APTD-1517

TRANSMISSION FOR ADVANCED AUTOMOTIVE SINGLE-SHAFT GAS TURBINE AND TURBO-RANKINE ENGINE

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air and Water Programs
Mobile Source Pollution Control Program
ed Automotive Power Systems Development Division
Ann Arbor, Michigan 48105

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FOREWORD

This report presents a summary of a study project performed by Mechanical Technology Incorporated with cooperation from the Ford Motor Company. The work was performed for the Environmental Protection Agency, Office of Air Programs, Division of Advanced Automotive Power Systems EPA/AAPS under Contract 68-04-0033.

A key consultant for the project was Mr. Edwin Charles in regard to the costing analyses. Acknowledgement is given to Mr. George DeLalio who submitted a proprietary design of a hydromechanical transmission; to Tracor for supplying information on their traction drive; and to the Rohr Corporation for supplying proprietary information on their transmission design.

Mr. R. C. Bowlin was responsible for overall project direction. Other major contributors from Mechanical Technology Incorporated were P. Lewis, H. Jones, Dr. A. Smalley and Ms. Linda Almstead.

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.

SECTION I

SUMMARY

The purpose of this study was to assess, on both a performance and cost basis, the transmission most suitable for use with two types of advanced automotive engines designed to power a medium sized family car. The advanced automotive engines were a conceptual single-shaft gas turbine and a turbo Rankine engine based upon design characteristics supplied by AiResearch and Aerojet, respectively.

The scope of the study consisted of a feasibility study to select a candidate transmission, a preliminary design of the selected transmission, a detailed performance analysis, control systems definition, cost analysis, and a specific recommendation concerning the desirability of the selected transmission for the engine types investigated.

As a result of considering eight different types of transmissions, a continuously variable ratio, power-splitting, hydromechanical transmission was selected as the best near-term transmission for application with the single-shaft gas-turbine and the turbo-Rankine engines. This type of transmission combines hydraulic elements with mechanical-gear elements to achieve a variable, stepless ratio that achieves torque multiplication and control by means of the hydraulic elements. The best long-term candidate was determined to be the Traction-type transmission. This transmission holds great promise for the future, but requires substantial development effort.

The selected hydromechanical transmission was designed in sufficient detail to ascertain performance, cost and physical characteristics. The design philosophy followed was to minimize the number of mechanical elements so as to achieve simplicity and reliability with minimum size, weight, and cost while still meeting specified vehicle performance goals. As a result, in comparison with a current automatic transmission for a medium-sized car, the selected transmission was only 3 percent heavier, 28 percent smaller in volume, and required 6 percent more parts.

Production cost is a key factor in considering a transmission for automotive applications. A detailed cost analysis, based upon costing procedures of the Ford Motor Company, showed that the "original equipment manufacture" (O.E.M.) cost of the selected transmission was 30 to 40 percent higher than a present day

automatic transmission in production quantities of 1,000,000 units/year. On a variable cost basis (more meaningful to the automotive companies on comparing designs) the increase in cost was 44 percent. Higher material cost was the major factor in causing the increased cost. However, since the weight of the selected transmission was approximately the same as an automatic transmission, this suggests that, with development of production techniques, future production costs for such a transmission could approach that of a present day automatic transmission.

From a performance viewpoint, the selected transmission was compatible with both engine types and provided smooth acceleration characteristics that met specified vehicle performance goals. Average transmission efficiency over the Federal Driving Cycle ranged from 71 to 74 percent with the turbo-Rankine and single-shaft gas turbine engines, respectively. Comparable efficiency of a conventional automatic transmission powered by an IC engine has been estimated at 78 percent The slightly lower efficiency of the selected transmission was due to speed dependent losses which cause the efficiency to be lower than that of a conventional automatic at cruise-power levels below 25 mph. With respect to fuel economy (miles/gallon), the selected transmission resulted in significant improvement compared with the conventional automatic transmission for the single-shaft gas turbine, but little improvement was shown for the turbo-Rankine engine. This was because the minimum specific fuel consumption characteristics of the single-shaft gas turbine changed much more with engine speed variations than did the turbo-Rankine engine used for this study. Thus, there was little advantage gained in maintaining engine speed of the turbo-Rankine with a variable-ratio transmission to achieve maximum miles/gallon.

For this reason, the most significant conclusion of this study was that further development of the selected variable ratio hydromechanical transmission is recommended for application to the single-shaft gas turbine and not for the turbo-Rankine engine, unless future design developments for the latter engine result in showing a significant change in SFC with engine speed.

SECTION II

INTRODUCTION

Recently various engine types have been considered as possible alternatives to present automotive engines in order to improve exhaust emissions. Two of the engine types are a single-shaft gas turbine and a Rankine-cycle engine with a turbine expander. Both of these engines require a transmission in order to achieve torque multiplication for adequate vehicle power. In addition, based upon fuel economy, the transmission should allow these engines to operate at conditions which minimize fuel consumption.

The purpose of this study was to select the most promising candidate transmission for these types of turbine engines and to determine the resultant performance and production cost of the selected transmission in comparison with a conventional automatic transmission.

Selection of the most promising transmission must take into account, the performance requirements of the vehicle as well as the constraints imposed by development time for new components, reliability, size and cost. On this basis an overall assessment pointed to the desirability of the powersplitting type of transmission. Even with this concept there are many possible variations which could increase the capabilities of the transmission and achieve higher efficiencies, etc. For this study, the selection and design philosophy concentrated on minimizing the number of mechanical elements to achieve simplicity and high reliability with state-of-the-art components within the constraints of size, weight, and cost. As a consequence, more complicated versions were not considered for this study.

A preliminary design of the selected transmission was made in sufficient detail to provide the basis for a reasonable estimate of transmission weight, volume, number of parts, and cost. In addition, the transmission control system was defined by a control schematic; control logic details were investigated in conjunction with determining power-train performance.

Engine characteristics used in this study were supplied by AiResearch based on their conceptual design for a single-shaft gas turbine and Aerojet based on their "prototype" design for a turbo-Rankine engine with an organic working fluid. The respective engine maps were incorporated into digital computer models in order to determine steady-state (cruise), maximum acceleration, and Federal Driving Cycle performance with the selected transmission.

The following sections of this report discuss a summary of the results and then present conclusions and recommendations. Detailed supporting information is contained in the last section of the report.

SECTION III

DISCUSSION OF RESULTS

This section summarizes and discusses the results of the study by considering in the following order:

A. Selection of candidate transmission
B. Description of selected transmission
C. Performance of selected transmission
D. Cost and physical comparisons

Additional supporting details are given by topical headings in Section V - Supporting Information.

A. Selection of Candidate Transmission

A number of different types of transmissions were specified by EPA/AAPS for for consideration as a candidate transmission. These types were:

- Mechanical
- Hydrostatic
- Combination of Mechanical and Hydrostatic
- Hydrokinetic
- Electrical
- Traction
- Belt Drive

As discussed in more detail in Section V-A, all of the above types were considered in order to select a candidate transmission for the single-shaft gas turbine and the turbo-Rankine engine. Several of the above were eliminated from serious consideration because of inherent limitations. For example, a purely mechanical gear-type transmission was eliminated on the basis that it would not provide the continuously variable ratio with stepless changes, which is required for smooth operation. Similarly the hydrostatic and electrical were eliminated because of large volume, high cost and relatively low efficiency for this application.

As a result, the possible candidate transmissions were narrowed down to eight specific types of transmission, which were:

1. Three-Speed Automatic with Variable Torque Converter Element
2. Advanced Hydromechanical
3. Conventional Hydromechanical
4. Traction - TRACOR
5. Traction - Power-Splitting
6. Friction - Composition Belt
7. Traction - Metal Belt
8. Three-Speed Automatic with Aerodynamic Torque Converter

Each of the above transmission types was reviewed and evaluated. Descriptive details for each of the transmission types and the details of the evaluation are given in Section V-A. Table 1 presents a summary of the descriptive evaluation for each transmission type and Table 2 presents an evaluation summary.

As a result, on an overall basis, the power-splitting hydromechanical transmission (subsequently described) was selected as the most promising candidate transmission on a near-term development (1974) basis. Key attributes of this type of transmission were relative simplicity, comparable size and weight to a three-speed automatic transmission which is currently used, weight (which would imply future cost comparable to the standard automatic), proven components well within the current state-of-the-art, and the ability to provide optimum engine speed throughout the desired operating range. Disadvantages of this type of transmission include excessive noise — particularly when operated at high hydraulic pressures — and reduced efficiency at low power levels if input speeds are high.

From a long-range viewpoint, with additional development work, the traction type of transmission was found to offer considerable promise as an alternate candidate transmission.

TABLE 1

DESCRIPTIVE EVALUATION OF CANDIDATE TRANSMISSIONS

	Life & Reliability	Noise & Smoothness	Cost	Development Status	Efficiency	Size & Weight	Restriction On Turbine Engine	Control Complexity	Driver Acceptability	Environmental Restrictions
1. 3 Speed Automatic w/variable element	Similar to existing (except for variable element)	Good - similar to existing	Additional cost for extra element	Developed - not in production	Low - Penalty because of variable element	Nominal increase over existing AT	Slight penalty due to steps	Minimal	Similar to existing AT	Similar to existing AT
2. Advanced Hydromech.	Similar to existing AT. Benefit of simplicity	Smoother Oper - Requires attention to noise	Slightly higher than existing	Developed - not in auto production	Higher than existing AT, except at low power levels	Comparable to existing AT	Minimal	Minimal	Similar to existing AT	Similar to AT, Reqires attention to shock design
3. Conventional Hydromech.	Similar to AT. Some penalty over (2)	Smoother Oper - Requires attention to noise	Higher than (2)	Developed - not in auto production	Higher than existing AT, except at low power levels	Higher than (2)	Minimal	Minimal	Similar to existing AT	Similar to AT, Requires attention to shock design
4. Traction - TRACOR	Not defined at this time	Smooth Oper - w/low noise potential	Higher than (2)	Not completely developed for auto hp range	Not as high as (2)	Higher than existing AT	Minimal	Minimal	Similar to existing AT	Similar to AT, Requires attention to shock design
5. Traction-Power-Splitting	Not defined at this time	Smooth Oper - w/low noise potential	Higher than (2)	Concept Only	Potentially similar to (2)	Higher than existing AT	Minimal	Minimal	Similar to existing AT	Low shock absorption capability
6. Traction - Composition Belt	Not defined at this time	Smooth Oper - w/low noise potential	Potential for low cost	Not developed for auto hp range	Potentially similar to (2)	Not defined for auto applic.	Possible speed range limitations	Minimal	Similar to existing AT	Similar to existing AT
7. Traction - Metal Belt	Not defined at this time	Smooth Oper - w/low noise potential	Potential for low cost	Not developed for auto hp range	Potentially similar to (2)	Not defined for auto applic.	Possible speed range limitations	Minimal	Similar to existing AT	Similar to existing AT
8. 3-Speed Automatic w/Aero torque converter - ROHR	Not defined at this time	Smooth Oper - w/low noise potential	Higher than (2)	Not developed as a transmission	Not as high as (1), significant penalty for Aero element	Not defined, should be higher than existing AT	Minimal	More complex	Similar to existing AT	Similar to existing AT

Note: AT designates current mass produced three-speed automatic transmission

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TABLE 2

EVALUATION SUMMARY

	TRANSMISSION TYPE	MAIN ADVANTAGE	MAIN DISADVANTAGE
1.	Three-Speed Automatic with Variable Torque converter element	Similar to existing automatic transmission	Lower efficiency than existing automatic
2.	Advanced Hydromechanical	Simple mechanical construction	Manufacturing techniques need to be developed and noise minimized
3.	Conventional Hydromechanical	Development experience existing	Manufacturing techniques need to be developed and noise minimized
4.	Traction - TRACOR	Future potential for high efficiency with low noise	
5.	Traction-Power Splitting	Future potential for high efficiency with low noise	Not to design stage
6.	Friction-Composition Belt	Manufacturing techniques developed	Life and efficiency undetermined
7.	Traction-Metal Belt	Future potential for high efficiency	Not to development stage
8.	Three-Speed Automatic with Aerodynamic Torque Converter-Rohr	Manufacturing technique developed for bulk of the transmission	Not to complete transmission design stage.

B. Description of Selected Transmission

The selected hydromechanical transmission is an infinitely variable, stepless unit that obtains torque multiplication and control by means of hydraulic elements (pump-motor-combination). The transmission is based upon a proprietary design of Mr. George DeLalio. Consequently, design of the transmission has not been included in this report. The unit differs from the conventional torque converter or fluid-coupling-type transmission, in that the hydraulic power is transferred by fluid static pressure at low flow as contrasted to the high dynamic action of fluid as utilized in hydrodynamic units. It also differs from a purely hydrostatic transmission in that the much more efficient mechanical elements transmit a significant portion of the power. It is a "hard" type of drive, in that slip is less than 2 percent under full load.

The design of the transmission was based upon minimizing the number of mechanical parts in order to: a) make the transmission as simple as possible, b) keep the size, weight and, particularly, the cost low, c) require the minimum amount of development, and d) achieve high reliability while retaining a reasonably high transmission efficiency. Additional gear trains and hydraulic functions could have been employed to increase the capabilities of the transmission and reduce the amount of power in the hydraulic elements. Clearly, however, this approach would have involved more parts, and a more costly transmission. It was more consistent with the stated design philosophy to concentrate on a straight forward, simple, design.

Figure 1 is a functional schematic of the selected transmission. As shown on this figure the transmission consists of an engine input shaft, a variable-displacement hydraulic element and a fixed-displacement hydraulic element. Additional components which are essential to the transmission operation are: a brake, which locks one element of the planetaries for low-speed operation; a clutch, which closes the mechanical power both for high-speed operation, and a control system.

As shown in Figure 1, the power path between input and output splits between hydrostatic and straight mechanical. The transmission has two operating ranges designated as low-range and high-range. At low power and output speed levels (up to approximately 0.4 of maximum output speed) the transmission operates in the low range and all power flows through the hydraulic elements. At higher power and speed conditions the transmission operates in high range and the power path is

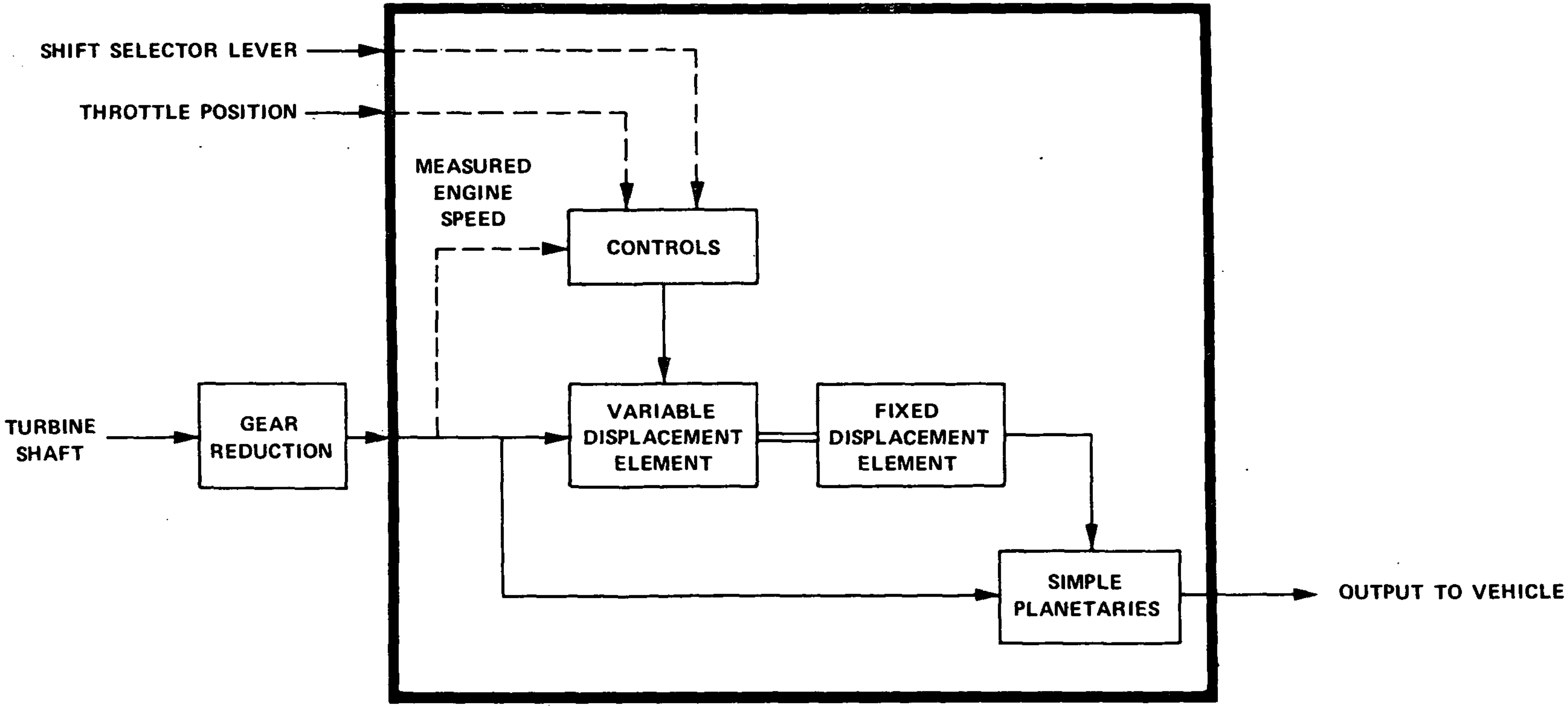

Fig. 1 Functional Schematic — Selected Hydromechanical Variable-Ratio Transmission

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split between the hydraulic elements and the mechanical path. At approximately 0.7 of maximum output speed, all the power flow is through the mechanical path.

An important feature of this design is the synchronous shift which operates as follows: The transmission has two operating ranges, low speed and high speed. Transfer between these two ranges is effected by the concurrent opening of a brake and closing of a clutch. The shift is ideally synchronous if it occurs under the following conditions:

1. In either low or high range the displacement of the variable hydraulic element is at the same maximum point.

2. There is no change in relative velocity between the two sides of the clutch.

3. The braked element is stationary whether the brake is applied or not.

In practice such effects as slip between hydraulic elements and control imperfections cause slight deviations from the ideally synchronous shift. Even so, there is never a significant change in momentum demanded of the rotating parts and the brake and clutch elements suffer little slippage or wear.

While theoretically unlimited , the practical low-speed torque ratio range lies between 5:1 and 2.5:1 and the high-speed (mechanical path) range lies between 2.5:1 and 1:1. Thus the overall ratio range for the selected transmission lies between 5:1 and 1:1. With the 3.08 rear-end ratio, the maximum torque ratio between transmission input and the wheels is 15.4.

From a driver viewpoint, operation of the selected transmission is similar to a present automatic transmission, except no "kick down" is required to downshift. Gear shift lever functions are identical to a present three speed automatic transmission and are described in detail in Section V-B.

The controls required for the hydromechanical transmission will be more sophisticated than those required to operate the present automobile automatic transmission. This results from the fact that to exploit full benefit from the continuously variable transmission it must be controlled to operate the turbo-rankine or gas turbine engine within the most economical fuel flow range. To accomplish this objective an engine speed regulator governor and an extra flow path through the transmission control valve are added.

The raw signals used to achieve optimum fuel economy are accelerator pedal position and engine speed. The accelerator pedal position signifies commanded vehicle speed and is translated via a cam into a signal, representing the corresponding optimum engine speed (desired speed) (determined by design analysis). The engine speed is translated, via the engine regulator governor, into a speed signal, and compared to the desired speed. The error signal (hydraulic) causes the piston actuator to move the swash plate cam, so establishing a new position of the variable hydraulic element and a different transmission ratio. Subject to limitations of the vehicle engine and control system time constants, optimum fuel consumption at the commanded vehicle speed is achieved (on a level road). Of course frequent acceleration and braking will tend to limit fuel economy as with any vehicle power train.

One of the key features of the transmission control system is that at engine idle speed the transmission is automatically disengaged from the engine, again minimizing fuel flow. The operation is accomplished by sending an idle engine speed signal from the engine engage governor through the additional flow path of the control valve and on to the engaging spool valve. The engaging valve then disengages the hydraulic element and the transmission from the engine.

When the engine speed is above idle the engage governor, through similar hydraulic flow paths, causes the transmission to engage.

The shift from low range to high range, or vice-versa, is designed to occur when the displacement of the variable element is at its extreme negative value. A particular valve (the clutch valve) is used for this purpose. A cam, controlled by the actuator piston has a profile which, over a short part of its travel, slews the clutch valve from its low range position to its high range position. The effect of this valve is to relieve pressure from the low-range brake and apply it to the high-range clutch, or vice-versa.

The controls hardware is described in more detail in Section V-B.

C. Performance of Selected Transmission

The performance of the selected hydromechanical transmission was determined over a wide range of operating conditions. Three categories of performance were investigated: 1) steady-state (cruise and constant power output), 2) Federal Driving Cycle, and 3) full-power accelerations. In each analysis the performance was determined for two different types of advanced automotive engines powering a medium-sized family car with power-train loading specified by EPA/AAPS vehicle design goals (1). Comparisons, where possible, were made with the corresponding performance of a similar power-train employing a conventional automatic transmission.

The engine characteristics were supplied by Aerojet and AiResearch as specified by EPA/AAPS as a result of work under separate EPA/AAPS contracts. The characteristics of the Aerojet engine were based upon their "prototype" design for a turbo-Rankine-cycle engine, which employs an organic working fluid. The AiResearch engine characteristics were based upon their conceptual design for a single-shaft gas turbine. Details of the engine maps and power-train loading are given in Section V-D.

It should be pointed out that: (1) the respective engine characteristics were based upon different design constraints not necessarily optimized for operation with the selected transmission and (2) the engine performance predictions have not been demonstrated experimentally. For these reasons comparisons between engines, based upon these study results, are not warranted.

1. Transmission Performance Characteristics

The resultant transmission efficiency* of the selected hydromechanical transmission is dependent upon the mode of operation. In low range, at low vehicle speeds, all the power passes through the hydraulic elements which limit the efficiency. At high speeds, where the majority of the power passes through the mechanical power path, the efficiency is higher. Peak efficiency occurs close to the "straight-through" point, where power flow through the hydraulic path is essentially zero.

*Transmission efficiency, as used throughout this report, relates the power output from the transmission (delivered to the differential) and the input power to the transmission input shaft.

The above characteristics can be seen by referring to Figure 2, which presents the variation of transmission efficiency with vehicle velocity under cruise-power conditions for both engines. At 20 mph the efficiency is between 57 to 60 percent, but increases to 90 percent at speeds of 50 mph or higher. The slight discontinuity in slope at 20 mph with the AiResearch engine, and at 25 mph with the Aerojet engine, indicates the cruise-power shift point between low and high speed ranges. The "straight-through" point occurs at 40 mph and 50 mph for the AiResearch and Aerojet engines, respectively.

The marked difference between the transmission efficiencies with the two engines at speeds below 50 mph reflects the differences in the manufacturer's specified operating line. Up to approximately 24 hp output from the engine, the transmission input speed with the AiResearch engine is 2300 rpm (minimum engaged speed). However, with the Aerojet engine, the transmission input speed lies between 2700 and 3000 rpm. This difference in speed means that speed dependent transmission losses are more significant with the Aerojet engine than with the AiResearch engine and, at the low power levels, was 13 percent at 30 mph and 3 percent at 20 mph.

The variation in efficiency at constant power levels as shown in Figures 3 and 4 reveals some differences from the cruise efficiency curves. This is particularly noticeable at low speeds. For example, at 10 mph, the transmission cruise efficiency is 51 percent; whereas, with an output at the road of 10 percent of rated power, the efficiency is 78 percent. This difference results from the fact that the cruise power demand at 10 mph is only 2 hp at the road, exaggerating the importance of speed dependent losses, which are of similar magnitude to the road load. A further observation from the constant-power efficiency curves is the reduced separation between the two engines at high power levels, indicating the reduced significance of speed-dependent losses at high power levels. For a cross-plot of transmission efficiency versus output power at constant speeds refer to Section V-D.

The peak transmission efficiency is close to 95 percent, under conditions of high power at the "straight through" speed condition. This high efficiency value is a result of the minimal losses which occur when all power passes through the

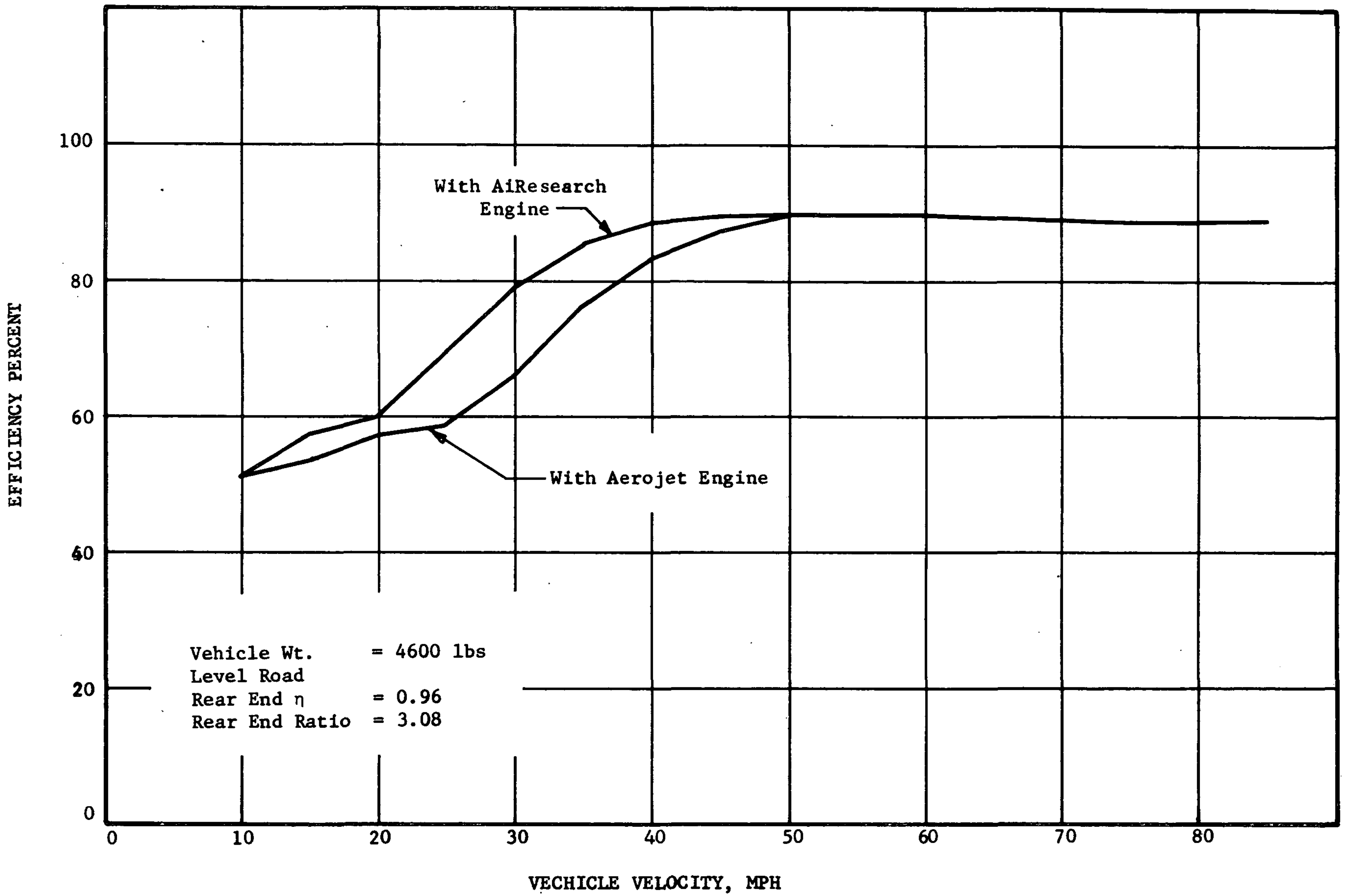

Fig. 2 Cruise-Power Efficiency of Selected Transmission

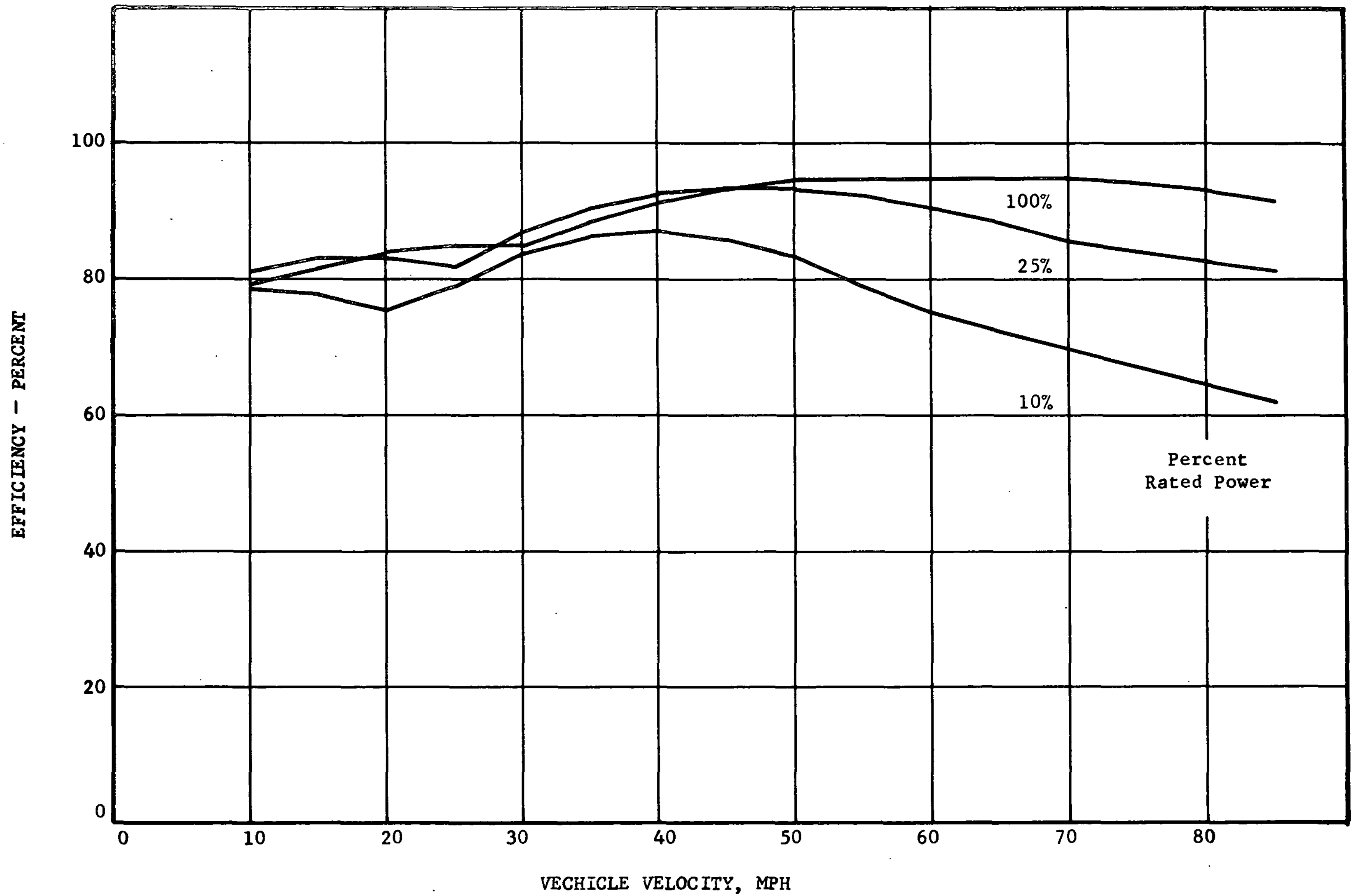

Fig. 3 Constant Power Efficiency of Selected Transmission — AiResearch Engine

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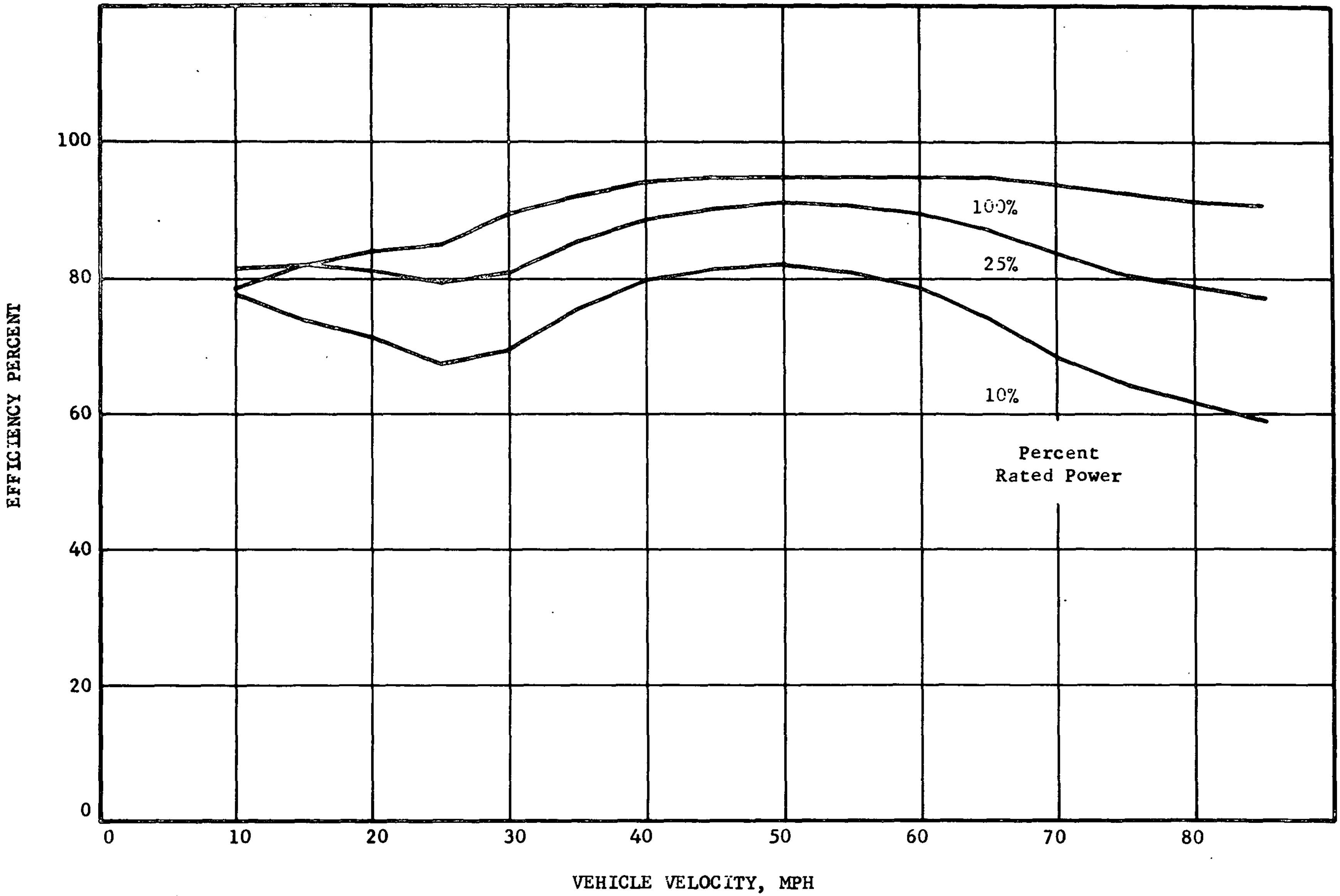

Fig. 4 Constant Power Efficiency of Selected Transmission — Aerojet Engine

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mechanical path. The 5 percent losses which are incurred are the sum of the cumulative losses in the output planetary, the residual power necessary to rotate the variable hydraulic element, even when no power is passing through it, and the "parasitic" losses such as the charge pump.

Figure 5 presents a comparison of the selected hydromechanical transmission cruise power efficiency with that of the conventional automatic. The conventional automatic used for this comparison was that currently selected by Aerojet for use with their turbo-Rankine engine. The shift points are therefore designed to produce optimum system performance for this engine. The wide spread of the shift points, even under the cruise conditions of Figure 5, is necessitated when a vehicle speed range for zero to 85 mph is to be provided by an engine whose ratio of maximum to idle speed is well below 2. In fact, the complete transmission incorporated by Aerojet includes a separate idle gear for use between 10 and 22 mph and a slipping clutch for speeds below 10 mph. Thus, in the range 10-85 mph, the so-called "conventional automatic" actually behaves as a 4-speed, rather than a 3-speed.

Below 30 mph the conventional automatic is significantly more efficient than the selected transmission - the difference reaching 24 efficiency points at 10 mph for the Aerojet engine. As shown in Section V-D (Figures D-10 and D-11), the efficiency of the selected hydromechanical transmission decreases rapidly below 20 percent load due to speed dependent losses. By contrast, the part load efficiencies for a torque converter transmission increase with decreasing load within the converter range. At increased load corresponding to vehicle speeds above 35 mph, it can be seen from Figure 5, that the selected transmission efficiency was equal to, and in some instances better than, the automatic transmission.

Additional insight into the performance of the selected transmission is provided by Table 3, which gives a detailed breakdown of the power flow and indicates the contribution of mechanical and hydraulic losses to performance. Vehicle speeds at 20 mph and 60 mph with the Aerojet engine were selected as typical operating conditions. The first of these speeds is slightly below the shift point, and the second speed is somewhat above the "straight-through" point. In both cases the amount of power flowing through the hydraulic path is very similar.

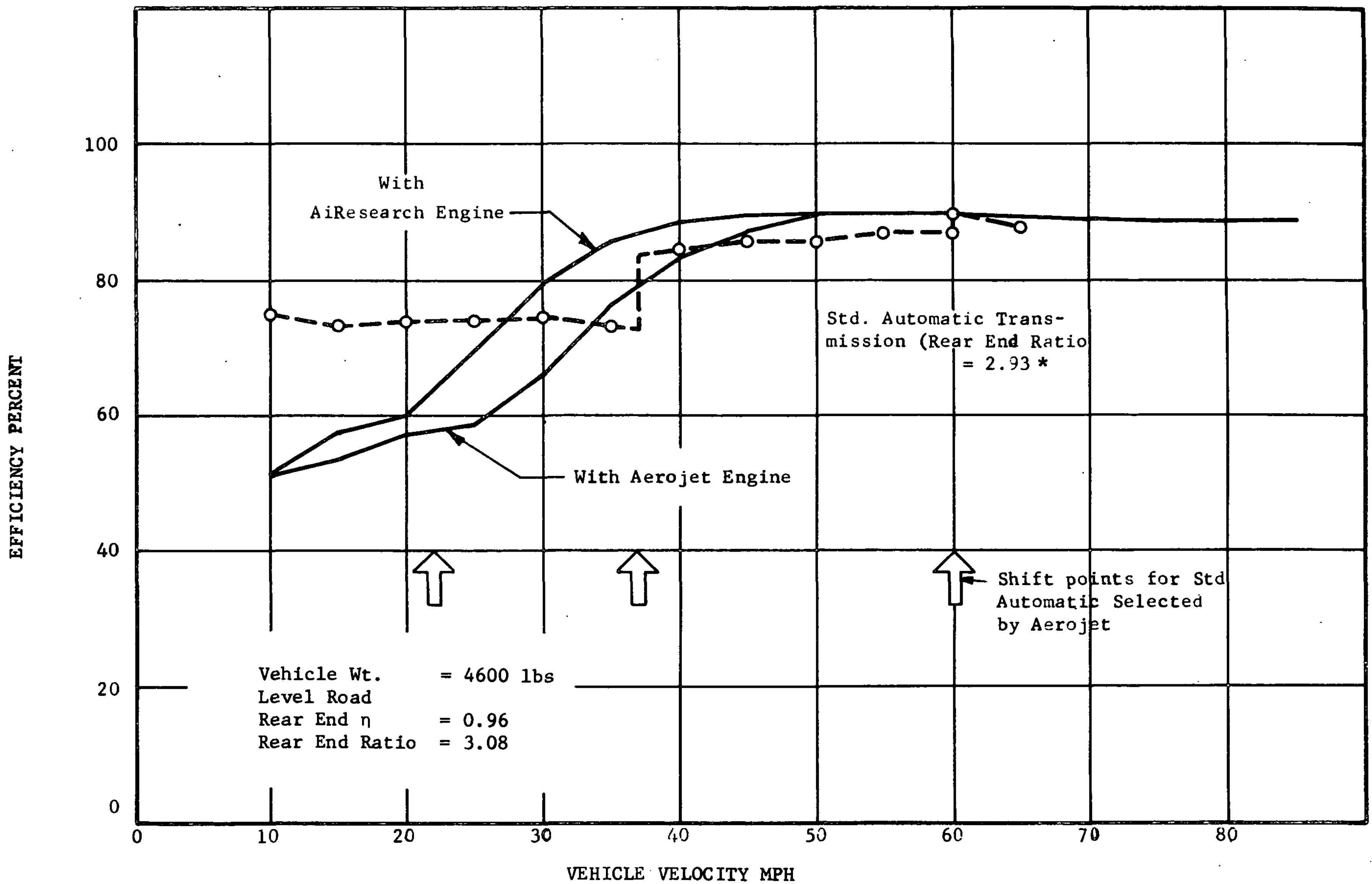

Fig. 5 Comparison of Transmission Efficiency — Cruise Power

* Different Rear End Ratio for Standard Automatics selected by Aerojet to give optimum performance with remainder of power train (engine + transmission)

TABLE 3

TYPICAL POWER FLOW BREAKDOWN - AEROJET ENGINE

WITH AIR CONDITIONER - CRUISE POWER

	20 MPH			60 MPH		
	AVAILABLE HP	HP USED & LOSSES	% OF TRANSMISSION INPUT HP	AVAILABLE HP	HP USED & LOSSES	% OF TRANSMISSION INPUT HP
ENGINE HP	13.114			39.908		
ACCESSORY HP		4.740	56.60		4.911	10.03
TRANSMISSION INPUT HP	8.374			34.996		
MECHANICAL PATH LOSSES		1.126	13.45		1.356	3.87
HYDRAULIC PATH LOSSES		2.472	29.52		1.814	5.18
TRANSMISSION OUTPUT HP	4.776			31.826		
DIFFERENTIAL LOSSES		0.184	2.20		1.224	3.50
ROAD HP	4.592			30.602		

However, at 20 mph the only power flow to the mechanical path is that necessary to overcome friction and to drive the charge pump - no output power is delivered by the mechanical path. At 60 mph most of the delivered power passes through the mechanical path. The influence of this difference in power split is reflected in the percentage contribution of the hydraulic and mechanical losses. At 20 mph hydraulic losses account for 29.5 percent of the transmission input power, and mechanical losses account for 13.5 percent. At 60 mph the hydraulic losses fall to 5.2 percent and the mechanical losses to 3.9 percent.

