U.S. DEPARTMENT OF COMMERCE
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PB-296 600

Study and Test to Confirm Automobile Drivetrain Components to Improve Fuel Economy. Volume II. The Drivetrain Design Process with an Automatic Transmission

Arthur D. Little, Inc., Cambridge, MA

Prepared for

Transportation Systems Center, Cambridge, MA

May 79

REPORT NO. DOT-TSC-NHTSA-79-11.II

PB 296600

DOT HS-803 856

STUDY AND TEST TO CONFIRM AUTOMOBILE DRIVETRAIN COMPONENTS TO IMPROVE FUEL ECONOMY
Volume II
The Drivetrain Design Process With An Automatic Transmission

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MAY 1979

INTERIM REPORT

Prepared for

U.S. DEPARTMENT OF TRANSPORTATION
NATIONAL HIGHWAY TRAFFIC SAFETY ADMINISTRATION
Office of Research and Development
Washington DC 20590

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Technical Report Documentation Page

1. Report No. DOT-HS-803 856	2. Government Accession No.
3. Recipient's Catalog No. PB296600	5. Report Date May 1979
4. Title and Subtitle STUDY AND TEST TO CONFIRM AUTOMOBILE DRIVETRAIN COMPONENTS TO IMPROVE FUEL ECONOMY VOLUME II, THE DRIVETRAIN DESIGN PROCESS WITH AN AUTOMATIC TRANSMISSION	6. Performing Organization Code
7. Author's) D. A. Hurter, P. G. Gott	8. Performing Organization Report No. DOT-TSC-NHTSA-79-11.II
9. Performing Organization Name and Address Arthur D. Little, Inc. Acorn Park Cambridge, MA 02140*	10. Work Unit No. (TRAIS) HS927/R9404
	11. Contract or Grant No. DOT/TSC 1046
12. Sponsoring Agency Name and Address U.S. Department of Transportation National Highway Traffic Safety Administration Office of Research and Development Washington DC 20590	13. Type of Report and Period Covered Interim June 1975 - December 1976
	14. Sponsoring Agency Code

15. Supplementary Notes *Under Contract to: U.S. Department of Transportation
Research and Special Programs Administration
Transportation Systems Center
Kendall Square, Cambridge MA 02140

16. Abstract

This report presents the key thought processes that are used by the drivetrain designer in matching an engine to a given vehicle with specific performance goals in mind. This report provides those uninitiated in this aspect of automotive design valuable insight into the compromises that must be made between acceleration, gradeability and top speed.

This report also presents a method by which the performance of any modern torque converter may be approximated with accuracy once only the stall torque ratio and diameter are known. These two parameters can be used to generate torque converter maps that almost duplicate actual maps for torque converters of any modern manufacturer, and greatly facilitate computer simulation.

17. Key Words	18. Distribution Statement
Gear Ratio, Drivetrain Design, Shift Point, Torque Converter, Simpson Gear Set, Performance, Automatic Transmission	DOCUMENT IS AVAILABLE TO THE U.S. PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 2216,

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	82	A05-A01

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

PREFACE

This interim report, prepared by Arthur D. Little, Inc., for the U. S. Department of Transportation, presents a study of U. S. Automatic Transmissions - their history and design philosophy. This report consists of two Volumes.

Volume I presents a history of the automobile transmission in the United States, with particular emphasis on the family tree of the U.S. automatic transmissions. Volume I also contains, in tabular form, a description of the 1970-1975 engine/transmission/rear axle ratio/vehicle size and weight combinations available in all U.S. cars of this time period.

Volume II is a handbook-like narrative on the selection of transmissions and rear axle gear ratios, shift points, shift quality, and other pertinent drivetrain design parameters.

Arthur D. Little, Inc., wishes to acknowledge and thank Mr. H. Gould and Mr. R. Colello of the Department of Transportation, Transportation Systems Center for their guidance and assistance in the preparation of these volumes. ADL also wishes to gratefully acknolwedge Chilton Company for their permission to use copyrighted material from the Automotive Industries from which much of Volume I, has been drawn.

The work on this project was completed under the sponsorship of the U.S. Department of Transportation, National Highway Traffic Safety Administration's Automotive Fuel Economy Regulatory Program at the Transportation Systems Center.

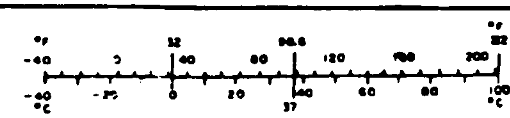
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
	MASS (weight)			
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
	VOLUME			
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
	TEMPERATURE (exact)			
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
	MASS (weight)			
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
	VOLUME			
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. INTRODUCTION

1.1 PURPOSE

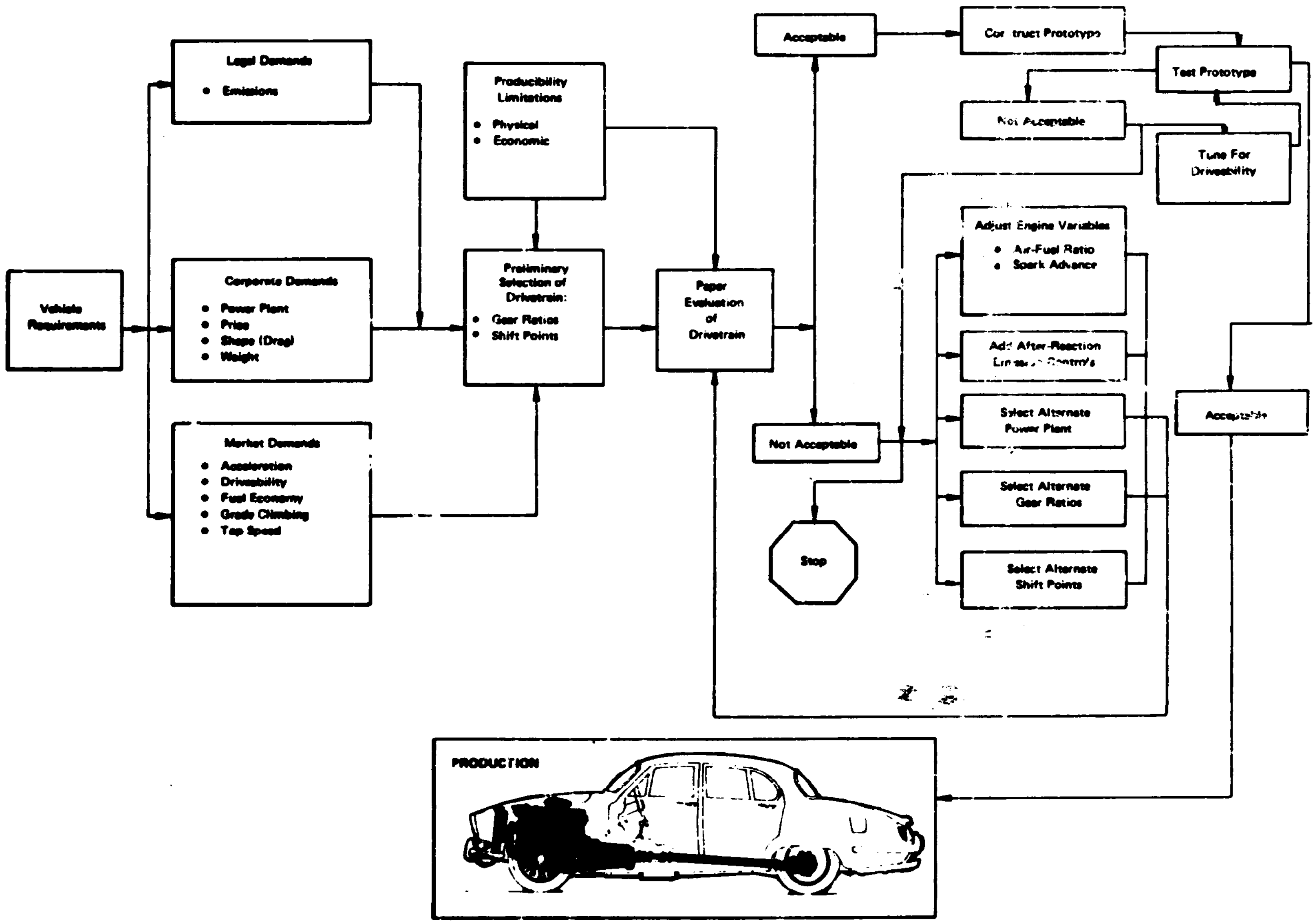
The drivetrain designer's basic task is to blend vehicular power requirements, engine characteristics, and "front office" mandates with each other to come up with a finished product which will allow the driver to convert rotational force into smooth linear motion with a slight movement of his right foot. Given the basic vehicle requirements such as acceleration, top speed, and grade climbing ability, the designer must also consider fuel consumption, emissions, driveability, vibration, noise, and price.

The purpose of this report is to present the design decision process. It will show how the transmission designer endeavors to match vehicle requirements with engine characteristics to arrive at the optimum compromise among the above parameters. As Figure 1.1, Decision Tree for Drivetrain Design, illustrates, the drivetrain design process consists of optimizing and/or compromising various desired vehicle characteristics in a step-by step process beginning with 'paper' drivetrains. The preliminary paper design is built and tested in prototype form, modified, and retested until the vehicle's drivetrain is optimized. Each step is represented by a box in the Decision Tree. This report will explain contents of each of these boxes. Appendix A will take the reader step-by-step through the preliminary drivetrain design process.

1.2 WHAT IS AN AUTOMATIC TRANSMISSION?

Before one can detail the design process, it is necessary to define an automatic transmission. More to the point, one must define exactly what an automatic transmission must do.

An automatic transmission must match the output of an engine having certain torque, power, and speed characteristics with widely different vehicular torque, power, and speed requirements. This is best illustrated by Figure 1.2. Available engine power may be modified by throttle position, and vehicle demands may be altered by the whims of the driver. The transmission must automatically mate the engine output with the



Source: Arthur D. Little, Inc.

FIGURE 1.1 DECISION TREE FOR DRIVETRAIN DESIGN vehicle demands. The manner in which this is done determines the performance and economy of the vehicle.

The transmission is the major link between the engine, the driver, and the vehicle, as shown in Figure 1.3. The driver senses the motion of the vehicle and, through the accelerator and/or gearshift, makes a demand for power and/or acceleration. Through the automatic transmission control circuit, the engine and transmission interact to move the vehicle in response to driver demands.

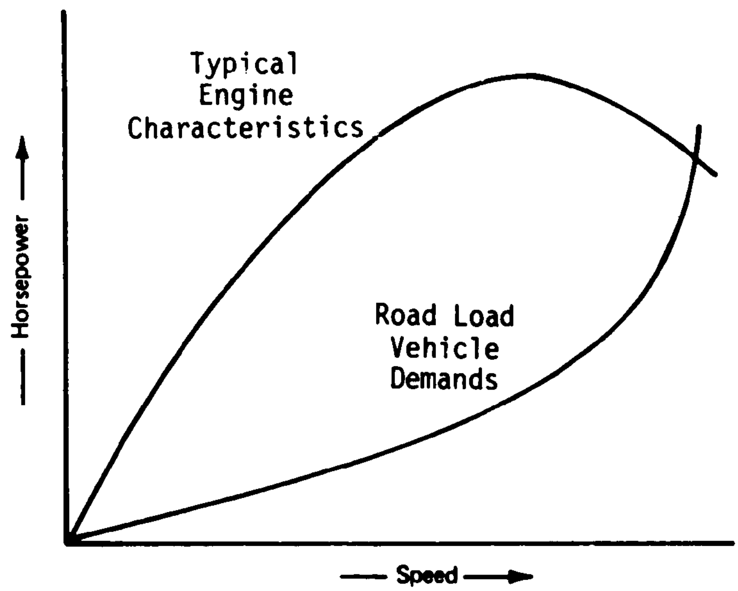


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FIGURE 1.2 ILLUSTRATION OF THE DIFFERENCE BETWEEN AVAILABLE ENGINE POWER AND VEHICLE NEEDS

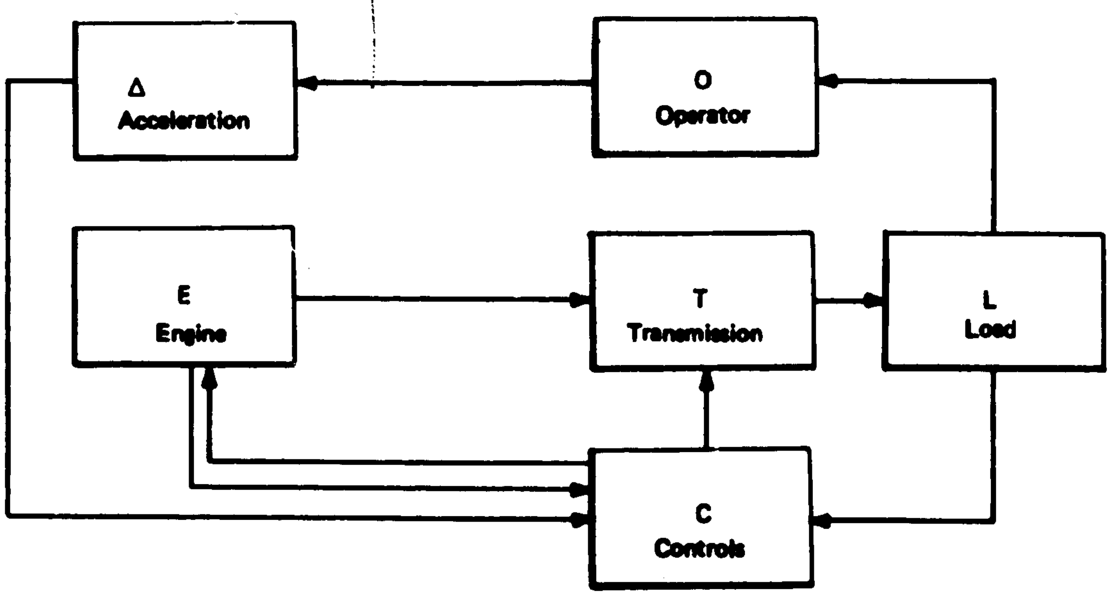

FIGURE 1.3 RELATIONSHIP BETWEEN ENGINE, TRANSMISSION, LOAD (VEHICLE), AND DRIVER

2. VEHICLE REQUIREMENTS

Vehicle requirements include both the physical characteristics of the particular vehicle and the desired performance of that vehicle (driver demands). Physical characteristics of the vehicle include weight, dynamics (over-the-road power requirements), and even a tentative engine with given torque, power, and speed relationships. Desired performance includes emissions and such saleable driver demands as acceleration, top speed, grade climbing ability, and economy. These, then, provide the starting points for a transmission design. Taken in a probable order of consideration in the design process, transmission design parameters are:

- Grade Climbing Ability - Vehicle must be able to start on a particular grade and traverse it at a minimum speed (i.e., 30% at 5 mph);
- Acceleration - Vehicle must get from a standing start to a given speed in a minimum time (i.e., 0-60 in 12 seconds);
- Economy - Does the vehicle utilize fuel most efficiently (i.e., is the engine running at minimum brake specific fuel consumption at design speeds?);
- Top Speed - (i.e., can vehicle reach or exceed 55 mph on a given grade?)

3. TRANSMISSION DESIGN - TORQUE AND GEAR RATIOS

3.1 AUTOMATIC TRANSMISSION CHARACTERISTICS

There are two basic power transmitting components of an automatic transmission: the gearset and the torque converter. The gear set simply multiplies the speed and torque of the transmission input by a numerical ratio dependent upon the number of teeth and diameters of the various gears, as in Figure 3.1.

The torque converter is a hydrodynamic device which multiplies engine torque when there is a large speed differential between the input and output shafts. The torque multiplication factor is greatest when the torque converter is "stalled"; that is, when the output shaft is not turning. Without going into the mechanics of torque converters, it is sufficient for this discussion to illustrate their characteristics in Figure 3.2. The point at which they cease to multiply torque and become a nonrigid connector between the input and output shafts is the "coupling" point. This is when they become a fluid coupling and not a torque multiplier. Appendix B illustrates the relationship between stall torque ratio and coupling point.

3.2 THE STARTING POINT

The first consideration in torque ratio selection is usually grade climbing ability. The steepest grade normally encountered is found on service station grease racks, and is approximately 30%. Hence, this determines the minimum vehicle gradeability.

With nominal vehicle weight and engine characteristics already given, the transmission designer must choose an overall first gear ratio, including torque converter and rear axle, which will allow the vehicle to start and move on a 30% grade. This establishes the maximum tractive effort available from the vehicle's wheels; that is, the maximum force exerted on the ground by the wheels.

This overall first gear ratio will include a torque converter operating with a stall torque ratio of approximately 2:1. On a 30% grade, the engine will operate well up on its torque vs. speed curve, but not necessarily at the peak. As the vehicle begins to move, the torque converter torque ratio may drop and the engine may speed up and climb up to the peak of its torque curve.

