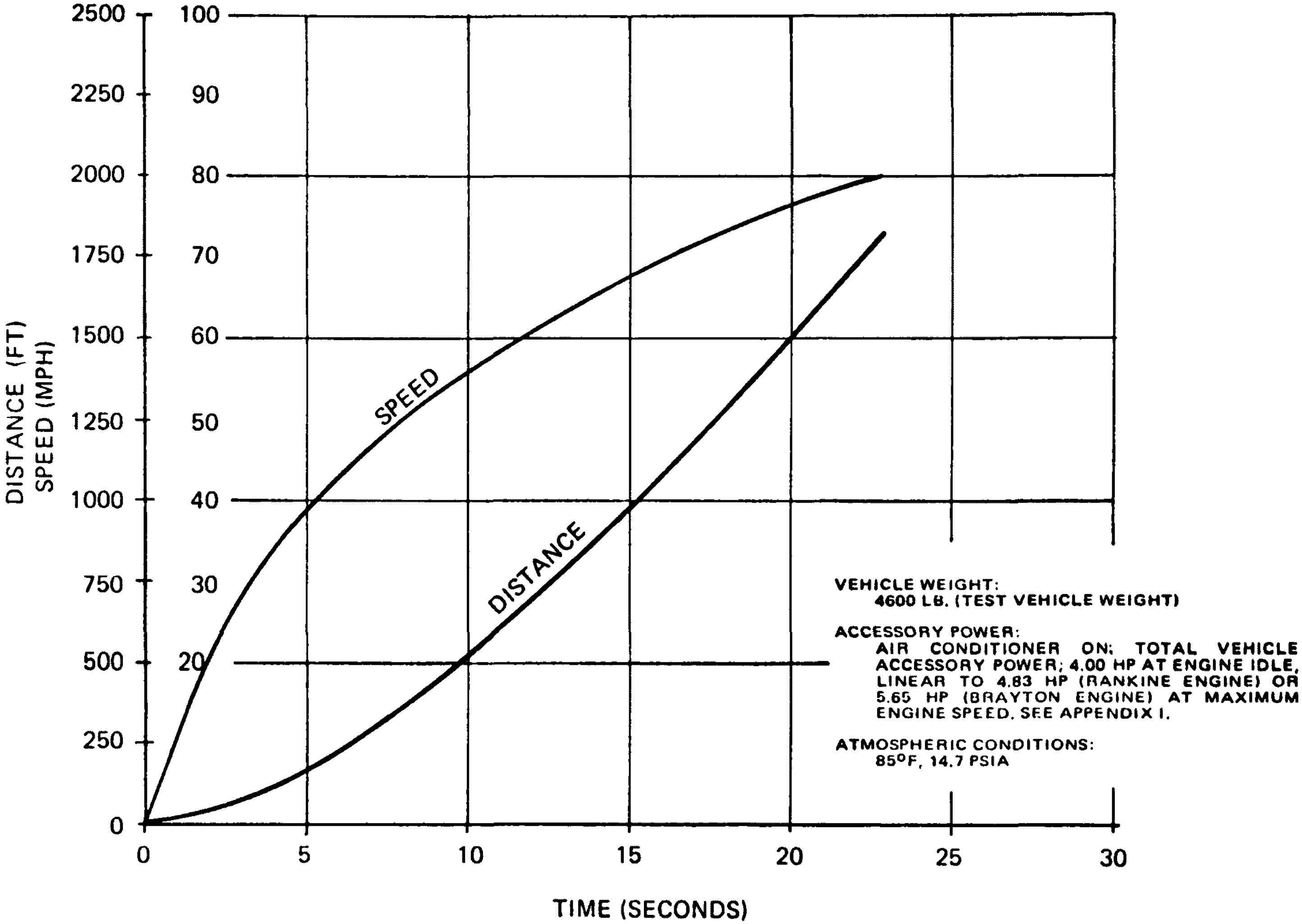

Figure VI-11 Hydromechanical Transmission Acceleration Brayton Engine

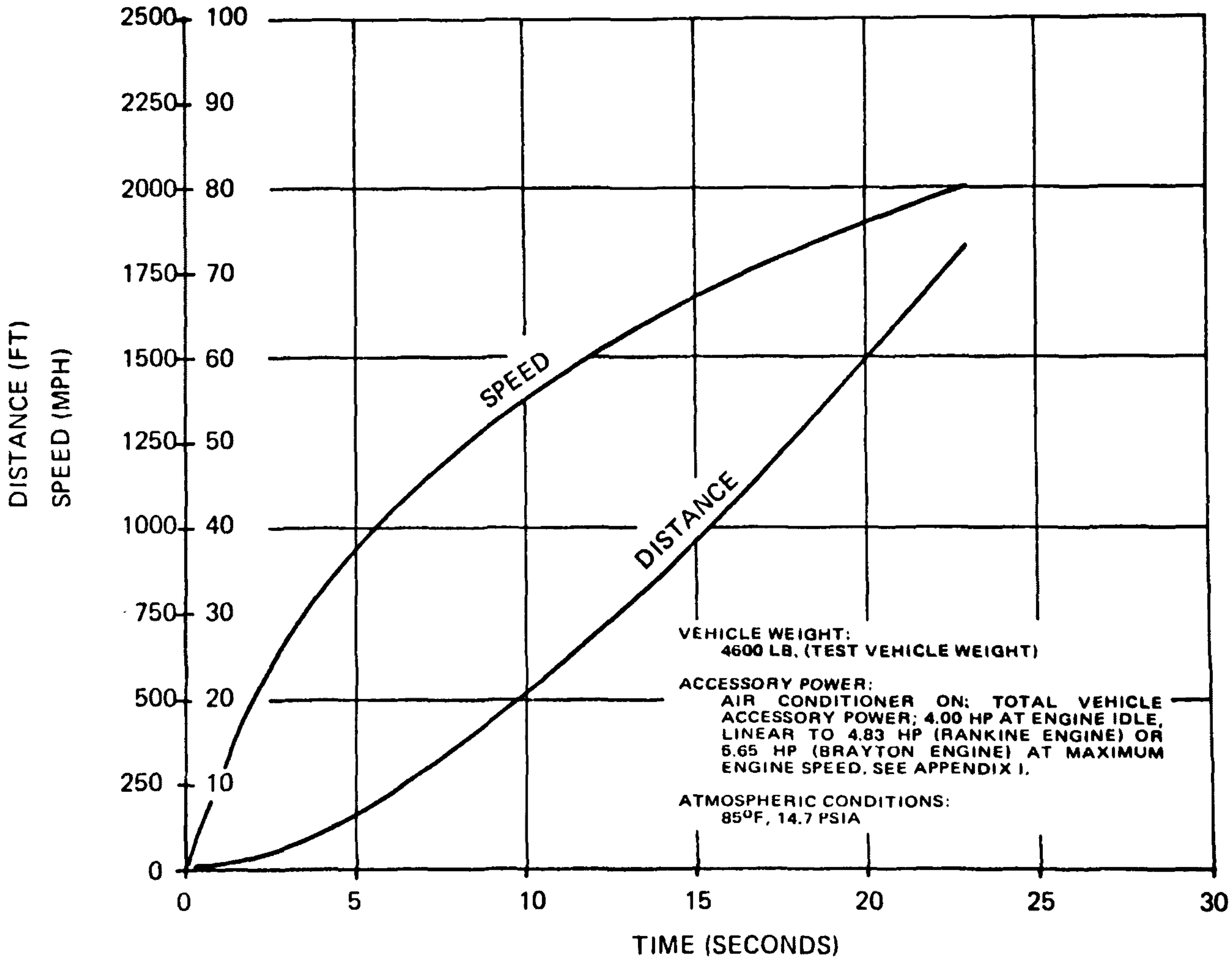

Figure VI-12 Traction Drive Transmission Acceleration Brayton Engine

E. Fuel Consumption at Constant Speed and Full and Part Loads

Constant speed fuel consumption and fuel consumption at full and part load in miles per gallon were calculated for systems using the tri-mode hydromechanical transmission and the traction drive transmission for both the Aerojet Rankine engine and the AiResearch Brayton engine.

Constant speed fuel consumption is defined as fuel consumption at zero vehicle acceleration.

Plots of constant speed fuel consumption based upon the simulated vehicle with its various configurations of transmissions and engines are presented in this section in Figures VI-13, VI-14, VI-15 and VI-16. Also presented here are plots of instantaneous fuel consumption in miles per gallon vs. vehicle speed at maximum tractive effort, as well as at 75%, 50%, 25%, and 10% of maximum tractive effort. This full and part load fuel consumption data is presented in Figures VI-17, VI-18, VI-19 and VI-20.

F. Fuel Consumption Summary

Average fuel consumption in terms of miles per gallon and BTU/mile has been calculated for both the tri-mode hydromechanical and the traction drive-torque converter transmission for both the Aerojet Rankine engine systems and the AiResearch Brayton engine systems.

A summary of the average fuel consumption over the Federal Driving Cycle both with and without the air conditioner operating is given in Table VI-3.

A breakdown of the fuel consumption over the Combined Driving Cycle is presented in Table VI-4. The Combined Driving Cycle consists of the Federal Driving Cycle, the Simplified Suburban Route, and the Simplified Country Route. Also presented in Table VI-4 is the fuel consumption that could be expected from an ideal transmission. The ideal transmission is infinitely variable, 100% efficient, and has no spin loss to load the engine at idle.

A fuel consumption breakdown of the Combined Driving Cycle in terms of BTU/mile is presented in Table VI-5.

Vehicle range, which is a function of fuel consumption, was calculated and is presented in Table VI-6. It was assumed that there was 25 gallons of fuel available initially. Vehicle range has been calculated for the Federal Driving Cycle at a constant 70 mph cruise, both with and without the air conditioner operating.

G. Tractive Effort Limits

Tractive effort limits for the simulated vehicle are established by a number of parameters. Among these are road adhesion of the tires, vehicle weight and configuration, maximum available transmission input power, maximum available transmission output torque, and transmission efficiency.

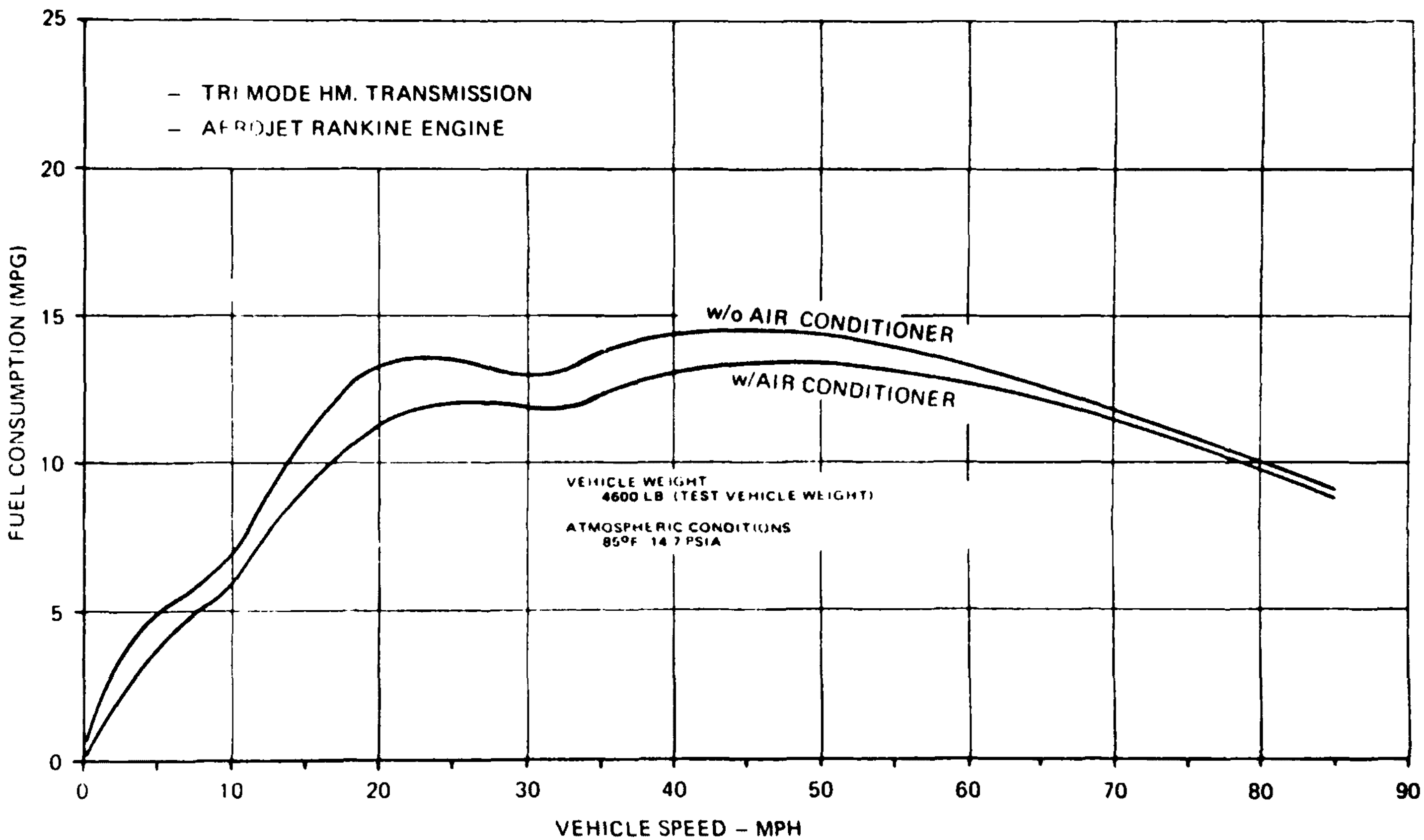

Figure VI-13 Hydromechanical Transmission Fuel Consumption at Constant Speed - Rankine Engine

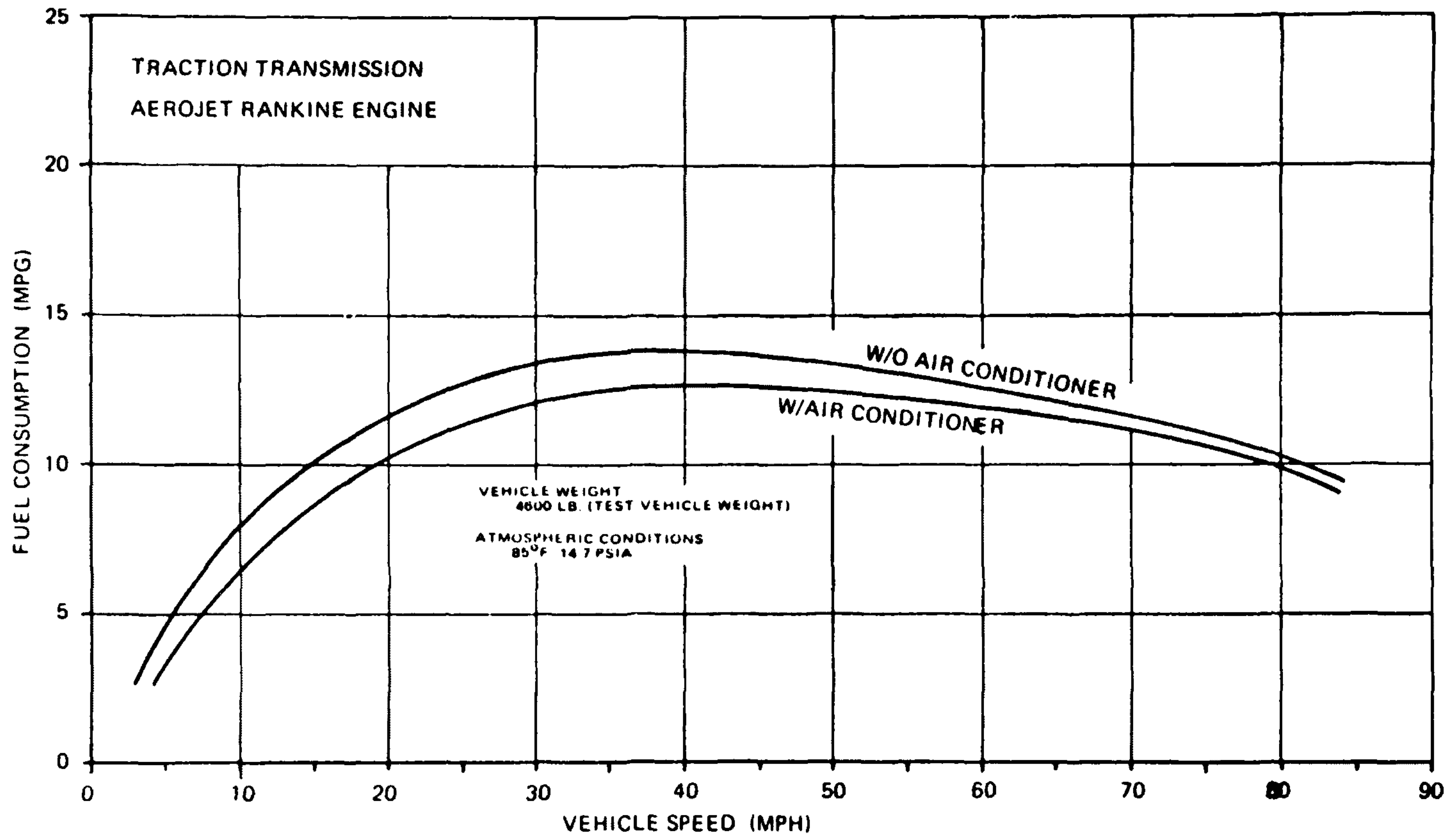

Figure VI-14 Traction Drive Transmission Fuel Consumption at Constant Speed - Rankine Engine

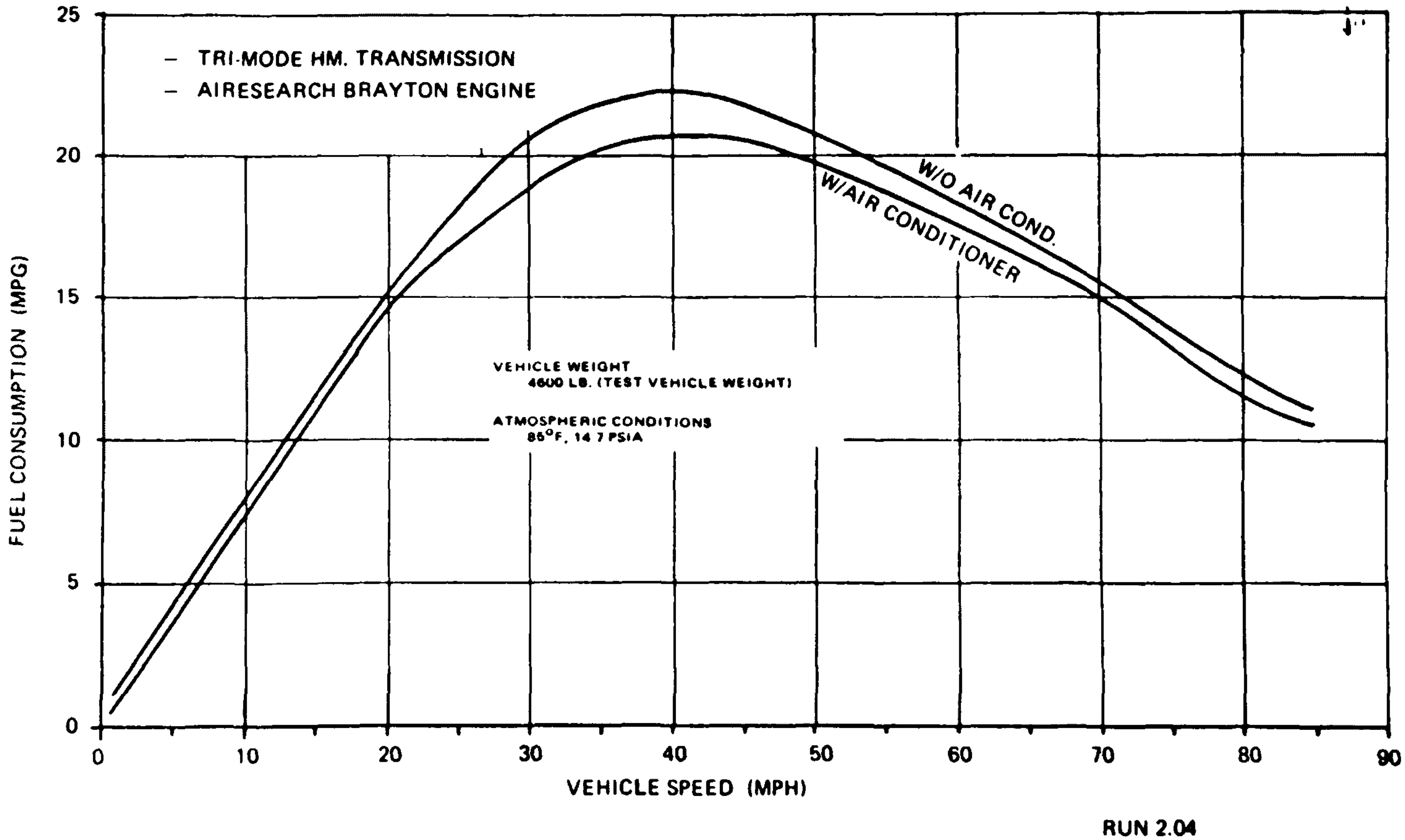

Figure VI-15 Hydromechanical Transmission Fuel Consumption at Constant Speed – Brayton Engine

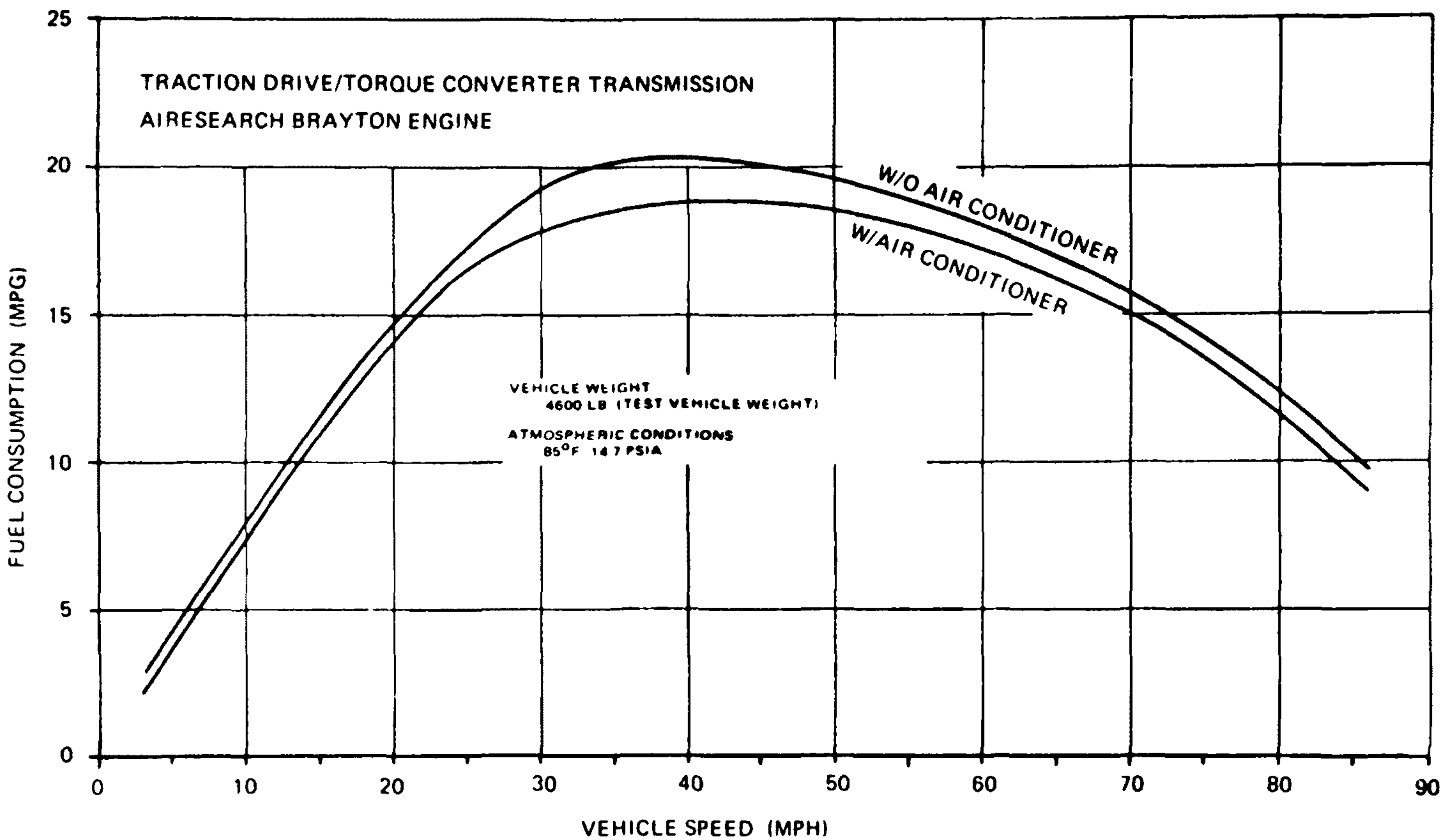

Figure VI-16 Traction Drive Transmission Fuel Consumption at Constant Speed - Brayton Engine

Sundstrand Aviation

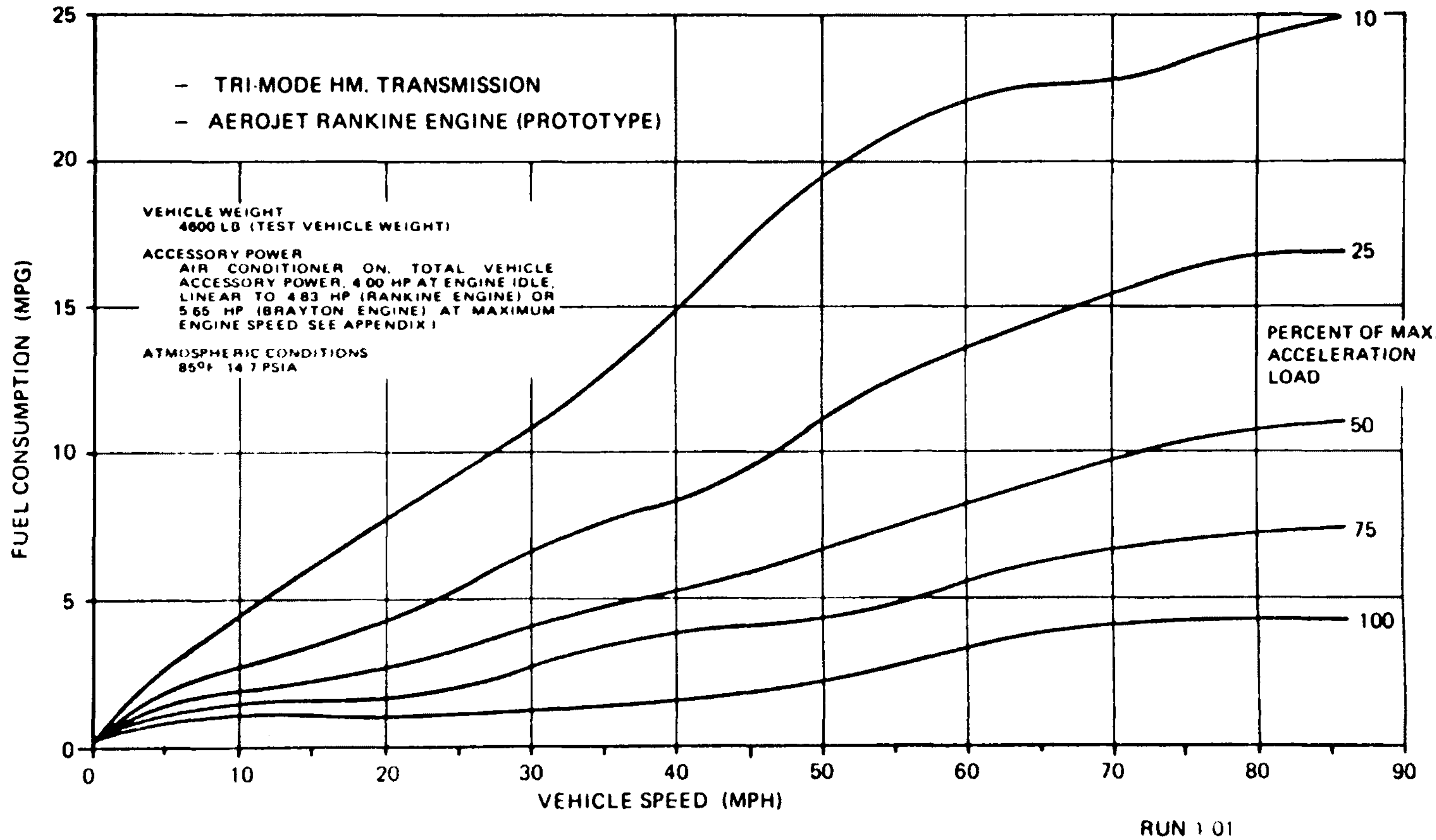

Figure VI-17 Hydromechanical Transmission Fuel Consumption at Full and Part Load – Rankine Engine

Sundstrand Aviation

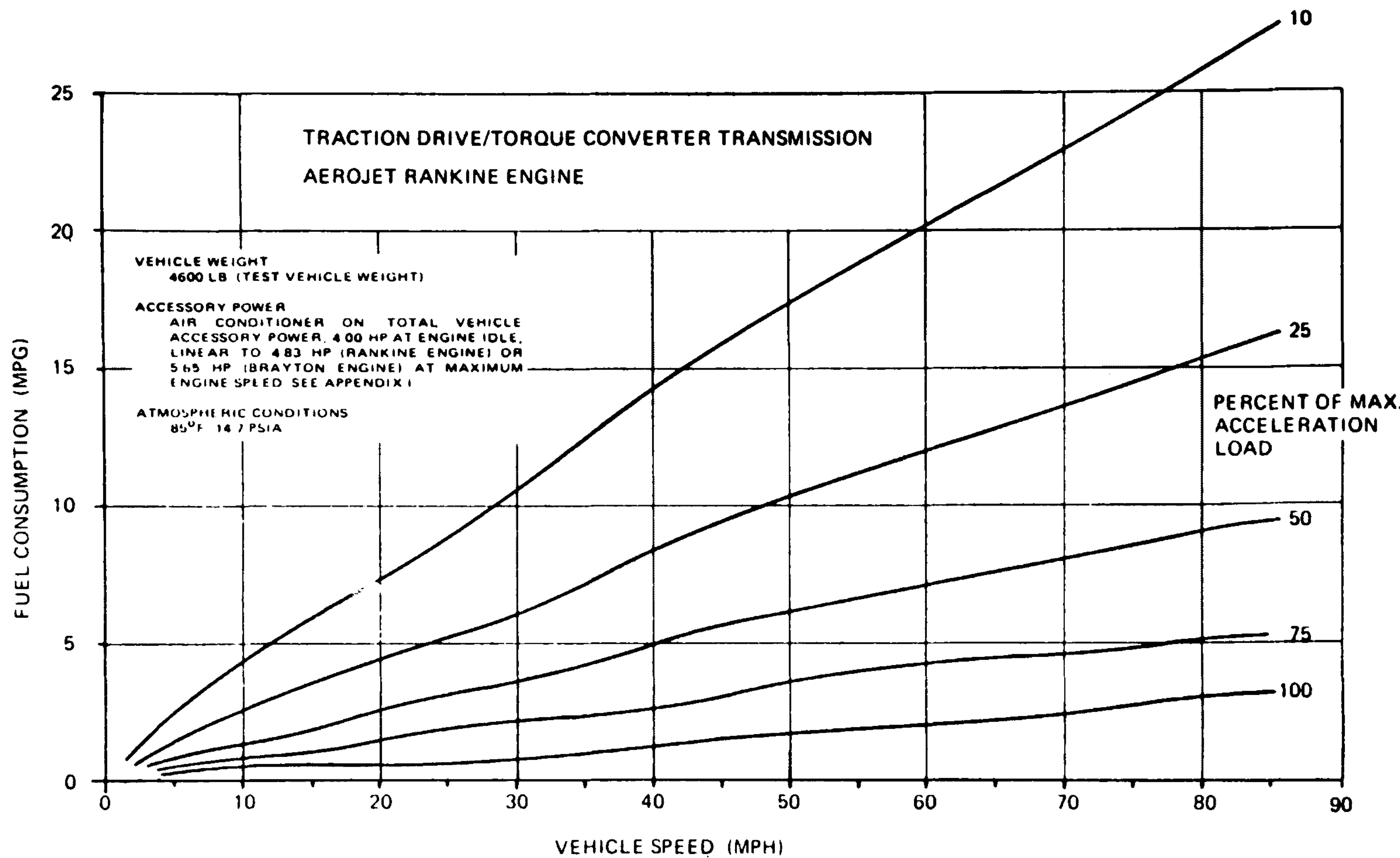

Figure VI-18 Traction Drive Transmission Fuel Consumption at Full and Part Load - Rankine Engine

Sundstrand Aviation

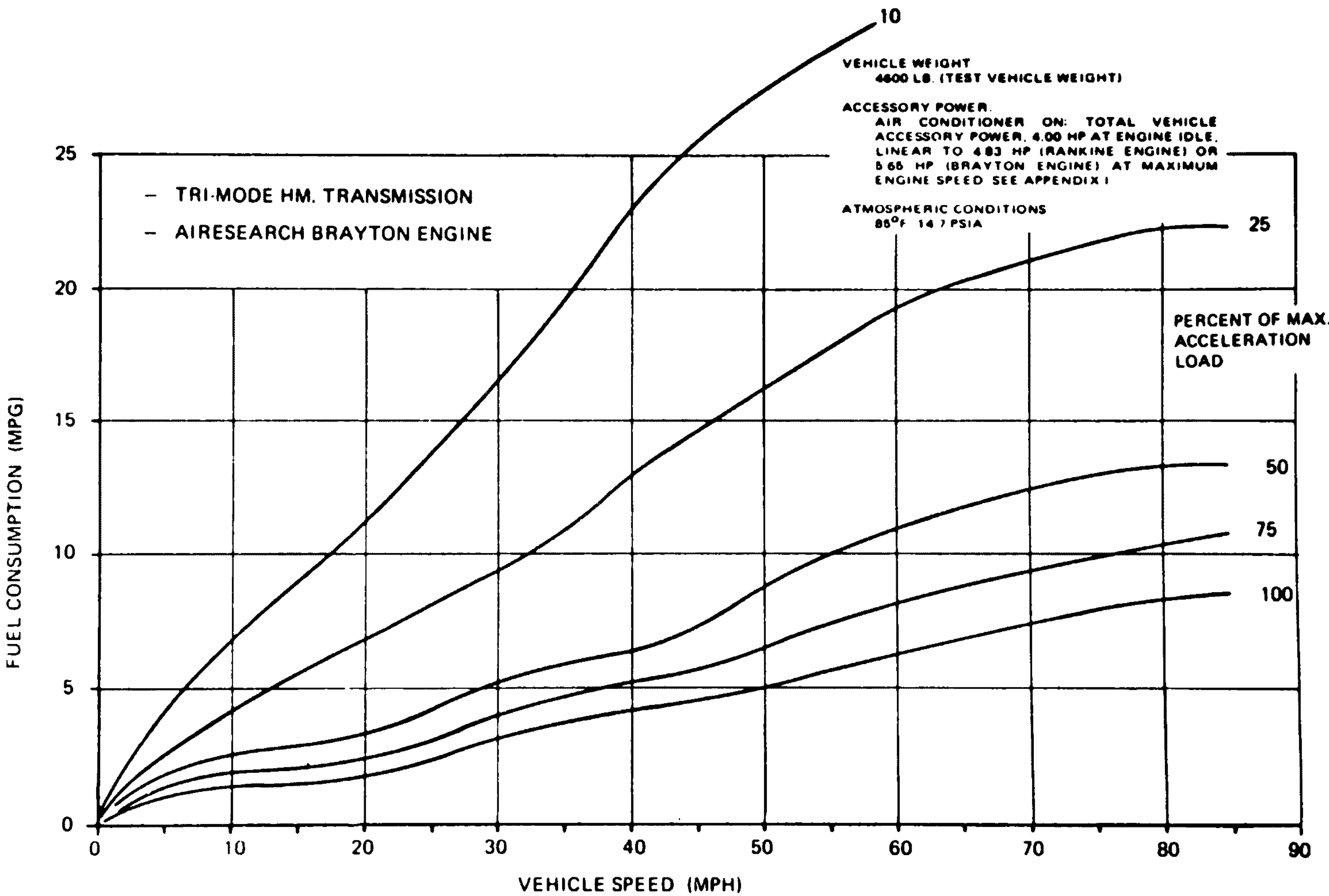

Figure VI-19 Hydromechanical Transmission Fuel Consumption at Full and Part Load – Brayton Engine

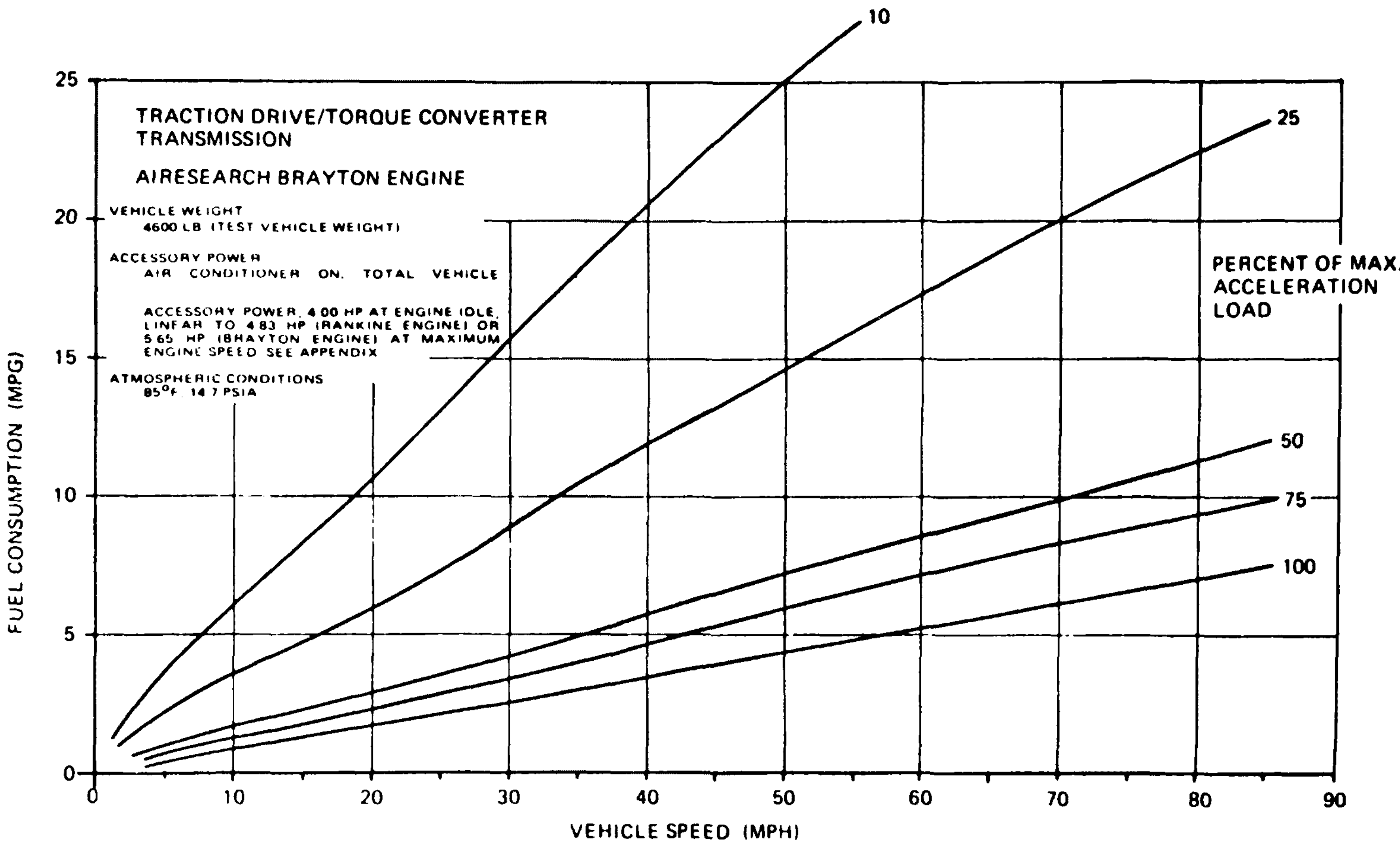

Figure VI-20 Traction Drive Transmission Fuel Consumption at Full and Part Load - Brayton Engine

TABLE VI-3

FEDERAL DRIVING CYCLE FUEL CONSUMPTION WITH AND WITHOUT AIR CONDITIONING

	Average Fuel Consumption Over The Federal Driving Cycle With and Without Air Conditioner On			
Accessory Power	Rankine Engine		Brayton Engine	
	HMT	TDT	HMT	TDT
Air Conditioner On	8.33	7.65	12.91	12.03
Air Conditioner Off	9.43	8.59	14.29	13.31

HMT – Hydromechanical Transmission (Tri-Mode)

TDT – Traction Drive Transmission (with Torque Converter)

Vehicle Weight – 4600 lb (Test Vehicle Weight)

Accessory Power – Air Conditioner On: Total Vehicle Accessory Power; 4.00 HP at Engine Idle, Linear to 4.83 HP (Rankine Engine) or 5.65 HP (Brayton Engine) at maximum engine speed.