The design operating pressure in the hydraulic elements of a hydromechanical transmission is important with respect to reliability (life) and noise. Maximum operating pressures above 3500 psi not only reduce the life of the elements but also cause unwanted excessive noise. Thus, low operating pressures are highly desirable to minimize transmission noise.

Typical design operating pressures for the selected transmission are shown by Figure 6. At cruise-power, operating pressures were low, reaching a maximum of 330 psi at 85 mph. Under full-power demands the pressure remains close to its limiting value of 3500 psi at speeds of 10 and 15 mph, reflecting the design limit corresponding to wheel slip. At higher speeds, even in low range, the full power can be transmitted through the hydrostatic path without exceeding the pressure limits, and the pressure begins to fall with speed. At a speed of 85 mph, the full-power pressure has decreased to about 530 psi. These results indicate that the selected transmission design was conservative with respect to operating pressure and therefore noise levels should be at a minimum, since the noise is most strongly influenced by pressure and speed. Even though the low operating pressures minimize noise, it should be recognized that they do not eliminate the noise problem. Isolation and noise insulation techniques will probably have to be developed to achieve driver aceptability.

2. Power-Train Performance - Aerojet Engine

The fuel economy of a vehicle power train, in mpg, is, of course, a meaningful measure of the overall efficiency with which fuel is being converted to useful work. The variable ratio selected transmission was controlled in a manner so as to maintain engine speed for minimum fuel consumption. As pointed out earlier,

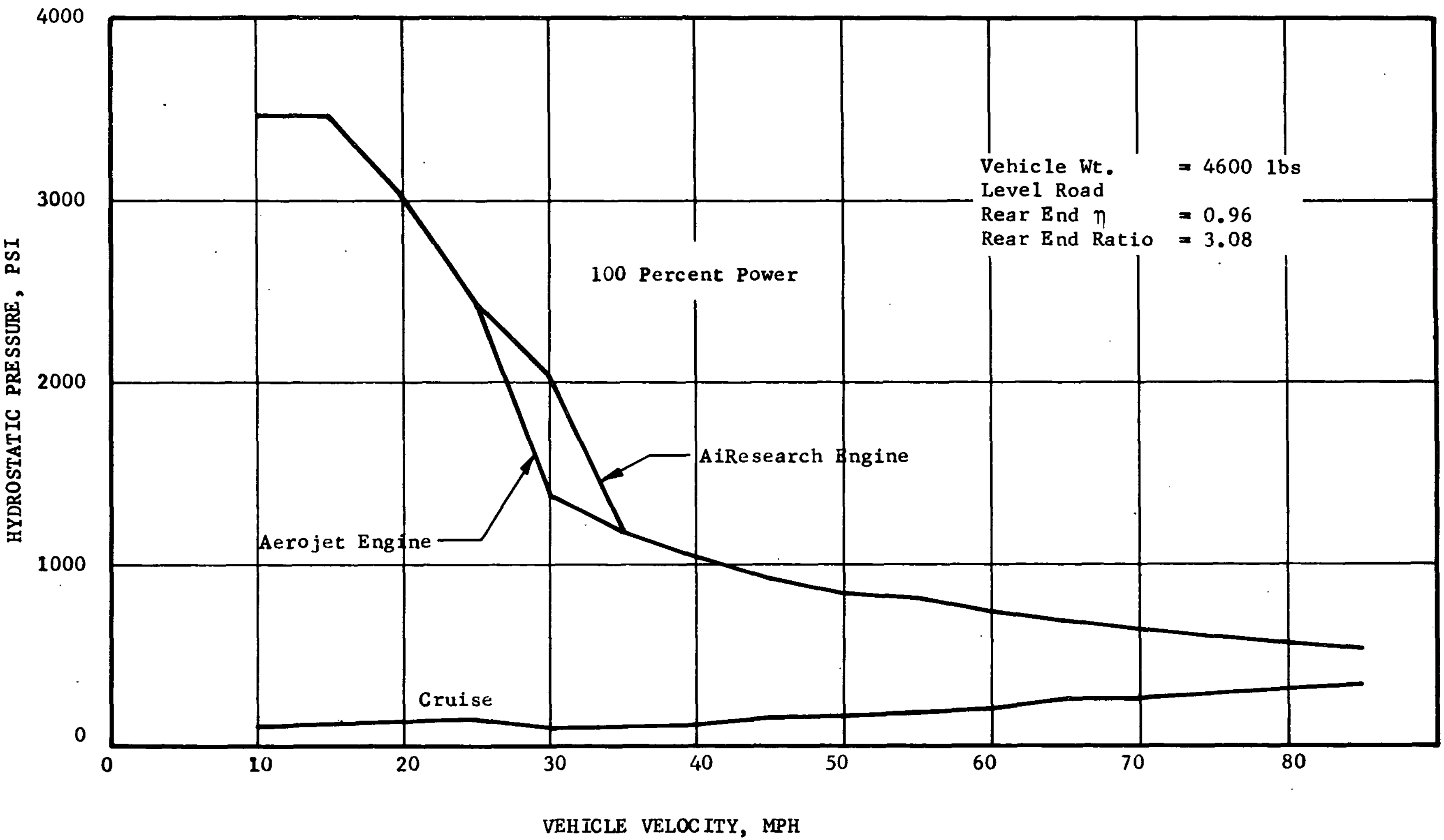

Fig. 6 Hydrostatic Pressures for Selected Transmission

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the transmission changes ratio (hence engine load) so that engine operation was maintained on a desired operating line except at large acceleration power levels. Under maximum power accelerations, the transmission automatically provides the largest necessary torque ratio.

Figure 7 presents the cruise-power fuel economy of the Aerojet engine with the selected transmission. Optimum fuel economy occured at 35 to 40 mph; 15.4 mpg with air conditioner and 17.4 without air conditioner.

A comparison in fuel economy (with air conditioner) to a torque-converter automatic transmission, with the Aerojet engine, is given by Figure 8 (details of the automatic transmission are given in Section V-D). The discontinuities in performance, associated with each shaft point for the automatic transmission are clearly shown. However, on an average basis it is apparent that, at low speeds, the fuel economy is slightly better with the automatic transmission, and at higher speeds the selected transmission produced slightly better fuel economy. The predominant reason for this similarity in power-train performance is the flat, symmetrical nature of the Aerojet engine performance map. Thus, typically, a 500 rpm deviation from the optimum engine speed, either up or down, causes only about 1 out of 17 deviation in engine efficiency. At low vehicle speeds the higher efficiency of the automatic transmission actually results in higher fuel economy when the engine is operating near its minimum SFC point.

Also shown by Figure 8, is the resultant fuel economy for an idealized situation where the selected transmission has 100 percent efficiency. By comparison, even with this perfect transmission and the Aerojet engine operating at maximum efficiency, 17.9 mpg is the best fuel economy - only 3 mpg or 20 percent better than the automatic transmission.

The Federal Driving Cycle provides an alternative operating condition to compare performance. These results for the Aerojet engine are given by Table 4, and show that the conventional automatic provides a small (4 percent) improvement in average fuel economy over the hydromechanical transmission. Thus, again the two transmissions are of similar benefit to the Aerojet engine. In this case a perfectly efficient hydromechanical could provide 21 percent better average fuel economy than the conventional automatic.

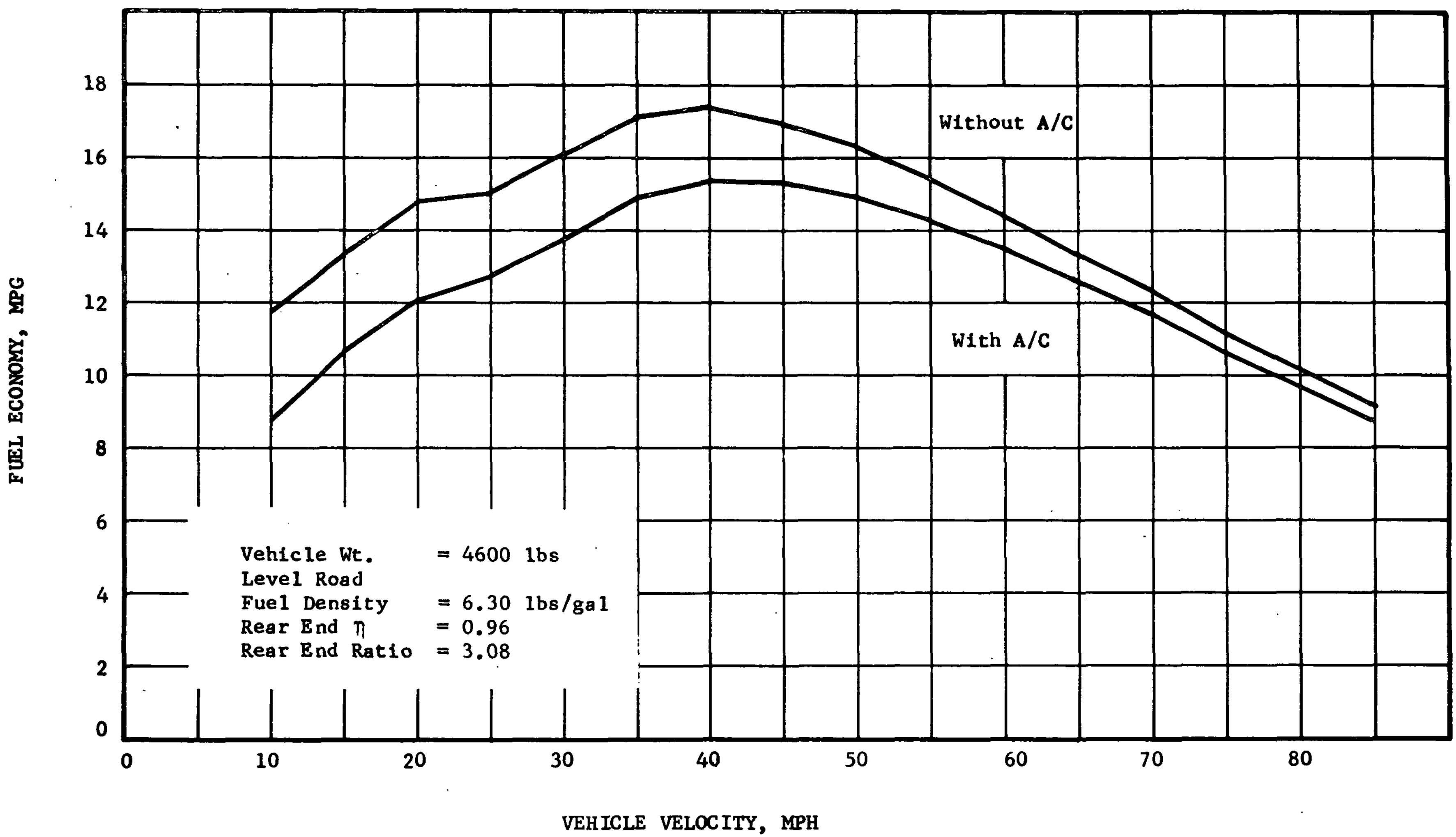

Fig. 7 Cruise Fuel Economy with Selected Transmission — Aerojet Engine

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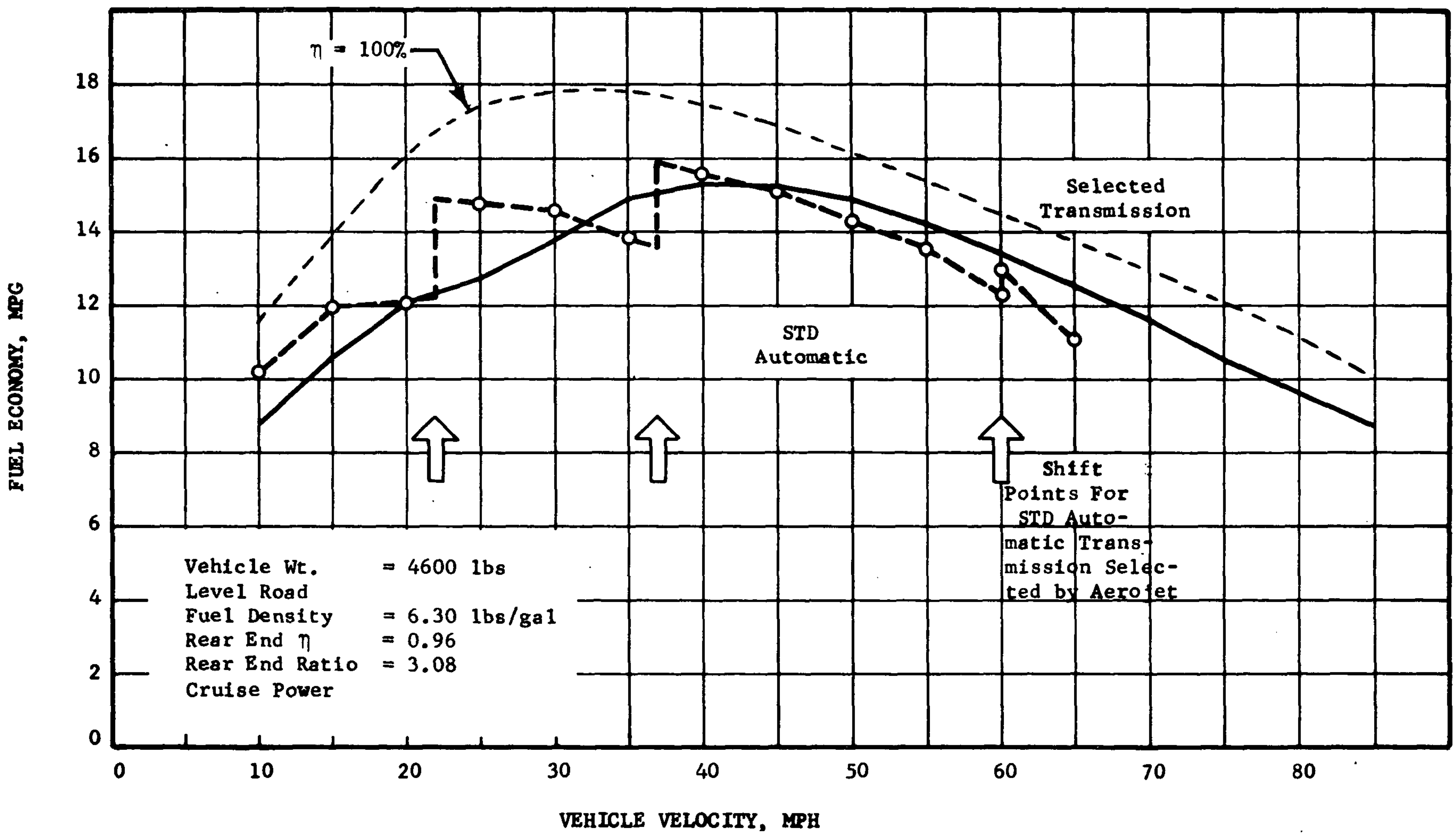

Fig. 8 Comparison of Fuel Economy — Aerojet Engine with Air Conditioner — Cruise Power

TABLE 4

AEROJET ENGINE — DRIVING-CYCLE PERFORMANCE

Quantity	Selected Transmission With A/C	Selected Transmission Without A/C	100% Transmission With A/C	Automatic* Transmission With A/C
Average MPG	9.55	10.85	12.02	9.9
Average Transmission	71%	71%	100%	-
Average Engine Power	16.76 hp	14.43 hp	13.16 hp	-
Average Road Power	8.38 hp	8.38 hp	8.38 hp	
Average Velocity	19.6 mph	19.6 mph	19.6 mph	

* Date provided by Aerojet (transmission efficiency engine power data not available)

Consider now the full-power acceleration performance of the selected transmission with the Aerojet engine. As shown by Table 5, the power train exceeds all of the EPA/AAPS maneuver specifications. The time history of vehicle velocity (Section V-D) shows smooth, stepless acceleration for the power train consisting of the Aerojet engine and the selected transmission.

TABLE 5

AEROJET MANEUVER PERFORMANCE

	EPA Specifications	Aerojet Engine with Selected Transmission
1. Distance traveled in 10 seconds	440 ft.	505 ft.
2. Time to reach 60 mph from standing start	13.5 sec	11.7 sec.
3. High speed merge (25-70 mph)	15.0 sec.	13.5 sec.
4. DOT passing maneuver (time and distance to overtake 50 mph truck)		
TIME	15.0 sec.	12.2 sec.
DISTANCE	1400 ft.	1166 ft.

3. Power-Train Performance - AiResearch Engine

Figure 9 presents the fuel economy of the AiResearch engine coupled with the selected transmission. Peak fuel economy was 28.3 mpg at 40 mph without air conditioner and 26 mpg with air conditioner.

The power-train performance of a vehicle incorporating the same automatic transmission as discussed for the Aerojet engine was computed. It is emphasized that the conventional automatic could never provide satisfactory kinematic performance wit the single shaft gas turbine but the comparison does provide an exagerated demonstration of the advantages of a continuously variable transmission to this type of engine. A comparison of fuel economy is shown by Figure 10. These results clearly show a considerable advantage in fuel economy with the selected transmission. The reason for the poor performance shown by the automatic transmission is the extreme sensitivity of the single-shaft, gas-turbine SFC to engine speed for a given power demand.

Performance over the Federal Driving Cycle for the AiResearch engine with the selected transmission is presented in Table 6. No data is avaiable for comparison with the conventional automatic. The average fuel economy values obtained with the selected transmission were 14.53 mpg with air conditioning and 15.76 without.

TABLE 6

DRIVING CYCLE PERFORMANCE — AIRESEARCH ENGINE

Quantity	Selected Transmission With A/C	Selected Transmission Without A/C
Average MPG	14.53	15.76
Average Transmission	74.4	73.5
Average Engine Power	15.94 hp	13.92
Average Road Power	8.38 hp	8.38 hp
Average Velocity	19.6 mph	19.6 mph

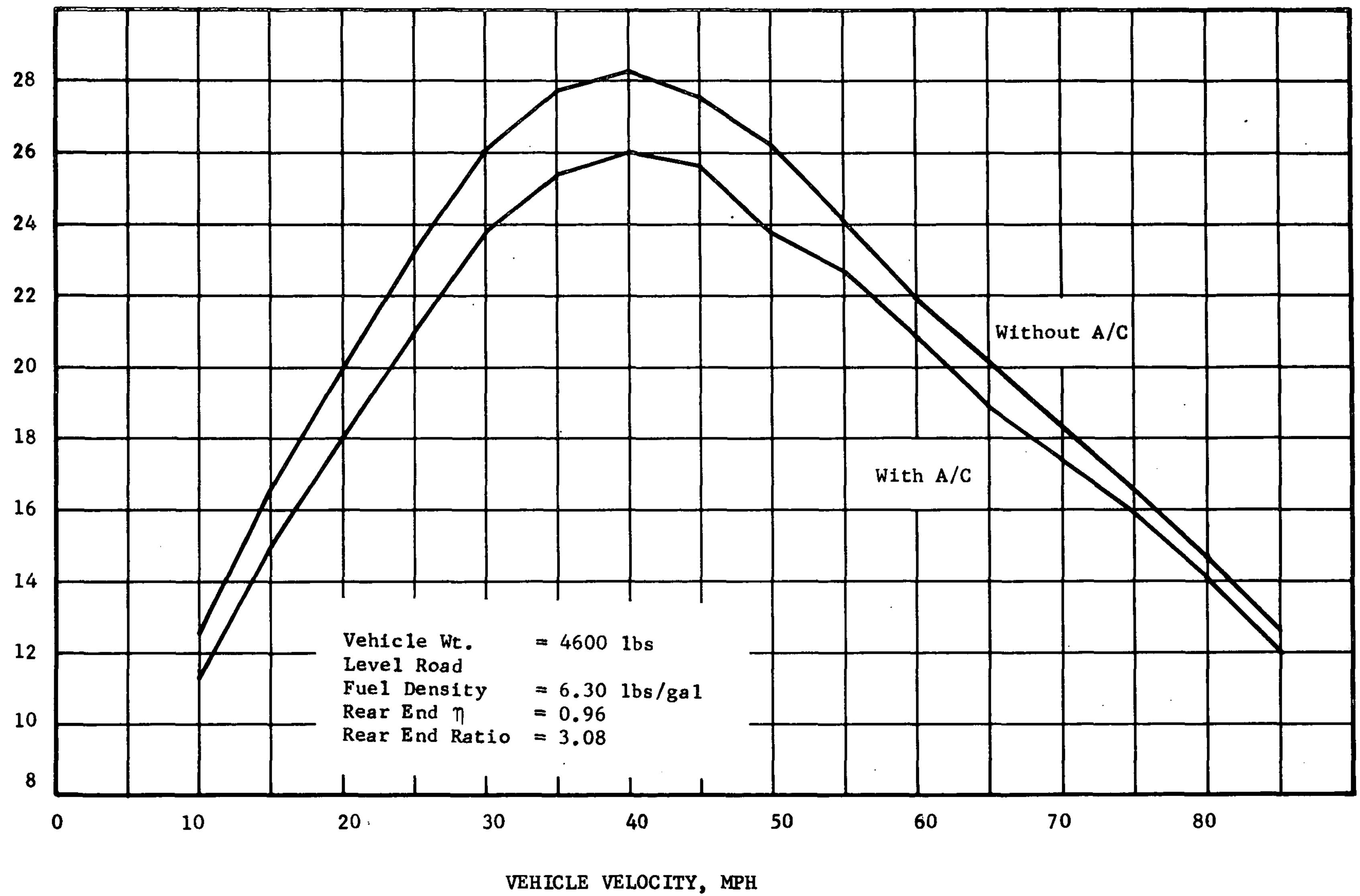

Fig. 9 Cruise Fuel Economy with Selected Transmission — AiResearch Engine

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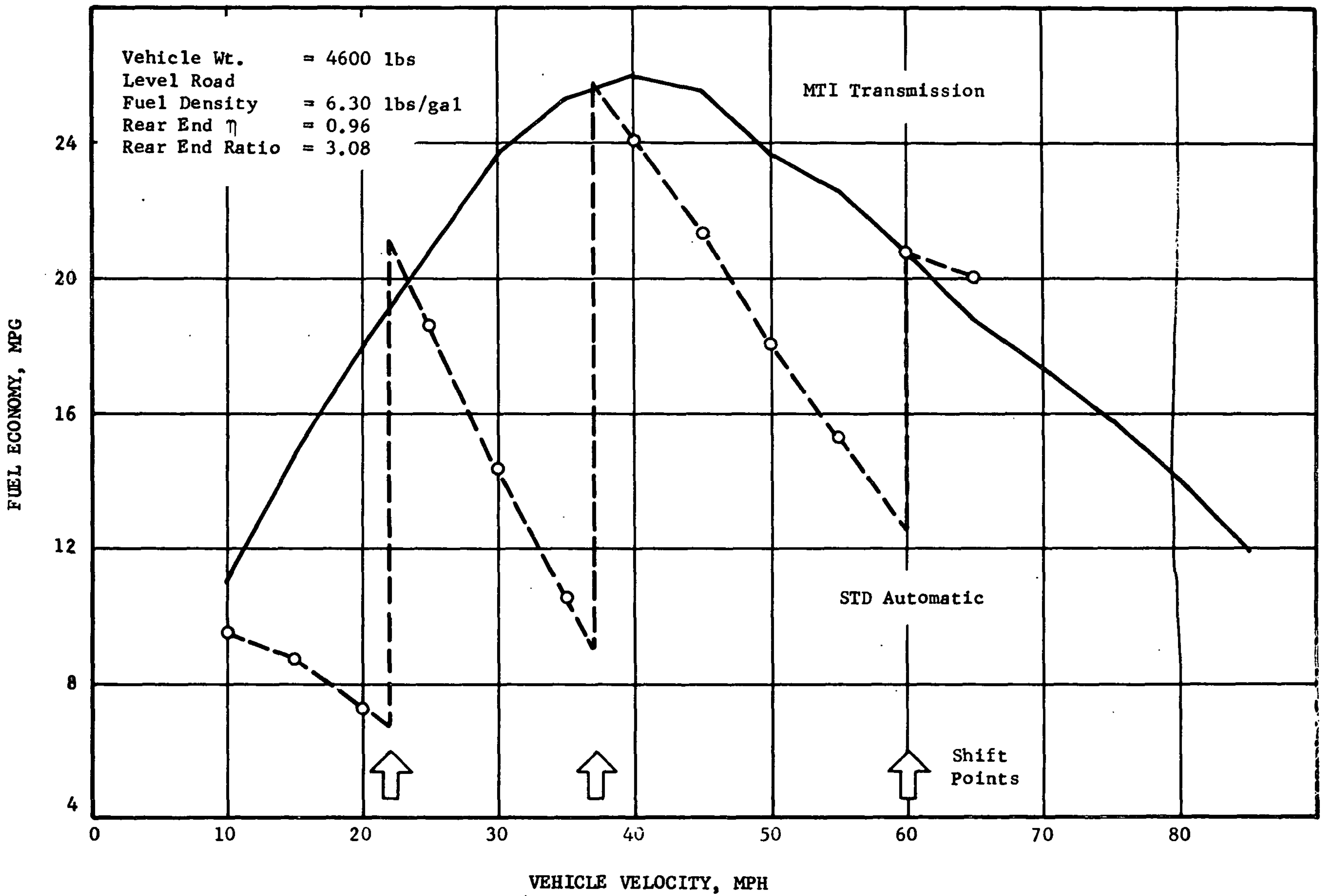


Fig. 10 Comparison of Fuel Economy — AiResearch Engine with Air Conditioner — Cruise Power

Full-power performance calculations showed that the AiResearch engine had to be scaled upward by 15 percent more power, to meet all of the EPA/AAPS specifications. The results are given in Table 7 below:

TABLE 7

AIRESEARCH MANEUVER PERFORMANCE

Maneuver	EPA Specifications	AiResearch Engine
1. Distance traveled in 10 seconds	440 ft.	447 ft.
2. Time to read 60 mph for standing effort	13.5 sec.	11.1 sec.
3. High speed merge (25-70 mph)	15.0 sec.	11.6 sec.
4. DOT passing maneuver (time and distance to overtake 50 mph truck)		
TIME	15.0 sec.	11.8 sec.
DISTANCE	1400 ft.	1139 ft.

As discussed in Section V-D, the primary reason for scaling up the engine power was in order to meet the requirement of 440 feet traveled in 10 seconds. This was caused by the inherently low starting torque characteristics of a single-shaft gas turbine.

The resulting time history of vehicle velocity (see Section V-D) showed smooth, stepless acceleration characteristics for the AiResearch engine when coupled to the selected transmission.

D. Cost and Physical Comparisons

One of the most important aspects of any transmission being considered for automotive applications is cost. Consequently, a detailed cost analysis was performed (refer to Section V-E for detail), using procedures currently practiced by the Ford Motor Company, in order to determine the production cost of the selected transmission. This analysis consisted of determining detailed cost estimates for approximately 200 separate parts. The resulting costs were then compared to the current cost of an automatic transmission for a medium-sized family car. Since the cost of the automatic transmission was based upon proprietary information of the Ford Motor Company, the results of the cost analysis are presented as ratios.

Detailed costs were determined on a "variable'cost" basis rather than an "original equipment manufacturer" (O.E.M.) basis; since this approach is more meaningful to the automotive industry in comparing designs. Variable cost includes: 1) purchased cost of a part, 2) direct labor required to get the part to a desired condition, 3) indirect labor associated with the manufacturing process, 4) variable overhead items which specifically relate to the manufacturing process, and 5) specific (programmed) overhead expenses, such as specific required testing.

The O.E.M. costs (specified by EPA/AAPS) were estimated from the aggregate variable cost. O.E.M. cost includes transfer costs such as capital investment, engineering development, facilities, etc. which are dependent upon management strategy decisions. Consequently, O.E.M. cost estimates were given a range to account for probable variations.

A summary of the cost analysis is presented in Table 8. These results show that, in production quantities of 1,000,000 units/year, the variable cost of the selected hydromechanical transmission will be 1.44 times the cost of a conventional three-speed automatic transmission - an increase in cost of 44 percent. On an O.E.M. cost basis, the increase ranged from 30 to 40 percent. For smaller production quantities (100,000 units/year) the increase in cost could be as high as 70-80 percent on an O.E.M. basis, when compared to larger production quantities of the automatic transmission.

Also shown in Table 8, is a breakdown in costs attributed to controls, labor, and material at production levels of 1,000,000 units/year. The selected transmission control cost increase was 28 percent, additional labor content 34 percent, and material content 53 percent. Thus, the major factor causing the increased costs was the material content of the transmission.

As subsequently discussed, the selected hydromechanical transmission weighs approximately the same as a conventional automatic transmission. Therefore, it is reasonable to believe that when the design and manufacturing skills of the automobile industry, which have been applied over many years to existing transmissions, are applied to the selected transmission, the cost will approach the cost/pound ratio of existing transmissions. This suggests that in the future the production cost of the selected transmission would approach the present cost of an automatic transmission.

Consider now a comparison of pertinent physical characteristics. As summarized in Table 9, the selected transmission design presented herein was 3 percent heavier, had 6 percent more parts, and required 28 percent less volume than a comparable three-speed automatic transmission. Figure 11 presents a comparison of volume envelopes which shows that the selected transmission is even smaller than a two-speed automatic transmission. Thus, it was concluded that the selected transmission was smaller but was slightly heavier than a comparable automatic transmission.

TABLE 8

TRANSMISSION COST ANALYSIS – COST RATIOS

	PRODUCTION LEVEL 1,000,000 UNITS PER YEAR		PRODUCTION LEVEL 100,000 UNITS PER YEAR	
	STANDARD AUTOMATIC TRANSMISSION WITH TORQUE CONVERTER*	POWER SPLITTING HYDROMECHANICAL TRANSMISSION	STANDARD AUTOMATIC TRANSMISSION WITH TORQUE CONVERTER*	POWER SPLITTING HYDROMECHANICAL TRANSMISSION
1. VARIABLE COST RATIO (TOTAL)	1.00*	1.44	1.29	1.86
a. CONTROL VARIABLE COST RATIO	1.00	1.28	1.29	1.65
b. LABOR CONTENT RATIO	1.00	1.34	1.50	2.02
c. MATERIAL CONTENT RATIO	1.00	1.53	1.20	1.84
2. OEM COST RATIO	1.00	1.30-1.40	1.25-1.35	1.70-1.80

*USED AS REFERENCE, PRODUCTION LEVEL OF 1,000,000 UNITS PER YEAR.

TABLE 9

TRANSMISSION PHYSICAL COMPARISONS

	POWER-SPLITTING TRANSMISSION	AUTOMATIC THREE SPEED WITH TORQUE CONVERTER TRANSMISSION	RATIO OF POWER-SPLITTING TO STANDARD AUTOMATIC THREE SPEED TRANSMISSION
TOTAL TRANSMISSION AND CONTROL VOLUME	1.10 FT^3	1.52 FT^3	.72
TRANSMISSION VOLUME – FT^3	1.00 FT^3	1.40 FT^3	.714
CONTROL VOLUME FT^3	.10 FT^3	.12 FT^3	.833
TOTAL TRANSMISSION AND CONTROL WEIGHT	146 LBS	142 LBS	1.03
TRANSMISSION WEIGHT – LBS.	130 LBS	123 LBS	1.057
CONTROL WEIGHT – LBS.	16 LBS	19 LBS	.842
TOTAL TRANSMISSION AND CONTROL PARTS	250	236	1.06
TRANSMISSION – NUMBER OF PARTS	175	166	1.054
CONTROL – NUMBER OF PARTS	75	70	1.071

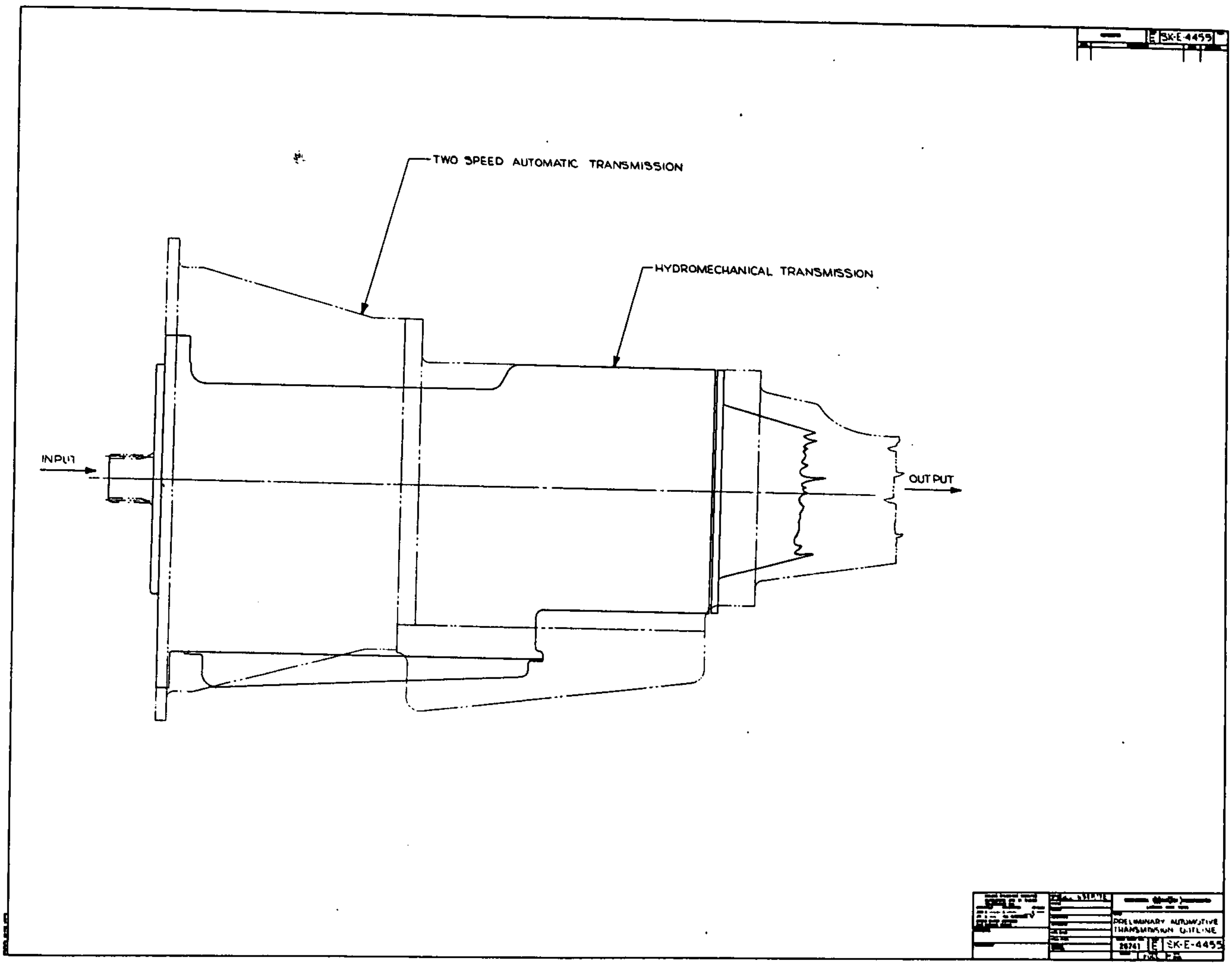


Fig. 11 Comparison of Transmission Envelopes

SECTION IV

CONCLUSIONS AND RECOMMENDATIONS

1. As a result of considering eight different types of transmissions, the variable ratio, power-splitting, hydromechanical transmission was selected as the most promising near-term (1974) transmission for application to both the single-shaft gas turbine and the turbo-Rankine engine. With future development, the traction type of transmission was considered a promising alternate on a long-term basis.

2. The selected transmission design was 28 percent smaller, 3 percent heavier and required only 6 percent more parts than a comparable conventional automatic transmission.

3. The selected transmission was compatible and feasible for both the single-shaft gas turbine and turbo-Rankine engines. Performance analysis demonstrated that the transmission maintained engine operation (SFC) along a prescribed operating line (for maximum miles/gallon) and the resulting smooth acceleration characteristics met all EPA/AAPS specified vehicle design goals.

4. Production cost is a key factor in considering a transmission for automotive applications. A detailed cost analysis, based upon costing procedures of the Ford Motor Company, showed that the "original equipment manufacture" (O.E.M.) cost of the selected transmission was 30 to 40 percent higher than a present day automatic transmission in production quantities of 1,000,000 units/year. On a "variable cost" basis (more meaningful to the automotive companies in comparing designs) the increase in cost was 44 percent. Higher material cost was the major factor in causing the increased cost. However, since the weight of the selected transmission was only slightly greater than a comparable automatic transmission, this suggests that, with development of production techniques, future production costs for such a transmission could approach that of a present day automatic transmission.

5. Detailed performance analysis showed that the resulting efficiency of the selected transmission at cruise power was 57 to 60 percent at 20 mph, 66 to 74 at 30 mph and 90 percent at 50 mph or higher speeds. Average transmission efficiency over the Federal Driving Cycle was 71 and 74 percent with the turbo-Rankine and single-shaft gas turbine, respectively. A comparable conventional automatic transmission (powered by a present-day automotive engine) has been estimated to have an average efficiency of 77.6 percent over the same driving cycle. Thus, it was concluded that the selected transmission had somewhat lower efficiency than the automatic transmission at low cruise-power levels (below 25 mph) and this was primarily caused by speed dependent losses.

6. The fuel economy calculated for the selected transmission with the single shaft gas turbine showed clear advantages relative to a conventional automatic. The automatic transmission is, of course, incapable of (a) following the critical minimum SFC line for the gas turbine and (b) providing satisfactory power to the wheels over the vehicle speed range required. The continuously variable transmission provides both capabilities very well.

7. The fuel economy calculated for the selected transmission with the Turbo-Rankine engine showed no advantage relative to a conventional automatic. Although the automatic cannot closely follow the minimum SFC line for this engine, there is no requirement that it should. The turbo-Rankine engine has relatively flat engine characteristics and neither fuel economy nor kinematic performance is sensitive to speed.

8. Although the selected transmission provided smooth acceleration and optimum engine operation for the turbo-Rankine engine, it was concluded that, since there was little improvement in fuel economy compared to a conventional automatic transmission, further development of the transmission for that engine application is not advantageous. However, if future turbo-Rankine engine design developments result in an engine characteristic where there is a significant change in SFC with engine speed, then development of the selected transmission for that engine would be beneficial.

9. Further development of the selected variable ratio, hydromechanical transmission is recommended for application with automotive single-shaft gas-turbine engines (and other similar types of engines) since, in addition to smooth vehicle operation, maximum fuel economy will be obtained.

SECTION V

SUPPORTING INFORMATION

A. SELECTION OF TRANSMISSION

The initial step in this study was to select, from a wide range of transmission types, the most suitable candidate transmission type for use with the specified single-shaft gas turbine and turbo-Rankine engine in a power train applicable to a medium-sized family car. One of the more important criteria, specified by EPA/AAPS, in the selection process was that the selected transmission should be available for engine testing early in 1974.

Included for consideration were the following seven basic types of transmissions:

- Mechanical
- Hydrostatic
- Combination of mechanical and hydrostatic
- Hydrokinetic
- Electrical
- Traction
- Belt Drive

As will be discussed, these and other variations were considered in reaching a point of establishing the techical and economic feasibility.

Some of the basic transmission types can be ruled out on a qualitative basis because of the inherent limitations which they have in this application. The mechanical gear type was ruled out, since it does not provide the infinitely variable ratio with stepless changes that are required for smooth, efficient operation. The hydrostatic and the electric transmission overcome this limitation. They can be considered to be similar in that they can provide an infinitely variable ratio with smooth operating characteristics. Both suffer from the limitations of high weight, relatively high cost and efficiency lower than that attainable with other transmission types. In the case of the electrical transmission, one often omitted consideration is the cost and weight of associated controls and power-conditioning equipment.

Once having eliminated these transmissions from active consideration, the above list of basic types was expanded to include the following:

Three-speed Automatic with Variable Element

Conventional Hydromechanical

Advanced Hydromechanical

Traction - TRACOR

Traction - Power Splitting

Traction - Composition Belt

Traction - Metal Belt

Three-speed Automatic with Aerodynamic Torque Converter - ROHR

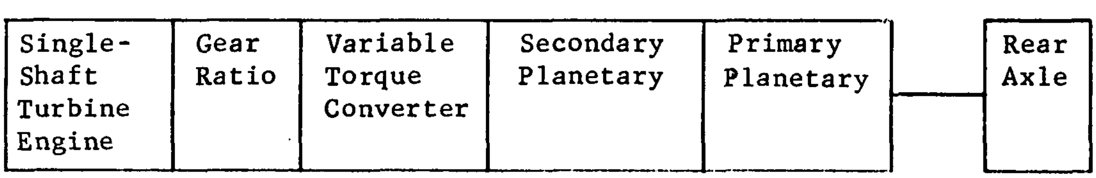
The standard three-speed automatic transmission was used in the latter part of the evaluation as the datum for performance comparisons. There is also merit in its consideration from the standpoint that it is widely used currently with the standard IC engine.

The candidate transmissions are described in the following paragraphs.

Transmission Descriptions

In this section each of the candidate transmissions is shown followed by a brief description.

1. Three-Speed Automatic Transmission with Variable Element



The standard three-speed automatic transmission is superior to a straight mechanical gear-type transmission. However, the steps or shift requirements still impose a penalty. The addition of a variable element would provide an infinitely variable ratio without steps. This basic arrangement is shown above.

A gear ratio is required ahead of the transmission to reduce engine output speed to approximately 4000 rpm which is a practical input speed for the torque converter.