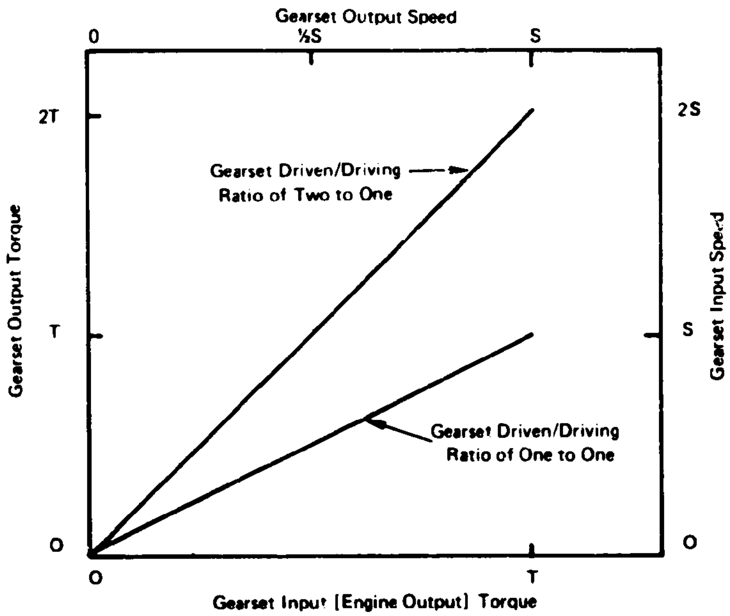

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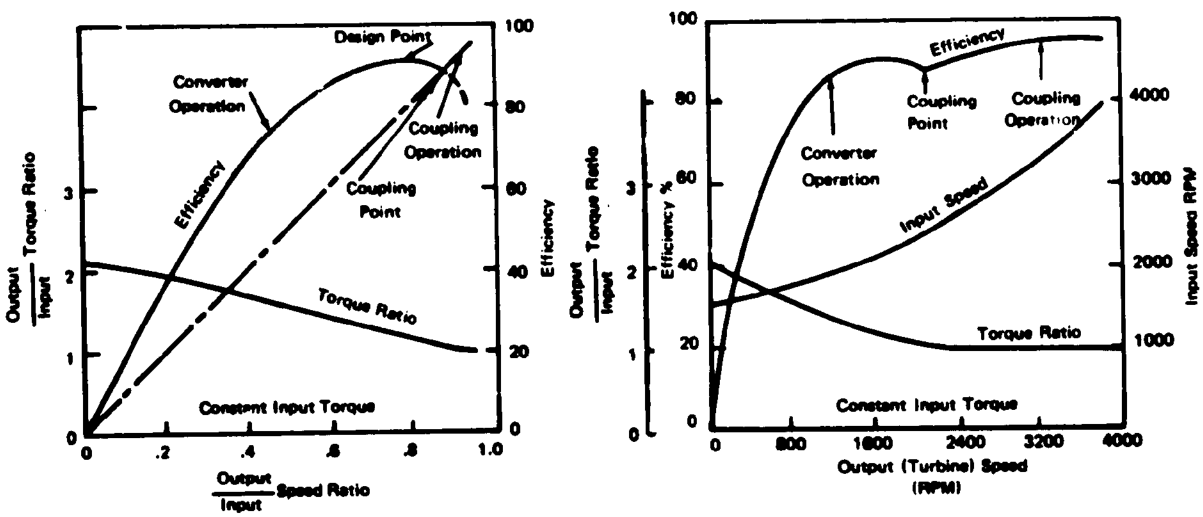
FIGURE 3.1 GEARSET CHARACTERISTICS



Source: Giles[13]

FIGURE 3.2 TORQUE-CONVERTER CHARACTERISTICS

The ratio of the lowest transmission gear, in fact, of all the transmission gears, is normally fixed. Usually the automaker manufactures only one set of gear ratios and the drivetrain designer is locked into using it. Normally the production costs of producing a second, alternate gearset are not warranted, as the drivetrain can usually be designed satisfactorily by juggling both torque converter and rear axle ratios.

The rear axle ratio is determined next by gradeability and the top speed, top gear performance requirements. In the past, top gear performance has been considered optimized when the maximum vehicle speed occurs just after the point of peak horsepower[2], providing adequate acceleration capability throughout the vehicle's speed range. However, the trend today is to design for fuel economy, so the designer may select a rear axle ratio which will operate the engine at or close to the lowest brake specific fuel consumption point for normal "cruise" speeds which will provide the necessary power in top gear. The power required in top gear will be based upon designed top gear grade climbing ability at speed and parasitic accessory (generator, air conditioning, water pump, etc.) loads, as well as vehicle dynamic loads such as wind drag and rolling resistance.

With the rear axle ratio now established and the first (low) gear ratio known, the designer can determine precisely the required torque converter output torque from the 30% grade climbing requirement. The torque converter stall torque ratio and stall speed are then determined as those values which will allow the engine to run at a speed at which it will generate sufficient torque which, when multiplied by the stall torque ratio, will provide the necessary torque converter output torque and allow the car to move on a 30% grade.

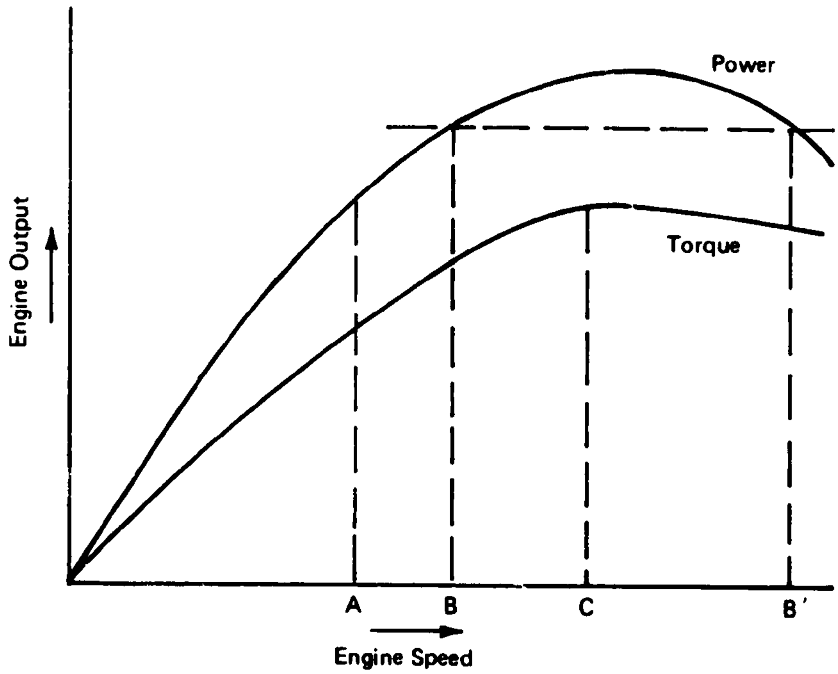
If the drivetrain designer is not "locked" into a specific set of transmission gear ratios, the intermediate gear ratios must now be determined. The usual goal for the selection of these ratios in most passenger vehicles is to get from low gear to top gear as smoothly as possible. At the same time, however, the public expects a car to accelerate from 0 to 60 mph in 12 to 18 seconds. The choice of intermediate gears may become a compromise between smoothness and acceleration.

3.3 SMOOTHNESS - A FUNCTION OF INTERMEDIATE GEAR RATIOS AND SHIFT LOGIC

A smooth gear change is one in which there is no noticeable change in the torque transmitted through the drivetrain; that is, the torque (and therefore the power) available from the drivetrain in the lower gear just before the upshift must match that available in the upper gear just after the gear change. Figures 3.3 and 3.4 illustrate this. Assume that the car is at a standstill and the driver opens the throttle to accelerate. The engine immediately speeds up to point A (Figure 3.3 & 3.4), the converter stall speed, and the vehicle begins to move. The engine then accelerates through point B (Figure 3.3) to peak torque (C), and then down the backside of the curve to lower torque speeds. If the engine speed is allowed to vary much beyond that point where peak torque is developed, the rate of vehicular acceleration will fall off. Therefore, the transmission designer causes the gear box to upshift when the available torque has dropped to a level some 3 to 5% below peak torque. Upshift at this point minimizes the change in rate of vehicle acceleration due to the reduction in engine torque with increasing engine speed.

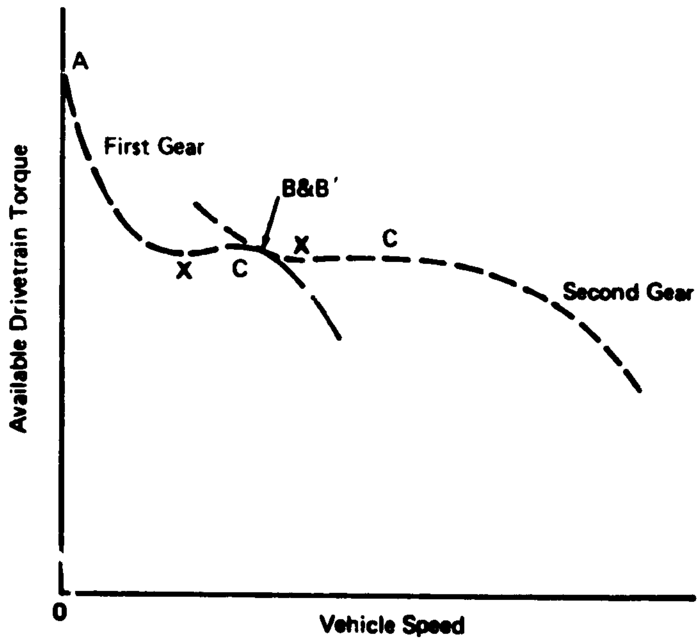
In order to effect a smooth shift, there should be no sudden change in drivetrain output torque or power. The designer chooses his intermediate gear ratio such that the torque available from the drivetrain at point B' before the shift is equal to the available drivetrain torque after the shift. Point B, Figure 3.4, a drivetrain map, illustrates this for an upshift from first to second gear. (The labeled points on Figure 3.4 correspond to the same engine operating points as on Figure 3.3 except that Figure 3.4 includes torque multiplication through the converter and gears whereas Figure 3.3 does not. Point X is the torque converter coupling point.) The shift point (B & B') is where the first and second gear curves are tangent to each other and where engine power output is constant.

To prevent excessive transmission "hunting" between gears if a car is driven along at exactly shift speed with a constant throttle opening, the transmission is normally scheduled to downshift at speeds of 3 to 5 mph below the upshift speeds.



Source: Arthur D. Little, Inc.

FIGURE 3.3 ENGINE CHARACTERISTICS FOR A GIVEN THROTTLE POSITION



Source: Arthur D. Little, Inc.

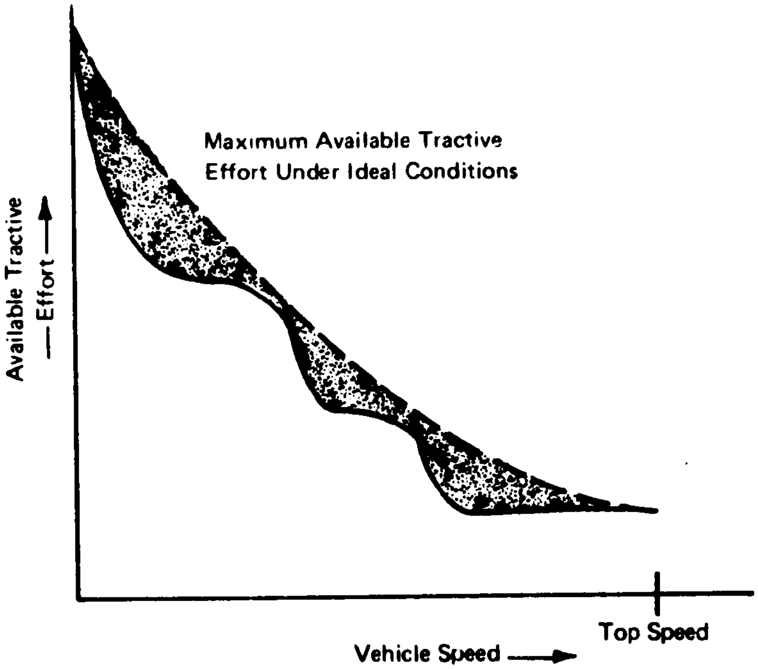
FIGURE 3.4 AVAILABLE DRIVETRAIN TORQUE IN FIRST AND SECOND GEARS VS. VEHICLE ROAD SPEED

3.4 PERFORMANCE

The drivetrain designer has now selected a smooth shifting torque converter/gear combination. Does it meet the requirement for adequate acceleration?

The rolling radius of the driving wheels divided by the available axle torque yields the tractive effort or force available to propel the car. This available force, or the friction force afforded by the tires on the pavement, whichever is less, divided by the mass of the car yields the acceleration. If this is not adequate, the drivetrain designer may choose to increase the available force by altering his choice of gears, shift points and/or engine.

The dashed line in Figure 3.5 depicts the maximum available tractive effort at the driving wheels. This would be generated if a transmission capable of maintaining the engine at its peak torque throughout the entire speed range were available. (Such would be the case with a properly designed continuously variable transmission.) Figure 3.5 also shows the available tractive effort resulting from the gear ratios selected. The shaded area between the maximum possible tractive effort and that available from the gear box is that power not available. From studying this illustration, it should be clear that the real curve may be brought closer to the ideal by upshifting at slightly lower vehicle speeds, by adding another gear between first and second, or by making the second and third gear ratios more equal. Of course, if the real curve already exceeds the friction force between the tires and road, the car will spin its wheels. In this case, the vehicle may be overpowered, and a less powerful engine may be desired.



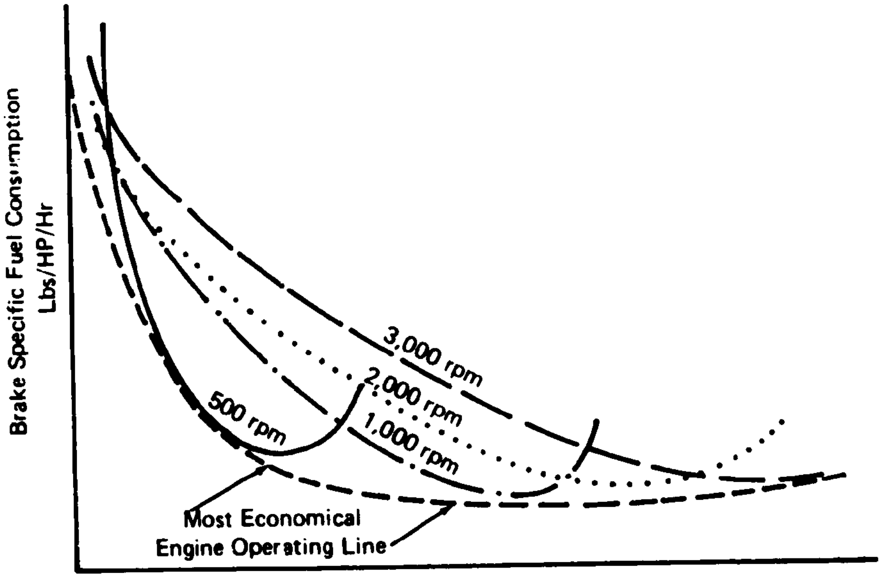
Source Arthur D. Little, Inc.

FIGURE 3.5 MAXIMUM AVAILABLE TRACTIVE EFFORT (WOT) VS. VEHICLE ROAD SPEED

3.5 FUEL ECONOMY

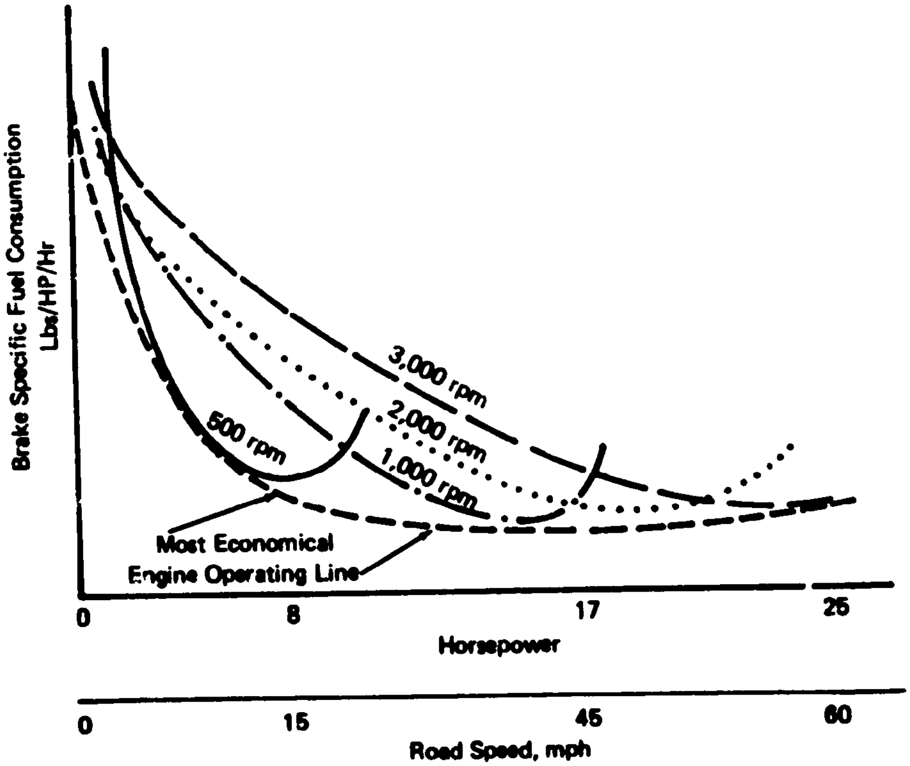
Another criteria for selecting a less powerful engine may be fuel economy. As a general rule, the harder an engine works, (the more open the throttle at a given load) the more efficiently it works. This may be visualized by looking at Figure 3.6. This hypothetical figure is typical of an automotive engine. The gear train in the car would operate the engine at the most economical point by following the indicated line. Road load would be matched with that engine speed producing the required power with the least quantity of fuel. In Figure 3.7, a scale of road load power required has been added to Figure 3.6 to illustrate this point. This figure shows that in this hypothetical case, the transmission should be designed to run the engine at 500 RPM at 12 mph, 1000 RPM at 40 mph, and 3000 RPM at 60 mph in order to obtain optimum economy.