Air Conditioner Off: Total Vehicle Accessory Power; 2.00 HP at Engine Idle, Linear to 2.23 HP (Rankine Engine)or 2.45 HP (Brayton Engine) at maximum engine speed. See Appendix I.

Atmospheric Conditions – 85^{o}F, 14.7 PSIA

TABLE VI-4
COMBINED DRIVING CYCLE FUEL CONSUMPTION

	Fuel Consumption In Miles Per Gallon Over the Individual And Combined Driving Cycles					
	Rankine Engine			Brayton Engine		
Cycle	HMT	TDT	Ideal	HMT	TDT	Ideal
Federal Driving Cycle	8.33	7.65	10.19	12.91	12.03	15.08
Simplified Suburban Route	12.23	11.77	14.68	18.33	17.30	19.84
Simplified Country Route	12.30	11.78	13.30	16.91	16.62	19.28
Combined Driving Cycle	10.62	10.01	12.43	15.71	14.95	17.79

HMT – Hydromechanical Transmission (Tri-Mode)

TDT – Traction Drive Transmission (with Torque Converter)

IDEAL – Hypothetical 100% Efficient Transmission

Vehicle Weight – 4600 lb. (Test Vehicle Weight)

Accessory Power – Air Conditioner On. Total Vehicle Accessory Power; 4.00 HP at Engine Idle, Linear to 4.83 HP (Rankine Engine) or 5.65 HP (Brayton Engine) at maximum engine speed. See Appendix I.

Atmospheric Conditions – $85^{\circ}F$, 14.7 PSIA

TABLE VI-5
COMBINED DRIVING CYCLE ENERGY CONSUMPTION

Cycle	Average Energy Consumption In BTU's Per Mile Over The Individual And Combined Driving Cycles			
	Rankine Engine		Brayton Engine	
	HMT	TDT	HMT	TDT
Federal Driving Cycle	13947.	15187.	8999.	9658.
Simplified Suburban Route	9500.	9871.	6338.	6716.
Simplified Country Route	9446.	9862.	6870.	6990.
Combined Driving Cycle	10940.	11606.	7395.	7771.

HMT – Hydromechanical Transmission (Tri-Mode)

TDT – Traction Drive Transmission (with Torque Converter)

Fuel Heating Value – 18500 BTU/LB

Fuel Density – 6.28 LB/GAL.

$$\frac{BTU}{MI} = \frac{BTU}{LB} \times \frac{LB}{GAL} \div \frac{MI}{GAL}$$

Vehicle Weight – 4600 LB (Test Vehicle Weight)

Accessory Power – Air Conditioner On. Total Vehicle Accessory Power; 4.00 HP at Engine Idle, Linear to 4.83 HP (Rankine Engine) or 5.65 HP (Brayton Engine) at maximum engine speed. See Appendix I.

Atmospheric Conditions – 85°F, 14.7 PSIA

TABLE VI-6
VEHICLE RANGE AT FEDERAL DRIVING CYCLE AND AT CRUISE

		Vehicle Range In Miles with and without Air Conditioning (25 gallons of fuel)			
		Rankine Engine		Brayton Engine	
Cycle	Accessory Power	HMT	TDT	HMT	TDT
Federal Driving Cycle	Air Conditioner On	208	191	323	301
Federal Driving Cycle	Air Conditioner Off	236	215	357	333
70 MPH Cruise	Air Conditioner On	285	280	376	378
70 MPH Cruise	Air Conditioner Off	324	291	393	398

HMT – Hydromechanical Transmission (Tri-Mode)

TDT – Traction Drive Transmission (with Torque Converter)

Fuel Tank Capacity – 25 gallons

The EPA "Prototype Vehicle Performance Specification" (Appendix I) Requires 200 Miles Minimum Range

Miles = MPG x Gallons

Vehicle Weight – 4600 lb (Test Vehicle Weight)

Accessory Power – Air Conditioner On: Total Vehicle Accessory Power; 4.00 HP at Engine Idle, Linear to 4.83 HP (Rankine Engine) or 5.65 HP (Brayton Engine) at Maximum Engine Speed
Air Conditioner Off: Total Vehicle Accessory Power; 2.00 HP at Engine Idle, Linear to 2.23 HP (Rankine Engine or 2.45 HP (Brayton Engine) at Maximum Engine speed. See Appendix I

Atmospheric Conditions – 85^{o}F, 14.7 PSIA

It was assumed that the absolute maximum tractive effort that the wheels could produce without slipping was 2500 pounds.

In the case of the systems with hydromechanical transmissions, a tractive effort of about 2400 pounds corresponds to a hydraulic system pressure of 4500 psi. For life, noise, and reliability reasons, it is recommended that system pressure be limited to 4500 psi at maximum tractive effort.

Systems with traction drive transmissions are limited by road adhesion to 2500 pounds, with the assumed 155 HP Brayton engine, or to 2450 pounds at start-up by engine power limitations with the Rankine engine.

At higher vehicle speeds, the tractive effort available from either transmission type is limited by engine power limitations and transmission efficiency. See graphs, Figures VI-21 and VI-22.

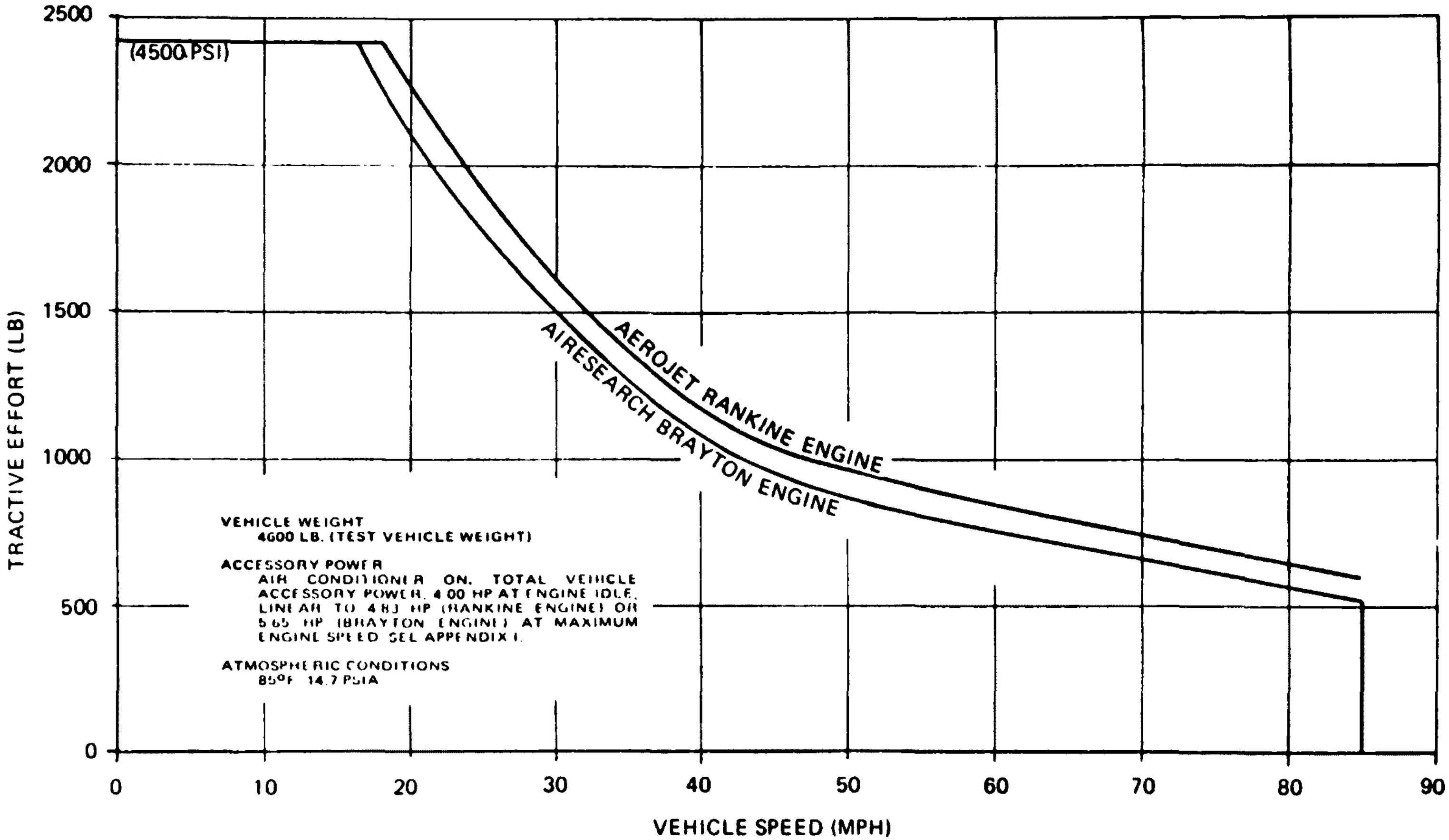

Figure VI-21 Maximum Tractive Effort - Hydromechanical Transmission

Sundstrand Aviation

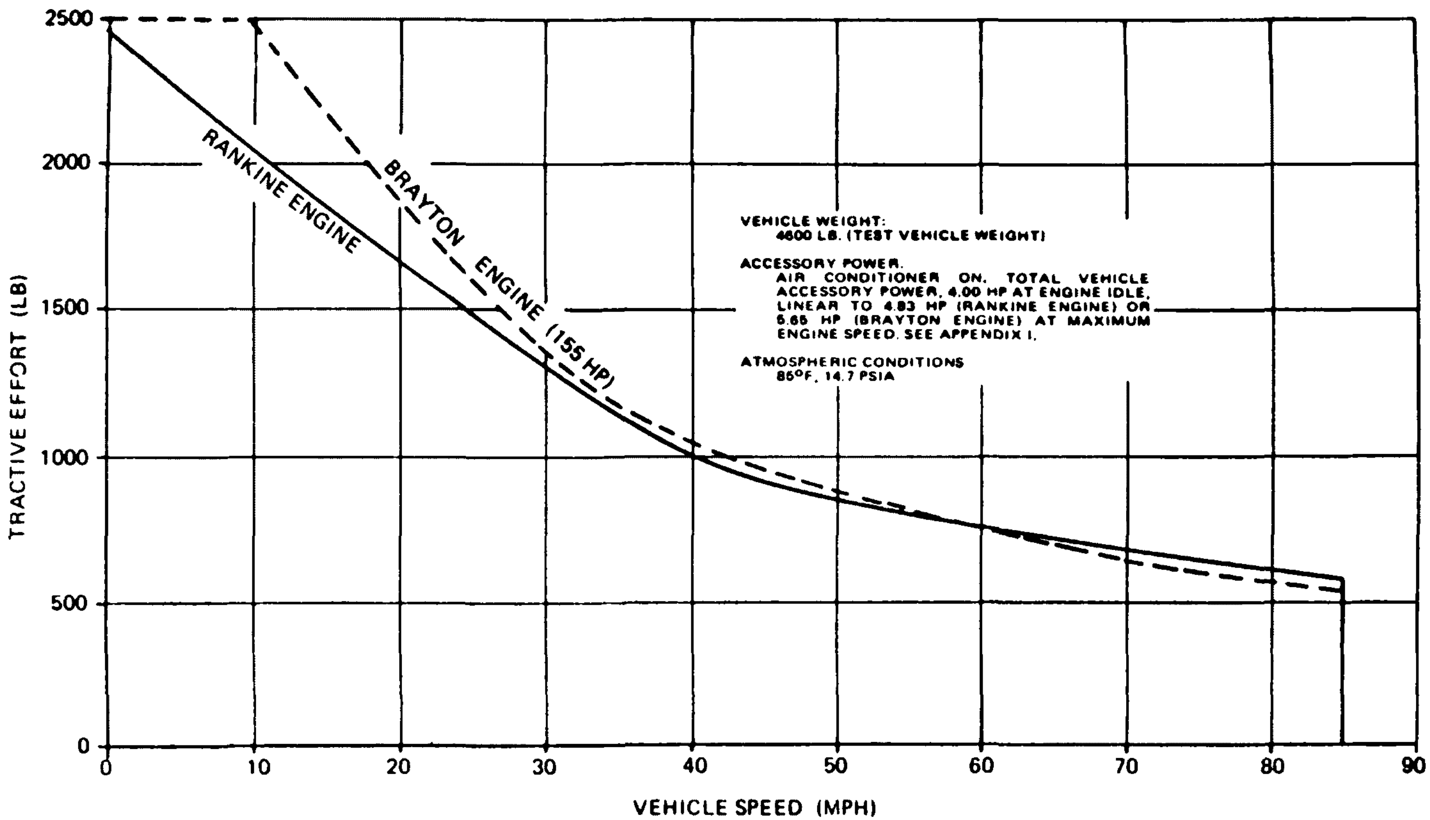

Figure VI-22   Maximum Tractive Effort - Traction Drive Transmission

VII. CONTROL SYSTEM ANALYSIS

The two engines under consideration, the Brayton and Rankine cycle, have similar ratings and narrow speed ranges from idle to maximum speed. The tri-mode and traction transmission are both infinitely variable ratio which provides the capability of operating the engine at speeds completely independent of vehicle road speed. For these reasons, the control system for either transmission when combined with either engine will be basically the same.

A. Control System Approach

Two fundamental methods of control available for use with infinitely variable ratio transmissions are speed control or torque control. With speed control the accelerator pedal position is directly related to transmission output speed and the output torque will attempt to reach the value which is proportional to the difference between the output speed being called for and the actual output speed. If this difference is great, the torque will be high and shock loading will occur with the possible stalling of the engine and excessive wear or ultimate failure of driveline parts. The prime advantage of speed control is that for a particular pedal position, the output speed will remain constant regardless of torque required up to the engine capability. If this feature is desirable or necessary, additional control devices must be incorporated to prevent engine stall or shock loading.

With torque control, the accelerator pedal position is directly related to transmission output torque and the output speed will change until the output torque equals the torque being called for. This control scheme prevents shock loading and engine stall unless the engine is at a speed where it cannot produce the torque called for. This type of control is very similar to that now used in the standard passenger car with a 3-speed automatic transmission. A governor within the transmission which controls the shift point prevents the engine from stalling; the shift to a higher gear does not occur until the engine is up to a speed where it can produce the required torque.

Protection from excessive torque (or pressure in the hydromechanical transmission) is provided inherently in the design of either type of transmission. For the traction drive, the torque converter is the "relief valve" and for the hydromechanical the control system prevents the pressure from exceeding 4500 psi by a pressure feedback which causes the variable wobbler to destroke at that pressure.

The control system must also consider and provide for lowest fuel consumption, emissions, and noise and for maximum acceleration and engine (dynamic) braking.

The block diagram of the control system selected to optimize the foregoing requirements is shown on Figure VII-1. The diagram is similar for both transmission and both engine combinations with only the addition of the shift device for the tri-mode transmission. The system is basically torque control with the torque at the output directly proportional to the output of the governor valve. The governor valve output is a function of accelerator pedal position and engine speed. These are combined in such a manner that each pedal position calls for a particular horsepower and for an engine speed which is optimum (lowest SFC) for that horsepower.

The speed sensor not only prevents the engine from being overloaded at any time but also allows the engine to accelerate to the desired speed and horsepower so that maximum vehicle acceleration can be obtained.

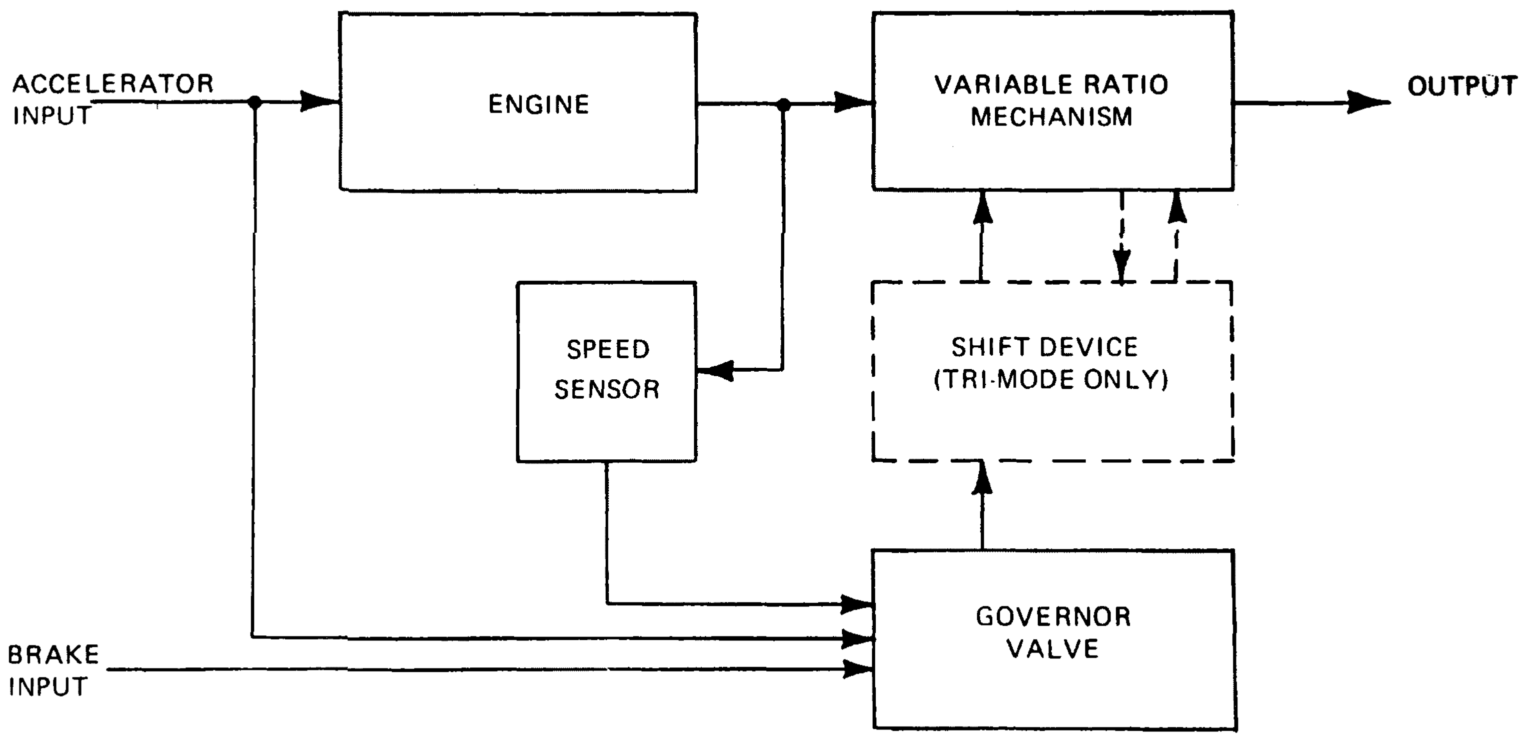

Figure VII-1 Control System Block Diagram

Dynamic engine braking is possible with either transmission type although the hydromechanical will transmit power in the reverse direction more efficiently than the torque converter traction type. With the accelerator pedal released, the brake input signal will place the control system in a state where vehicle inertia will be capable of driving the engine up to its maximum speed for dynamic engine braking.

Either transmission will have a manual control lever very similar to the present automotive control for park, reverse, neutral, and drive (forward) operation. The tri-mode hydromechanical transmission has two automatic mode changes (shifts) in forward which do not affect system performance or stability analysis.

B. Description of Operation

The engine will be started with the control lever only in park or neutral position. In either position the control system is designed so that the existence of any torque at the output will cause the transmission ratio to change in the direction to reduce that torque. This is accomplished by porting working pressure directly to the hydraulic unit stroking pistons.
When the operator depresses the accelerator pedal, two functions are accomplished. The engine fuel control produces an engine torque which is a function of engine speed. Also, the transmission governor valve controls the transmission output torque until the engine reaches the speed for that particular accelerator pedal position, and maintains that speed. The combination of controlling engine torque and speed provides a constant horsepower out of the engine which will accelerate the vehicle until the road load equals this horsepower. A change in accelerator pedal position or load will also change the transmission ratio and engine speed.

For the hydromechanical transmission, the controls automatically change to the second and third mode when the transmission reaches the shift ratio. The transition from one mode to another is completed very smoothly as the load is shared by both hydraulic and mechanical load paths.

When the driver removes his foot from the accelerator pedal, the engine will run at the minimum speed compatible with the output speed and the vehicle will slow down until the engine reaches idle speed. If the driver desires to reduce speed more rapidly, he will apply pressure to the brake pedal. This will first bias the governor valve so that the vehicle will drive the engine to its maximum rated speed for dynamic braking. Further pressure on the brake pedal will actuate the vehicle service brakes.

C. Stability Analysis

A simulation of the control system was run on a hybrid computer. The parameters, equations, and engine performance used are included in Appendix VII of this report. Also included are representative traces of the computer readout for a 20% of maximum throttle (0:2 power unit) acceleration at 50% load and no load.

The simulation was of the hydromechanical transmission in the start-up or hydrostatic mode. The results indicate that the response is good and that the system is stable even when subjected to a maximum transient. Experience dictates that similar results will exist in either hydromechanical mode and also with the traction transmission.

D. Safety Analysis

A failure and effect analysis has been carried out for each component associated with the control system. The results of this study follow:

1. Accelerator Input Failure

If the input to both engine and transmission is lost, there will be no response when the accelerator is depressed.

If only the input to the engine is lost, the engine will not be able to support any load and will not accelerate away from idle.

If only the input to the transmission is lost, the transmission will try to keep the engine at idle by assuming a minimum transmission rate and, therefore, the output power will be quite low.

2. Brake Input Failure

If the input linkage from the brake pedal fails, the system will perform normally except there will be no added assist from the engine while braking. The engine will run at the minimum speed possible for the vehicle speed. Normal friction brakes will not be affected.

3. Speed Sensor Failure

If the signal is lost to the speed sensor, or if the speed sensor sticks in the underspeed position, the transmission will not be able to load the engine and when the accelerator is depressed, the engine will accelerate as it does when in neutral.

If the speed sensor is stuck in the overspeed position, the transmission will load, stall the engine, as the accelerator is depressed.

4. Governor Valve Failure

If the governor valve sticks in the high speed position, the engine will be loaded down. At high speeds, the transmission will go to minimum ratio (minimum engine speed) and exhibit low power output. At low speeds, or when standing still, the engine speed will be driven down until it stalls.

If the governor valve sticks in the low speed position, the engine speed will be driven up. If this happens at high vehicle speeds, the engine speed could be driven beyond a safe speed. This is the only potentially dangerous failure in the system. It is unlikely that this will happen while the vehicle is running, and if it does, the operator can correct the overspeed by shifting into neutral. If this failure occurs while shutdown, the vehicle will appear to remain in neutral after the engine is started.

5. Shift Device Failure (Tri-Mode Only)

If the shift device sticks in the hydrostatic mode, everything will be normal except the vehicle will not be able to accelerate to any speed above the shift point.

If the shift device sticks in either other position, the transmission will remain in that range and the engine will stall as the vehicle slows down or will not start if this occurs when stopped.

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VIII. ESTIMATED TOTAL MANUFACTURING COSTS

A. Definition of the Cost Analysis

The EPA Contract Specification paragraph 6.5 requires an original equipment manufacturer (OEM) cost estimate for the transmission in quantities of 100,000 and 1,000,000 per annum and a cost comparison made with a "conventional" (unspecified) multi-speed automatic transmission with torque converter.

The figures shown in the following cost analysis are for the "total manufacturing cost" which can be broadly defined as the cost of labor and materials, along with the operation and maintenance of existing plant and tooling.

The price includes – cost of materials and purchased subcomponents, direct and indirect labor (such as administration, supervision, production control, quality control, plant maintenance, production engineering, etc.), and supplies and utilities for plant operation. Tooling and plant amortization, and taxes for existing plant and equipment are also included. This price does not include engineering and development, advertising, sales, distribution, interest or profit.

B. Costing Procedure

Although Sundstrand is not a supplier of transmissions to the automobile industry, large quantities of transmissions for the trucking, farm equipment, construction and garden equipment industry are produced. Personnel with cost estimating experience in the automotive automatic transmission industry are available also.

This experience, coupled with cost data available from related product lines, is the basis for the estimate of production rates of 1,000,000 per annum. Additionally, a "judgment factor" was applied to arrive at figures for 100,000 per annum production rates. This "judgment factor" acounted for the degree of complexity, type of processing, and the degree of process simplification possible with high volume production for each type of component within the transmission.

In the area of the hydraulic units, Sundstrand produces approximately 30,000 units per annum of a similar size and type as used in this study, and again "judgment factors" were applied to this cost data to arrive at figures for the production rates required in this study.

Traction drive components were estimated by similarity to automotive parts as much as possible with due consideration for the accurate form grinding required on the toroids and rollers.

All of the above cost estimating assumed the use of highly automated machine tools and material handling equipment used in very high volume production.

C. Results of Cost Analysis

The cost for a typical three speed automatic transmission with torque converter was estimated on a major subassembly basis, and is included for reference along with the hydromechanical and traction transmission costs in Table VIII-1.

For a major component cost breakdown and comparison for the hydromechanical, traction, and a "typical" 3 speed automatic transmission, see Table VIII-2.

Table VIII-1 Transmission Cost Comparison

	Yearly Production Rate	
	100,000	1,000,000
Hydromechanical Tri-Mode Transmission	$182	$122
Traction Drive—Torque Converter Transmission	$149	$105
"Typical" 3 Speed Automatic Transmission	—	$ 89

Table VIII-2 Transmission Manufacturing Cost Breakdown Comparison

	"Typical" 3 Speed Automatic $	Tri-Mode Hydro— Mechanical $	Traction Drive/ Converter $
PLANETARY GEAR SET	12.00	12.40	—
SHAFTING	6.00	8.50	5.30
TRANSFER GEARS (Includes synchronizer assembly for traction transmission)	—	11.20	9.50
CLUTCHES	21.00	15.60	—
HYDRAULIC UNIT (Excludes Bearings, Shafts)	—	22.80	—
CONTROLS SYSTEM (Valve Body, Charge Pump, Linkage)	7.00	7.00	6.50
HOUSINGS, BULKHEADS, COVERS (Includes sound isolators)	14.00	16.50	16.20
TORQUE CONVERTER	14.00	—	13.00
TRACTION DRIVE UNIT (Excludes shaft, includes steering assembly)	—	—	30.20
ANTI-FRICTION BEARINGS (Excludes planet bearings)	1.00	11.00	6.80
MISCELLANEOUS (Bolts, seals, gaskets, filter, etc.)	2.00	2.00	2.00
TRANSMISSION ASSEMBLY AND TEST	12.00	15.00	15.00
TOTAL (To Nearest Dollar)	$89.00	$122.00	$105.00
(COSTS BASED ON 1 MILLION UNITS PER YEAR)			

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IX. REFERENCES

1. Prospects for Electric Vehicles – A Study of Low Pollution Potential Vehicles – Electric – National Air Pollution Control Administration and Arthur D. Little, Inc. 1969.

2. Final Report – U. S. Army Contract DA-11-022-AMC-2269(T) U. S. Army Tank Automotive Center – April – 1966.

3. Final Report – U. S. Army Contract DA-11-022-AMC-695(T) U. S. Army Tank Automotive Center – June 30, 1966.

4. Phase 1 – Final Report – Contract 68-04-0034 EPA, Office of Air Programs, Advanced Automotive Power Systems Division – February, 1972.

5. "Design Practice – Passenger Car Automatic Transmissions" Part 1 and 2, issued by S.A.E.

6. "Tractive Capacity and Efficiency of Rolling Contacts", Hewko, L. O., Rounds, F. G., Scott, R. L., Proceedings of the Symposium on Rolling Contact Phenomena, Joseph B. Bidwell, Ed., Elsevier, N. Y., 1962.

7. "Design and Test of the First Aerodynamic Torque Converter", C.C. Hill, R.A. Mercure, and C.D. Cole, ASME Paper 69-GT-109; March 1969.

APPENDICES

APPENDIX I-1, I-2, I-3, I-4, I-5, I-6, I-7 and I-8; referenced in Section I.

APPENDIX V-1, V-2, V-3, V-4, V-5 and V-6; referenced in Section V.

APPENDIX VI-1, VI-2 and VI-3; referenced in Section VI.

APPENDIX VII-1, VII-2, VII-3, VII-4, VII-5, VII-6(1) and VII-6(2); referenced in Section VII.

There are no appendices for Sections II, III, IV, VIII and IX.

APPENDIX I-1 Attachment 1, Scope of Work, Contract 68-04-0034

Task 6 - Transmission Study for Turbine and Rankine Cycle Engines

This task is generally similar to the effort in tasks 1, 2, 3, 4, 5 and is an extension of the study to cover transmissions, for the gas turbine and rankine cycle engines. For each engine the contractor shall assess quantitatively the technical and economic feasibility of existing and potential types of transmissions most suitable for the particular engine. Based on this study an optimum transmission for each engine shall be recommended and thoroughly evaluated as outlined below:

6.1 - Requirement

6.1.1 - The transmission systems considered shall be suitable for application in a full size family car. The specifications of this vehicle are given in an attachment to the original statement of work entitled "vehicle design goals". Vehicle weight for performance calculation shall be the same as for previous tasks.

6.1.2 - The transmission study for each of the two heat engines, that is 1) gas turbine 2) rankine cycle, shall be based on the engine characteristics and accessories requirements to be supplied by the project officer within one week from the initiation of this study.

6.2 - Technical Feasibility Study

The contractor shall conduct technical and economic feasibility analysis of the various types of transmissions for the two engines specified in section 6.1.2. The transmission types for each engine considered shall include existing and potential transmission such as:

1. Mechanical
2. Hydrostatic
3. Combination of mechanical and hydrostatic
4. Hydrokinetic
5. Electrical
6. Traction
7. Belt Drive

EPA may propose specific transmissions for consideration in this study.

The contractor shall provide, when requested by the project officer, but not earlier than 30 days from the effective date of the contract layout drawings of the transmission or parts of the transmission, in order that independent checks of stress analysis, thermal analysis and safety analysis can be made.

6.3 Performance Analysis

6.3.1. - Steady State Efficiency and Fuel Consumption

The contractor shall calculate and provide graphical plots of steady state part load and full load efficiency of the two selected transmissions for vehicle speed ranging from 0 - 80 mph. (1/10, 1/4, 1/2 and 3/4 full load). The corresponding plots of fuel consumptions shall also be provided. This shall include plot of transmission efficiency and fuel consumptions for cruise speed, on level road, ranging from 0-80 mph with and without air conditioning load.

6.3.2. - Driving Cycle Efficiency and Fuel Consumption

The contractor shall calculate the average efficiency and the corresponding average fuel consumption for the two selected transmissions over the Federal driving cycle, with and without air conditioning load. The detailed procedure and methods for calculating above efficiencies and fuel consumptions shall be included, in a separate appendix attached to the final report.

6.4 Control System Definition and Analysis

The contractor shall conduct control systems analysis on the entire transmission/engine/vehicle system. Control system analysis shall include:

a) a cursory stability analysis

b) safety analysis

c) analysis of possible "pathological case" operator induced instability.

The transmission for each engine shall be readily adaptable to the corresponding heat engine power system presently investigated by EPA. The contractor is responsible for liason with EPA Rankine system contractor such that the recommended transmission and its accessories as a whole unit is adaptable for integration in the vehicle. EPA will provide necessary information on Brayton Cycle system. All cooling systems, control system, etc for the transmission shall match the overall vehicle system thus avoiding unnecessary complications and duplications of sub-systems. The contractor shall provide sufficiently detailed drawings of the transmission control inputs, cooling system and accessories and shall indicate how the total transmission with all its control inputs and accessories fit the overall vehicle system.

6.5 Cost Analysis

The contractor shall perform cost analysis of the various transmission concepts. The quantity of transmissions in units per year to be considered are 100,000 and 1,000,000. This shall be original equipment manufacturer (OEM) cost. The reference transmission, against which all cost and performance comparisons shall be made, is the conventional multi-speed torque converter ("Automatic") transmissions.
The detailed procedure and method for cost estimate shall be included in a separate appendix attached to the report.

6.6 Transmission Recommendation

A recommendation of an optimum transmission based on the system cost and efficiency shall be made. This recommendation shall include designs of the optimum transmission in such detail that accurate cost estimates required in 6.5 above can be made. The recommendation shall include heat engine optimum operational mode and control.

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APPENDIX I-2

ENVIRONMENTAL PROTECTION AGENCY

PROTOTYPE VEHICLE PERFORMANCE SPECIFICATION

January 3, 1972

Division of Advanced Automotive
Power Systems Development

2929 Plymouth Road
Ann Arbor, Michigan 48105

Approved: George M. Thur
Mr. George M. Thur
Chief, Power Systems Branch

Approved: John J. Brogan
John J. Brogan
Director, Div. of Advanced Automotive Power Systems Development

ADVANCED AUTOMOTIVE POWER SYSTEMS (AAPS)

PROTOTYPE VEHICLE PERFORMANCE SPECIFICATION *

January 3, 1972

The AAPS Vehicle performance design specification presented below is intended to provide:

A common objective for prospective contractors.

Criteria for evaluating proposals and selecting a contractor.

Criteria for evaluating competitive power systems for entering first generation system hardware.

Advisory criteria to assist the contractor in such areas as rolling resistance, vehicle air drag etc.

The derived criteria are based on typical characteristics of the class of passenger automobiles with the largest market volume produced in the U.S. during the model years 1969 and 1970. It is noted that emissions, volume and most weight characteristics presented are maximum values while the performance characteristics are intended as minimum values. Contractors and prospective contractors who take exceptions must justify these exceptions and relate these exceptions to the technical goals presented herein.

*Supersedes "Vehicle Design Goals - Six Passenger Automobile" (Revision C - May 28, 1971)

CONTENTS

INTRODUCTION

The design of an automobile from a total systems standpoint could be expected to result in major benefits in cost, safety, and performance. This specification is intended as a step along that path, describing a propulsion system that can be installed into engine compartments as they now exist. Integration of the vehicle accessories within the propulsion system are highly desirable. Following the successful demonstration of this development further optimization of the propulsion system with the power train, suspension, and vehicle styling would be possible.

1. VEHICLE WEIGHT WITHOUT PROPULSION SYSTEM - W_o

W_o is the weight of the vehicle excluding the propulsion system. This weight includes, but is not limited to the frame, body, glass, trim, suspension, wheels (rims and tires), service brakes, seats, upholstery, sound absorbing materials, insulation, dashboard instruments, accessory ducting and wiring, accessories, and all other components not included as part of the propulsion system.

Accessories are defined as driver assistance and passenger convenience components and subsystems not essential to propulsion system operation. Included are power steering systems, power brake systems and passenger compartment heating and air conditioning systems.

W_o is fixed at 2700 lbs.

2. PROPULSION SYSTEM WEIGHT - W_p

W_p includes the energy storage subsystem (including fuel, containment, and supply and deliver ducting), power conversion subsystem (including auxiliaries and control) and power transmitting subsystem (including transmission and drive train to the driven wheels).

Auxiliaries are defined as components and subsystems essential to the operation of the power conversion system. Included are electric power generating subsystems, starting subsystems, exhaust subsystems, motors fans, blowers, pumps, and fluids.

Lightweight propulsion system are highly desirable, however, the maximum allowable propulsion system weight, W_{pm}, is 1500 lbs.

3. VEHICLE CURB WEIGHT - W_c

$W_c = W_o + W_p$.

The maximum allowable vehicle curb weight, W_{cm}, is 4200 lbs. (2700 + 1500 max. = 4200).

4. VEHICLE TEST WEIGHT - W_t

$W_t = W_c + 400$ lbs. W_t is the vehicle weight at which all accelerative maneuvers, fuel economy and emissions are to be calculated. (Items 9c,11c,11d,11e, 11f, 12,13).

The maximum allowable test weight, W_{tm}, is 4600 lbs. (2700 + 1500 max. + 400 = 4600).

5. GROSS VEHICLE WEIGHT - W_g

$W_g = W_c + 1100$ lbs. W_g is the gross vehicle weight at which sustained velocity capability at 5 percent and 30 % grades is to be calculated. (Item 11f). The 1100 lbs. load simulates a full load of passengers and baggage.

The maximum allowable gross vehicle weight, W_{gm}, is 5300 lbs. (2700 + 1500 max. + 1100 = 5300).

6. PROPULSION SYSTEM VOLUME - V_p

V_p is the volume allotted for all items identified under item 2. The propulsion system shall be packagable in such a way that the volume encroachment on either the passenger or luggage compartment does not exceed the following:

a) The transmission tunnel may not be widened so as to decrease the selected production vehicle clearance between the accelerator pedal and the tunnel. The accelerator pedal may not be relocated.

b) Intrusion of the tunnel into the passenger side of the vehicle may be increased by a maximum of 1.5 inches.

c) The tunnel height may be increased by a maximum of 2 inches but without affecting the full fore and aft adjustment of the front seat of the vehicle. The front seat may not be raised.

The propulsion system shall not violate the vehicle ground clearance lines as established by the manufacturer of the vehicle used for propulsion system/vehicle packaging. Additionally, the propulsion system shall not violate the space allocated for wheel jounce motions and vehicle steering clearances. Necessary external appearance (styling) changes will be minor in nature. The propulsion system shall also be packagable in such a way that the handling characteristics of the vehicle are not degraded.

7. AIR DRAG

The product of the drag coefficient, C_d, and the frontal area, A_f, is to be used in air drag calculations. The product of C_dA_f has a value of 12 ft^2. The air density used in computations shall correspond to the applicable ambient air temperature.

8. ROLLING RESISTANCE

Rolling resistance, R, is expressed in the equation $R = (W/65)\,[\,1 + (1.4 \times 10^{-3}V) + (1.2 \times 10^{-5}V^2)\,]$ lbs. V is the vehicle velocity in ft/sec. W is the vehicle weight in lbs.

9. PROPULSION SYSTEM EMISSIONS

The vehicle is to be tested for emissions in accordance with the procedure in the July 2, 1971 Federal Register and as further described in the Code of Federal Regulations Title 40 Part 85 for model year 1976 light duty vehicles (CFR 40-85). The vehicle test weight shall be W_t and the accessory load as defined in Section 14a. Ambient conditions are 14.7 psia and 85°F. Emission tests will be run with fuel specified in Section 10.

The Federal emissions standards are:

Hydrocarbons	-	0.41	grams/mile maximum
Carbon Monoxide	-	3.40	grams/mile maximum
Oxides of Nitrogen*	-	0.40	grams/mile maximum

Prototype vehicles are to meet the following emissions goals to allow for production tolerances and life degradation. Measurement of emissions is to be taken after the system has operated for 100 hours.

Hydrocarbons	-	0.20	grams/mile maximum
Carbon Monoxide	-	1.70	grams/mile maximum
Oxides of Nitrogen*	_	0.20	grams/mile maximum

*Oxides of nitrogen are to be measured or computed as NO_2

Production of smoke, odors, aldehydes, ammonia, particulates or other undesirable emissions not now specified in the July 2, 1971 Federal Register are undesirable.

10. FUEL

Emission tests will use the fuel specified below, however, the power system shall have the capability of meeting emission levels using commercially available unleaded fuels.