A variable hydrodynamic torque converter makes use of a reactor element which is varied in position to extend the normal torque multiplication range of the standard hydrodynamic torque converter. The additional reactor element is simple and inexpensive. The impeller and turbine members are similar to those used in existing torque converters and thus adaptable to the same relatively simple manufacturing and assembly techniques.

The remainder of the construction is similar to the standard three-speed automatic transmission.

This configuration has the obvious advantage of maintaining the basic configuration of and the majority of the parts of the standard automatic, although an additional sub-assembly is required. Most particularly the poor efficiency of the variable torque converter element was considered to rule out this candidate.

2. Conventional Hydromechanical

<table>
<tr><td rowspan="2">Single-Shaft Turbine Engine</td><td rowspan="2">Gear Ratio</td><td>Gear Ratio</td><td>Variable Hydraulic Element</td><td>Variable Hydraulic Element</td><td>Gear Ratio</td><td>Low-Range Planetary</td><td rowspan="2">High-Range Planetary</td><td rowspan="2">Rear Axle</td></tr>
<tr><td colspan="5">High Range Mechanical Path.</td></tr>
</table>

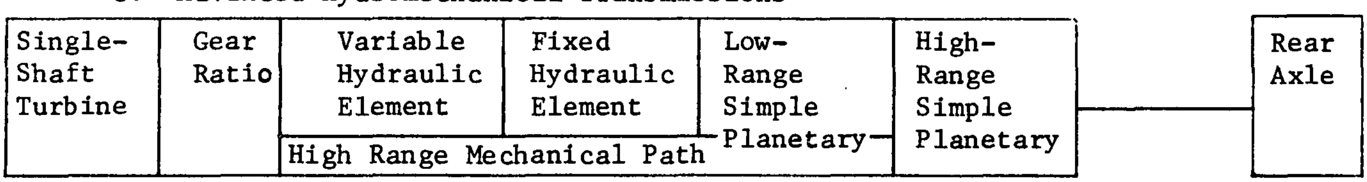
This transmission utilizes a simple hydrostatic pump-motor circuit in combination with a planetary gear train to provide a wide continuously variable operating range.

The design has two operating ranges. In the low-output-speed range, the hydraulic circuit operates as a straight hydrostatic system driving through the planetary gear set. This provides high output torque and variable operation in both the forward and reverse directions. In high range, the planetary gear set and hydrostatic circuit function as a split torque or hydromechanical system. This extends the range of output speed, increases the efficiency, and also provides positive control. The low range is achieved by braking one of the planetary elements. The high range is achieved by disengaging the brake and engaging a clutch which closes the high range mechanical path.

By utilizing a combination hydrostatic-hydromechanical construction, the range and pressure level over which the hydrostatic circuit must operate are minimized. This substantially reduces the displacement and size of the hydrostatic elements, which increases the efficiency and at the same time provides a smaller and more compact design.

In the design, the planetary gear train is constructed so that the clutch and brake elements are in synchronization during transition from one stage to the other. This eliminates high inertial loads on the clutch and brake, minimizes slippage and wear, and provides positive and even drive over the entire operating range.

3. Advanced Hydromechanical Transmissions



The advanced hydromechanical operates on basically the same principles described above for the conventional hydromechanical. However, differences to be observed are greater simplicity in design and controls, fewer parts, greater reliability and lower cost. Note in particular the elimination of the gear ratios at input to and output from the hydraulic elements.

The hydraulic units are operated at the reduced pressure of 3600 psi maximum and speed of 3500 rpm maximum to achieve several objectives.

Minimize noise levels which must be carefully considered in this type of design.

Achieve reliability by minimizing piston loading.

This design approach, however, does result in larger hydraulic units which impose some weight and cost penalties, and some reduction in efficiency.

The noise problem, while minimized by the reduced pressure, may require additional insulation or isolation to achieve driver acceptability.

4. Traction Drive - TRACOR

Single-Shaft Turbine Engine	Gear Ratio	Toroid Drive	Primary Planetary	Torque Converter	Secondary Planetary	—	Rear Axle

The TRACOR traction drive, which has progressed beyond the model stage, was considered as typical of all traction devices. A transmission schematic, shown above, was made and evaluated using the TRACOR traction drive.

The essential element of the traction transmission is a toroid drive which provides a continuously variable ratio by changing the relative radii at which power is delivered to and taken from a set of rotating disks. Fig. A-4 presents the important details of the mechanism.

The theoretically available range of torque ratio from input to output varies from 3:1 to 1:3 - a factor of 9 variation. TRACOR has proposed an automotive gas turbine transmission with a factor of 6 variation, which from the present studies would appear very adequate. Continuous variability over this ratio with the toroid drive results in a very high maximum output speed from the drive, and the implications of this high speed are discussed further below.

Since the toroid drive itself is more effective when operating at high speeds, the gear ratio shown ahead of the drive is a very simple low reduction gear set. It is very possible that this gear reduction could be eliminated entirely.

The TRACOR traction drive must be disconnected to change ratio when the output is at zero speed, and does not have internal provisions for reverse operation. A torque converter was added to allow disconnection of the traction drive from the rear end when at zero output speed. The torque converter provides an additional benefit of further increasing the available torque ratio range.

Since the TRACOR traction drive operates at a high speed, a primary planetary gear train to reduce the speed to that practical for the torque converter input was required. A second planetary gear set was added to provide transmission output at the appropriate speed and torque and to provide the reverse operation.

The advantages of the TRACOR transmission are the wide ratio range and relatively quiet operation. The disadvantages are the need for an additional planetary set and the torque converter, and the associated efficiency reduction. In addition, since a complete traction transmission has not been built for this power range, significant development effort is to be anticipated before this transmission type could be available for high production automotive applications.

5. Traction Drive - Power Splitting

Single-Shaft Turbine Engine	Gear Ratio	Toroid Drive / Clutch	Low-Range Simple Planetary	High-Range Simple Planetary		Rear Axle

During the transmission study, several purely conceptual transmissions were reviewed. One was the power-splitting traction transmission.

Since the existing larger size traction elements do not directly provide idle and reversing, an alternative scheme was generated which added this capability and also reduced the amount of power going through the traction drive. This reduction of power makes possible the use of a smaller traction element. The block diagram of such a transmission is shown above.

The operation of the transmission is conceived as very similar to that of the hydromechanical power-splitting transmission; that is, the traction units perform the same function as the hydraulic units.

In the low output horsepower and low-speed range, the traction circuit operates as a straight traction system driving through the planetary gear set. This provides high output torque and variable operation in both the forward and reverse directions. The clutch, which is shown in the diagram, is used during the idle periods and during the switching from forward to reverse. This clutch may not be necessary since several of the manufacturers of smaller traction units claim that their units have built-in features that permit reversing and operation with zero output speed.

In the high range the planetary gear set and traction drive function as a split torque system. By utilizing a combination of traction drives and planetaries, the range of power over which the traction drive must operate is minimized. With the proper combination of planetaries and traction drives, the power through the traction drive can be reduced to as low as 15 percent of the output power. Thus, smaller and much more versatile traction units can be used.

The remaining elements of the transmission are similar in operational and constructional features to the hydromechanical power-splitting transmission.

Since this is a conceptual transmission only, it should not be considered for near-future application. One disadvantage in common with the power split hydromechanical is the likelihood of more parts than a conventional automatic. However, the high-efficiency, high-speed operation and low-noise-level potential of the traction element do encourage its exploration for a transmission of the future.

6. Traction Drive - Composition Belt

Single-Shaft Turbine Engine	Gear Ratio	Clutch	Comp. Variable Belt Drive	Secondary Planetary	Primary Planetary	Rear Axle

Transmissions using composition belt drives as the variable elements have been used in many low-horsepower, off-the-road vehicles. Most of these drives have operated on vehicles requiring less than 50 horsepower. Several companies are now considering the belt transmission for automotive application. These studies are proprietary and were not made available for this review.

A possible approach is presented in the above diagram. Here the variable-speed belt drive is used as the hydraulic units are used in the hydro-mechanical transmission.

A gear train is required to reduce the gas-turbine output speed to 6,000 rpm, which is a practical speed for variable speed belt drives.

Existing variable-speed belt drives do not operate effectively at zero output speed (belts cannot be "slipped"); therefore, a clutch is provided to protect the belt drive during idle operations. As the variable-speed belt drives are developed for higher horsepower, the clutch may not be required or may be integral with the belt drive.

The scheme presented is simply a concept and has not been reduced to a design. Several of the areas that require exploration are:

1. The belt material and construction required to provide a life of 3500 hours. (The load, heat, speed and rate of speed change all tend to limit the life of existing belt transmissions.)
2. The packaging of the belt drives into an envelope suitable for automotive application.
3. The ability to achieve high efficiencies predicted by manufacturers of variable-speed belt drives, especially when operating at the extremes of the speed and torque range.
4. The probability of the composition belt acting as a traction drive and not a friction drive. (The wear associated with friction would severely limit the load capacity and life.)
5. Design approaches consistent with maintenance requirements.

One other approach would be to replace the torque converter of an existing three-speed automatic transmission with a variable-speed composition belt drive. Such a scheme would require that the belt drive handle more power than the power-splitting approach. The advantages of using most of the existing automotive transmission would make such an arrangement a worthwhile investigation.

Although there are many areas of the belt transmission that require exploration, the possibility of achieving transmission efficiencies of around 92% over a broad operating range suggests that a preliminary design study should be started.

7. Traction Drive - Metal Belt

Single-Shaft Turbine Engine	Gear Ratio	Metal Belt Variable Drive	Clutch	Secondary Planetary	Primary Planetary	Rear Axle

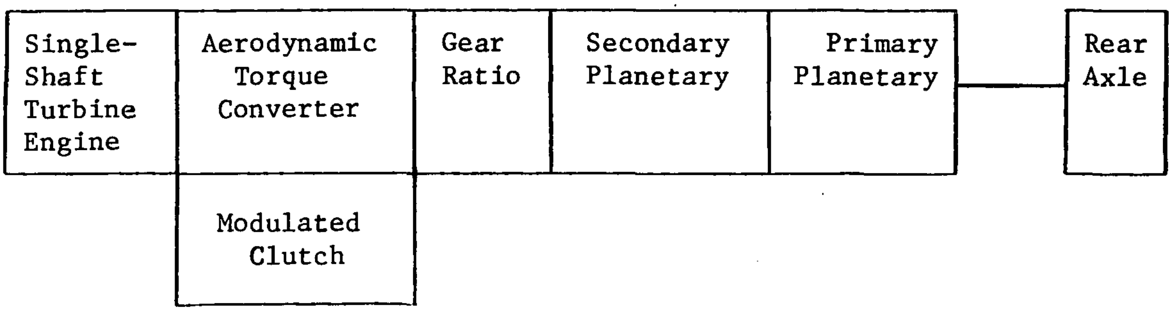
The composition-belt and metal-belt transmission are very similar in design. However, the metal belt can be considered a traction-type element and, in fact, several companies have built prototype transmissions using the belt as the traction element.

The diagram is an arrangement which proposes that the torque converter of an existing, three-speed, automatic transmission be replaced with a metal-belt variable drive. A clutch is provided for the idling mode when the output of the variable drive is at zero speed.

Many of the factors requiring exploration for the composition belt are not a problem with the metal belt. However, the packaging of a metal-belt drive into an envelope suitable for an automobile and the possibility of achieving the high efficiencies predicted by the manufacturers are two areas that have not been resolved.

The arrangement presented above is a concept and has not been reduced to a design. Therefore, additional study is required before a meaningful evaluation can be accomplished.

8. Three-Speed Automatic Transmission with Aerodynamic Torque Converter--ROHR INDUSTRIES, INCORPORATED



An automotive transmission for use with a gas-turbine engine has been proposed by Rohr Industries, Incorporated. A block diagram of the proposed transmission is shown above. The normal hydraulic torque converter used with existing automotive automatic transmissions has been replaced with an aerodynamic torque converter (ATC) as developed by Rohr Industries, Incorporated. Further, the ATC is shunted by a modulated clutch.

The ATC resembles the hydraulic torque converter in operation except it operates directly at the high speeds associated with the gas-turbine engine and does not require a geared reduction between the engine and the torque converter.

Since the ATC operates most effectively at high speeds, a gear reduction between the ATC output and the planetary gears is required. An additional reduction would be used to drive the required accessories at their normal operating speeds. To perform adequately throughout all vehicle modes at good transmission efficiency, the ATC is shunted by a modulated clutch. By judicious pressurization of the ATC in combination with engagement of the modulated clutch, satisfactory performance is envisioned to be achieved. The secondary and primary planetaries are similar to those used in automotive power-shift transmissions. They provide the additional torque ratio required and the reversing mode of operation.

In considering a gas-turbine engine, there are several advantages when using such a transmission:

1. The ATC can operate quite efficiently and effectively at the high speeds of a gas turbine.

2. The ATC allows the gas turbine to operate at the most efficient speeds required to satisfy the road load.

3. The ability of the ATC to regenerate its losses to the gas turbine by return of coolant flow poses the possibility of recovering some of the transmission losses.

4. The components all use existing mass production techniques.

Also, there are several questions that must be considered:

1. When considering system overall efficiency throughout the driving cycle, would a free power turbine and a three-speed shift transmission be more efficient than the ATC transmission?

2. Can an ATC transmission be arranged which would not require an evacuation pump, modulated clutch, additional reduction gears, and high speed seals; thereby, eliminating the increases in weight and loss in reliability and efficiency associated with such components?

3. Can the control of the ATC and return of coolant flow be accomplished without additional complexity?

Although the initial development of the ATC is complete, the development of a transmission using the ATC has not started. Therefore, such a transmission should not be considered as a possibility for application during the 1974 to 1975 period.

Selection Procedure

The first step in the selection was to define the evaluation factors. These are given in Table A-1 below:

TABLE A-1

EVALUATION FACTORS

1. Life and Reliability.	a. Number of parts
	b. Application of parts
	c. Design Simplicity
2. Noise and Smoothness	
3. Cost.	a. Number of parts
	b. Size
	c. Weight
	d. Manufacturing precision required
4. Development Status.	a. Paper Study
	b. Complete Transmission Built & Tested
	c. In production
5. Efficiency	
6. Size and Weight	
7. Restriction on Turbine Engine.	a. Performance capability

(Table A-1 continued)

8. Control Complexity

9. Drive Acceptability

10. Environmental Restrictions. . . . . a. Shock
b. Temperature

In applying these factors, it was important to bear in mind the short-term implementation requirement (1974), which makes factor 4, Development Status, one of major importance.

Although many of the transmissions must necessarily be treated somewhat qualitatively and subjectively, it was important that some objective procedure be adopted for making the final comparisons and selection using the factors shown in Table A-1 . Table A-2 shows an unweighted overall evaluation summary of these transmissions in which a score of 1-10 has been assigned for each factor. It will be noted from Table A-2 that the three-speed automatic with a variable element, the advanced hydromechanical and the conventional hydromechanical rated very close to each other.

These transmissions were reviewed to determine their major strengths and deficiencies. Table A-3 gives a short synopsis of the main advantage and disadvantage of each of the transmissions.

Discussion

The evaluation of transmission types previously discussed shows that the hydromechanical and traction-type transmissions were most worthy of consideration. It is appropriate, therefore, to present some additional comparisons between these two basic approaches.

TABLE A-2

EVALUATION SUMMARY

TRANSMISSION TYPE	OVERALL RATING* (Unweighted)
1. Three-speed Automatic with Variable Element	60
2. Advanced Hydromechanical	63
3. Conventional Hydromechanical	58
4. Traction-TRACOR	48
5. Traction-Power Splitting	35
6. Traction-Composition Belt	43
7. Traction-Metal Belt	32
8. Three-speed Automatic with Aerodynamic Torque Converter Rohr	28
9. Electric - Alternator and Motor	21

TABLE A-3

EVALUATION SUMMARY

TRANSMISSION TYPE	MAIN ADVANTAGE	MAIN DISADVANTAGE
1. THREE-SPEED AUTOMATIC WITH VARIABLE ELEMENT	SIMILAR COMPONENTS TO EXISTING AUTOMATIC TRANSMISSION	LOWER EFFICIENCY THEN EXISTING AUTOMATIC
2. ADVANCED HYDROMECHANICAL	SIMPLE MECHANICAL CONSTRUCTION	PRODUCTION TECHNIQUES WILL HAVE TO BE DEVELOPED
3. CONVENTIONAL HYDROMECHANICAL	DEVELOPMENT EXPERIENCE EXISTING	PRODUCTION TECHNIQUES WILL HAVE TO BE DEVELOPED
4. TRACTION – TRACOR	FUTURE POTENTIAL FOR HIGH EFFICIENCY	NOT TO DEVELOPMENT STAGE
5. TRACTION – POWER SPLITTING	FUTURE POTENTIAL FOR HIGH EFFICIENCY	NOT TO DESIGN STAGE
6. FRICTION – COMPOSITION BELT	BELT MANUFACTURING TECHNIQUES DEVELOPED	LIFE AND EFFICIENCY NOT FULLY DETERMINED
7. TRACTION – METAL BELT	FUTURE POTENTIAL FOR HIGH EFFICIENCY	NOT TO DEVELOPMENT STAGE
8. THREE-SPEED AUTOMATIC WITH AERODYNAMIC TORQUE CONVERTER–ROHR	MANUFACTURING TECHNIQUES DEVELOPED FOR BULK OF THE TRANSMISSION	NOT TO COMPLETE TRANSMISSION DESIGN STAGE

Several different kinds of traction devices were studied. Some of these claimed:

1. Infinitely variable ratio.
2. Ratio change with the output member at zero speed (eliminating need for clutch or torque converter).
3. Continuous operation through idle conditions (eliminating need for special reversing element).

These drives were, at most, only to the very preliminary small-size-model stage. As a result, they could not be evaluated for the present study. Therefore, the TRACOR traction drive, which has progressed beyond the model stage, was considered as typical of all traction devices.

Performance data for the traction drive was provided by the TRACOR Company and is presented in Figure A-1. Using the traction drive data, MTI estimated the performance of a traction transmission such as that shown in the diagram on page A-5. The performance data calculated in this manner is shown in Figure A-2.

Figure A-3 compares the performance with a comparable hydromechanical transmission. The results indicate that the efficiency of the hydromechanical is superior, particularly at lower power levels. It should be noted that this is largely due to the high-speed ratios and the use of a torque converter with this traction transmission concept.

Selected Transmission

Based upon these reviews, the hydromechanical (power-splitting) transmission was selected. The key features of the selected transmission are:

- Simple Construction
- Proven Components
- Low Cost Potential
- Minimal Development
- Size and Weight Compatible

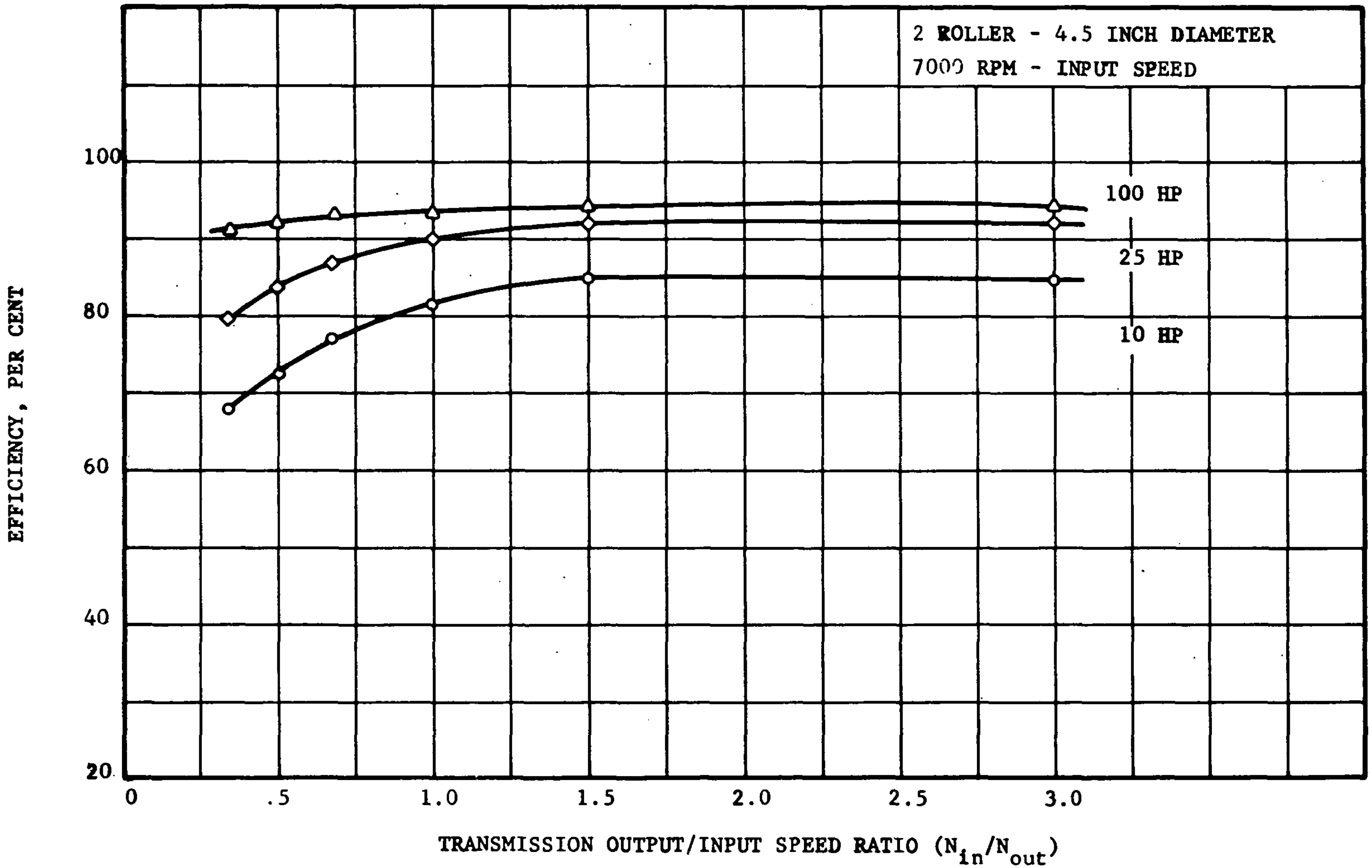


Fig. A-1 TRACOR Traction Drive Performance Data

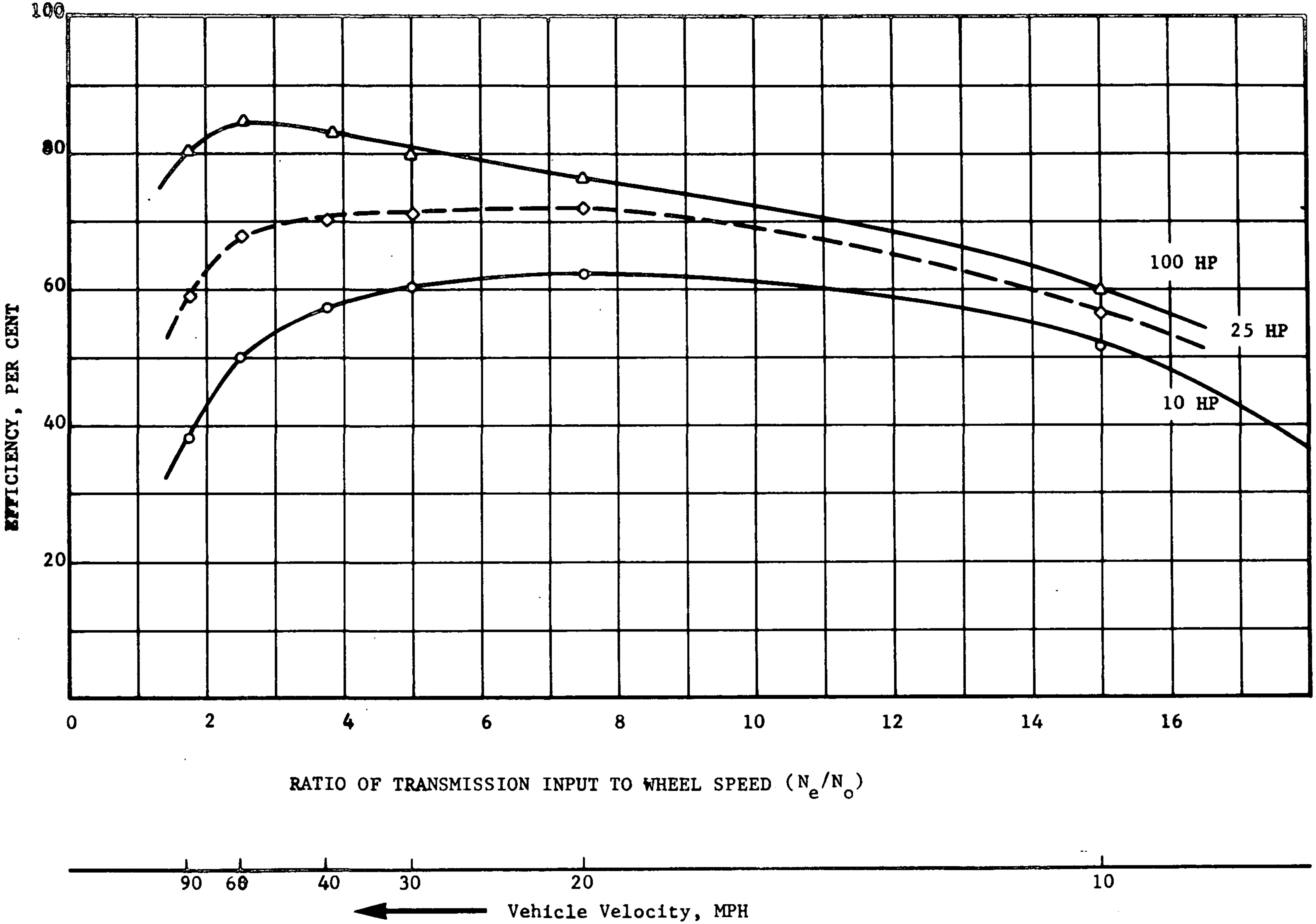


Fig. A-2 Estimate of Traction Transmission Performance

MTI-14661

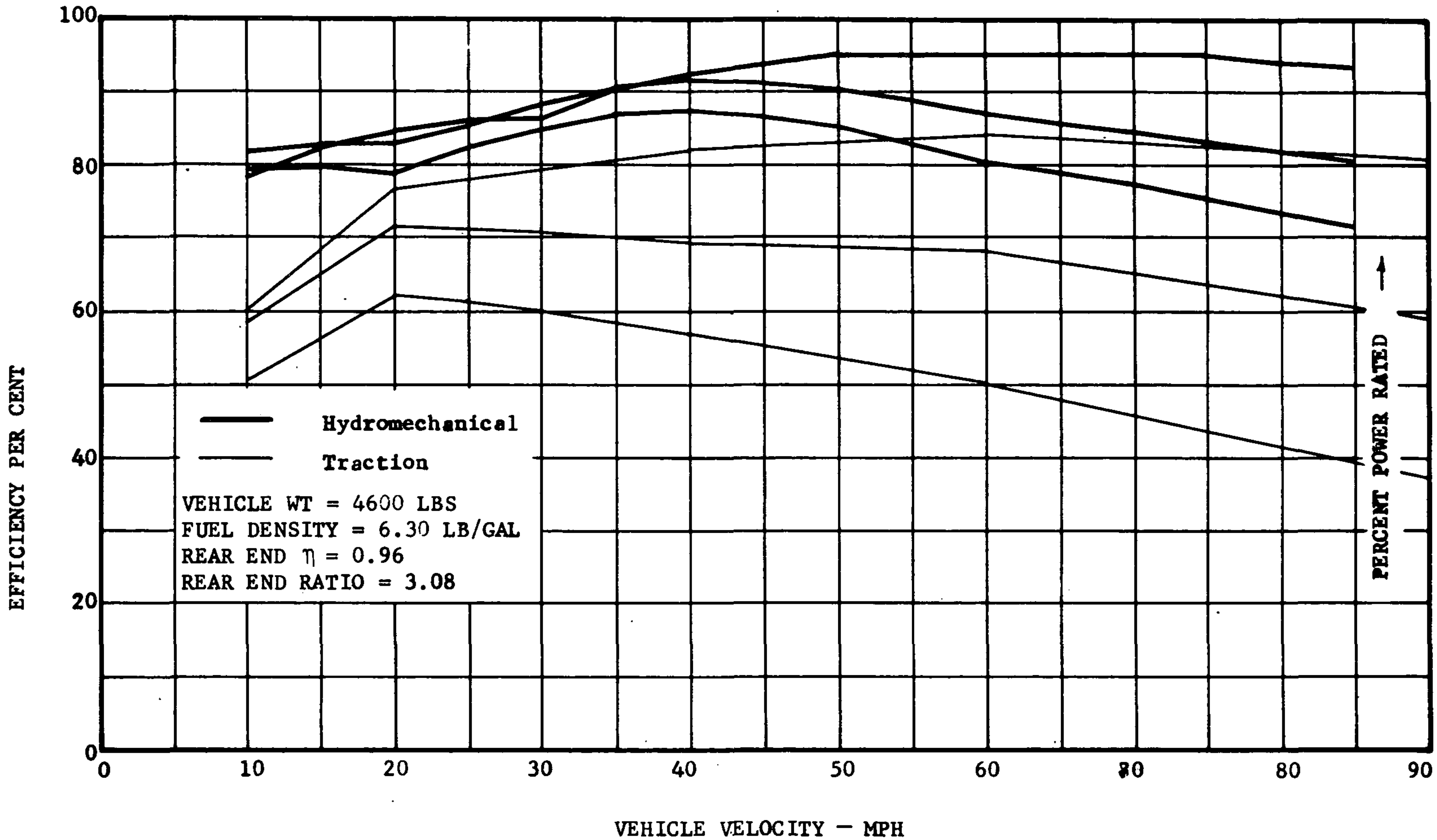


Fig. A-3 Comparison of Hydromechanical and Traction Transmissions

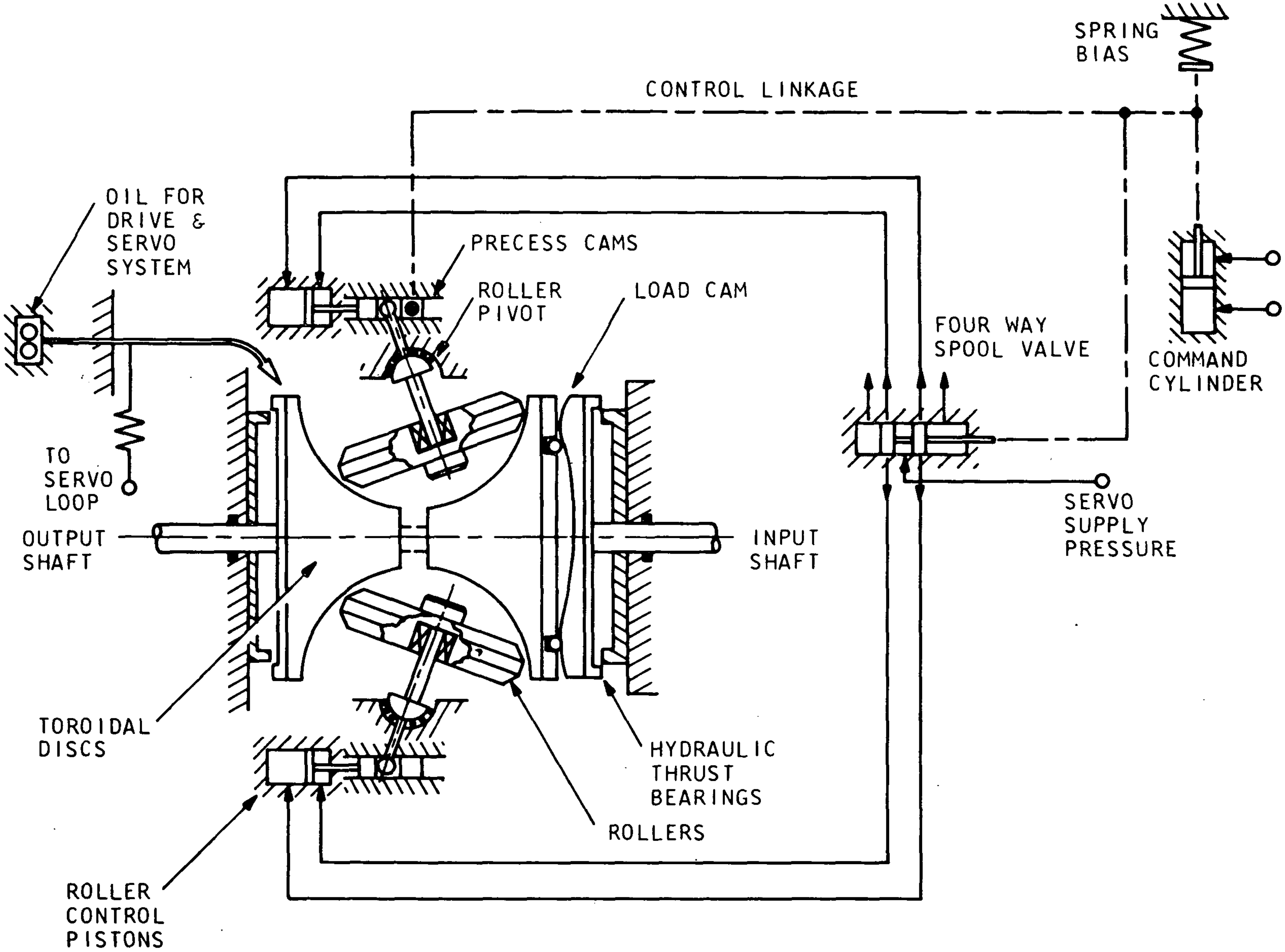


Fig. A-4 Traction Drive Schematic

B. DESCRIPTION OF TRANSMISSION & CONTROLS

This section contains a description of the selected hydromechanical transmission and its associated controls. Design details have been omitted, since the transmission is based upon a proprietary design of Mr. George DeLalio. In the following discussion, some of the features of the transmission are described with a subsequent discussion of the controls.

The selected hydromechanical transmission is an infinitely variable, stepless unit that obtains torque multiplication and control by means of hydraulic elements (pump-motor combination). The unit differs from the conventional torque converter or fluid-coupling-type transmission in that the hydraulic power is transferred by fluid static pressure at low flow as contrasted to the high dynamic action of fluid as utilized in hydrodynamic units. It also differs from a purely hydrostatic transmission in that the much more efficient mechanical elements transmit a significant portion of the power. It is a "hard" type of drive in that slip is less than 2 percent under full load.

1. Description of Transmission

As pointed out earlier in this report, the design of the selected hydromechanical transmission was based upon minimizing the number of mechanical parts in order to: a) make the transmission as simple as possible, b) keep the size, weight and, particularly, the cost low, c) require the minimum amount of development, and d) achieve high reliability while retaining a reasonably high transmission efficiency. Additional gear trains and hydraulic functions could have been employed to increase the capabilities of the transmission and reduce the amount of power in the hydraulic elements; however, it was considered more realistic to concentrate on a unique, straight-forward simple design in order to minimize development time.

Figure B-1 is a functional schematic of the selected transmission. As shown on this figure the transmission consists of an engine input shaft, and an output carrier and shaft. Other essential components are a brake for locking one element of the planetary to achieve low range operation; a clutch for closing the high speed mechanical power path; and a control system for varying the displacements of the elements, for engaging, and for shifting range.

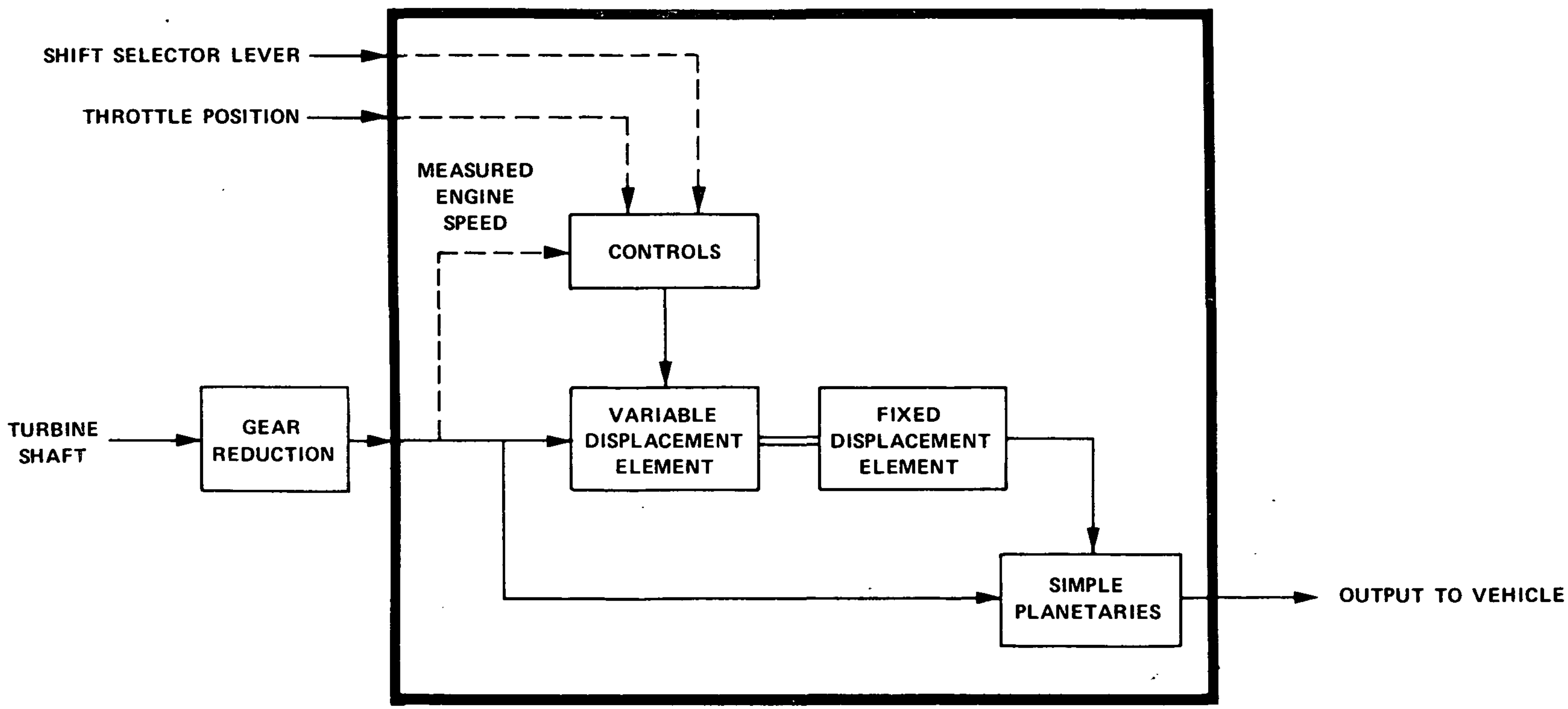


Fig. B-1 Functional Schematic — Selected Hydromechanical Variable-Ratio Transmission

MTI-14628

As shown by Figure B-1, the power path between input and output splits between hydrostatic and straight mechanical. The transmission has two operating ranges designed as low-range and high-range. At low power and output speed levels (up to approximately 0.4 of maximum output speed) the transmission operates in the low range and all power flows through the hydraulic elements. At higher power and speed conditions the transmission operates in high range and the power path is split between the hydraulic elements and the mechanical path. At approximately 0.7 of maximum output speed, all the power flow is through the mechanical path.

An important feature of this design is the synchronous shift which operates as follows: The transmission has two operating ranges, low speed and high speed. Transfer between these two ranges is effected by the concurrent opening of a brake and closing of a clutch. The shift is ideally synchronous if it occurs under the following conditions:

1. In either low or high range the displacement of the variable hydraulic element is at the same maximum point.
2. There is no change in relative velocity between the two sides of the clutch.
3. The braked element is stationary whether the brake is applied or not.

In practice such effects as slip between hydraulic elements and control imperfections cause slight deviations from the ideally synchronous shift. Even so there is never a significant change in momentum demanded of the rotating parts and the motor and clutch elemnts suffer little slippage or wear.

The introduction of the Mechanical connection reduces the range over which the hydraulic elements must operate at 100 percent power level to approximately 40 percent of the total speed range. Since the mechanical path elements are more efficient, compact, and less costly to produce,.this approach provides an optimum wherein the transmission is completely variable over its range, the size of the hydraulic elements is minimized, the mechanical construction is kept simple, and the range change is synchronous without any steps, slippage or wear of brake and clutch elements.

The maximum torque ratio available when the power path is through the hydraulic branch is 5.0 from engine input to transmission output and 15.4 from engine input to rear-axle output when a standard rear-axle ratio of 3.08:1 is used.

The maximum torque ratio available when the power path is through the mechanical branch is 2.5 from engine input to transmission output and is 7.7 from engine input to rear-axle output.

In the hydraulic units, the movement of the swashplate, which controls the pump displacement, is a continuous function; therefore, the transmission ratio control is completely stepless and infinitely variable within the operating range.

A preliminary design layout of the transmission was made in sufficient detail in order to determine size, weight, number of parts, and cost. Figure B-2 presents the design layout without the proprietary design details.

In the design, all the gears are made of automotive gear materials such as forged and surface hardened AISI 8620 steel and are machined to an AGMA gear quality level of 8 with a final polish. The manufacturing is accomplished using automotive practices.

The two hydraulic elements are based upon a design originated by Dr. H. Ebert* and recommended by Mr. G. DeLalio. They are considered to be of a high power-density construction which produces the maximum capacity using the minimum volume and weight. Typical power densities are approximately 20 horsepower per cubic inch. The construction of the unit is shown by Figures B-3 and B-4. The compact size of these hydraulic elements is achieved by using a rolling-element bearing for the swashplate bearing** and close coupling it to the drum.

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* Independent consultant. Dr. Ebert formally supervised the development of hydromechanical transmissions at Daimler-Benz, Austin, NSU and Allgaier in Germany.

** The design of the swashplate bearing was established by many hours of testing in Germany and at the Stratos Division of the Fairchild Corporation.