Of course, no over-the-road vehicle operates at steady state conditions. The roadways are never level, and the driver is continually making speed adjustments, forcing the vehicle to periodically accelerate



Source: Arthur D. Little, Inc.

FIGURE 3.6 TYPICAL BRAKE SPECIFIC FUEL CONSUMPTION VS. POWER OUTPUT



Source: Arthur D. Little, Inc.

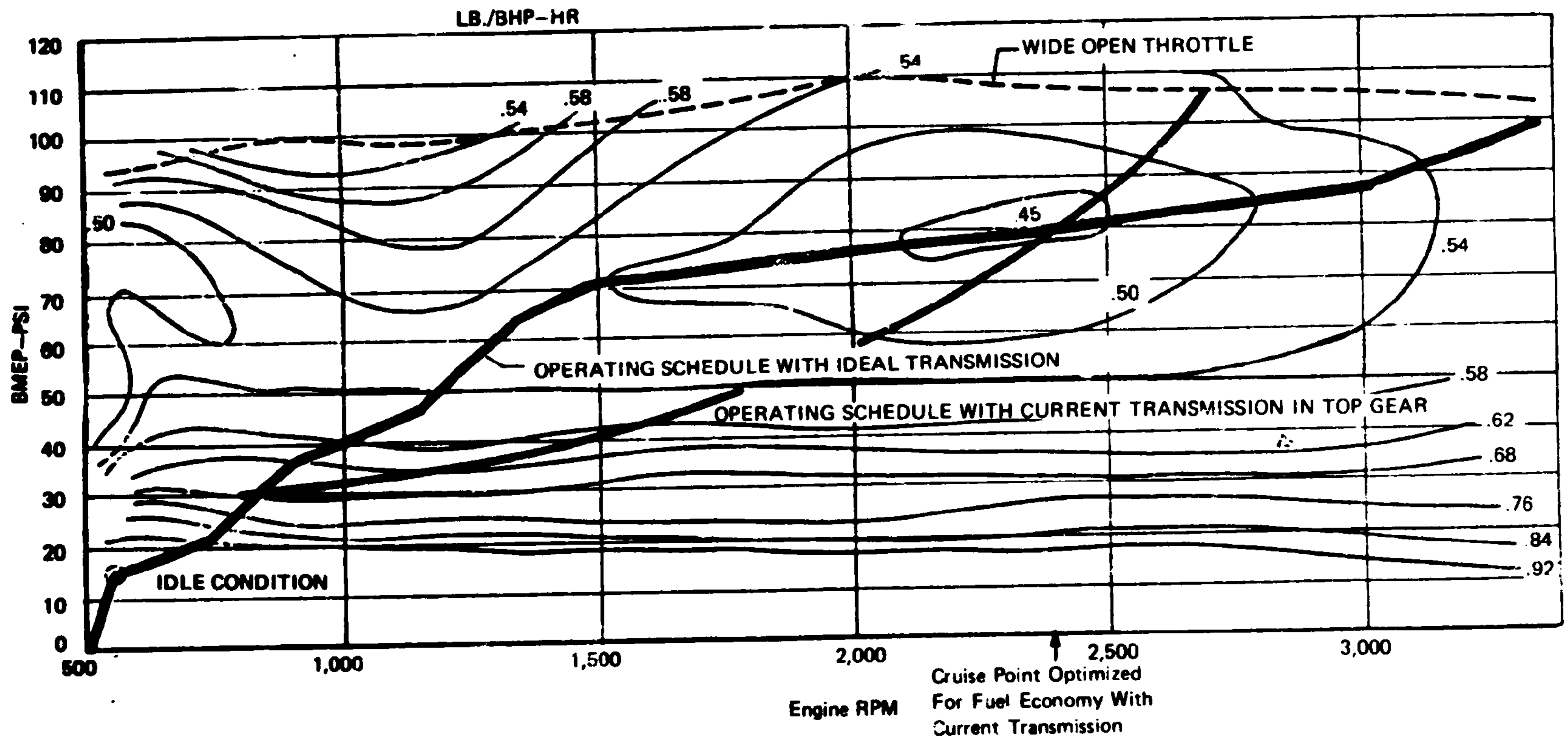
FIGURE 3.7 TYPICAL BRAKE SPECIFIC FUEL CONSUMPTION VS. REQUIRED HORSEPOWER AND VEHICLE ROAD SPEED AT CRUISE CONDITION

and decelerate. It is not enough, then, to design a transmission to keep the engine operating on the line of minimum bsfc vs. speed as in Figure 3.7. This line of minimum fuel consumption can never really be followed, and it is necessary to optimize the fluctuation around that line. For this optimization process, an engine map of brake specific fuel consumption on a plot of output vs. speed is needed. Such a map is shown in Figure 3.8. A line representing an ideal transmission would run through the islands of lowest bsfc at normal cruise conditions and correspond to the bottom-most line of Figures 3.6 and 3.7. However, the true engine demands will fluctuate around this line. During acceleration, the operating point will be above this line, and during deceleration, below it. The transmission designer will select his top gear ratio such that he operates at a point of minimum bsfc. This operating line will be optimized if the normal cruise speed fluctuations and the average grades encountered are such that the engine fluctuates evenly about the ideal point under normal driving conditions most of the time as shown. This optimum is rarely achieved, however, for no two drivers are alike, nor are the grades on every road identical. Furthermore, accessory loads are constantly changing and demanding more or less power from the engine.

3.6 EMISSIONS

Table 3.1 from Patterson[4] illustrates, in a very simplified way, those engine operating variables that affect air pollution emissions from gasoline spark ignition engines. It also details the direction in which pollutant concentrations go with an increase in those variables. A horizontal line indicates no significant change, while an upturned arrow indicates an increase and vice versa. Wherever the intake manifold mass flow is shown to increase, however, total emissions of those pollutants with a horizontal line or upturned arrow will also increase because the total mass flow has increased while the concentration of pollutants in the air stream has not decreased.

The choice of gear ratios and shift points used in an automatic transmission will directly affect engine speed and load. The other variables are only indirectly related to speed and load and therefore may be used



Source: Orshansky[3]

FIGURE 3.8 SPECIFIC FUEL CONSUMPTION MAP OF A V-8 ENGINE (300 CUBIC INCH) to tune an engine to minimize the emissions of a given engine/drivetrain package.

TABLE 3.1

EFFECT OF DESIGN AND OPERATING VARIABLES ON EXHAUST EMISSIONS AND ENGINE AIR FLOW

Variable Increased	HC Conc.	CO Conc.	NO Conc.	CH_2O Conc.	Intake Mass Flow Constant Load
Air-Fuel Ratio					↑
Load	–	–	↑		↑
Speed	↓	–	↑ ↓		↑
Spark Retard	↓	–	↓	↓	↑
Exhaust Back Pressure	↓	–	↓		↑
Valve Overlap	↓	–	↓		↑
Intake Manifold Pressure	–	–	↑		↑
Combustion Chamber Deposits	↑	–	↑		–
Surface to Volume Ratio*	↑	–	–		–
Combustion Chamber Area	↑	–	–		–
Stroke to Bore Ratio	↓	–	–		↑
Displ. Per Cylinder	↓	–	–		–
Compression Ratio	↑	–	↑		↓
Air Injection	↓	↓	–↑	↑ ↓	↑
Fuel Injection	↓	↓	↑		–
Coolant Temperature	–↓	–↓	↑		–

*Engine changes which decrease surface to volume ratio reduce heat loss to the coolant. As a result, NO concentration may increase

To design a drivetrain which will minimize emissions, the designer starts out with a set of emissions maps for a given engine in much the same way as he approaches the fuel economy problem. Figure 3.9 is such a map for Nitrogen Oxides which shows the nitrogen oxide emissions rates as a function of engine output (brake mean effective pressure) and engine speed. The transmission designer would optimize his choice of ratios by attempting to match the load demands made by the vehicle and accessories with the points of minimum NOx emissions. An ideal operating line is sketched onto Figure 3.9 as are typical operating points.

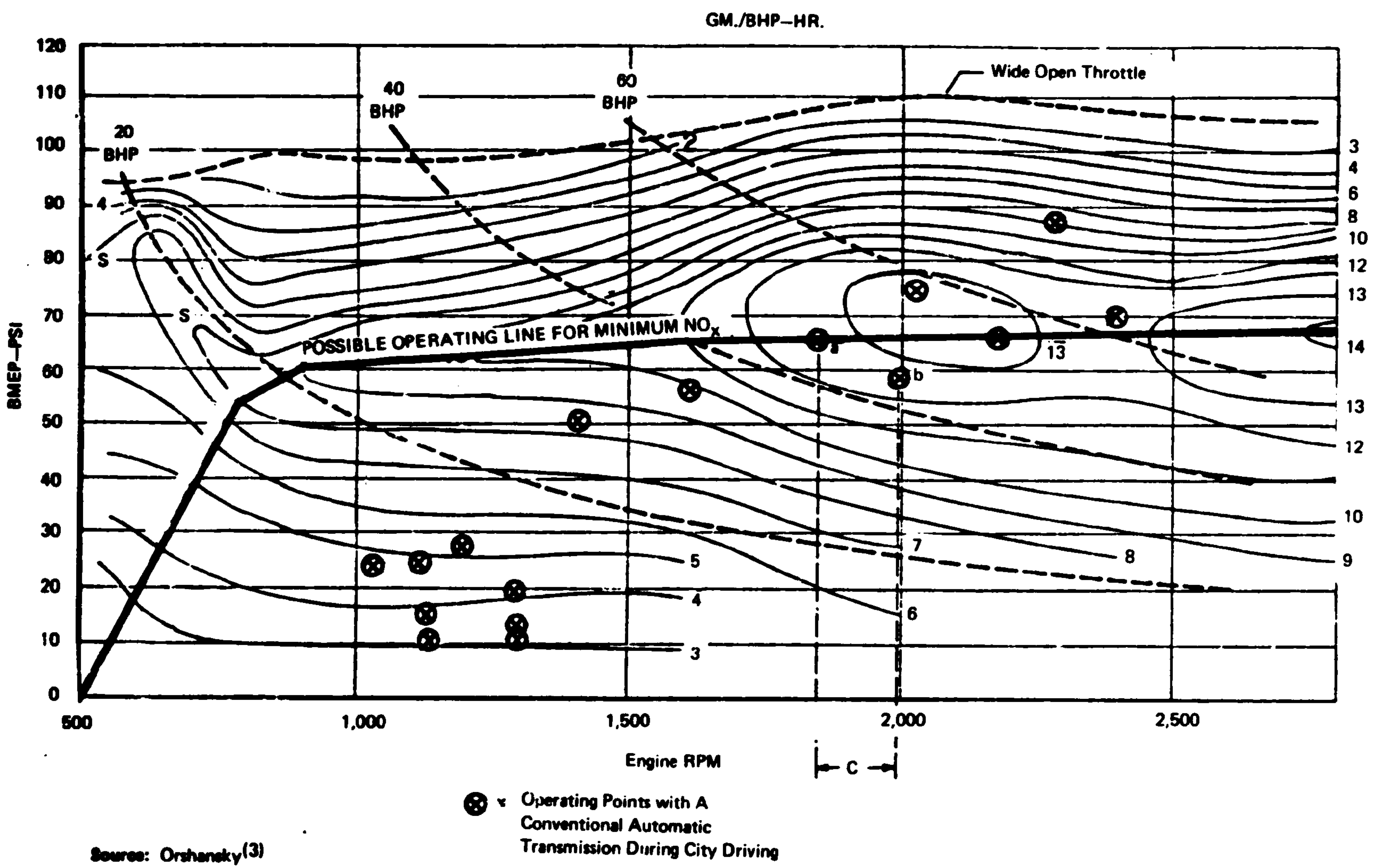


FIGURE 3.9 EMISSION RATE OF OXIDES OF NITROGEN FROM A V-8 ENGINE (300 CUBIC INCH)

Unfortunately, operating lines which minimize NOx do not necessarily minimize hydrocarbons or carbon monoxide. The designer must stay on the narrow path which minimizes all three pollutants.

4. FINE TUNING FOR EMISSIONS, FUEL ECONOMY AND PERFORMANCE

4.1 VARIABLES

The ideal power train would propel a vehicle with minimal pollutant emission, minimum fuel consumption, and have at the same time a maximum acceleration capability. As yet, there are no combinations of engines and transmissions available which can meet all three criteria. Two must suffer for the sake of one. Engine design and operational parameters can be adjusted to bring all three into an acceptable compromise, however.

Today, this compromise is reached in practice by combining an engine already meeting the regulatory emissions requirements with the desired chassis in such a way that the given vehicle requirements (a marketing or "front office" demand) are met. The drivetrain designers are then faced with the task of minimizing fuel consumption without forcing emissions and performance beyond legal and marketable limitations, respectively. These three vehicle requirements are optimized by juggling those parameters that are readily altered on the production line:

- Rear axle ratio (in certain discreet production steps);
- Shift points;
- Air-fuel ratio;
- Spark timing;
- Exhaust after-reaction devices;
- Deceleration coupling

The rear axle ratio and shift points are usually established first as they are the primary factors affecting all three performance requirements. The other parameters are then adjusted to "fine tune" the powertrain. If these six parameters cannot be juggled in such a manner to place the vehicle within the acceptable limits of emissions, economy, and performance, the remaining parameters of Table 3.1 must be adjusted to make the vehicle a saleable product. Alteration of these parameters, however, usually means that a new engine must be designed if one is not now available, or that such a vehicle is unbuildable and the vehicle must either be scrapped, the legal limits for emissions raised, and/or the market demands for gradeability, acceleration, and fuel economy reduced. Let us see how the operating parameters within the control of today's powertrain designer may be altered to optimize the compromise between fuel economy, emissions, and performance.

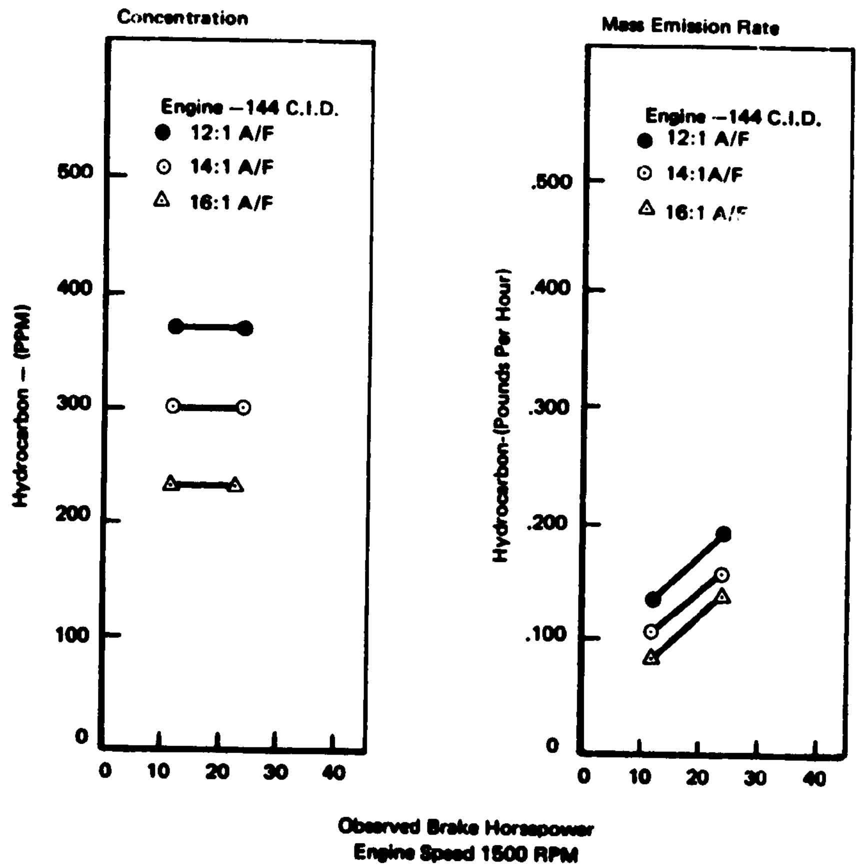
4.2 REAR AXLE RATIO

The overall speed ratio in any given gear determines the engine speed and torque output. Since a drivetrain designer has certain predetermined gear ratios thrus upon him in today's mass production world, he can readily alter only the rear axle ratio to change the engine speed vs. vehicle speed (N/V) ratio.