Item	ASTM Designation	Specification
Octane, Research, min.	D1656	91-93
Pb. (Organic), gm/U.S. gal	D 525	< .02
Distillation range	D 86	-
I.B.P., °F	-	100-115
10 percent point, °F	-	140-150
50 percent point, °F	-	240-250
90 percent point, °F	-	330-340
E.P. °F (max)	-	425
Sulfur, Wt. percent max.	D1266	0.10
Phosphorous, theory	-	0.0
R.V.P. lb.	D 323	5.5-7.5
Washed gum (max) mgm/gal	D 381	4.0
Corrosion (not lower than)	D 130	1B
Oxidation stability (not less than)	D 525	240+
Hydrocarbon composition	D1319	-
Olefins, percent, max.	-	30
Aromatics, percent, max.	-	40
Saturates	-	Remainder

For computation purposes the lower heating values of this fuel is to be assumed as 18500 Btu/lb. The cost to be assumed for system cost analysis is $0.31/gallon. An A.P.I. gravity of 56.0 is to be assumed in all calculations.

11. START UP, ACCELERATION, AND GRADE VELOCITY PERFORMANCE

a. Start Up:

The vehicle must be capable of being tested in accordance with the procedure outlined in the July 2, 1971 Federal Register without special driver startup/warmup procedures. The accessory load shall be as defined in Section 14b.

The maximum time from "key on" to reach 65 percent of full power level is 45 sec. Ambient conditions are 14.7 psia 60°F. The vehicle is to be soaked at this temperature for a minimum of 12 hours prior to initiation of start test.

Powerplant starting procedures in low ambient temperatures shall be equivalent to or better than the typical automobile spark-ignition engine. After a 24 hour soak at -20°F and 14.7 psia the engine shall achieve a self-sustaining idle condition without further driver input within 25 seconds. No starting aids external to the normal vehicle system shall be needed at or above - 20°F.

b. Idle operation conditions:

The idle creep torque shall not result in level road operation of the vehicle at a speed in excess of 18 mph in high gear, with the entire propulsion system at steady state operating temperature and ambient conditions of 14.7 psia and 85°F. The accessory load shall be as defined in Section 14a.

c. Acceleration from a standing start:

The minimum distance to be covered in 10.0 sec. is 440 ft. The maximum time to reach a velocity of 60 mph is 13.5 sec. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14a. Acceleration is on zero grade and initiated with the engine at the normal idle condition.

d. Acceleration in merging traffic:

The maximum time to accelerate from a constant velocity of 25 mph to a velocity of 70 mph is 15.0 sec. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14a. Acceleration is on zero grade and time starts when the accelerator pedal is depressed.

e. Acceleration, DOT High Speed Pass Maneuver:

The maximum time and distance to go from an initial velocity of 50 mph with the front of the automobile (18 foot length assumed) 100 feet behind the back of a 55 foot truck traveling at a constant 50 mph, to a position where the back of the automobile is 100 feet in front of the front of the 55 foot truck, is 15 sec. and 1400 ft. The entire maneuver takes place in a traffic lane adjacent to the lane in which the truck is operated. Vehicle is accelerated until the maneuver is completed or until a maximum speed of 80 mph is attained, whichever occurs first. Vehicle acceleration ceases when a speed of 80 mph is attained, the maneuver then being completed at a constant 80 mph. (This does not imply a design requirement limiting the maximum vehicle speed to 80 mph). Time starts when the accelerator pedal is depressed. Ambient conditions are 14.7 psia, 85°F. Vehicle weight is W_t and accessory load as defined in Section 14a. Acceleration is on zero grade.

f. Grade velocity:

The vehicle must be capable of starting from rest on a thirty percent (30%) grade and ascending the grade at a minimum speed of 5 mph. The vehicle must be capable of this maneuver in both the forward and reverse directions at a vehicle weight of W_g, with the accessory load as defined in Section 14a.

The minimum cruise velocity that can be continuously maintained on a five percent (5%) grade shall be not less than 65 mph with a vehicle weight of W_g and accessory load as defined in Section 14a.

The minimum cruise velocity that can be continuously maintained on a zero percent (0%) grade shall be not less than 85 mph with a vehicle weight of W_t and with the accessory load as defined in Item 14a.

Ambient conditions for all grade specifications are 14.7 psia and 85°F.

Performance degradation attributable to loss of powerplant efficiency at extreme temperatures shall not exceed ten percent (10%) relative to the performance values specified at 85°F. This limitation applies to ambient temperatures from -20°F to 105°F.

The wind velocity is to be less than 10 mph for all acceleration and grade tests.

12. MINIMUM VEHICLE RANGE

Minimum vehicle range without refueling will be 200 miles (maximum fuel capacity is 25 U.S. gallons). The minimum range shall be calculated for, and applied to each of the following modes:

1. Cyclic mode is: The Federal driving cycle which is in accordance with the July 2, 1971 Federal Register. The range may be calculated for one cycle and ratioed to 200 miles.

2. Cruise mode is: A constant 70 mph cruise on a zero grade for 200 miles.

The vehicle weight for both modes shall be W_t initially and with accessory power levels as specified in Section 14 . The ambient conditions shall be a pressure of 14.7 psia, and a temperature of -20^oF (air conditioner off) and 105^oF (air conditioner on).

13. FUEL CONSUMPTION

Using the fuel specified in Section 10,a "fuel economy" figure shall be calculated based on 1) miles per gallon and 2) the number of Btu per mile required to drive the vehicle through the following modes of operation:

		Avg. Speed	Hours	% of Time
1)	Federal Driving Cycle	19.84	1750	50
2)	Simplified Surburban Route (equal times at constant 20,30 and 40 mph speeds).	30.00	1150	33
3)	Simplified Country Route (equal times at constant 50,60 and 70 mph speeds).	60.00	600	17
	Totals	30	3500	100

In all cases the system fuel consumption shall be calculated for a vehicle weight of W_t initially, and power levels as specified in item 14a. Ambient conditions are 14.7 psia and 85^oF.

It is desirable that the fuel consumption rate at idle operating condition not exceed 7 lbs/hour.

14. ACCESSORY POWER REQUIREMENTS

a. Accessory power requirements with the air conditioning in operation are defined as 15 hp at maximum engine speed and 4 hp at engine idle speed, with a linear relationship between these two points.

b. Accessory power requirements without the air conditioning in operation are defined as 5 hp at maximum engine speed and 2 hp at engine idle speed, with a linear relationship between these two points.

15. PROPULSION SYSTEM OPERATING TEMPERATURE AND PRESSURE RANGE

The propulsion system shall be operable within an expected ambient temperature range of -40° to 125°F.

The propulsion system shall be operable within an expected environmental pressure range of 9 psia to 15 psia.

16. PASSENGER COMFORT REQUIREMENTS

Heating and air conditioning of the passenger compartment shall be at a rate equivalent to that provided in the present (1970) standard full size family car.

Present practice for maximum passenger compartment heating rate is approximately 30,000 Btu/hr. For an air conditioning system at 110°F ambient, 80°F and 40% relative humidity air to the evaporator, the rate is approximately 13,000 Btu/hr.

17. NOISE STANDARDS

a. Maximum noise test:*

The maximum noise generated by the vehicle shall not exceed 77 dbA when measured in accordance with SAE J986a. Note that the noise level is 77 dbA whereas in the SAE J986 the level is 86 dbA.

b. Low speed noise test:*

The maximum noise generated by the vehicle shall not exceed 63 dbA when measured in accordance with SAE J986a except that a constant vehicle velocity of 30 mph is used on the pass-by.

c. Idle noise test:*

The maximum noise generated by the vehicle shall not exceed 62 dbA when measured in accordance with SAE J986a except that the engine is idling (clutch disengaged or in neutral gear) and the vehicle is stationary. A 360° survey shall be made, the microphone being 10 feet from the vehicle perimeter.

* The air conditioner will not be in operation during noise tests.

18. OPERATIONAL LIFE

The design lifetime of the propulsion system in normal operation will be 3500 hours minimum.

Termination of the operational life of an engine shall be determined by structural or functional failure. Functional failure is defined as power degradation exceeding 25 percent of maximum output of the rear wheels.

19. RELIABILITY AND MAINTAINABILITY

The reliability and maintainability of the vehicle shall equal or exceed that of the spark-ignition automobile. The mean-time-between-failure should be maximized to reduce the number of unscheduled service trips. No failure modes shall present a serious safety hazard during vehicle operation and servicing. Failure propagation should be minimized. The power plant should be designed for ease of maintenance and repairs to minimize costs, maintenance personnel education, and downtime.

20. COST OF OWNERSHIP

The initial cost and net cost of ownership of the vehicle shall be minimized for ten years and 105,000 miles of operation.

21. SAFETY STANDARDS

The vehicle shall comply with all Department of Transportation Federal Motor Vehicle Safety Standards in force when the selected test vehicle was manufactured.

TABLE I

TEST CONDITIONS
(ENGINE DYNAMOMETER, CHASSIS DYNAMOMETER, ROAD)

Section	Performance Requirements	Accessory Power	Weight	Temperature	Pressure
9. EMISSIONS	HC 0.20 grams per mile**	14a	W_t	85°F	14.7
	CO 1.70 grams per mile**	14a	W_t	85°F	14.7
	NO_2 0.20 grams per mile**	14a	W_t	85°F	14.7
11. PERFORMANCE					
a. Start Up	65%* power in 45 sec**	14b	W_t	60°F	14.7
	Driver Assistance 25 sec**	14b	W_t	-20°F	14.7
b. Idle	Creep 18 mph**	14a	W_t	85°F	14.7
c. Accel (0-60)	13.5 sec** to 60 MPH	14a	W_t	85°F	14.7
	440 ft* in 10.0 sec.	14a	W_t	85°F	14.7
d. Accel (25-70)	15.0 sec** from 25 to 70 MPH	14a	W_t	85°F	14.7
e. Accel (50-80)	15.0 sec** and 1400 Ft** From 50 to 80 MPH**	14a	W_t	85°F	14.7
f. Grade (30%)	0 to 5 MPH*	14a	W_g	85°F	14.7
(5%)	65 MPH*	14a	W_g	85°F	14.7
(0%)	85 MPH*	14a	W_t	85°F	14.7
12. VEHICLE RANGE	200 MI*				
	1. During ~~DCC~~ FDC	14b and 14a	W_t	-20°F and 105°F	14.7
	2. At 70 MPH	14b and 14a	W_t	-20°F and 105°F	14.7
13. FUEL CONSUM.	MPG During FDC	14a A/C	W_t	85°F	14.7
	MPG at 20,30, and 40 MPH	14a "	W_t	85°F	14.7
	MPG at 50,60, and 70 MPH	14a "	W_t	85°F	14.7

NOTES: Emission tests will be run with fuel specified in Section 10.
Road test wind conditions shall not exceed 10 MPH in any direction.
*Minimum values
**Maximum values

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APPENDIX A

DHEW URBAN DYNAMOMETER DRIVING SCHEDULE

(Speed versus Time Sequence)

Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)
0	0.0	84	28.6	168	16.5
1	0.0	85	29.3	169	19.8
2	0.0	86	29.8	170	22.2
3	0.0	87	30.1	171	24.3
4	0.0	88	30.4	172	25.8
5	0.0	89	30.7	173	26.4
6	0.0	90	30.7	174	25.7
7	0.0	91	30.5	175	25.1
8	0.0	92	30.4	176	24.7
9	0.0	93	30.3	177	25.0
10	0.0	94	30.4	178	25.2
11	0.0	95	30.8	179	25.4
12	0.0	96	30.4	180	25.8
13	0.0	97	29.9	181	27.2
14	0.0	98	29.5	182	26.5
15	0.0	99	29.8	183	24.0
16	0.0	100	30.3	184	22.7
17	0.0	101	30.7	185	19.4
18	0.0	102	30.9	186	17.7
19	0.0	103	31.0	187	17.2
20	0.0	104	30.9	188	18.1
21	3.0	105	30.4	189	18.6
22	5.9	106	29.8	190	20.0
23	8.6	107	29.9	191	22.2
24	11.5	108	30.2	192	24.5
25	14.3	109	30.7	193	27.3
26	16.9	110	31.2	194	30.5
27	17.3	111	31.8	195	33.5
28	18.1	112	32.2	196	36.2
29	20.7	113	32.4	197	37.3
30	21.7	114	32.2	198	39.3
31	22.4	115	31.7	199	40.5
32	22.5	116	28.6	200	42.1
33	22.1	117	25.3	201	43.5
34	21.5	118	22.0	202	45.1
35	20.9	119	18.7	203	46.0
36	20.4	120	15.4	204	46.8
37	19.8	121	12.1	205	47.5
38	17.0	122	8.8	206	47.5
39	14.9	123	5.5	207	47.3
40	14.9	124	2.2	208	47.2
41	15.2	125	0.0	209	47.0
42	15.5	126	0.0	210	47.0
43	16.0	127	0.0	211	47.0
44	17.1	128	0.0	212	47.0
45	19.1	129	0.0	213	47.0
46	21.1	130	0.0	214	47.2
47	22.7	131	0.0	215	47.4
48	22.9	132	0.0	216	47.9
49	22.7	133	0.0	217	48.5
50	22.6	134	0.0	218	49.1
51	21.3	135	0.0	219	49.5
52	19.0	136	0.0	220	50.0
53	17.1	137	0.0	221	50.6
54	15.8	138	0.0	222	51.0
55	15.8	139	0.0	223	51.5
56	17.7	140	0.0	224	52.2
57	19.8	141	0.0	225	53.2
58	21.6	142	0.0	226	54.1
59	23.2	143	0.0	227	54.6
60	24.2	144	0.0	228	54.9
61	24.6	145	0.0	229	55.0
62	24.9	146	0.0	230	54.9
63	25.0	147	0.0	231	54.6
64	24.6	148	0.0	232	54.6
65	24.5	149	0.0	233	54.8
66	24.7	150	0.0	234	55.1
67	24.8	151	0.0	235	55.5
68	24.7	152	0.0	236	55.7
69	24.6	153	0.0	237	56.1
70	24.6	154	0.0	238	56.3
71	25.1	155	0.0	239	56.6
72	25.6	156	0.0	240	56.7
73	25.7	157	0.0	241	56.7
74	25.4	158	0.0	242	56.5
75	24.9	159	0.0	243	56.5
76	25.0	160	0.0	244	56.5
77	25.4	161	0.0	245	56.5
78	26.0	162	0.0	246	56.5
79	26.0	163	0.0	247	56.5
80	25.7	164	3.3	248	56.4
81	26.1	165	6.6	249	56.1
82	26.7	166	9.9	250	55.8
83	27.5	167	13.2	251	55.1

APPENDIX A—Continued

Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)
252	54.6	339	0.0	426	8.5
253	54.2	340	0.0	427	5.2
254	54.0	341	0.0	428	1.9
255	53.7	342	0.0	429	0.0
256	53.6	343	0.0	430	0.0
257	53.9	344	0.0	431	0.0
258	54.0	345	0.0	432	0.0
259	54.1	346	0.0	433	0.0
260	54.1	347	1.0	434	0.0
261	53.8	348	4.3	435	0.0
262	53.4	349	7.6	436	0.0
263	53.0	350	10.9	437	0.0
264	52.6	351	14.2	438	0.0
265	52.1	352	17.3	439	0.0
266	52.4	353	20.0	440	0.0
267	52.0	354	22.5	441	0.0
268	51.9	355	23.7	442	0.0
269	51.7	356	25.2	443	0.0
270	51.5	357	26.6	444	0.0
271	51.6	358	28.1	445	0.0
272	51.8	359	30.0	446	0.0
273	52.1	360	30.8	447	0.0
274	52.5	361	31.6	448	3.3
275	53.0	362	32.1	449	6.6
276	53.5	363	32.8	450	9.9
277	54.0	364	33.6	451	13.2
278	54.9	365	34.5	452	16.5
279	55.4	366	34.6	453	19.8
280	55.6	367	34.9	454	23.1
281	56.0	368	34.8	455	26.4
282	56.0	369	34.5	456	27.8
283	55.8	370	34.7	457	29.1
284	55.2	371	35.5	458	31.5
285	54.5	372	36.0	459	33.0
286	53.6	373	36.0	460	33.6
287	52.5	374	36.0	461	34.8
288	51.5	375	36.0	462	35.1
289	51.5	376	36.0	463	35.6
290	51.5	377	36.0	464	36.1
291	51.1	378	36.1	465	36.0
292	50.1	379	36.4	466	36.1
293	50.0	380	36.5	467	36.2
294	50.1	381	36.4	468	36.0
295	50.0	382	36.0	469	35.7
296	49.6	383	35.1	470	36.0
297	49.5	384	34.1	471	36.0
298	49.6	385	33.5	472	35.6
299	49.5	386	31.4	473	35.5
300	49.1	387	29.0	474	35.4
301	48.6	388	25.7	475	35.2
302	48.1	389	23.0	476	35.2
303	47.2	390	20.3	477	35.2
304	46.1	391	17.5	478	35.2
305	45.0	392	14.5	479	35.2
306	43.8	393	12.0	480	35.2
307	42.6	394	8.7	481	35.0
308	41.5	395	5.4	482	35.1
309	40.3	396	2.1	483	35.2
310	38.5	397	0.0	484	35.5
311	37.0	398	0.0	485	35.2
312	35.2	399	0.0	486	35.0
313	33.8	400	0.0	487	35.0
314	32.5	401	0.0	488	35.0
315	31.5	402	0.0	489	34.8
316	30.6	403	2.6	490	34.6
317	30.5	404	5.9	491	34.5
318	30.0	405	9.2	492	33.5
319	29.0	406	12.5	493	32.0
320	27.5	407	15.8	494	30.1
321	24.8	408	19.1	495	28.0
322	21.5	409	22.4	496	25.5
323	20.1	410	25.0	497	22.5
324	19.1	411	25.6	498	19.8
325	18.5	412	27.5	499	16.5
326	17.0	413	29.0	500	13.2
327	15.5	414	30.0	501	10.3
328	12.5	415	30.1	502	7.2
329	10.8	416	30.0	503	4.0
330	8.0	417	29.7	504	1.0
331	4.7	418	29.3	505	0.0
332	1.4	419	28.8	506	0.0
333	0.0	420	28.0	507	0.0
334	0.0	421	25.0	508	0.0
335	0.0	422	21.7	509	0.0
336	0.0	423	18.4	510	0.0
337	0.0	424	15.1	511	1.2
338	0.0	425	11.8	512	3.5

APPENDIX A—Continued

Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)
513	5.5	600	21.6	686	0.0
514	6.5	601	22.0	687	0.0
515	8.5	602	22.4	688	0.0
516	9.6	603	22.5	689	0.0
517	10.5	604	22.5	690	0.0
518	11.9	605	22.5	691	0.0
519	14.0	606	22.7	692	0.0
520	16.0	607	23.7	693	0.0
521	17.7	608	25.1	694	1.4
522	19.0	609	26.0	695	3.3
523	20.1	610	26.5	696	4.4
524	21.0	611	27.0	697	6.5
525	22.0	612	26.1	698	9.2
526	23.0	613	22.3	699	11.3
527	23.8	614	19.5	700	13.5
528	24.5	615	16.2	701	14.6
529	24.9	616	12.9	702	16.4
530	25.0	617	9.6	703	16.7
531	25.0	618	6.3	704	16.5
532	25.0	619	3.0	705	16.5
533	25.0	620	0.0	706	18.2
534	25.0	621	0.0	707	19.2
535	25.0	622	0.0	708	20.1
536	25.6	623	0.0	709	21.5
537	25.8	624	0.0	710	22.5
538	26.0	625	0.0	711	22.5
539	25.6	626	0.0	712	22.1
540	25.2	627	0.0	713	22.7
541	25.0	628	0.0	714	23.3
542	25.0	629	0.0	715	23.5
543	25.0	630	0.0	716	22.5
544	24.4	631	0.0	717	21.6
545	23.1	632	0.0	718	20.5
546	19.8	633	0.0	719	18.0
547	16.5	634	0.0	720	15.0
548	13.2	635	0.0	721	12.0
549	9.9	636	0.0	722	9.0
550	6.6	637	0.0	723	6.2
551	3.3	638	0.0	724	4.5
552	0.0	639	0.0	725	3.0
553	0.0	640	0.0	726	2.1
554	0.0	641	0.0	727	0.5
555	0.0	642	0.0	728	0.5
556	0.0	643	0.0	729	3.2
557	0.0	644	0.0	730	6.5
558	0.0	645	0.0	731	9.8
559	0.0	646	2.0	732	12.5
560	0.0	647	4.5	733	14.0
561	0.0	648	7.8	734	16.0
562	0.0	649	10.2	735	18.0
563	0.0	650	12.5	736	19.0
564	0.0	651	14.0	737	21.5
565	0.0	652	15.3	738	23.1
566	0.0	653	17.5	739	24.5
567	0.0	654	19.6	740	25.5
568	0.0	655	21.0	741	26.5
569	3.3	656	22.2	742	27.1
570	6.6	657	23.3	743	27.6
571	9.9	658	24.5	744	27.9
572	13.0	659	25.3	745	28.3
573	14.6	660	25.6	746	28.6
574	16.0	661	26.0	747	28.6
575	17.0	662	26.1	748	28.3
576	17.0	663	26.2	749	28.2
577	17.0	664	26.2	750	28.0
578	17.5	665	26.4	751	27.5
579	17.7	666	26.5	752	26.8
580	17.7	667	26.5	753	25.5
581	17.5	668	26.0	754	23.5
582	17.0	669	25.5	755	21.5
583	16.9	670	23.6	756	19.0
584	16.6	671	21.4	757	16.5
585	17.0	672	18.5	758	14.0
586	17.1	673	16.4	759	12.5
587	17.0	674	14.5	760	9.4
588	16.6	675	11.6	761	6.2
589	16.5	676	8.7	762	3.0
590	16.5	677	5.8	763	1.5
591	16.6	678	3.5	764	1.5
592	17.0	679	2.0	765	0.5
593	17.6	680	0.0	766	0.0
594	18.5	681	0.0	767	3.0
595	19.2			768	6.3
596	20.2	682	0.0	769	9.6
597	21.0	683	0.0	770	12.9
598	21.1	684	0.0	771	15.8
599	21.2	685	0.0	772	17.5

Table APP-I-3 DHEW Urban Dynamometer Driving Cycle

APPENDIX A—Continued

Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)
773	18.4	861	29.1	948	24.0
774	19.8	862	29.0	949	22.0
775	20.7	863	28.1	950	20.1
776	22.0	864	27.5	951	16.9
777	23.2	865	27.0	952	13.6
778	25.0	866	25.8	953	10.3
779	26.8	867	25.0	954	7.0
780	27.6	868	24.5	955	3.7
781	28.0	869	24.8	956	0.4
782	28.3	870	25.1	957	0.0
783	28.9	871	25.6	958	0.0
784	28.9	872	25.7	959	0.0
785	28.9	873	26.2	960	2.0
786	28.6	874	26.9	961	5.3
787	28.5	875	27.6	962	8.6
788	28.3	876	27.8	963	11.9
789	28.3	877	28.4	964	15.2
790	28.3	878	29.0	965	17.5
791	28.2	879	29.2	966	18.6
792	27.6	880	29.1	967	20.0
793	27.5	881	29.0	968	21.1
794	27.5	882	28.9	969	22.0
795	27.5	883	28.5	970	23.0
796	27.5	884	28.1	971	24.5
797	27.5	885	28.0	972	26.3
798	27.5	886	28.0	973	27.5
799	27.6	887	27.6	974	28.1
800	28.0	888	27.2	975	28.4
801	28.5	889	26.6	976	28.5
802	30.0	890	27.0	977	28.5
803	31.0	891	27.5	978	28.5
804	32.0	892	27.8	979	27.7
805	33.0	893	28.0	980	27.5
806	33.0	894	27.8	981	27.2
807	33.6	895	28.0	982	26.8
808	34.0	896	28.0	983	26.5
809	34.3	897	28.0	984	26.0
810	34.2	898	27.7	985	25.7
811	34.0	899	27.4	986	25.2
812	34.0	900	26.9	987	24.0
813	33.9	901	26.6	988	23.0
814	33.6	902	26.5	989	21.5
815	33.1	903	26.5	990	21.5
816	33.0	904	26.5	991	21.8
817	32.5	905	26.3	992	22.5
818	32.0	906	26.2	993	23.0
819	31.9	907	26.2	994	22.8
820	31.6	908	25.9	995	22.8
821	31.5	909	25.6	996	23.0
822	30.6	910	25.6	997	22.7
823	30.0	911	25.9	998	22.7
824	29.9	912	25.8	999	22.7
825	29.9	913	25.5	1,000	23.5
826	29.9	914	24.6	1,001	24.0
827	29.9	915	23.5	1,002	24.6
828	29.6	916	22.2	1,003	24.8
829	29.5	917	21.6	1,004	25.1
830	29.5	918	21.6	1,005	25.5
831	29.3	919	21.7	1,006	25.6
832	28.9	920	22.6	1,007	25.5
833	28.2	921	23.4	1,008	25.0
834	27.7	922	24.0	1,009	24.1
835	27.0	923	24.2	1,010	23.7
836	25.5	924	24.4	1,011	23.2
837	23.7	925	24.9	1,012	22.9
838	22.0	926	25.1	1,013	22.5
839	20.5	927	25.2	1,014	22.0
840	19.2	928	25.3	1,015	21.6
841	19.2	929	25.5	1,016	20.5
843	20.9	930	25.2	1,017	17.5
844	21.4	931	25.0	1,018	14.2
845	22.0	932	25.0	1,019	10.9
846	22.6	933	25.0	1,020	7.6
847	23.2	934	24.7	1,021	4.3
848	24.0	935	24.5	1,022	1.0
849	25.0	936	24.3	1,023	0.0
850	26.0	937	24.3	1,024	0.0
851	26.6	938	24.5	1,025	0.0
852	26.6	939	25.0	1,026	0.0
853	26.8	940	25.0	1,027	0.0
854	27.0	941	24.8	1,028	0.0
855	27.2	942	24.6	1,029	0.0
856	27.8	943	24.1	1,030	0.0
857	28.1	944	24.5	1,031	0.0
858	28.8	945	25.1	1,032	0.0
859	28.9	946	25.6	1,033	0.0
860	29.0	947	25.1	1,034	0.0

APPENDIX A—Continued

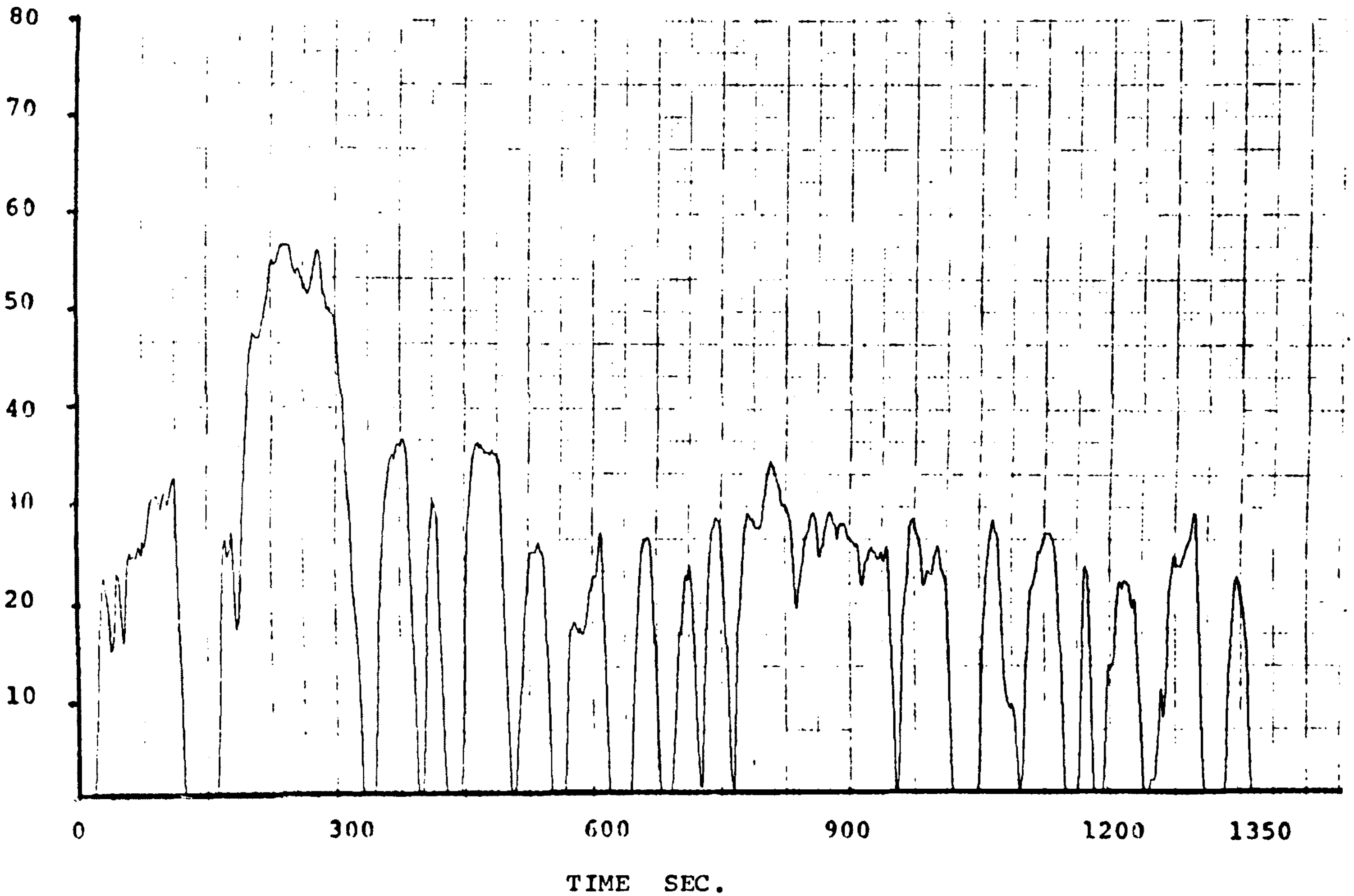
Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)
1,035	0.0	1,122	25.0	1,209	14.0
1,036	0.0	1,123	24.9	1,210	15.5
1,037	0.0	1,124	24.8	1,211	17.0
1,038	0.0	1,125	25.0	1,212	18.6
1,039	0.0	1,126	25.4	1,213	19.7
1,040	0.0	1,127	25.8	1,214	21.0
1,041	0.0	1,128	26.0	1,215	21.5
1,042	0.0	1,129	26.4	1,216	21.8
1,043	0.0	1,130	26.6	1,217	21.8
1,044	0.0	1,131	26.9	1,218	21.5
1,045	0.0	1,132	27.0	1,219	21.2
1,046	0.0	1,133	27.0	1,220	21.5
1,047	0.0	1,134	27.0	1,221	21.8
1,048	0.0	1,135	26.9	1,222	22.0
1,049	0.0	1,136	26.8	1,223	21.9
1,050	0.0	1,137	26.8	1,224	21.7
1,051	0.0	1,138	26.5	1,225	21.5
1,052	0.0	1,139	26.4	1,226	21.5
1,053	1.2	1,140	26.0	1,227	21.4
1,054	4.0	1,141	25.5	1,228	20.1
1,055	7.3	1,142	24.6	1,229	19.5
1,056	10.6	1,143	23.5	1,230	19.2
1,057	13.9	1,144	21.5	1,231	19.6
1,058	17.0	1,145	20.0	1,232	19.8
1,059	18.5	1,146	17.5	1,233	20.0
1,060	20.0	1,147	16.0	1,234	19.5
1,061	21.8	1,148	14.0	1,235	17.5
1,062	23.0	1,149	10.7	1,236	15.5
1,063	24.0	1,150	7.4	1,237	13.0
1,064	24.8	1,151	4.1	1,238	10.0
1,065	25.6	1,152	0.8	1,239	8.0
1,066	26.5	1,153	0.0	1,240	6.0
1,067	26.8	1,154	0.0	1,241	4.0
1,068	27.4	1,155	0.0	1,242	2.5
1,069	27.9	1,156	0.0	1,243	0.7
1,070	28.3	1,157	0.0	1,244	0.0
1,071	28.0	1,158	0.0	1,245	0.0
1,072	27.5	1,159	0.0	1,246	0.0
1,073	27.0	1,160	0.0	1,247	0.0
1,074	27.0	1,161	0.0	1,248	0.0
1,075	26.3	1,162	0.0	1,249	0.0
1,076	24.5	1,163	0.0	1,250	0.0
1,077	22.5	1,164	0.0	1,251	0.0
1,078	21.5	1,165	0.0	1,252	1.0
1,079	20.6	1,166	0.0	1,253	1.0
1,080	18.0	1,167	0.0	1,254	1.0
1,081	15.0	1,168	0.0	1,255	1.0
1,082	12.3	1,169	2.1	1,256	1.0
1,083	11.1	1,170	5.4	1,257	1.6
1,084	10.6	1,171	8.7	1,258	3.0
1,085	10.0	1,172	12.0	1,259	4.0
1,086	9.5	1,173	15.3	1,260	5.0
1,087	9.1	1,174	18.6	1,261	6.3
1,088	8.7	1,175	21.1	1,262	8.0
1,089	8.6	1,176	23.0	1,263	10.0
1,090	8.8	1,177	23.5	1,264	10.5
1,091	9.0	1,178	23.0	1,265	9.5
1,092	8.7	1,179	22.5	1,266	8.5
1,093	8.6	1,180	20.0	1,267	7.6
1,094	8.0	1,181	16.7	1,268	8.8
1,095	7.0	1,182	13.4	1,269	11.0
1,096	5.0	1,183	10.1	1,270	14.0
1,097	4.2	1,184	6.8	1,271	17.0
1,098	2.6	1,185	3.5	1,272	19.5
1,099	1.0	1,186	0.2	1,273	21.0
1,100	0.0	1,187	0.0	1,274	21.8
1,101	0.1	1,188	0.0	1,275	22.2
1,102	0.6	1,189	0.0	1,276	23.0
1,103	1.6	1,190	0.0	1,277	23.6
1,104	3.6	1,191	0.0	1,278	24.1
1,105	6.9	1,192	0.0	1,279	24.5
1,106	10.0	1,193	0.0	1,280	24.5
1,107	12.8	1,194	0.0	1,281	24.0
1,108	14.0	1,195	0.0	1,282	23.5
1,109	14.5	1,196	0.0	1,283	23.5
1,110	16.0	1,197	0.2	1,284	23.5
1,111	18.1	1,198	1.5	1,285	23.5
1,112	20.0	1,199	3.5	1,286	23.5
1,113	21.0	1,200	6.5	1,287	23.5
1,114	21.2	1,201	9.8	1,288	24.0
1,115	21.3	1,202	12.0	1,289	24.1
1,116	21.4	1,203	12.9	1,290	24.6
1,117	21.7	1,204	13.0	1,291	24.7
1,118	22.5	1,205	12.6	1,292	25.0
1,119	23.0	1,206	12.8	1,293	25.4
1,120	23.8	1,207	13.1	1,294	25.6
1,121	24.5	1,208	13.1	1,295	25.7

APPENDIX A—Continued

Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)	Time (sec.)	Speed (m.p.h.)
1,296	26.0	1,322	0.0	1,347	20.3
1,297	26.2	1,323	0.0	1,348	21.3
1,298	27.0	1,324	0.0	1,349	21.9
1,299	27.8	1,325	0.0	1,350	22.1
1,300	28.3	1,326	0.0	1,351	22.4
1,301	29.0	1,327	0.0	1,352	22.0
1,302	29.1	1,328	0.0	1,353	21.6
1,303	29.0	1,329	0.0	1,354	21.1
1,304	28.0	1,330	0.0	1,355	20.5
1,305	24.7	1,331	0.0	1,356	20.0
1,306	21.4	1,332	0.0	1,357	19.6
1,307	18.1	1,333	0.0	1,358	18.5
1,308	14.8	1,334	0.0	1,359	17.5
1,309	11.5	1,335	0.0	1,360	16.5
1,310	8.2	1,336	0.0	1,361	15.5
1,311	4.9	1,337	0.0	1,362	14.0
1,312	1.6	1,338	1.5	1,363	11.0
1,313	0.0	1,339	4.8	1,364	8.0
1,314	0.0	1,340	8.1	1,365	5.2
1,315	0.0	1,341	11.4	1,366	2.5
1,316	0.0	1,342	13.2	1,367	0.0
1,317	0.0	1,343	15.1	1,368	0.0
1,318	0.0	1,344	16.8	1,369	0.0
1,319	0.0	1,345	18.3	1,370	0.0
1,320	0.0	1,346	19.5	1,371	0.0
1,321	0.0				

Table APP-I-3 DHEW Urban Dynamometer Driving Cycle (Continued)

Sundstrand Aviation



MILES/HR.

Figure APP-I-3 Plot of Federal Driving Cycle

Sundstrand Aviation

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APPENDIX I-4. RANKINE ENGINE DATA

To evaluate the performance characteristics of the EPA car with the Aerojet Rankine engine and an infinitely variable transmission, it was necessary to define in detail the characteristics of the engine. The optimum engine operating speed as a function of required engine power was determined, as well as the specific fuel consumption characteristics.

From the engine performance map supplied by Aerojet and a knowledge of transmission performance characteristics, it was possible to determine an engine operating speed curve as a function of required engine power. The curve selected offers a good compromise in balancing the engine efficiency and the transmission efficiency in the effort to maximize system efficiency (see graph, Figure I-4).

Specific fuel consumption data in pounds of fuel per horsepower hour as a function of engine output power and vehicle velocity was supplied by Aerojet (see graph, Figure I-4A).

Engine output power as used here means total engine output power, that is, input power to the transmission plus vehicle accessory power.

The engine reduction gear mesh has already been accounted for in the specific fuel consumption data.