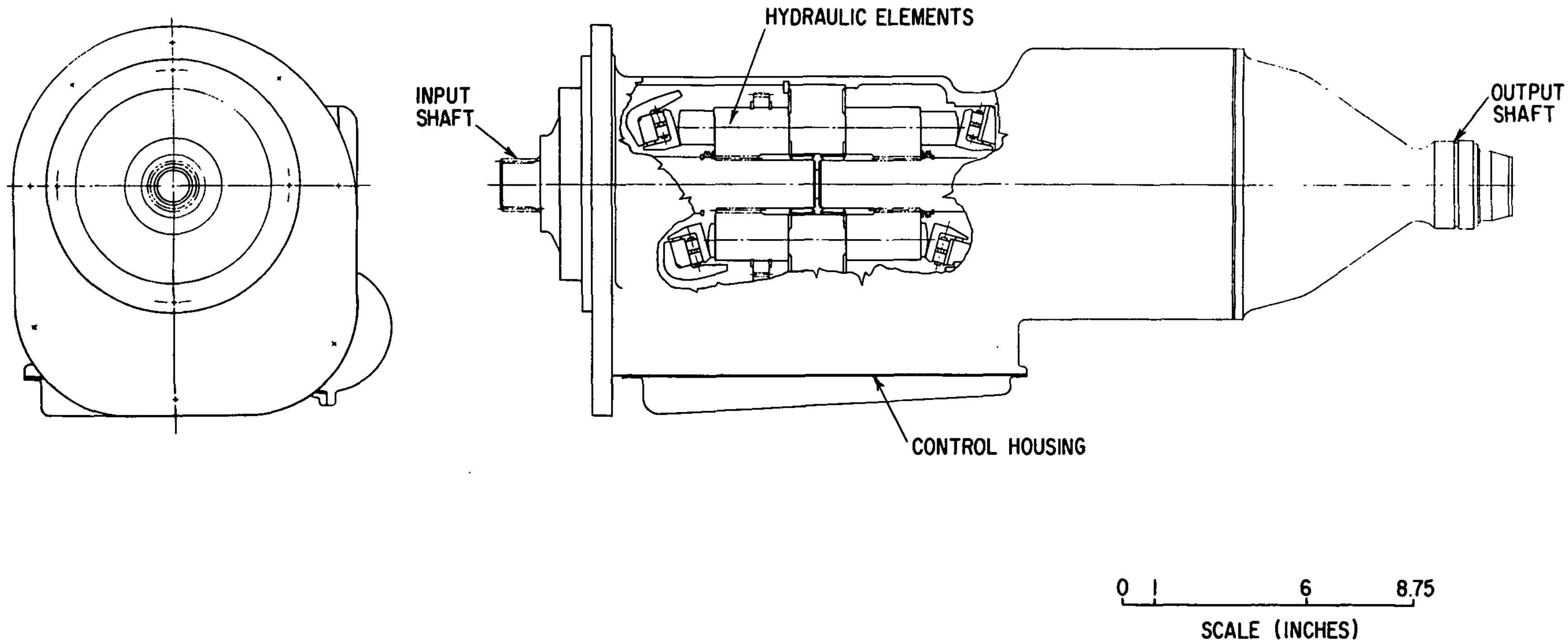


Fig. B-2 Preliminary Design Layout of the Hydromechanical Transmission

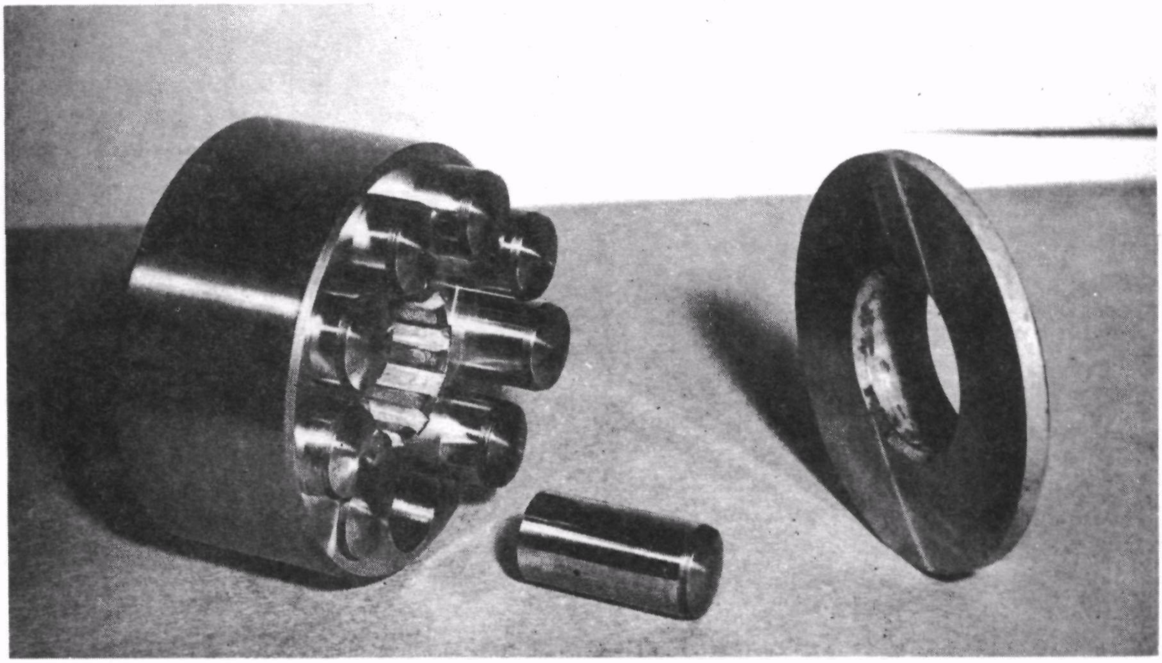


Fig. B-3 Typical Axial Piston Hydraulic Element

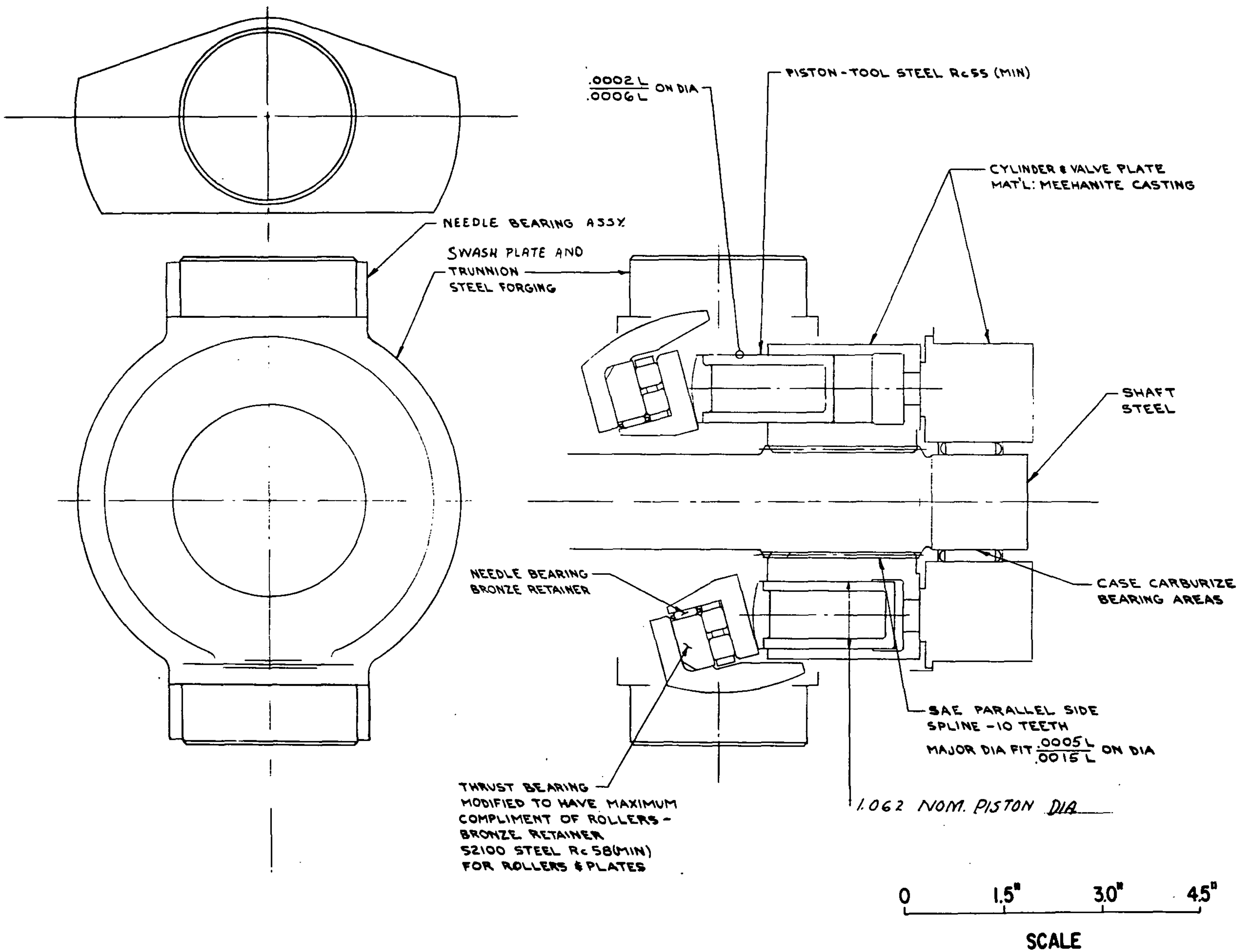


Fig. B-4 Drawing of High-Density Hydraulic Element (SK-C-4266)

MTI-12520

This type of hydraulic element has been successfully used in selected applications for aircraft constant-speed drives, tractor transmissions, truck transmissions and in postal vehicle transmissions. For example, these types of hydraulic elements were specifically used in the following programs conducted by the Stratos Division of the Fairchild Corporation:

1. 25-horsepower, constant-speed drive electric supply and hydraulic pump for Fairchild Goose Missile.

2. 200-horsepower hydromechanical transmission for trucks evaluated by Detroit Arsenal.

3. 150-horsepower hydromechanical transmission for M-34 trucks evaluated by Detroit Arsenal under Contract DA-30-069-ORD-2340.

4. 50-horsepower hydrostatic transmission for off-road vehicles evaluated by Detroit Arsenal.

Tests conducted in various applications of the hydraulic elements have established the requirements that must be met to achieve long life. These results have shown that, for the7.5 in ^{3}/rev size element, the system should operate at pressures of 2500 psi or lower and speeds of 3000 rpm or lower for 90% of the load schedule if a life greater than 3500 hours is expected. Pressures to 3500 psi and speeds of 350 rpm for 10% of the load schedule will not reduce the life of the elements below the 3500 hours.

With regard to structural aspects of the transmission, the main housings, control housings and mounting plates will all be aluminum pressure die castings. Automotive practices of thin-wall design, intricate sections for less machining, high strength, maximum heat disipation and favorable economics will be followed.

The high-range clutch and low-range brake are of conventional automotive construction used in existing transmissions.

The automotive practice of using sleeve bearings or needle bearings to support the radial and thrust loads has been followed.

As in the case of the gears, the shafts were constructed using a forged steel similar to AISI 8620. The bearing raceways and splines are hardened.

The displacement of the hydraulic unit is varied using a piston actuator which moves a cam plate linked to the trunnion and swashplate of the element. The actuator is similar in construction to that of the automobile transmission actuators used to engage clutches and brakes. The cam plate is steel with hardened cam tracks. A cam roller bearing is used to link the cam to the trunnion.

2. Description of Transmission Controls

A control hardware implementation schematic is shown in Figure B-5.

The basic components are:

- Shift lever
- Control cams
- Engine regulator governor
- Engage governor
- Control valve
- Spool valve
- Shuttle valve
- Clutch valve
- Element I swashplate cam
- Swashplate piston actuator
- Engaging valve

The manual shift lever is used to select the operating mode. The shift lever, through control cams, acts directly on the control valve, spool valve and regulator governor. A bias is applied to the regulator governor spring while the control valve and spool valve, which are positioned by the shift lever, direct pressures to the control elements as required for each mode.

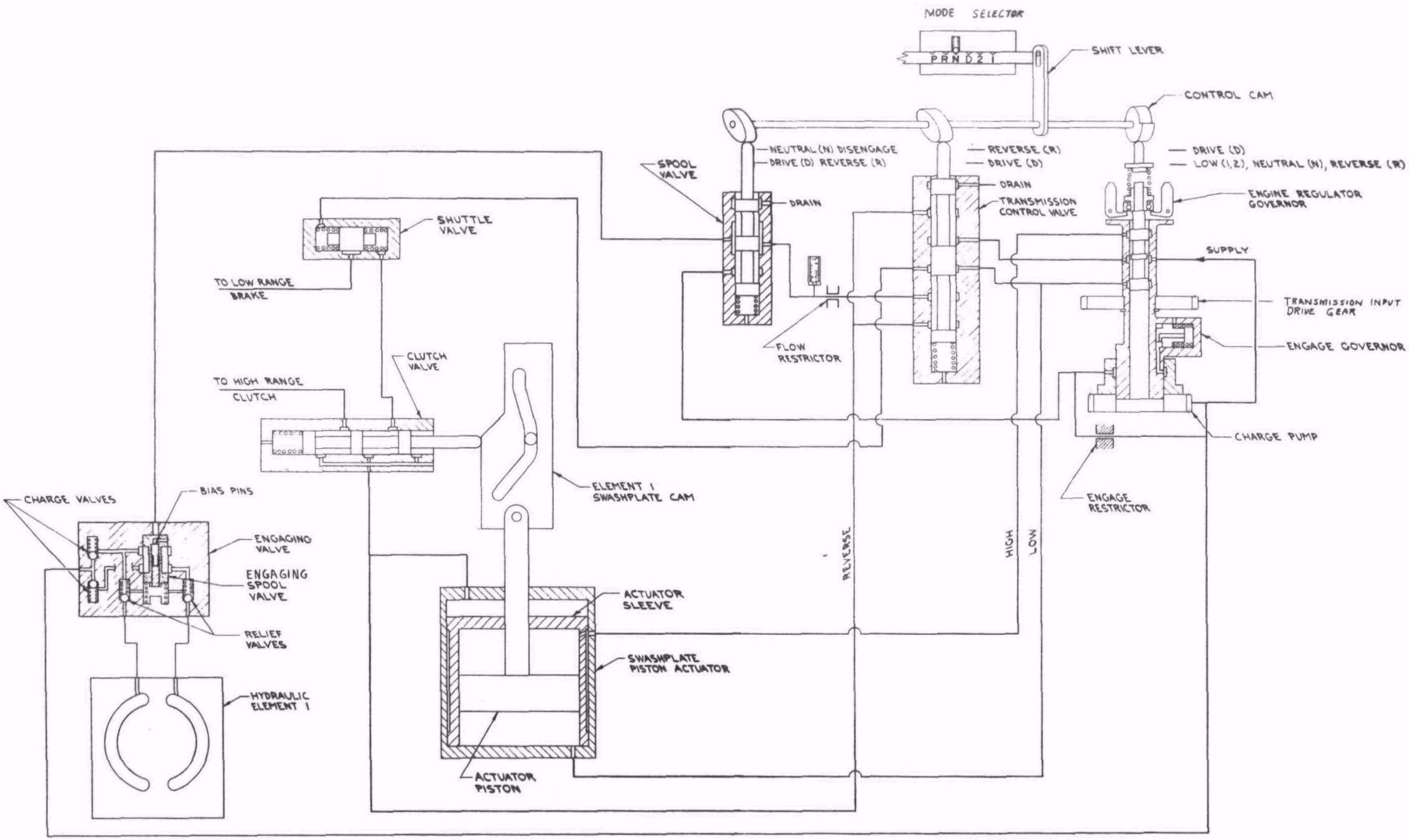


Fig. B-5 Hydromechanical Transmission Control Schematic

The engaging valve has several functions. It is a charge valve for the pump and motor, a bypass valve for the pump, and is also a fast-acting relief valve for the pump and motor.

The bypass function is controlled by the engage governor. The governor is geared to the transmission input shaft to sense engine RPM and give a positive neutral. When the engine RPM is too low, the governor causes the valve to remain in the position to bypass. The bias pins and spring react against the housing to open the engaging spool valve. As the speed of the engine and the engage governor pressure increase the engaging spool valve is moved to engage the hydraulic elements of the transmission. As the hydraulic system working pressure builds up, it further reacts against the bias pins which tend to open the engaging spool valve and effect a smooth modulated engage action as a function of speed and working pressure.

The hydraulic pump and motor relief valves, which are also located in the engaging valve housing, are set at approximately 3600 psi. The control system should hold the pressures below 2500 psi during normal operation for maximum efficiency and minimum wear. Pressure surges are held below the relief valve settings, during normal operation, by limiting the rate at which the actuator varies postion. Shock loadings, accidental shifting into reverse, or overloading greater than the full torque-speed ratio, will not harm the transmission, since the pressure relief valve is of a low inertia construction which prevents over pressurization of the hydraulic system.

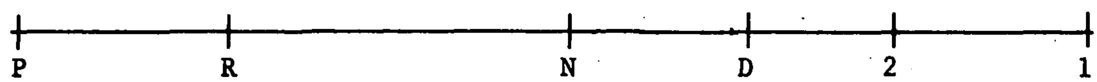
The charge pump is geared to the transmission input shaft and supplies charge pressure to the low side port of the pump and motor through the charge valves.

The discussion has been limited to the controls required to accomplish the primary functions of the transmission.

Although not discussed, additional hydraulic elements, and controls, for secondary functions such as bypass valves, flow restrictions, pressure regulators, filters, and interlocks are required.

Operation

The shift pattern schematic as shown below follows the present day automobile shift pattern as closely as practical. The control sequences for the various modes are discussed below.



P - Park

R - Reverse

N - Neutral

D - Drive

2 - Second

1 - Low

Neutral

In neutral, pressure is directed by the control valve to the piston actuator to move the sleeve to the position which limits piston stroke. Pressure is transmitted through the shuttle valve to engage the low-range brake. The engaging valve is in the open position and the swashplate of Element I is near zero displacement, which allows this element to rotate freely to effect zero output torque. A cam, driven by the shift lever, sets the engine regulator governor bias load to overcome the flyweight force, thus moving the governor valve. The resultant position of the governor valve provides flow to the actuator which moves the piston to the maximum torque position. Accordingly, in neutral, the control system presets the primary swashplate and low-range brake for low-ratio output for initial vehicle acceleration from a stopped position.

Drive

When the control valve is shifted to the drive position, the fluid pressure connections to the low-range brake and high-range clutch remain the same as in the neutral position. Pressure from the engage governor is transmitted through the spool valve to the engage valve. The output pressure from the engage governor increases as engine speed increases. This pressure acts on the engage valve to engage the transmission at 2100 to 2300 rpm transmission input speed.

During drive operation the engine regulator governor controls hydraulic flow to the opposite sides of the actuator piston to effect movement of the Element I swashplate. The movement of the governor valve is regulated in part by the position of the cam which adjusts the speed setting as a function of the accelerator pedal position. Accordingly, for every throttle position, the governor valve mechanism continuously controls the position of the actuator piston to vary the operating ratio to maintain a set engine speed. This provides a means for ideally matching the engine and vehicle speeds as a function of throttle position to provide optimum engine performance.

It will be noted that, in initially accelerating, the clutch valve rides upon the upper surface of the control cam, thereby providing fluid pressure through the shuttle valve to engage the low-range brake. When the transmission output speed increases, movement of the cam causes the clutch valve to move down to the lower surface of the cam. In the extended position of this valve, the fluid pressure to the low-range brake is vented, thereby releasing the low-range brake. Also, when this valve is extended, fluid pressure is provided to the high-range clutch, thereby engaging this clutch. This transition is made while both the high-range clutch and low-range brake are in synchronization to effect very smooth operation. This provides wear-free operation.

In Mode 2 the control cam introduces a bias to the engine regulator governor which varies the engine speed setpoint at which the transmission switches from low to high range and, in effect, holds the transmission in the low range. In Mode 1 (Low) a further bias is introduced, enforcing an even higher minimum torque ratio.

Reverse

In reverse, fluid pressure is transmitted from the control valve through the shuttle valve, to engage the low-range brake. The control cam is positioned by the mode selector to provide additional force on the engine regulator governor spring thus moving the engine regulator governor valve downward and providing fluid pressure to the bottom of the piston actuator. Pressure in the top chamber of the actuator is vented through the control valve. Thus, the piston and sleeve of the actuator both move to the top position. The additional stroke of the cam plate due to movement of the sleeve causes the pin connected with the swashplate housing to move into the negative angle portion of the cam slot, thereby positioning Element 1 swashplate at a negative angle for reverse output.

It will be noted that, when the control valve is moved to the reverse position, fluid pressure is directed to the engage valve so as to engage the hydraulic elements in the transmission.

Park

Park, which opens the engage valve to bypass fluid around the hydraulic motor and pump, permits the engine to idle without transmitting any torque to the transmission output shaft. The engine idles at 2100 rpm. In addition, the parking lock is positioned to lock the transmission output shaft to the rear wheels.

C. DESCRIPTION OF METHODS FOR DETERMINING PERFORMANCE

This section presents: 1) a description of the method employed to determine transmission performance, 2) a description of the computer program used for determining steady-state performance, 3) a description of the computer program used for driving-cycle analysis, and 4) a description of the computer program for simulation of full-power acceleration of the vehicle (used for calculating maneuver performance).

1. Method for Determining Transmission Performance

The selected hydromechanical transmission was considered to have the following components:

- Spur gears
- Planetary gears
- Hydraulic pumps
- Hydraulic motor

In transmitting power either under steady or transient conditions, each of these components is a source of power loss. The losses considered may be grouped into three types:

a. Mechanical losses

Mechanical losses always act to oppose rotation of a shaft, and arise from such sources as friction in the bearings, friction at gear teeth, and windage. All of the components listed above are subject to mechanical losses. Part of the mechanical loss acts as a function of its transmitted power (load dependent) and part is a function of speed and independent of transmitted power (speed dependent loss).

b. Flow losses

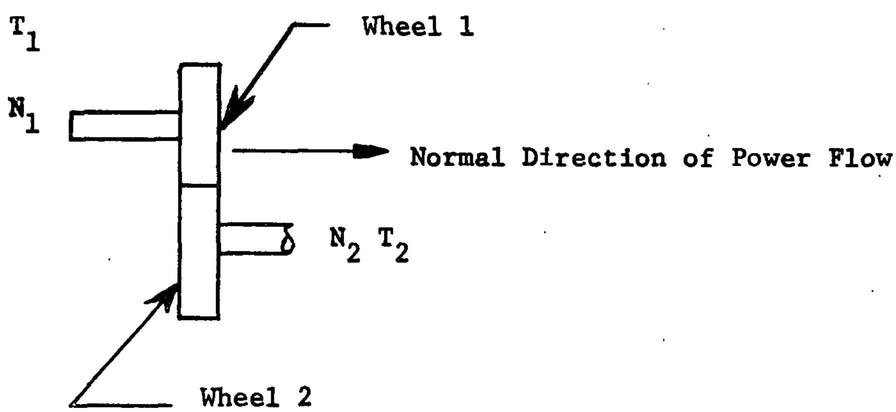
Flow losses are a loss in pressure head in a pump-motor combination (due to entrance), exit fluid inertial losses, and viscous pipe losses. The direction in which flow losses act is determined by the direction of power flow. Only hydraulic units are subject to flow losses. Flow losses are predominantly speed-dependent.

c. Compressibility and leakage losses

Compressibility and leakage losses represent deviations from ideal performance in the transfer of flow from one hydraulic unit to the other. Thus, while nominally the flow transferred to the motor equals the flow generated by the pump, the actual or effective flows differ by a small amount due to compressibility of the fluid and leakage through seals and past the pistons. The direction in which compressibility and leakage losses act is also determined by the direction of power flow. Only the hydraulic units are subject to compressibility and leakage losses.

The treatment of the various components including losses is as follows:

Spur gears



Speed equation:

$$N_2 = RN_1 \qquad \text{(C-1)}$$

Mechanical loss equation:

$$T_2 = [T_1 - |(1 - \eta) T_1| \operatorname{sign}(N_1)] / R \qquad \text{(C-2)}$$

where

R is the gear ratio

T_1, T_2 are the input and output torques acting in the normal direction of rotation

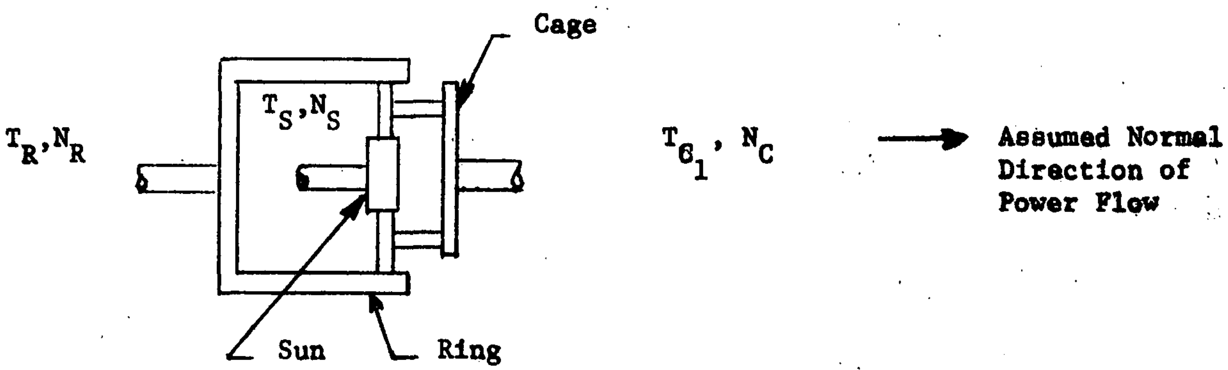
N_1, N_2 are the input and output speed

η is the efficiency with which power is transmitted through the gear pair

$|(1 - \eta)T_1|$ represents the absolute value of the quantity $(1 - \eta)T_1$

Sign (N_1) represents the algebraic sign of the quantity N_1; i.e., if N_1 is positive, Sign $(N_1) = +1$; if N_1 is negative, Sign $(N_1) = -1$.

<u>Planetary gears</u>



Speed equation:

$$N_C = N_S\left(\frac{r_S}{r_S + r_R}\right) + N_R\left(\frac{r_R}{r_S + r_R}\right) \tag{C-3}$$

Mechanical loss equations:

$$\left.\begin{aligned} T_S' &= T_S - T_{LS} \\ T_R' &= T_R - T_{LR} \\ T_C' &= T_C + T_{LC} \end{aligned}\right\} \tag{C-4}$$

Torque relationships:

$$\begin{aligned} T_C' &= 2\left(\frac{r_C}{r_S}\right) T_S' \\ T_R' &= \left(\frac{r_R}{r_S}\right) T_S' \end{aligned} \tag{C-5}$$

Mechanical loss definition:

$$T_{LS} = \left| (1-\eta_S)\, T_S \right| \text{sign } (N_S)$$

$$T_{LR} = \left| (1-\eta_R)\, T_R \right| \text{sign } (N_R)$$

$$T_{LC} = \left| (1-\eta_C)\, T_C \right| \text{sign } (N_C)$$

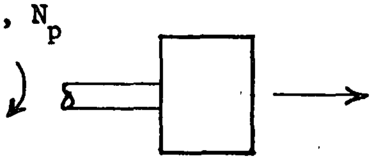
where

T_S, T_R	are the input torques to the sun and ring gears
T_C	is the output torque from the cage
T_S', T_C', T_R'	are the corresponding effective torques
T_{LS}, T_{LR}, T_{LC}	are loss torques associated with the sun, ring and cage
η_S, η_R, η_C	are corresponding efficiencies with which power is transmitted via sun, ring and cage (see Fig. C-2 for numerical values)
r_S, r_R, r_C	are radii of sun, ring and cage.

The above system of equations allows the cage output torque and speed and ring torque to be calculated from the torque and speed input to the sun and ring gears.

Pumps

T_p, N_p



Q_p, P Normal Power Flow Direction $\longrightarrow$

Mechanical loss equation:

$$T_p' = T_p - T_{LP} \qquad \text{(C-6)}$$

where

$$T_{LP} = \left| T_p \left(K_{2M} + K_{3M}\, \bar{D}_p\right) \right| \text{sign } (N_p)$$

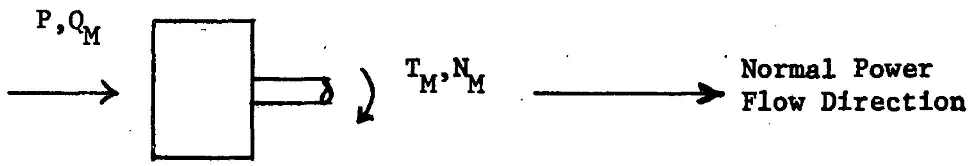
Flow loss equation:

$$P = T_p'/D_p - K_f \left(\left| \bar{N}_p \; \bar{D}_p \right| \right)^{\alpha} P_{max} \; \text{sign} \; (D_p) \qquad \text{(C-7)}$$

where σ is an exponent taken to 1.5. (1.5 is a value often used where both viscous (1st power) and inertial (2nd power) losses must be simply accounted for)

Note that all flow loss for the pump-motor combination is taken in the pump (This was simply a computational convenience contributing negligible error in the calculation).

<u>Motors</u>



Mechanical loss equation:

$$T_M = T_M' - T_{LM} \qquad \text{(C-8)}$$

where

$$T_M' = P \; D_M$$

$$T_{LM} = \left| T_M' \; (K_{2M} + K_{3M} \; \bar{D}_p) \right| \; \text{sign} \; (N_M) \qquad \text{(C-9)}$$

Transfer of flow from pump to motor

$$Q_M = Q_p - Q_{LC} \qquad \text{(C-10)}$$

where

$$Q_{LC} = L_1 p / p_{max}$$

$$L_1 = [K_{1L} + K_{2L} + K_{3L} \; \bar{D}_1] \left| \hat{Q}_p \right| + [K_{1L}] \left| \hat{Q}_M \right.$$

In calculating hydraulic transmission performance, the pump and motor are considered in combination. Equations C-6 through C-10 relate the six values of torque, speed and displacement for the pump and motor in such a way that, given four of the six values, the remaining unknown two may be determined.

In the above treatment of hydraulic units, the following definitions apply:

- T_p, T_M are the input torque to pump and output torque from the motor, respectively.
- D_p, D_M are the pump and motor displacements.
- $\overline{D}_p$ is the ratio of pump displacement to its maximum value for the unit.
- N_p, N_M are pump and motor speeds, rad/sec.
- $\overline{N}_p, \overline{N}_M$ are ratios of pump and motor speeds to the maximum values for the unit.
- P is pressure, lb/in^2.
- P_{max} is maximum pressure value for the unit.
- K_f is a flow loss coefficient (see below for values imposed for all loss coefficients).
- K_{2M}, K_{3M} are mechanical loss coefficients.
- K_{1L}, K_{2L}, K_{3L} are coefficients relating to compressibility and leakage losses.
- Q_p, Q_M are the effective pump and motor flows, in^3/sec.
- Q_{LC} is a leakage flow.

d. Speed dependent losses

In addition to the losses in gears and hydraulic units defined by equations C-1 through C-10 the following speed dependent losses are applied:

- Charge pump ,windage,and control losses (M)
- Valve plate friction losses (H)
- Clutch and brake losses (M)
- Main thrust bearing losses (H)
- Journal bearing losses (M)

These are broken into two sets of lumped losses, one associated with the hydraulic power path (items H) and one associated with the mechanical power path (M). These are located as shown in Figures C-1 and C‑2.

Numerical values for efficiencies and loss coefficients

The loss coefficients and mechanical efficiency numbers for the considered components were generated from previous experimental work.

Values for mechanical losses were typical for the type of gearing specified in the transmission design. The numerical values of gear efficiency used for performance calculations are shown in Figures C-1 and C-2 for the low-speed and high-speed ranges, respectively.

The equations defining losses in all hydraulic elements (presented earlier) were correlated to experimental data obtained for similar hydraulic units operating in various different transmissions that ranged in power rating from 25 to 200 horsepower. The resultant values of hydraulic element loss coefficients employed in all performance analysis calculations were as follows:

$K_f = 0.018$ $\qquad$ $K_{1L} = 0.0075$

$K_{2M} = 0.025$ $\qquad$ $K_{2L} = 0.025$

$K_{3M} = 0.005$ $\qquad$ $K_{3L} = 0.005$

The speed dependent losses described above are as specified in Table C-1 for two speeds.

TABLE C-1

SPEED DEPENDENT LOSSES

Speed RPM	Mechanical HP Loss	Hydraulic HP Loss	Total HP Loss
2100	1.00	0.5	1.5
3500	1.25	1.0	2.25

Between these speeds linear variation of losses with speed is assumed.

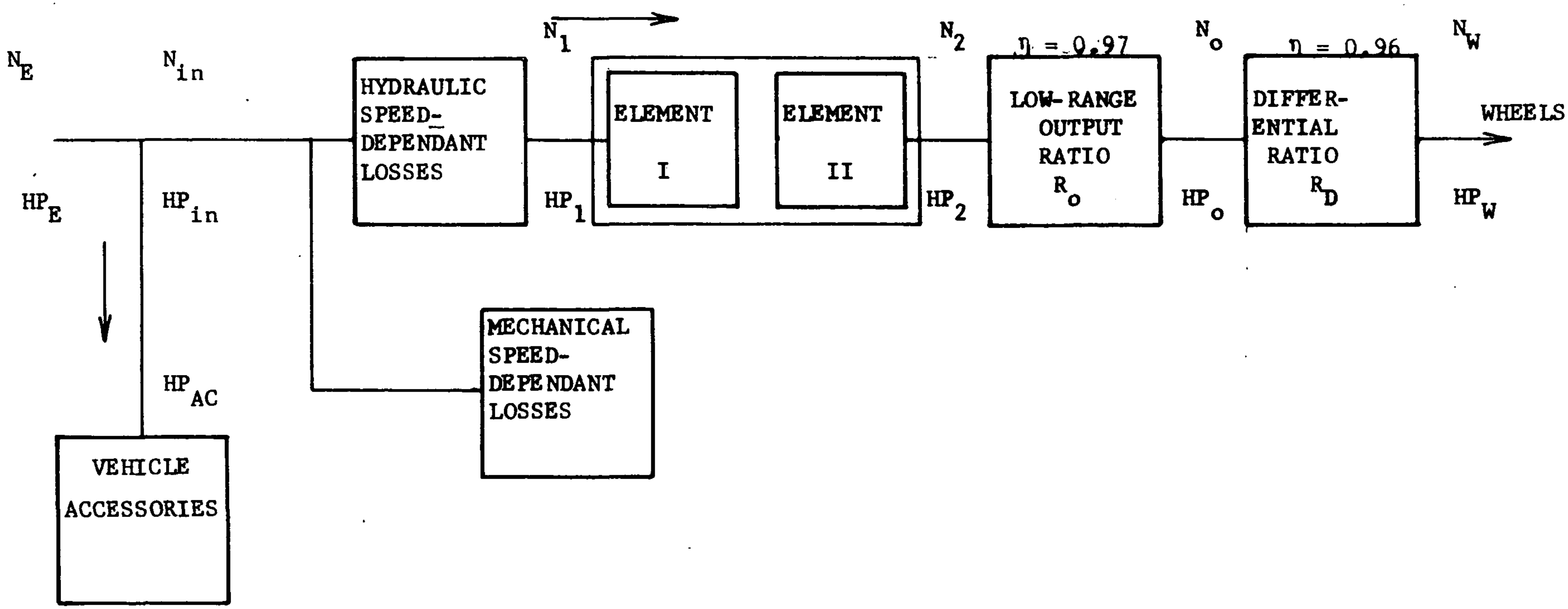


Fig. C-1 Power-Splitting Transmission Low-Speed Range Diagram

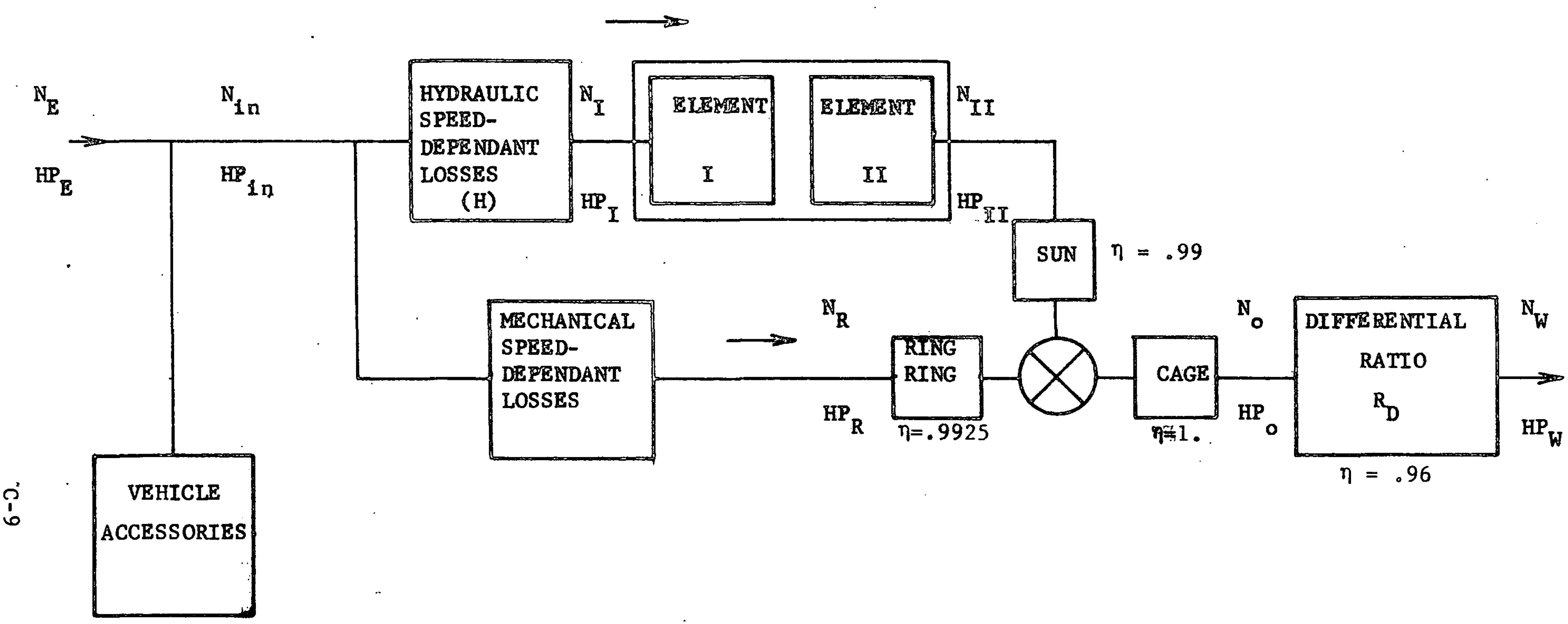


Fig. C-2 Power-Splitting Transmission High-Speed Range Diagram

2. Description of Steady-State Performance Computer Program

This computer program was used to compute constant-speed performance of the transmission and power train. Two types of load can be applied to the power train: 1) cruise road load as specified by EPA (1); or 2) constant power, limited at low speed by slip of the wheels.

The program operates by starting with a required power at the wheels and moving back towards the engine. At each point in the transmission in turn the program calculates the power required to overcome all losses between that point and the wheels and to supply the power demanded at the wheels. In determining the required power output from the engine speed reducer, the vehicle accessory load as specified by EPA (1) is added. With the continuously variable transmission under consideration, the transmission input speed (directly related to engine speed) for a given power requirement, is determined by an engine operating line, specified by the engine manufacturer. In general this line represents the condition of minimum SFC (maximum engine efficiency). At low and high speeds the limits of the transmission ratio range and engine speed range may force deviation from the operating line. The mechanics of this deviation will be discussed below.

An additional result of the analysis is the setting of the variable displacement unit necessary to achieve the particular operating condition. Thus, in addition to calculating performance, this computer program provides a means of defining the swashplate displacement schedules as a function of speed.

Detailed program procedure

The following sequence of operations describes the actual procedures followed by the steady-state computer program. The terms upstream and downstream describe relative locations which respectively follow or oppose the natural flow of power. Operation of the transmission in low range (Figure C-1) is described first (items 1-12) followed by modifications (items 13-15) to handle the high-range operation (Figure C-2).

1. For each vehicle speed of interest the program initially calculates the resistance torque to be overcome at the wheels, which must, therefore, be supplied to the wheels by the transmission. This torque is based on EPA specifications.

2. That transmission input speed is calculated which will provide the required power with the engine on the desired operating line. This first calculation of speed is made assuming no intermediate losses apart from wheel resistance.

3. Using equations C-1 and C-2, the speed and torque acting immediately upstream of the rear axle differential are computed.

4. Using equations C-1 and C-2, the speed and torque acting immediately upstream of the output gear ratio (R_o) are computed.

5. The mechanical power path speed-dependent losses are calculated as a function of the current value of transmission input speed. For the low-speed range these losses are the only input required to the mechanical power path.

6. Using equations C-6, C-7, C-8, C-9 and C-10 with pump and motor speed, motor torque and motor displacement specified, the pump displacement and pump torque are determined accounting for all losses in the hydraulic pump-motor combination.

7. Hydraulic speed-dependent torque losses based on Table C-1 are added to the pump torque to determine the torque input to the hydraulic power path.

8. The torque inputs to the mechanical and hydraulic power paths are added and multiplied by speed to give the transmission input power.

9. The accessory HP, which is a function of engine speed based on EPA specifications, is added to the transmission input power, giving the power required as output from the engine speed reducer.

10. From the engine performance map the engine speed to provide this power while running on the desired operating line is determined.

11. The above procedure, starting with item 3, is repeated until engine speed and displacement values are repeatable between successive iterations within 1 part in 10,000.

12. Using the manufacturers engine performance map, the SFC, fuel flow and MPG corresponding to the calculated power-speed condition of the engine are determined.

For operation of the transmission in the high range, the following steps replace steps 4 and 5.

13. With ring (transmission input) speed and cage (upstream of R_O) speed specified, equation C-3 is used to calculate the sun speed - which is equal to the speed of element II.