4.2.1 Effects on Emissions

Increased engine speed at a constant power output generally reduces the mass emission rate of hydrocarbons. This is because the higher speeds create more turbulence within both the combustion chamber and exhaust system. This promotes improved mixing which enhances combustion in the cylinder and after reactions (hydrocarbon oxidation processes) in the exhaust gases. Partially offsetting this effect, however, is the increased air and fuel flow required through the engine to supply the power absorbed by the increased friction losses in the engine and drivetrain due to the higher rotational speeds. Another offsetting factor is the reduced residence time in the exhaust system which limits the available time for the oxidation of hydrocarbons in the exhaust system. Hence, increased engine speeds do not reduce the total mass emissions of hydrocarbons as much as might be expected.

The effect mentioned above of the increased air flow necessary to offset the increased friction loads also illustrated the effect on hydrocarbon emissions of increasing the engine's power output at a constant engine speed. As load is increased, the throttle must be opened more in order to maintain a constant speed. This will increase the flow of air (and fuel) through the engine. If all else remains the same, the concentration of hydrocarbons in the exhaust will not increase, but since the overall mass flow rate of gases through the engine has increased, the mass flow rate of hydrocarbons will also increase. This is illustrated in Figure 4.1.



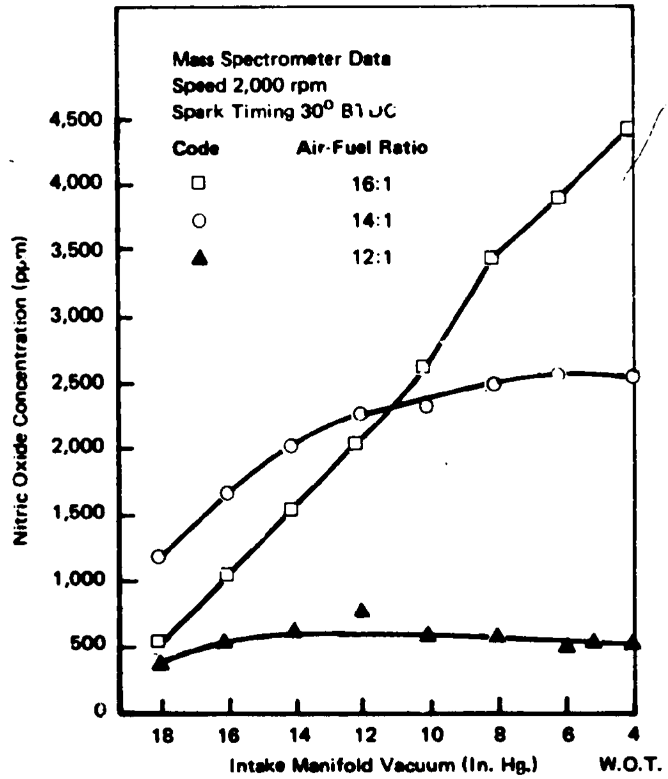
Source: Hagen and Holiday (5)

FIGURE 4.1 EFFECT OF HORSEPOWER ON EXHAUST GAS HYDROCARBON EMISSIONS

Unlike hydrocarbons, the formation of carbon monoxide is limited by chemical reaction kinetics rather than by the physical limitations of mixing. Thus, the formation rate of CO per pound of air-fuel mixture is independent of engine speed and load. However, since air flow is strongly influenced by load (at constant speed) and, to a lesser extent, by engine speed (at constant load), the total mass emission rate of carbon monoxide will increase directly with power output and only slightly with speed.

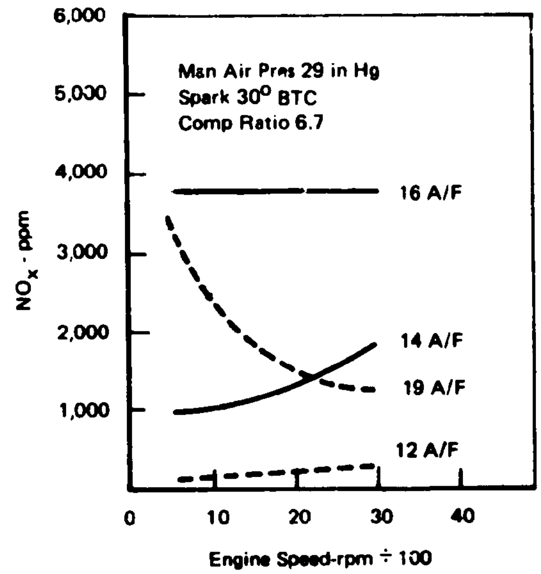
Nitric Oxides (NOx) formation is strongly influenced by the level and duration of the temperatures developed during the combustion process. Factors which increase these temperatures and/or times increase NOx formation and vice versa. Increasing engine load increases compression temperatures, and the concurrent air flow increase reduces the mass of spent fuel remaining in the cylinder due to increased turbulence, mixing, and scavenging. The reduction in volume of spent fuel and higher initial temperature increases the burning rate such that more fuel is burned near top dead center. This results in higher combustion temperatures (because there is less time and surface area available to remove heat from the mixture) which in turn increases the concentration of nitric oxides in the exhaust gases. Such an increase is shown in Figure 4.2. (Note that manifold pressure is proportional to engine load.) Of course, the actual mass emission rate will increase much more markedly due to the increased air flow through the engine.

Engine speed also increases NOx emissions for similar reasons. The greater the engine speed at constant power levels, the greater the turbulence. For commonly used air-fuel ratios of 14 or 15 to 1, this results in an increase in flame speed which causes more of the burning to occur near TDC. This tends to raise combustion pressure and temperature by reducing the time and surface area available for heat transfer away from the mixture. The effect of engine speed upon nitrogen oxide concentrations is shown in Figure 4.3. The slight effect of increasing air flow with increasing engine speed at constant load will tend to exacerbate these NOx increases.



Source: Patterson[4]

FIGURE 4.2 EFFECT OF ENGINE LOAD ON NO_x CONCENTRATION LEVELS IN EXHAUST GASES



Source: Patterson [4]

FIGURE 4.3 EFFECT OF ENGINE SPEED ON NITROGEN OXIDE CONCENTRATIONS IN THE EXHAUST GAS

4.2.2 Effects on Fuel Economy

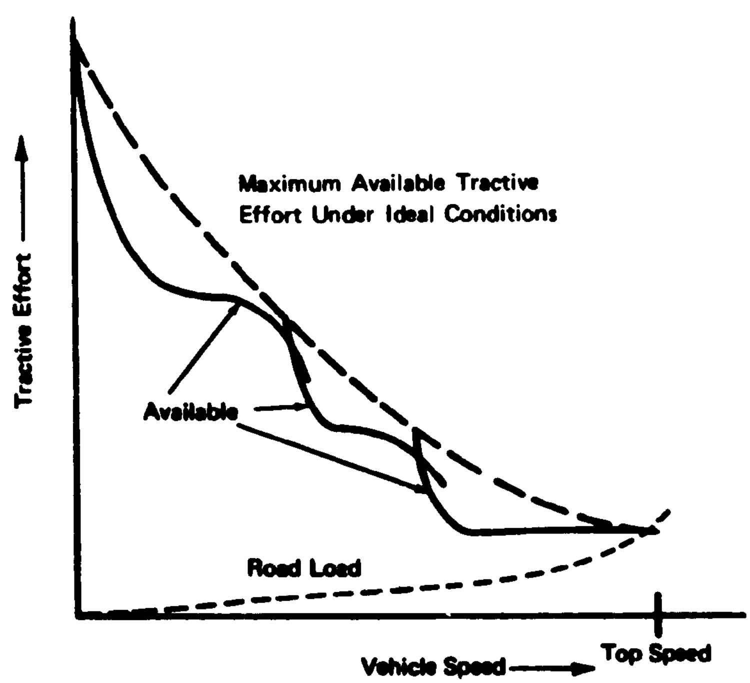
A review of Figure 4.3 will serve to illustrate the effects of rear axle ratio on fuel economy. The brake mean effective pressure is proportional to engine (vehicle) load. The rear axle ratio will determine the engine rotational speed at which the vehicle load is met. The closer the drivetrain designer can force the engine to operate on the ideal line shown on Figure 3.8, the better the fuel economy. Since current production practice is to operate below this line, it can be stated that, in general, the lower the rear axle gear ratio, the better the fuel economy because the spread between the actual and the ideal curve has been reduced.

4.2.3 Effects on Performance

Consider Figure 3.5 again, redrawn as Figure 4.4, with the addition of a line representing the required tractive effort (road load) vs. vehicle speed. The rear axle ratio will determine the vertical location of the line of available tractive effort. The dashed line representing the maximum attainable tractive effort under ideal conditions. The solid line is that effort actually available at wide-open throttle at any road speed, while the dotted line is that effort required to propel the vehicle at a steady speed. The vertical space between these latter two lines represents the excess power available to accelerate the vehicle. The rear axle ratio will move the solid line up or down, vertically. The higher the axle ratio (numerically), the higher the solid line and the more power available for acceleration.

4.3 SHIFT POINTS

Shift points affect engine speed and load in much the same way as rear axle ratios. By altering a shift point up or down, one can alter the engine vs. road speed relationship. Hence, if higher engine speeds are desired, the shift points are raised in order to force the engine to run at the desired speed for the same range of road speeds. Unlike rear axle ratio changes, however, the effects of shift point changes are only effective over the road speed range defined by the old and new shift points.



Source: Arthur D. Little, Inc.

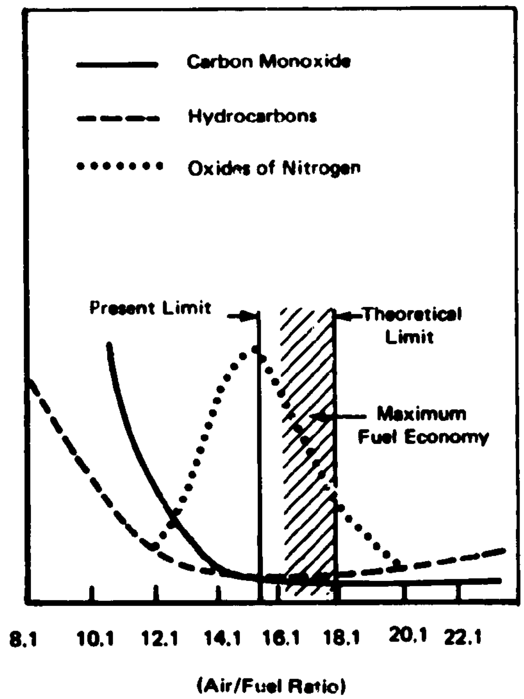
FIGURE 4.4 TRACTIVE EFFORT VS. VEHICLE ROAD SPEED

4.4 AIR-FUEL RATIO

4.4.1 Effects on Emissions

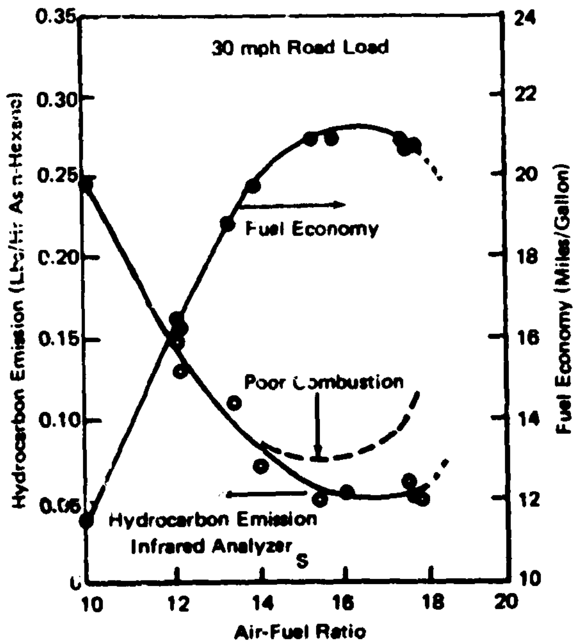
The air-fuel ratio affects the thickness of the quenched layer of fuel-air mixture near the "cold" combustion chamber walls, the concentration of fuel within that unburned layer, peak combustion temperature, and the quantity of excess oxygen available in the exhaust gases to oxidize unburned hydrocarbons and carbon monoxide. With rich mixtures, there is insufficient oxygen to completely burn all the fuel and completely oxidize the carbon, resulting in high emission rates of unburned hydrocarbons and carbon monoxide. Rich mixtures also burn relatively cold, which further promotes incomplete combustion and inhibits the formation of nitrogen oxides.

At an air-fuel ratio of approximately 15.5 to 1, peak combustion temperatures are reached. Here, too, there is sufficient oxygen to completely burn most of the fuel and fully oxidize the carbon. Thus, an air-fuel ratio of 15.5 to 1 results in peak NOx output and minimal hydrocarbon and carbon monoxide emissions. Current automotive carburetion and ignition systems are limited to this lean limit of 15.5 to 1 due to fuel distribution and flame propagation difficulties. If, however, an automotive engine could be made to run at even leaner mixtures, nitrogen oxide emissions would again be reduced as combustion temperatures dropped. Above air-fuel ratios of 18 to 1, "poor" combustion becomes a problem as temperatures are low enough to inhibit flame propagation and quenching becomes significant even under ideal conditions. This is illustrated in Figures 4.5 and 4.6.



Source: Chrysler Corp. (6)

FIGURE 4.5 EFFECT OF AIR/FUEL RATIO ON EMISSIONS AND FUEL ECONOMY



Source: Patterson[4]

FIGURE 4.6 RELATIONSHIP BETWEEN FUEL ECONOMY, HYDROCARBON EMISSIONS, AND AIR-FUEL RATIO

4.4.2 Effects on Fuel Economy

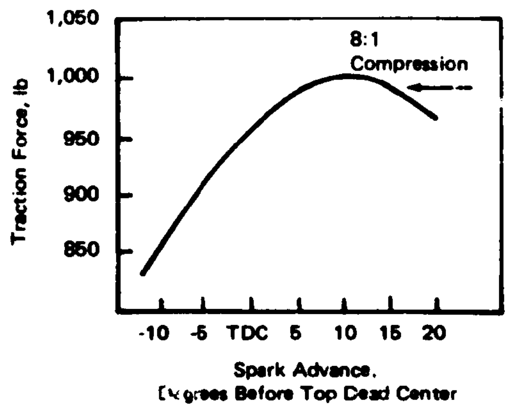
Maximum fuel economy is achieved when there is sufficient oxygen present to insure that all the fuel is consumed. This occurs in lean mixtures which have an excess quantity of air (more air [oxygen] is present than is necessary to completely oxidize all the fuel under ideal, [air-fuel ratio of 15 to 1] stoichiometric conditions). As shown in Figures 4.5 and 4.6, this occurs at an air-fuel ratio of approximately 16 to 1. Beyond this limit, combustion temperatures are reduced and mixing is limited to the point that complete consumption of all the fuel becomes increasingly difficult. Beyond a ratio of 18 to 1, these effects become dominant, and fuel economy drops markedly.

4.4.3 Effects on Performance

Just as maximum economy is achieved when all the fuel is consumed, maximum performance (power) is achieved when all of the oxygen

4.5.3 Effects on Performance

Accompanying the reduction in BMEP due to retarded spark timing is a reduction in power output which is illustrated in Figure 4.9.



Source: Mark's Handbook(8)

FIGURE 4.9 EFFECT OF SPARK TIMING ON AVAILABLE TRACTIVE EFFORT (ENGINE POWER OUTPUT) AT WOT FROM A TYPICAL AUTOMOBILE ENGINE AT A GIVEN SPEED

4.6 EXHAUST AFTER-REACTION DEVICES

4.6.1 Effects on Emissions

These emission control devices consist of either air injection units or catalysts which enhance the oxidation of hydrocarbons and carbon monoxide in the exhaust system. Air injection systems usually consist of an air pump which directs a stream of air into the exhaust manifold just downstream of the exhaust valve. Here temperatures are sufficiently high that the added air will initiate combustion of carbon monoxide and oxidation of unburned hydrocarbons. The actual effectiveness of these systems varies with the actual fuel-air ratio and initial exhaust gas temperature.