Sundstrand Aviation

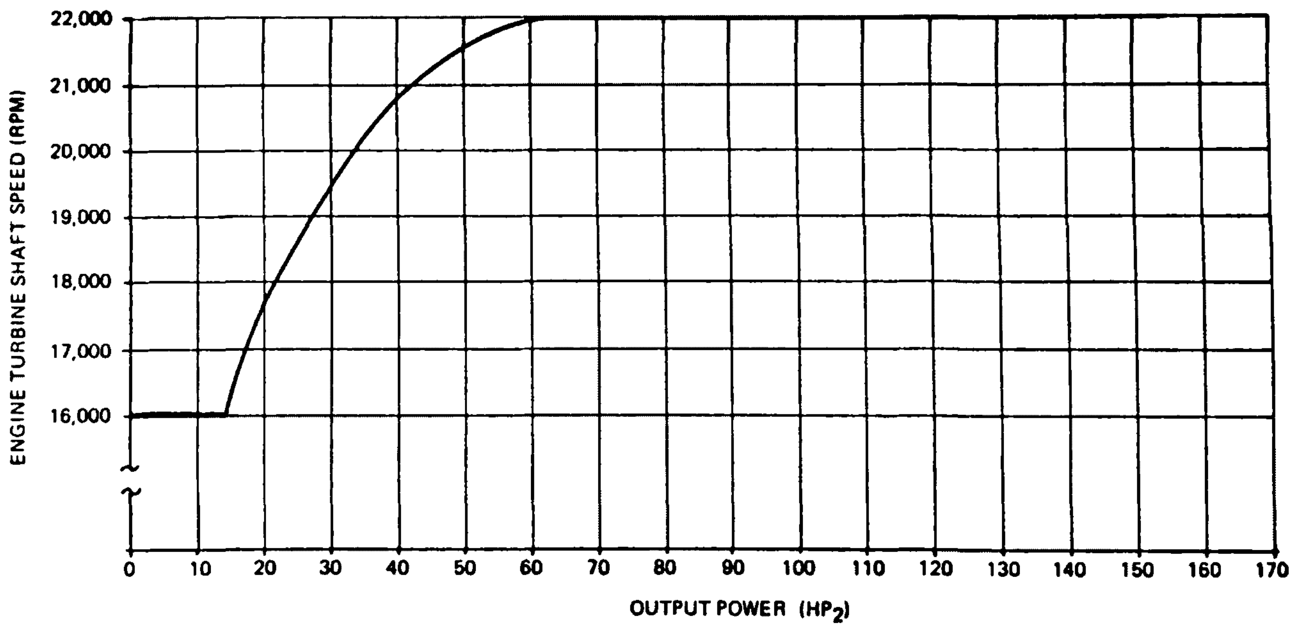

Figure I-4 Rankine Engine Performance - Speed vs. Output Horsepower

Sundstrand Aviation

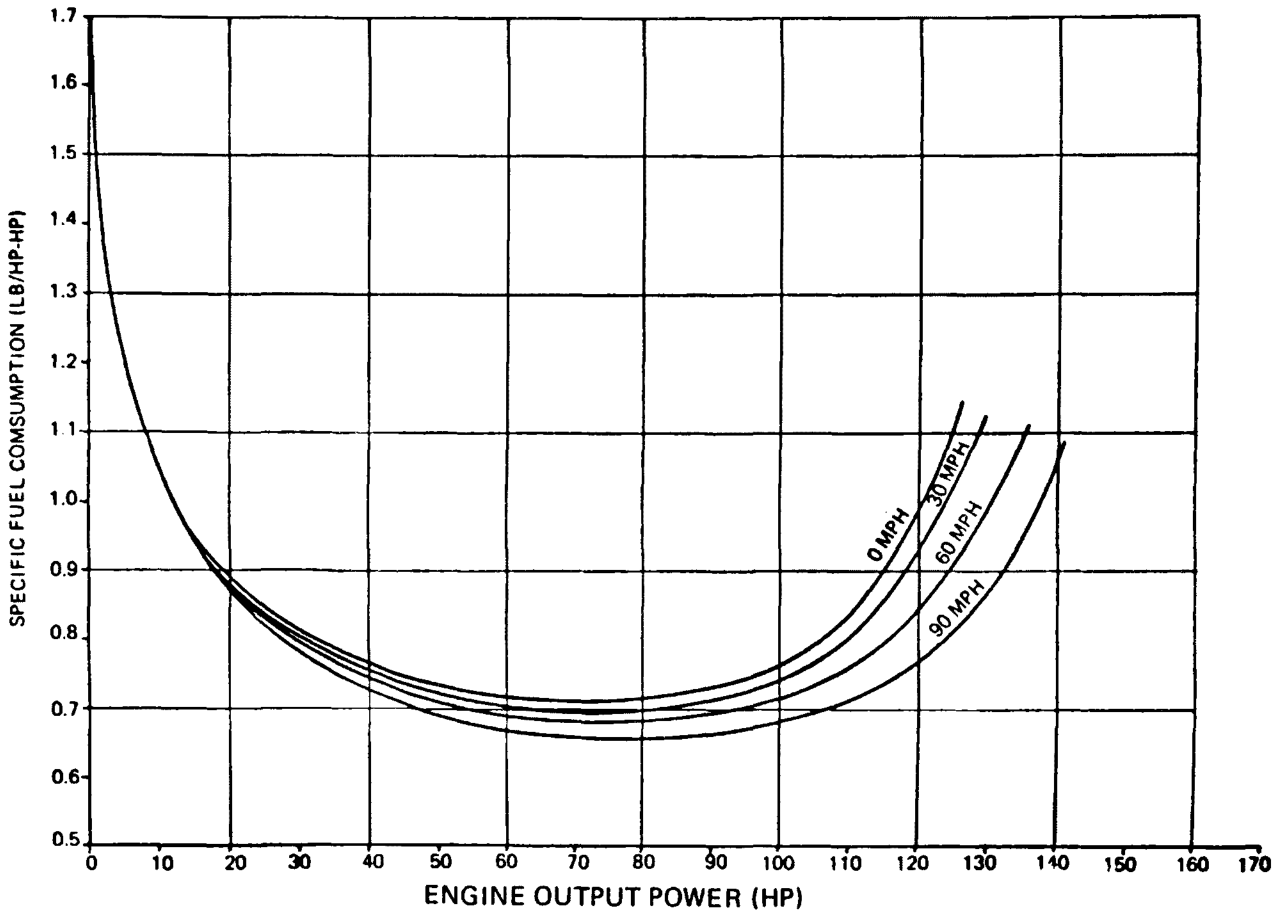

Figure I-4A   Rankine Engine Performance - Specific Fuel Consumption vs. Output Horsepower

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APPENDIX I-5. BRAYTON ENGINE DATA

To evaluate the performance characteristics of the EPA car with the AiResearch Brayton engine and an infinitely variable transmission, it was necessary to define in detail the characteristics of the engine. The optimum engine operating speed as a function of required engine power was determined, as well as the specific fuel consumption characteristics.

Assumed engine data is based upon information supplied by AiResearch. Engine speed ranges from 60% speed at idle to 105% speed for short duration bursts at maximum power.

However, within the normal operating speed range of 60% to 100% speed, the highest continuous permissible operating conditions of temperature and inlet guide vane position are 1700°F and 1.00 respectively.

From 0 to 15 HP, the minimum specific fuel consumption (min. SFC) is obtained at 60% speed. Above 15 HP the min. SFC engine speed curve was assumed to coincide with the curve that defines the 1700°F/1.00 curve mentioned above, since operating above that curve for any length of time is detrimental to the life of the engine (see graph, Figure I-5).

When the power requirements exceed approximately 76 HP, engine speed is allowed to increase above 100% speed for short periods of acceleration (see graph, Figure 1-5B).

Specific fuel consumption in pounds per horsepower hour was calculated from data supplied by AiResearch (see graphs, Figure 1-5 A).

Engine power as used by Sundstrand implies total engine output power, that is, input power to the transmission plus vehicle accessory power. Since AiResearch had assumed a constant 4 HP vehicle accessory load, it was necessary to add 4 HP to all the data to find total engine power.

No engine reduction gear mesh had been assumed by AiResearch, therefore, it was necessary to include this power loss in the fuel consumption calculations.

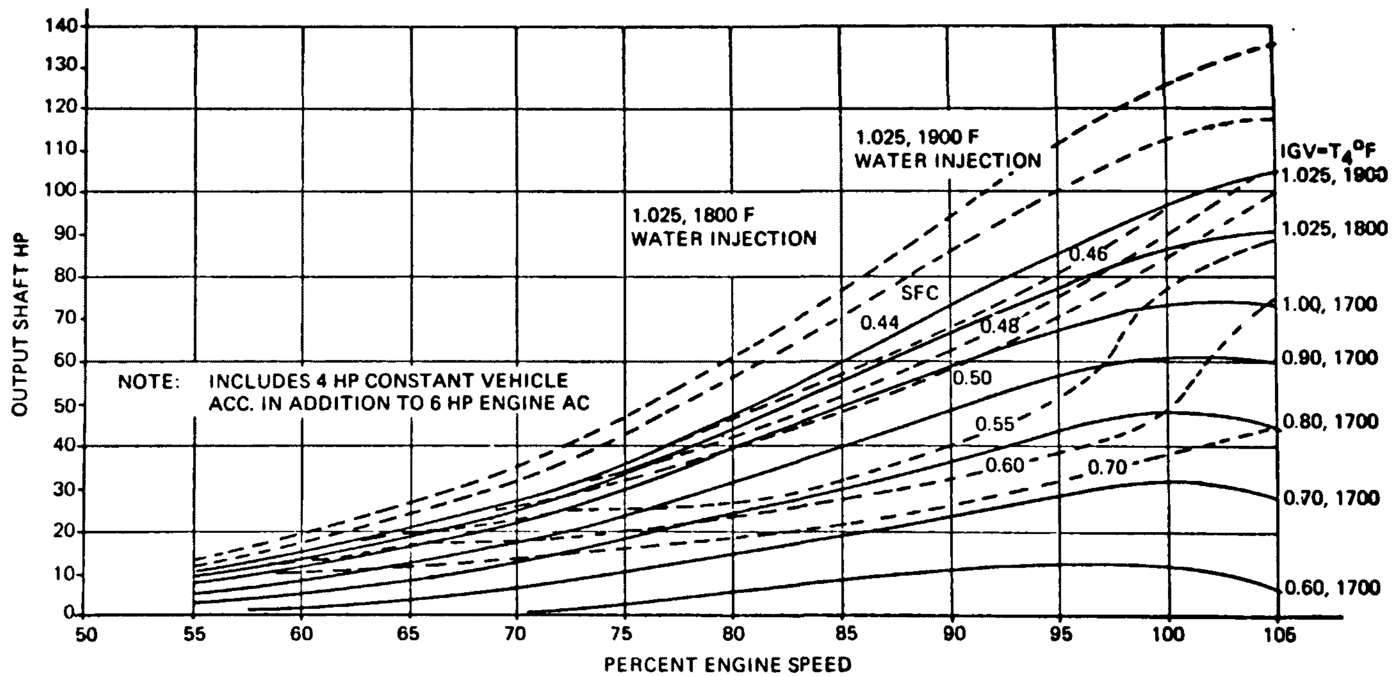


Figure I-5 Brayton Engine Output Shaft HP vs. Percent Engine Speed

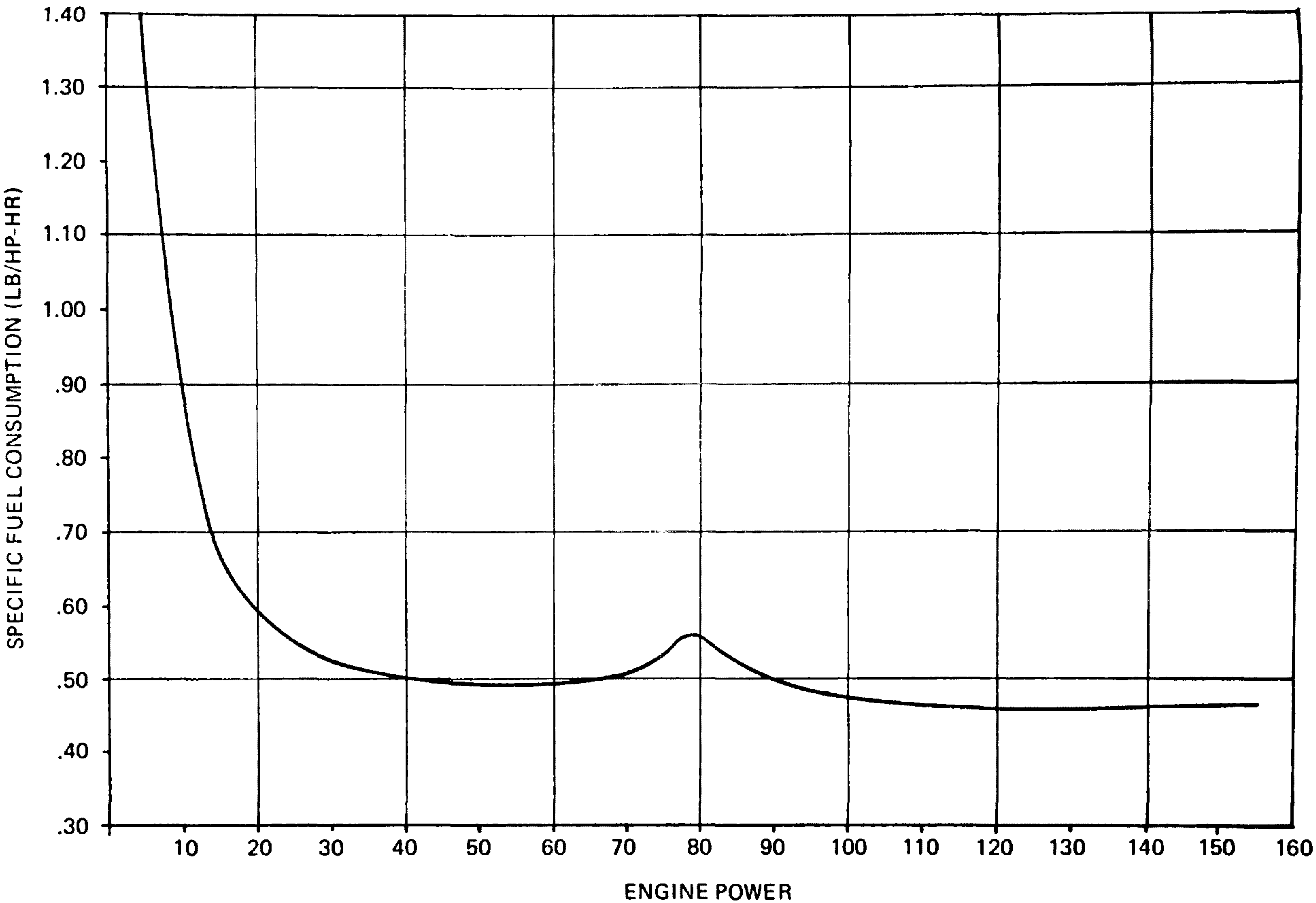


Figure I-5A Brayton Engine Specific Fuel Consumption vs. Engine Output HP

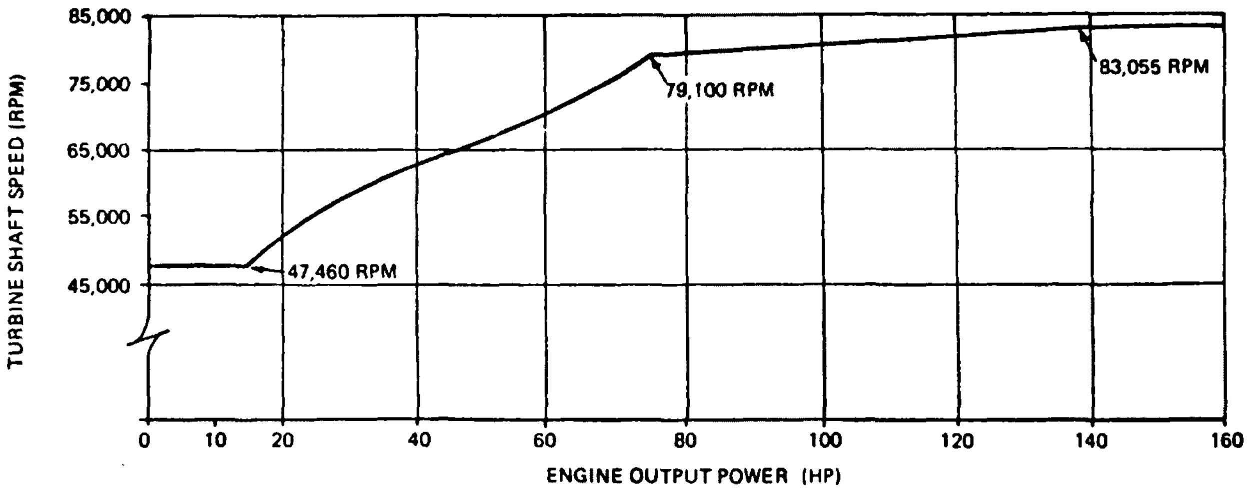


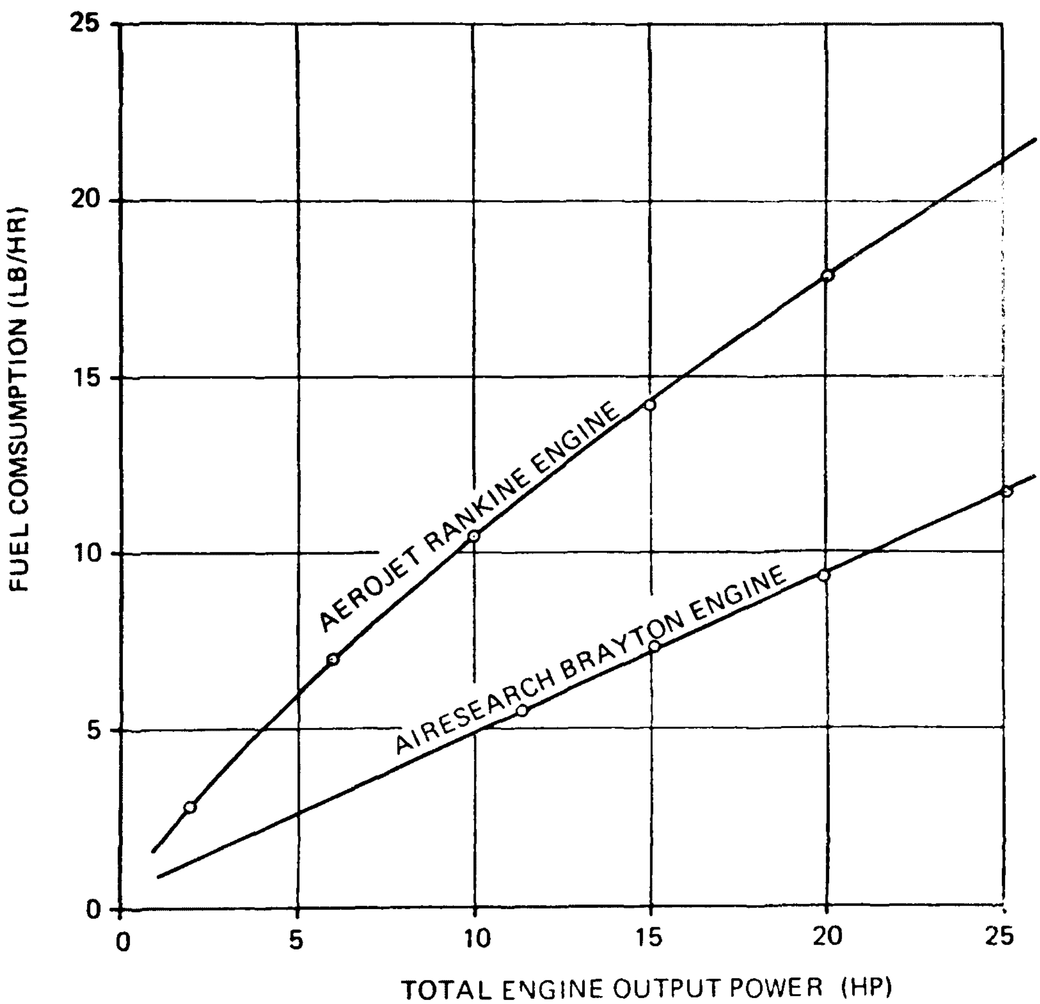
Figure I-5B Brayton Engine Speed vs. Engine Output HP

APPENDIX I-6. IDLE FUEL CONSUMPTION

Idle fuel consumption has proven to be an important parameter in this study, since a good deal of the duty cycle is at idle. It was calculated from the specific fuel consumption data presented in Appendicies I-4 and I-5. The formula for fuel consumption rate is: specific fuel consumption ÷ engine power.

$$\frac{LB}{HR} = \frac{\frac{LB}{HP-HR}}{HP_e}$$

Engine power is total output engine power, that is, it includes the power absorbed by the transmission at engine idle, and vehicle accessory power.



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APPENDIX I-7 MAXIMUM TOTAL ENGINE POWER VS. VEHICLE SPEED

The engine data received from EPA indicated that the Aerojet Rankine cycle engine had a maximum output power that increased with vehicle speed. This increase was due to the ram air effect on condenser cooling. The Airesearch gas turbine has a maximum output power which is not affected by vehicle speed. The power characteristics of the engine are shown in Figure I-7.

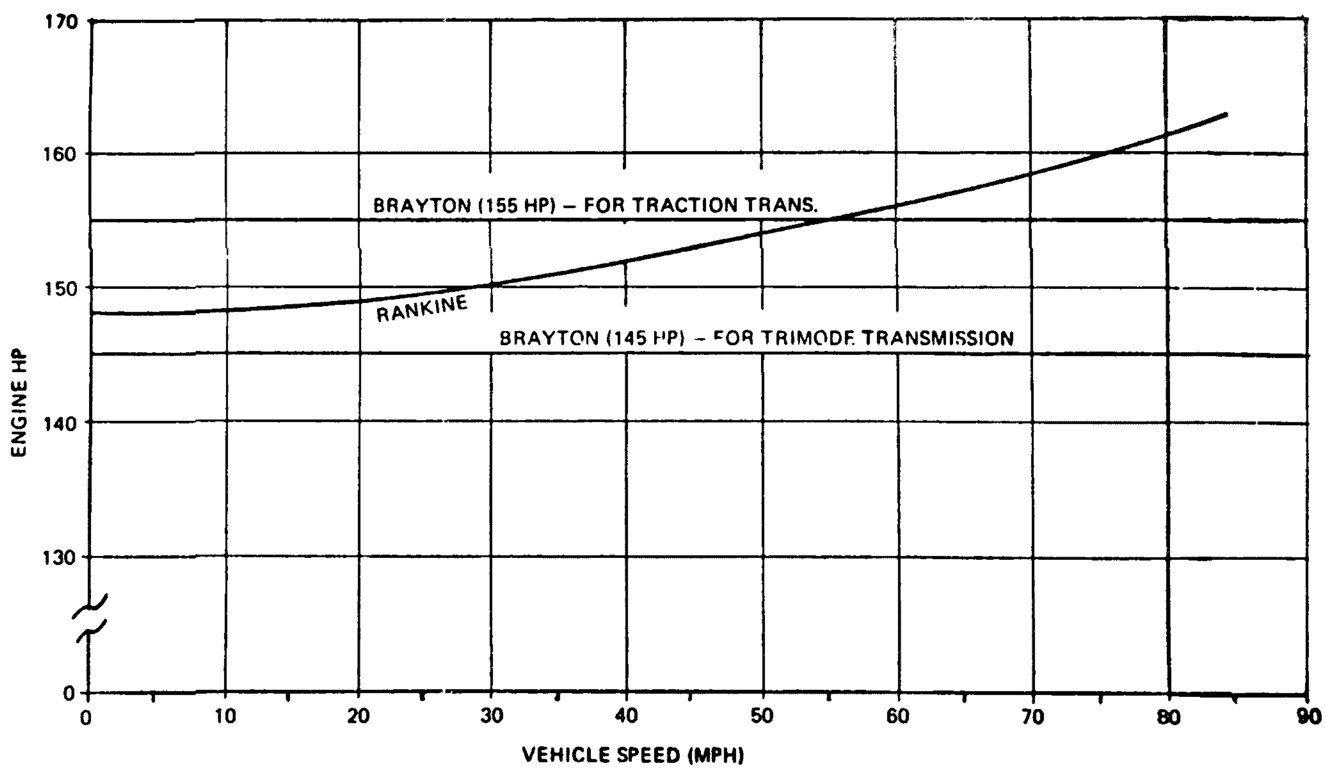


Figure I-7   Maximum Total Engine Power vs. Vehicle Speed

APPENDIX I-8. VEHICLE ACCESSORY POWER REQUIREMENTS

The vehicle accessory power requirements are defined in the Prototype Vehicle Performance Specification, Section 14, page 10, as follows:

Accessory Power Requirements

a. Accessory power requirements with the air conditioning in operation are defined as 15 HP at maximum engine speed and 4 HP at engine idle speed, with a linear relationship between these two points.

b. Accessory power requirements without the air conditioning in operation are defined as 5 HP at maximum engine speed and 2 HP at engine idle speed, with a linear relationship between these two points.

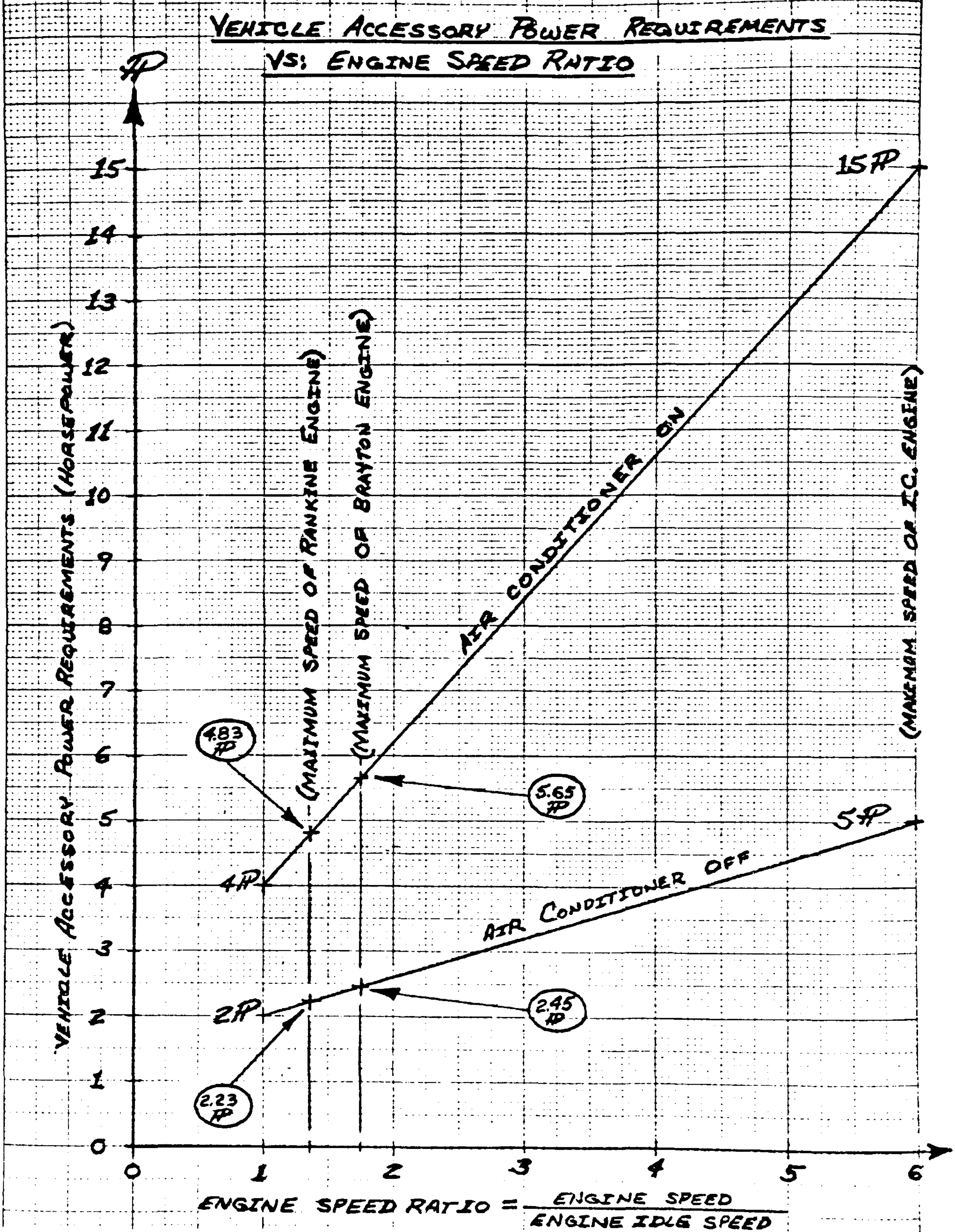
These accessory loads were based upon the operating speed range of a typical internal combustion (I.C.) engine. (Operating speed range is the maximum engine speed divided by the engine idle speed.) The operating speed range of a typical I.C. engine is 6:1.

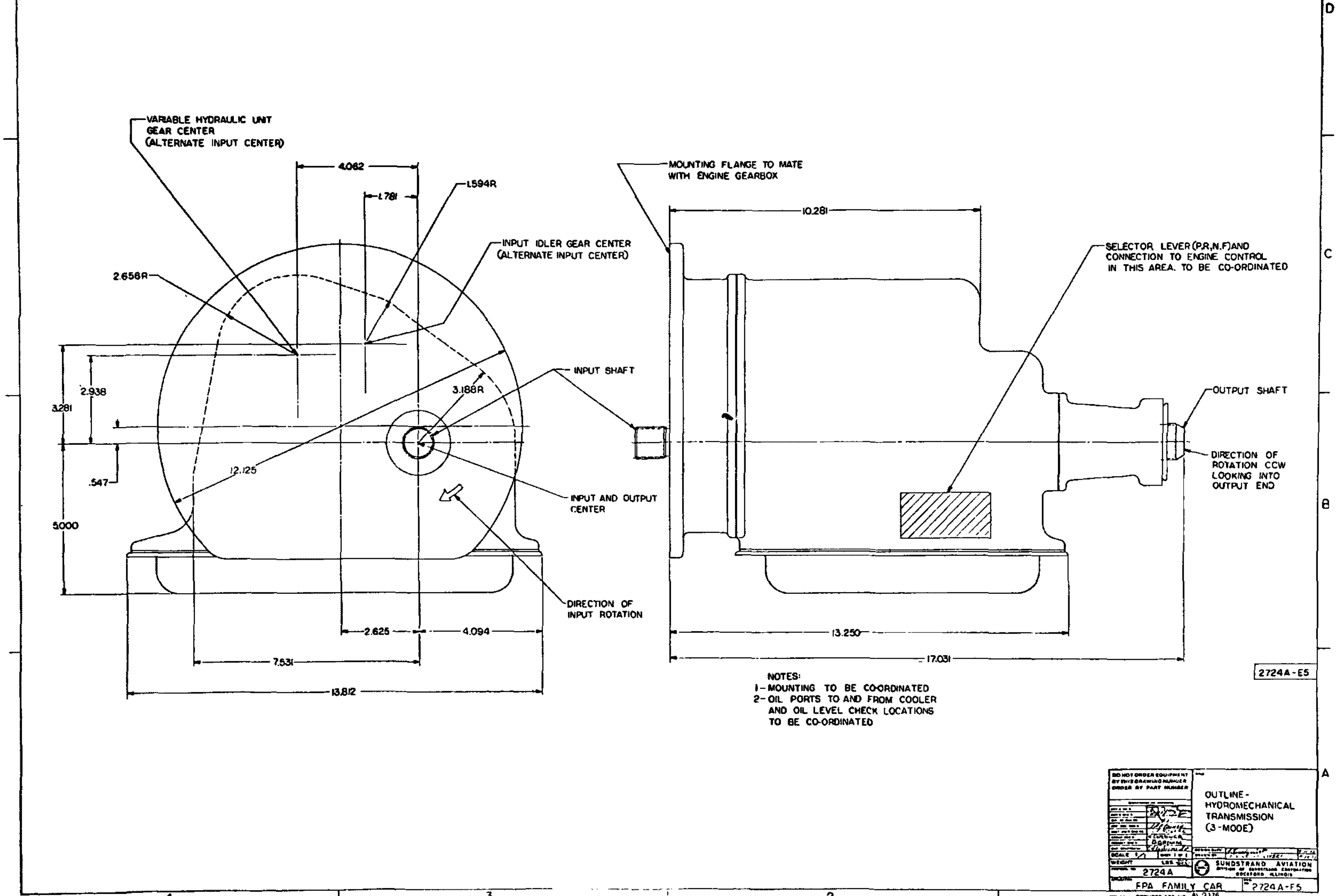
However, the operating speed range of the Rankine and Brayton engines used in this study are considerably narrower. (1.375:1 for the Rankine engine, and 1.750:1 for the Brayton engine.)

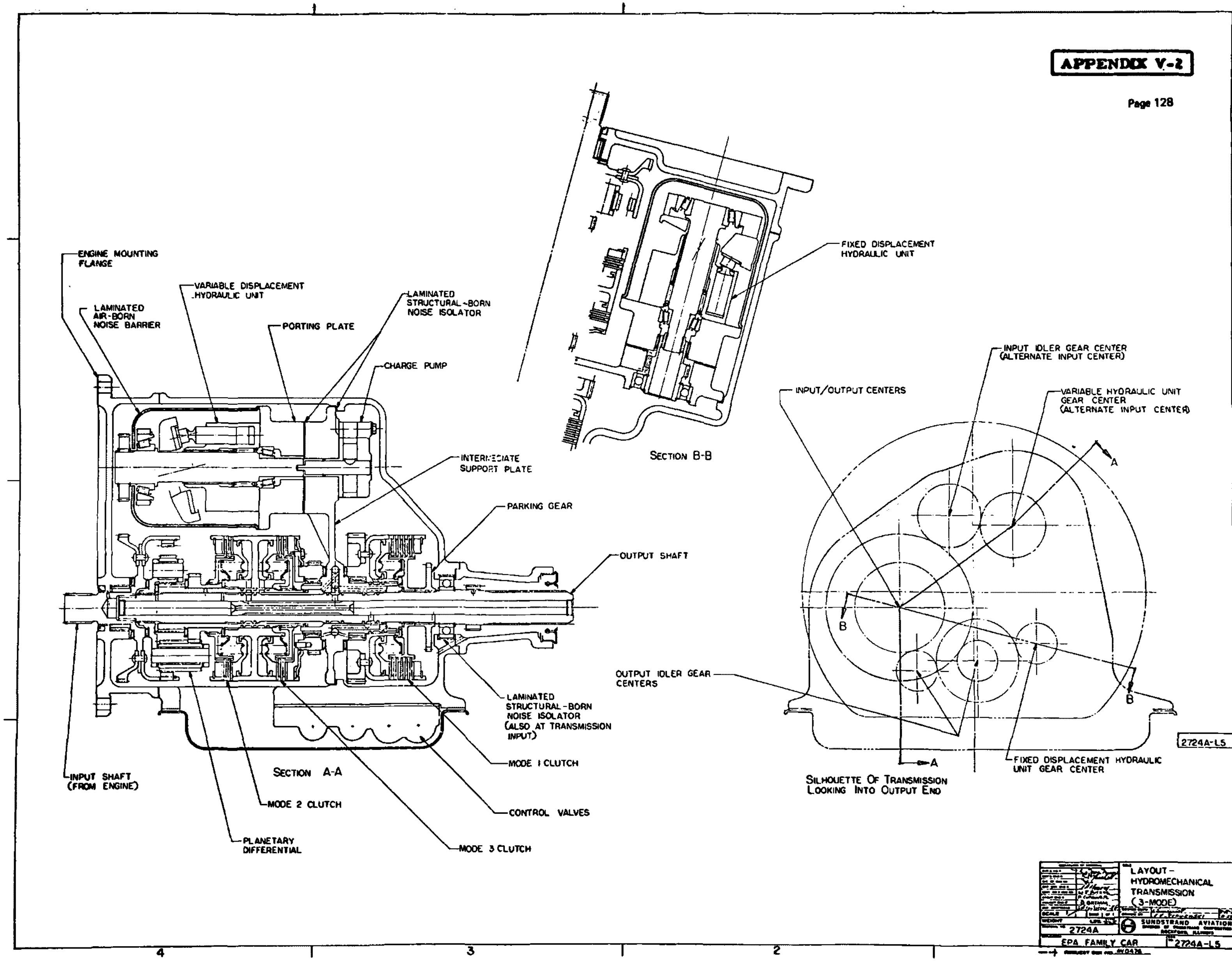
It was assumed that vehicle accessory loads, and therefore vehicle accessory speeds, would be the same at engine idle regardless of the type of engine that was used.

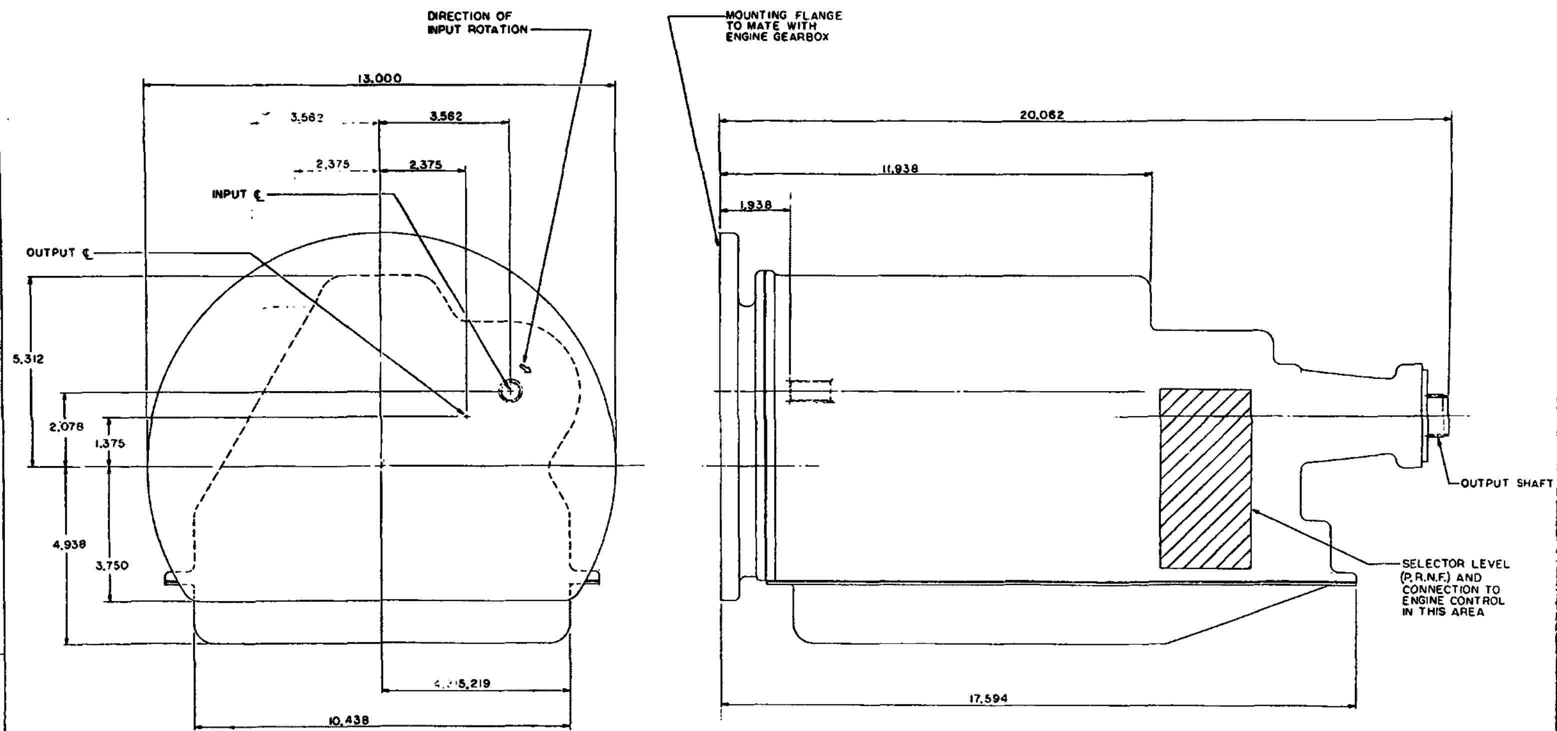
Then obviously at maximum engine speed, the vehicle accessory speeds and loads should be less with an engine with a narrower speed range. Therefore, it seemed unreasonable to apply the same maximum accessory power requirement to different engines with different speed ranges.