14. Using equations C-4 and C-5, the sun and ring torques for the output planetary are calculated. The sun torque is the torque of element II. The ring torque (T_R) is the torque downstream of the mechanical speed dependent losses. The cage torque is equal to the torque upstream of the rear end differential and known (from step 3).

15. The mechanical power path loss torque is calculated as a function of the current value of transmission input speed and added to T_R to give the torque input to the mechanical power path.

Apart from the above modifications to handle the output planetary, the treatment for high-range operating parallels that for low-range operation.

The following additional constraints apply:

- If any of the engine speed requirements call for a displacement of unit I which lies outside the upper or lower limits, then the value of displacement is set at the limiting value and the equations solved for engine speed. However, if the resultant engine speed falls below the minimum engaged speed, then this condition will cause the transmission to partially disengage and is not directly calculable by the procedures described. The latter condition only occurs at speeds below 10 mph. To obtain efficiencies at speeds below 10 mph, linear interpolation between zero at zero mph and the value calculated for 10 mph is used.

- If the engine under consideration has been scaled up in power by some factor, F, it is assumed that for a given engine speed the SFC for a given power demand, P, is that corresponding to the power P/F at the same speed on the unscaled (original map).

<u>Output on steady-state program</u>

The following quantities are calculated and printed by the computer program as a function of vehicle velocity:

Wheel Speed
Engine Speed
Transmission Output Speed
Wheel Torque
Road Torque
Engine Input Torque
Engine Output Torque
Transmission Output Torque
Engine HP
Road HP
Transmission Input HP
Accessory HP
Overall Efficiency
Transmission Efficiency
Fuel Flow
Specific Fuel Consumption
MPG
Speed, Torque, HP at Hydraulic Units

Hydraulic Pressure

Displacements of Hydraulic Units

Ideal Hydraulic Flows

Input for steady-state program

Input to the computer program consists of:

Wheel Radius

Car Frontal Area

Wind Resistance Coefficient

Ambient Pressure Temperature

Vehicle Weight

Accessory HP Tables as a Function of Engine Speed

Engine Performance Tables

Gear Ratios

Sun, Gear, Cage Radii for Output Planetary

Mechanical Efficiencies

Compressibility, Leakage and Flow Coefficients

Limits of Displacements and Speed Pressure

3. Description of Driving Cycle Performance Computer Program

This computer program is used to calculate the time-varying and cumulative performance of the vehicle power train over any driving cycle specified in terms of velocity values at a sequence of discrete points in time. In this case the load applied to the vehicle is calculated on the basis of cruise road load at the appropriate velocity added to the power at the wheels necessary to accelerate* the vehicle according to the driving cycle. The driving cycle used for all performance calculations is that specified in the Federal Register dated, July 2, 1971.

The program processes each interval of the driving cycle in turn, starting off by calculating the power required at the wheels, then moving back towards the engine. The procedure followed is very similar to that previously defined for the steady-state program except that the transmission is treated as a

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* The required acceleration over each time increment of the driving cycle is obtained by numerical differentiation ($a=(V_{I+1}-V_I)/\Delta T$) where V_I, V_{I+1}, are velocities at beginning and end of Ith increment.

single component with an efficiency defined as a function of speed and power by the steady-state program. Thus the detailed calculational procedure is exactly as defined for the steady-state program except that steps 4, 5, 6, 7, 8, 13, 14 and 15 are replaced by:

16. Using tables of efficiency data generated by the steady-state program the transmission input torque is calculated and multiplied by speed to give the transmission input power.

Thus, as for the steady-state program, the engine is operated on the operating line specified by the engine manufacturer. The implicit assumption when applying this approach on the driving cycle is that the control system is perfect. Thus the results of this driving cycle analysis may be slightly optimistic in relation to the performance of a real power train - control system combination.

Certain special conditions peculiar to the driving cycle analysis are handled as follows:

1. Deceleration

   If the deceleration is so mild that the negative vehicle inertia force remains less than the steady-state road load, the net output power from the transmission must remain positive. In this case the treatment is the same as for any condition in which the power flow is positive.

   If the deceleration is sufficient for the negative vehicle inertia force to exceed the steady-state road load, then it is assumed that positive braking is required; that no power output is required from the transmission; that the transmission disengages; and that the engine speed falls to idle. The only power output from the engine under idle condition is the accessory load (2.00 hp without air conditioning, 4.00 hp with air conditioning).

2. Low-speed operation of the vehicle

At speeds below approximately 10 mph, the transmission is partially disengaged. It is assumed that transmission efficiency can be interpolated linearly between zero at zero vehicle speed and the value obtained at the minimum engaged speed (subject to an arbitrary minimum of 2 1/2 percent efficiency). This treatment is regarded as conservative.

3. Transmission out of range

If the operating line requirement calls for a transmission ratio above the maximum value (~5.5:1) then the engine speed is calculated to satisfy this ratio. However, if the resultant engine speed falls below the minimum engaged speed (implying a vehicle speed below 10 mph), then the engine speed is held at the minimum engaged value and the efficiency is interpolated as described above.

Having iterated to establish the power and speed from the engine speed reducer, the fuel flow is calculated by interpolating from the manufacturer's engine map as for the steady-state program.

The above description applies to each individual time interval. Additional calculations involve the calculation of cumulative values of work done at various points in the transmission, distance travelled, and fuel used.

Output

The following quantities are printed by the program as a function of time:

Vehicle velocity
Wheel speed
Transmission output speed
Transmission input speed
Acceleration power
Steady-state power

Road power

Transmission efficiency

Overall efficiency

Fuel flow

SFC

MPG

Together with cumulative values of

Time of trip

Distance travelled

Road work done

Engine work done

Total fuel consumed

And average values of

Velocity

Road power

Engine power

SFC

MPG

Transmission efficiency

Input

Inputs to the Driving Cycle Program are:

Weight of car

Ambient pressure

Ambient temperature

$A \times C_d$

Minimum engaged speed

Idle speed

Wheel radius

Differential efficiency

Differential gear ratio

Accessory power

Fuel density

Engine operating line (speed vs. hp)

Engine map (fuel flow vs. speed, power)

Transmission efficiency map (efficiency vs. output, speed, power)

4. Description of Full-Power Acceleration Performance Computer Program

This computer program calculates the histories of acceleration, velocity and distance travelled for a vehicle accelerating between two specified velocities. This program is used to calculate the performance of the vehicle relative to the EPA maneuver specifications.

The program solves the combined equations of motion for the engine and vehicle, subject to the assumption of perfect controls. The maximum engine power as a function of speed is obtained from the engine map, and the engine is operated for as much of the maneuver time as possible at the speed providing maximum power.

The combined equation of motion for the engine-inertia system is as follows: (more detailed explanation)(follows nomenclature).

$$T_e = T_a + J_e\dot{N}_e + RR_oJ_o\dot{N}_o + RR_oT_{res} + RR_oT_{lo} + RT_{11} \qquad \text{(C-11)}$$

$$N_o = R_oRN_e \qquad \text{(C-12)}$$

$$\dot{N}_o = R_o\dot{R}N_e + R_oR\dot{N}_e \qquad \text{(C-13)}$$

where:

T_e is the output torque from the engine speed reducer.

T_a is the accessory torque.

J_e is the engine inertia as reflected at the transmission input shaft.

N_e is the engine speed (rad/sec).

R is the transmission ratio.

R_o	is the differential ratio.
J_o	is the vehicle inertia as reflected at the rear wheels.
N_o	is the wheel speed.
T_{res}	is the wheel torque required to overcome rolling and air resistance (as specified by EPA).
T_{10}	is the torque loss in the transmission as reflected to the transmission output shaft.
T_{11}	is the torque loss in the differential as reflected to the rear axle.
$\dot{N}_o$, $\dot{N}_e$	are wheel and engine accelerations, respectively.

Equation C-11 provides a means of determining the distance travelled, velocity and acceleration history of the engine and vehicle provided that some relationship defining the variable ratio R in equations C-11, 12, 13 as a function of time is available. Equation C-11 expresses the fact that the available engine torque (which we know, as a function of engine speed) must, in general, provide torque to accelerate engine and vehicle, and overcome internal losses and the road load. According to the relationship for R (discussed below) certain terms drop out and the equation is rearranged to provide values either for acceleration of the engine alone the vehicle above or some combined acceleration of the two. So as to simplify the problem relationships for R are considered only for the following 3 special cases

1. Engine Speed = Constant
   Under this condition $\dot{N}_e=0$ and equation C-11 becomes a one-inertia equation for vehicle speed. The identity $\dot{N}_e=0$ is substituted into equations C-12, 13; these are, in turn, substituted into C-11, and C-11 is rearranged to make $\dot{N}_o$ its subject.

2. Vehicle Speed = Constant

Under this condition $\dot{N}_o=0$, and equation C-11 becomes a one-inertia equation for engine speed. The identity $\dot{N}_o=0$ is substituted into equations C-12, 13; these are in turn substituted into C-11, and C-11 is rearranged to give $\dot{N}_e$ as its subject.

3. Transmission Ratio = Constant

Under this condition $\dot{R}=0$, and equation C-13 allows either $\dot{N}_e$ or $\dot{N}_o$ to be eliminated from equation C-11, so that a single effective inertia problem again exists. C-11 provides a relationship with either $\dot{N}_o$ or $\dot{N}_e$ (arbitrary) as the subject.

Adequate performance results for the EPA maneuvers can be generated using combinations of these three conditions. Figure C-3 shows the relationship between engine and wheel speed which has been imposed to cover all acceleration maneuvers. Five regions (I, II, III, IV, V) are identified on this figure, and are defined below.

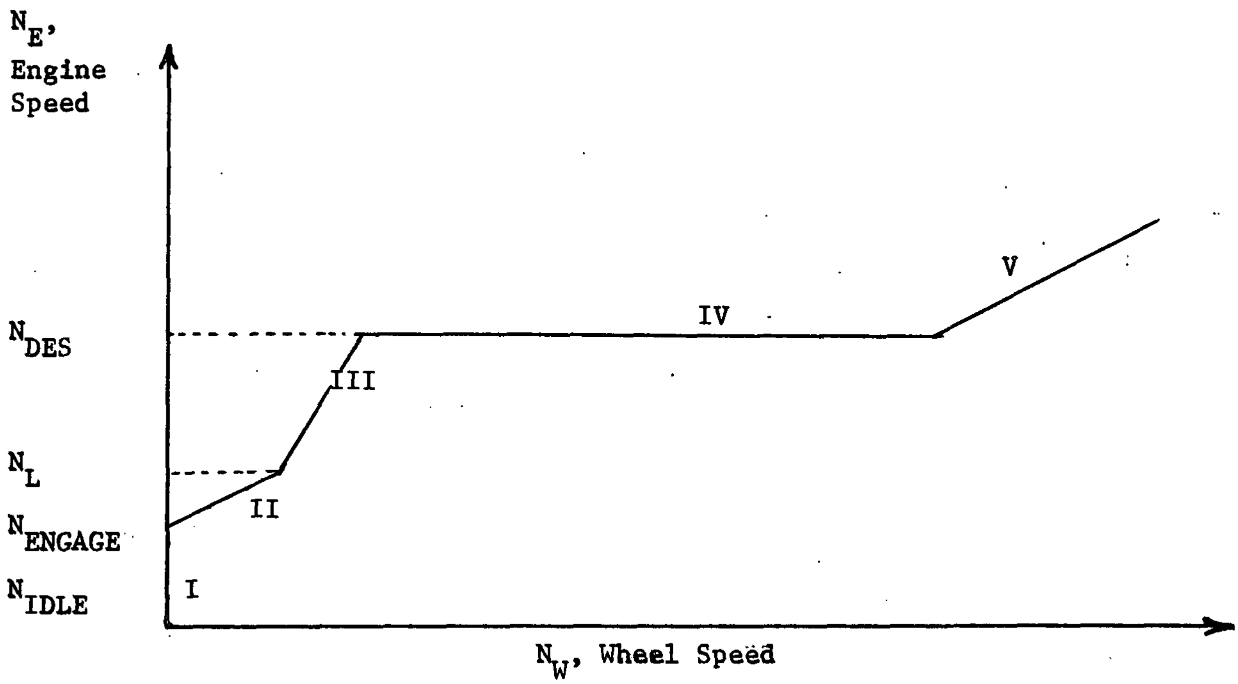

Fig. C-3 Relationship Between Engine Speed and Wheel Speed for Acceleration from Standing Start

Region I

The engine is accelerated from idle to some speed, N_{engage} which is intended to correspond to the point in a real system where the transmission engagement valve is closed. During this initial acceleration the vehicle is maintained stationary. The delay involved before the vehicle starts to move is, in all cases, less than 1/4 sec., which is considered an acceptable short time (N_{engage} is 200 rpm above idle speed for the Airesearch engine, 100 rpm above idle speed for the Aerojet engine).

Region II

During region II the engine and vehicle are considered to follow a linear relationship until the ratio of engine speed to wheel speed reaches a value corresponding to the maximum steady-state transmission ratio. The purpose of region II is to bridge the gap between the condition when the engagement valve is closed but there is 100 percent slip and the point where the slip falls to a steady-state value (~10-12 percent slip for full power at 10 mph).

In region II equations C-12 and C-13 are replaced by the condition

$$\begin{aligned} N_e &= AN_o + B \\ \dot{N}_e &= A\dot{N}_o \end{aligned} \qquad \text{(C-14)}$$

A and B are constants during a particular acceleration and are determined by the values imposed for N_{engage}, N_{idle} and the maximum steady-state transmission ratio. Equation C-14 is substituted into equation C-11 to eliminate one of $\dot{N}_e$ or $\dot{N}_o$ and the problem is solved as a single-inertia problem.

Region III

During region III the ratio R remains constant at the maximum value for the transmission. Equation C-11 is solved as described above for a constant ratio. Region III ends when the engine speed reaches the desired operating speed for full-power acceleration.

Region IV

During region IV the engine speed remains constant at the desired operating speed for full-power acceleration. Equation C-11 is solved as described above for constant engine speed. Region IV normally continues to the completion of the maneuver.

Region V

Only in the case of the Aerojet engine, towards the end of the high-speed passing maneuver (50-80 mph), does the engine get forced, by the lowerlimit of the transmission, to go above the desired operating speed (~2970 rpm). The system then operates in region V which is a constant-ratio region, solved exactly as for region III.

The above idealization is intended as a simplified representation of the more gradual, smooth changes in condition occurring in a real power-train-control system. For example, the actual engagement process will start below the speed N_{engage}, causing a build-up in hydraulic pressure, torque output from the transmission, and acceleration of the vehicle, thus eliminating the first discontinuity in slope of Figure C-3 between region I and II. Similar smoothing of the curve of Figure C-3 will occur over its entirety.

During the early stages of acceleration from standing start, if all available engine power were transmitted to the rear wheels they would skid. This was controlled in the analysis in a manner analagous to the operation of the transmission (in which relief valves limit pressure to the value corresponding to wheel slip). Thus in the analysis torques corresponding both to limiting hydraulic pressure and to wheel slip were calculated and the transmitted torque was not allowed to exceed either value.

The numerical calculations were performed using the Runge-Kutta 4th order integration method.

D. PERFORMANCE ANALYSIS

This section presents details of the performance which was determined for the selected hydromechanical transmission and the resultant power-train performance of two selected advanced automatic engines with the transmission. Three catagories of performance analysis are considered:

- Steady-state analysis (cruise and constant output power)
- Driving-cycle analysis
- Full-power acceleration (standing start and DOT maneuvers)

In each analysis the performance was calculated for the selected engines (specified by AiResearch and Aerojet) with the selected transmission powering or medium-sized family car, as specified by EPA/AAPS vehicle design goals (1)*. Comparisons, where possible, are made with the corresponding performance of a similar power-train employing a conventional automatic transmission.

1. Specified Engine Characteristics

The characteristics of the two advanced automotive engines used for this study were based upon engines selected by EPA/AAPS in order to ascertain the performance gain offered by an advaneed transmission. The engine characteristics were supplied by Aerojet and AiResearch as a result of their work under seperate EPA/AAPS contracts. These characteristics are based upon differing design constraints, not necessarily optimized for the selected variable-ratio transmisson, and have not been demonstrated experimentally.

The characteristics of the Aerojet engine are based upon their "prototype" design for a turbo Rankine-cycle engine which employs an organie working fluid. Aerojet is currently testing and developing the "pre-prototype" version of this engine under an EPA/AAPS contract. Figure D-1 shows the

*Denotes references listed in Section F.

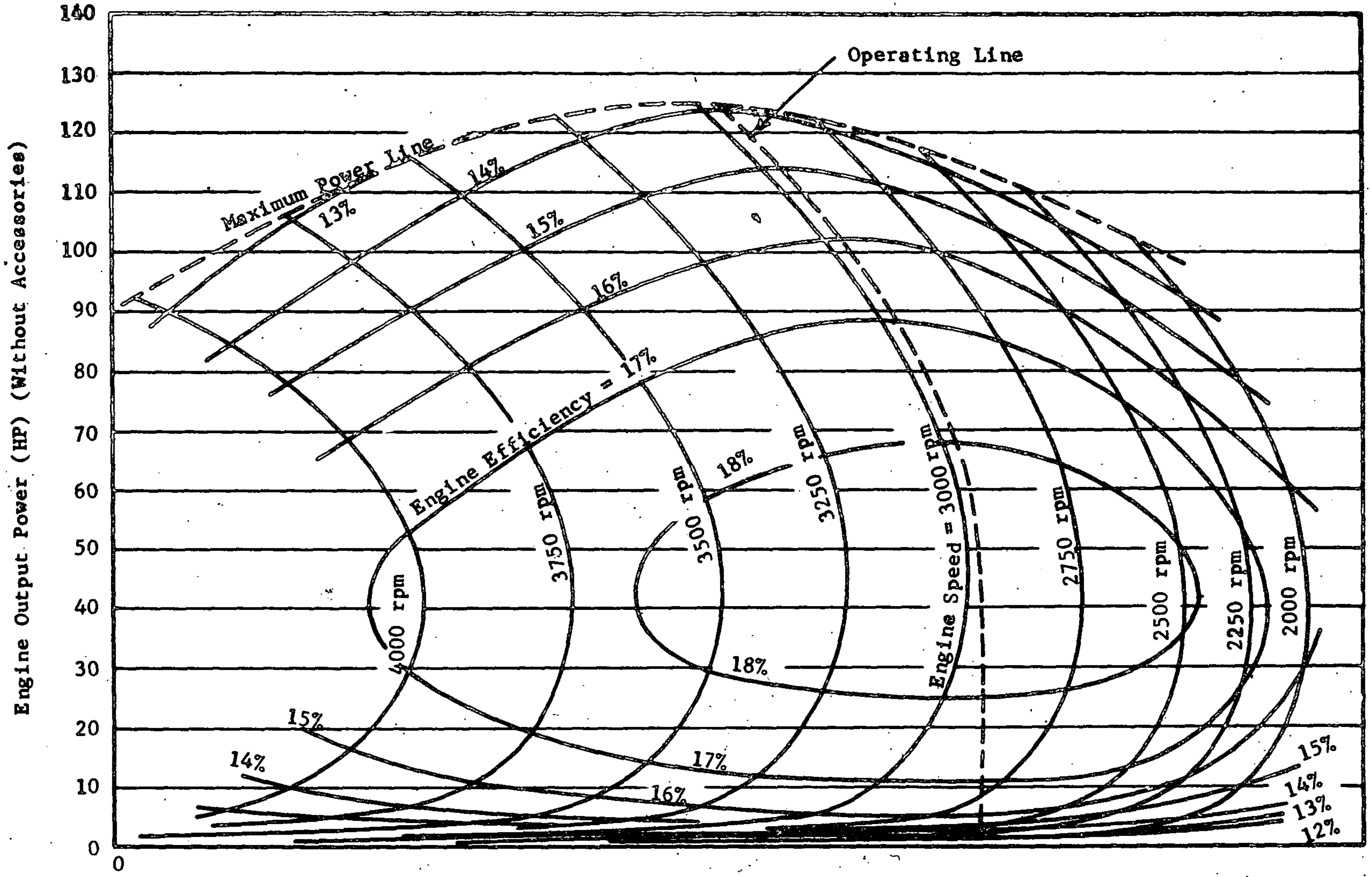

Fig. D-1 Aerojet Engine Performance Map
(Ram Air Velocity = 0)

performance map for the Aerojet Rankine-cycle engine. Engine power level (defined as the power output from the engine speed reducer) is plotted against overall engine efficiency and speed reducer output speed. Figure D-2 shows the percentage increase in power and efficiency as a function of vehicle speed due to ram air velocity.

The AiResearch engine characteristics are for a single-shaft, gas turbine as described in their report to EPA/AAPS (2). Appendix A defines, in tabular form, the performance of the AiResearch single-shaft gas turbine. However, to illustrate the nature of the power variation with speed and to define the operating line, a partial map (without fuel flow data) is plotted in Figure D-3 The engine, as analyzed, was scaled up in power output by 15 percent. The scaling law used was that, for given speed and SFC, the engine could put out 15 percent more power than specified by the map. The purpose in scaling up the engine power is to allow achievement of all EPA maneuver specifications. This is discussed subsequently in more detail. It is to be noted that the AiResearch Engine data already has 4 hp subtracted to account for accessory load. Since a variable accessory power was imposed in the present study, this 4 hp was re-included in the engine power, before scaling.

Additional engine and transmission interface data are given in Table D-1.

TABLE D-1

ENGINE AND TRANSMISSION INTERFACE DATA

Manufacturer	AiResearch	Aerojet
Type	SS Gas Turbine	Rankine Cycle
100% Transmission input speed	3500 rpm	3500 rpm
Engine inertia (reflected at transmission input)	0.766 lb.ft.sec^2	0.306 lb.ft.sec^2
Transmission input speed at idle	2100 rpm	2100 rpm
Max. power	147 hp at 100% speed (scaled up 15%)	125 hp at 0 mph 133 hp at 60 mph
Idle fuel flow	3.23 lb/hr without A/C 3.81 lb/hr with A/C	2.42 lb/hr without A/C 3.55 lb/hr with A/C

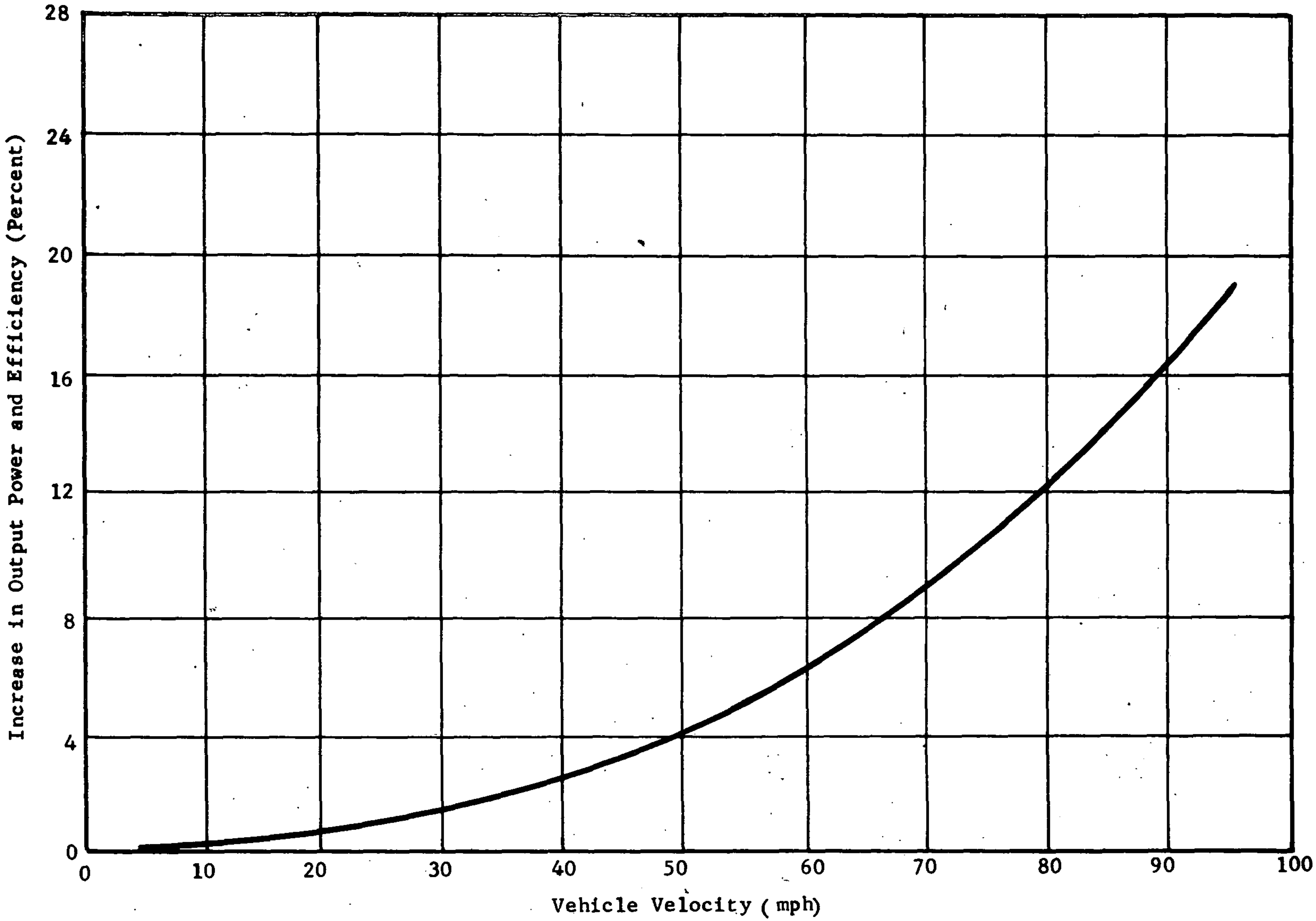

Fig. D-2 Effect of Ram Air Velocity — Aerojet Engine

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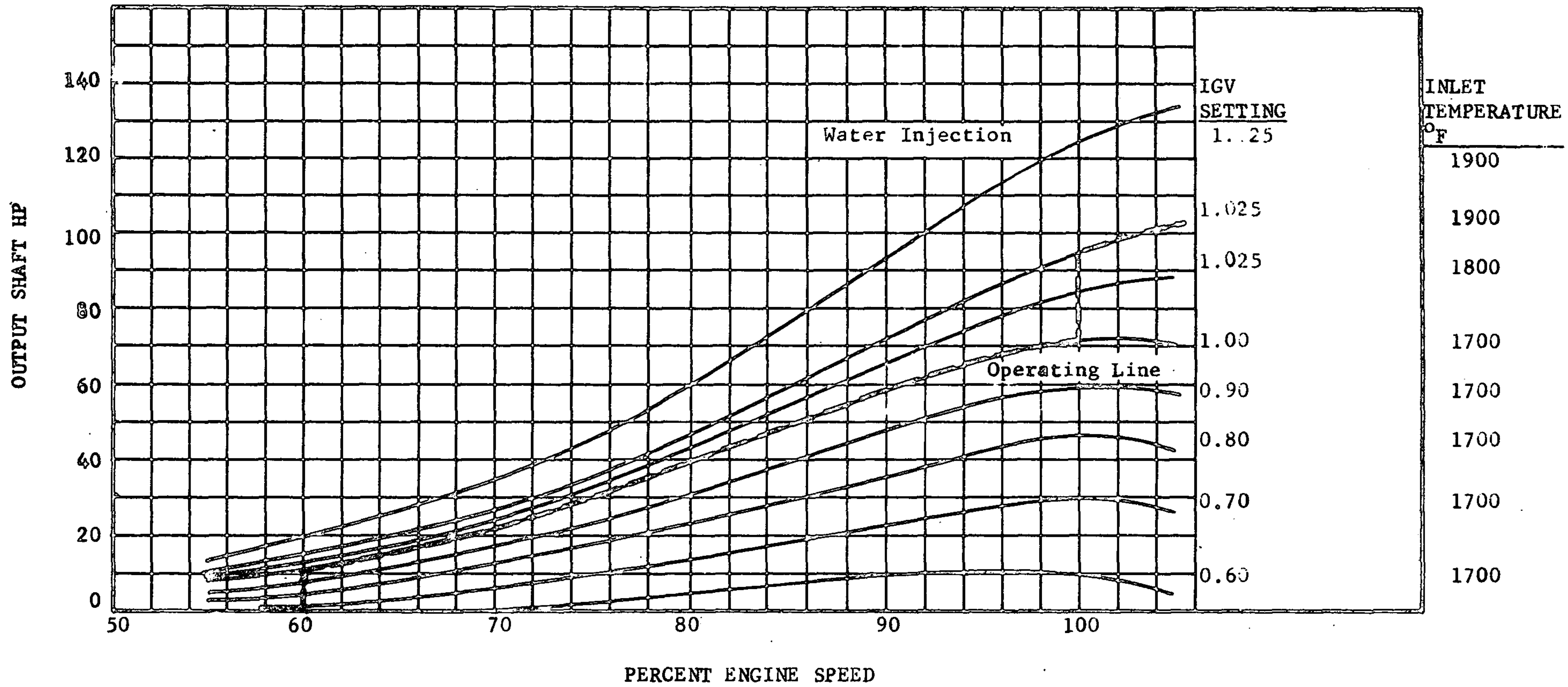

Fig. D-3 AiResearch Engine Performance Map
(Includes Constant 4-hp Accessories)

Superimposed on each of the aforementioned engine maps (Figures D-1 and D-3) is an operating line which uniquely defines desired engine speed as a function of power level. The operating line was specified by the engine manufacturer as the desired steady-state operating condition of the engine. The operating line tends to coincide with the maximum engine efficiency (minimum SFC) under steady-state conditions. As described elsewhere in this report, in the computation of steady-state and driving-cycle performance, the continuously variable ratio transmission is adjusted (controlled) to maintain engine speed (by varying engine load) on the selected operating line, within the extreme limits of the transmission.

2. Power-Train Loading

Under steady-state cruise conditions the road load is a combination of rolling resistance and air resistance which is specified as a function of velocity by EPA/AAPS (1). The variation of this load with vehicle velocity is plotted in Figure D-4. In addition the engine is subjected to Accessory loads, also based on EPA specifications as a function of engine speed, for the cases of with and without air conditioner. The accessory load variation with engine speed is shown in Figure D-5.

For constant-power loading, a range of power-level demands, each related by some fraction to the full-power load (taken to be 100 hp at the road), is applied in turn. These loads are limited at low vehicle velocity by the constraint that the implied tractive force at the wheels should be less than 2150 lb. The resultant road power level variation with vehicle velocity is shown in Figure D-6.

Under driving-cycle and full-power acceleration conditions the same road load as described for steady-state cruise conditions was applied. In addition the power to accelerate a 4600 lb car and appropriate engine inertia were applied. The power available to accelerate the vehicle is, in both cases, limited firstly by the maximum engine power under wide-open throttle conditions and secondly, by the wheel slip condition (maximum tractive force of wheels = 2150 lb).

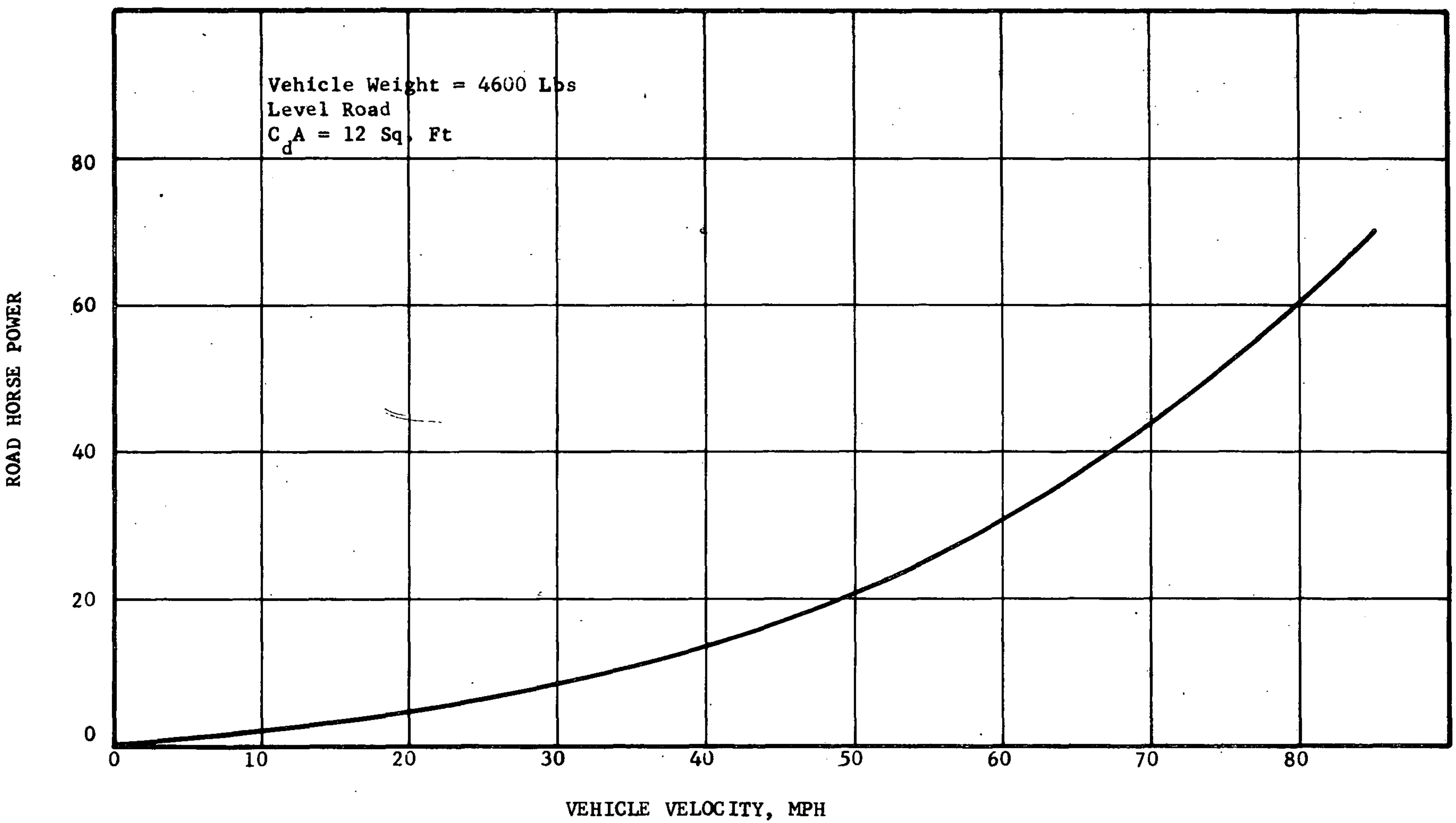

Fig. D-4 Road Horsepower — Steady-State Cruise Conditions

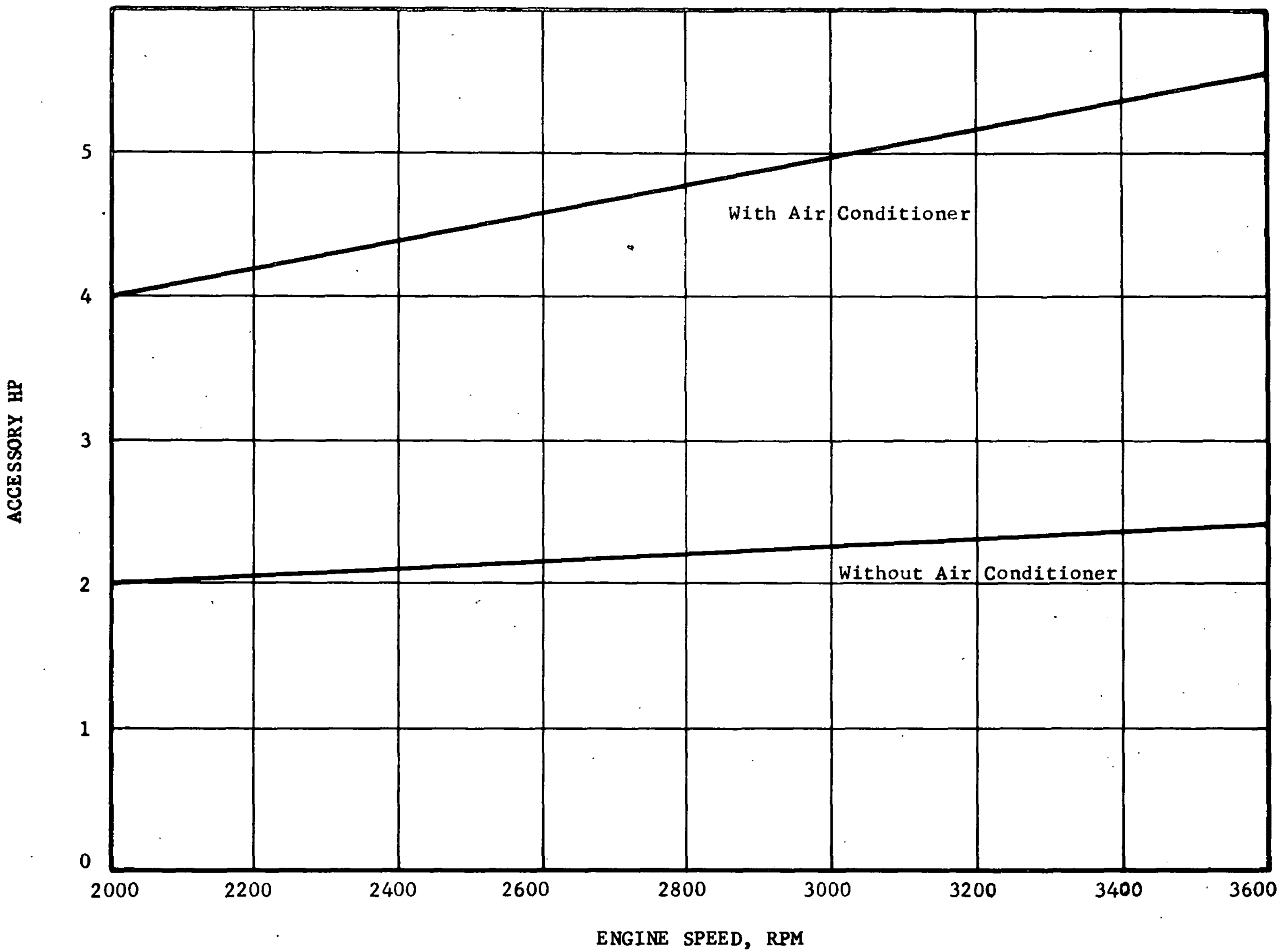

Fig. D-5 Accessory Load — EPA/AAPS Specification

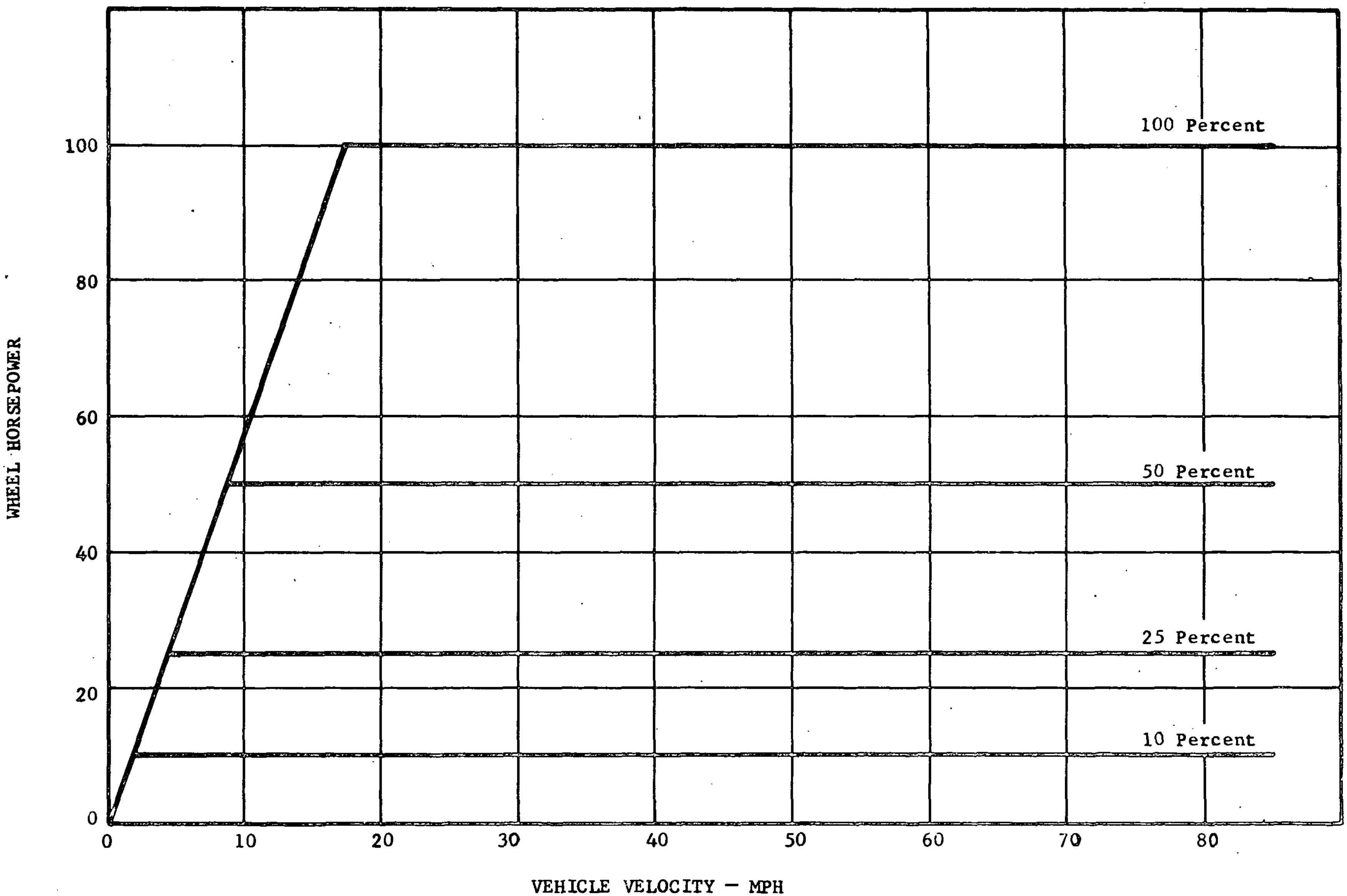

Fig. D-6 Constant Power — Low Speed Limits

3. Transmission Performance Characteristics

The efficiency of the selected split-power path hydromechanical transmission is highly dependent upon the mode of operation. At high speeds, where the majority of power passes through the mechanical power path, the efficiency is high. The shift point between low- and high-speed ranges occurs at between 20 and 30 mph. Peak efficiency tends to occur close to the "straight-through" point, where power flow through the hydraulic path is essentially zero. The "straight-through" point occurs at between 35 and 50 mph.