Catalysts may also be used to oxidize carbon monoxide and hydrocarbons. Depending upon conditions, catalysts are capable of controlling emission rates to less than 1% of base (uncontrolled) engine rates.[4]

4.6.2 Effects on Fuel Economy and Performance

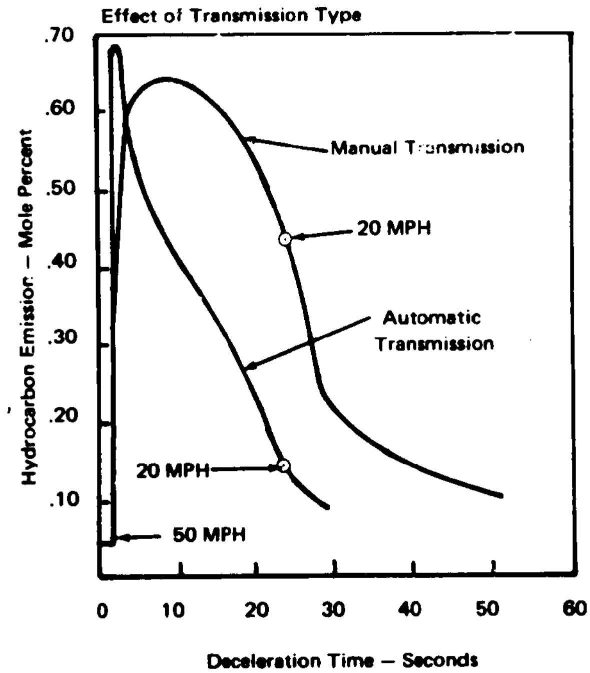
Air injection systems require engine power to run the pump and hence increase the engine load while decreasing the power available to propel the vehicle. Some of the power loss may be regained by running at a richer fuel-air ratio than a non-injected engine which must run lean to reduce emissions to levels that are better achieved with air injection. However, both the power absorbed by the pump and the richer mixture may contribute to increased fuel consumption.

Catalysts, on the other hand, do not require any power from the engine. In addition, they allow an engine to operate at fuel-air ratios and spark timing for optimized economy and NOX emissions. Hence, the use of catalysts allows the engine to be tuned for economy (or performance) within NOx emission limits while controlling HC and CO emissions without parasitic power losses.

4.7 DECELERATION COUPLING

During deceleration under engine braking, the throttle is closed and the intake manifold pressure may be quite low. When this occurs, considerable quantities of burned exhaust gases remain in the combustion chamber and inhibit proper combustion.[9] The more rigid the coupling between the engine and the rear wheels, the more air-fuel mixture is pumped through the engine, the lower the manifold pressure, and the more unburned fuel is emitted from the exhaust pipe. This coupling effect is illustrated in Figure 4.10 which demonstrates the difference in deceleration hydrocarbon emissions from two identical "uncontrolled" vehicles except one is equipped with a manual transmission (rigid coupling) and one has an automatic transmission (non-rigid coupling).

The drivetrain designer could solve this problem quite simply by completely decoupling the engine from the rear wheels during deceleration. In this way, manifold (and therefore cylinder) pressures would never be



Source: District[9]

FIGURE 4.10 EXHAUST HYDROCARBON EMISSION DURING DECELERATION FROM 50 MPH, EFFECT OF TRANSMISSION TYPE

low enough to retain significant quantities of exhaust gas. The feeling of free-wheeling, however, is not acceptable to most drivers, and other means must be used to combat this problem. To this end, throttle openers which open or retard the closing of the throttle may be employed to keep manifold pressures above critical levels. Alternatively, idle circuit fuel shut-off devices which effectively switch off the fuel supply during deceleration and thereby stop unburned hydrocarbons at their source. This latter method actually improves engine braking, and there is absolutely no force trying to turn over the engine other than the vehicle inertia. The former method reduces engine braking by reducing the pressure differential across the piston. (Lower than atmospheric manifold pressures create lower than atmospheric cylinder pressures during the induction stroke. Since atmospheric pressure is normally maintained in the crankcase, the pressure difference across the pistons retards their motion.) In addition, some amount of motive power is still provided by combustion during the power stroke. To aid in engine braking, this power may be minimized by retarding the spark during deceleration in an effort to maintain the "feel" of engine braking.

4.8 THE BIG COMPROMISE

It is obvious from the above discussion that there is no perfect cure for a particular powertrain fault. If a given system meets all but one of the three desired drivetrain requirements of performance, fuel economy, and emissions, usually a certain amount of the other two must be sacrificed in order to accommodate all three. The requirement that is optimized will be based upon market or legal demands. From 1968 through 1974, emissions were optimized to meet legal demands. Now attempts are being made to improve fuel economy without exceeding the legal limitations on emissions. The ultimate success of this effort involves juggling many, many more operational parameters than those under the control of the drivetrain designer.

Nonetheless, the drivetrain system will be designed in such a way that the various demands made upon the vehicle will be met, and the design process will be one of compromise. This compromise will be the result of many redesigns and evaluations of the original plan involving both paper and hardware studies as shown in the design flow path outlined in the Decision Tree for Drivetrain Design, in Figure 1.1.

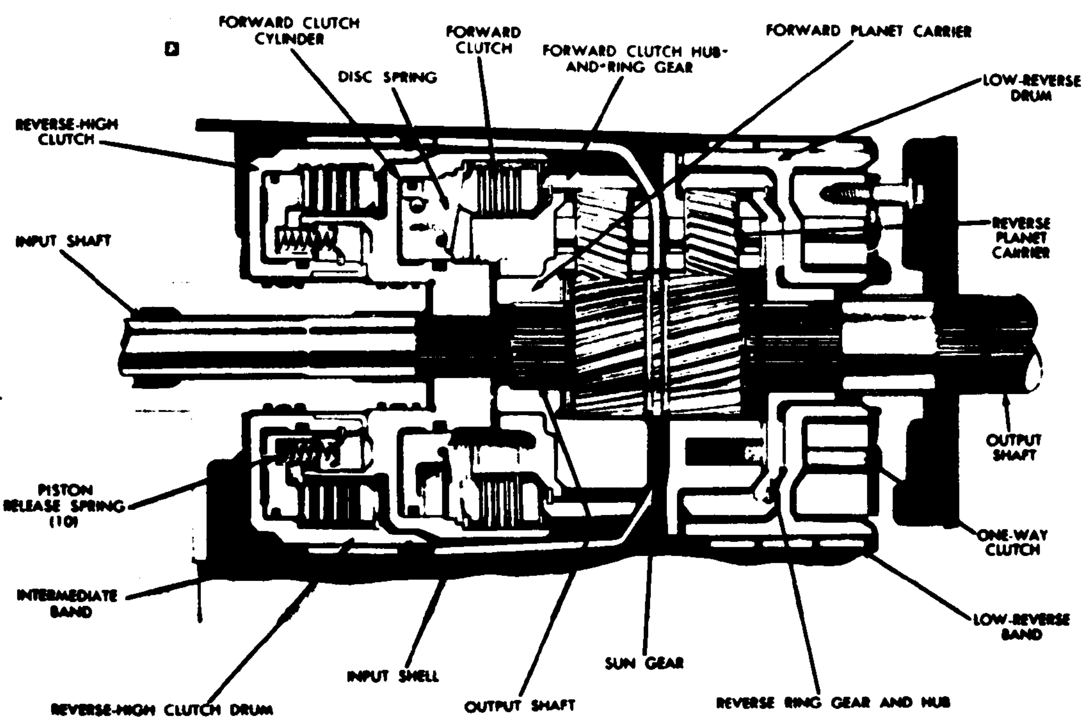
5. TRANSMISSION DESIGN LIMITATIONS

5.1 BACKGROUND

As the Decision Tree indicates, the drivetrain designer is constrained by economic and physical limitations as to his choices of components. The early parts of this text have already mentioned that he must usually work with those transmission gear ratios currently in production in order to avoid the expense of two separate production units. Other factors such as tooling costs and physical dimensions also constrain the design.

5.2 ECONOMIC FACTORS

Current production line practice is to utilize a "Simpson" type planetary gear set. This configuration utilizes two identical planetary gear sets having a common sun gear and interchangeable planet and ring gears. Only one production line is needed to manufacture each, whereas if two or more sizes of each were used, the number of gear cutting lines would double at least. A Simpson gear set as utilized by Ford is illustrated in Figure 5.1.



Source: Commercial Trades Institute[16]

FIGURE 5.1 SIMPSON PLANETARY GEAR SET AS USED IN A FORD AUTOMATIC TRANSMISSION

Use of such a Simpson-type gear set limits the choice of gear ratios available to the drivetrain designer. The method of operating a Simpson gear set dictates that if "N" is the ratio of low gear, "N-1" is the ratio of second gear, and top gear is direct. Hence, the designer is limited to the choice of gear steps in order that they can be produced least expensively.

The flexibility of transmission gear ratio choices is limited by the economies of production. It is estimated that in order to economically justify production of two different gear sets, at least 500,000 transmission units utilizing such gears would have to be produced every year.[10]

In order to justify a new transmission, the drivetrain designer must show that the benefits associated with new gear ratios are worth the increased cost. There would be a 10-15% cost increase to produce 3-speed Simpson transmission units with ratio ranges wider than 2.77 or for any four speed. Such units require very large annular gears which would dictate larger cases (larger physical size) in order to transmit the same torque as today's units. The transmission industry estimates that a cost increase of 10-15% would result from the production of large quantities of four-speed units with torque carrying capacity equal to current units. There would be no significant unit cost increase if a different ratio 3-speed transmission were produced as long as it could fit into current casings where the diameter of the annular gear limits the maximum range to about 2.77.[10] Little is to be gained, however, in moving to 2.77 from current units whichalready have ranges of around 2.5. As one moves higher than 2.5, also, one encounters driveability problems which are to be discussed later. In general, benefits which justify a unit cost increase of 10 to 15% will allow production of a new transmission.

5.3 PHYSICAL LIMITATIONS

We have already seen that present transmission cases limit the gear ratio range. Units with ranges above 2.77 will not fit into current cases if they are to handle current engine output torques. Physical size limits the choice of gear ratios in several ways other than dollars and cents. Dimensions affect producibility, weight, durability, and suitability. There is no clear-cut distinction between all four parameters, however. If overall size is immaterial, almost anything can be produced. For instance, in order to manufacture a hypoid rear-axle gear set suitable for use in a car, one must limit the lower ratios to about 2.2 or the gears become undercut. Larger gears would allow lower ratios, but ground clearance would become a problem because of the ring gear size.(11) Durability considerations limit the ratio of suitably sized hypoid gear sets to 2.5.(12) Much lower ratios tend to have unsatisfactory service life.

Durability also limits the ratio range in the transmission. Small ratio changes in a Simpson gear set require large changes in the physical size of the gears. The shear strength of a shaft changes with the cube of its diameter. If one is going to economically machine a sun gear onto a shaft, the shaft must be made of relatively mild steel and hence it must be fairly large. With current technology, shaft root diameters of less than one inch will not reliably transmit the output of today's American V-8 engines. Hence, the shaft limits the size of the sun gear, and the transmission designer is again faced with a factor limiting the overall gear range. Ranges exceeding 2.77 require unsuitably small sun gears(13) or uneconomically large ring gears, and therefore, cases.

The durability or strength of a gear may be enhanced by making the gear wider. While the strength will increase linearly with gear width, so will the weight, which is undesirable.

Weight is a penalty from two standpoints. The heavier a component, the more fuel required to drag it around. In the drivetrain, the effective weight (the equivalent inertial weight during acceleration or deceleration) increases with the square of the shaft speed, so weight is more critical here than anywhere else in the vehicle. The equivalent weight of any rotating part may be determined by the following formula:

Equivalent weight (lb) = $\frac{IG^2}{R^2}$ where

I = rotary inertia lb. - ft.^{2} of parts in question;

G = number of revolutions of the rotating component for each revolution of driving wheel; and

R = radius of driving wheel (ft)[1]

In addition, increased weight of transmission components decreases the durability of clutches and bands because of the greater force required to accelerate or decelerate parts during gear changes. The greater the required force, the less smooth the shift.

Hence, the drivetrain designer is limited to a very narrow range in his choice of gear ratios. Economics, durability, producilibity, suitability, and weight all set bounds on what he may do. The fact that most American cars drive very smoothly and will respond quickly to the touch of the accelerator in spite of these limits is probably due as much to the fact that U.S. engines have very broad, smooth torque and power curves as to the gear-train design itself.

6. DRIVEABILITY

6.1 GOALS

When all is said and done, the vehicle must be driveable. This means that it must be acceptable--no, even desirable--to the public. The vehicle must live up to all their expectations: the power flow must be smooth and vibration-free, the shifts must not be felt, the noise level must be such that the wind is the only auditory intrusion into this living room on wheels. All this is the ultimate in driveability in the eyes of the American public. Above all else, to be marketable, the drivetrain must strive toward these goals.

As the Decision Tree in Section 1 indicates, when the drivetrain package is designed and refined as much as possible, a prototype is built and then tested and tuned for "driveability." This tuning exercise involves searching out all sources of torsional and harmonic vibration, setting the shift points and shift feel. Even before this stage, however, the drivetrain designer strives to make his paper hardware meet all the standard ground rules for driveability.

6.2 TORSIONAL VIBRATION

A reciprocating, internal combustion engine is not a smooth power source but rather it is a pulse generator. The flywheel goes a long way toward smoothing out the individual pulses into continuous rotary motion. If, however, the engine with a massive flywheel was coupled directly and rigidly with the rear wheels, there would still be times when these power pulses would be felt. This is torsional vibration.

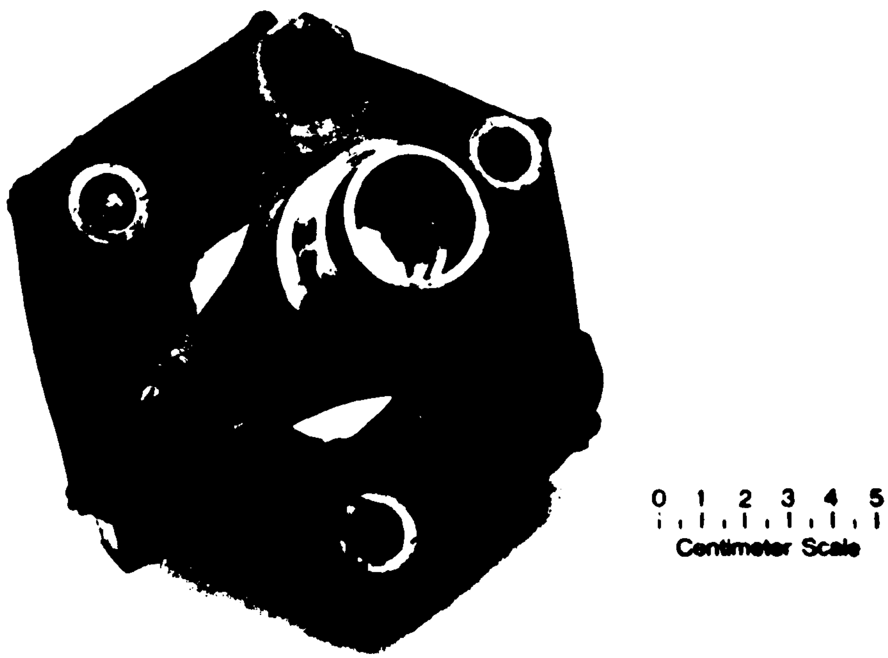
A motor vehicle can be described and visualized as a system of masses and springs interconnected with each other and with considerable damping. Under average driving conditions (which are part throttle and relatively high engine speeds), the frequency of the power pulses coming from the engine is too high, and the amplitude too low, to excite any of these spring-mass units into motion. Under high loads at low speeds, however, the pulses are of such a frequency and amplitude that they can excite some or all of the spring mass systems in the car, and severe vibration could result. This is the form in which the power pulses, or torsional vibrations, manifest themselves, and it is commonly called "lugging".

When a prototype vehicle is tuned for driveability, much time and effort is expended in damping or absorbing these torsional vibrations. The problem is usually attacked in one of two areas, or sometimes both. Compliance can be built into the drivetrain to absorb the torsional pulsations or the suspension of the driven wheels can be designed to dampen the vibrations before they are transmitted to the chassis. Driveline compliance is normally built into the connection between the engine and transmission. In severe cases, compliance can also be designed into the driveshaft and/or axle shafts. In an automatic transmission/torque converter combination, the fluid in the torque converter acts as both a compliant member and absorber. With a mechanical coupling, however, compliance is normally built into the unit in the form of a series of coil springs running concentrically with the shaft, as shown in Figure 6.1. The springs are actually the only connection between the outer, friction surface and the shaft spline. On hard acceleration, however, it is not unusual for these springs to completely bottom out and additional compliance may be necessary. This

Source: Arthur D. Little, Inc.

FIGURE 6.1 TORSIONALLY DAMPED CLUTCH

compliance is normally built into the driveshaft in the form of a rubber, compliant universal joint as shown in Figure 6.2, or a two part compliant driveshaft. The compliant driveshaft consists of two concentric shafts, one inside the other, with approximately 1/4 inch between the inside of one and the outside of the inner shaft. This space is filled with rubber which is bonded to both shafts. In both types of compliant driveshaft, power is transmitted through the rubber which serves to damp and/or absorb the vibrations.