Consequently, with the agreement of the EPA, the accessory power requirement at maximum engine speed was factored down linearly as a function of maximum engine speed ratio (see graph, Figure


VEHICLE ACCESSORY POWER REQUIREMENTS
VS: ENGINE SPEED RATIO
HP
VEHICLE ACCESSORY POWER REQUIREMENTS (HORSEPOWER)
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
15 HP
AIR CONDITIONER ON
(MAXIMUM SPEED OF RANKINE ENGINE)
(MAXIMUM SPEED OF BRAYTON ENGINE)
(MAXIMUM SPEED OF I.C. ENGINE)
4.83 HP
5.65 HP
4 HP
5 HP
AIR CONDITIONER OFF
2 HP
2.45 HP
2.23 HP
0
1
2
3
4
5
6
ENGINE SPEED RATIO = ENGINE SPEED / ENGINE IDLE SPEED


VARIABLE HYDRAULIC UNIT GEAR CENTER (ALTERNATE INPUT CENTER)
4.062
1.781
1.594R
INPUT IDLER GEAR CENTER (ALTERNATE INPUT CENTER)
2.656R
2.938
3.281
3.188R
INPUT SHAFT
12.125
.547
5.000
INPUT AND OUTPUT CENTER
DIRECTION OF INPUT ROTATION
2.625
4.094
7.531
13.812
MOUNTING FLANGE TO MATE WITH ENGINE GEARBOX
10.281
SELECTOR LEVER (P,R,N,F) AND CONNECTION TO ENGINE CONTROL IN THIS AREA. TO BE CO-ORDINATED
OUTPUT SHAFT
DIRECTION OF ROTATION CCW LOOKING INTO OUTPUT END
13.250
17.031
NOTES:
1-MOUNTING TO BE CO-ORDINATED
2-OIL PORTS TO AND FROM COOLER AND OIL LEVEL CHECK LOCATIONS TO BE CO-ORDINATED
2724A-E5
DO NOT ORDER EQUIPMENT BY THIS DRAWING NUMBER ORDER BY PART NUMBER
OUTLINE-HYDROMECHANICAL TRANSMISSION (3-MODE)
SUNDSTRAND AVIATION
ROCKFORD, ILLINOIS
2724A
EPA FAMILY CAR
2724A-E5
D
C
B
A
4
3
2
1


ENGINE MOUNTING FLANGE
LAMINATED AIR-BORN NOISE BARRIER
VARIABLE DISPLACEMENT HYDRAULIC UNIT
LAMINATED STRUCTURAL-BORN NOISE ISOLATOR
PORTING PLATE
CHARGE PUMP
INTERMEDIATE SUPPORT PLATE
PARKING GEAR
OUTPUT SHAFT
LAMINATED STRUCTURAL-BORN NOISE ISOLATOR (ALSO AT TRANSMISSION INPUT)
MODE 1 CLUTCH
CONTROL VALVES
MODE 3 CLUTCH
PLANETARY DIFFERENTIAL
MODE 2 CLUTCH
INPUT SHAFT (FROM ENGINE)
SECTION A-A
FIXED DISPLACEMENT HYDRAULIC UNIT
SECTION B-B
INPUT IDLER GEAR CENTER (ALTERNATE INPUT CENTER)
INPUT/OUTPUT CENTERS
VARIABLE HYDRAULIC UNIT GEAR CENTER (ALTERNATE INPUT CENTER)
OUTPUT IDLER GEAR CENTERS
FIXED DISPLACEMENT HYDRAULIC UNIT GEAR CENTER
SILHOUETTE OF TRANSMISSION LOOKING INTO OUTPUT END
A
B
2724A-L5
LAYOUT - HYDROMECHANICAL TRANSMISSION (3-MODE)
SUNDSTRAND AVIATION
2724A
EPA FAMILY CAR
2724A-L5



NOTES
1-ENGINE MOUNTING TO BE CO-ORDINATED
2-OIL PORTS TO AND FROM COOLER AND OIL LEVEL CHECK LOCATIONS TO BE CO-ORDINATED

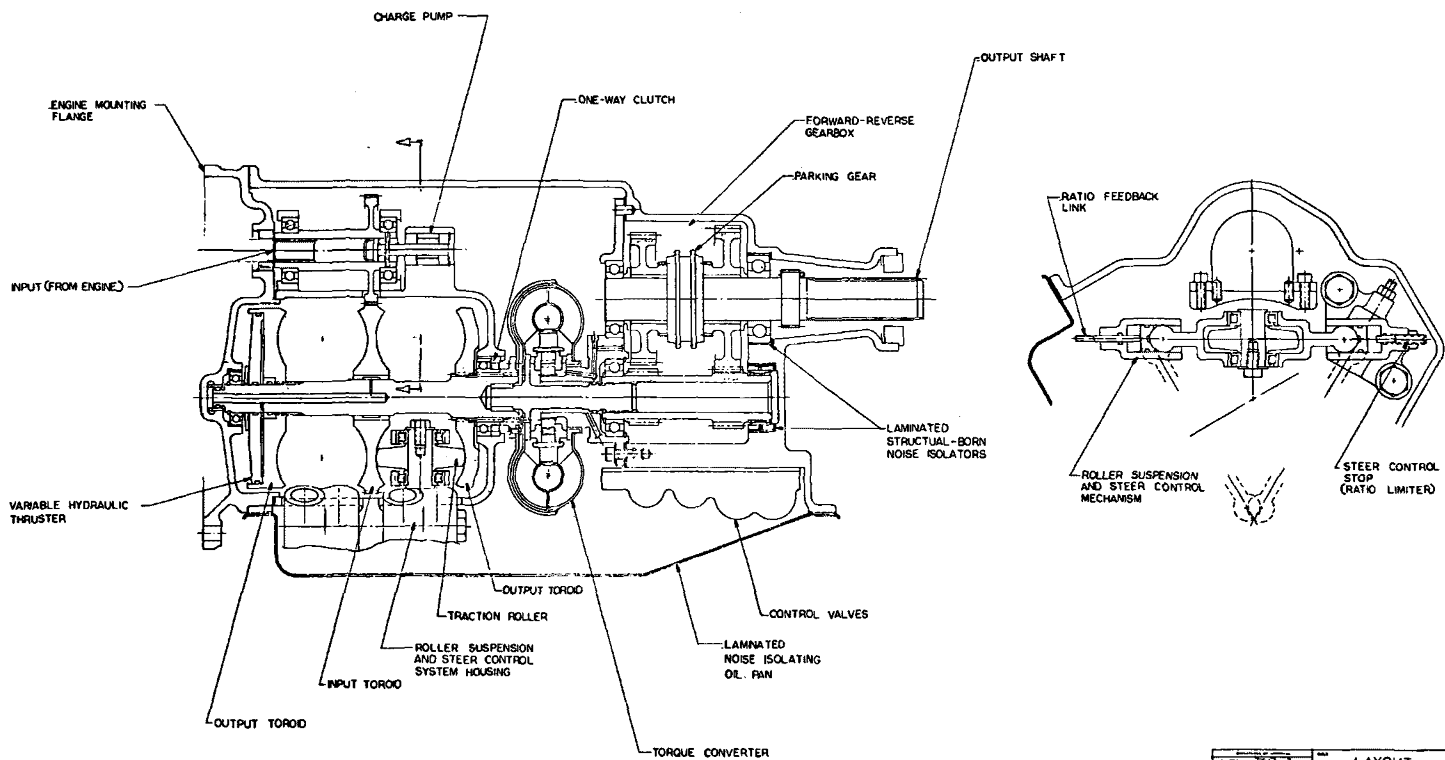
OUTLINE TRACTION TRANSMISSION

SCALE FULL

SUNDSTRAND AVIATION

EPA FAMILY CAR

2724A-E4



LAYOUT, TRACTION TRANSMISSION

SCALE FULL

SUNDSTRAND AVIATION

EPA FAMLY CAR

2724A-L4

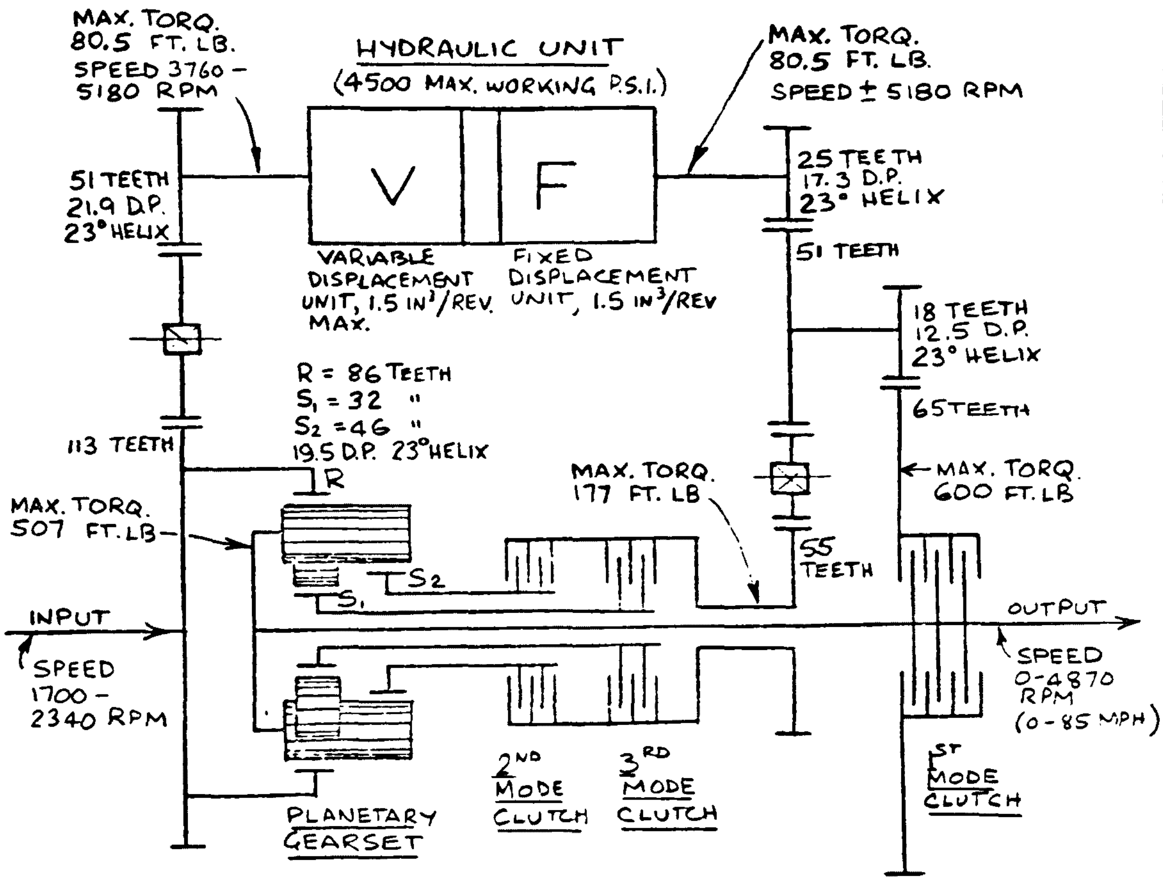
AV0473

2724A-L4

SHEET 1 OF 3
DATE
BY

TRI-MODE HYDROMECHANICAL TRANSMISSION
- AEROJET RANKINE ENGINE APPLICATION

SCHEMATIC SUMMARY



Form G 7736

SHEET 2 OF 3
DATE
BY

HYDROMECHANICAL TRANSMISSION
FOR AEROJET RANKINE ENGINE

COMPONENT SIZING SUMMARY

1. CLUTCHES
   - SYNCHRONOUS SHIFTING – SIZED BY MAX. STATIC TORQUE, NOT ENERGY DISSIPATION
   - MAXIMUM CLUTCH LINING UNIT PRESSURE = 550 PSI
   - STATIC COEFFICIENT OF FRICTION = .12
   - OVERLOAD CAPACITY AT MAX. TORQUE CONDITION — 36% MODE 1 CLUTCH, 80% MODE 2 AND 3 CLUTCHES.

2. SHAFTS AND SPLINES
   - MAXIMUM SHEAR STRESS DOES NOT EXCEED 60,000 PSI AT MAX. TORQUE.

3. GEARS
   - FACE WIDTH SIZED BY THE GREATER OF TWO CRITERIA —
     (1) MAXIMUM ALLOWABLE A.G.M.A. BENDING STRESS OF 120,000 PSI AT MAX. TORQUE.
     (2) MINIMUM HELICAL CONTACT RATIO OF 1.1 (FOR GEAR NOISE CONSIDERATION)

Form G 7756

SHEET 3 OF 3
DATE
BY

HYDROMECHANICAL TRANSMISSION
FOR AEROJET RANKINE ENGINE

COMPONENT SIZING SUMMARY

4. ANTI-FRICTION BEARINGS

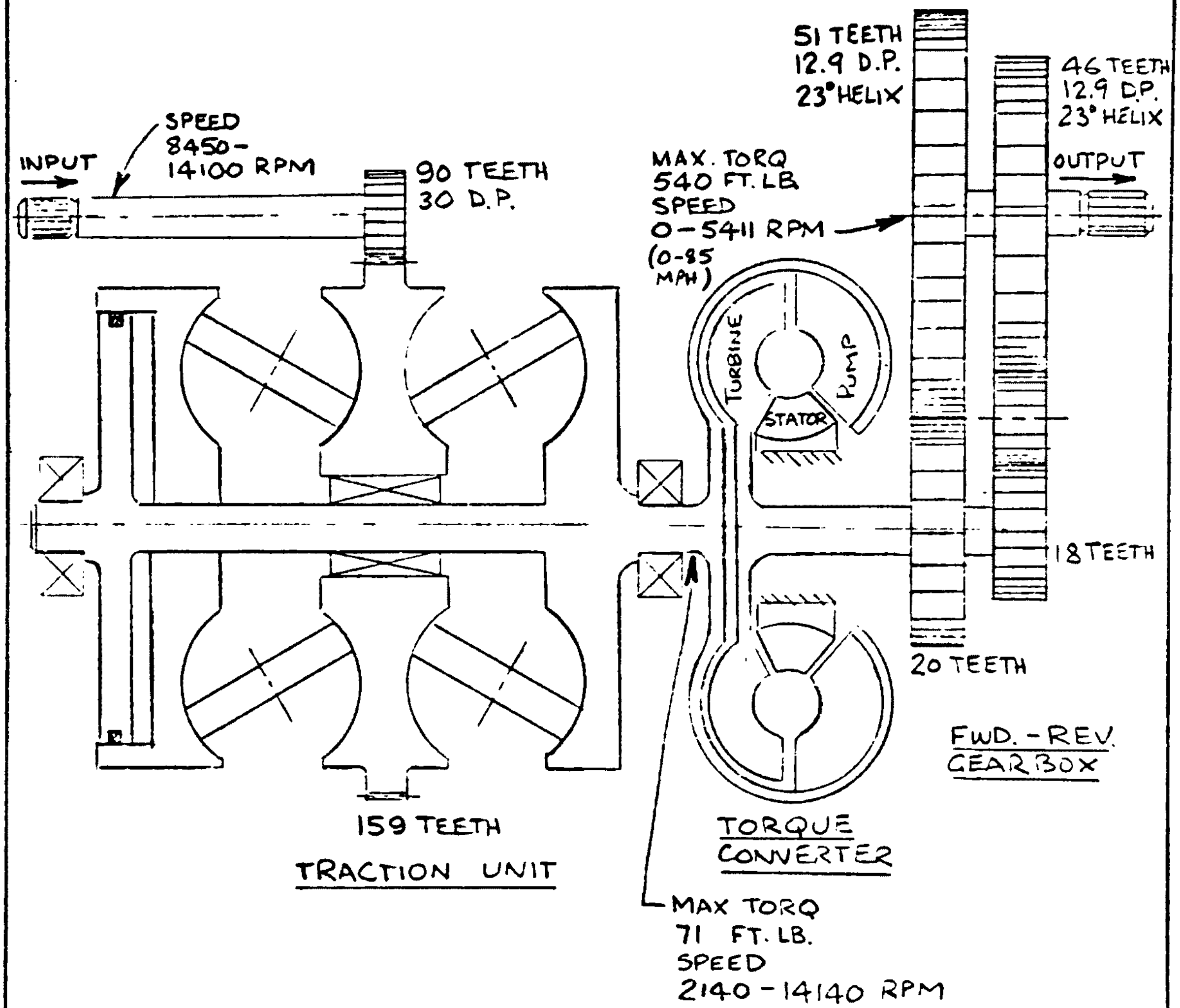
- MEAN LOAD USED FOR LIFE-SIZING ARBITARILY ASSUMED TO BE 1/5 OF MAX. LOAD. (NOTE, THE MEAN INPUT POWER TO THE TRANS. OVER THE FULL EPA SPECIFIED DRIVING CYCLE IS ABOUT 20 HP. THE ENGINE CAN PUT OUT 160 HP)
- A.F.B.M.A. PROCEDURES USED TO SIZE THE BEARINGS FOR A 3500 HOURS MINIMUM LIFE.

Form G 7756

SHEET 1 OF 2
DATE
BY

TRACTION DRIVE TRANSMISSION
- AEROJET RANKINE ENGINE APPLICATION

SCHEMATIC SUMMARY



Form G 7?.

SHEET 2 OF 2

DATE

BY

TRACTION DRIVE TRANSMISSION
- AEROJET RANKINE ENGINE

COMPONENT SIZING SUMMARY

SHAFTS, SPLINES, GEARS AND BEARINGS WERE SIZED TO THE SAME CRITERIA AS SHOWN FOR THE HYDROMECHANICAL TRANSMISSION.

Form G 7756

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APPENDIX VI-1. HYDROMECHANICAL TRANSMISSION/COMPUTER PERFORMANCE PROGRAM

The purpose of this program was to determine transmission and vehicle performance of a simulated vehicle with a hydromechanical transmission for a given duty cycle. The program also computes speeds, torques, horsepower, working pressure in the hydraulic units, power losses, and efficiency.

The required input parameters are listed below:

- Vehicle parameters
- Engine specification
- Fuel consumption data
- Accessory specification
- Duty cycle definition
- Planetary dimensions
- Gear ratios and efficiency
- Hydraulic unit displacement
- Charge pump pressure range

The program is a discrete simulation which calculates the conditions necessary within the system to obtain the desired response. For each duty cycle condition, the program calculates the required data. The following paragraphs explain in detail how the program works.

For each incremental time interval the vehicle speed, acceleration, drag, and tractive effort are defined by given input data or predetermined duty cycles available as subroutines. An initial estimate of required engine speed is made, and a check is made to be sure that the estimate is greater than the minimum possible engine speed for the vehicle speed in question.

Next, the speeds of the various components of the system are calculated. Since the transmission is a multi-mode transmission, a check is made to determine which mode is correct for the calculated speed conditions.

After the speeds have been determined, the torques and horsepower in the various transmission elements are calculated. Horsepower loss in the hydraulic units is calculated by a separate subroutine. Torque and horsepower are found by a trial and error procedure. A working pressure (which controls system torque) is assumed and hydraulic unit losses are calculated. Then the equations of dynamic equilibrium are solved to find the unknown torques in the system. The working pressure is then recalculated. If the recalculated working pressure differs by more than 10 psi from the assumed working pressure, the assumed working pressure is modified, and the whole process is repeated until it iterates to a solution.

The horsepower flow in the elements of the transmission are found from the torques and speeds determined above.

Then the other power losses that occur in the transmission are calculated. These losses include: charge pump power requirements, planetary gearset losses, transfer gear losses, and open clutch power loss.

The total power loss, the total power required from the engine to the transmission, and the transmission efficiency are calculated. The total engine power includes the vehicle accessory power requirements.

The calculated engine speed is compared with the engine speed that would yield the minimum specific fuel consumption for the required power level. If the engine speeds do not correlate within one percent, a new engine speed is assumed, and the process is repeated until it iterates to a solution.

When a solution is achieved at minimum specific fuel consumption, engine speed, fuel consumption and energy efficiency are calculated. Then the calculated output is printed, and the program goes on to the next condition of the duty cycle.

When the driving cycle is completed, a summary of fuel consumption, energy efficiency, average power requirements, distance, and time is printed.

A typical readout is included as part of this appendix. The example presented here is of the EPA car at test weight. The engine is the Aerojet prototype configuration Rankine engine. The assumed vehicle accessory load includes the air conditioner, but the accessory load at maximum engine speed was scaled down as a function of maximum engine speed ratio because of the narrow operating speed range of the engine (see Appendix I-8).

The transmission featured in this computer run was the Sundstrand Tri-Mode hydromechanical transmission.

The duty cycle assumed was the complete Combined Driving Cycle, which includes the Federal Driving Cycle, the Simplified Suburban Route, and the Simplified Country Route.

The readout includes the input parameters for each driving cycle, the output parameters for each driving cycle condition, and a summary of the results of each of the driving cycles. (Only the first 127 seconds and the last 135 seconds of the Federal Driving Cycle are shown here.) Also included at the end of the readout is a summary of the combined duty cycle.

```
FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION
--------------------------------------------------

 8-17-72 RUN 1.03 EPA TURBINE CAR  FEDRAL DRIVING CYCLE

 AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

INPUT PARAMETERS
----------------

VEHICLE WEIGHT (LB)..................................WT=  4600.000
WHEEL INERTIA (FT-LB-SEC2)...........................WI=    11.200
FRONTAL AREA (FT2)...................................AF=    20.000
COEFFICIENT OF DRAG..................................CD=     0.600
FUEL DENSITY (LB/GAL)................................DEN=    6.290
ENGINE CODE..........................................ENG=    3.000
DUTY CYCLE CODE......................................DUTY=   2.000
OUTPUT CODE..........................................OUT=    2.000

NOMOGRAPH DIMENSION..................................A=  0.4100
NOMOGRAPH DIMENSION..................................B=  0.6300
NOMOGRAPH DIMENSION..................................C=  1.0000
NEMAX/NOMAX.......................................SRATE=  4.3021
LB/HR AT  0 MPH...................................FPH1 =  7.0000
LB/HR AT 85 MPH...................................FPH2 = 14.2000

ENGINE SPEED / ENGINE LINK SPEED.....................R1=  9.4184 (0.MESH)
V-UNIT SPEED / V-UNIT LINK SPEED.....................R2=  2.2157 (2.MESH)
MODE 3 F-UNIT SPEED / MODE 3 F-UNIT LINK SPEED.......R3=  2.2000 (3.MESH)
MODE 2 F-UNIT SPEED / MODE 2 F-UNIT LINK SPEED.......R4=  2.2000 (3.MESH)
MODE 1 F-UNIT SPEED / MODE 1 F-UNIT LINK SPEED.......R5=  7.3667 (2.MESH)
TRANS OUTPUT LINK SPEED / TRANS OUTPUT SPEED.........R6=  1.0000 (0.MESH)
TRANS OUTPUT SPEED / AXLE SPEED......................FA=  4.5000 (0.950EFF)

HYD UNIT DISPLACEMENT................................DISP=    1.500
WHEEL DIAMETER.......................................DW=     26.400
GEAR MESH EFFICIENCY.................................EG=      0.985
MODE 2 PLANETARY CARRIER LINK........................ICAR2=   4.000
MODE 3 PLANETARY CARRIER LINK........................ICAR3=   1.000
MAXIMUM CLUTCH LOSS HORSEPOWER.......................FLCLK=   1.500
MINIMUM CHARGE PUMP PRESSURE.........................PCN=    60.000
MAXIMUM CHARGE PUMP PRESSURE.........................PCX=   250.000
```

APPENDIX VI -1 COMPUTER RUN (first 127 seconds)

FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION

8-17-72 RUN 1.03 EPA TURBINE CAR FEDERAL DRIVING CYCLE

AEROJET RANKINE ENGINE 1MT W/1.5 IN. LCG

OUTPUT PARAMETERS

VEHICLE					ENGINE				TRANSMISSION			ROAD		FUEL			
SPEED MPH	ACCEL F/S/S	TIME INC	TIME CUM	DIST FT	SPEED RPM	ACCES HP	TOTAL HP	TOTAL FT-LB	SPEED RATIO	DELTA PRES	TOTAL EFF	DRAG LB	TRACT LB	LB/HPH	LB/HP	MPG INST	MPG CUM
0.0	0.0	1.0	1.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	2.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	3.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	4.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	5.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	6.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	7.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	8.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	9.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	10.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	11.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	12.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	13.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	14.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	15.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	16.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	17.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	18.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.0	0.0	1.0	19.0	0.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	0.0
0.4	2.16	1.0	20.0	1.	16000.	4.00	7.0	2.3	0.006	991.	13.9	71.	400.	1.125	7.87	0.29	0.02
2.9	4.33	1.0	21.0	5.	16000.	4.00	14.1	4.6	0.045	1429.	53.9	71.	730.	0.960	13.54	1.37	0.13
5.5	4.22	1.0	22.0	13.	17321.	4.26	21.7	6.4	0.081	1381.	67.5	73.	714.	0.874	19.00	1.94	0.33
8.7	4.11	1.0	23.0	26.	19286.	4.45	29.7	8.1	0.111	1396.	67.7	74.	699.	0.913	24.12	2.27	0.57
11.5	4.03	1.0	24.0	43.	20557.	4.63	38.7	9.9	0.139	1211.	65.4	77.	690.	0.764	29.56	2.44	0.81
14.2	3.56	1.0	25.0	64.	20721.	4.65	41.0	10.4	0.169	1221.	74.7	79.	691.	0.754	30.94	2.88	1.06
16.6	2.42	1.0	26.0	88.	19376.	4.44	30.5	9.3	0.212	811.	80.8	82.	450.	0.805	24.54	4.26	1.34
17.5	0.88	1.0	27.0	114.	16928.	4.13	17.5	5.4	0.255	318.	79.7	83.	217.	0.913	15.97	6.88	1.64
18.2	1.76	1.0	28.0	141.	18663.	4.37	26.0	7.3	0.241	607.	83.4	84.	351.	0.836	21.71	5.28	1.88
19.9	2.64	1.0	29.0	170.	20267.	4.59	36.2	9.4	0.242	882.	86.2	85.	488.	0.773	27.97	4.47	2.09
21.5	1.61	1.0	30.0	202.	18882.	4.40	27.2	7.6	0.291	696.	88.6	88.	334.	0.826	22.44	6.02	2.33
22.1	0.59	1.0	31.0	234.	17110.	4.15	17.7	5.4	0.319	554.	81.9	89.	179.	0.911	16.09	8.62	2.59
22.4	-0.07	1.0	32.0	267.	16000.	4.00	11.3	3.7	0.345	784.	67.1	90.	79.	1.014	11.57	12.22	2.87
22.0	-0.73	1.0	33.0	299.	0.	0.0	0.0	0.0	0.0	0.	0.0	89.	-23.	0.0	8.96	15.59	3.15
21.5	-0.77	1.0	34.0	331.	0.	0.0	0.0	0.0	0.0	0.	0.0	88.	-29.	0.0	8.92	15.30	3.40
20.9	-0.91	1.0	35.0	361.	0.	0.0	0.0	0.0	0.0	0.	0.0	87.	-35.	0.0	8.77	14.99	3.64
20.3	-1.65	1.0	36.0	391.	0.	0.0	0.0	0.0	0.0	0.	0.0	86.	-155.	0.0	8.72	14.60	3.85
18.7	-2.49	1.0	37.0	418.	0.	0.0	0.0	0.0	0.0	0.	0.0	84.	-295.	0.0	8.58	13.68	4.05
17.1	-2.02	1.0	38.0	443.	0.	0.0	0.0	0.0	0.0	0.	0.0	82.	-224.	0.0	8.45	12.70	4.22
15.9	-1.54	1.0	39.0	467.	0.	0.0	0.0	0.0	0.0	0.	0.0	81.	-153.	0.0	8.35	11.99	4.36
15.1	-0.55	1.0	40.0	489.	0.	0.0	0.0	0.0	0.0	0.	0.0	80.	-4.	0.0	8.28	11.43	4.48
15.2	0.44	1.0	41.0	511.	16000.	4.00	13.0	4.3	0.234	197.	69.8	80.	147.	0.980	12.74	7.49	4.56
15.6	0.81	1.0	42.0	534.	16347.	4.05	15.8	5.1	0.235	313.	75.2	81.	207.	0.974	14.80	6.60	4.62
16.3	1.17	1.0	43.0	559.	17353.	4.19	19.4	5.9	0.232	436.	78.3	81.	260.	0.953	17.31	5.91	4.67
17.2	2.05	1.0	44.0	583.	18355.	4.41	27.8	7.7	0.224	703.	81.8	83.	395.	0.823	22.86	4.74	4.67
19.1	2.53	1.0	45.0	611.	20493.	4.62	35.1	9.8	0.230	975.	85.0	85.	531.	0.764	25.13	4.12	4.64
21.0	2.13	1.0	46.0	642.	19591.	4.51	33.1	9.6	0.263	699.	87.7	87.	411.	0.754	25.49	5.17	4.66
22.0	1.32	1.0	47.0	674.	18505.	4.34	24.3	6.9	0.293	297.	83.5	89.	290.	0.847	20.63	6.70	4.73
22.8	0.55	1.0	48.0	709.	17149.	4.16	17.5	5.5	0.327	508.	81.2	90.	174.	0.909	15.21	8.82	4.86
22.7	-0.22	1.0	49.0	741.	16000.	4.00	10.0	3.3	0.350	232.	59.0	90.	57.	1.044	10.49	13.63	4.98
22.4	-1.43	1.0	50.0	774.	0.	0.0	0.0	0.0	0.0	0.	0.0	90.	-128.	0.0	8.97	15.61	5.13
20.8	-2.64	1.0	51.0	804.	0.	0.0	0.0	0.0	0.0	0.	0.0	87.	-314.	0.0	8.76	14.91	5.26
19.0	-2.49	1.0	52.0	832.	0.	0.0	0.0	0.0	0.0	0.	0.0	85.	-295.	0.0	9.41	13.87	5.38
17.4	-2.35	1.0	53.0	858.	0.	0.0	0.0	0.0	0.0	0.	0.0	83.	-274.	0.0	8.47	12.90	5.47
16.1	-0.49	1.0	54.0	881.	16000.	4.00	6.8	2.2	0.249	105.	14.2	81.	9.	1.132	7.67	13.21	5.56
16.7	1.30	1.0	55.0	906.	17957.	4.26	21.5	6.3	0.231	507.	79.8	82.	294.	0.872	18.82	5.60	5.56
17.8	2.13	1.0	56.0	932.	19159.	4.43	29.0	7.9	0.229	725.	82.9	83.	407.	0.815	23.61	4.74	5.53
19.6	2.86	1.0	57.0	961.	20494.	4.62	38.1	9.8	0.235	952.	85.8	85.	521.	0.764	29.10	4.24	5.49
21.5	2.39	1.0	58.0	993.	20174.	4.57	35.4	9.2	0.263	779.	88.4	89.	451.	0.776	27.47	4.92	5.46

APPENDIX VI -1 COMPUTER RUN (first 127 seconds)

VEHICLE SPEED MPH	VEHICLE ACCEL F/S/S	VEHICLE TIME INC	VEHICLE TIME CUM	VEHICLE DIST FT	ENGINE SPEED RPM	ENGINE ACCES HP	ENGINE TOTAL HP	ENGINE TOTAL FT-LB
22.9	1.51	1.0	59.0	1026.	15552.	4.49	31.5	9.4
24.1	1.21	1.0	60.0	1061.	18722.	4.37	25.9	7.2
24.5	0.51	1.0	61.0	1097.	17406.	4.19	19.8	5.7
24.8	0.15	1.0	62.0	1134.	16000.	4.00	15.0	4.9
24.7	-0.22	1.0	63.0	1170.	16000.	4.00	11.0	3.6
24.6	-0.07	1.0	64.0	1206.	16000.	4.00	12.5	4.1
24.6	0.07	1.0	65.0	1242.	16000.	4.00	14.1	4.6
24.7	0.04	1.0	66.0	1279.	16000.	4.00	13.7	4.5
24.7	0.0	1.0	67.0	1315.	16000.	4.00	13.3	4.4
24.7	-0.04	1.0	68.0	1351.	16000.	4.00	12.9	4.2
24.6	-0.07	1.0	69.0	1387.	16000.	4.00	12.5	4.1
24.7	0.33	1.0	70.0	1423.	16909.	4.12	16.9	5.3
25.1	0.73	1.0	71.0	1460.	18015.	4.28	21.7	6.3
25.5	0.29	1.0	72.0	1498.	16992.	4.14	17.1	5.3
25.5	-0.15	1.0	73.0	1535.	16000.	4.00	12.2	4.0
25.4	-0.22	1.0	74.0	1572.	16000.	4.00	11.3	3.7
25.2	-0.29	1.0	75.0	1609.	16000.	4.00	10.4	3.4
25.1	0.22	1.0	76.0	1646.	16502.	4.07	16.0	5.1
25.5	0.73	1.0	77.0	1684.	19081.	4.29	22.1	6.4
25.9	0.26	1.0	78.0	1722.	16969.	4.13	17.0	5.3
25.8	-0.22	1.0	79.0	1759.	16000.	4.00	11.6	3.9
25.8	0.26	1.0	80.0	1797.	16936.	4.13	16.9	5.2
26.2	0.73	1.0	81.0	1836.	18196.	4.30	22.7	6.5
26.8	1.06	1.0	82.0	1875.	18333.	4.40	27.1	7.5
27.6	1.39	1.0	83.0	1915.	19740.	4.51	32.1	8.5
28.6	1.14	1.0	84.0	1957.	19414.	4.47	30.0	8.1
29.2	0.88	1.0	85.0	2000.	19027.	4.42	27.5	7.6
29.8	0.66	1.0	86.0	2044.	18589.	4.37	25.3	7.1
30.1	0.44	1.0	87.0	2088.	18334.	4.32	22.9	6.6
30.4	0.33	1.0	88.0	2133.	18147.	4.30	21.8	6.3
30.5	0.22	1.0	89.0	2177.	17792.	4.24	20.6	6.1
30.7	0.0	1.0	90.0	2222.	17053.	4.15	18.2	5.6
30.5	-0.27	1.0	91.0	2267.	16000.	4.00	14.4	4.7
30.4	-0.11	1.0	92.0	2312.	16805.	4.11	16.5	5.2
30.4	0.0	1.0	93.0	2356.	17045.	4.14	17.9	5.5
30.4	0.0	1.0	94.0	2401.	17045.	4.14	17.9	5.5
30.4	0.0	1.0	95.0	2446.	17045.	4.14	17.9	5.5
30.3	-0.33	1.0	96.0	2490.	16000.	4.00	12.9	4.2
29.9	-0.44	1.0	97.0	2534.	16000.	4.00	8.7	2.9
29.6	-0.04	1.0	98.0	2577.	16332.	4.13	15.6	5.2
29.9	0.59	1.0	99.0	2621.	18565.	4.35	24.5	6.9
30.3	0.51	1.0	100.0	2666.	18506.	4.34	24.0	6.8
30.6	0.44	1.0	101.0	2711.	18433.	4.33	23.4	6.7
30.9	0.22	1.0	102.0	2756.	17801.	4.25	21.0	6.2
30.9	0.0	1.0	103.0	2801.	17309.	4.18	18.3	5.6
30.8	-0.40	1.0	104.0	2846.	16000.	4.00	11.9	3.9
30.3	-0.81	1.0	105.0	2891.	0.	0.0	0.0	0.0
29.9	-0.26	1.0	106.0	2935.	16000.	4.00	13.9	4.6
30.0	0.29	1.0	107.0	2979.	17377.	4.27	20.9	6.1
30.2	0.51	1.0	108.0	3023.	18457.	4.34	24.0	6.8
30.7	0.73	1.0	109.0	3068.	19031.	4.42	27.3	7.5
31.2	0.73	1.0	110.0	3114.	19140.	4.43	27.9	7.7
31.7	0.73	1.0	111.0	3160.	19254.	4.45	28.5	7.9
32.1	0.37	1.0	112.0	3207.	18489.	4.36	24.3	6.9
32.2	0.0	1.0	113.0	3255.	17464.	4.20	17.7	5.9
32.0	-1.12	1.0	114.0	3302.	0.	0.0	0.0	0.0
30.4	-2.64	1.0	115.0	3346.	0.	0.0	0.0	0.0
28.4	-3.74	1.0	116.0	3388.	0.	0.0	0.0	0.0
25.3	-4.84	1.0	117.0	3425.	0.	0.0	0.0	0.0
22.0	-4.84	1.0	118.0	3457.	0.	0.0	0.0	0.0
18.7	-4.84	1.0	119.0	3485.	0.	0.0	0.0	0.0
15.4	-4.84	1.0	120.0	3507.	0.	0.0	0.0	0.0
12.1	-4.84	1.0	121.0	3525.	0.	0.0	0.0	0.0
8.8	-4.84	1.0	122.0	3539.	0.	0.0	0.0	0.0
5.5	-4.84	1.0	123.0	3546.	0.	0.0	0.0	0.0
2.5	-3.23	1.0	124.0	3550.	0.	0.0	0.0	0.0
1.1	-1.61	1.0	125.0	3551.	0.	0.0	0.0	0.0
0.1	-0.81	1.0	126.0	3551.	0.	0.0	0.0	0.0
0.0	0.0	1.0	127.0	3551.	0.	0.0	0.0	0.0

VEHICLE TIME CUM	TRANSMISSION SPEED RATIO	TRANSMISSION DELTA PRES	TRANSMISSION TOTAL EFF	ROAD DRAG LB	ROAD TRACT LB	FUEL LB/HPH	FUEL LB/HR	FUEL MPG INST	FUEL MPG CUM
59.0	0.288	543.	90.6	90.	380.	0.758	25.10	5.73	5.47
60.0	0.317	758.	87.0	92.	276.	0.835	21.59	7.01	5.51
61.0	0.349	464.	80.6	93.	171.	0.957	16.89	9.13	5.59
62.0	0.383	331.	73.1	94.	116.	0.945	14.22	10.97	5.67
63.0	0.331	217.	59.3	93.	60.	1.023	11.20	13.88	5.73
64.0	0.373	262.	66.8	93.	82.	0.990	12.36	12.51	5.87
65.0	0.390	308.	71.5	93.	104.	0.961	13.55	11.43	5.96
66.0	0.390	297.	70.4	93.	99.	0.967	13.23	11.69	6.04
67.0	0.391	285.	69.3	93.	93.	0.974	12.99	11.94	6.12
68.0	0.390	274.	68.1	93.	88.	0.982	12.69	12.22	6.21
69.0	0.390	262.	66.9	93.	82.	0.990	12.37	12.51	6.29
70.0	0.369	396.	77.6	93.	143.	0.919	15.56	9.96	6.35
71.0	0.343	549.	83.1	94.	206.	0.865	18.85	8.34	6.39
72.0	0.370	381.	76.9	95.	139.	0.917	15.70	10.21	6.45
73.0	0.393	242.	63.1	95.	72.	0.956	12.14	13.17	6.53
74.0	0.391	219.	59.4	94.	61.	1.015	11.48	13.89	6.61
75.0	0.388	195.	54.9	94.	50.	1.036	10.76	14.71	6.70
76.0	0.375	355.	75.1	94.	127.	0.931	14.92	10.56	6.75
77.0	0.349	543.	83.1	95.	206.	0.866	19.09	9.39	6.78
78.0	0.377	359.	76.0	95.	134.	0.918	15.61	10.43	6.83
79.0	0.399	221.	59.0	95.	62.	1.008	11.70	13.88	6.91
80.0	0.375	369.	74.1	95.	134.	0.920	15.54	10.42	6.96
81.0	0.355	537.	83.0	96.	207.	0.860	19.51	8.43	6.99
82.0	0.349	650.	85.6	97.	259.	0.825	22.37	7.51	7.00
83.0	0.345	760.	87.3	99.	310.	0.793	25.45	6.82	6.99
84.0	0.363	668.	85.7	100.	273.	0.805	24.17	7.42	7.00
85.0	0.379	586.	83.5	101.	235.	0.822	22.60	8.11	7.02
86.0	0.393	517.	80.9	103.	203.	0.838	21.21	8.81	7.05
87.0	0.405	453.	77.5	103.	170.	0.857	19.61	9.64	7.09
88.0	0.413	426.	75.2	104.	154.	0.867	18.87	10.11	7.14
89.0	0.423	402.	72.0	104.	139.	0.877	18.11	10.59	7.19
90.0	0.443	350.	64.0	104.	104.	0.903	16.42	11.73	7.24
91.0	0.471	115.	58.3	104.	71.	0.956	13.77	13.96	7.31
92.0	0.446	316.	59.9	104.	97.	0.924	15.28	12.50	7.37
93.0	0.440	344.	64.5	104.	104.	0.907	16.22	11.77	7.42
94.0	0.440	344.	64.5	104.	104.	0.907	16.22	11.77	7.47
95.0	0.440	344.	64.5	104.	104.	0.907	16.22	11.77	7.52
96.0	0.467	53.	51.1	104.	54.	0.982	12.69	15.02	7.59
97.0	0.461	130.	4.6	103.	3.	1.076	9.35	20.09	7.67
98.0	0.431	319.	64.4	102.	97.	0.723	15.33	12.13	7.72
99.0	0.397	493.	79.9	103.	192.	0.844	20.70	9.07	7.74
100.0	0.403	477.	78.5	104.	182.	0.848	20.36	9.36	7.76
101.0	0.407	460.	77.1	104.	171.	0.852	19.95	9.63	7.79
102.0	0.427	407.	71.5	105.	138.	0.874	18.36	10.56	7.82
103.0	0.440	350.	64.6	105.	105.	0.902	16.51	11.76	7.86
104.0	0.475	22.	47.3	105.	43.	1.001	11.95	16.19	7.93
105.0	0.0	0.	0.0	104.	-19.	0.0	9.57	19.91	8.00
106.0	0.461	84.	54.7	103.	64.	0.965	13.38	14.03	8.06
107.0	0.411	411.	74.6	103.	148.	0.875	18.31	10.29	8.08
108.0	0.403	476.	78.6	104.	182.	0.848	20.32	9.34	8.10
109.0	0.399	555.	81.2	104.	216.	0.822	22.49	8.57	8.10
110.0	0.402	562.	80.9	106.	217.	0.818	22.85	8.57	8.11
111.0	0.406	568.	80.6	107.	218.	0.814	23.23	8.57	8.12
112.0	0.429	466.	77.2	107.	163.	0.845	20.52	9.84	8.14
113.0	0.454	371.	62.8	109.	108.	0.988	17.47	11.59	8.17
114.0	0.0	0.	0.0	107.	-54.	0.0	9.71	20.73	8.24
115.0	0.0	0.	0.0	104.	-298.	0.0	9.59	19.94	8.31
116.0	0.0	0.	0.0	100.	-469.	0.0	9.41	18.97	8.37
117.0	0.0	0.	0.0	94.	-642.	0.0	9.14	17.38	8.41
118.0	0.0	0.	0.0	89.	-647.	0.0	8.86	15.59	8.45
119.0	0.0	0.	0.0	84.	-652.	0.0	8.53	13.79	8.48
120.0	0.0	0.	0.0	80.	-656.	0.0	8.30	11.65	8.49
121.0	0.0	0.	0.0	77.	-659.	0.0	8.02	9.47	8.50
122.0	0.0	0.	0.0	74.	-662.	0.0	7.75	7.14	8.49
123.0	0.0	0.	0.0	73.	-664.	0.0	7.47	4.63	8.47
124.0	0.0	0.	0.0	71.	-420.	0.0	7.21	2.16	8.45
125.0	0.0	0.	0.0	71.	-174.	0.0	7.09	0.97	8.42
126.0	0.0	0.	0.0	71.	-82.	0.0	7.01	0.12	8.39
127.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.39