The above cruise efficiency characteristics are shown very clearly in Figure D-7, which presents the variation of transmission efficiency with vehicle velocity under cruise conditions for the two engines. At 20 mph the efficiency is between 57 and 60 percent, but rises to 90 percent at 50 mph. The slight discontinuity in slope at 20 mph with the AiResearch engine, and at 25 mph with the Aerojet engine indicates the shift point. The "straight-through" point occurs at 40 mph and 50 mph for the AiResearch and Aerojet engines, respectively.

The marked difference between the transmission efficiencies with the two engines at speeds below 50 mph reflects the differences in the manufacturer's specified operating line. Up to approximately 24 hp output from the engine the transmission input speed with the AiResearch engine is 2300 rpm (minimum engaged speed). However, with the Aerojet engine the transmission input speed lies between 2700 and 3000 rpm. This difference in speed means that speed dependent transmission losses are more significant with the Aerojet engine than with the AiResearch engine and, at the low power levels demanded by the cruise condition, the effect of these losses on efficiency was 13 percent at 30 mph. From a systems point of view this suggests the possibility of improved overall performance resulting from operating the Aerojet engine at a speed below optimum. However, this possibility has not been explored under the present study.

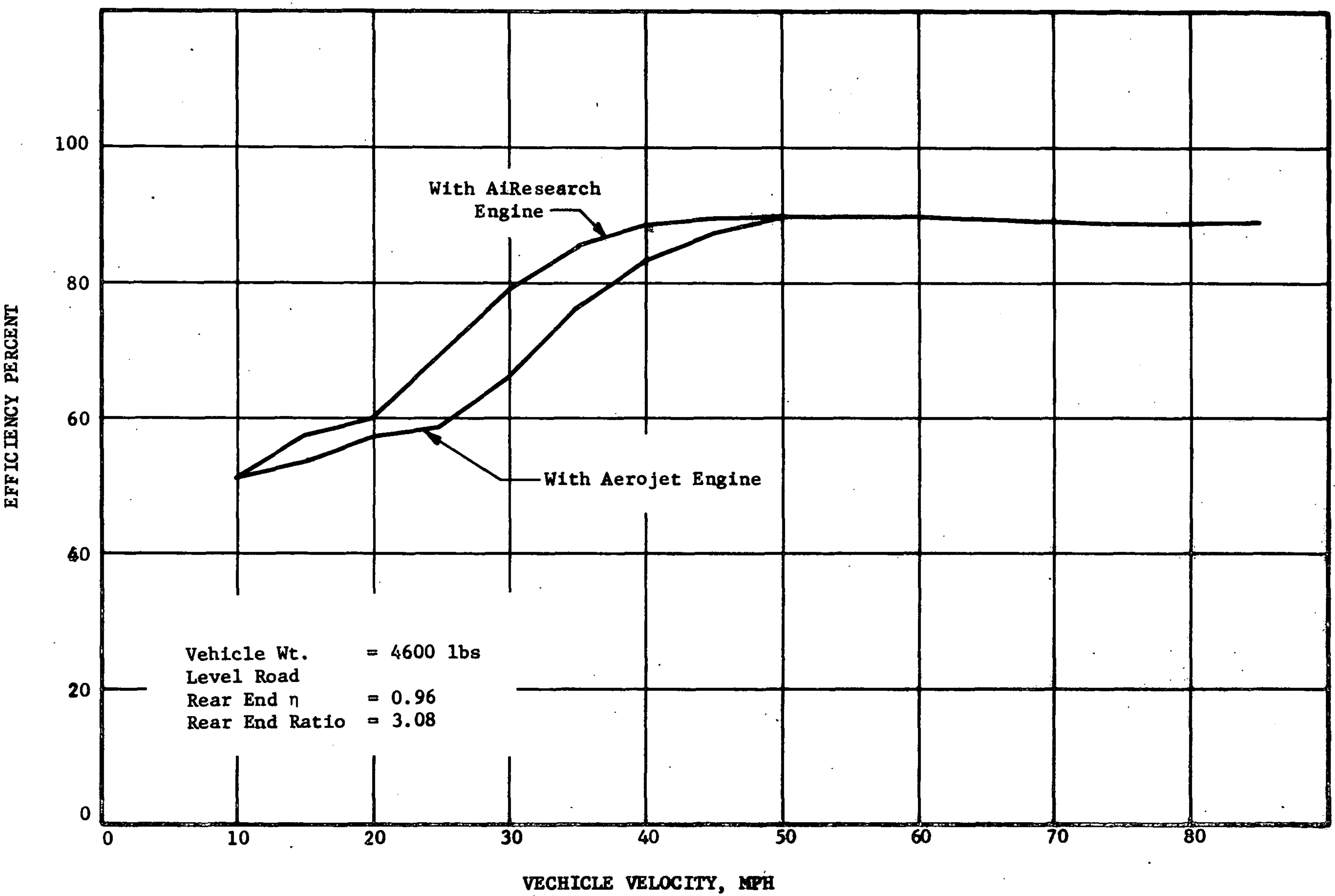

Fig. D-7 Cruise-Power Efficiency of Selected Transmission

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The maximum transmission efficiency under cruise conditions is close to 90 percent. It occurs near to the "straight through" point (40 mph and 50 mph for AiResearch and Aerojet respectively) and remains almost constant at speeds above this.

The variation in efficiency at constant power levels as shown by Figures D-8 and D-9 reveals some differences from the cruise efficiency curves. This is particularly noticeable at low speeds. For example, at 10 mph, the transmission cruise efficiency is 51 percent; whereas, with an output at the road of 10 percent of rated power, the efficiency is 78 percent. This difference results from the fact that the cruise power demand at 10 mph is only 2 hp at the road, exaggerating the importance of speed dependent losses, which are of similar magnitude to the road load. A further observation from the constant-power efficiency curves is the reduced separation between the two engines at high power levels, indicating the reduced significance of speed-dependent losses at high power levels.

The peak transmission efficiency is close to 95 percent, under conditions of high power at the "straight through" speed condition. This high efficiency value is a result of the minimal losses which occur when all power passes through the mechanical path. The 5 percent loss which is incurred represents the sum of all losses in the output planetary, the residual power necessary to turn the variable hydraulic element even when no power is passing through it, and the "parasitic" losses such as the charge pump.

Figures D-10 and D-11 are cross-plots of transmission efficiency vs output power at 3 constant speeds, which reinforce the significance of speed-dependent losses at low power levels. At 85 mph the speed-dependent losses are highest, and at 10 hp output, the 85-mph efficiency is clearly the lowest. The fact that the 50-mph efficiency is consistently highest over the output power range reflects the closeness of this speed to the "straight through" point.

Figure D-12 presents a comparison of the selected hydromechanical transmission cruise power efficiency with that of a conventional automatic. The conventional automatic used for this comparison was that currently selected by Aerojet for use with their Turbo-Rankine engine. The shift points are therefore designed to produce optimum system performance for this engine. The wide spread of the

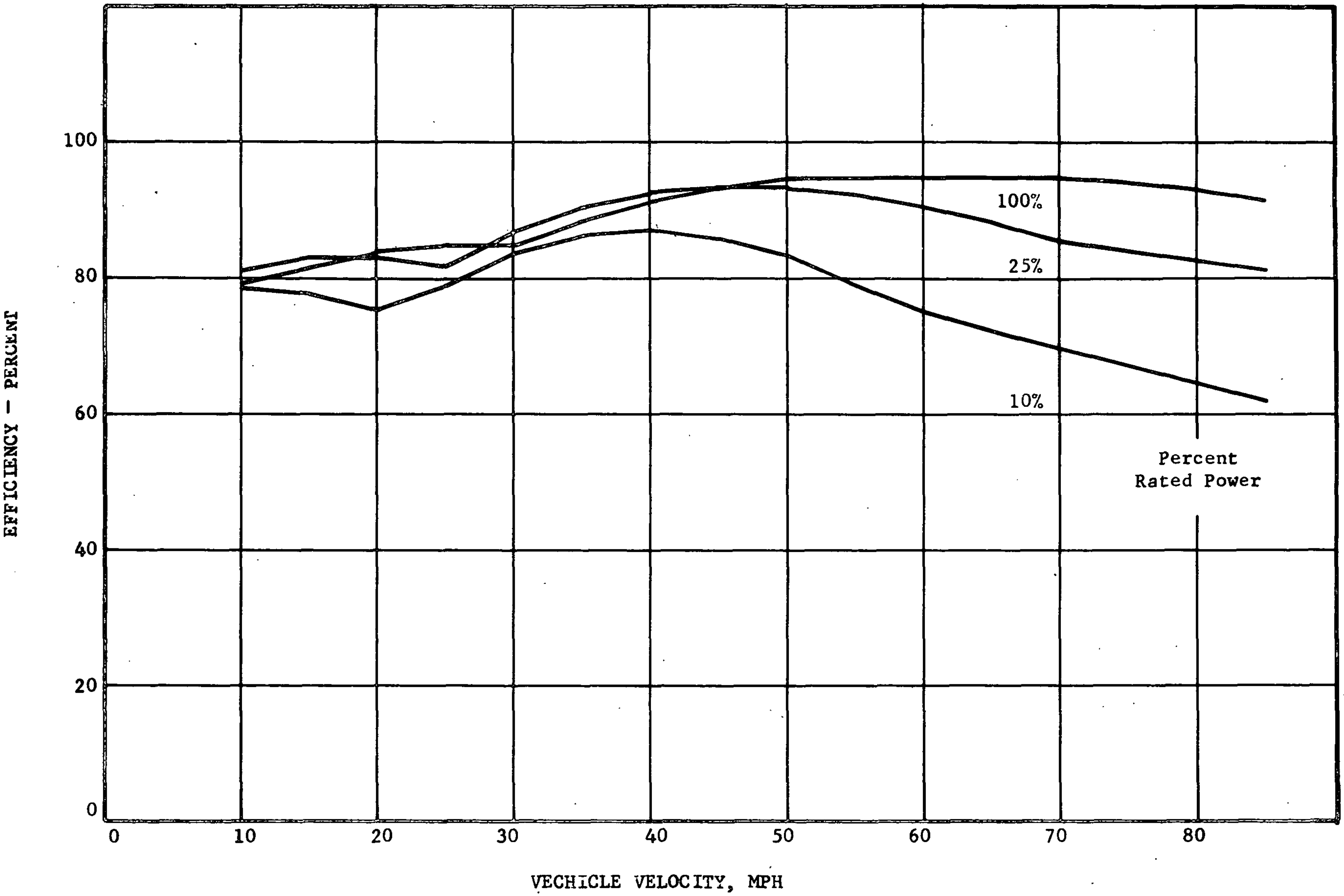

Fig. D-8 Constant Power Efficiency of Selected Transmission — AiResearch Engine

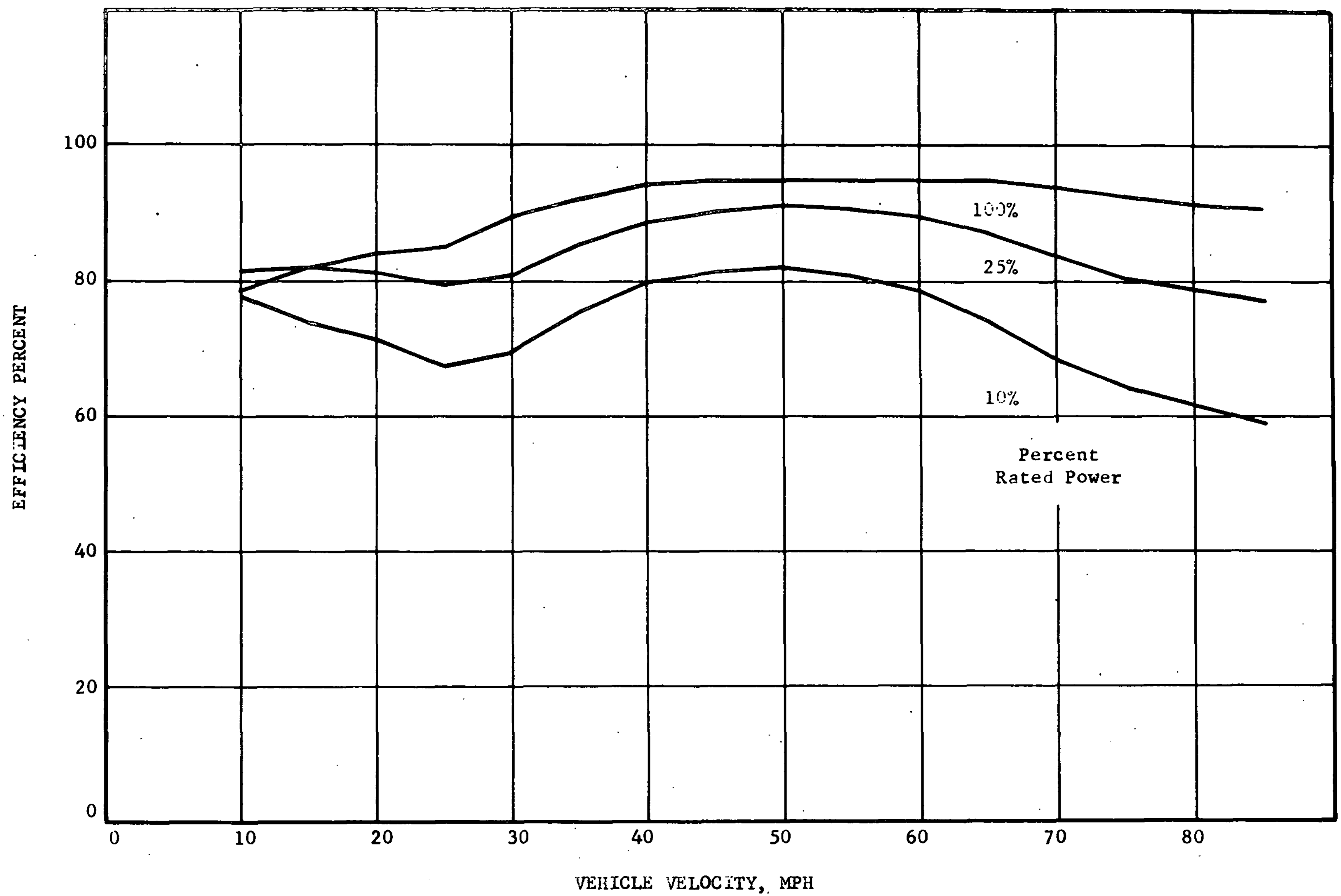

Fig. D-9 Constant Power Efficiency of Selected Transmission — Aerojet Engine

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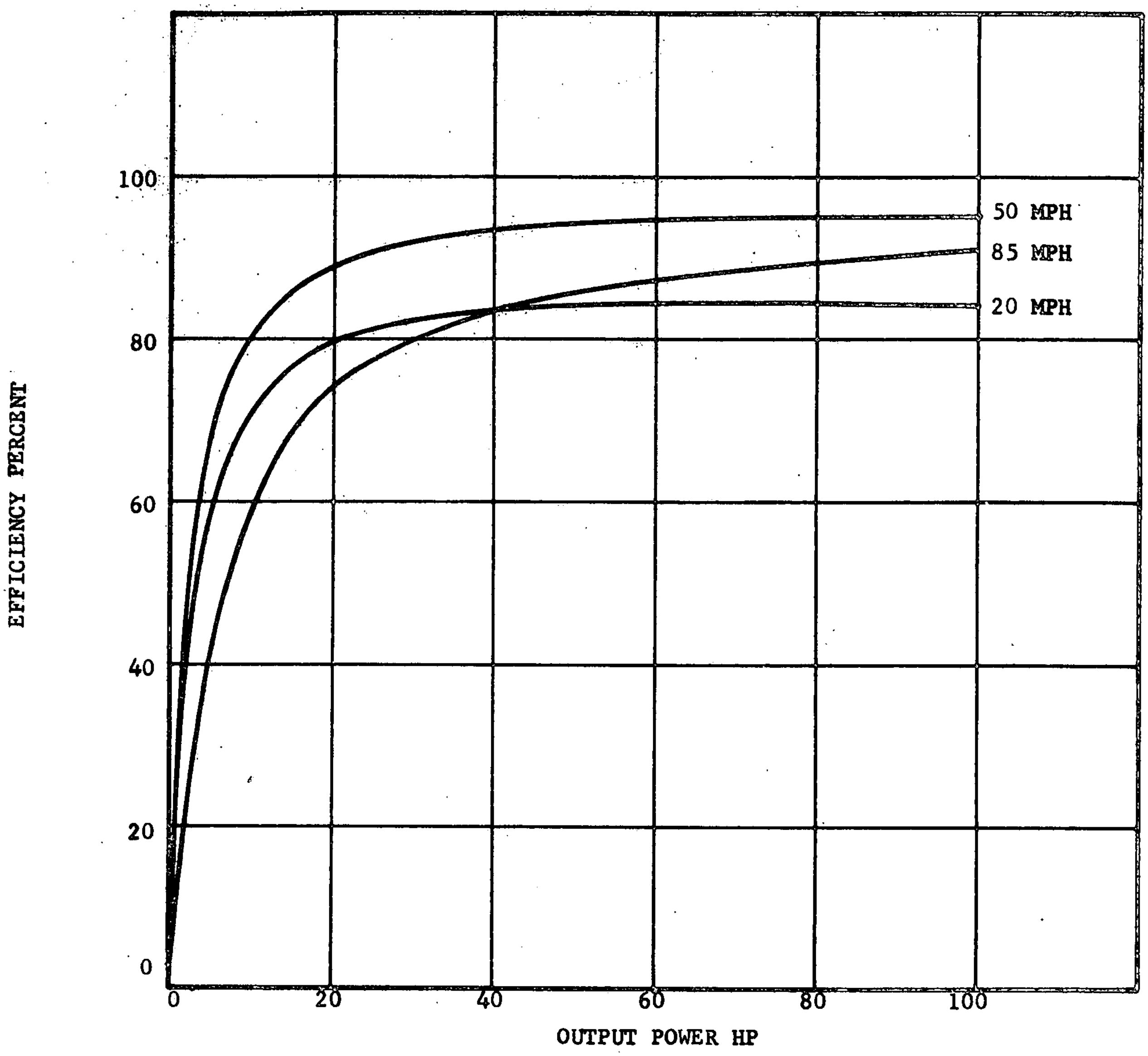

Fig. D-10 Constant-Speed Efficiency of Selected Transmission — Aerojet Engine

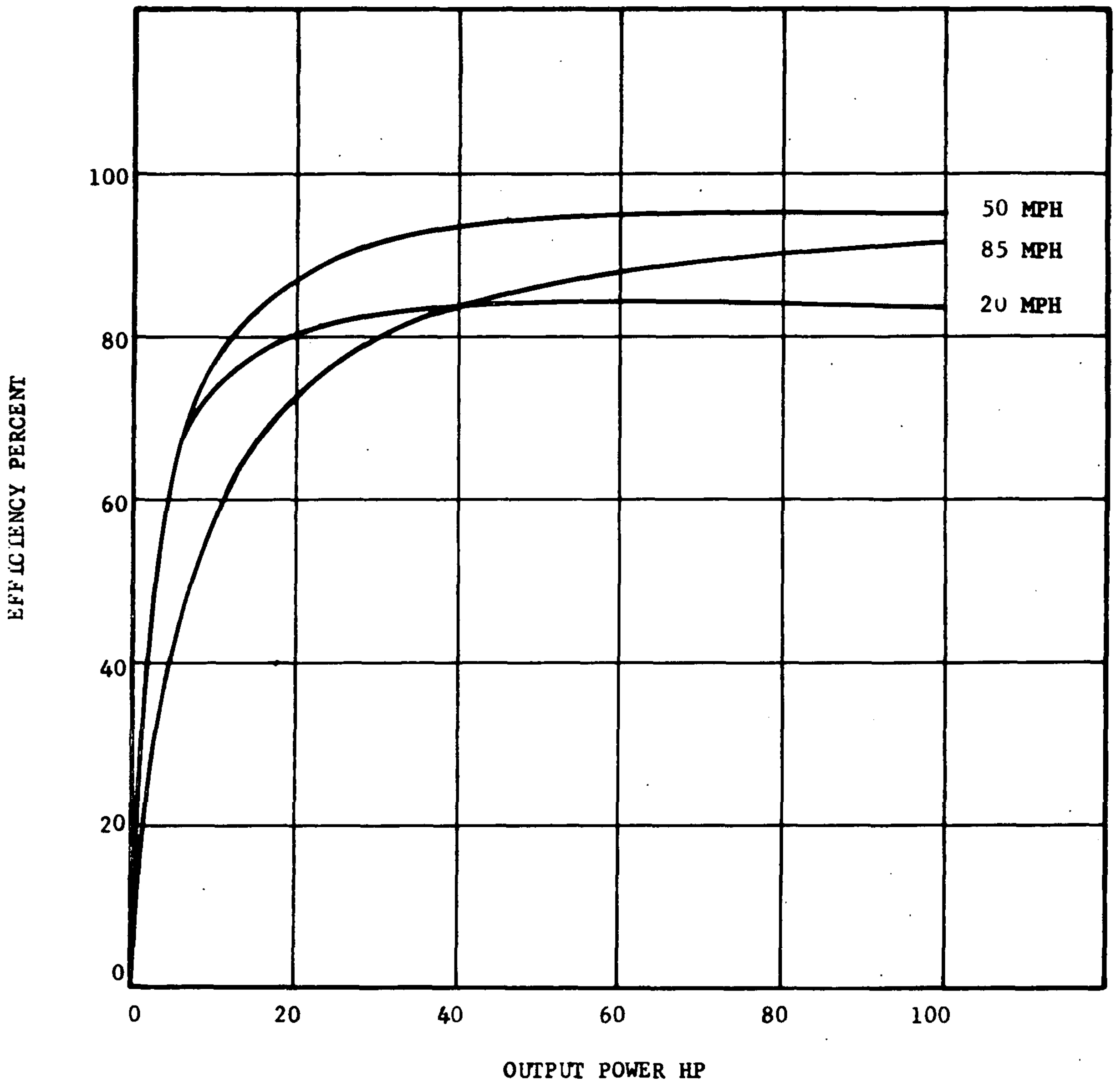

Fig. D-11 Constant-Speed Efficiency of Selected Transmission — AiResearch Engine

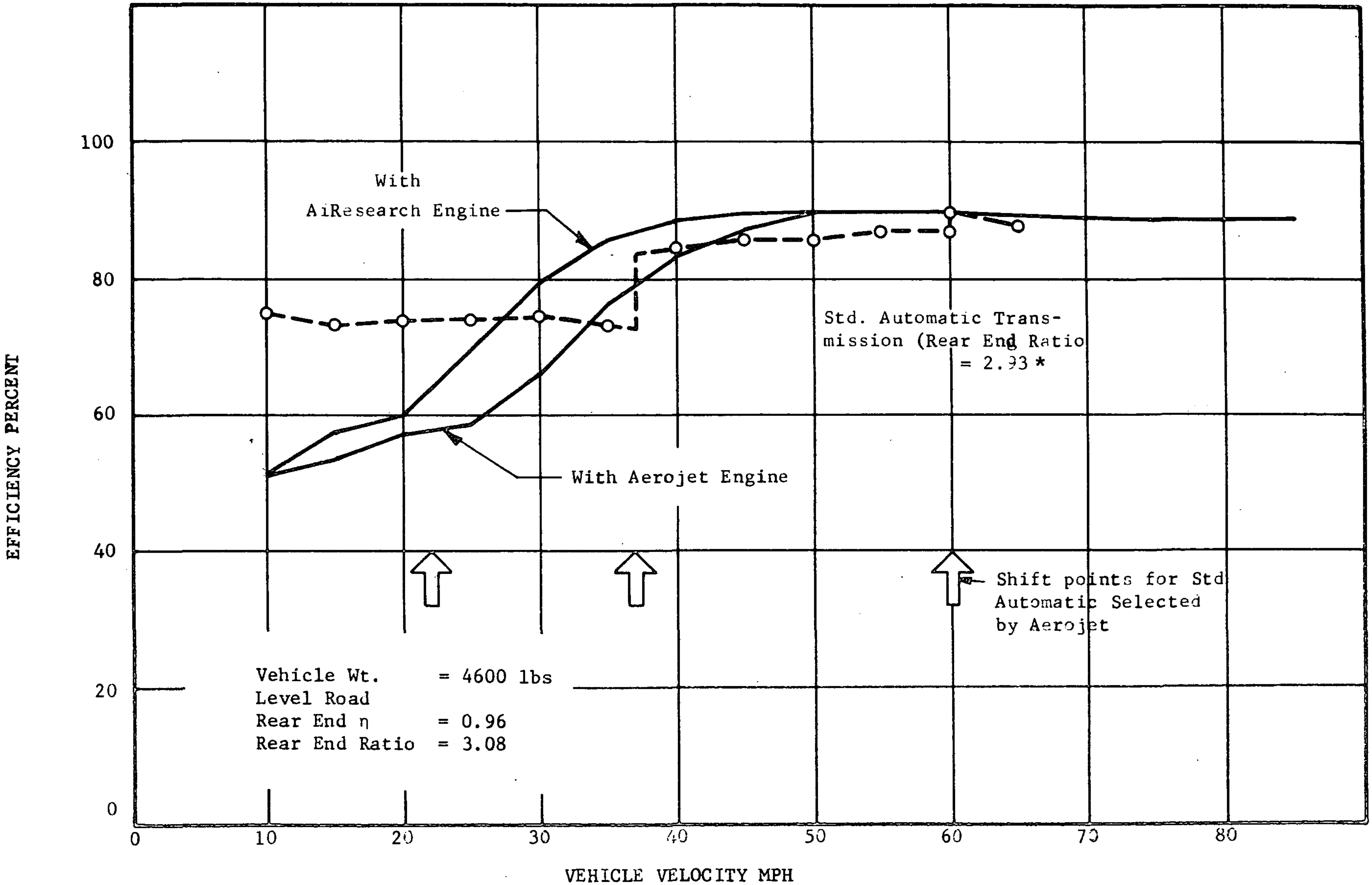

Fig. D-12 Comparison of Transmission Efficiency — Cruise Power

* Different Rear End Ratio for Standard Automatics selected by Aerojet to give optimum performance with remainder of power train (engine + transmission)

MTI-14632

shift points, even under the cruise conditions of Fig. 5, is necessitated when a vehicle speed range from zero to 85 mph is to be provided by an engine whose ratio of maximum to idle speed is well below 2. In fact the complete transmission incorporated by Aerojet includes a separate idle gear for use between 10 and 22 mph and a slipping clutch for speeds below 10 mph. Thus, in the range 10-85 mph, the so-called "conventional automatic" actually behaves as a 4-speed rather than a 3-speed.

Below 30 mph the conventional automatic is significantly more efficient than the selected transmission - the difference reaching 24 efficiency points at 10 mph for the Aerojet engine. The implications of this difference in efficiency are discussed subsequently.

Additional measures of behavior, or performance which reflect most clearly the operation of the selected hydromechanical transmission are the fraction of transmission input power which goes through the hydrostatic power path and the operating pressure in the hydraulic units.

Figures D-13 and D-14 show the variation of hydrostatic power with vehicle velocity for the two engines, respectively. In each case, up to the shift point, the majority of the power goes to the hydrostatic path - 99 percent under full power conditions and about 80 percent under cruise conditions. The small amount of power going to the mechanical power path under low-range conditions is that associated with mechanical path speed-dependent losses such as clutch friction.

At speeds above the shift point the fraction of power passing through the hydrostatic path falls rapidly to a minimum at the "straight-through" point. At speeds above the "straight-through" point the hydrostatic power increases again to about 35 percent at 85 mph.

The main differences in power split characteristics for the two engines can be associated with the difference in transmission shift behaviors. For the Aerojet engine the sharp reduction in hydraulic power corresponding to the shift point occurs at similar speeds, both for cruise and for 100 percent operation. However, for the AiResearch engine, the sharp reduction at the shift point occurs at 20 mph under cruise conditions and at 30 mph under 100 percent power. This difference is a result of the insensitivity to power level of the desired Aerojet

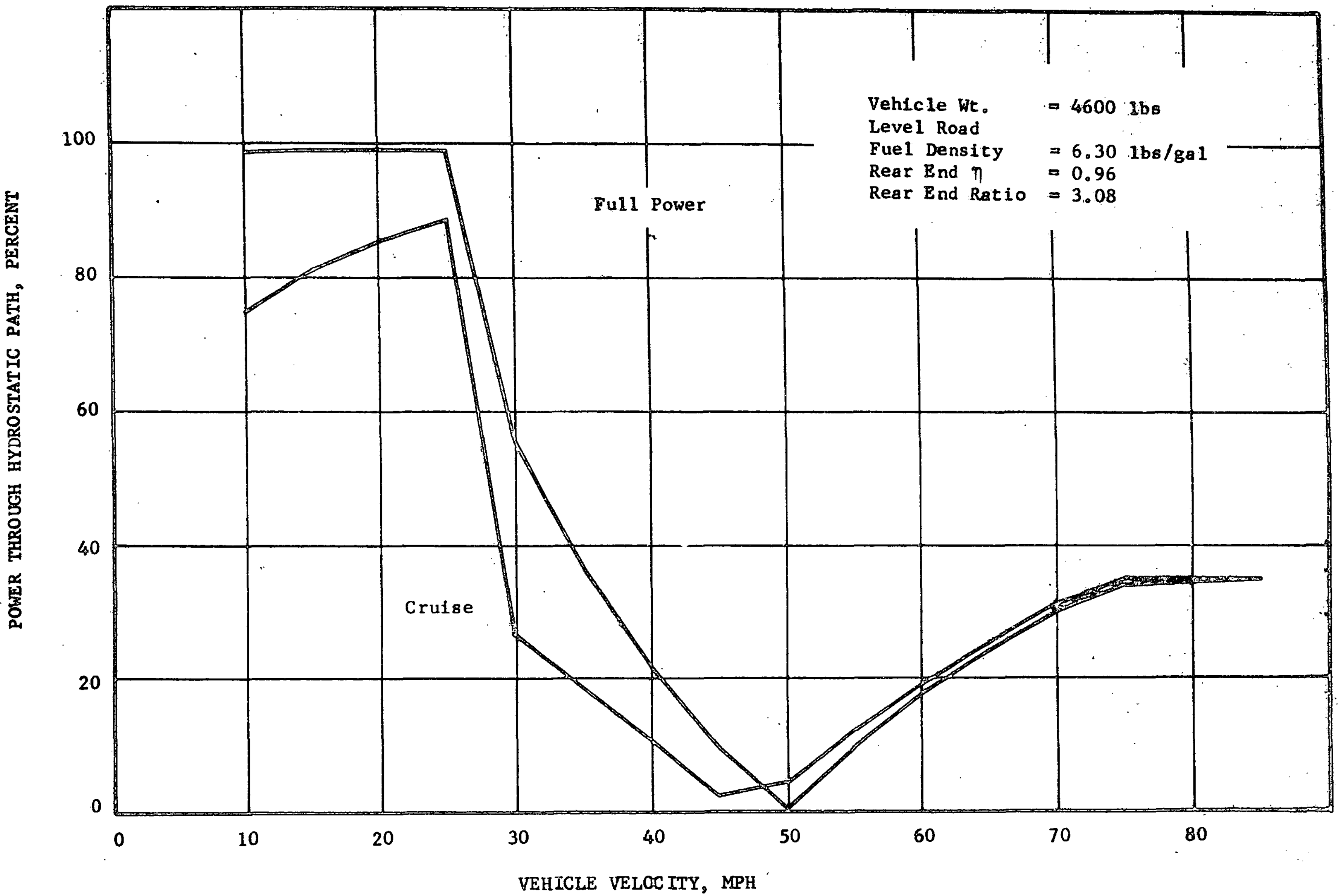

Fig. D-13 Power Through Hydrostatic Path of Selected Transmission with Aerojet Engine (with Air Conditioner)

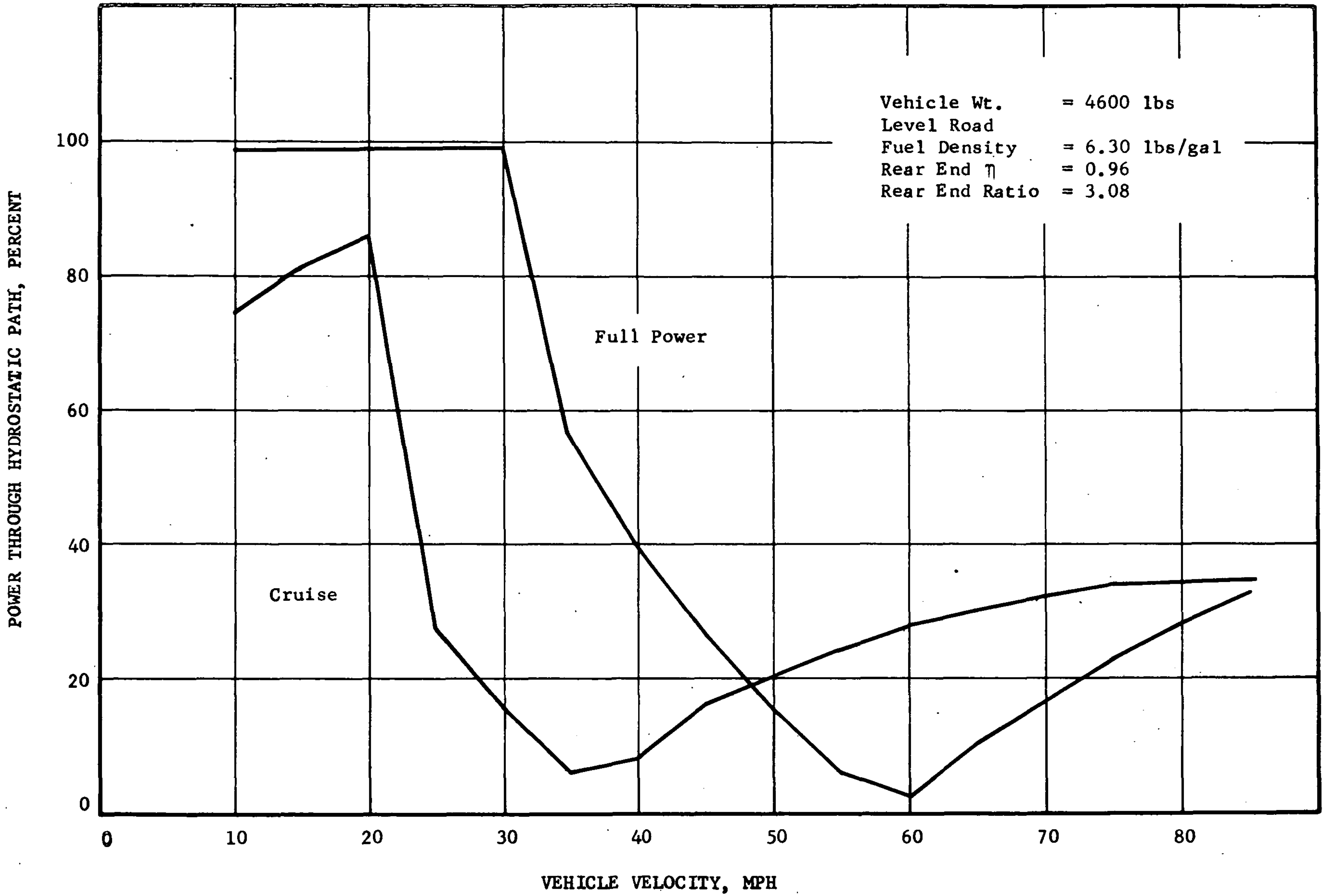

Fig. D-14 Power Through Hydrostatic Path of Selected Transmission with AiResearch Engine (with Air Conditioner)

MTI-14604

engine speed, and the corresponding sensitivity to power level of the AiResearch engine speed. Power level influences the shift point much less for the Aerojet engine than for the AiResearch engine.

Table D-2 provides a more detailed breakdown of the power flow and indicates the contribution of mechanical and hydraulic losses to performance. This has been done for the two vehicle speeds of 20 mph and 60 mph with the Aerojet engine. The first of these speeds is slightly below the shift point, as the second speed is somewhat above the "straight-through" point. In both cases the amount of power flowing through the hydraulic path is very similar. However, at 20 mph the only power flowing to the mechanical path is that necessary to overcome friction and to drive the charge pump - no output power is delivered by the mechanical path. The influence of this difference in power split is reflected in the percentage contribution of the hydraulic and mechanical losses. At 20 mph hydraulic losses account for 29.5 percent of the transmission input power, and mechanical losses account for 13.5 percent. At 60 mph the hydraulic losses fall to 5.2 percent and the mechanical losses to 3.9 percent.

Consider the operating pressure in the hydraulic elements. It is important to point out that maximum operating pressures above 3500 psi not only reduce the life (and reliability) of the hydraulic elements, but also cause unwanted excessive noise in the elements. Thus, low operating pressures are highly desirable to reduce transmission noise to a minimum.

Figure D-15 shows the variation in hydrostatic pressure as a function of vehicle speed for the selected hydromechanical transmission. It can be seen that, for cruise conditions, the operating pressure was below 400 psi. Note that the cruise pressure drops slightly following the shift point, then remains nearly constant up to the straight-through" point, and steadily increases in pressure up to a maximum of 330 psi at 85 mph. Under full-power demands the pressure remains close to its limiting value of 3500 psi at speeds of 10 and 15 mph, reflecting the design limit corresponding to wheel slip. At higher speeds, even in low range (17-30 mph) the full power can be transmitted through the hydrostatic path without exceeding the pressure limits, and the pressure begins to fall with speed. At a speed of 85 mph the full-power pressure has decreased to about 530 psi.

TABLE D-2

TYPICAL POWER FLOW BREAKDOWN — AEROJET ENGINE
WITH AIR CONDITIONER — CRUISE POWER

	20 MPH			60 MPH		
	AVAILABLE HP	HP USED & LOSSES	% OF TRANSMISSION INPUT HP	AVAILABLE HP	HP USED & LOSSES	% OF TRANSMISSION INPUT HP
ENGINE HP	13.114			39.908		
ACCESSORY HP		4.740	56.60		49.11	14.03
TRANSMISSION INPUT HP	8.374			34.996		
MECHANICAL PATH LOSSES		1.126	13.45		1.356	3.87
HYDRAULIC PATH LOSSES		2.472	29.52		1.814	5.18
TRANSMISSION OUTPUT HP	4.776			31.826		
DIFFERENTIAL LOSSES		0.184	2.20		1.224	3.50
ROAD HP	4.592			30.602		

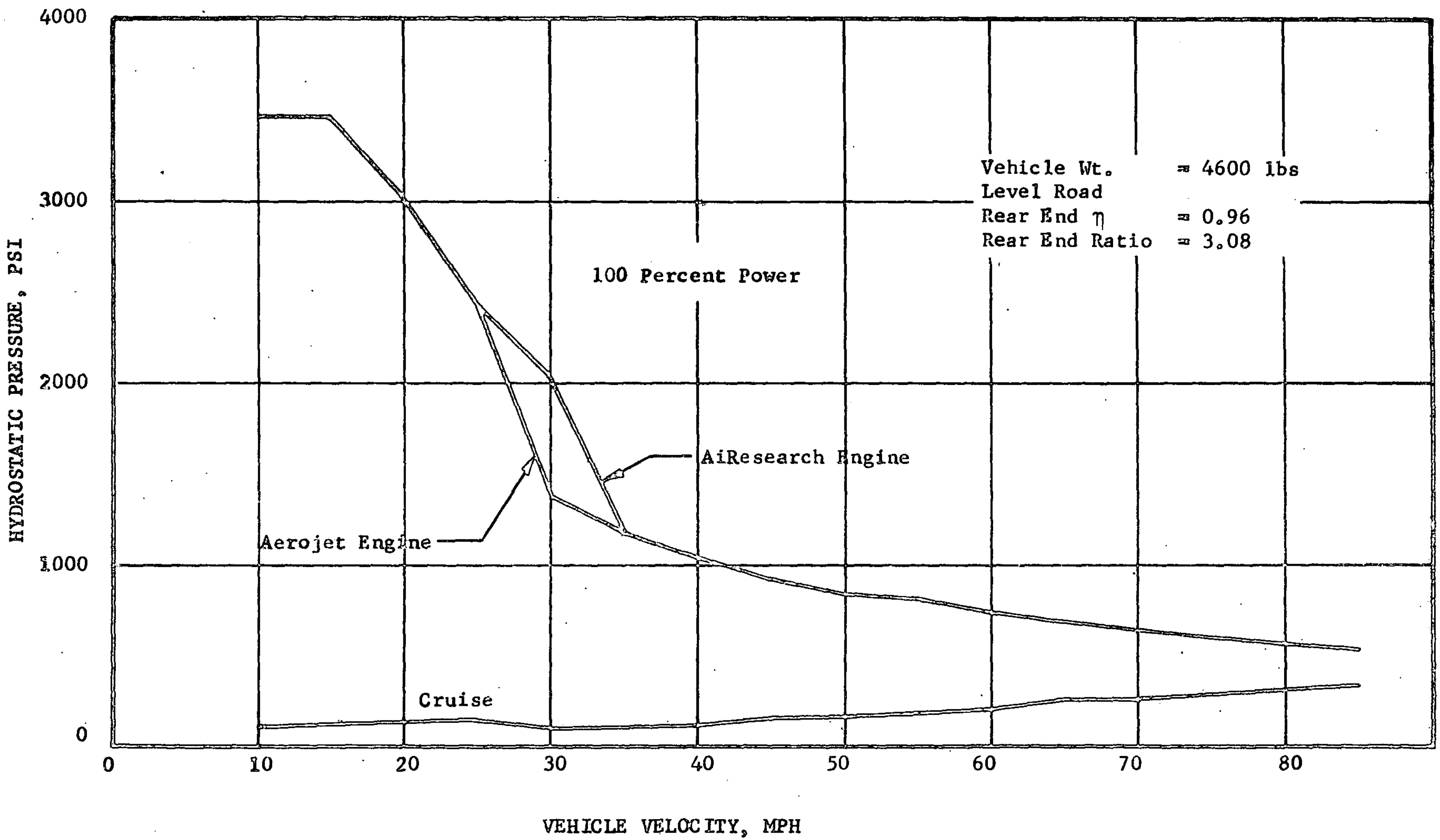

Fig. D-15 Hydrostatic Pressures for Selected Transmission

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The only significant difference in pressure level between the two engines occurs at the shift point as a result of the differences in the shift velocity for the two engines. Apart from this point, at which the transmission is actually in a different mode for the two engines, the pressure is dictated almost directly by the power demand at the wheels.