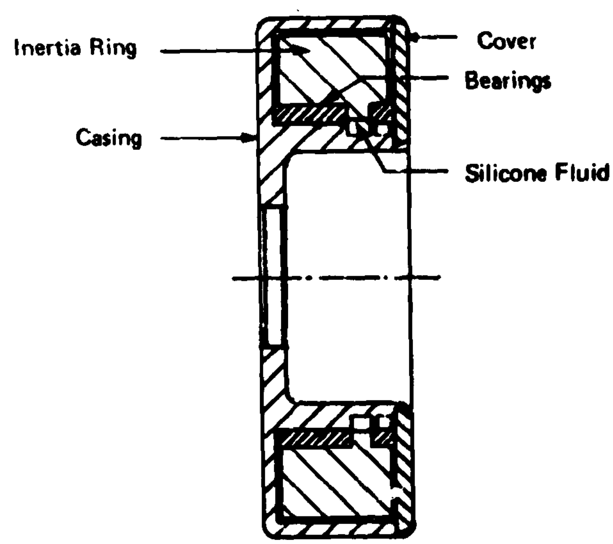
Source: Arthur D. Little, Inc.

FIGURE 6.2 COMPLIANT UNIVERSAL JOINT (ALFA ROMEO)

Torsional vibration, which consists of rotary, power impulses, is transmitted to the chassis by the suspension of the driven wheels when there is not enough compliance in the driveline. The power pulses cause the driving wheels to speed up and slow down and hence push the driven axle forward and back. If driveline compliance cannot reduce the vibrations to acceptable levels, this fore and aft motion must be damped in the suspension. Rubber is used here, again, to isolate the axle locating links from the chassis.

Torsional vibration can sometimes be partially damped through the use of vibration dampers. These units typically consist of a fairly massive (2-5 lbs) element connected through a viscous or elastic medium to the engine crankshaft. This element has enough inertia to rotate at a constant speed and smooth out the torsional pulses by "dragging" the crankshaft along with it via the viscous or elastic connection.

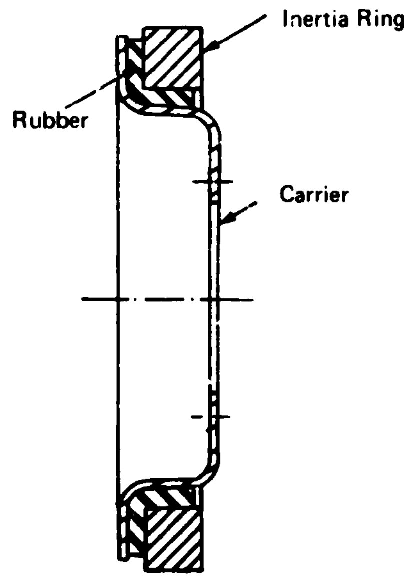
A viscous damper is illustrated in Figure 6.3. The massive inertia ring is driven by a very viscous silicone fluid much in the same manner as a very tight fluid coupling.



Source: Smith[14]

FIGURE 6.3 ARRANGEMENT OF VISCOUS-TYPE VIBRATION DAMPER

The elastic unit, pictured in Figure 6.4, is simpler and more popular. In some automobiles (Chrysler Corp., for instance), the inertia ring doubles as the crankshaft driving pulley.



Source: Smith[14]

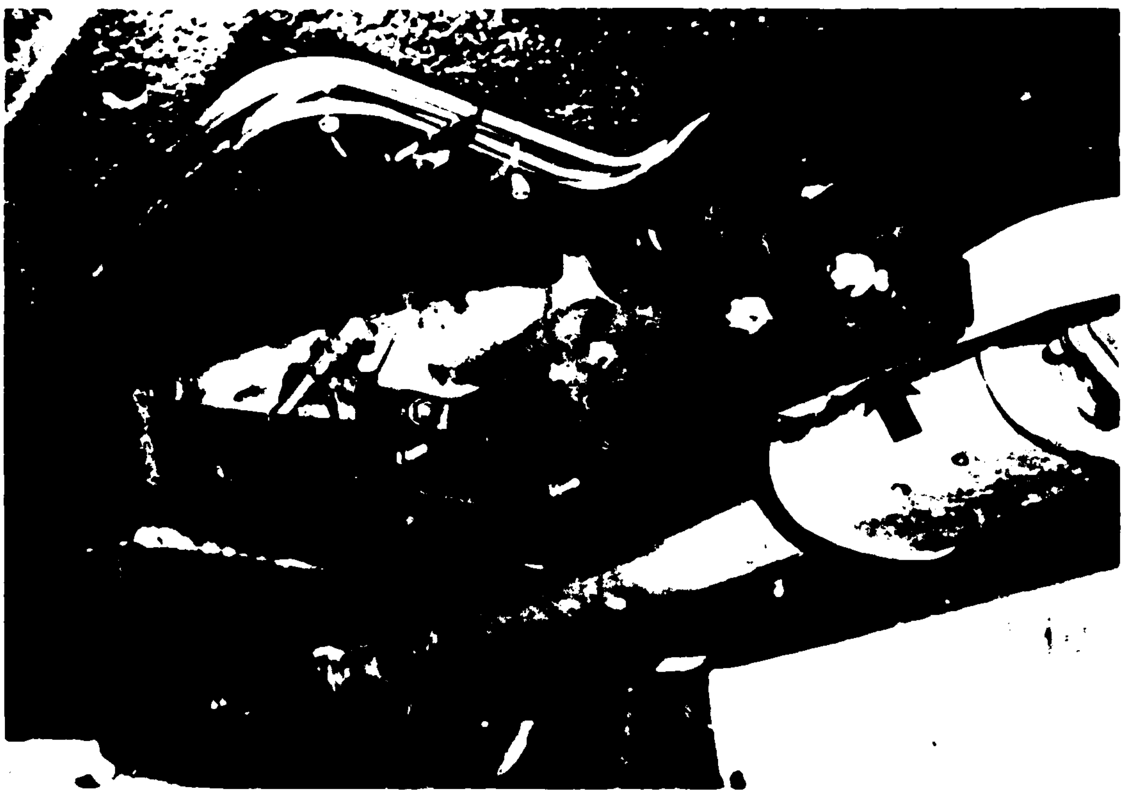
FIGURE 6.4 ARRANGEMENT OF VIBRATION DAMPER WITH BONDED-RUBBER CONNECTING MEDIUM

6.3 HARMONIC VIBRATION

The reciprocating components of the engine can generate another form of disturbance - harmonic vibration. The inherent imbalance or rotating and reciprocating forces in the engine will excite the engine/transmission unit into sympathetic or harmonic oscillatory motion. Unless the power unit is properly located and isolated from the chassis, this motion will cause the entire vehicle to vibrate. Ideally, the engine/transmission package is mounted along the fore and aft axis having the lowest moment of inertia. This axis typically runs from high up on the motor at the front to a low spot on the rear, passing through the center of the flywheel. Considerations of accessory placement and chassis design usually prohibit this, however, and the entire job of damping is left to the rubber engine mounts.

The size and shape of the motor and transmission mount is dependent upon the rotational speeds, the locus of the center of mass, and the weight of the power plant. In the end, however, the final selection for optimum isolation is the result of cut and try tuning.

Harmonic vibrations may also be damped with harmonic balancers, or absorbers. These are nothing more than weights placed on the engine or transmission. They can be a massive front pulley or they can be separate weights suspended on a pendulum arm off the back of the transmission, as shown in Figure 6.5.



Source: Arthur D. Little, Inc.

FIGURE 6 5 HARMONIC BALANCER ON A DODGE COLT

6.4 SHIFTING GEARS

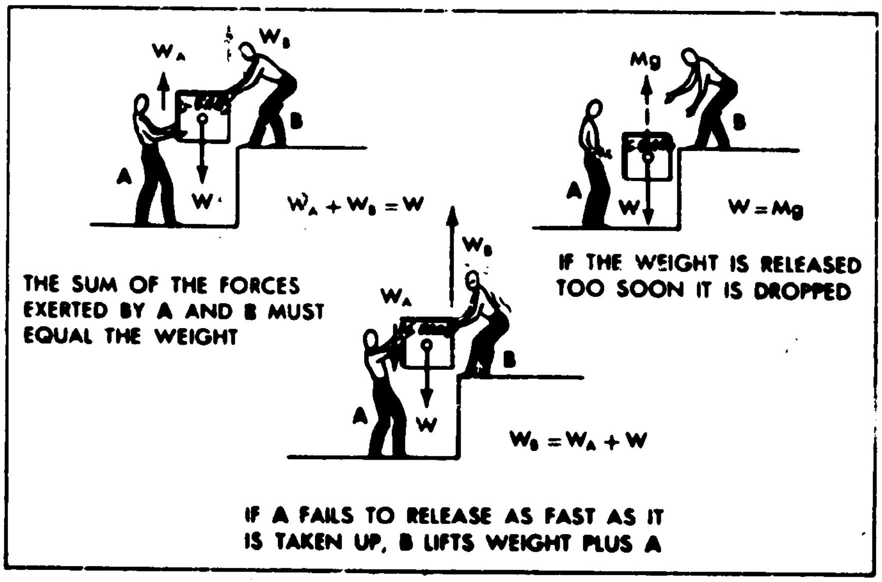
The "Detroit" automakers believe that the ultimate shift is one which is undetectable. While the actual preference of the auto buying public may be open to debate, the end result of it all is that subjective quality known as shift "feel". This shift feel is that which is characteristic of the behavior of the vehicle as the engine speed is changed rapidly while the vehicle speed changes relatively slowly or not at all.

The degree to which a shift is or is not acceptable is dependent upon three factors presented by Winchell and Route:[15]

- "Expectation - The disturbance accompanying an expected shift is more acceptable than that occuring from an unexpected shift. For example, the disturbance consequent to the deliberate movement of the shift lever or downshift detent has a higher degree of acceptability than the disturbance arising from the automatic upshift scheduled by the transmission engineer.
- Sense - The disturbance must be positive and not negative; that is, increased, not decreased, acceleration when the driver expects acceleration, and vice versa. For an example of negative sense in the disturbance, the driver depresses the throttle detent to downshift and experiences a momentary loss of power, or with the throttle adjusted for the desired acceleration, there is a momentary reduction in acceleration during the upshift.
- Environment - An acceptable disturbance for a WOT upshift in the presence of a high level of engine and wind noise, combined with the driver's attention to his speed, will be much greater than that acceptable for the relatively peaceful environment of a light-throttle upshift or coast downshift."

The feel of a shift can best be described by the analogy promulgated by Messrs. Winchell and Route[15] and illustrated in Figure 6.6. If one lifts a box and passes it to another person, the transfer is smoothest if the combined forces of both people never exceed the weight of the box.

If the total force exerted becomes less than the weight of the box, the box falls. Conversely, if the total force is even greater than the weight, the box is jerked upwards. If the first person fails to let go of the box, the second person strains to lift the combined weight of the box and the person and nothing moves. In the same way, if the output torque of a drive-train is constant during a shift, the fact that the gear ratios have been changed is undetectable. If, however, the output torque drops, the car slows down; if the torque is increased, the car speeds up. If one gear fails to release while another engages, the transmission binds up and the car stops.



Source: Winchell and Route (15)

FIGURE 6.6 ANALOGY OF A GEAR CHANGE

The transmission designer's job is to change gears as smoothly as possible without dropping or jerking the box and without binding up any gears. This is done most smoothly in a Simpson gear set which requires only minimal engagement of various clutches and/or bands to progressively shift. The rate at which the clutches are engaged determines the smoothness with which the box is passed. With other gear sets one clutch may have to be released when another is applied. The rate of application and release of two clutches simultaneously complicates the passing of the box and, unless things are done under perfect control, the box will drop or the system will bind up.

While determining the rate of application of the clutches or bands is a cut and try effort, there are certain things a designer can do to minimize the chances of unacceptable shifts. Basically, there are three things that can be controlled on paper:

- The complexity of the shift (the number of boxes);
- The inertia to be overcome (the weight of the box);
- The gear ratio changes (the height to which the box must be lifted).

The complexity of the shift has already been discussed. The more clutches and/or bands that must be simultaneously applied and/or released increases the chances that the timing will be off and a box may be dropped or gears will temporarily bind. Hence, the Simpson gear set has been almost universally adopted as the simplest gear set to shift smoothly.

The inertia of the individual transmission components plays a more significant role than is commonly appreciated. During an upshift or downshift, the speed of various gear sets must be changed in a matter of a few tenths of a second. It is easy to see that a heavy gear set will demand or release more power to downshift or upshift than a light one. This power is removed from or added to the driveline and can be large enough to cause a change in vehicle speed. While it can be overcome to a certain extent by lengthening the engagement time, this solution decreases the life of the engaging, friction element. The longer duration of the shift also contributes to poor shift feel as the car appears to hesitate between gears.

Of all the details that the designer can control, perhaps the most significant is the magnitude of the step from one gear to another. On the surface, it would appear that the magnitude of the ratio step should not make any difference in shift feel as long as the shift occurs at a point where the before and after output torques are equal. When one starts to consider how this shift is accomplished, however, it becomes clear that it is not quite so simple. The larger the step, the greater the difference in engine speed and noise and the more noticeable the fact that a shift is occurring. Also, as the speed differential increases, more and more power is required to accelerate the gear train--the required power increases with the square of the rotational speed--and the greater the power input to, or loss from, the drivetrain. Of course, the acceptability of these effects is purely a matter of opinion, and the maximum allowable ratio change depends upon the philosophy of the car builder. General Motors, for instance, will not accept a ratio change over 1.8[11] while Borg-Warner is willing to try steps as wide as a factor of 2[10].

7. SUMMARY

If two things are clear about the process of drivetrain design, they should be:

1. There is no single path to follow in drivetrain design, but rather a multitude of forks in the road. The final design depends as much upon those factors that the designer feels will be acceptable to the car buyer as those vehicle parameters of top speed, acceleration, and grade climbing ability.

2. The drivetrain design process is not and cannot be done entirely on paper. The final design parameters and tuning details can only be done by cut and try prototype construction and testing.

The characteristics of the final product depend upon the operational vehicle parameter which has been optimized. To review, Table 7.1 presents the optimized parameters and the generalized drivetrain characteristics which are inherent in that optimization process. As the table illustrates, no one set of characteristics can optimize all the vehicle parameters: a choice must be made. The choice made by the designer will ultimately determine who will purchase the vehicle he has engineered. The marketplace, then, is the ultimate drivetrain designer.

TABLE 7.1 THE OPTIMIZATION PROCESS

OPTIMIZED VEHICLE PARAMETER	DRIVETRAIN CHARACTERISTICS
Acceleration and Gradeability	High numerical gear ratios High speed shifts Richer than stoichiometric fuel-air ratios Advanced spark timing Low equivalent weights No parasitic engine devices
Fuel Economy	Low numerical gear ratios Low speed shifts Leaner than stoichiometric fuel-air ratios Advanced spark timing No parasitic engine devices
Emissions:	
Low Hydrocarbons and Carbon Monoxide	High numerical gear ratios High speed shifts Leaner than stoichiometric fuel-air ratios Retarded spark timing Parasitic engine devices (air pumps) Loose engine/transmission deceleration coupling
Low Nitrogen Oxides	Low numerical gear ratios Low speed shifts Much richer or much leaner than stoichiometric fuel-air ratios Advanced spark timing
Driveability:	
Low Noise Good Acceleration Smooth Shifts	Mid-range numerical gear ratios Slightly rich fuel-air ratios Low speed shifts Low equivalent weights Parasitic engine devices (air conditioners) Tight engine/transmission deceleration coupling

APPENDIX A

EXAMPLE OF THE PRELIMINARY SELECTION OF A DRIVETRAIN

Introduction

This appendix will lead the novice drivetrain designer through the preliminary steps of designing the torque converter speed and torque ratios, the transmission gear ratios, and the rear axle ratio. In this design process it is assumed that the "corporate demands" have already defined the vehicle and specified the engine. This preliminary design process will provide the first cut of a drivetrain which will meet the primary "market demands" of acceleration, grade climbing ability (gradeability), and top speed. Once this preliminary design has been achieved, the designer can utilize the information in the preceeding text to fine-tune the preliminary design.

The Vehicle and Its Performance

The corporate "front office" has decided to market a 2000 pound vehicle with an engine having the WOT torque and power characteristics shown in Figure A-1 and is equipped with emission control systems suitable for a car of this weight. The vehicle is equipped with tires having a nominal rolling radius of one foot.

The body designer and chassis engineers have plotted a preliminary road load function shown in Figure A-2. Marketing has demanded a minimum top speed of 90 mph on a 10% grade because of its feeling for the desires of the people who will purchase such a car.

The Drivetrain Design

Gradeability

In order to start on a 30% grade (16.7°) a vehicle of weight W will require a traction force as shown in Figure A-2. This force times the radius of the wheel, R, is the torque required to start the car on a 30% grade.