APPENDIX VI -1 COMPUTER RUN (last 135 seconds)

VEHICLE					ENGINE				TRANSMISSION			ROAD		FUEL			
SPEED MPH	ACCEL F/S/S	TIME INC	TIME CUM	DIST FT	SPEED RPM	ACCES HP	TOTAL HP	TOTAL FT-LB	SPEED RATIO	DELTA PRES	TOTAL EFF	DRAG LB	TRACT LB	LB/ HPH	LB/ HR	MPG INST	MPG CUM
19.7	0.11	1.0	1232.0	36949.	16000.	4.00	11.7	3.8	0.304	475.	74.2	96.	102.	1.007	11.74	10.57	8.50
19.6	-0.22	1.0	1233.0	36978.	16000.	4.00	9.7	2.9	0.303	394.	60.6	96.	52.	1.075	9.40	13.13	8.50
19.3	-1.58	1.0	1234.0	37006.	0.	0.0	0.0	0.0	0.0	0.	0.0	95.	-155.	0.0	8.63	14.02	8.51
17.5	-2.93	1.0	1235.0	37031.	0.	0.0	0.0	0.0	0.0	0.	0.0	93.	-362.	0.0	8.49	12.96	8.51
15.4	-3.48	1.0	1236.0	37054.	0.	0.0	0.0	0.0	0.0	0.	0.0	90.	-449.	0.0	8.30	11.65	8.51
12.8	-4.03	1.0	1237.0	37077.	0.	0.0	0.0	0.0	0.0	0.	0.0	79.	-536.	0.0	8.09	9.91	8.51
10.1	-3.48	1.0	1238.0	37088.	0.	0.0	0.0	0.0	0.0	0.	0.0	75.	-454.	0.0	7.85	8.07	8.51
8.0	-2.93	1.0	1239.0	37098.	0.	0.0	0.0	0.0	0.0	0.	0.0	74.	-372.	0.0	7.69	6.54	8.51
6.0	-2.75	1.0	1240.0	37109.	0.	0.0	0.0	0.0	0.0	0.	0.0	73.	-346.	0.0	7.51	5.04	8.51
4.3	-2.57	1.0	1241.0	37114.	0.	0.0	0.0	0.0	0.0	0.	0.0	72.	-319.	0.0	7.36	3.63	8.51
2.6	-2.20	1.0	1242.0	37118.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-263.	0.0	7.22	2.23	8.50
1.3	-1.83	1.0	1243.0	37120.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-208.	0.0	7.11	1.10	8.50
0.2	-0.92	1.0	1244.0	37120.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-69.	0.0	7.01	0.14	8.50
0.0	0.0	1.0	1245.0	37120.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.49
0.0	0.0	1.0	1246.0	37120.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.49
0.0	0.0	1.0	1247.0	37120.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.49
0.0	0.0	1.0	1248.0	37120.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.48
0.0	0.0	1.0	1249.0	37120.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.48
0.1	0.37	1.0	1250.0	37120.	16000.	4.00	6.4	2.1	0.001	862.	0.9	71.	127.	1.147	7.29	0.05	8.48
0.5	0.73	1.0	1251.0	37121.	16000.	4.00	6.6	2.2	0.008	574.	9.8	71.	182.	1.138	7.52	0.42	8.47
0.9	0.37	1.0	1252.0	37122.	16000.	4.00	6.7	2.2	0.014	421.	12.1	71.	127.	1.133	7.64	0.77	8.47
1.0	0.0	1.0	1253.0	37124.	16000.	4.00	6.6	2.2	0.015	319.	7.8	71.	71.	1.139	7.48	0.84	8.47
1.0	0.0	1.0	1254.0	37125.	16000.	4.00	6.5	2.2	0.015	319.	7.9	71.	71.	1.135	7.48	0.84	8.47
1.0	0.0	1.0	1255.0	37127.	16000.	4.00	6.6	2.2	0.015	313.	7.8	71.	71.	1.135	7.49	0.86	8.46
1.1	0.73	1.0	1256.0	37128.	16000.	4.00	7.1	2.3	0.017	505.	18.6	71.	183.	1.122	7.96	0.89	8.46
2.0	1.47	1.0	1257.0	37131.	16000.	4.00	8.6	2.8	0.031	672.	36.3	71.	234.	1.080	9.23	1.35	8.46
3.0	1.47	1.0	1258.0	37136.	16000.	4.00	9.6	3.2	0.046	643.	44.2	71.	295.	1.054	10.12	1.86	8.45
4.0	1.47	1.0	1259.0	37142.	16000.	4.00	10.7	3.5	0.062	627.	49.5	72.	295.	1.025	10.99	2.29	8.45
5.1	1.83	1.0	1260.0	37149.	16000.	4.00	12.9	4.2	0.079	723.	56.2	72.	351.	0.982	12.65	2.51	8.44
6.5	2.20	1.0	1261.0	37159.	16575.	4.09	16.4	5.2	0.097	833.	60.4	73.	408.	0.928	15.70	2.69	8.44
8.0	2.02	1.0	1262.0	37170.	17236.	4.17	18.5	5.6	0.114	807.	59.3	74.	381.	0.904	16.74	2.99	8.44
9.3	1.83	1.0	1263.0	37184.	17729.	4.24	20.3	6.0	0.129	784.	57.0	75.	354.	0.866	18.03	3.22	8.43
10.2	0.18	1.0	1264.0	37199.	16000.	4.00	11.3	3.7	0.157	92.	40.5	75.	103.	1.014	11.49	5.59	8.43
9.5	-1.47	1.0	1265.0	37213.	0.	0.0	0.0	0.0	0.0	0.	0.0	75.	-146.	0.0	7.80	7.64	8.43
8.6	-0.62	1.0	1266.0	37225.	0.	0.0	0.0	0.0	0.0	0.	0.0	74.	-20.	0.0	7.73	7.02	8.43
8.6	0.22	1.0	1267.0	37238.	16000.	4.00	11.8	3.9	0.133	325.	33.7	74.	108.	1.005	11.83	4.59	8.43
9.1	2.02	1.0	1268.0	37251.	17687.	4.23	20.5	6.2	0.127	832.	58.5	75.	391.	0.881	19.41	3.11	8.42
11.4	3.81	1.0	1269.0	37268.	20352.	4.60	36.9	9.5	0.138	1147.	65.0	76.	657.	0.772	28.51	2.51	8.41
14.0	3.92	1.0	1270.0	37289.	20653.	4.64	40.5	10.3	0.167	1203.	74.2	79.	676.	0.756	30.62	2.88	8.40
16.8	4.03	1.0	1271.0	37313.	21176.	4.71	45.2	11.2	0.195	1281.	80.7	92.	695.	0.740	33.45	3.15	8.39
19.3	2.96	1.0	1272.0	37342.	20440.	4.61	37.6	9.7	0.233	952.	85.3	95.	520.	0.766	28.84	4.20	8.39
20.6	1.69	1.0	1273.0	37372.	18954.	4.40	27.2	7.6	0.269	545.	87.2	97.	344.	0.826	22.49	5.77	8.38
21.7	1.28	1.0	1274.0	37404.	18404.	4.33	23.8	6.8	0.291	320.	88.9	89.	284.	0.852	20.27	6.73	8.38
22.4	0.88	1.0	1275.0	37437.	17944.	4.25	20.7	6.1	0.309	685.	85.7	90.	223.	0.875	18.23	7.72	8.38
23.0	0.84	1.0	1276.0	37470.	17878.	4.26	21.0	6.7	0.317	640.	84.6	91.	219.	0.876	18.37	7.86	8.38
23.5	0.81	1.0	1277.0	37505.	17901.	4.26	21.1	6.2	0.324	607.	84.1	91.	214.	0.875	18.46	8.01	8.38
24.1	0.55	1.0	1278.0	37540.	17402.	4.19	18.8	5.7	0.341	483.	81.1	92.	176.	0.857	16.90	8.94	8.38
24.3	0.29	1.0	1279.0	37576.	16550.	4.09	16.3	5.2	0.361	393.	76.8	93.	137.	0.928	15.10	10.11	8.38
24.4	-0.22	1.0	1280.0	37612.	16000.	4.00	10.9	3.5	0.376	217.	60.1	93.	59.	1.027	11.06	13.86	8.39
24.0	-0.73	1.0	1281.0	37647.	0.	0.0	0.0	0.0	0.0	0.	0.0	92.	-19.	0.0	9.03	14.69	8.39
23.6	-0.37	1.0	1282.0	37681.	16000.	4.00	8.9	2.9	0.363	175.	48.5	91.	36.	1.072	9.49	15.59	8.39
23.5	0.0	1.0	1283.0	37716.	16000.	4.00	12.6	4.2	0.362	291.	69.7	91.	91.	0.987	12.48	11.82	8.39
23.5	0.0	1.0	1284.0	37750.	16000.	4.00	12.6	4.2	0.362	291.	69.7	91.	91.	0.987	12.49	11.82	8.40
23.5	0.0	1.0	1285.0	37785.	16000.	4.00	12.6	4.2	0.362	291.	69.7	91.	91.	0.987	12.49	11.82	8.40
23.5	0.19	1.0	1286.0	37819.	16000.	4.00	14.6	4.8	0.363	347.	74.5	91.	119.	0.953	13.88	12.64	8.40
23.8	0.37	1.0	1287.0	37854.	16767.	4.11	16.7	5.2	0.349	415.	75.2	92.	147.	0.923	15.39	9.70	8.40
24.0	0.37	1.0	1288.0	37889.	16849.	4.12	16.8	5.3	0.351	414.	78.3	92.	148.	0.920	15.51	9.72	8.40
24.3	0.37	1.0	1289.0	37925.	16931.	4.13	17.0	5.3	0.353	412.	79.3	93.	148.	0.918	15.63	9.74	8.40
24.5	0.37	1.0	1290.0	37961.	16985.	4.14	17.2	5.3	0.356	410.	79.3	93.	149.	0.916	15.75	9.77	8.41
24.8	0.37	1.0	1291.0	37997.	17040.	4.14	17.4	5.4	0.358	409.	79.3	93.	149.	0.914	15.88	9.79	8.41
25.0	0.40	1.0	1292.0	38034.	17203.	4.17	18.0	5.5	0.358	420.	79.0	94.	155.	0.907	16.29	9.64	8.41
25.3	0.44	1.0	1293.0	38071.	17370.	4.19	18.6	5.6	0.357	432.	79.5	94.	161.	0.900	16.73	9.50	8.41
25.6	0.37	1.0	1294.0	38108.	17227.	4.17	18.0	5.5	0.366	406.	78.3	95.	151.	0.906	16.30	9.86	8.41
25.8	0.29	1.0	1295.0	38146.	17056.	4.15	17.3	5.3	0.373	387.	76.8	95.	140.	0.914	15.83	10.23	8.41
26.0	0.51	1.0	1296.0	38184.	17713.	4.24	20.0	5.9	0.362	454.	80.5	95.	174.	0.885	17.70	9.24	8.41
26.5	0.73	1.0	1297.0	38223.	18247.	4.31	23.0	6.6	0.353	536.	82.0	96.	208.	0.857	19.69	8.45	8.41
27.0	0.84	1.0	1298.0	38263.	18541.	4.35	24.8	7.0	0.359	571.	83.9	97.	226.	0.843	20.97	8.13	8.41
27.6	0.95	1.0	1299.0	38303.	18977.	4.40	26.7	7.4	0.361	607.	84.6	99.	244.	0.828	22.13	7.65	8.41
28.3	0.77	1.0	1300.0	38345.	18622.	4.36	25.1	7.1	0.374	547.	82.9	100.	217.	0.840	21.09	8.42	8.41

APPENDIX VI -1 COMPUTER RUN (last 135 seconds)

VEHICLE					ENGINE				TRANSMISSION			ROAD		FUEL			
SPEED MPH	ACCEL F/S/S	TIME INC	TIME CUM	DIST FT	SPEED RPM	ACCES HP	TOTAL HP	TOTAL FT-LB	SPEED RATIO	DELTA PRES	TOTAL EFF	DRAG LB	TRACT LB	LB/ HPH	LB/ HR	MPG INST	MPG CUM
28.7	0.59	1.0	1301.0	34387.	18344.	4.32	23.3	5.7	0.386	493.	80.6	100.	190.	0.954	15.69	9.06	8.41
29.0	0.0	1.0	1302.0	34429.	16965.	4.12	16.5	5.1	0.424	321.	66.3	101.	101.	0.924	15.27	11.93	8.41
28.7	-0.59	1.0	1303.0	34472.	16000.	4.00	9.0	3.0	0.442	148.	19.0	100.	11.	1.067	9.65	13.69	8.42
27.9	-2.92	1.0	1304.0	34512.	0.	0.0	0.0	0.0	0.0	0.	0.0	99.	-330.	0.0	9.36	13.72	8.42
24.8	-5.06	1.0	1305.0	34549.	0.	0.0	0.0	0.0	0.0	0.	0.0	94.	-676.	0.0	9.10	17.14	8.43
21.4	-4.55	1.0	1306.0	34580.	0.	0.0	0.0	0.0	0.0	0.	0.0	89.	-665.	0.0	8.91	15.26	8.43
18.1	-4.84	1.0	1307.0	34607.	0.	0.0	0.0	0.0	0.0	0.	0.0	84.	-653.	0.0	8.53	13.32	8.43
14.8	-4.84	1.0	1308.0	34629.	0.	0.0	0.0	0.0	0.0	0.	0.0	80.	-657.	0.0	8.25	11.26	8.43
11.5	-4.84	1.0	1309.0	34645.	0.	0.0	0.0	0.0	0.0	0.	0.0	77.	-660.	0.0	7.97	9.06	8.44
8.2	-4.84	1.0	1310.0	34657.	0.	0.0	0.0	0.0	0.0	0.	0.0	74.	-662.	0.0	7.69	6.69	8.43
4.9	-4.84	1.0	1311.0	34665.	0.	0.0	0.0	0.0	0.0	0.	0.0	72.	-664.	0.0	7.42	4.15	8.43
1.9	-3.01	1.0	1312.0	34667.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-336.	0.0	7.16	1.63	8.43
0.8	-1.17	1.0	1313.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-108.	0.0	7.07	0.71	8.43
0.1	-0.59	1.0	1314.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-18.	0.0	7.01	0.09	8.42
0.0	0.0	1.0	1315.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.42
0.0	0.0	1.0	1316.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.42
0.0	0.0	1.0	1317.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.42
0.0	0.0	1.0	1318.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.41
0.0	0.0	1.0	1319.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.41
0.0	0.0	1.0	1320.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.41
0.0	0.0	1.0	1321.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.40
0.0	0.0	1.0	1322.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.40
0.0	0.0	1.0	1323.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.40
0.0	0.0	1.0	1324.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.39
0.0	0.0	1.0	1325.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.39
0.0	0.0	1.0	1326.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.39
0.0	0.0	1.0	1327.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.39
0.0	0.0	1.0	1328.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.39
0.0	0.0	1.0	1329.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.38
0.0	0.0	1.0	1330.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.38
0.0	0.0	1.0	1331.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.37
0.0	0.0	1.0	1332.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.37
0.0	0.0	1.0	1333.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.37
0.0	0.0	1.0	1334.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.36
0.0	0.0	1.0	1335.0	34669.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	71.	0.0	7.00	0.0	8.36
0.1	0.55	1.0	1336.0	34669.	16000.	4.00	6.3	2.1	0.001	772.	1.7	71.	154.	1.148	7.27	0.08	8.36
0.8	1.10	1.0	1337.0	34670.	16000.	4.00	7.0	2.3	0.012	644.	16.7	71.	238.	1.125	7.87	0.60	8.36
1.8	2.57	1.0	1338.0	34673.	16000.	4.00	9.9	3.3	0.029	1792.	45.1	71.	523.	1.047	10.38	1.10	8.35
4.8	4.84	1.0	1339.0	34680.	17563.	4.23	20.4	6.1	0.067	1549.	67.2	72.	808.	0.866	19.13	1.66	8.35
8.0	4.29	1.0	1340.0	34691.	18067.	4.42	28.2	7.8	0.104	1432.	68.5	74.	726.	0.823	23.23	2.16	8.34
10.6	3.74	1.0	1341.0	34707.	18842.	4.53	33.9	9.0	0.132	1327.	65.6	76.	645.	0.787	26.70	2.50	8.33
13.1	3.19	1.0	1342.0	34726.	19345.	4.54	33.8	8.9	0.162	984.	70.9	78.	563.	0.787	26.58	3.10	8.32
15.0	2.64	1.0	1343.0	34748.	19385.	4.47	30.8	8.3	0.191	860.	77.0	80.	492.	0.804	24.75	3.81	8.32
16.7	2.31	1.0	1344.0	34773.	19240.	4.45	29.6	9.1	0.215	779.	81.0	82.	433.	0.811	24.00	4.38	8.31
18.1	1.98	1.0	1345.0	34799.	18991.	4.41	27.9	7.7	0.236	679.	83.4	84.	385.	0.822	22.95	4.97	8.31
19.4	1.65	1.0	1346.0	34828.	18661.	4.37	25.9	7.3	0.257	556.	85.2	85.	336.	0.836	21.66	5.66	8.31
20.4	1.32	1.0	1347.0	34858.	18231.	4.31	23.4	6.7	0.275	418.	86.4	87.	287.	0.856	19.89	6.41	8.30
21.2	0.55	1.0	1348.0	34889.	17764.	4.24	20.0	5.9	0.295	105.	88.0	88.	233.	0.866	17.74	7.52	8.30
21.7	0.59	1.0	1349.0	34921.	17029.	4.14	17.3	5.4	0.314	569.	82.0	89.	178.	0.914	15.86	8.69	8.30
22.0	0.26	1.0	1350.0	34953.	16000.	4.00	14.4	4.7	0.340	394.	76.2	89.	128.	0.956	13.76	10.06	8.31
22.0	-0.07	1.0	1351.0	34986.	16000.	4.00	11.2	3.7	0.340	291.	67.1	89.	79.	1.017	11.39	12.17	8.31
21.9	-0.37	1.0	1352.0	35018.	16000.	4.00	8.3	2.7	0.338	202.	47.4	89.	33.	1.086	9.02	15.28	8.31
21.5	-0.66	1.0	1353.0	35049.	0.	0.0	0.0	0.0	0.0	0.	0.0	88.	-12.	0.0	8.83	15.33	8.31
21.1	-0.73	1.0	1354.0	35080.	0.	0.0	0.0	0.0	0.0	0.	0.0	88.	-24.	0.0	8.79	15.07	8.32
20.5	-0.81	1.0	1355.0	35110.	0.	0.0	0.0	0.0	0.0	0.	0.0	87.	-36.	0.0	8.74	14.76	8.32
20.0	-0.95	1.0	1356.0	35140.	0.	0.0	0.0	0.0	0.0	0.	0.0	86.	-59.	0.0	8.69	14.43	8.32
19.3	-1.10	1.0	1357.0	35168.	0.	0.0	0.0	0.0	0.0	0.	0.0	85.	-82.	0.0	8.63	14.01	8.32
18.5	-1.28	1.0	1358.0	35195.	0.	0.0	0.0	0.0	0.0	0.	0.0	84.	-111.	0.0	8.56	13.64	8.33
17.5	-1.47	1.0	1359.0	35221.	0.	0.0	0.0	0.0	0.0	0.	0.0	83.	-140.	0.0	8.48	12.96	8.33
16.5	-1.65	1.0	1360.0	35245.	0.	0.0	0.0	0.0	0.0	0.	0.0	82.	-169.	0.0	8.39	12.32	8.33
15.3	-1.83	1.0	1361.0	35267.	0.	0.0	0.0	0.0	0.0	0.	0.0	80.	-199.	0.0	8.29	11.55	8.33
13.8	-3.12	1.0	1362.0	35287.	0.	0.0	0.0	0.0	0.0	0.	0.0	79.	-395.	0.0	8.17	10.60	8.33
11.0	-4.40	1.0	1363.0	35304.	0.	0.0	0.0	0.0	0.0	0.	0.0	76.	-593.	0.0	7.93	8.71	8.33
8.0	-4.22	1.0	1364.0	35315.	0.	0.0	0.0	0.0	0.0	0.	0.0	74.	-567.	0.0	7.69	6.57	8.33
5.3	-4.03	1.0	1365.0	35323.	0.	0.0	0.0	0.0	0.0	0.	0.0	72.	-541.	0.0	7.46	4.43	8.33
2.7	-2.93	1.0	1366.0	35327.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-375.	0.0	7.23	2.34	8.33
1.3	-1.83	1.0	1367.0	35329.	0.	0.0	0.0	0.0	0.0	0.	0.0	71.	-208.	0.0	7.11	1.10	8.33

APPENDIX VI -1 COMPUTER RUN (analysis)

FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION

8-17-72 RUN 1.03 EPA TURBINE CAR FEDRAL DRIVING CYCLE

AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

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PERFORMANCE SUMMARY (MILES PER GALLON , MILES , SECONDS)
   MPG FOR ALL +HPO (ENGINE DRIVING)   CONDITIONS   8.355 (  5.976MI,   852.SEC)
   MPG FOR ALL -HPO (VEHICLE COASTING) CONDITIONS   8.213 (  1.473MI,   515.SEC)

   MPG OVERALL FOR ALL CONDITIONS                   8.326 (  7.449MI,  1367.SEC)

ENGINE DRIVING MODE

   TOTAL WHEELS-TO-ROAD ENERGY,HP.SEC              11581.05
   TOTAL ENGINE ENERGY (INCL.ACCESS.),HP.SEC       19214.43

   AVG. WHEELS TO ROAD HP                             13.59
   AVG. ENGINE HP (INCL.ACCESS)                       22.55

   TRANSMISSION ENERGY EFFICIENCY                     78.205

ENGINE COASTING MODE

   TOTAL ROAD-TO-WHEELS ENERGY,HP.SEC              -3774.99

   AVG. ROAD-TO-WHEELS HP                             -7.33
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FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION

8-17-72 RUN 1.03 EPA TURBINE CAR SIMPLIFIED SUBURBAN ROUTE

AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

INPUT PARAMETERS

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VEHICLE WEIGHT (LB)................................WT=    4600.000
WHEEL INERTIA (FT-LB-SEC2).........................WI=      11.200
FRONTAL AREA (FT2).................................AF=      20.000
COEFFICIENT OF DRAG................................CD=       0.600
FUEL DENSITY (LB/GAL)..............................DEN=      6.290
ENGINE CODE........................................ENG=      3.000
DUTY CYCLE CODE....................................DUTY=     1.000
OUTPUT CODE........................................OUT=      2.000

NOMOGRAPH DIMENSION................................A=   0.4100
NOMOGRAPH DIMENSION................................B=   0.6300
NOMOGRAPH DIMENSION................................C=   1.0000
NEMAX/NOMAX........................................SPATF=  4.3021
LB/HR AT  0 MPH....................................FPH1 =  7.0000
LB/HR AT 85 MPH....................................FPH2 = 14.2000

ENGINE SPEED / ENGINE LINK SPEED...................R1=   9.4184 (0.MESH)
V-UNIT SPEED / V-UNIT LINK SPEED...................R2=   2.2157 (2.MESH)
MODE 3 F-UNIT SPEED / MODE 3 F-UNIT LINK SPEED.....R3=   2.2000 (3.MESH)
MODE 2 F-UNIT SPEED / MODE 2 F-UNIT LINK SPEED.....R4=   2.2000 (3.MESH)
MODE 1 F-UNIT SPEED / MODE 1 F-UNIT LINK SPEED.....R5=   7.7667 (2.MESH)
TRANS OUTPUT LINK SPEED / TRANS OUTPUT SPEED.......R6=   1.0000 (0.MESH)
TRANS OUTPUT SPEED / AXLE SPEED....................RA=   4.5000 (0.950EFF)

HYD UNIT DISPLACEMENT..............................DISP=     1.500
WHEEL DIAMETER.....................................DW=      26.400
GEAR MESH EFFICIENCY...............................EG=       0.985
MODE 2 PLANETARY CARRIER LINK......................ZCAR2=    4.000
MODE 3 PLANETARY CARRIER LINK......................ZCAR3=    1.000
MAXIMUM CLUTCH LOSS HORSEPOWER.....................FLCLX=    1.500
MINIMUM CHARGE PUMP PRESSURE.......................FCN=     60.000
MAXIMUM CHARGE PUMP PRESSURE.......................FCX=    250.000
```

FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION

8-17-72 RUN 1.03 EPA TURBINE CAR SIMPLIFIED SUBURBAN ROUTE

AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

OUTPUT PARAMETERS

VEHICLE					ENGINE				TRANSMISSION			ROAD		FUEL			
SPEED MPH	ACCEL F/S/S	TIME INC	TIME CUM	DIST FT	SPEED RPM	ACCES HP	TOTAL HP	TOTAL FT-LB	SPEED RATIO	DELTA PRES	TOTAL EFF	DRAG LB	TRACT LB	LB/ HPH	LB/ HR	MPG INST	MPG CUM
20.0	0.0	299.4	299.4	8784.	16000.	4.00	10.8	3.4	0.308	406.	70.7	86.	86.	1.025	11.11	11.30	11.30
30.0	0.0	299.4	598.9	21959.	17026.	4.14	17.5	5.4	0.434	336.	65.1	103.	103.	0.912	15.83	11.93	11.61
40.0	0.0	299.4	898.3	39526.	17976.	4.27	22.3	6.5	0.548	339.	78.5	126.	126.	0.855	19.19	13.09	12.23

APPENDIX VI-1 COMPUTER RUN (analysis)

FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION

8-17-72 RUN 1.03 EPA TURBINE CAR SIMPLIFIED SUBURBAN ROUTE

AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

PERFORMANCE SUMMARY (MILES PER GALLON , MILES , SECONDS)

MPG FOR ALL +HPO (ENGINE DRIVING) CONDITIONS	12.228	(7.486MI,	898.SEC)
MPG FOR ALL -HPO (VEHICLE COASTING) CONDITIONS	0.0	(0.0 MI,	0.SEC)
MPG OVERALL FOR ALL CONDITIONS	12.228	(7.486MI,	898.SEC)

ENGINE DRIVING MODE

TOTAL WHEELS-TO-ROAD ENERGY,HP.SEC	7875.40
TOTAL ENGINE ENERGY (INCL.ACCESS.),HP.SEC	15163.45
AVG. WHEELS TO ROAD HP	8.77
AVG. ENGINE HP (INCL.ACCESS)	16.88
TRANSMISSION ENERGY EFFICIENCY	72.424

ENGINE COASTING MODE

TOTAL ROAD-TO-WHEELS ENERGY,HP.SEC	0.0
AVG. ROAD-TO-WHEELS HP	0.0

FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION

8-17-72 RUN 1.03 EPA TURBINE CAR SIMPLIFIED COUNTRY ROUTE

AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

INPUT PARAMETERS

VEHICLE WEIGHT (LB)	WT=	4600.000
WHEEL INERTIA (FT-LB-SEC2)	WI=	11.200
FRONTAL ARVE (FT2)	AF=	22.000
COEFFICIENT OF DRAG	CD=	0.600
FUEL DENSITY (LB/GAL)	DEN=	6.290
ENGINE CODE	ENG=	3.000
DUTY CYCLE CODE	DUTY=	1.000
OUTPUT CODE	OUT=	2.000
NOMOGRAPH DIMENSION	A=	0.4100
NOMOGRAPH DIMENSION	B=	0.6300
NOMOGRAPH DIMENSION	C=	1.0000
NEMAX/NOMAX	SRATE=	4.3021
LB/HR AT 0 MPH	FPH1	7.0000
LB/HR AT 85 MPH	FPH2 =	14.2000
ENGINE SPEED / ENGINE LINK SPEED	R1=	9.4184 (0.MESH)
V-UNIT SPEED / V-UNIT LINK SPEED	R2=	2.2157 (2.MESH)
MODE 3 F-UNIT SPEED / MODE 3 F-UNIT LINK SPEED	R3=	2.2000 (3.MESH)
MODE 2 F-UNIT SPEED / MODE 2 F-UNIT LINK SPEED	R4=	2.2000 (3.MESH)
MODE 1 F-UNIT SPEED / MODE 1 F-UNIT LINK SPEED	R5=	7.3647 (2.MESH)
TRANS OUTPUT LINK SPEED / TRANS OUTPUT SPEED	R6=	1.0000 (0.MESH)
TRANS OUTPUT SPEED / AXLE SPEED	RA=	4.5000 (0.950EFF)
HYD UNIT DISPLACEMENT	DISP=	1.500
WHEEL DIAMETER	DW=	26.400
GEAR MESH EFFICIENCY	EG=	0.985
MODE 2 PLANETARY CARRIER LINK	ZCAR2=	4.000
MODE 3 PLANETARY CARRIER LINK	ZCAR3=	1.000
MAXIMUM CLUTCH LOSS HORSEPOWER	FLCLX=	1.500
MINIMUM CHARGE PUMP PRESSURE	FCN=	60.000
MAXIMUM CHARGE PUMP PRESSURE	FCX=	250.000

FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION

8-17-72 RUN 1.03 EPA TURBINE CAR SIMPLIFIED COUNTRY ROUTE

AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

OUTPUT PARAMETERS

VEHICLE					ENGINE				TRANSMISSION			ROAD		FUEL			
SPEED MPH	ACCEL F/S/S	TIME INC	TIME CUM	DIST FT	SPEED RPM	ACCES HP	TOTAL HP	TOTAL FT-LB	SPEED RATIO	DELTA PRES	TOTAL EFF	DRAG LB	TRACT LB	LB/ MPH	LB/ HR	MPG INST	MPG CUM
50.0	0.0	156.2	156.2	11457.	19252.	4.45	29.6	8.1	0.639	440.	86.9	156.	156.	0.802	23.77	13.21	13.21
60.0	0.0	156.2	312.5	25205.	20552.	4.63	39.7	10.2	0.720	399.	91.7	191.	191.	0.744	25.57	12.74	12.95
70.0	0.0	156.2	468.7	41245.	21762.	4.79	55.8	13.5	0.793	951.	89.8	233.	233.	0.651	39.54	11.41	12.36

```
FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION
-------------------------------------------------

8-17-72 RUN 1.03 EPA TURBINE CAR  SIMPLIFIED COUNTRY ROUTE

AEROJET RANKINE ENGINE TMT W/1.5 IN. LOG

PERFORMANCE SUMMARY (MILES PER GALLON , MILES , SECONDS
   MPG FOR ALL +HPO (ENGINE DRIVING)    CONDITIONS  12.304 (  7.811MI,   469.SEC)
   MPG FOR ALL -HPO (VEHICLE COASTING) CONDITIONS   0.0   (  0.0  MI,     0.SEC)

   MPG OVERALL  FOR ALL CONDITIONS                 12.304 (  7.811MI,   469.SEC)

ENGINE DRIVING MODE

   TOTAL  WHEELS-TO-ROAD ENERGY,HP.SEC            14815.53
   TOTAL  ENGINE ENERGY (INCL.ACCESS.),HP.SEC     19546.58

   AVG. WHEELS TO ROAD HP                            31.61
   AVG. ENGINE HP (INCL.ACCESS)                      41.70

   TRANSMISSION ENERGY EFFICIENCY                   89.735

ENGINE COASTING MODE

   TOTAL  ROAD-TO-WHEELS ENERGY,HP.SEC                0.0

   AVG. ROAD-TO-WHEELS HP                             0.0
```

```
FUEL ECONOMY / TRANSMISSION PERFORMANCE ANALYSIS
FOR A VEHICLE WITH A HYDROMECHANICAL TRANSMISSION
-------------------------------------------------

COMBINED FEDRAL DRIVING CYCLE,SIMPLIFIED SUBURBAN CYCLE,AND SIMPLIFIED COUNTRY CYCLE

PERFORMANCE SUMMARY (MILES PER GALLON , MILES , SECONDS
   MPG FOR ALL +HPO (ENGINE DRIVING)    CONDITIONS  10.941 ( 21.274MI,  2219.SEC)
   MPG FOR ALL -HPO (VEHICLE COASTING) CONDITIONS   8.213 (  1.473MI,   515.SEC)

   MPG OVERALL  FOR ALL CONDITIONS                 10.621 ( 22.746MI,  2734.SEC)

ENGINE DRIVING MODE

   TOTAL  WHEELS-TO-ROAD ENERGY,HP.SEC            34271.98
   TOTAL  ENGINE ENERGY (INCL.ACCESS.),HP.SEC     53924.47

   AVG. WHEELS TO ROAD HP                            15.44
   AVG. ENGINE HP (INCL.ACCESS)                      24.30

   TRANSMISSION ENERGY EFFICIENCY                   81.227

ENGINE COASTING MODE

   TOTAL  ROAD-TO-WHEELS ENERGY,HP.SEC            -3774.99

   AVG. ROAD-TO-WHEELS HP                            -7.33
```

APPENDIX VI -2

ZB32 - VEHICLE PERFORMANCE PROGRAM, TORQUE CONVERTER AND TRACTION DRIVE TRANSMISSIONS

Program Language: Fortran IV

Purpose: To determine vehicle performance (including fuel consumption) for any given vehicle with a traction drive-torque converter, or automatic shifting gearbox-torque converter type transmission and with any given engine for (I) any given vehicle output specified driving cyle, or (II) a standing start acceleration run under the application of the given engine output.

Required Input

Environment parameters: Air temperature, road grade

Vehicle parameters: Weight, drive wheel radius, and total wheel inertia, frontal area and aerodynamic drag factor. (Rolling resistance factors built into program)

Engine parameters: Engine HP versus specific fuel consumption map, desired engine speed versus engine HP operating curve, maximum engine speed, torque and power, "closed throttle" fuel consumption at idle and some other engine speed, fuel density, engine speed versus vehicle accessory HP curve, and engine inertia.

Transmission parameters: Transmission and drive line gear ratios and efficiencies, traction drive unit ratio range, reference torque converter characteristics, and required torque converter diameter. (Traction drive efficiency computed in a separate sub-routine)

Driving cycle: Federal driving cycle, simplified suburban and country routes specified in separate sub-routines. Output conditions can also be specified point by point on punched cards in terms of vehicle speed, acceleration and and time increment, or in terms of required tractive effort, speed, and time increment.

Driving cycle: (continued)

If the output performance conditions are not specified, then they will be computed for the case of a vehicle starting from rest under the application of (I) the given engine speed-torque curve in the case of a fixed ratio-torque converter transmission or (II) the given engine at its maximum power point for a traction drive-converter transmission.

Computed Output

The program initially prints out some transmission limiting parameters. Such as conditions at the maximum power stall point, maximum creep speed, and some maximum system speeds.

The program then takes the vehicle through each point in the requested driving cycle, printing out various speed, torques, ratios, powers etc. in the engine-drive line system, as well as speed, time, distance and instantaneous and cumulative fuel consumption. At completion, a summary for the course, and the sections within the course are printed out.

Operation

Following is a brief general description of how the program operates.

From the given vehicle speed and acceleration (or tractive effort) for each given driving cycle point, the program calculates the required driveline speeds and torques from the road wheels to the torque converter output shaft. Knowing the torque converter size and characteristics, the torque converter input speed and torque can be calculated,and thus, the output conditons at the traction drive unit.

The engine speed is determined by the power required to satisfy the particular duty cycle point and the given engine power versus engine speed operating curve. Thus, the speed to the traction drive input can be determined. The required instantaneous traction drive ratio is then determined from the required input and output speeds.

Operation: (continued)

This ratio must be within the ratio range specified as input data. If this range is exceeded, the unit will go to its maximum ratio and the engine will be run at some speed off its required operating line. (Part of the system optimization is to choose a ratio range that coincides very closely with that required by all the vehicle performance limits.)

The traction drive efficiency is calculated as a function of the traction drive input torque, which means the input speed must be determined by an iterative process of first assuming an efficiency, calculating the required engine power, and traction drive input conditions and then actual efficiency. This is repeated until the actual and assumed efficiencies are sufficiently close to each other.

Instantaneous fuel consumption is computed for each point from the calculated engine power and speed and the given specific fuel consumption map.

Time, distance, fuel consumption and energy are accumulated through the duty cycle for the completed driving cycle performance summary printout.

If a duty cycle is not given, the problem is worked in reverse, that is,instead of calculating the required engine and transmission conditions to give the specified vehicle speed and acceleration, the speed and acceleration will be computed with the specified engine output.

Sample Output

Following is a sample printout of the program, giving the first 69 seconds, and the last 33 seconds of the Federal Driving Cycle, the Simplified Suburban and Country Routes, and the resultant combined course summary for the traction drive-torque converter transmission and Aerojet Rankine cycle engine.

APPENDIX VI -2 FIRST 69 SECONDS COMPUTER RUN

```
1130-B32,TORQUE CONVERTER-FLUID COUPLING SIZING AND PERFORMANCE ANALYSIS
VEHICLE PERFORMANCE VERSION-------------------------------------------REVISION F

VEHICLE.......EPA TURBINE CAR  COMBINED DUTY CYCLE
ENGINE........AEROJET RANKINE ENGINE
TRANSMISSION..TRACTION DRIVE/TORQUE CONVERTER
 CONVERTER.....FORD 3.00 S.T.R.
---------------------------------------------------------------------------------
                                   INPUT DATA
                                   ----------

VEHICLE TAKE, OVER EPA COMPOUND DRIVING CYCLE

DIAMETER OF CONVERTER/COUPLING TO BE SCALED .............   6.12 INS.

VEHICLE WEIGHT.........................................  4600.

DRIVE WHEEL RADIUS,FT..................................  1.100

TOTAL WHEEL INERTIA,SLUG-FT-FT.........................  11.20

FRONTAL AREA,SQ.FT.....................................  20.00

AIR TEMP.,DEG.F........................................   85.0

AERODYNAMIC DRAG FACTOR................................  0.600

ROAD GRADE,PERCENT.....................................   0.0

M.I.T. ROLLING RESISTANCE FORMULAE USED
```


```
1130-B32,TORQUE CONVERTER-FLUID COUPLING SIZING AND PERFORMANCE ANALYSIS
VEHICLE PERFORMANCE VERSION-------------------------------------------REVISION F

VEHICLE.......EPA TURBINE CAR  COMBINED DUTY CYCLE
ENGINE........AEROJET RANKINE ENGINE
TRANSMISSION..TRACTION DRIVE/TORQUE CONVERTER
 CONVERTER.....FORD 3.00 S.T.R.
---------------------------------------------------------------------------------
```


```
ENGINE ACCESSORY POWER
----------------------
       ENGINE RPM
       16000.18064.20019.22000.

       ACCESSORY HP
         4.00  4.28  4.55  4.82

FUEL MAP FOR ENGINE ID.................................3
ENGINE IDLE SPEED,RPM..............................16000.0
IDLE FUEL RATE,LB/HR...................................8.000
ENGINE SPEED N2,RPM................................22000.0
FUEL RATE AT N2,LB/HR.................................12.000

FUEL DENSITY,LB/GAL...................................  6.28

TRACTION DRIVE RATIO RANGE............................  5.000
MAX.ENGINE TORQUE,FT.LF...............................  28.6
MAX.ENGINE HP.........................................  120.0
MAX.ENGINE RPM.......................................22000.0

GEAR RATIOS,RG1,RG2,RG3,RG4..........   2.750   1.000   1.000 12.780
EFFICIENCIES,EG1,EG2,EG3,EG4.........   1.000   1.000   0.985  0.950

DESIRED ENGINE HP-SPEED OPERATION
---------------------------------
    ENGINE HP
      14.     14.     17.     20.     25.     30.     40.     50.     55.     60.     65.     70.     71.