4. Power-Train Performance — Aerojet Engine

The fuel economy of the vehicle power-train, in MPG, is the most meaningful measure of the efficiency with which fuel is being converted into useful work. The cruise fuel economy for the Aerojet engine is shown in Figure D-16. The two different lines reflect the influence of the air conditioner on fuel economy, which can reach almost 3 mpg at low vehicle speeds, but falls to less than 0.5 mpg at high speeds.

The optimum fuel economy occurs at 35-40 mph. Here the Aerojet engine gives 15.4 mpg with air conditioner, and 17.4 mpg without air conditioner. The drop-off in fuel economy at low and high speeds is pronounced. At low speeds the main reason for the fall in fuel economy is the increase in significance of transmission losses and vehicle accessories. At high speeds the reason for the fall in fuel economy is the square-law dependence of air resistance and rolling resistance with vehicle velocity.

Aerojet has designated an automatic transmission for use with their engine. This transmission includes a 3-speed automatic gear box, a torque converter, and an idler gear which provides additional speed reduction between 10 mph and 22 mph. Using data for the efficiency of the 3-speed automatic gear box and torque converter as supplied by EPA/AAPS (3) and assuming 97 percent efficiency for the idler gear, the fuel economy of the Aerojet power-train employing this automatic transmission has been calculated and compared with that for the selected transmission.

The comparison is shown in Figure D-17, and demonstrates the discontinuities in performance associated with each shift point of the automatic. However,

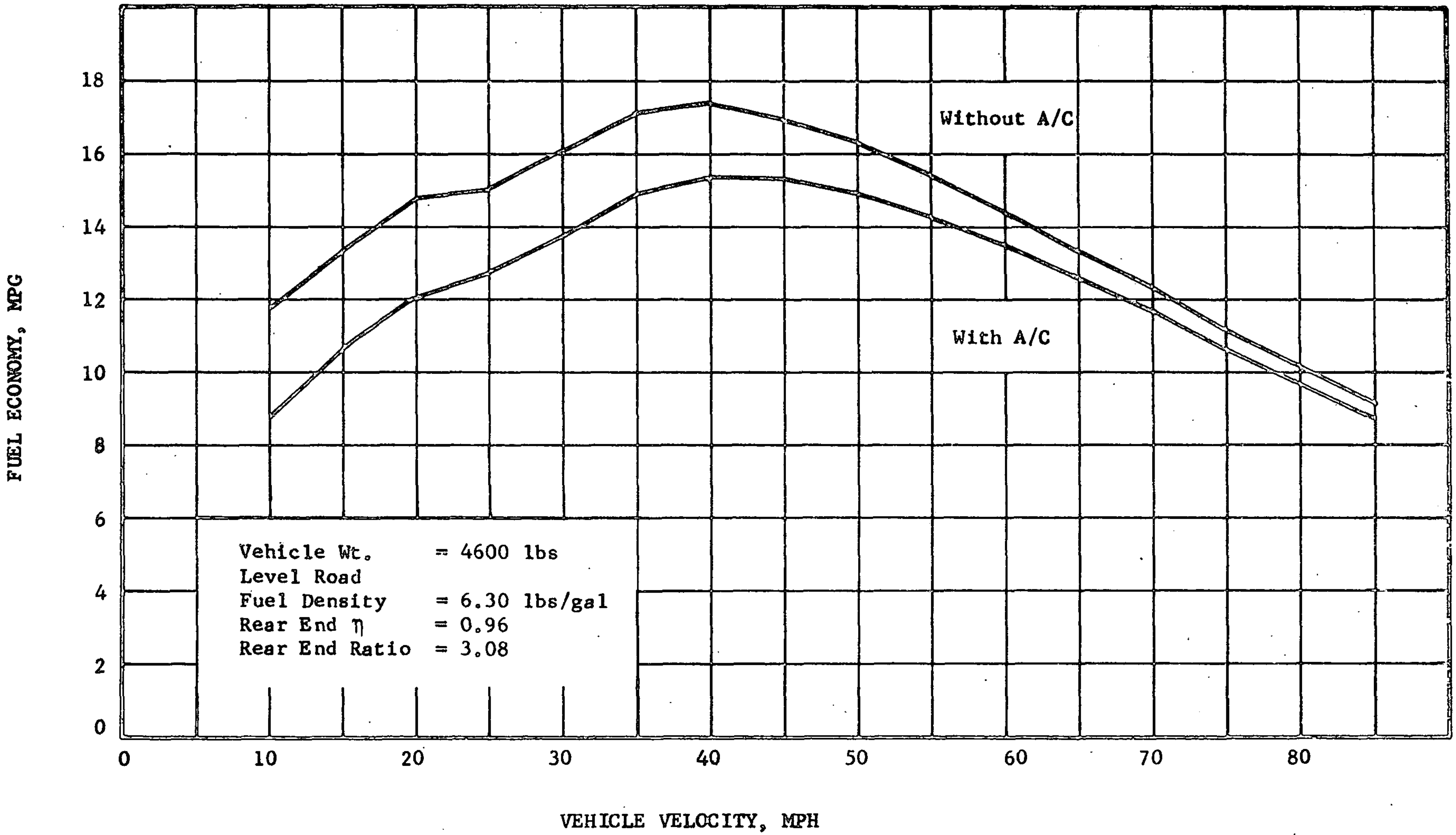

Fig. D-16 Cruise Fuel Economy with Selected Transmission — Aerojet Engine

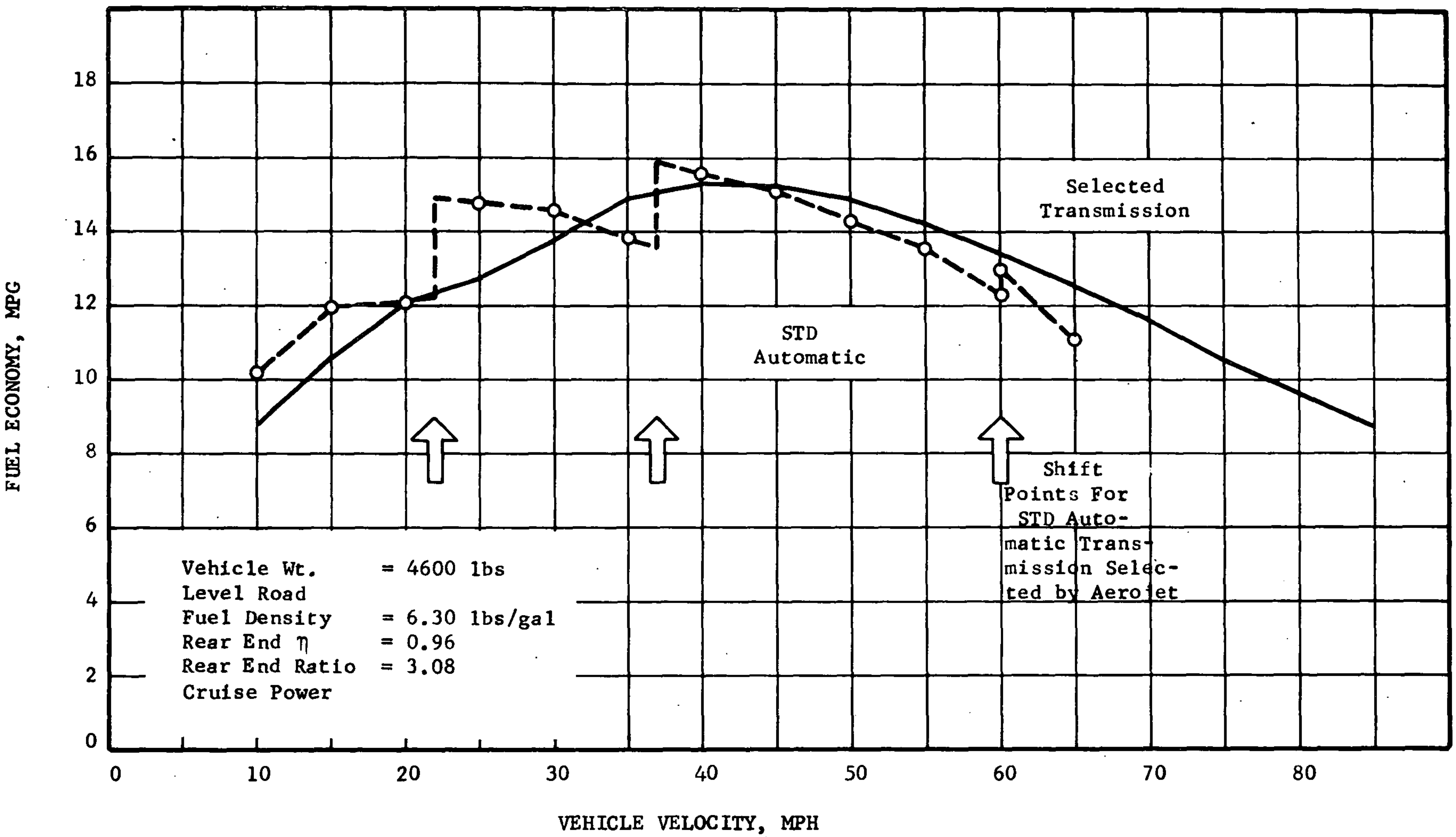

Fig. D-17 Comparison of Fuel Economy — Aerojet Engine with Air Conditioner — Cruise Power

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by comparing average levels, it is apparent that, at low speeds, the fact economy is slightly better with the automatic and, at high speeds, the fuel economy is slightly better with the selected transmission. The predominant reason for this similarity in power-train performance is the flat, symmetrical nature of the Aerojet engine performance map. Thus, typically, a 500 rpm deviation from the optimum engine speed, either up or down, causes only about a 1 out of 17 deviation in engine efficiency. At low vehicle speeds the higher efficiency of the automatic transmission actually results in higher fuel economy when the engine is operating near its minimum SFC point.

As an extreme illustration the power-train performance with a 100 percent efficient hydromechanical transmission was calculated. The comparison of power-train performance with the automatic transmission and the idealized hydromechanical transmission is presented in Figure D-18.

The conclusion from this comparison is that, even with a perfectly efficient transmission and the engine always operating at maximum efficiency, 17.9 mpg is the best cruise fuel economy which can be achieved - that is only 3.0 mpg or 20 percent better than with the standard automatic.

The ability of the selected hydromechanical transmission to keep the engine operating at minimum SFC is demonstrated in Figure D-19. The solid lines on this plot represent the engine SFC with and without air conditioner. The engine SFC with the conventional automatic is superimposed on this plot and only below 15 mph does the SFC with conventional automatic fall below that with the selected transmission. The exception occurs because, at 10 mph, the standard transmission actually allows the engine to operate closer to the minimum SFC line then the selected transmission.

The Federal Driving cycle provides an alternative operating condition under which to measure power-train performance. On the basis of the cruise performance, it is not to be expected that the driving cycle will reveal any significant performance advantages for the selected hydromechanical transmission. Table D-3, which summarizes the Aerojet driving-cycle performance, and compares it with available information for the automatic transmission confirms this expectation.

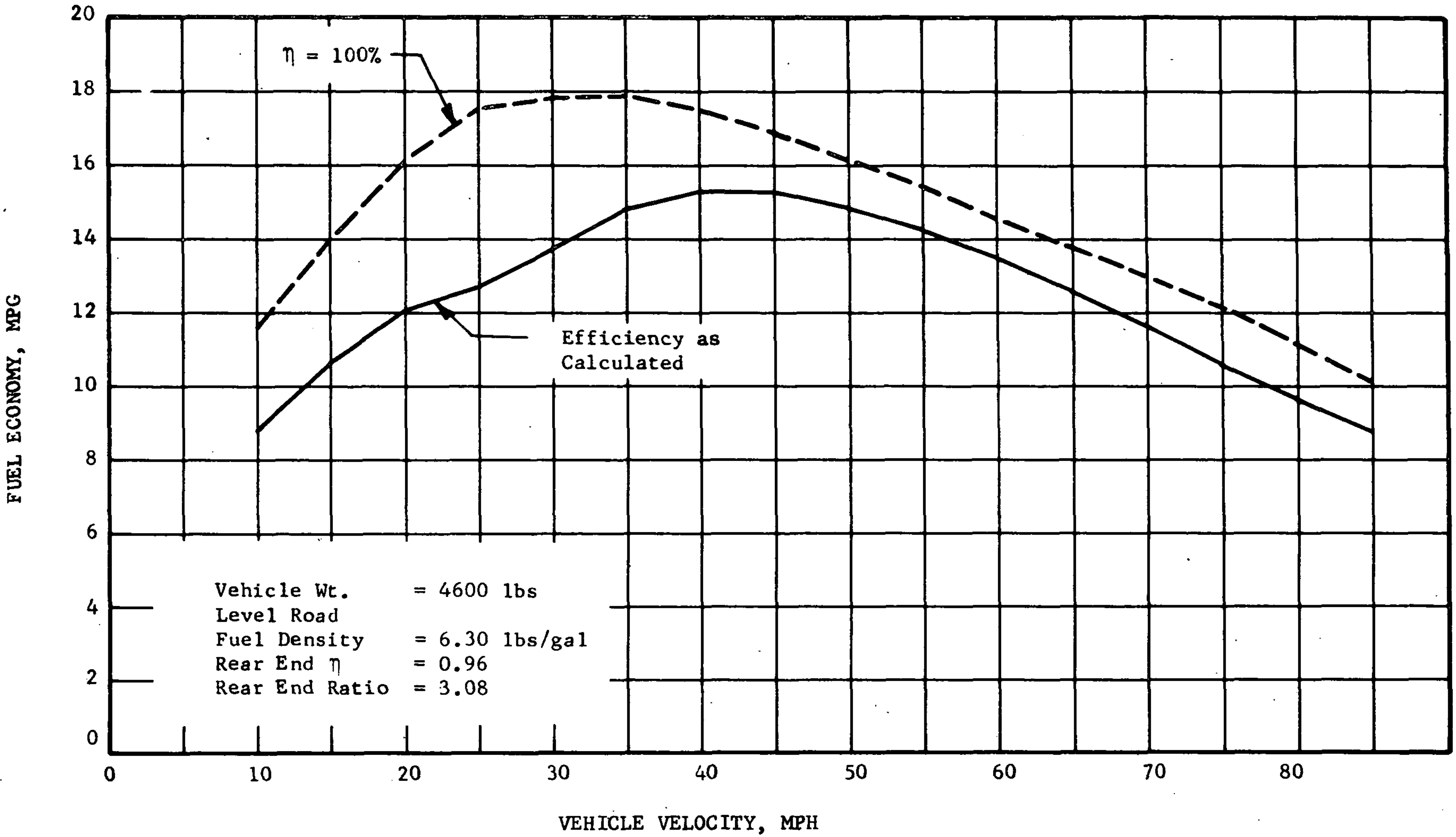

Fig. D-18 Effect of Transmission Efficiency — Aerojet Engine with Air Conditioner

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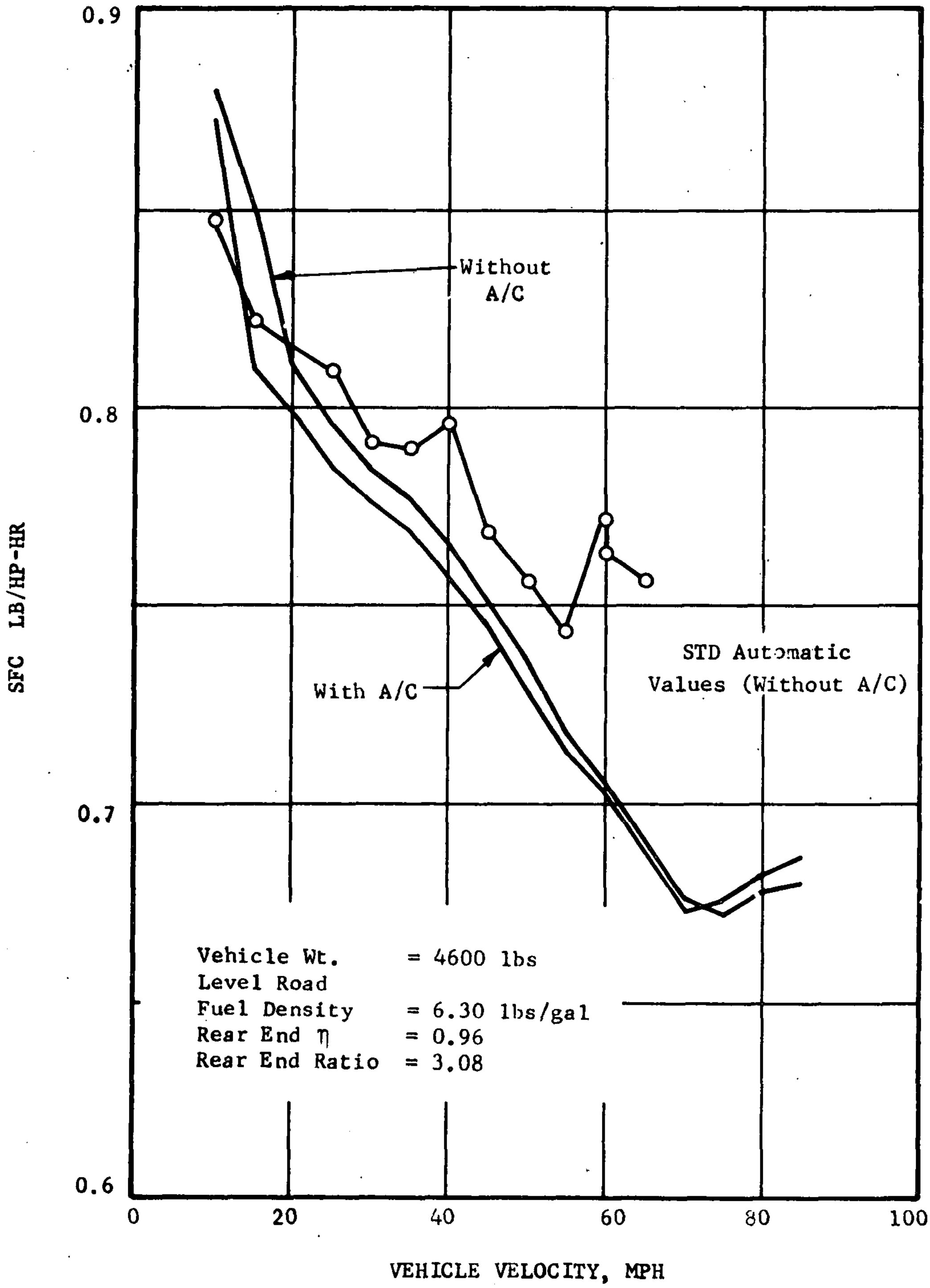

Fig. D-19 Specific Fuel Consumption with Selected Transmission – Aerojet Engine

TABLE D-3

AEROJET ENGINE — DRIVING-CYCLE PERFORMANC

Quantity	Selected Transmission With A/C	Selected Transmission Without A/C	100% η Transmission With A/C	Automatic* Transmission With A/C
Average MPG	9.55	10.85	12.02	9.9
Average Transmission	71%	71%	100%	-
Average Engine Power	16.76 hp	14.43 hp	13.16 hp	-
Average Road Powe	8.38 hp	8.38 hp	8.38 hp	
Average Velocity	19.6 mph	19.6 mph	19.6 mph	

Note* Data Provided by Aerojet. (Transmission efficiency and engine power data not available)

On the basis of this data, the Aerojet engine provides marginally better performance with the automatic transmission than with the selected transmission. It is of benefit to rationalize this conclusion as follows.

The average driving cycle vehicle velocity is 19.6 mph. On the basis of cruise performance at 20 mph the automatic transmission might be expected to provide significantly better fuel economy (see Figure D-17). However, it should be noted that the average road power over the driving cycle (8.38 hp) is 83 percent higher than the cruise road power at 20 mph (4.59 hp). This higher average power level results in a higher average transmission efficiency (71 percent vs 62 percent) for the selected transmission, and is the reason why the selected transmission gives very similar driving-cycle fuel economy in comparison with the conventional automatic transmission.

The effect of a perfectly efficient hydromechanical transmission is to give an average fuel economy of 12.02 mpg with air conditioner - a gain of 2.47 mpg (25 percent) relative to performance with the actual transmission efficiencies.

Consider now the full-power acceleration performance of the Aerojet engine with the selected hydromechanical transmission. As shown by Table D-4, the power-train exceeds all of the EPA/AAPS maneuver specifications.

TABLE D-4

AEROJET MANEUVER PERFORMANCE

	Maneuver	EPA Specifications	Aerojet Engine with Selected Transmission
1.	Distance travelled in 10 seconds	440 ft.	505 ft
2.	Time to reach 60 mph from standing start	13.5 sec.	11.7 sec.
3.	High speed merge (25-70 mph)	15.0 sec.	13.5 sec.
4.	DOT passing maneuver (time and distance to overtake 50 mph truck)		
	TIME	15.0 sec.	12.2 sec.
	DISTANCE	1400 ft.	1166 ft.

Thus, no special optimization of the transmission, differential or overall power-train is necessary to satisfy these full-power vehicle performance requirements. They are achieved without a requirement to exceed the maximum temperature or 100 percent speed.

Typical time domain plots of engine velocity (rpm), vehicle velocity (ft/sec), and distance traveled are shown in Figures D-20, D-21, and D-22, respectively. The engine speed plot, in particular, reflects the imposed control law described in Section C - Description of Methods for Determining Performance. During region I, in which the engine alone is accelerated to 100 rpm above idle, the high power available and relatively low engine inertia make this an almost instantaneous acceleration. During region II, in which the vehicle is accelerated from a standing condition to a speed relative to the engine corresponding to the maximum transmission ratio, the increase in engine speed is greatly slowed, taking 1.1 seconds to provide a 200 rpm increase in engine speed. The reason for this slowing is that the vehicle inertia, as seen by the engine, is increasing and that there is an energy increase associated with the increase in inertia itself. During region III the engine and vehicle are accelerated together at a constant transmission ratio. This again results in a very rapid (0.3 sec) increase in engine velocity from 2300 to 2970 rpm. Finally, in region IV the constant engine speed is seen.

It is to be noted that, in spite of the sharp discontinuities in slope shown by the engine speed variation, the vehicle acceleration of the vehicle is limited to a value corresponding to wheel slip.

The distance vs time curve again closely confirms the maneuver results of Table D-4. The relatively flat power-speed curve of the Aerojet engine allows very rapid engine acceleration as discussed above. As a result, the vehicle responds early and the resultant distance covered, $\int V\,dt$, in 10 seconds was 505 ft or 15 percent further than the EPA/AAPS specification.

5. Power-Train Performance — AiResearch Engine

The resultant fuel economy of the AiResearch engine with the selected hydro-mechanical transmission is shown by Figure D-23. Peak performance occurs at 40 mph, at which speed the fuel economy is 26 mpg with air conditioning,

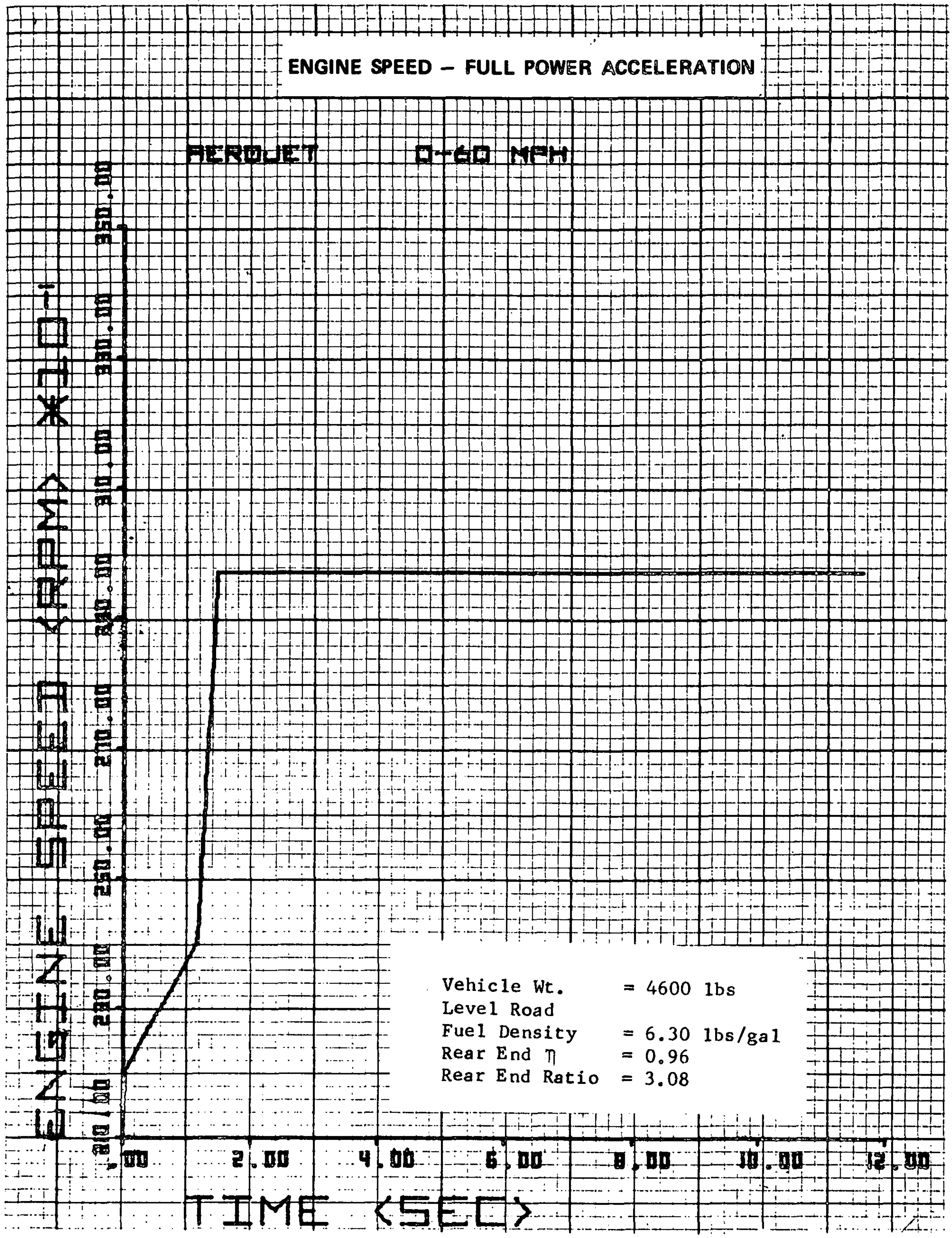

Fig. D-20. Engine Speed Under Full-Power Acceleration With Selected Transmission (0-60 MPH) - Aerojet Engine

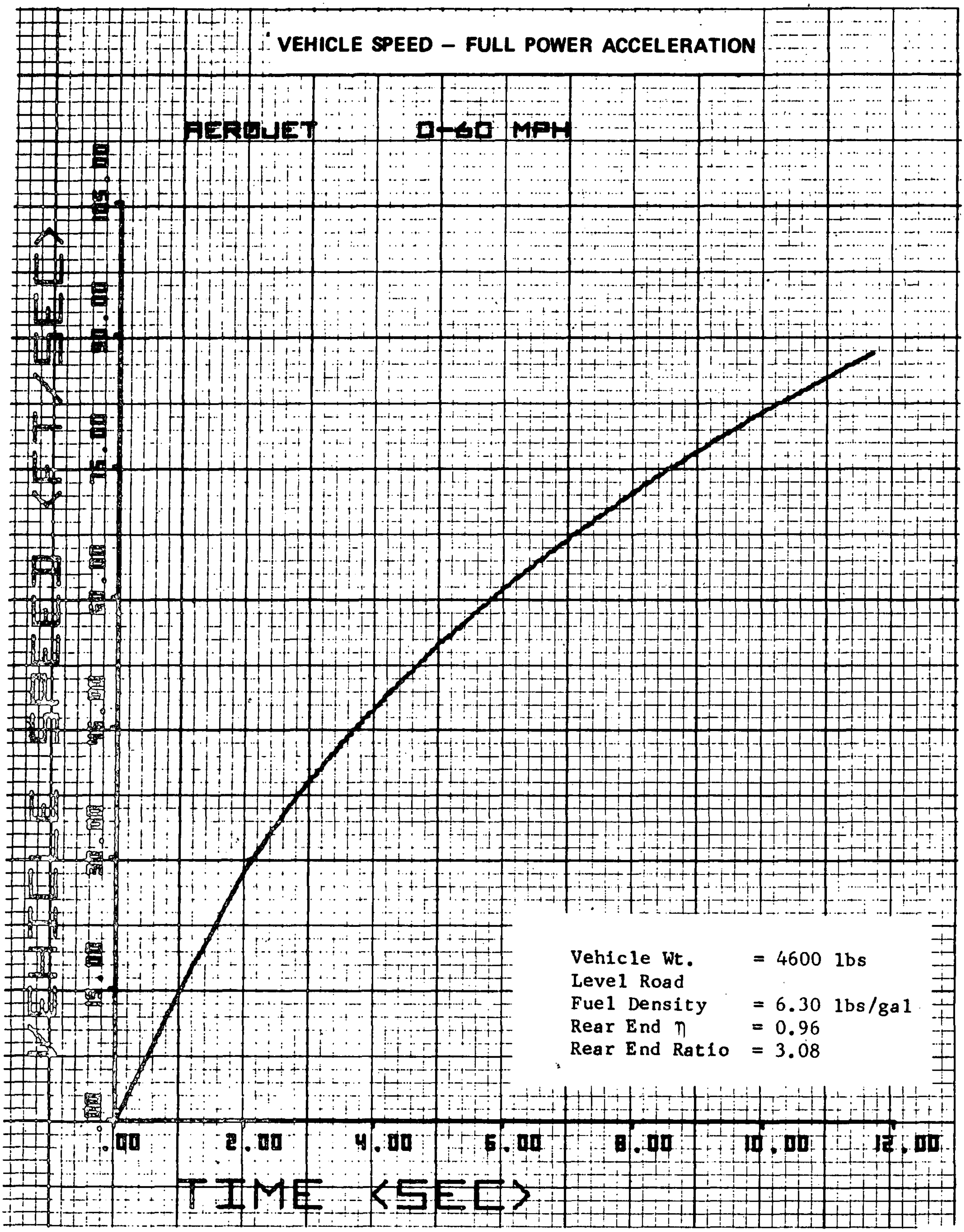

Fig. D-21. Vehicle Velocity Under Full-Power Acceleration With Selected Transmission (0-60 MPH) — Aerojet Engine

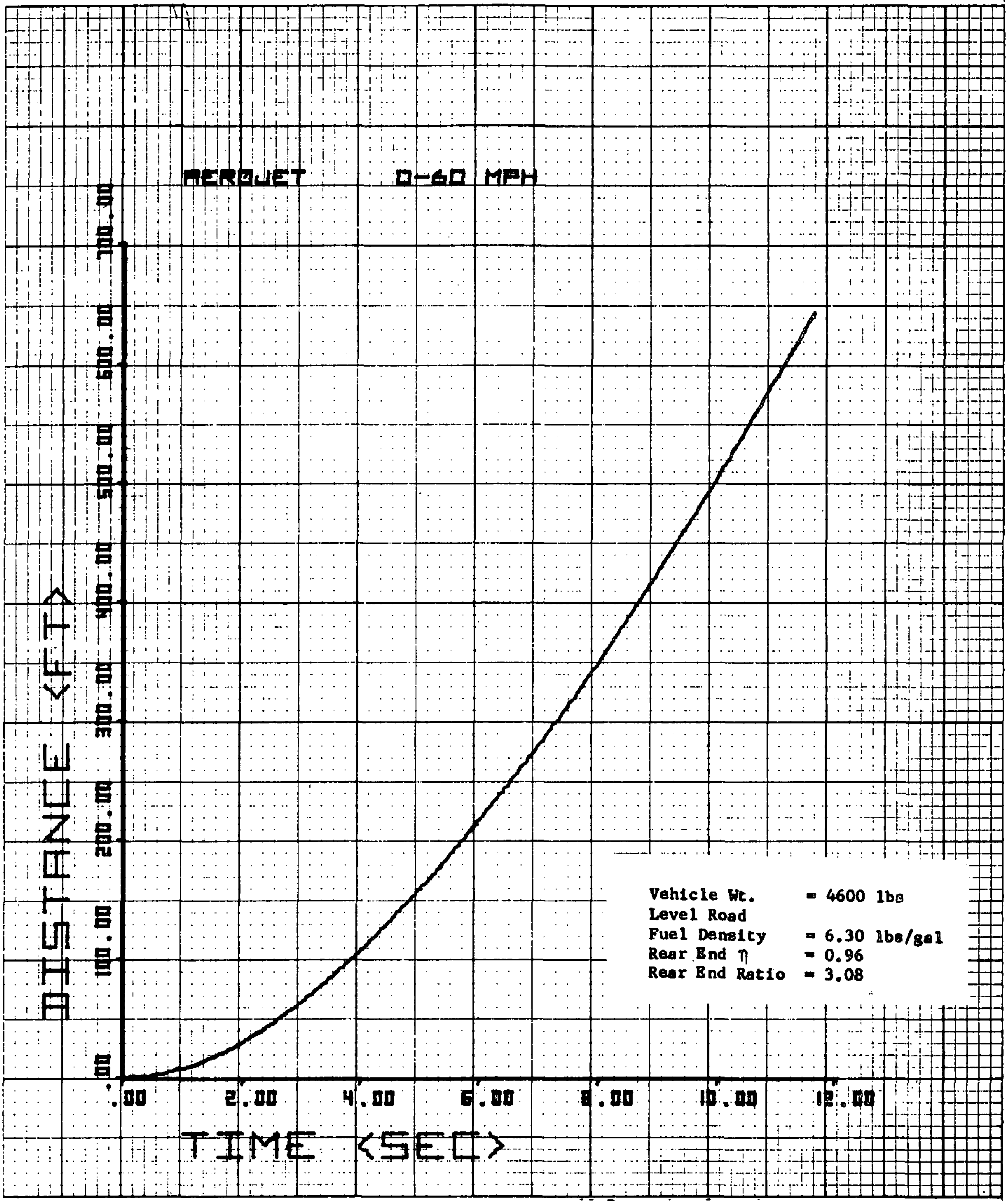

Fig. D-22 Distance Travelled Under Full-Power Acceleration with Selected Transmission (0-60 MPH) — Aerojet Engine

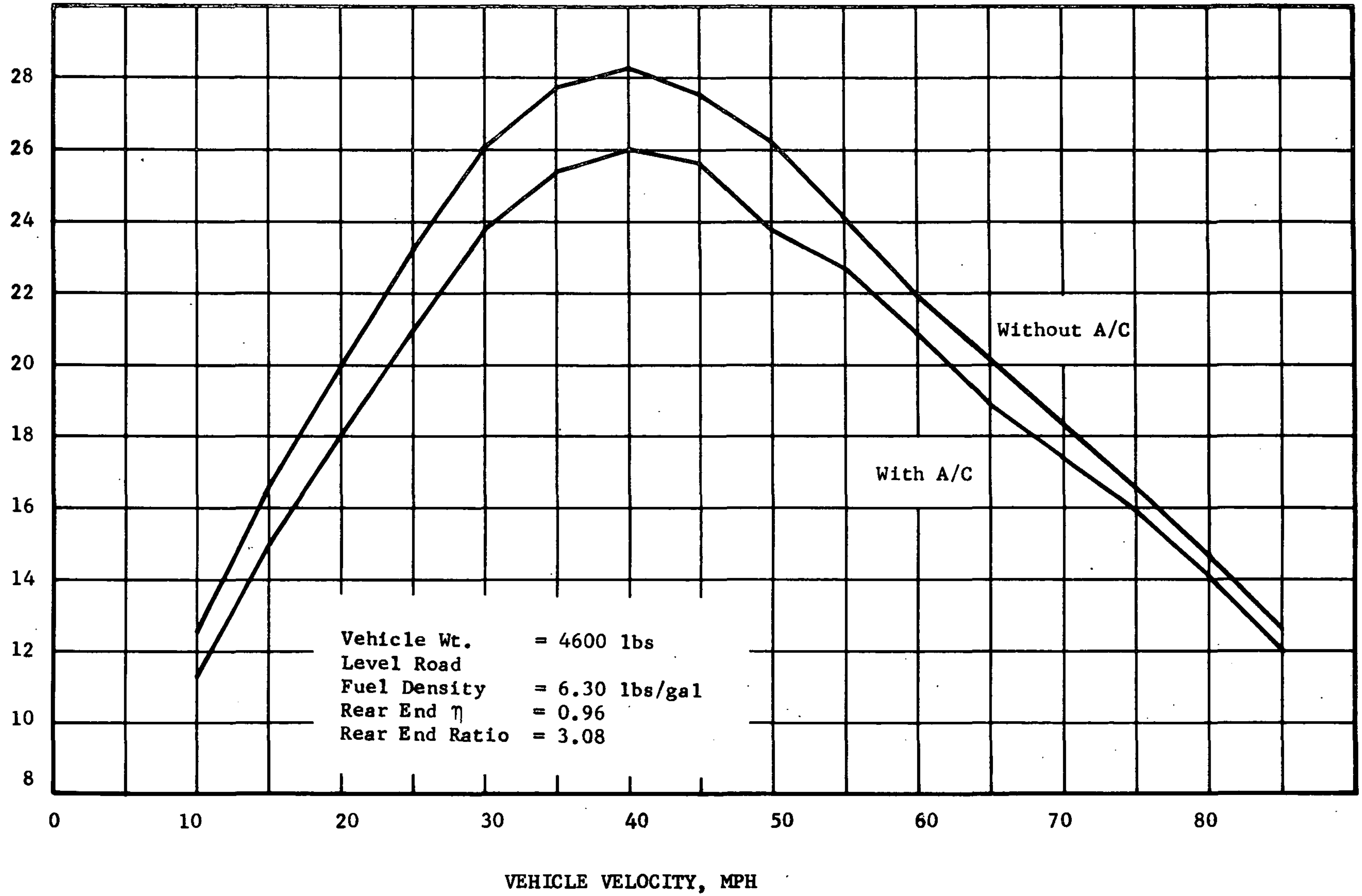

Fig. D-23 Cruise Fuel Economy with Selected Transmission — AiResearch Engine and 28.3 mpg without. It is noticeable that air conditioning has less of an effect than with the Aerojet engine. This difference is attributable to the lower operating speed of the AiResearch engine, and the correspondingly lower accessory power level.

The power-train performance of a vehicle incoprorating the same automatic transmission as discussed for the Aerojet engine has been calculated. It is recognized that this transmission has not been optimized in any way for the single-shaft, gas turbine engine and, indeed that a conventional automatic could never provide satisfactory kinematic performance with this engine. However, the calculation does provide an exaggerated demonstration of the benefits of the selected transmission for single shaft gas turbine application. The comparison of the fuel economy for the two transmissions with the AiResearch engine is shown in Figure D-24. Clearly the selected transmission offers considerable advantages in this application. The reason for the poor power-train performance with the conventional automatic transmission is the extreme sensitivity of the single-shaft, gas-turbine SFC to engine speed for a given power demand.

This is confirmed in Figure D-25 in which specific fuel consumption is plotted as a function of vehicle velocity. In most cases the selected transmission produces substantially lower specific fuel consumption than the standard automatic.

The performance over the Federal driving cycle of the AiResearch engine train with the selected transmission is presented in Table D-5.