575 lbs x 1 ft = 575 ft lbs required

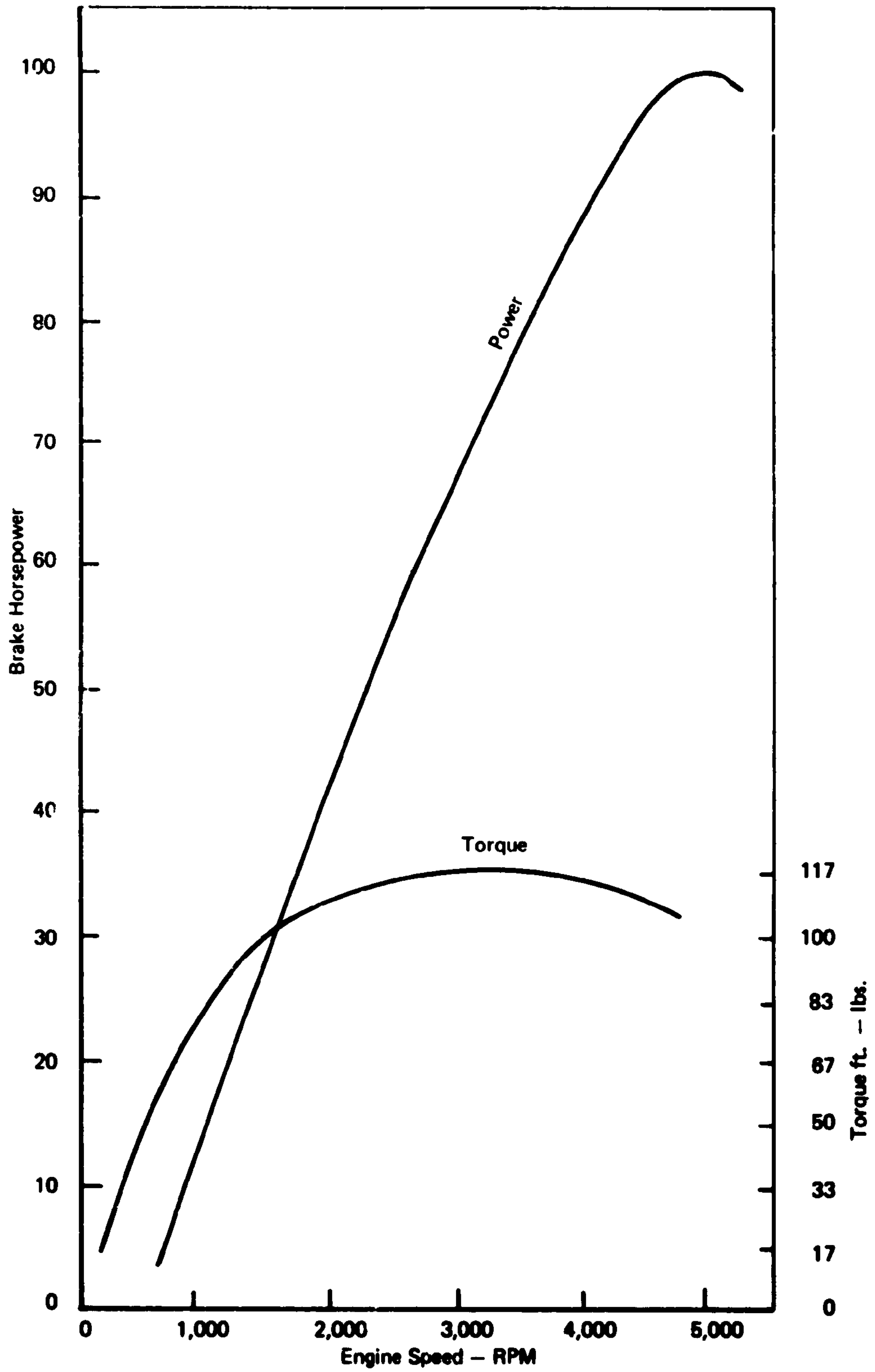


FIGURE A–1 ENGINE OUTPUT 2000 CC. (WIDE OPEN THROTTLE)

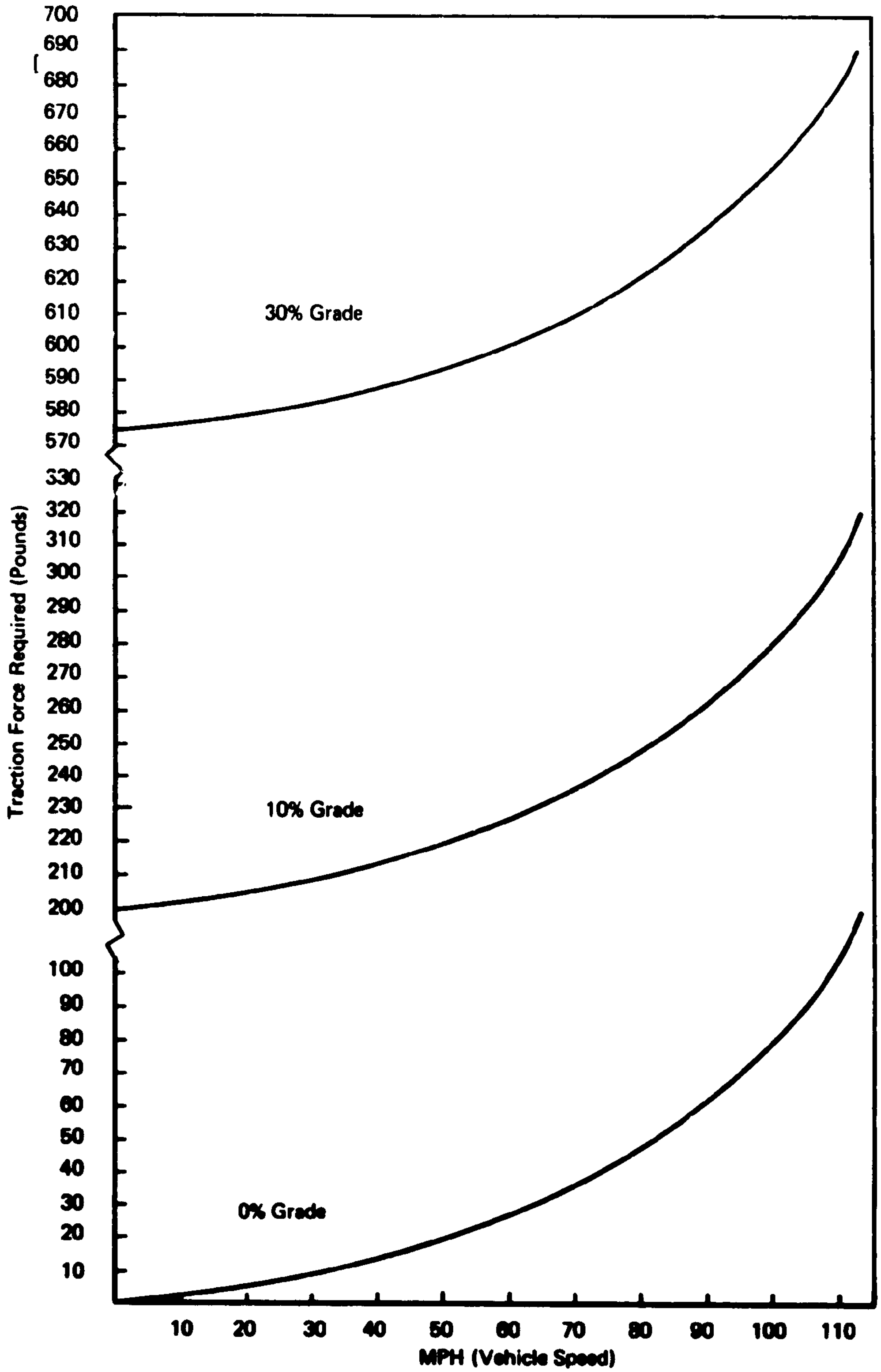


FIGURE A–2 ROAD LOAD FOR DESIGN VEHICLE

Top Speed

The traction effort required at the designated top speed on a 10% grade is obtained from Figure A-2 and is 260 pounds. This force, multiplied by the rolling radius of the tire, determines the torque required of the power train to propel the car and that speed.

$$260 \text{ lbs} \times 1 \text{ ft} = 260 \text{ ft lbs}$$

The power required is:

$$\frac{270 \text{ lbs} \times 1.5 \text{ miles/min} \times 5280 \text{ ft/mile}}{33000 \text{ ft lbs/min/HP}} = 62.6 \text{ HP}$$

Clearly, the engine is capable of propelling the car at speeds higher than 90 mph. The choice must be made between using a smaller engine or accepting a higher top speed. The higher top speed is chosen in this case.

For reasons explained in Section 3.2, we want the engine to operate just beyond peak power and vehicle top speed. At this point, the available power is 99 HP and 5200 rpm. This will propel the car at approximately 115 mph on a 10% grade (Figure A-2).

Rear Axle Ratio

The rear axle ratio must allow the engine to develop the required horsepower at the top vehicle speed, assuming that the transmission is in top gear and the torque converter is coupled. At 115 mph, the one foot radius wheel is turning at a rate of

$$\frac{115 \text{ mph}}{60} \times 5280 \text{ ft/mile} \times \frac{1}{2\pi(1)}$$

The engine must run at 5200 rpm, therefore the required rear axle ratio is

$$\frac{5200}{1611} = 3.23 \text{ to } 1$$

Torque Converter and First Gear Ratio

The engine output, rear axle ratio, and gradeability requirement will govern the first gear and torque converter ratios. The gradeability requirement is that the vehicle start on a 30% grade and move and a speed of at least 5 mph. The required starting torque from Figure A-2 is

$$575 \text{ lbs} \times 1 \text{ foot tire radius} = 575 \text{ ft lbs}$$

The power required for 5 mph operation is:

$$576 \text{ lbs} \times \frac{5 \text{ mph}}{60} \times 5280 \times \frac{\text{HP}}{33000 \text{ ft lbs/min}} = 7.7 \text{ HP}$$

Figure A-1 indicates that the engine must run at at least 800 rpm in order to obtain the desired power. Obviously, if we can obtain the necessary torque anywhere above this speed, obtaining adequate power will be no problem.

The torque required is 576 ft lbs at 5 mph, while at startup it is 575 ft lbs. Clearly, in order to propel the car at any speed along a 30% grade, we must design the gears so that adequate engine torque is available, once the vehicle begins to move, to propel the car at some speed.

We will therefore desire to operate the engine at a speed less than that at which peak torque is developed, 3200 rpm. At the same time, we do not want to require a torque converter with a large stall ratio because we know that they are not as efficient in coupling as those with low ratios. (See Figure B-4 in Appendix B)

If we arbitrarily allow the engine to run at 1000 rpm at startup to minimize noise, the overall first gear ratio required is:

$$\frac{\text{Required Torque}}{\text{Available Torque}} = \frac{575 \text{ ft lbs}}{70 \times 3.23} = 2.55$$

This overall first gear ratio can be obtained with a torque converter alone, a first gear alone, or any combination of the two. Since it is much lower than most overall first gear ratios normally required, we know that the 30% grade requirement is not limiting for this vehicle design. Again we have the choice of using a smaller engine or accepting a greater grade climbing ability. We will do the latter.

We will choose our overall first gear ratio to maximize acceleration without allowing wheel spin. The coefficient of friction of the rear wheels is approximately .8, and each carries a load of 500 pounds. The maximum torque that can be transmitted to the rear wheels before they spin is therefore:

$$[(500 \times .8)+(500 \times .8)]\times 1 \text{ ft} = 800 \text{ ft lbs.}$$

Again, if we allow the engine to operate at 1000 rpm at startup, the overall first gear ratio will be:

$$\frac{800 \text{ ft lbs}}{70 \times 3.23} = 3.54$$

Again, any combination of torque converter and first gear ratio can be used to achieve this.

In this example, the designer will use the corporate standard gear box with a first gear ratio of 2.45 to 1. The required torque converter stall torque ratio is therefore:

$$\frac{3.54}{2.45} = 1.44 \text{ to } 1$$

and the stall speed is 1000 rpm.

The performance characteristics of the torque converter as the speed ratio increases from 0 to 1 are fixed by the stall torque ratio. For further discussion, the reader is referred to Appendix B and the SAE Design Practice handbook for Passenger Car Automatic Transmissions.

Second or Intermediate Gear Ratios

As explained in Section 5.1, cost will dictate that a Simpson gear set be utilized in this transmission. This automatically sets the second, or intermediate gear ratio at 1.45 to 1.

Shift Points

In the main part of the text, Section 3.3 illustrates the method which would be used to set the shift points of an engine with a broad power curve as shown in Figure 5. In this example, only the wide open throttle power curve is shown, and it is markedly peaked. It would be obviously impractical to shift over an engine speed range of 4600-5200 rpm. At wide open throttle, the upshift will occur at an engine speed of 4800 rpm, corresponding to 95% of peak torque as explained in Section 3.3, and the fact that the shift will not be smooth will have to be accepted. Part throttle shifts are determined in the same way using part throttle engine output curves.

Summary

The preliminary drivetrain design has been developed for this hypothetical vehicle. The reader has seen that most of the decisions made in the drivetrain design have been arbitrary, and that the alternative choices would have been just as good. The basic philosophy of the car builder will govern the choices made. In this example, we chose the preliminary design for optimum performance with minimum noise. The alert reader will have no doubt noticed that the gradeability requirements could have been met with an off-the-shelf torque converter having a stall ratio of 2.23, completely eliminating the need for gear reductions in the transmission. Depending upon the desire of the car builder, this would have been a choice as equally viable as the one we made.

APPENDIX B

DEVELOPING TORQUE CONVERTER CHARACTERISTICS WHEN ACTUAL DATA IS NOT AVAILABLE

Introduction

Previous sections have described the process of selecting the stall torque ratio and speed of the torque converter for use in a vehicle.

It is often desirable and, in the case of vehicle computer simulation work, mandatory to know the complete operational characteristics of a torque converter over its full operational speed ratio range. Unfortunately, this data is not always readily available and sometimes cannot be readily obtained due to its proprietary nature. This section presents a method by which torque converter characteristics may be approximated once the stall torque ratio and diameter are known.

The section entitled Torque Converter/Engine Map will be useful even for those who have actual converter data. This section details the means by which engine and torque conversion operating points may be determined.

Governing Characteristics

The fundamental equation governing torque converter design for a given speed ratio is:

$$T_i = CN_i^2 D_i^5 \qquad (1)$$

where:

T_i = input torque
N_i = input, or impeller speed
D_i = impeller diameter
C = Converter capability coefficient for the particular converter at a given speed rate.

The coefficient "C" characterizes the torque characteristics of the converter at a given speed ratio. The value of "C" depends upon the design of the impeller, the turbine, the stator, and the flow circuit. Jandasek[17] has indicated that the impeller exit angle is the major factor influencing converter characteristics, turbine and stator characteristics presumably are designed for optimum performance under coupling conditions.

Figure B-1 indicates how C varies with impeller exit angle and stall conditions.

The impeller exit angle can in turn be determined from the stall torque ratio in Figure B-2 as presented by Jandasek.

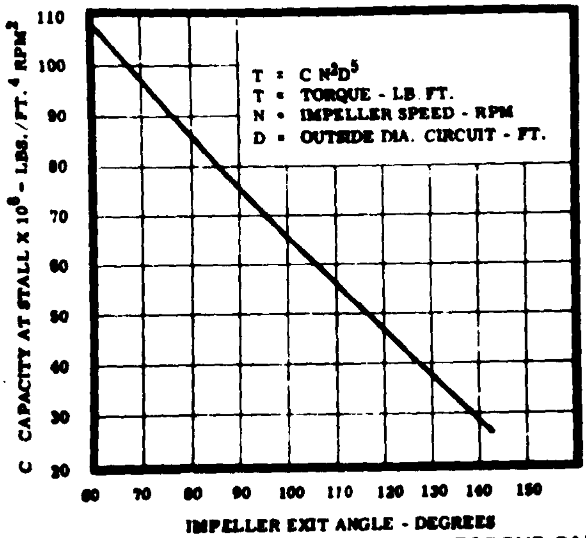


FIGURE B-1 VARIATION OF CONVERTER TORQUE CAPACITY AT STALL WITH IMPELLER EXIT ANGLE

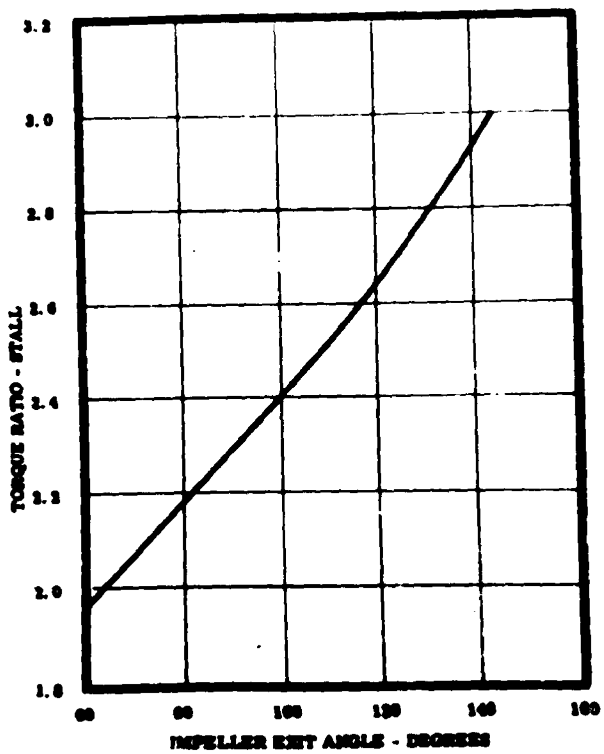


FIGURE B-2 RELATIONSHIP OF STALL-TORQUE RATIO AND IMPELLER EXIT ANGLE

The value of C is also dependent upon speed ratio. In order to determine the converter characteristics over the full speed ratio range, the change in C with speed ratio for a given impeller exit angle must be determined. Jandasek shows this relationship as that shown in Figure B-3.