    ENGINE RP
     16000. 16000. 16930. 17670. 18570. 19400. 20740. 21530. 21790. 21930. 21990. 22000. 22000.
 12.00 IN. REFERENCE CONVERTER/COUPLING WAS GIVEN AS INPUT DATA
-----------------------------------------------------------------
       SPEED RATIO
        0.0   0.100 0.200 0.300 0.400 0.450 0.500 0.550 0.600 0.650 0.700 0.750
        0.800 0.850 0.860 0.900 0.920 0.940 0.950 0.960 0.970 0.980 0.990 0.992
       TORQUE RATIO
        3.000 2.600 2.300 2.010 1.800 1.690 1.560 1.500 1.370 1.300 1.200 1.120
        1.030 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000
       CAPACITY FACTOR,K
         196.  183.  180.  172.  167.  165.  164.  165.  167.  169.  172.  176.
         183.  196.  200.  214.  225.  250.  278.  320.  400.  800. 2000. 4000.
  6.12 INCH CONVERTER/COUPLING SCALED FROM   12.00 INCH REFERENCE UNIT
  ------------------------------------------------------------------
       SPEED RATIO
        0.0   0.100 0.200 0.300 0.400 0.450 0.500 0.550 0.600 0.650 0.700 0.750
        0.800 0.850 0.860 0.900 0.920 0.940 0.950 0.960 0.970 0.980 0.990 0.992
       TORQUE RATIO
        3.000 2.600 2.300 2.010 1.800 1.690 1.560 1.500 1.370 1.300 1.200 1.120
        1.030 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000
       CAPACITY FACTOR,K
        1055. 1009.  966.  923.  897.  886.  883.  886.  897.  904.  923.  948.
         985. 1055. 1077. 1152. 1211. 1346. 1497. 1723. 2153. 4307.10767.21534.
```


Sundstrand Aviation
division of Sundstrand Corporation

APPENDIX VI -2 FIRST 69 SECONDS OF COMPUTER RUN (continued)

1130-832, TORQUE CONVERTER - FLUID COUPLING SIZING AND PERFORMANCE ANALYSIS REVISION F

VEHICLE.......EPA TURBINE CAR COMBINED DUTY CYCLE
ENGINE........AEROJET RANKINE ENGINE
TRANSMISSION..TRACTION DRIVE/TORQUE CONVERTER
CONVERTER.....FORD 3.00 S.T.R.

CALCULATED CREEP SPEED,MPH................. 13.0

MAX.POWER STALL CONDITIONS
ENGINE SPEED.............................22000.0
ENGINE HP TO TRANS.I/P................... [illegible].2
CONVERTER I/P TORQ,FT.LB................. [illegible]
CONVERTER I/P SPEED...................... 363[illegible].8

TRACT.DRIVE I/P TORQ,FT.LB............... 75.6
TRACT.DRIVE I/P SPEED.................... 50[illegible]0.0
TRACT.DRIVE SPEED RATIO,IN/OUT........... 0.[illegible]27
TRACT.DRIVE EFFICIENCY................... [illegible] 57

TRACTIVE EFFORT AT ROAD,LB............... 2162.0

HP ABSORBED BY CONVERTER AT IDLE........... 3.01
MAX,CONVERTER I/P SPEED....................17888.5
MAX.TRACT.DRIVE I/P SPEED.................. 9000.0
MAX.THEORETICAL VEHICLE SPEED,MPH.......... 110.0

VEHICLE SPEED MPH ACCEL F/S/S	TIME INC TIME CUM	DIST FT	ENGINE RPM	ENGINE TOTAL HP ACCES HP	ENGINE TOTAL FTLB ACCES FTLB	TRACT.DRIVE RPM IN RPM OUT	TRACT.DRIVE SPEED RATIO RATIO RANGE	TRACT.DRIVE FTLB IN EFF	CONVERTER RPM IN RPM OUT	CONVERTER SPEED RATIO TORQ RATIO	CONVERTER FTLB IN EFF	GBOX RPM OUT TORQ OUT	TRANS EFF BRAKE LB	ROAD DRAG LB TRACT LB	FUEL LB/HPH LB/HR	FUEL MPG INST MPG CUM
0.0 0.0	1.00 [illegible]	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 2.00	0.	15070.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 3.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 4.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 5.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 6.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 7.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 8.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	
0.0 0.0	1.00 9.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.0[illegible]	
0.0 0.0	1.00 10.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	
0.0 0.0	1.00 11.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	
0.0 0.0	1.00 12.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 13.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 14.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 15.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 16.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 17.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 18.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.0 0.0	1.00 19.00	0.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 0.0
0.4 2.16	1.00 20.00	1.	16531.	16 4.2	5 1.3	6011 3655	1.64 1.36	10.4 0.70	3655 59	0.02 2.92	12.0 0.05	59 34	0.03 0	70 379	0.931 15.006	0.15 0.01
2.3 4.33	1.00 21.00	5.	19748.	32 4.6	8 1.2	7177 4982	1.44 1.55	20.2 0.83	4982 479	0.10 2.61	24.3 0.25	479 62	0.21 0	71 689	0.799 25.757	0.72 0.11
5.2 4.22	1.00 22.00	13.	20109.	34 4.6	9 1.2	7312 5012	1.44 1.53	21.7 0.84	5012 956	0.19 2.33	26.7 0.44	956 61	0.37 0	72 675	0.784 27.317	1.35 0.2[illegible]
8.7 4.11	1.00 23.00	26.	20497.	37 4.7	9 1.2	7453 5054	1.47 1.52	23.4 0.85	5054 1415	0.28 2.05	29.5 0.58	1415 59	0.49 0	74 661	0.769 29.118	1

VEHICLE			ENGINE			TRACT. DRIVE			CONVERTER			GBOX		ROAD	FUEL	
SPEED MPH	TIME INC	DIST FT	RPM	TOTAL HP	TOTAL FTLB	RPM IN	SPEED RATIO	FTLB IN	RPM IN	SPEED RATIO	FTLB IN	RPM OUT	TRANS EFF	DRAG LB	LB/HPH	MPG INST
ACCEL F/S/S	TIME CU			ACCES HP	ACCES FTLB	RPM OUT	RATIO RANGE	EFF	RPM OUT	TORQ RATIO	EFF	TORQ OUT	BRAKE LB	TRACT LB	LB/HR	MPG CUM
11.5	1.00	43.	20826.	40	10	7573	1.45	25.1	5127	0.36	32.1	[illegible]	0.58	76	0.756	2.34
4.03	24.00			4.7	1.2	5127	1.51	0.67	1856	1.37	0.66	59	0	652	30.889	0.66
14.2	1.00	64.	21116.	44	10	7575	1.47	26.9	5227	0.44	34.7	2309	0.65	79	0.744	2.72
3.96	25.00			4.7	1.2	5227	1.52	0.88	2309	1.71	0.75	58	0	644	37.795	0.87
16.5	1.00	88.	19975.	33	8	7263	1.55	21.2	4661	0.59	27.6	2706	0.68	81	0.786	3.93
2.42	26.00			4.6	1.2	4661	1.44	0.84	2706	1.43	0.87	38	0	427	26.611	1.11
17.5	1.00	114.	17827.	20	6	6482	1.69	13.3	3859	0.74	16.8	2846	0.62	82	0.880	6.02
0.88	27.00			4.3	1.3	3859	1.33	0.75	2846	1.14	0.84	18	0	208	18.264	1.36
18.2	1.00	141.	19346.	29	8	7034	1.57	16.8	4469	0.66	24.2	2968	0.66	83	0.810	4.77
1.76	28.00			4.5	1.2	4469	1.42	0.82	2968	1.27	0.85	30	0	335	24.024	1.58
19.7	1.00	170.	20834.	45	10	7576	1.48	25.1	5116	0.63	32.2	3236	0.71	85	0.753	4.06
2.64	29.00			4.7	1.2	5116	1.51	0.87	3236	1.32	0.84	41	0	463	30.811	1.76
21.5	1.00	202.	19772.	32	8	7139	1.5	20.4	4760	0.74	25.7	3501	0.69	88	0.792	5.26
1.61	30.00			4.6	1.2	4760	1.48	0.83	3501	1.14	0.84	28	0	318	25.708	1.97
22.1	1.00	234.	17915.	21	5	6514	1.55	13.6	4215	0.85	15.9	3594	0.64	89	0.874	7.49
0.59	31.00			4.3	1.3	4215	1.45	0.76	3594	1.00	0.85	15	0	172	18.542	2.19
22.4	1.00	267.	16000.	12	4	5818	1.51	7.6	3847	0.95	7.3	3641	0.59	89	0.990	11.36
-0.07	32.00			4.0	1.3	3847	1.41	0.63	3641	1.00	0.95	7	0	79	12.378	2.43
22.7	1.00	299.	16000.	3	1	5818	2.17	0.0	2683	1.33	0.0	3578	0.59	88	1.249	17.27
-0.73	33.00			4.4	1.5	2683	1.03	0.80	3578	1.00	0.75	0	0	-15	8.000	2.68
21.5	1.00	331.	16000.	3	1	5818	2.22	0.0	2622	1.33	0.0	3496	0.59	88	1.256	16.87
-0.77	34.00			4.5	1.5	2622	1.01	0.80	3496	1.00	0.75	0	0	-21	8.000	2.92
20.9	1.00	361.	16000.	0	0	5818	2.24	0.0	2601	1.31	0.0	3407	0.59	87	0.0	15.45
-0.81	35.00			4.0	1.3	2601	1.00	0.80	3407	1.00	0.75	0	0	-27	8.000	3.16
20.3	1.00	391.	16000.	0	0	5818	2.24	0.0	2601	1.27	0.0	3294	0.59	86	0.0	15.90
-1.65	36.00			4.0	1.3	2601	1.00	0.80	3294	1.00	0.75	0	0	-149	8.000	3.34
18.7	1.00	418.	16000.	0	0	5818	2.24	0.0	2601	1.17	0.0	3041	0.59	84	0.0	14.58
-2.49	37.00			4.0	1.3	2601	1.00	0.80	3041	1.00	0.75	0	0	-271	8.000	3.52
17.1	1.00	443.	16000.	0	0	5818	2.24	0.0	2601	1.07	0.0	2778	0.59	82	0.0	13.41
-2.02	38.00			4.0	1.3	2601	1.00	0.80	2778	1.00	0.75	0	0	-205	8.000	3.67
15.9	1.00	467.	16000.	0	0	5818	2.24	0.0	2601	1.00	0.0	2594	0.59	80	0.0	12.52
-1.54	39.00			4.0	1.3	2601	1.00	0.80	2594	1.00	0.75	0	0	-139	8.000	3.81
15.1	1.00	489.	16000.	5	2	5818	-2.24	2.0	2601	0.94	[illegible]	2451	0.74	79	1.151	[illegible]
-0.55	40.00			4.0	1.3	2601	1.00	0.05	2451	1.00	0.94	3	0	1	8.000	3.93
15.2	1.00	511.	16133.	15	4	5966	1.79	9.7	3272	0.76	11.9	2472	0.56	80	0.946	6.73
0.44	41.00			4.2	1.4	3272	1.25	0.58	2472	1.11	0.84	12	0	142	14.191	4.00
15.5	1.00	534.	17314.	18	5	6295	1.75	11.8	3594	0.70	15.	2531	0.60	80	0.903	5.86
0.81	42.00			4.3	1.3	3594	1.29	0.73	2531	1.19	0.8[illegible]	17	0	195	16.671	4.05
16.3	1.00	558.	16197.	22	6	6564	1.68	14.3	3914	0.68	18.4	2651	0.64	81	0.867	5.31
1.17	43.00			4.4	1.3	3914	1.33	0.77	2651	1.25	0.84	22	0	248	19.270	4.10
17.2	1.00	583.	19619.	31	8	7134	1.57	19.8	4557	0.62	25.7	2805	0.68	82	0.799	4.31
2.05	44.00			4.5	1.2	4557	1.43	0.83	2805	1.35	0.83	34	0	375	25.118	4.10
19.1	1.00	611.	21015.	42	10	7541	1.47	26.2	5203	0.60	33.7	3106	0.71	84	0.746	3.75
2.93	45.00			4.7	1.2	5203	1.52	0.87	3106	1.34	0.82	45	0	503	32.004	4.09
21.7	1.00	642.	20419.	37	9	7425	1.50	23.1	4965	0.69	29.3	3409	0.71	87	0.768	4.61
2.13	46.00			4.6	1.2	4965	1.50	0.85	3409	1.23	0.84	35	0	391	28.586	4.11

APPENDIX VI -2 FIRST 69 SECONDS OF COMPUTER RUN (continued)

VEHICLE SPEED MPH / ACCEL F/S/S	TIME IN / TIME CUM	DIST FT	ENGINE RPM	TOTAL HP / ACCES HP	TOTAL FTLB / ACCES FTLB	TRACT. DRIVE RPM IN / RPM OUT	SPEED RATIO / RATIO RANGE	FTLB IN / EFF	CONVERTER RPM IN / RPM OUT	SPEED RATIO / TORQ RATIO	FTLB IN / EFF	GBOX RPM OUT / TORQ OUT	TRANS EFF / BRAKE LB	ROAD DRAG LB / TRACT LB	FUEL LB/HPH / LB/HR	MPG INST / MPG CUM
22.0	1.00	674.	19340.	29	8	7050	1.51	18.9	4654	0.77	23.6	3578	0.67	88	0.807	5.72
1.32	47.00			4.5	1.2	4654	1.49	0.32	3578	1.09	0.83	25	0	277	24.167	4.17
22.5	1.00	708.	17855.	20	6	6492	1.54	13.4	4227	0.88	15.5	3703	0.65	90	0.877	7.80
0.55	48.00			4.3	1.3	4227	1.46	0.75	3703	1.00	0.86	15	0	168	18.330	4.26
22.7	1.00	741.	16000.	10	3	5818	1.51	6.2	3863	0.96	5.4	3700	0.55	90	1.026	12.88
-0.22	49.00			4.0	1.3	3863	1.48	0.58	3700	1.00	0.96	5	0	58	11.092	4.39
22.4	1.00	774.	16709.	0	0	6108	2.24	0.0	2731	1.33	0.0	3642	0.59	89	0.0	16.48
-1.43	50.00			4.1	1.3	2731	1.00	0.80	3642	1.00	0.75	0	0	-114	8.532	4.53
20.4	1.00	804.	16000.	0	0	5818	2.24	0.0	2601	1.30	0.0	3383	0.59	87	0.0	16.33
-2.64	51.00			4.0	1.3	2601	1.00	0.80	3383	1.00	0.75	0	0	-289	8.000	4.66
19.7	1.00	837.	16000.	0	0	5819	2.24	0.0	2601	1.19	0.0	3094	0.59	84	0.0	14.93
-2.47	52.00			4.0	1.3	2601	1.00	0.80	3094	1.00	0.75	0	0	-271	8.000	4.77
17.4	1.00	856.	16000.	0	0	5818	2.24	0.0	2601	1.09	0.0	2830	0.59	82	0.0	13.66
-2.35	53.00			4.0	1.3	2601	1.00	0.80	2830	1.00	0.75	0	0	-252	8.000	4.86
16.1	1.00	881.	16000.	5	2	5818	2.16	1.9	2638	0.98	1.[illegible]	2621	0.28	81	1.155	12.65
-0.48	54.00			4.0	1.3	2638	1.03	0.29	2621	1.00	0.98	1	0	13	8.000	4.95
16.7	1.00	906.	18545.	24	6	6729	1.64	15.8	4095	0.67	20.3	2724	0.65	81	0.847	5.05
1.39	55.00			4.4	1.3	4096	1.36	0.79	2724	1.27	0.85	25	0	280	20.832	4.95
17.8	1.00	932.	19627.	32	9	7209	1.55	20.6	4649	0.62	26.7	2899	0.68	83	0.791	4.31
2.13	56.00			4.6	1.2	4649	1.44	0.84	2899	1.34	0.83	35	0	387	25.971	4.93
19.5	1.00	961.	21023.	42	10	7444	1.47	26.3	5216	0.61	33.7	3196	0.71	85	0.746	3.85
2.83	57.00			4.7	1.2	5216	1.53	0.87	3196	1.35	0.83	44	0	494	32.051	4.69
21.5	1.00	993.	20814.	40	10	7565	1.47	25.0	5150	0.58	31.0	3500	0.72	88	0.753	4.41
2.33	58.00			4.7	1.2	5150	1.52	0.86	3500	1.24	0.84	38	0	428	30.667	4.87
22.9	1.00	1026.	20472.	37	9	7444	1.47	23.3	5071	0.73	29.2	3724	0.71	90	0.765	4.99
1.91	59.00			4.7	1.2	5071	1.52	0.85	3724	1.14	0.84	32	0	362	28.823	4.87
24.1	1.00	1061.	17579.	31	8	7119	1.47	19.6	4857	0.81	23.3	3916	0.67	92	0.799	6.07
1.21	60.00			4.5	1.2	4857	1.53	0.83	3916	1.32	0.83	24	0	265	24.900	4.91
24.5	1.00	1097.	17956.	21	6	6529	1.47	13.7	4449	0.90	15.2	3993	0.67	93	0.872	8.26
0.51	61.00			4.3	1.3	4448	1.52	0.75	3993	1.00	0.90	15	0	166	18.669	4.97
24.8	1.00	1134.	16576.	16	5	6071	1.41	10.6	4305	0.94	10.5	4039	0.65	93	0.925	10.23
0.15	62.00			4.2	1.3	4305	1.59	0.70	4039	1.00	0.94	10	0	114	15.249	5.06
24.7	1.00	1170.	16000.	11	3	5815	1.39	6.8	4133	0.96	5.7	4025	0.57	93	1.010	13.35
-0.22	63.00			4.0	1.3	4108	1.61	0.60	4025	1.00	0.96	5	0	61	11.642	5.16
24.6	1.00	1206.	16000.	13	4	5818	1.38	8.4	4209	0.95	7.6	4005	0.61	93	0.972	11.83
-0.07	64.00			4.1	1.3	4209	1.62	0.65	4005	1.00	0.95	7	0	82	13.072	5.24
24.6	1.00	1242.	16253.	15	4	5910	1.39	10.0	4252	0.94	9.5	4009	0.64	93	0.940	10.69
0.07	65.00			4.2	1.3	4252	1.61	0.59	4039	1.00	0.94	9	0	103	14.476	5.32
24.7	1.00	1279.	16115.	14	4	5860	1.38	9.7	4243	0.95	9.1	4016	0.63	93	0.947	10.97
0.04	66.00			4.2	1.4	4243	1.62	0.68	4016	1.00	0.95	8	0	98	14.142	5.40
24.7	1.00	1315.	16026.	14	4	5827	1.37	9.3	4239	0.95	8.6	4017	0.63	93	0.955	11.24
0.0	67.00			4.1	1.4	4239	1.63	0.67	4017	1.00	0.95	8	0	93	13.798	5.48
24.7	1.00	1351.	16000.	13	4	5813	1.37	8.9	4229	0.95	8.[illegible]	4016	0.62	93	0.963	11.53
-0.04	68.00			4.1	1.4	4229	1.64	0.66	4016	1.00	0.95	7	0	88	13.449	5.56
24.6	1.00	1387.	16000.	13	4	5819	1.38	8.5	4213	0.95	7.6	4009	0.61	93	0.972	11.84
-0.07	69.00			4.1	1.3	4213	1.62	0.65	4009	1.00	0.95	7	0	82	13.079	5.64

APPENDIX VI -2 LAST 33 SECONDS COMPUTER RUN

VEHICLE SPEED MPH ACCEL F/S/S	VEHICLE TIME INC TIME CUM	VEHICLE DIST FT	ENGINE RPM	ENGINE TOTAL HP ACCES HP	ENGINE TOTAL FTLB ACCES FTLB	TRACT.DRIVE RPM IN RPM OUT	TRACT.DRIVE SPEED RATIO RATIO RANGE	TRACT.DRIVE FTLB IN EFF	CONVERTER RPM IN RPM OUT	CONVERTER SPEED RATIO TORQ RATIO	CONVERTER FTLB IN EFF	GBOX RPM OUT TORQ OUT	TRANS EFF BRAKE LB	ROAD DRAG LB TRACT LB	FUEL LB/HPH LB/HR	FUEL MPG INST MPG CUM
0.0 0.0	1.00 1335.00	38669.	16000.	0 4.0	1 1.3	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0.0	0.0 0.0	0 0	0.0 0	0 0	0.0 8.000	0.0 7.69
0.1 0.55	1.00 1336.00	38669.	16000.	7 4.0	2 1.3	5818 2601	-2.24 1.00	3.4 0.48	2601 15	0.01 2.97	4.6 0.02	15 17	0.01 0	70 149	1.101 8.569	0.07 7.69
0.8 1.10	1.00 1337.00	38670.	16000.	10 4.0	3 1.3	5818 2842	2.05 1.09	5.2 0.59	2842 121	0.04 2.81	7.5 0.17	121 20	0.07 0	70 228	1.024 11.157	0.42 7.68
1.8 2.97	1.00 1338.00	38673.	18066.	22 4.3	6 1.3	6569 4206	1.56 1.43	14.1 0.76	4206 295	0.07 2.70	16.9 0.19	295 44	0.14 0	71 495	0.873 19.216	0.59 7.68
4.8 4.84	1.00 1339.00	38680.	20561.	39 4.7	9 1.2	7476 5284	1.41 1.58	23.7 0.66	5238 740	0.15 2.45	28.7 0.36	780 69	0.30 0	72 763	0.768 29.492	1.02 7.67
8.0 4.29	1.00 1340.00	38691.	20556.	38 4.7	9 1.2	7474 5121	1.46 1.53	23.7 0.66	5121 1302	0.25 2.14	29.6 0.54	1302 62	0.46 0	73 686	0.767 29.425	1.71 7.66
10.6 3.76	1.00 1341.00	38707.	20424.	37 4.6	9 1.2	7425 4939	1.50 1.49	23.1 0.65	4939 1732	0.35 1.90	29.5 0.67	1732 55	0.56 0	75 610	0.771 28.727	2.33 7.65
13.1 3.19	1.00 1342.00	38726.	20221.	35 4.6	9 1.2	7353 4767	1.54 1.45	22.2 0.85	4767 2131	0.45 1.70	28.9 0.76	2131 48	0.63 0	77 533	0.777 27.740	2.97 7.65
15.0 2.64	1.00 1343.00	38748.	19900.	33 4.5	8 1.2	7236 4623	1.57 1.43	20.9 0.94	4623 2439	0.53 1.54	27.4 0.81	2439 41	0.67 0	79 457	0.789 26.304	3.58 7.64
16.7 2.31	1.00 1344.00	38773.	19867.	33 4.6	8 1.2	7224 4644	1.56 1.44	20.7 0.84	4644 2723	0.59 1.40	27.0 0.82	2723 37	0.68 0	81 411	0.790 26.147	4.02 7.64
18.1 1.93	1.00 1345.00	38799.	19652.	31 4.6	8 1.2	7146 4585	1.56 1.43	19.9 0.83	4585 2952	0.64 1.31	25.8 0.84	2952 33	0.69 0	83 366	0.798 25.243	4.52 7.63
19.4 1.55	1.00 1346.00	38828.	19434.	30 4.5	8 1.2	7066 4549	1.55 1.44	19.1 0.82	4549 3162	0.70 1.21	24.4 0.84	3162 29	0.68 0	85 321	0.806 24.359	5.01 7.63
20.4 1.32	1.00 1347.00	38858.	19075.	27 4.5	7 1.2	6936 4465	1.55 1.44	17.8 0.81	4465 3318	0.74 1.13	22.4 0.84	3318 24	0.67 0	86 275	0.821 22.963	5.58 7.63
21.2 0.95	1.00 1348.00	38889.	18625.	25 4.4	7 1.2	6765 4354	1.55 1.44	16.1 0.79	4354 3454	0.79 1.04	19.8 0.83	3454 20	0.64 0	87 224	0.841 21.194	6.29 7.63
21.7 0.59	1.00 1349.00	38921.	17878.	21 4.3	6 1.3	6500 4177	1.56 1.44	13.5 0.75	4177 3529	0.84 1.00	15.8 0.85	3529 15	0.63 0	88 172	0.876 18.413	7.40 7.63
22.0 0.26	1.00 1350.00	38953.	16748.	15 4.2	5 1.3	6090 3963	1.54 1.46	10.7 0.70	3963 3585	0.90 1.00	11.6 0.90	3585 11	0.63 0	89 125	0.924 15.333	9.03 7.63
22.0 -0.07	1.00 1351.00	38986.	16000.	12 4.0	4 1.3	5818 3793	1.53 1.46	7.5 0.63	3793 3586	0.95 1.00	7.2 0.95	3586 7	0.58 0	89 78	0.992 12.277	11.28 7.63
21.9 -0.37	1.00 1352.00	39018.	16000.	8 4.0	2 1.3	5818 3668	1.58 1.47	4.2 0.51	3668 3570	0.97 1.00	3.4 0.97	3570 3	0.48 0	88 36	1.077 9.326	14.78 7.63
21.5 -0.66	1.00 1353.00	39049.	16000.	4 4.3	1 1.4	5818 2623	2.21 1.01	0.0 0.30	2628 3505	1.33 1.00	0.0 0.75	3505 0	0.59 0	88 -5	1.240 8.000	16.92 7.64
21.1 -0.73	1.00 1354.00	39080.	16000.	0 4.0	0 1.3	5818 2601	2.24 1.00	0.0 0.30	2601 3429	1.32 1.00	0.0 0.75	3429 0	0.59 0	87 -17	0.0 8.000	16.55 7.64
20.5 -0.81	1.00 1355.00	39110.	16000.	0 4.0	0 1.3	5818 2601	2.24 1.00	0.0 0.60	2601 3342	1.28 1.00	0.0 0.75	3342 0	0.59 0	86 -28	0.0 8.000	16.13 7.64
20.0 -0.95	1.00 1356.00	39140.	16000.	0 4.0	0 1.3	5818 2601	2.24 1.00	0.0 0.60	2601 3249	1.25 1.00	0.0 0.75	3249 0	0.59 0	86 -50	0.0 8.000	15.68 7.65
19.3 1.10	1.00 1357.00	39168.	16000.	0 4.0	0 1.3	5818 2601	2.24 1.00	0.0 0.80	2601 3131	1.20 1.00	0.0 0.75	3131 0	0.59 0	85 -72	0.0 8.000	15.11 7.65

VEHICLE SPEED MPH / ACCEL F/S/S	VEHICLE TIME INC / TIME CUM	VEHICLE DIST FT	ENGINE RPM	ENGINE TOTAL HP / ACCES HP	ENGINE TOTAL FTLB / ACCES FTLB	TRACT.DRIVE RPM IN / RPM OUT	TRACT.DRIVE SPEED RATIO / RATIO RANGE	TRACT.DRIVE FTLB IN / EFF	CONVERTER RPM IN / RPM OUT	CONVERTER SPEED RATIO / TORQ RATIO	CONVERTER FTLB IN / EFF	GBOX RPM OUT / TORQ OUT	TRANS EFF / BRAKE LB	ROAD DRAG LB / TRACT LB	FUEL LB/HPH / LB/HR	FUEL MPG INST / MPG CUM
18.5	1.00	39195.	16000.	0	0	5818	2.24	0.0	2601	1.15	0.0	3004	0.59	84	0.0	14.57
-1.24	1358.00			4.0	1.3	2601	1.00	0.80	3004	1.00	0.75	0	0	-99	8.000	7.65
17.9	1.00	39221.	16000.	0	0	5818	2.24	0.0	2601	1.09	0.0	2846	0.59	82	0.0	13.74
-1.47	1359.00			4.0	1.3	2601	1.00	0.80	2846	1.00	0.75	0	0	-126	8.000	7.65
16.5	1.00	39245.	16000.	0	0	5818	2.24	0.0	2601	1.03	0.0	2678	0.59	81	0.0	12.93
-1.65	1360.00			4.0	1.3	2601	1.00	0.80	2678	1.00	0.75	0	0	-154	8.000	7.66
15.3	1.00	39267.	16000.	0	0	5818	2.24	0.0	2601	0.95	0.0	2480	0.59	80	0.0	11.97
-1.33	1361.00			4.0	1.3	2601	1.00	0.80	2480	1.00	0.75	0	0	-181	8.000	7.66
13.9	1.00	39287.	16000.	0	0	5818	2.24	0.0	2601	0.86	0.0	2241	0.59	78	0.0	10.82
-3.12	1362.00			4.0	1.3	2601	1.00	0.80	2241	1.00	0.75	0	0	-366	8.000	7.66
11.0	1.00	39304.	16000.	0	0	5818	2.24	0.0	2601	0.69	0.0	1789	0.59	76	0.0	8.63
-4.40	1363.00			4.0	1.3	2601	1.00	0.80	1789	1.00	0.75	0	0	-552	8.000	7.66
8.3	1.00	39315.	16000.	0	0	5818	2.24	0.0	2601	0.50	0.0	1306	0.59	73	0.0	6.30
-4.22	1364.00			4.0	1.3	2601	1.00	0.80	1306	1.00	0.75	0	0	-528	8.000	7.66
5.3	1.00	39323.	16000.	0	0	5818	2.24	0.0	2601	0.33	0.0	853	0.59	72	0.0	4.12
-4.03	1365.00			4.0	1.3	2601	1.00	0.80	853	1.00	0.75	0	0	-503	8.000	7.66
2.7	1.00	39327.	16000.	0	0	5818	2.24	0.0	2601	0.17	0.0	437	0.59	71	0.0	2.11
-2.93	1366.00			4.0	1.3	2601	1.00	0.80	437	1.00	0.75	0	0	-347	8.000	7.66
1.3	1.00	39329.	16000.	0	0	5818	2.24	0.0	2601	0.08	0.0	203	0.59	70	0.0	0.98
-1.83	1367.00			4.0	1.3	2601	1.00	0.80	203	1.00	0.75	0	0	-190	8.000	7.65

1130-832, TORQUE CONVERTER - FLUID COUPLING SIZING AND PERFORMANCE ANALYSIS
REVISION F

VEHICLE..... ..EPA [illegible] CAR COMBINED DUTY CYCLE
ENGINE.........[illegible] ENGINE
TRANSMISSION...TRACTION DRIVE/TORQUE CONVERTER
CONVERTER.....[illegible] S.T.R.

ENGINE DRIVING MODE

TOTAL WHEELS-TO-ROAD ENERGY,HP.SEC	11268.36
TOTAL ENGINE ENERGY (INCL.ACCESS.),HP.SEC	21591.88
AVG. WHEELS-TO-ROAD HP	13.12
AVG. ENGINE HP (INCL.ACCESS)	25.14
TRANSMISSION ENERGY EFFICIENCY	0.666
TIME,SEC	859.00000
MILES	6.02767
GALLONS	0.78589
MPG	7.67

ENGINE COASTING MODE

TOTAL ROAD-TO-WHEELS ENERGY,HP.SEC	-3464.00
AVG. ROAD-TO-WHEELS HP	-6.82
TIME,SEC	508.00000
MILES	1.42104
GALLONS	0.18746
MPG	7.58

COMPLETE COURSE

TIME,SEC	1367.00000
MILES	7.44863
GALLONS	0.97335
MPG	7.65

MAX.TRACTION DRIVE RATIO	2.236	(AT 1336. SEC.)
MIN.TRACTION DRIVE RATIO	0.644	(AT 251. SEC.)
USEFUL TRACTION DRIVE RATIO RANGE	3.473	

APPENDIX VI-2 SIMPLIFIED SUBURBAN ROUTE, SIMPLIFIED COUNTRY ROUTE, SUMMARY OF COMBINED DUTY CYCLE

VEHICLE			ENGINE			TRACT. DRIVE			CONVERTER			GBOX		ROAD	FUEL	
SPEED MPH	TIME INC	DIST FT	RPM	TOTAL HP	TOTAL FTLB	RPM IN	SPEED RATIO	FTLB IN	RPM IN	SPEED RATIO	FTLB IN	RPM OUT	TRANS EFF	DRAG LB	LB/HPH	MPG INST
ACCEL F/S/S	TIME CUM			ACCES HP	ACCES FTLB	RPM OUT	RATIO RANGE	EFF	RPM OUT	TORQ RATIO	EFF	TORQ OUT	BRAKE LB	TRACT LB	LB/HR	MPG CUM
20.0	299.44	48112.	16000.	12	4	5816	1.65	7.6	3507	0.93	7.9	3253	0.57	86	0.991	10.18
0.0	1666.44			4.0	1.3	3507	1.31	0.63	3253	1.00	0.93	7	0	86	12.338	8.02
30.0	299.44	61288.	16974.	17	5	6172	1.21	11.0	5096	0.96	9.5	4879	0.67	103	0.916	11.99
0.0	1965.89			4.2	1.3	5096	1.65	0.71	4879	1.00	0.96	9	0	103	15.714	8.63
40.0	299.44	78855.	18328.	23	6	6664	0.59	15.1	6727	0.97	11.6	6506	0.74	126	0.849	12.58
0.0	2265.32			4.4	1.3	6727	2.26	0.76	6506	1.00	0.97	11	0	126	19.961	9.28
50.0	156.23	90312.	197[illegible]3.	31	8	7164	0.66	20.1	6374	0.97	14.3	8132	0.80	155	0.798	12.46
0.0	2421.55			4.6	1.2	6374	2.61	0.83	8132	1.00	0.97	14	0	155	25.204	9.59
60.0	156.23	104060.	21031.	43	10	7647	0.76	26.3	10018	0.97	17.6	9759	0.84	191	0.731	11.97
0.0	2577.78			4.7	1.2	10018	2.93	0.67	9759	1.00	0.97	17	0	191	31.490	9.85
70.0	156.23	120099.	21859.	57	13	7948	0.68	34.5	11665	0.98	21.4	11385	0.88	232	0.699	11.19
0.0	2734.01			4.8	1.2	11665	3.28	0.91	11385	1.00	0.98	21	0	232	39.281	10.01

```
1130-832, TORQUE CONVERTER - FLUID COUPLING SIZING AND PERFORMANCE ANALYSIS
-----------------------------------------------------------------REVISION F

VEHICLE.......EPA TURBINE CAR  COMBINED DUTY CYCLE
ENGINE........AEROJET ENGINE
TRANSMISSION..TRACTION DRIVE/TORQUE CONVERTER
 CONVERTER.....FORD 3.0 S.T.R.
---------------------------------------------------------------------------

FEDERAL DRIVING CYCLE

   TIME,SEC                                          1367.00000
   MILES                                                7.44563
   GALLONS                                              0.97335
   MPG                                                  7.65

SIMPLIFIED SUBURBAN ROUTE

   TIME,SEC                                           898.39981
   MILES                                                7.45500
   GALLONS                                              0.63591
   MPG                                                 11.77

SIMPLIFIED COUNTRY ROUTE

   TIME,SEC                                           468.69994
   MILES                                                7.81150
   GALLONS                                              0.66322
   MPG                                                 11.78

RESULTANT COMPOUND DRIVING CYCLE

   TOTAL WHEELS TO ROAD ENERGY,HP,SEC                 33951.16
   TOTAL ENGINE ENERGY (INCL.ACCESS.),HP,SEC          50136.37

   AVG. WHEELS-TO-ROAD HP                                15.25
   AVG. ENGINE HP (INCL.ACCESS)                          20.12

   TRANSMISSION ENERGY EFFICIENCY                         0.738

   TIME,SEC                                          2734.00000
   MILES                                               22.74309
   GALLONS                                              2.27248
   MPG                                                 10.01

SCALE FACTOR FOR 3500 HRS. VEHICLE LIFE =    4608.63
```

APPENDIX VI-3. VEICLE PERFORMANCE WITH A TYPICAL 3 SPEED AUTOMATIC TRANSMISSION

Vehicle performance was computed for the vehicle specified in the "Prototype Vehicle Performance Specification" (Apendix I) using data for a typical 350 cubic inch displacement internal combustion engine and a typical 3 speed automatic transmission supplied by EPA under Phase I of this contract.

A. Applicable Data

1) Engine power and SFC curve (Figure VI-3A).

2) Vehicle drag and resistance forces (see "Prototype Vehicle Performance Specification," Appendix I.

3) Remaining transmission and vehicle data, see Figures VI-3B and Figure VI-3C.

4) "Typical" 3-speed automatic transmission, vehicle vs. transmission efficiency curves, see Figures VI-3D and VI-3E.

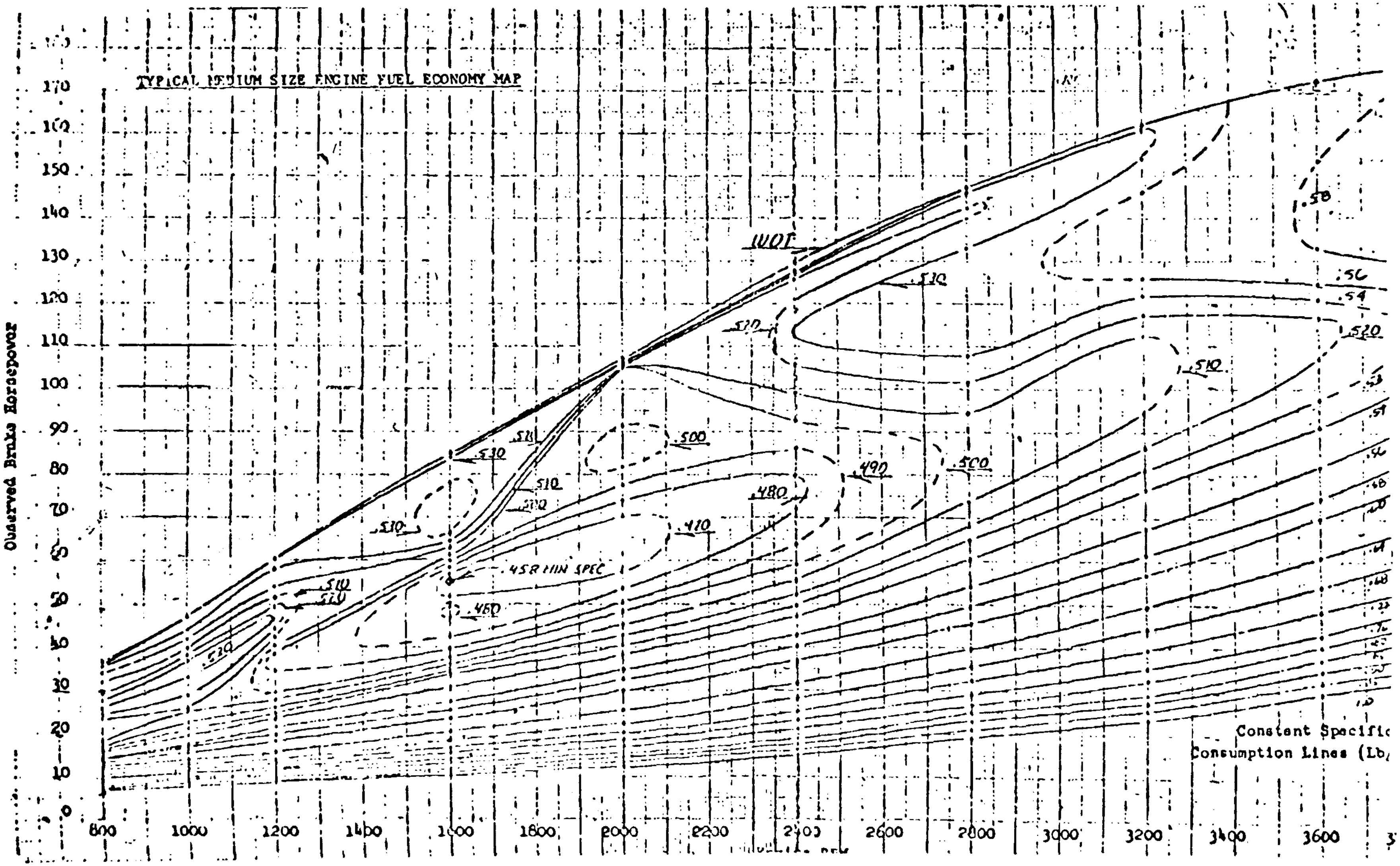


Figure VI-3A Engine Power and SFC Curve

Sundstrand Aviati

Appendix VI-3   Figure VI-3B

```
1130-P32,TORQUE CONVERTER-FLUID COUPLING SIZING AND PERFORMANCE ANALYSIS
VEHICLE PERFORMANCE VERSION-------------------------------------REVISION F

VEHICLE........FULL SIZED CAR (PER EPA)
ENGINE.........350 CU. IN V8 (PER EPA)   (AIR CONDITIONER ON)
TRANSMISSION..3 SPEED AUTOMATIC (2.5 1ST,1.5 2ND) (PER EPA)
CONVERTER.....11.75 INS DIA ,2.0 STR (PER EPA)
------------------------------------------------------------------------------

ENGINE ACCESSORY POWER
-------------------
      ENGINE RPM
        800. 1600. 2400. 3200.