TABLE D-5

DRIVING CYCLE PERFORMANCE - AIRESEARCH ENGINE

Quantity	Selected Transmission With A/C	Selected Transmission Without A/C
Average MPG	14.53	15.76
Average Transmission η	74.4	73.5
Average Engine Power	15.94 hp	13.92 hp
Average Road Power	8.38 hp	8.38 hp
Average Velocity	19.6 mph	19.6 mph

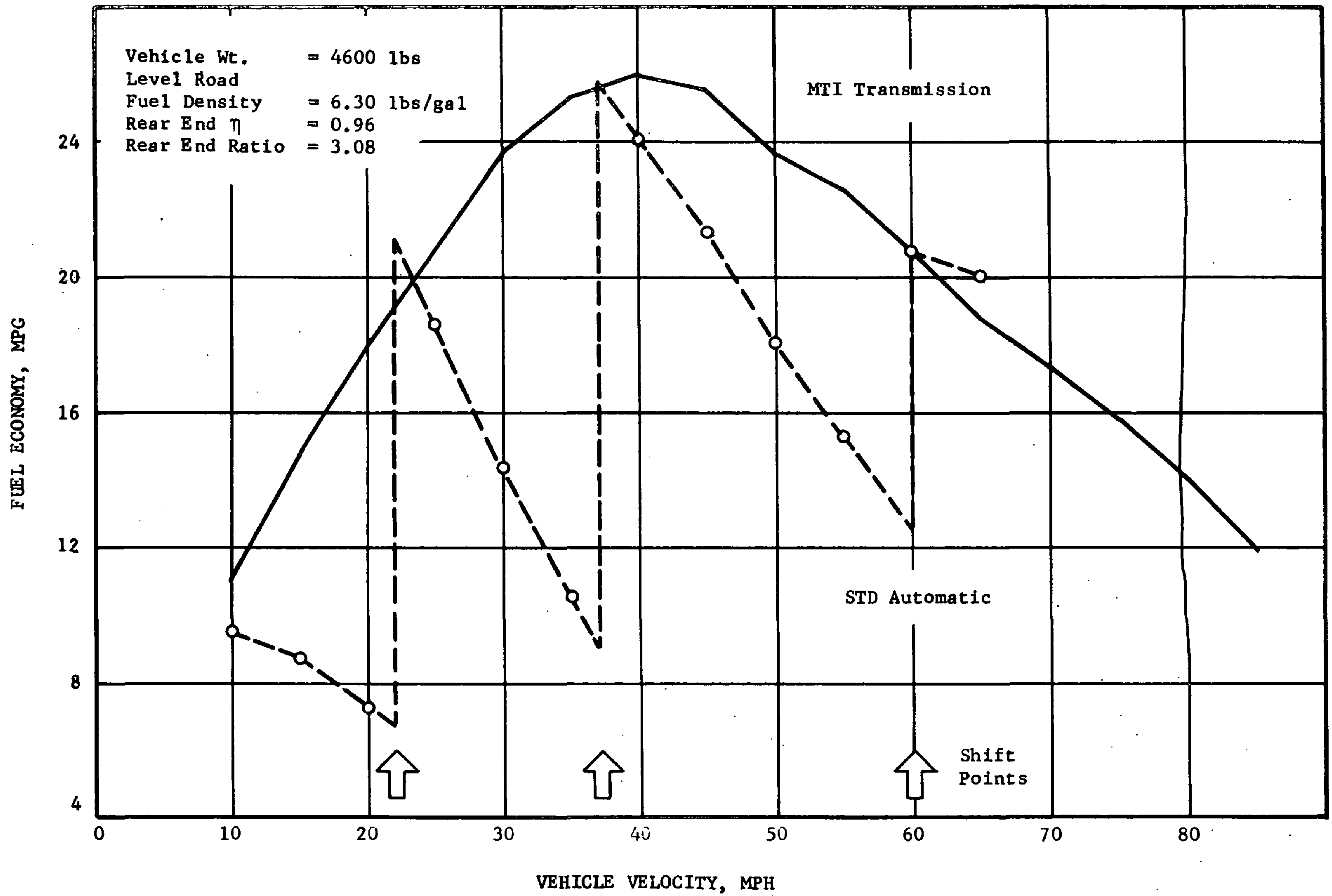

Fig. D-24 Comparison of Fuel Economy — AiResearch Engine with Air Conditioner — Cruise Power

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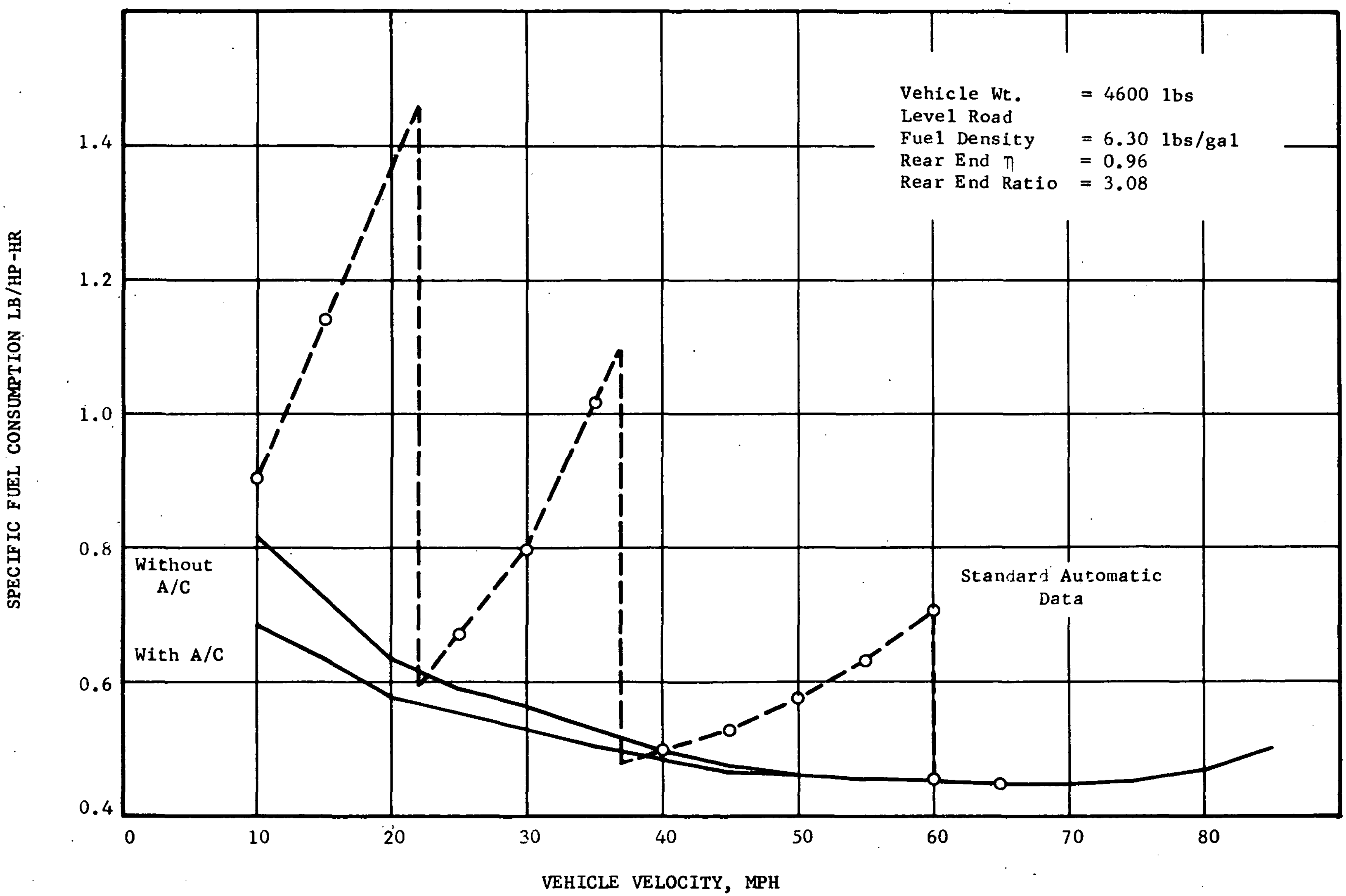

Fig. D-25. Specific Fuel Consumption With Selected Transmission — AiResearch Engine

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For operation both with and without air conditioner, the average driving-cycle fuel economy is close to 5 mpg less than the corresponding cruise fuel consumption. The air conditioner changes the average fuel economy by 1.4 mpg or 9 percent. No data are available for performance of the AiResearch engine over the driving cycle with a standard automatic transmission.

Full-power performance predictions with the AiResearch engine are given in Table D-6 below.

TABLE D-6

AIRESEARCH MANEUVER PERFORMANCE

Maneuver	EPA Specifications	AiResearch Engine
1. Distance traveled in 10 seconds	440 ft.	447 ft.
2. Time to read 60 mph from standing start	13.5 sec.	11.1 sec.
3. High speed merge (25-70 mph)	15.0 sec.	11.6 sec.
4. DOT passing maneuver (time and distance to overtake 50 mph truck)		
Time	15.0 sec.	11.8 sec.
Distance	1400 ft.	1139 ft.

As discussed in relation to the engine data, the AiResearch engine power level has been scaled up in order to meet these maneuver specifications. The most critical maneuver requirement was found to be the distance traveled in 10 seconds; it may be seen that all other requirements are very comfortably met by the power-train. The reason for this distance problem is the nature of the variation with speed of available power with the AiResearch engine. As shown in Figure D-3 the available power falls off very sharply at engine speeds below 100 percent. Thus, the velocities reached in the first few seconds of the acceleration are low, as demonstrated in Figure D-26, which gives vehicle velocity as a function of time, in a 0-60 mph acceleration. During the latter part of the acceleration maneuver the higher maximum power of the scaled-up AiResearch engine produces higher accelerations so enabling it to meet the 0-60 mph requirements.

It is noted that Rosbach (5) shows the same AiResearch engine, unscaled, as being able to meet the distance requirement. However, close examination of this reference reveals that an idle speed equal to 70 percent of design speed was used. The present studies used a value of 60 percent, and showed that acceleration of the engine up to 70 percent speed took almost 0.8 seconds, and that the vehicle moved only 4 feet in this period (see Figures D-26 and D-27). In terms of distance traveled in 10 seconds the increased idle speed used by Reference (5) will result in approximately 65 additional feet. However, the penalties for using this increased idle speed are substantial. As an example, a 300 rpm increase in the minimum engaged speed was found to cause over 4 mpg decrease (20 percent) in fuel economy at 30 mph.

Figure D-28, which plots engine speed vs time for a 0-60 mph acceleration, shows that with the selected variable ratio transmission, it took 3.3 seconds for the engine to reach 100 percent speed (3675 rpm at transmission input), due to the low initial engine torque characteristic typical of single-shaft gas turbines.

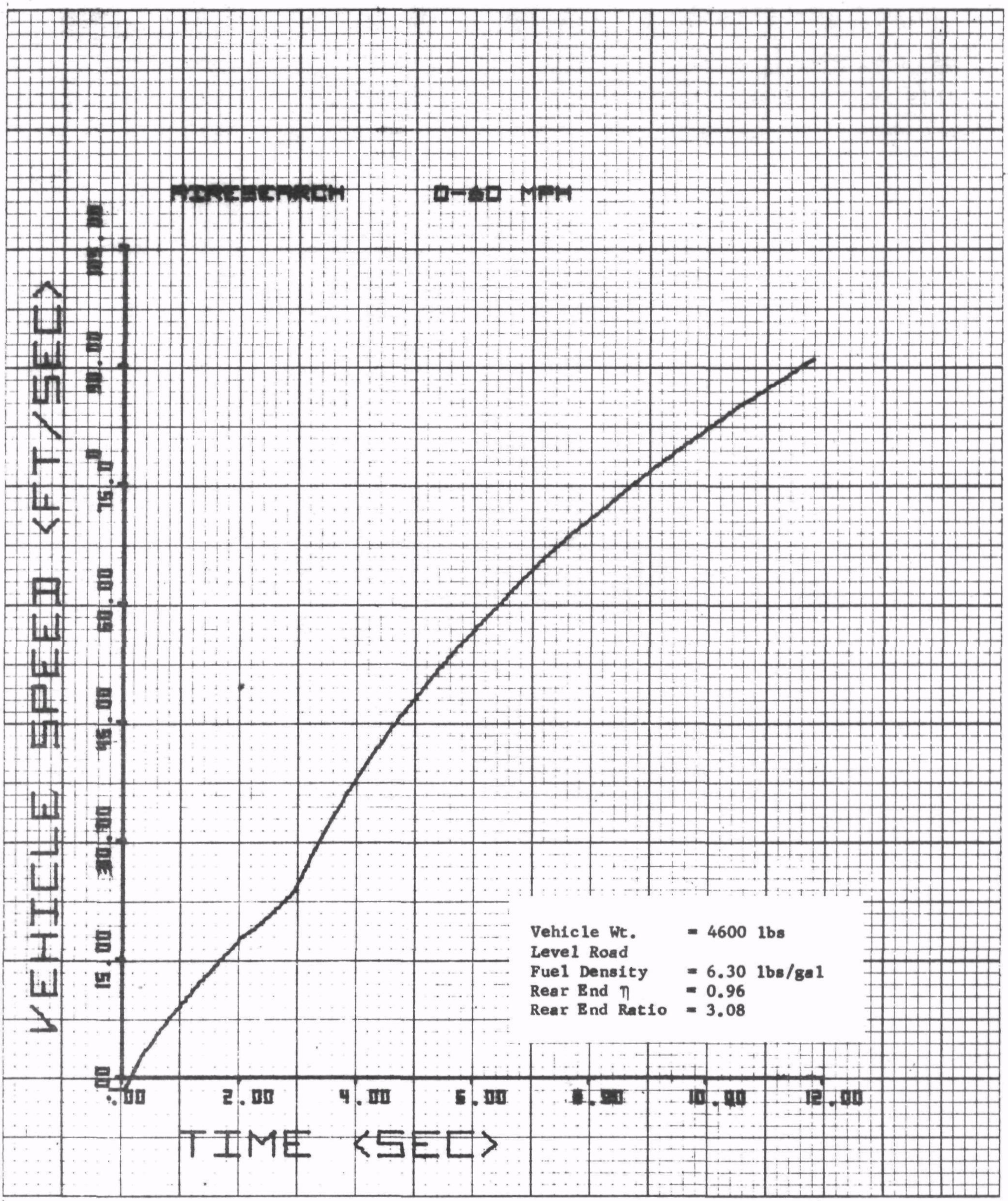

Fig. D-26 Vehicle Speed Under Full-Power Acceleration with Selected Transmission (0-60 MPH) — AiResearch Engine

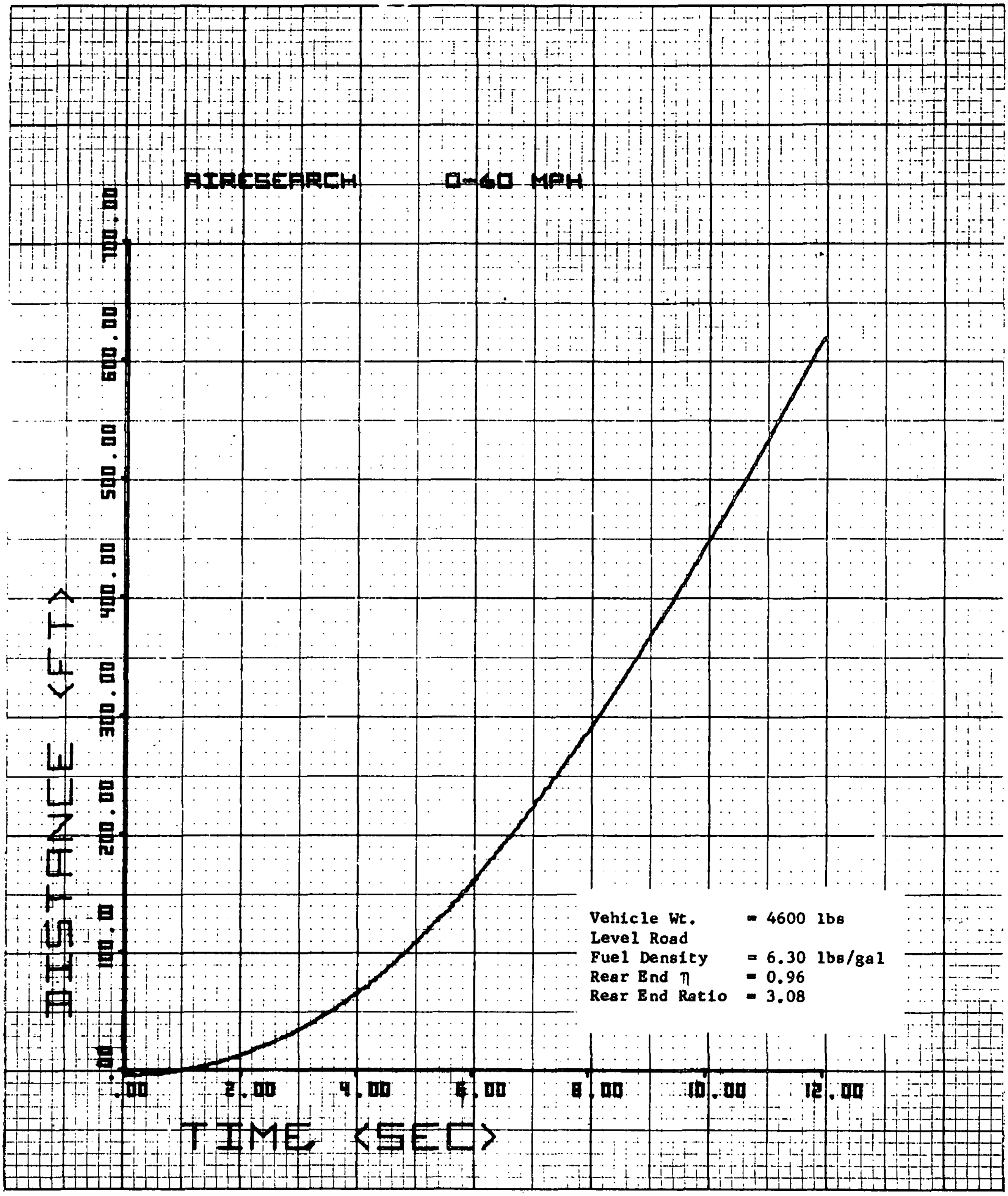

Fig. D-27 Distance Travelled Under Full-Power Acceleration with Selected Transmission (0-50 MPH) — AiResearch Engine

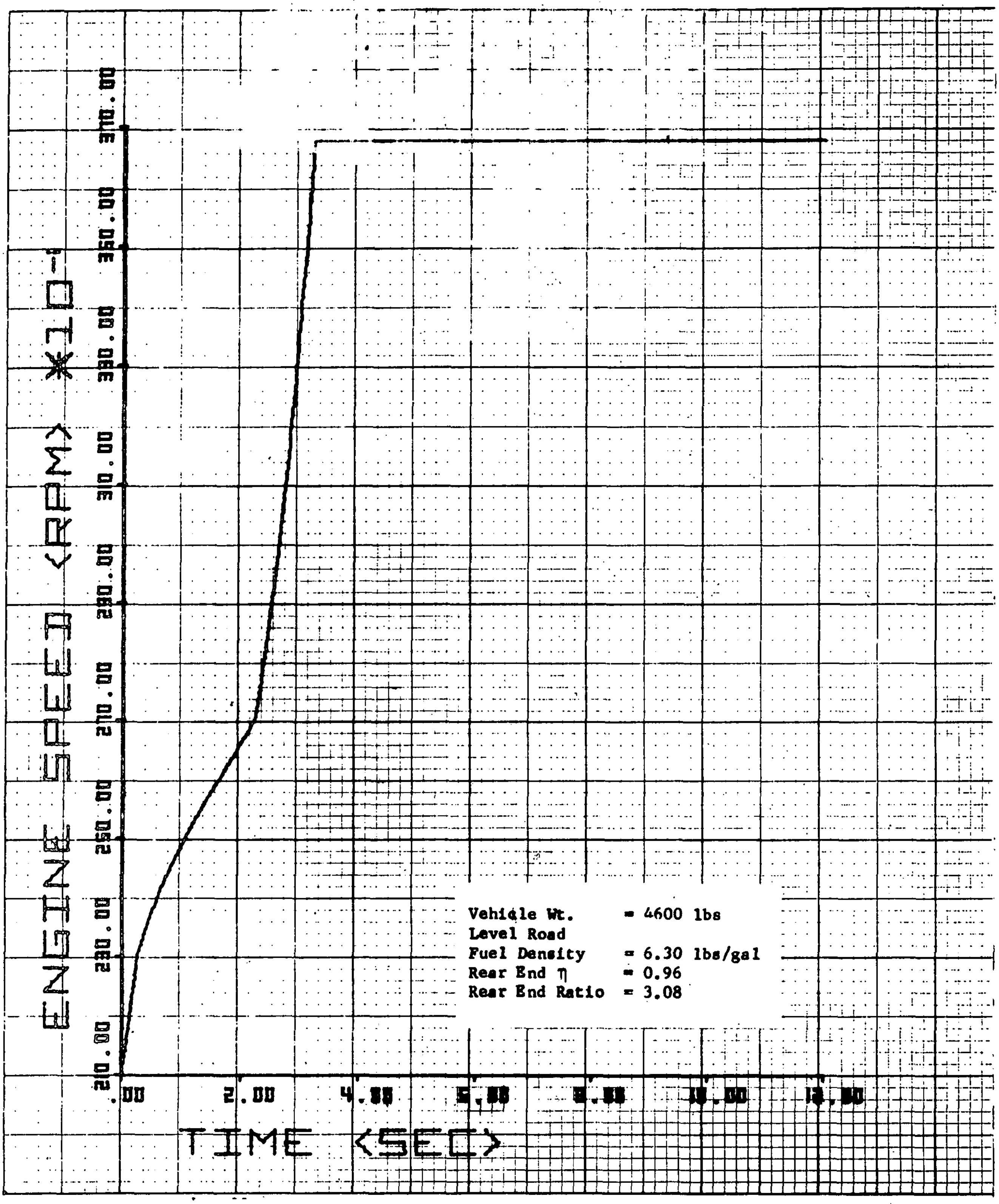

Fig. D-28 Engine Speed Under Full-Power Acceleration with Selected Transmission (0-60 MPH) — AiResearch Engine

E. COST ANALYSIS

Included in this section are a description of the methods employed to determine transmission costs and a discussion of the results. The cost data presented were generated using the experience of automotive cost consultants based upon cost information and practices of the Ford Motor Company. Therefore, this procedure provided a sound approach to comparing the cost of the selected hydromechanical transmission with that of the standard-multispeed torque converter (automatic) currently in mass production.

The objectives of the cost analysis were to determine the original equipment manufacturer cost (OEM) for production quantities of 100,000 and 1,000,000 units per year of the hydromechanical transmission and then compare that cost to similar costs for a multispeed (automatic) transmission.

The procedure used to determine costs was to estimate the detail manufacturing cost of all components of the selected hydromechanical transmission on a variable-cost basis rather than an OEM basis. Then, as subsequently discussed, the total cost of the transmission on a variable-cost basis was converted to an estimated-cost range on an OEM basis.

A variable-cost estimating approach is commonly used in the automotive industry when making decisions on implementation of a new design or replacement system. The major value of this approach is that it eliminated certain transfer costs which may be affected in various ways. The OEM or transfer costs would include cost allocations for fixed burden, scrap, factory cost adjustments, general and administrative costs, profit and capital investment. Capital investment would include costs for facilities, tooling and engineering expense. Since many of these transfer costs would vary with different automotive companies, the OEM data were not as basic as the variable cost data and therefore, were not considered as reliable when comparing information from different sources.

The items which are included in a variable-cost comparison are the purchased cost of the part, direct labor required to get to the desired condition, the indirect labor associated with the manufacturing process, variable overhead items which specifically relate to the manufacturing process and programmed overhead expenses such as specific testing required.

Aside from the need to generate cost data on a basis which was consistent and meaningful in the automotive industry, there was the requirement that proprietary information be protected. For this reason, the cost analysis is presented in the form of ratios using the standard automatic transmission as the datum.

In addition, transfer costs for facilities would not reflect the same in the transmission cost ratios. For instance, the cost of the facilities for the automatic and hydromechanical transmissions could be the same.

As an example, let:

1.0 = Variable costs of automatic transmission
1.44 = Variable costs of power-splitting transmission
0.1 = Facilities costs for either transmission.

Variable cost ratio without facilities include $= \frac{1.44}{1.0} = 1.44$

OEM or cost ratio with facilities included $= \frac{1.54}{1.1} = 1.40$

Therefore, when dealing with ratios the increased cost of the basic transmission is not correctly identified if only the OEM cost ratios are presented. OEM cost ratios are estimated. However, these are given a range to account for possible variances.

1. Typical Details of Costing Procedure

The technique for obtaining the costs for the selected transmission is outlined and a sample sheet is included as Table E-1. The cost of 175 components, some of which were assembles of more than one item (an example would be the park gear lock assembly) was reviewed in order to obtain a valid cost comparison of the transmission. Approximately 25 percent of these were components now used in the standard automatic transmission. The few examples presented on the sample sheet were, in most instances, selected to present the costs of the hydraulic components unique to the power-splitting transmission.

An overall design layout of the selected transmission was made in sufficient detail in order to establish costs. Components, such as the hydrostatic pumps and motors not normally found in an automobile transmission, were detailed with sufficient dimensional and material information for an accurate cost estimate. The approach is similar to that used by high-volume car manufacturers and is discussed in the following paragraphs.

The initial column of Table E-1 describes the part or function to be costed.

Columns 2 and 3 are the part number and number of such parts called out on the transmission parts list as given by the transmission design layouts.

Column 4 presents the method of manufacturing the part consistent with current automotive practices for mass production of transmissions.

In column 5 the material costs were established, and they include all costs to bring the part to the "as-purchased" condition. For example, a die cast component would have rough weight established to develop material cost. The material cost was the actual purchase price in the "as-purchased" condition.

TABLE E-1

COST SAMPLE SHEET

	TOTAL PER ASSEMBLY							REMARKS
COLUMN 1	2	3	4	5	6	7	8	9
DESCRIPTION	ITEM NO	QTY	MAKE BUY	MAT'L COST (DOLLARS)	LABOR MIN.	LABOR COST (VAR) (DOLLARS)	TOTAL COST (DOLLARS)	
Transmission Ass'y.	1A	1	A	—	48.05	8.385	8.385	Incl. Sub. Ass'y. not Specified (Cost Sht 1)
Shaft Engine Input	3	1	PR	.992	8.50	1.483	2.475	Forging - AISI 8620 Steel (Cost Sht 1)
Valve Plate	7	1	PR	.990	7.60	1.326	2.316	Meehanite Casting (Shell Mold) (Cost Sht 1)
Cylinder Block	8	2	PR	2.440	18.00	3.140	5.580	Meehanite Casting (Shell Mold) (Cost Sht 1)
Piston - Motor-Pump	9	18	PS/F	2.592	4.50	.785	3.377	Cold Extrusion (H.T.) (Cost Sht 1)
Trunnion - Swashplate	10	1	PR	1.510	12.00	2.094	3.604	Nodular Iron (H.T.) (Pearl.Mall.) (Cost Sht 1)
Swashplate - Motor-Pump	11	2	PF	4.800	—	—	4.800	Heavy Coined Stpg. for Blank Torr. (Cost Sht 1)
Support - Swashplate	13	1	PR	.600	7.02	1.225	2.125	Cast Iron (Cost Sht 1)
Gear - Motor - Planetary	14	1	PR	1.550	14.20	2.478	4.028	Forging - AISI 8620 Steel (Cost Sht 1)
Carrier - Planetary "B"	21	1	PR	1.540	6.80	1.187	2.727	Malleable Iron (Cost Sht 4)
Planetary Carrier - Sub Ass'y.	20 & 21	1	A	—	4.42	.772	.772	Sub-Assembly (Cost Sht 4)
Ring Gear - Planetary "B"	48	1	PR	1.484	9.40	1.640	3.124	Heat Treat Nodular Iron (Cost Sht 4)
Planet Gear - Planetary "B"	50	3	M	.360	5.30	.925	1.285	Stl. Bar (Cost Sht 4)
Governor - Engage	—	1	PF	1.611	10.25	1.789	3.400	(Cost Sht 2)
Control Body Ass'y - Including Valves, Sleeves, Springs, and Linkage		1	P.F./P.R. M/A	5.620	37.00	6.480	12.100	Typical of Automatic Transmission Control (Cost Sht 2)

NOMENCLATURE

- P.F. Purchased as finished item
- P.R. Purchased in rough condition, such as casting, and forgings
- P.S./F Purchased on a semi-finished item
- M Manufactured in house
- A Assemble

The "in-house" manufacturing costs to finish a specific part are developed on extension of variable-minute costs times labor-minute content. Variable-minute costs include direct labor, indirect labor and non-variable burden.

In Column 6, the actual "in-house" number of labor minutes to complete the manufacturing task were listed and in Column 7 the variable-minute costs were listed.

Column 8 is the total cost in dollars for each item or task labeled in Column 1, and is the sum of Columns 5 and 7.

Tabulation of costing sheets similar to that shown by Table E-1 provided the basis for the variable cost of the selected hydromechanical transmission. It should be noted that all cost-saving design improvements suggested by the automotive cost consultants were factored into the results.

2. Resultant Cost Ratios

As a result of the detailed costing procedure outlined above, cost ratios were established as shown by Table E-2. All ratios presented are the cost of the selected hydromechanical transmission divided by the cost of a conventional multispeed torque converter (automatic transmission) for medium-size family car as currently mass produced. The range in OEM cost ratios is shown in Table E-2 account for the estimated variation associated with these costs.

Detailed cost estimates were made for production levels of 100,000 and 1,000,000 units/year with tooling and facilities appropriate for each of these production rates.

The results given by Table E-2 show that for 1,000,000 units/year the ratio of the variable cost of the selected hydromechanical transmission to that of a typical presently produced, automatic transmission was 1.44 - a 44 percent increase in cost. On an OEM basis the increase in cost ranged between 30 to 40 percent.

TABLE E-2

TRANSMISSION COST ANALYSIS — COST RATIOS

	PRODUCTION LEVEL 1,000,000 UNITS PER YEAR		PRODUCTION LEVEL 100,000 UNITS PER YEAR	
	STANDARD AUTOMATIC TRANSMISSION WITH TORQUE CONVERTER*	POWER SPLITTING HYDROMECHANICAL TRANSMISSION	STANDARD AUTOMATIC TRANSMISSION WITH TORQUE CONVERTER*	POWER SPLITTING HYDROMECHANICAL TRANSMISSION
1. VARIABLE COST RATIO (TOTAL)	1.00*	1.44	1.29	1.86
a. CONTROL VARIABLE COST RATIO	1.00	1.28	1.29	1.65
b. LABOR CONTENT RATIO	1.00	1.34	1.50	2.02
c. MATERIAL CONTENT RATIO	1.00	1.53	1.20	1.84
2. OEM COST RATIO	1.00	1.30-1.40	1.25-1.35	1.70-1.80

*USED AS REFERENCE, PRODUCTION LEVEL OF 1,000,000 UNITS PER YEAR.

At lower production rates, 100,000 units/year, it can be seen from Table E-2 that the variable cost of the automatic transmission would be 1.29 times the cost for 1,000,000 units/year. Thus it follows that the variable cost of the selected hydromechanical in quantities of 100,000 units/year was 1.86 times (1.44 x 1.29 = 1.86) the cost of the automatic transmission produced at the rate of 1,000,000 units/year.

Also shown by Table E-2 is a breakdown in costs attributed to controls, labor, and material. At production levels of 1,000,000 units/year, the control cost increase was 28 percent, additional labor content 34 percent, and material content cost increased 53 percent.

Several features of the hydromechanical transmission can be cited as contributors to the cost increase:

1. The additional governor required to control engine speed.

2. The infinitely variable ratio of the transmission is achieved by using the planetary gearing. Thus, the gears of a power splitting transmission are always loaded, even under 1:1 conditions. A present day automatic locks up in high gear and therefore the gears are unloaded for a good proportion of driving time. This difference in operation requires that the gears of the selected transmission be heavier in construction.

3. The design and manufacturing techniques of a power-splitting hydromechanical transmission have not been developed to the degree of those of the automatic transmission. For example, highly developed, economical, stamping and brazing techniques such as used in current torque converter manufacture could not be considered at this time.

When an article is manufactured in high production, it is common practice to assign a cost to the item as so many dollars per pound. The power-splitting transmission weights 146 pounds versus 140 pounds for the standard automatic transmission. Therefore, it is reasonable to believe that when the design and manufacturing skills developed by the automobile industry, over a period of several years of manufacturing the existing transmission are applied to the selected hydromechanical transmissions, the cost will decrease. Assuming an equivalent cost per pound for both transmissions, the variable cost percent increase would be 4 percent.

In summary, it was concluded that the selected hydromechanical transmission would initially cost 44 percent more than the standard automatic transmission and finally, after several years of production and refinement, would approach 4 percent more.

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APPENDIX A

PERFORMANCE OF THE AIRESEARCH SINGLE-SHAFT, GAS-TURBINE ENGINE

Table A-1 presents the performance data for the single-shaft, gas-turbine engine as received from the engine manufacturer. This data accounts for losses in the engine speed reducer and for a 4 hp constant accessory load. All torque and speed data refer to the engine shaft.

TABLE A-1

PERFORMANCE DATA FOR THE SINGLE-SHAFT GAS-TURBINE ENGINE

Engine Scaled for 4600 Lb Vehicle 105 F Day, Sea Level

IGV Setting	Torque (Ft-Lb)	Fuel Flow (Lb/Hr)	Engine RPM	Turbine Inlet Temp (R)
Water Injection				
1.02500	8.476	65.17	83064.70000	2359.70000
1.02500	8.236	57.03	79109.20000	2359.70000
1.02500	7.605	48.53	75153.70000	2359.70000
1.02500	6.869	40.83	71198.20000	2359.70000
1.02500	5.89	33.47	67242.70000	2359.70000
1.02500	5.001	27.21	63287.20000	2359.70000
1.02500	4.103	21.50	59331.70000	2359.70000
1.02500	3.258	16.14	55376.20000	2359.70000
1.02500	2.676	13.38	51420.70000	2359.70000
1.02500	2.081	10.40	47465.20000	2359.70000
1.02500	1.587	7.96	43509.70000	2359.70000
Water Injection				
1.02500	7.322	61.6	83064.70000	2259.70000
1.02500	7.328	54.47	79109.20000	2259.70000
1.02500	6.844	46.59	75153.70000	2259.70000
1.02500	6.248	39.28	71198.20000	2259.70000
1.02500	5.395	32.30	67242.70000	2259.70000
1.02500	4.568	26.19	63287.20000	2259.70000
1.02500	3.763	20.77	59331.70000	2259.70000
1.02500	2.969	16.17	55376.20000	2259.70000
1.02500	2.428	12.92	51420.70000	2259.70000
1.02500	1.864	10.03	47465.20000	2259.70000
1.02500	1.411	7.68	43509.70000	2259.70000
1.02500	6.52048	50.13635	83064.70000	2359.70000
1.02500	6.33683	43.87011	79109.20000	2359.70000
1.02500	5.85136	37.33375	75153.70000	2359.70000
1.02500	5.28404	31.40953	71198.20000	2359.70000
1.02500	4.53156	25.75446	67242.70000	2359.70000
1.02500	3.84759	20.92541	63287.20000	2359.70000
1.02500	3.15644	16.54436	59331.70000	2359.70000
1.02500	2.50573	12.87627	55376.20000	2359.70000
1.02500	2.05890	10.29384	51420.70000	2359.70000
1.02500	1.60082	8.00597	47465.20000	2359.70000
1.02500	1.22117	6.12664	43509.70000	2359.70000

(TABLE A-1 CONT'D)

N III 2V ENGINE SCALED FOR 4600 LB VEHICLE 105 F DAY, SEA LEVEL

IGV Setting	Torque (Ft-Lb)	Fuel Flow (Lb/Hr)	Engine RPM	Turbine Inlet Temp (R)
1.02500	5.63231	47.39872	83064.70000	2259.70000
1.02500	5.63714	41.90301	79109.20000	2259.70000
1.02500	5.26518	35.83623	75153.70000	2259.70000
1.02500	4.80603	30.22312	71198.20000	2259.70000
1.02500	4.15052	24.85373	67242.70000	2259.70000
1.02500	3.51416	20.15443	63287.20000	2259.70000
1.02500	2.89535	15.97729	59331.70000	2259.70000
1.02500	2.28427	12.43825	55376.20000	2259.70000
1.02500	1.86830	9.93734	51420.70000	2259.70000
1.02500	1.43460	7.71882	47465.20000	2259.70000
1.25000	1.08504	5.90734	43509.70000	2259.70000
1.00000	4.50329	43.45813	83064.70000	2159.70000
1.00000	4.74686	39.16062	79109.20000	2159.70000
1.00000	4.54973	33.55177	75153.70000	2159.70000
1.00000	4.21023	28.38931	71198.20000	2159.70000
1.00000	3.66930	23.42959	67242.70000	2159.70000
1.00000	3.10374	19.01482	63287.20000	2159.70000
1.00000	2.58053	15.15423	59331.70000	2159.70000
1.00000	2.00324	11.78439	55376.20000	2159.70000
1.00000	1.62480	9.37452	51420.70000	2159.70000
1.00000	1.23159	7.27073	47465.20000	2159.70000
1.00000	.88790	5.53629	43509.70000	2159.70000
.95000	4.15621	40.04014	83064.70000	2159.70000
.95000	4.36602	36.05083	79109.20000	2159.70000
.95000	4.20550	31.04099	75153.70000	2159.70000
.95000	3.84409	26.13879	71198.20000	2159.70000
.95000	3.33133	21.47865	67242.70000	2159.70000
.95000	2.80127	17.35890	63287.20000	2159.70000
.95000	2.30842	13.86534	59331.70000	2159.70000
.95000	1.81989	10.82391	55376.20000	2159.70000
.95000	1.40877	8.51587	51420.70000	2159.70000
.95000	1.06671	6.62678	47465.20000	2159.70000
.95000	.74329	4.99698	43509.70000	2159.70000
.90000	3.69192	36.97856	83064.70000	2159.70000
.90000	3.93939	33.37912	79109.20000	2159.70000
.90000	3.84652	28.79323	75153.70000	2159.70000
.90000	3.46176	24.14606	71198.20000	2159.70000
.90000	3.00534	19.88584	67242.70000	2159.70000
.90000	2.53520	16.06917	63287.20000	2159.70000
.90000	2.05547	12.76527	59331.70000	2159.70000
.90000	1.51023	10.03611	55376.20000	2159.70000
.90000	1.20657	7.76594	51420.70000	2159.70000
.90000	.89830	6.05864	47465.20000	2159.70000
.90000	.59617	4.57479	43509.70000	2159.70000

(TABLE A-1 CONT'D)

IGV Setting	Torque (Ft-Lb)	Fuel Flow (lb/Hr)	Engine RPM	Turbine Inlet Temp (R)
.85000	3.21515	33.88733	83064.70000	2159.70000
.85000	3.54201	30.83968	79109.20000	2159.70000
.85000	3.44061	26.47738	75153.70000	2159.70000
.85000	3.04786	22.13567	71198.20000	2159.70000
.85000	2.63594	18.22650	67242.70000	2159.70000
.85000	2.23652	14.79780	63287.20000	2159.70000
.85000	1.77420	11.68450	59331.70000	2159.70000
.85000	1.37177	9.08968	55376.20000	2159.70000
.85000	1.00330	7.08779	51420.70000	2159.70000
.85000	.72302	5.49914	47465.20000	2159.70000
.85000	.45745	4.16674	43509.70000	2159.70000
.80000	2.70715	30.80587	83064.70000	2159.70000
.80000	3.09817	28.22065	79109.20000	2159.70000
.80000	2.95335	24.09902	75153.70000	2159.70000
.80000	2.60773	20.20578	71198.20000	2159.70000
.80000	2.24579	16.56778	67242.70000	2159.70000
.80000	1.90405	13.42425	63287.20000	2159.70000
.80000	1.52317	10.69869	59331.70000	2159.70000
.80000	1.09418	8.21204	55376.20000	2159.70000
.80000	.78042	6.39494	51420.70000	2159.70000
.80000	.53901	4.91990	47465.20000	2159.70000
.80000	.31242	3.73569	43509.70000	2159.70000
.75000	2.27605	27.88643	83064.70000	2159.70000
.75000	2.56293	25.47337	79109.20000	2159.70000
.75000	2.42355	21.70740	75153.70000	2159.70000
.75000	2.15166	18.16663	71198.20000	2159.70000
.75000	1.88119	14.99190	67242.70000	2159.70000
.75000	1.54983	12.05540	63287.20000	2159.70000
.75000	1.22536	9.61813	59331.70000	2159.70000
.75000	.82010	7.29660	55376.20000	2159.70000
.75000	.55712	5.67738	51420.70000	2159.70000
.75000	.34899	4.36702	47465.20000	2159.70000
.75000	.14381	3.33508	43509.70000	2159.70000
.70000	1.70357	24.80789	83064.70000	2159.70000
.70000	1.99172	22.63508	79109.20000	2159.70000
.70000	1.85513	19.30451	75153.70000	2159.70000
.70000	1.65484	16.11233	71198.20000	2159.70000
.70000	1.46790	13.31192	67242.70000	2159.70000
.70000	1.17852	10.78126	63287.20000	2159.70000
.70000	.90137	8.46226	59331.70000	2159.70000
.70000	.51862	6.36979	55376.20000	2159.70000
.70000	.31933	4.96326	51420.70000	2159.70000
.70000	.13693	3.82038	47465.20000	2159.70000
.70000	- .01083	2.90044	43509.70000	2159.70000

(TABLE A-1 CONT'D)

IGV Setting	Torque (F1-1b)	Fuel Flow (1b/hr)	Engine RPM	Turbine Inlet Temp (R)
.65000	1.12997	21.82657	83064.70000	2159.70000
.65000	1,32501	19.79218	79109.20000	2159.70000
.65000	1.28382	16.89241	75153.70000	2159.70000
.65000	1.21992	14.25839	71198.20000	2159.70000
.65000	1.02301	11.66346	67242.70000	2159.70000
.65000	.80923	9.43478	63287.20000	2159.70000
.65000	.55255	7.31575	59331.70000	2159.70000
.65000	.23529	5.48314	55376.20000	2159.70000
.65000	.08813	4.26976	51420.70000	2159.70000
.65000	-.06539	3.29390	47465.20000	2159.70000
.65000	-.18341	2.49250	43509.70000	2159.70000
.60000	.33612	18.45764	83064.70000	2159.70000
.60000	.70237	17.08063	79109.20000	2159.70000
.60000	.71189	14.61040	75153.70000	2159.70000
.60000	.67152	12.22382	71198.20000	2159.70000
.60000	.51866	10.02985	67242.70000	2159.70000
.60000	.39334	8.05270	63287.20000	2159.70000
.60000	.16514	6.15717	59331.70000	2159.70000
.60000	-.05371	4.60849	55376.20000	2159.70000
.60000	-.16955	3.60177	51420.70000	2159.70000
.60000	-.26051	2.78586	47465.20000	2159.70000
.60000	-.33485	2.12841	43509.70000	2159.70000

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15. SUPPLEMENTARY NOTES

16. ABSTRACT

The purpose of this study was to assess, on both a performance and cost basis, the transmission most suitable for use with two types of advanced automotive engines designed to power a medium sized family car. As a result of considering eight different types of transmissions, a continuously variable ratio, power-splitting, hydromechanical transmission was selected as the best near-term transmission for application with the single-shaft gas-turbine and the turbo-Rankine engines. The best long-term candidate was determined to be the Traction-type transmission. This transmission holds great promise for the future, but requires substantial development effort. The selected hydromechanical transmission was designed in sufficient detail to ascertain performance, cost and physical characteristics. Production cost is a key factor in considering a transmission for automotive applications. With development of production techniques, future production costs for such a transmission could approach that of a present day automatic transmission. The most significant conclusion of this study was that further development of the selected variable ratio hydromechanical transmission is recommended for application to the single-shaft gas turbine and not for the turbo-Rankine engine, unless future design developments for the latter engine result in showing a significant change in Specific Fuel Consumption with engine speed.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
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