      ACCESSORY HP
        4.00  6.20  8.40 10.60

FUEL MAP FOR ENGINE NO...................................      1
ENGINE IDLE SPEED,RPM....................................  800.0
IDLE FUEL RATE,LB/HR.....................................  6.000
ENGINE SPEED N2,RPM...................................... 4800.0
FUEL RATE AT N2,LB/HR....................................  [illegible].000

FUEL DENSITY,LB/GAL......................................   6.28

SHIFT POINT DATA
---------------
      NO. OF SHIFT POINTS..................      2
      ROAD HP AT DETENT SHIFT.............  120.0
      MIN. SHIFT POINT SPEEDS,MPH.........    5.0    15.0
      DETENT SHIFT POINT SPEEDS,MPH.......   33.0    55.0
      MAX SHIFT POINT SPEEDS..............   50.0    75.0

 11.75 IN. REFERENCE CONVERTER/COUPLING WAS GIVEN AS INPUT DATA
-------------------------------------------------------------
      SPEED RATIO
       0.0   0.200 0.400 0.600 0.800 0.900 0.925 0.950 0.975 0.985 0.990
      TORQUE RATIO
       2.000 1.800 1.600 1.370 1.120 1.000 1.000 1.000 1.000 1.000 1.000
      CAPACITY FACTOR,K
        106.  112.  120.  131.  151.  171.  190.  222.  343.  380.  422.
```

Appendix VI-3 Figure VI-3C

```
1120-B32,TORQUE CONVERTER-FLUID COUPLING SIZING AND PERFORMANCE ANALYSIS
VEHICLE PERFORMANCE VERSION--------------------------------------REVISION F

VEHICLE.......FULL SIZED CAR (PER EPA)
ENGINE........350 CU.IN V8 (PER EPA)  (AIR CONDITIONER ON)
TRANSMISSION..3 SPEED AUTOMATIC (2.5 1ST,1.5 2ND) (PER EPA)
CONVERTER.....11.75 IN DIA ,2.0 STR (PER EPA)
------------------------------------------------------------------------------
                              INPUT DATA
                              ----------

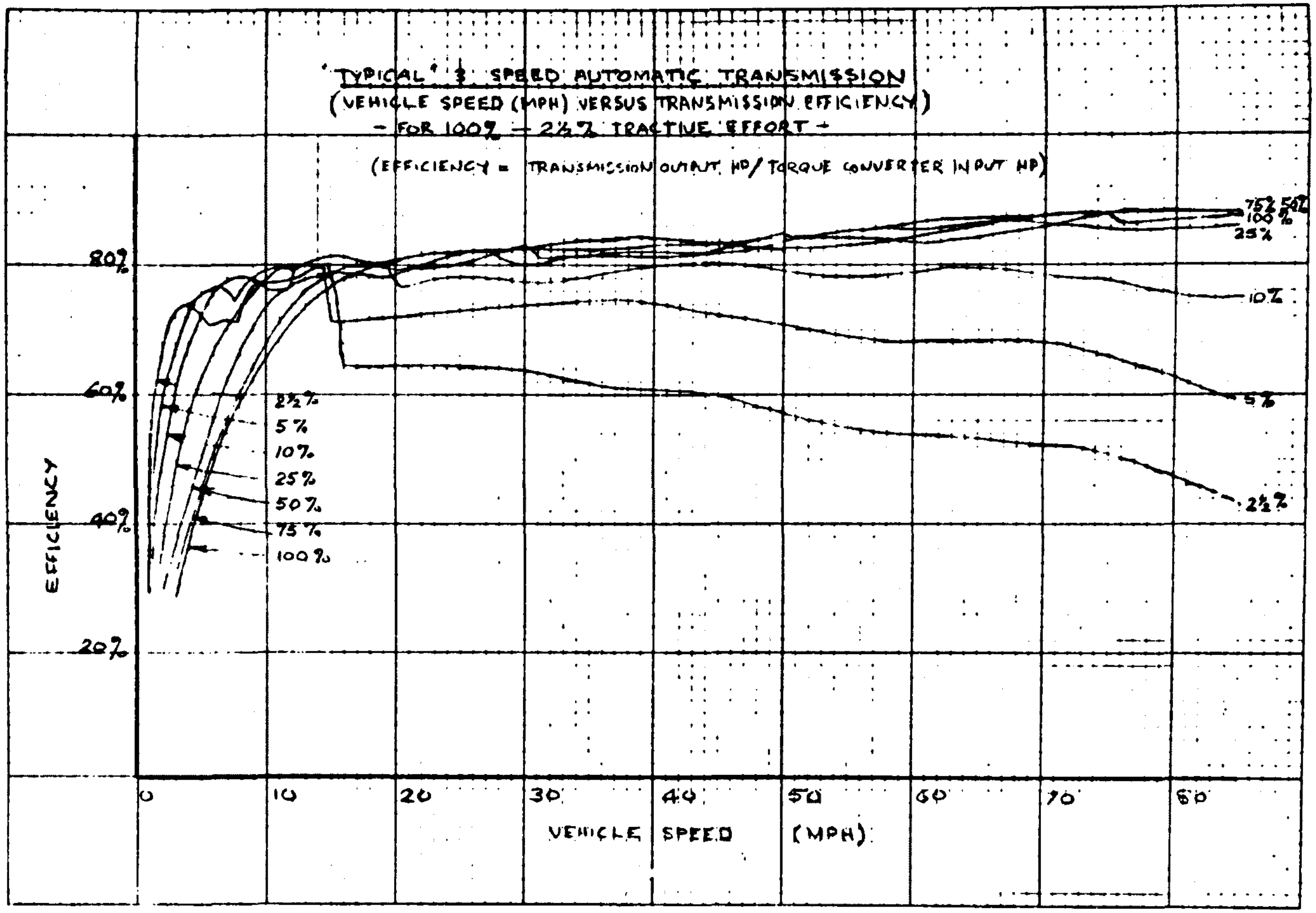
VEHICLE ACCELERATION PERFORMANCE  (AND EPA COMPOUND DRIVING
                                   CYCLE PERFORMANCE)
AXLE RATIO.......................................... 2.750
AXLE EFFICIENCY..................................... 0.950
VEHICLE WEIGHT...................................... 4600.
DRIVE WHEEL RADIUS,FT............................... 1.100
TOTAL WHEEL INERTIA,SLUG-FT-FT...................... 11.20
FRONTAL AREA,SQ.FT.................................. 20.00
AIR TEMP.,DEG.F..................................... 85.0
AERODYNAMIC DRAG FACTOR............................. 0.600
ROAD GRADE,PERCENT.................................. 0.0
RATIO ENG.SPEED/TRANS.INPUT SPEED................... 1.000
M.I.T. ROLLING RESISTANCE FORMULAE USED
INPUT CONDITIONS
----------------
      INPUT INERTIA      0.30000 SLUG-FT-FT

      INPUT SPEED,RPM
       800. 1200. 1600. 2000. 2500. 3000. 3500. 4000. 4500.

      INPUT TORQUE, LB-FT
       237.  267.  279.  281.  282.  272.  258.  230.  195.
TRANSMISSION
------------
      RATIOS
       2.500 1.500 1.000

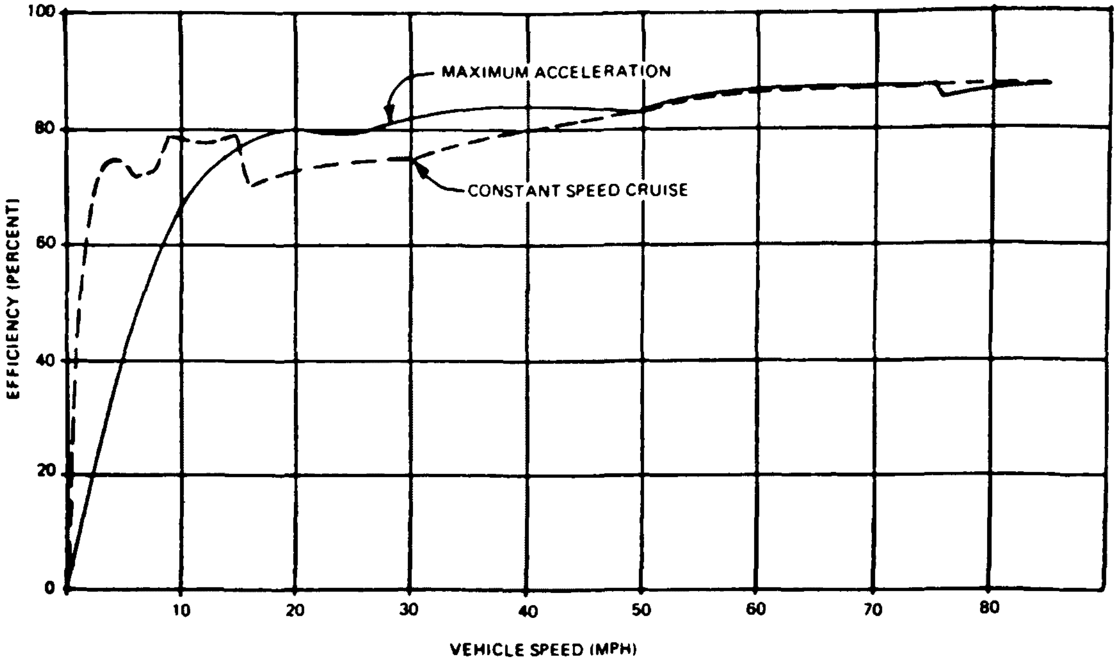
      EFFICIENCIES
       0.970 0.970 0.970

      SHIFT SPEEDS,MPH (ENGINE RPM IF GREATER THAN 100.)
         50.   75.    (WIDE OPEN THROTTLE SHIFT POINTS)
SPEED FOR TIRE SLIP FACTORS,MPH
         0.0  10.0  20.0  30.0  40.0  50.0  60.0
TIRE SLIP FACTORS
       0.890 0.910 0.940 0.960 0.970 0.980 0.990
         (USED FOR MAX. POWER PERFORMANCE ONLY.
          OTHER WISE FACTOR = 1.0, AT ANY SPEED)
```



Appendix VI-3D ''Typical'' 3 Speed Automatic Transmission
(Vehicle Speed versus Transmission Efficiency)

Sundstrand Aviation
division of Sundstrand Corporation



Appendix VI-3E Transmission Efficiency vs. Vehicle Speed
''Typical'' 3 Speed Automatic Transmission

B. Performance Summary for the "Typical 3-Speed Automatic Transmission"

	Weight	Performance	
		Air Cond. On	Air Cond. Off
Idle Creep Speed	1	18 3	–
Accel.			
Time to 60 MPH	1	11.74 sec.	–
Dist. in 10 Sec.	1	460 ft.	–
Time 25 → 70 MPH	1	12.69 sec.	–
Time 50 → 80 MPH (D.O.T. HI. – SPD. PASS)	1	12.0 sec.	–
Dist. 50 → 80 MPH (D.O.T. HI – SPD PASS)	1	1150 ft.	–
Grade Velocity			
30%	2	19 mph	–
5%	2	84 mph	–
0%	1	115 mph	–
Fuel Consumption			
Fed. Driv. Cycle	1	11.94 mpg	12.60 mpg
Simplified Suburban Route	1	17.86 mpg	19.38 mpg
Simplified Country Route	1	15.99 mpg	17.14 mpg
Combined EPA Driv. Cycle	1	14.85 mpg	15.87 mpg

1 Vehicle Weight 4600 lb.

3 In Third Gear

2 Vehicle Weight 5300 lb.

Total Vehicles Accessory Power with Air Conditioner on – 4.0 HP at min eng. speed (800 RPM), Linear to 15.0 HP at max. eng. speed (4800 RPM) – with Air Conditioner Off – 2.0 HP at min. eng. speed, Linear to 5.0 HP at max. eng. speed.

Atmospheric Conditions, 85^{O}F, 14.7 PSIA

(SPACER PAGE – INTENTIONALLY BLANK)

VII-1 Control System Parameters

a – Governor Valve Porting Area Coefficient = 2 x 0.078 In.

A_g – Governor Valve Spool Area = 0.1963 In^2

A_v – Variable Unit Control Piston Area = 6.52 In^2

B_1B_n – Bulk Modulus of Type A Hydraulic Fluid

k_o – Porting Area Flow Coefficient

Temperature °F	Pressure PSI	B - PSI	k_o
0	0	3.0×10^5	91
100	0	2.2×10^5	93
150	0	1.8×10^5	94
200	0	1.5×10^5	95
250	0	1.23×10^5	96
0	3000	3.6×10^5	91
100	3000	2.7×10^5	93
150	3000	2.3×10^5	94
200	3000	2.0×10^5	95
250	3000	1.7×10^5	96
200	6000	2.4×10^5	95

B_g – Governor Valve Damping Coefficient = 0.0449 lb/in/sec

C_V – Variable Unit Control Damping Coefficient = 0.006 lb/in/sec

D_F – Fixed Unit Displacement (10 in^3/rev) = 1.59 in^3/rad.

D_V – Variable Unit Displacement per Unit Stroke = 1.275 in^2/rad.

J_e – Polar Moment of Inertia of Engine = 4.0 in-lb-sec^2

J_s – Polar Moment of Inertia of Vehicle Reflected to Sun Gear of Transmission = 381 in-lb-sec^2

k_g – Governor Valve Spring Coefficient = 47.8 lb/in

k_v – Variable Unit Control Piston Spring Coefficient = 60 lb/in

VII-1 Control System Parameters (continued)

K_e — Engine Governor Pressure Coefficient = 0.000741 PSI/(rad/sec)^2

K_R — Pressure Regulator Valve Coefficient = 2.04 PSI/PSI

L — Vari./Fixed Unit Leakage Coefficient = 0.00585 c.i.s./p.s.i.

L_V — Control Piston Leakage Coefficient = 0.0001 c.i.s./p.s.i.

M_g — Mass of Governor Valve Spool = 4.46×10^{-4} lb/in/sec^2

M_x — Effect Mass @ Control Piston = 0.02435 lb/in/sec^2

R_1 — Input Gear Ratio (N_V/N_e) = 0.8242

R_2 — Gear Ratio (N_F/N_S) = 1.975

$V_1 = V_2$ — Volume of Fluid In Control Circuit = 12 in^3

VII-2 Equations

1. Vehicle Velocity, MPH

   $MPH = 0.04996\, W_s$ m.p.h.

2. Angular Velocity of Sun Gear, W_s

   $W_s = 1/J_s \int (R_2 T_F - T_L - T_{AR} - T_{RR})\ dt$ rad/sec

3. Fixed Unit Torque, T_F

   $T_F = D_F\, P_w$ in-lb

4. System Working Pressure, P_w

   $P_w = B_w/V_w \int (D_v X_v W_v - D_F R_2 W_s - L\, P_w)\ dt$ p.s.i.

5. Variable Unit Angular Velocity, W_v.

   $W_v = R_1\, W_e$ rad/sec

6. Variable Unit Control Stroke, X_v

   $X_v = 1/m_v \iint (-C_v \dot{X}_v - k_v X_v + A_v P_c - F_w)\ dt\ dt$ INCHES

7. Control Pressure, P_c.

   $P_c = P_1 - P_2$ p.s.i.

   where:

   $P_1 = B/V_i \int (a_1 k_o \sqrt{P_R - P_1} - a_3 k_o \sqrt{P_1 - P_d} - A_v \dot{X}_v - L_v P_c)\ dt$ p.s.i.

   $P_2 = B/V_2 \int (-a_2 k_o \sqrt{P_2 - P_d} + a_4 k_o \sqrt{P_R - P_2} + A_v \dot{X}_v + L_v P_c)\ dt$ p.s.i.

8. Governor Valve porting areas, a_1, a_2, a_3, & a_4

   $a_1 = a_2$ & $a_3 = a_4 = a\, X_g$ $IN.^2$

9. Regulated Pressure, P_R

   $P_R = K_R\, P_{NE}$ p.s.i.

   where:

   $P_{NG} = K_e\, W_e^2$ p.s.i.

10. Governor Valve Spool Position, X_g.

$$X_g = 1/m_g \iint (-B_g X_g - k_g X_g + A_g P_{NE} - F_{PL} - k_g X_{in})\, dt\, dt \text{ INCHES}$$

where:

X_{in} = 0 to 0.5 in., Throttle input.

11. Engine Angular Velocity, W_e

$$W_e = 1/J_e \int (T_e - R_1 T_v)\, dt \text{ in-lb.}$$

12. Variable Unit Torque, T_v.

$$T_v = D_v X_v P_w \text{ In-Lb.}$$

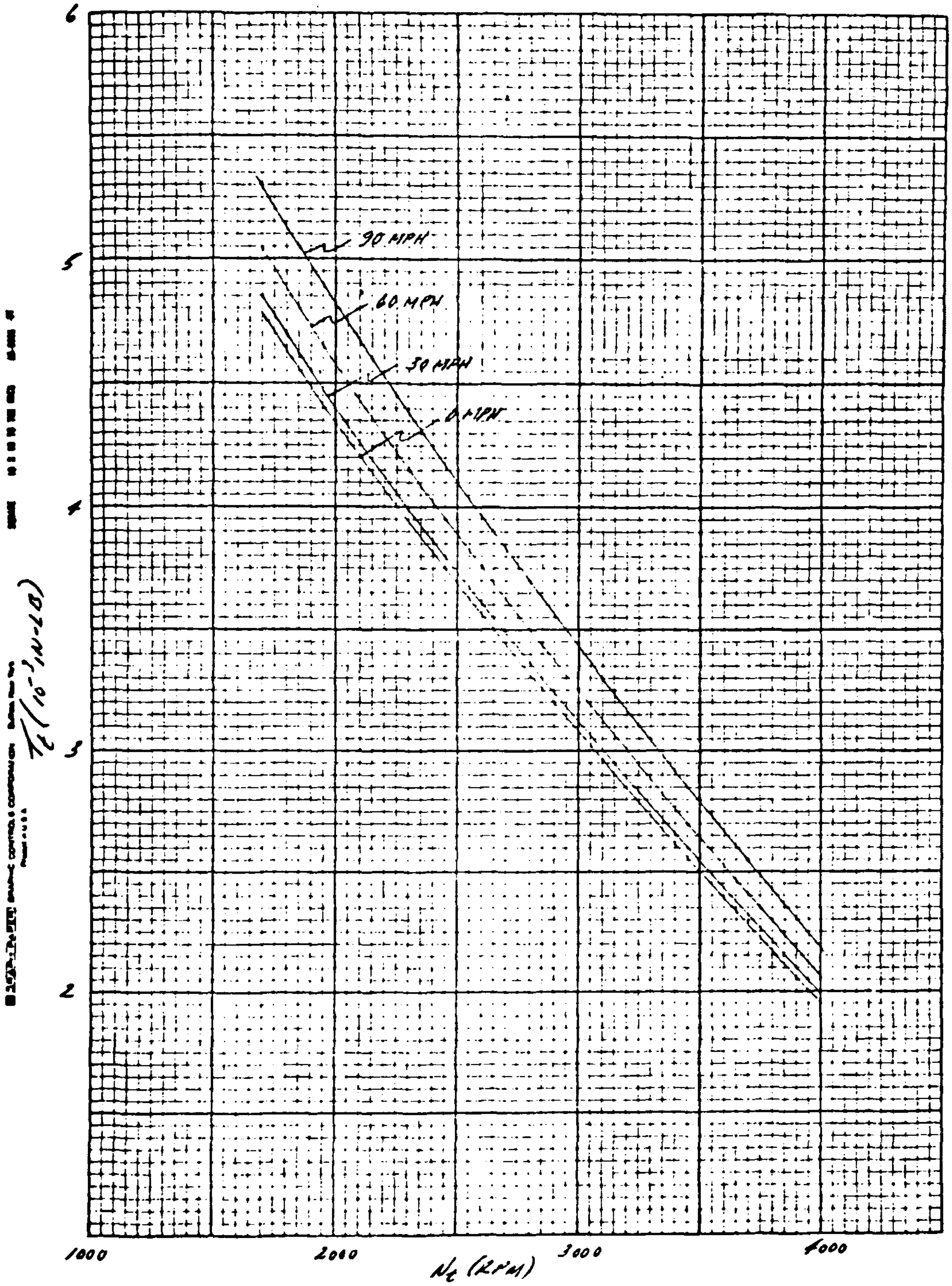
13. Engine Torque, T_e

$$T_e = f\,(W_e, \text{MPH}, X_{in}) \text{ In-Lb}$$

14. Variable Unit Wobbler Force, F_w

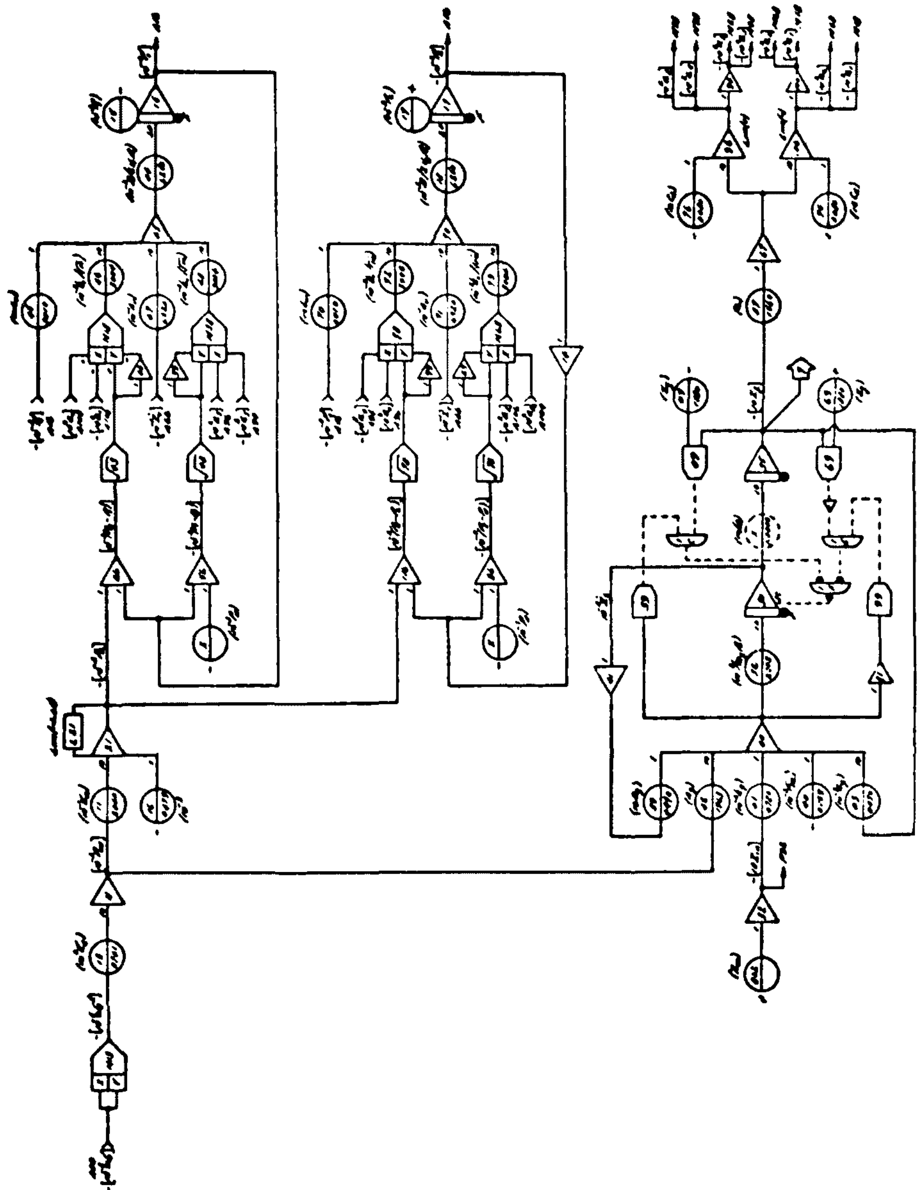
$$F_w = 0.045\, A_v P_w \text{ Lb.}$$

VII-3 TURBINE TORQUE VS. TURBINE SPEED - RANKINE ENGINE



```
      SCALED FRACTION NX(2),NY(2),X,Y,FXY
      CALL QSHYIN(IERR,680)
      CALL QSC(1,IERR)
   10 CALL QRBADS(X,1,1,IERR)
      CALL QRBADS(Y,2,1,IERR)
      CALL XN(X,NX,IRFG)
      IF(IRFG.NE.1) GO TO 50
      CALL YN(Y,NY,IRFG)
      IF(IRFG.NE.1) GO TO 60
      CALL FXYG(NX,NY,FXY)
      CALL QWJDAS(FXY,1,IERR)
      GO TO 10
   50 TYPE 200,X,NX,IRFG
      PAUSE 10
      GO TO 10
   60 TYPE 201,Y,NY,IRFG
  200 FORMAT(18H ARG NORM ERR   X=,S7,5X,S7,5X,I3)
  201 FORMAT(18H ARG NORM ERR   Y=,S7,5X,S7,5X,I3)
      PAUSE 20
      GO TO 10
      END
      SUBROUTINE XN(IX,INX,IERFG)
      SCALED FRACTION XT(9)
      DATA XT(1)/.1779S/,XT(2)/.2093S/,XT(3)/.2407S/,
     1     XT(4)/.2721S/,XT(5)/.293S/,XT(6)/.3244S/,
     2     XT(7)/.3558S/,XT(8)/.3872S/,XT(9)/.4186S/
      CALL VBNS(XT(1),XT(9),XT(5))
      RETURN
      END
      SUBROUTINE YN(IY,INY,IERFG)
      SCALED FRACTION YT(4)
      DATA YT(1)/.0S/,YT(2)/.3S/,YT(3)/.6S/,YT(4)/.9S/
      CALL VBNS(YT(1),YT(4),YT(2))
      RETURN
      END
      SUBROUTINE FXYG(INX,INY,IFXY)
      SCALED FRACTION FT(36)
      DATA FT(1)/.4781S/,FT(2)/.4331S/,FT(3)/.3939S/,FT(4)/.3572S/,
     1     FT(5)/.333S/,FT(6)/.2973S/,FT(7)/.2626S/,FT(8)/.2287S/,
     2     FT(9)/.1953S/
      DATA FT(10)/.4855S/,FT(11)/.4388S/,FT(12)/.3988S/,
     1     FT(13)/.3615S/,FT(14)/.3364S/,FT(15)/.3S/,
     2     FT(16)/.2655S/,FT(17)/.2319S/,FT(18)/.1989S/
      DATA FT(19)/.5040S/,FT(20)/.4568S/,FT(21)/.4153S/,
     1     FT(22)/.3756S/,FT(23)/.3499S/,FT(24)/.3124S/,
     2     FT(25)/.2768S/,FT(26)/.2414S/,FT(27)/.2074S/
      DATA FT(28)/.5314S/,FT(29)/.4820S/,FT(30)/.4388S/,
     1     FT(31)/.3974S/,FT(32)/.3701S/,FT(33)/.3302S/,
     2     FT(34)/.2924S/,FT(35)/.2546S/,FT(36)/.2177S/
      CALL FNGN2(9,FT(1))
      RETURN
      END
```

VII-5 ANALOG COMPUTER WIRING DIAGRAM (sheet 1 of 2)



VII-5 ANALOG COMPUTER WIRING DIAGRAM (sheet 2 of 2)

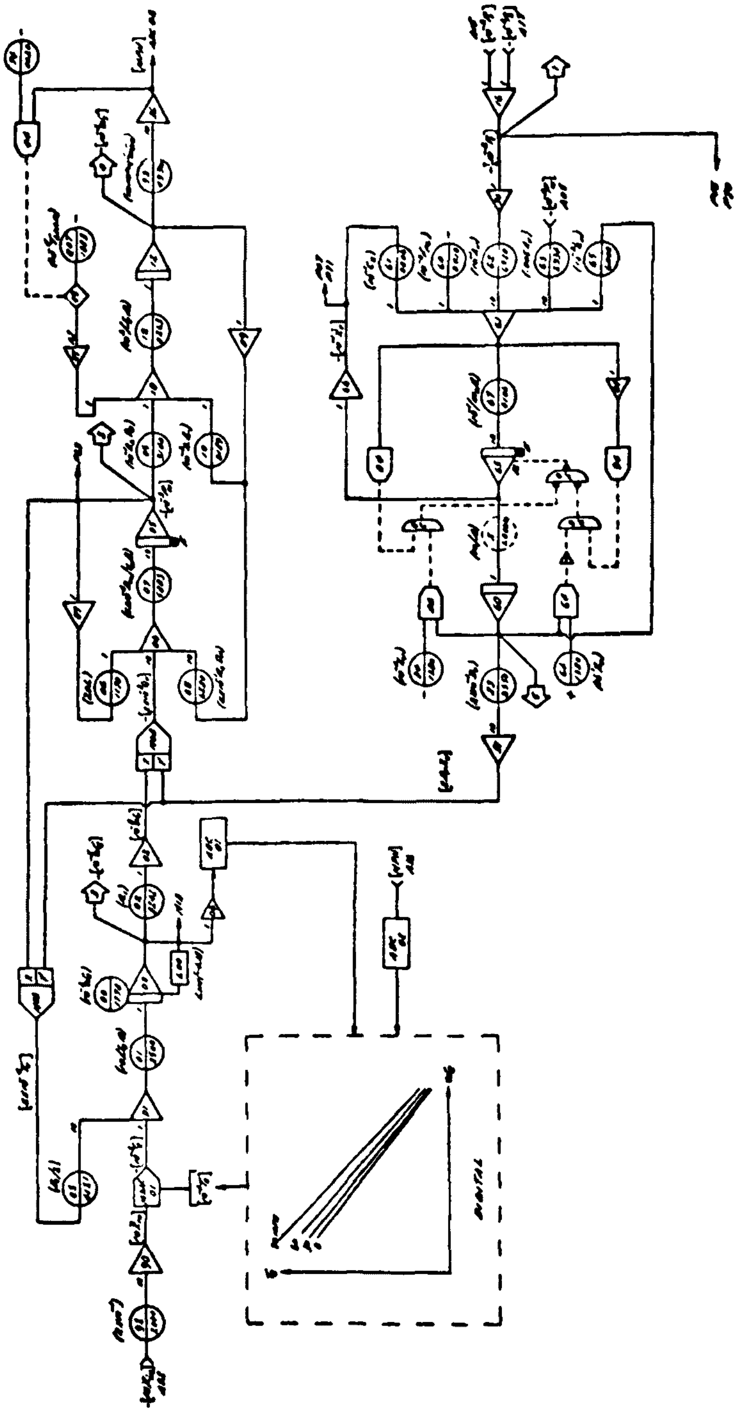


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FIGURE VII-6 (1) COMPUTER READOUT FOR A 0.2 UNIT THROTTLE ACCELERATION AND 50% LOAD

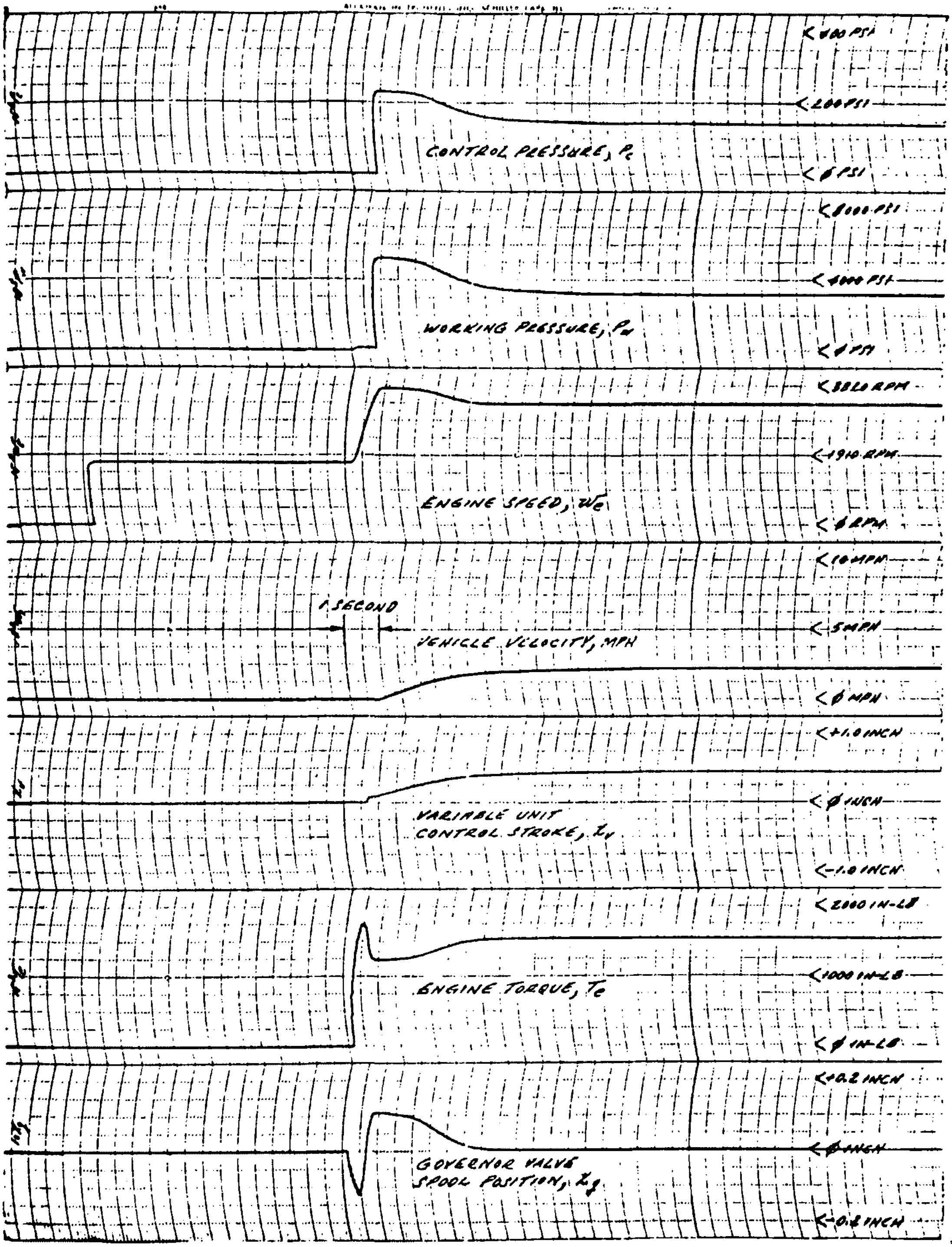

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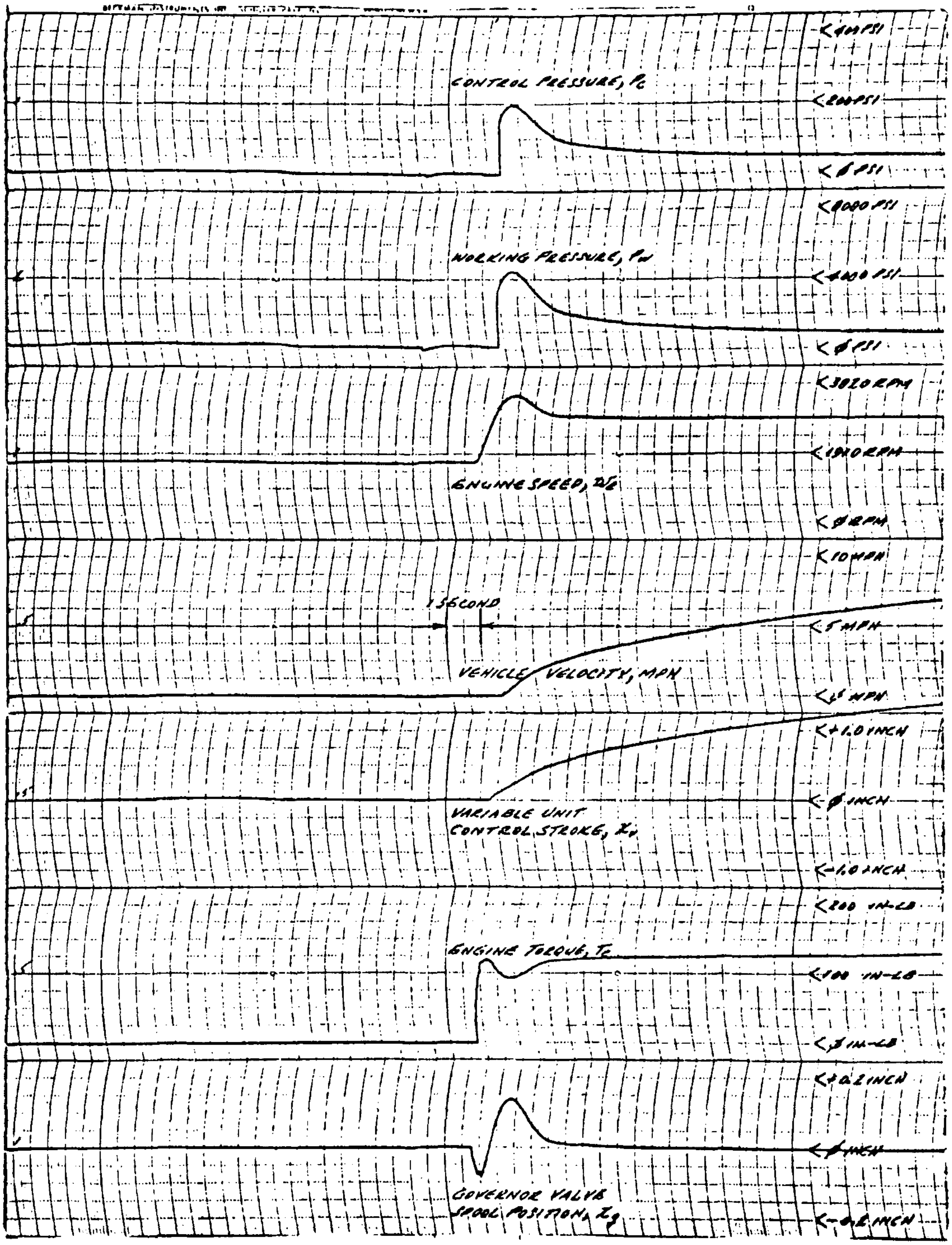
FIGURE VII-6 (2) COMPUTER READOUT FOR A 0.2 UNIT THROTTLE ACCELERATION AND ZERO LOAD



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BIBLIOGRAPHIC DATA SHEET	1. Report No. APTD-1558	2.	3. Recipient's Accession No.

4. Title and Subtitle
Transmission Study for Turbine and Rankine Cycle Engines

5. Report Date
December 15, 1972

6.

7. Author(s)
M. A. Cordner and D. H. Grimm

8. Performing Organization Rept. No. AER 657

9. Performing Organization Name and Address
Sundstrand Aviation
division of Sunstrand Corporation
Rockford, Illinois 61101

10. Project/Task/Work Unit No.

11. Contract/Grant No.
68-04-0034

12. Sponsoring Organization Name and Address
ENVIRONMENTAL PROTECTION AGENCY
Office of Air Programs
Division of Advanced Automotive Power Systems
Ann Arbor, Michigan 48105

13. Type of Report & Period Covered

14.

15. Supplementary Notes

16. Abstracts
A study was initiated to quantitatively assess the technical and economic feasibility of existing and potential types of transmissions most suitable for the gas turbine and Rankine cycle engines. Application of the engine/transmission was to a full size family car. The study was accomplished through a two-phase, multi-task program which included: (1) evaluation of transmission types through a feasibility study and ultimate selection of a transmission type; (2) evaluation of the selected transmission type through design calculations and layouts, performance analysis, control system analysis, and cost analysis. A number of different types of transmission were initially evaluated including conventional multi-shift, hydrostatic, hydrokinetic, electric, belt/chain, hydromechanical, and traction types. Requirements, scope of work, and other data utilized in and pertinent to the study are included in the appendices.

17. Key Words and Document Analysis. 17a. Descriptors
Air pollution
Automotive transmissions
Engines
Turbines
Rankine Cycle
Feasibility
Economic analysis
Performance tests
Cost analysis
Design

17b. Identifiers/Open-Ended Terms
Control system analysis

17c. COSATI Field/Group 13B

18. Availability Statement
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19. Security Class (This Report) UNCLASSIFIED

20. Security Class (This Page) UNCLASSIFIED

21. No. of Pages 207

22. Price

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