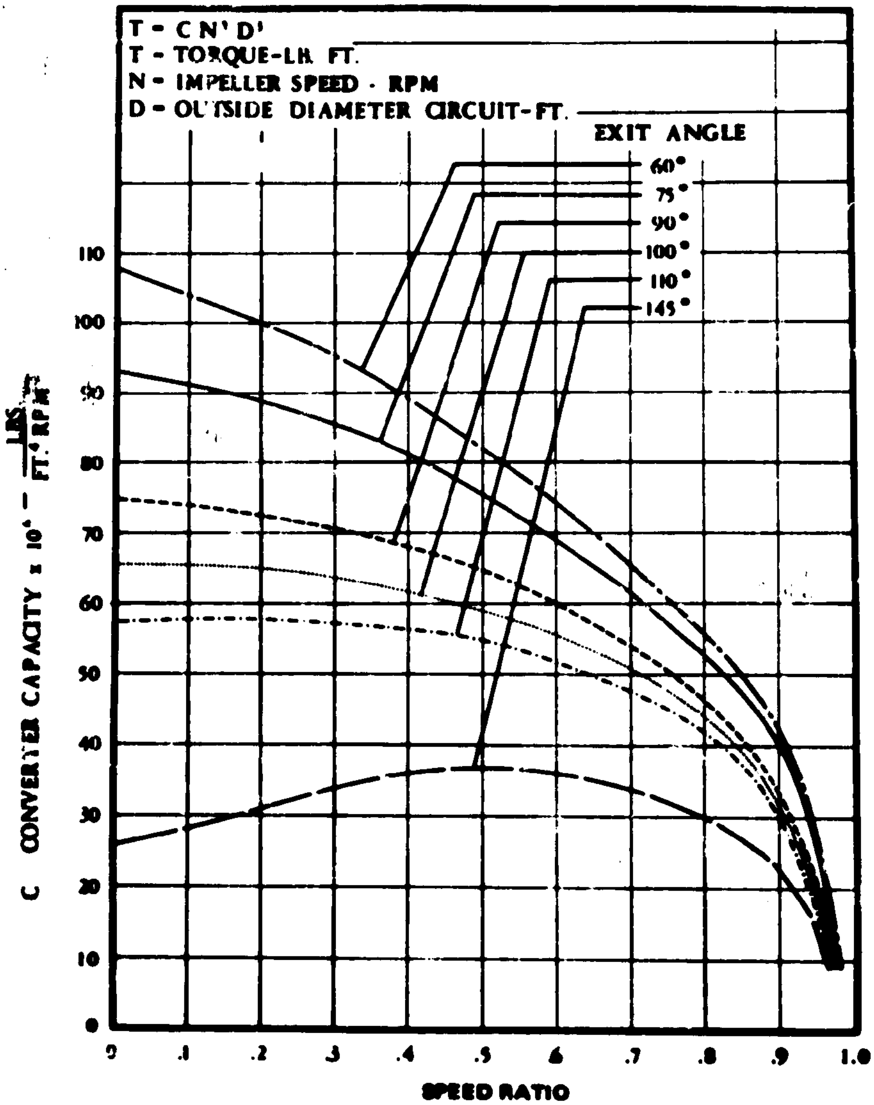


FIGURE B-3 VARIATION OF "C" WITH SPEED RATIO

The input torque associated with a given input speed and converter speed ratio can now be determined from equation 1. Alternatively, if the input torque and speed for stall is known, and the value of "C" determined from Figures B-1 and B-2, the size of the torque converter can be determined.

The output torque as a function of speed ratio and impeller exit angle can be determined from Figure B-4, as can the coupling point. We have simplified this as Figure B-5 and present that here as a tool to be used to characterize torque convertors when only the stall torque ratio is known. By drawing a straight line on Figure B-5 through point "M" from the known stall torque ratio, an approximate plot of torque ratio vs. speed ratio can be developed. (A line for a torque converter with a stall torque ratio of 2.02 is shown.) That point at which the torque ratio is 1 is the coupling point. (The author has tried this for convertors of several different sizes, stall torque ratios, and manufacturers and the results are quite accurate!) This tool is most useful between a stall torque ratio of 1.95 to 2.6. Outside these values, the designer will have to plot connect lines, following the basic pattern laid out by Jandasek in Figure B-4.

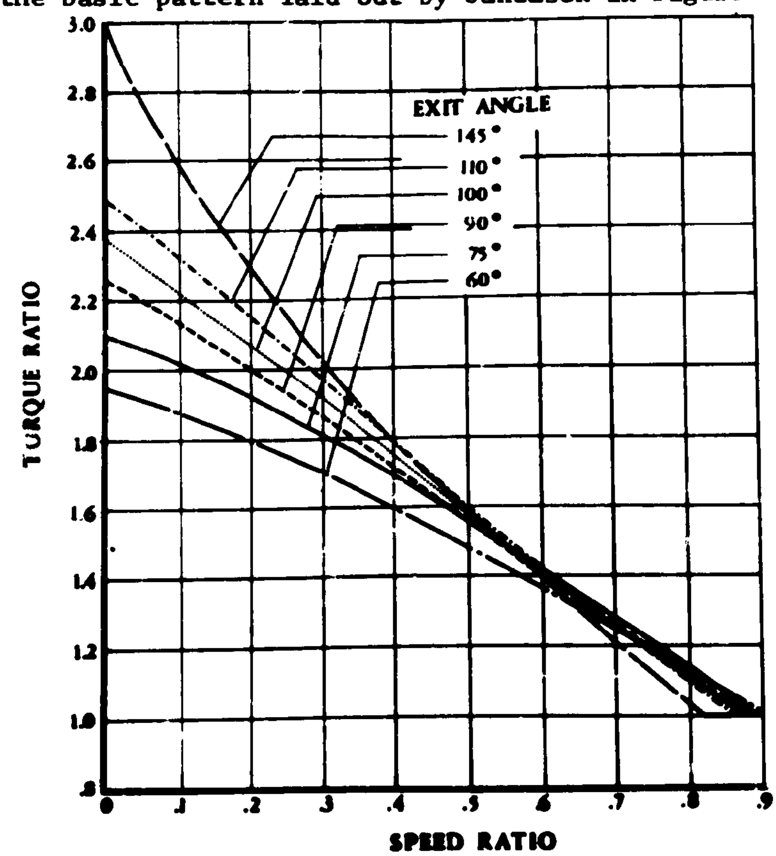
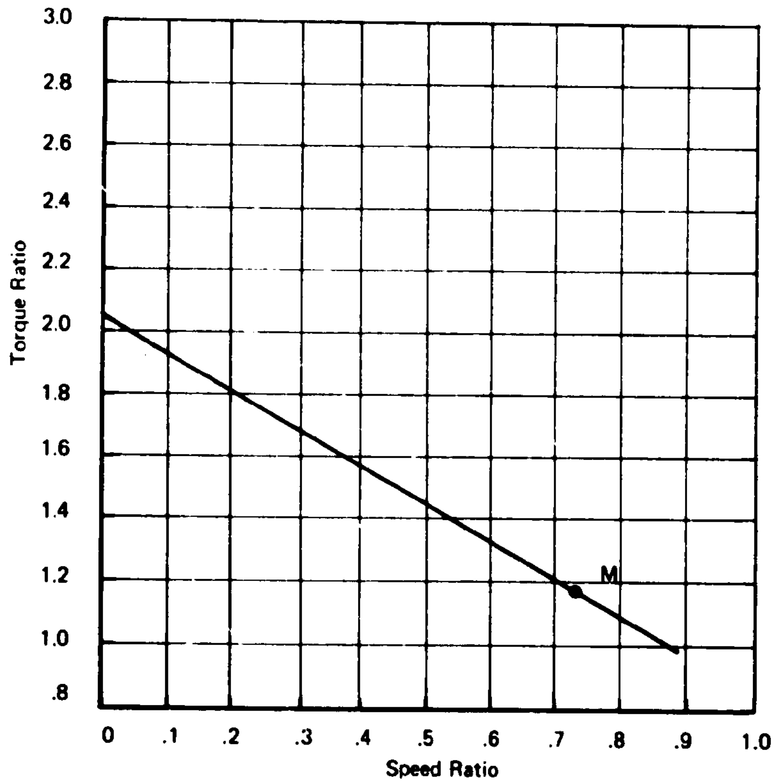


FIGURE B-4 RELATIONSHIP OF TORQUE RATIO TO SPEED RATIO FOR VARIOUS IMPELLER EXIT ANGLES



Source: Arthur D. Little, Inc.

FIGURE B-5 WORKING CHART FOR USE BY THE READER FOR DETERMINING SPEED RATIO AND COUPLING POINT CHARACTERISTICS OF A TORQUE CONVERTER FROM PREVIOUSLY DETERMINED STALL TORQUE RATIO (see text)

A useful tool for the determination of torque convertor/engine performance is the so called "capacity factor", "K"--not to be confused with the convertor capacity, "C". This capacity factor is unique function of the torque and speed ratios. For any given speed ratio, K describes the torque-speed relationship of the torque converter. As such, K may be thought of in terms of defining the torque handling capability--the torque capacity--of a torque converter at a given speed and speed ratio, hence, the name "capacity factor". "K" is determined by the following equation:

$$K = \frac{N}{\sqrt{T}} \tag{2}$$

where: N = either input or output speed

T = either input or output torque.

If input values of N and T are used, the calculation of K is the input capacity factor; if output values are used, K is the output capacity factor. K is constant for any unique combination of speed and torque ratios over the entire range of input and output speeds and torques. Previous sections have detailed the generation of torque converter characteristics based on input values, and we will hence limit this discussion to the input capacity factor.

From equations 1&2, it is obvious that K may also be expressed as:

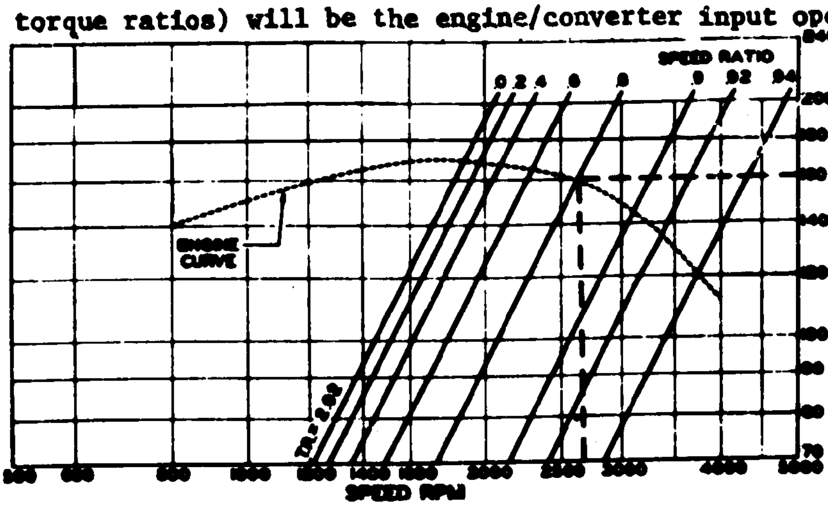
$$K = \sqrt{\frac{1}{CD^5}} \tag{3}$$

Thus, K may be determined from equation 3 for any speed ratio using Figure B-3 as long as the impeller exit angle is known.

Torque Converter/Engine Map

A map for converter input speed and torque can easily be generated once K is known. To do this, lines of constant "K" for given incremental speed ratios are plotted on a graph of input torque vs. input speed using equation 2. Such a plot is illustrated in Figure B-6. Each line represents a constant speed ratio and torque ratio. The torque ratios corresponding to each line are determined from Figure B-4 and/or B-5.

The engine speed and torque relationships for the desired throttle position are also plotted on this map as illustrated in Figure B-6. The points where the engine curve and the constant "K" curves (lines of constant speed and torque ratios) will be the engine/converter input operating points.



Source: Upton

FIGURE B-6 TYPICAL INPUT CAPACITY PLOT USED TO DETERMINE OPERATING CONDITIONS OF ENGINE AND TORQUE CONVERTER COMBINED

The torque converter output speed and torque can be determined from the known torque and speed ratios associated with each constant "K" line. The reader is urged to skip back to Figure B-5 for a moment, and study the straight line plotted through point M from a stall torque ratio of 2.02. This line defines the torque ratio as a function of speed ratio for the converter mapped in Figure B-6. From Figure B-5 the torque ratio associated with each input capacity speed ratio line on Figure B-6 may be determined. For instance, at an engine operating speed of 2700 rpm the speed ratio is .8 and the torque ratio is 1.1 from Figure B-5. The torque converter output is therefore:

Output Torque = 170 lbs ft x 1.1 x 187 lbs ft

Output Speed = 2700 rpm x .8 = 2160 rpm

Summary

This section has detailed the steps by which one can approximate torque converter characteristics knowing only the stall torque ratio and diameter. This methodology depends upon the assumption that all torque converters are designed in a similar manner and that the primary means of altering the torque/speed relationships are by changing the impeller exit angle. The author has tested this methodology against several current production torque converters of different diameters, stall torque ratios, and manufacturers and has found remarkable agreement. As a result, it may be assumed that this methodology is reasonably valid for current state of the art units. After all, this physical law governing fluid dynamics and converter design is universal, and the task of the torque converter is the same in all vehicles. It can be concluded, therefore, that the means of accomplishing the task will be quite similar from one design to the next. There is no substitute, however, for the actual data.

APPENDIX C

DETERMINING THE RATIOS OF A SIMPSON GEAR TRAIN

Figure C-1 details the ratios achieved with a Simpson gear set as found in the Chrysler A-904 transmission. Also shown are the formulas used to determine the given ratios. The following key will be helpful in using Figure C-1.

- A_1 & A_2 - Number of teeth on the Annular Gears
- S_1 & S_2 - Number of teeth on the Sun Gears
- C_1 & C_2 - Carriers for the planet gears
- P_1 & P_2 - Number of teeth on the planet gears
- C_1 and A_2 are directly connected to the output shaft
- B_1 & B_2 are bands for the front and rear gear sets, respectively.
- CL_1 and CL_2 are the front and rear clutches
- The upper figures in each row are the speeds of the unit indicated if it has the number of teeth specified in parenthesis.
- The formula given in the lower part of each row is the equation used to determine the speed of the unit.
- Unit speeds are presented as fractions of the input speed.

As an example, it is desired to determine the output shaft speed when the transmission is in second gear and the input shaft is operating at 1450 rpm.

The output shaft is connected to the second annular gear (A_2) and the first carrier (C_1). In second gear the proper equation from the Table is: $\frac{A}{A_1 + S_1}$. Also from the Table, $A_1 = 62$ (62 teeth are on the annular gears) and $S_1 = 28$. Units A_2 and C_1 (and the output shaft) will revolve $\frac{62}{62 + 28}$ $= .68889$ revolutions for every revolution of the input shaft. The rotational speed of the output shaft (units A_2 and C_1) will therefore be $1450 \times .6889 = 998.9$ rpm. The second gear ratio of this transmission is therefore $\frac{1}{.6889} = \frac{1450}{998.9}$ 1.45 to 1.

Gear	Activated Clutch or Band	S_1 & S_2 (28 T)	P_1 (17T)	C_1 & A_2	A_1 (62 T)	P_2 (17T)	C_2
1	CL_2, B_2	−0.9032	2.1594	0.4079	1.0	1.4876	0
		$\frac{A_1 A_2}{S_1(A_2+S_2)+S_2 A_1}$	$\frac{A_1 S_1 (A_2+S_2)}{P_1 [S_1 (A_2+S_2) + A_1 S_2}$	$\frac{A_1 S_2}{S_1 (A_2+S_2) + S_2 A_1}$	1.0	$\frac{A_1 A_2 S_2}{P_2[S_1(A_2+S_2) + A_1 S_2]}$	0
2	CL_1, B_1	0	1.1346	0.6889	1.0	0.7816	0.4745
		0	$\frac{S_1 A_1}{P_1 (A_1+S_1)}$	$\frac{A_1}{A_1+S_1}$	1.0	$\frac{A_1 A_2 S_2}{P_2(A_1+S_1)(A_2+S_2)}$	$\frac{A_1 A_2}{(A_1+S_1)(A_2+S_2)}$
N		1.0	−1.6471	0	−0.4516	−1.1346	0.3111
		1.0	$-\frac{S_1}{P_1}$	0	$-\frac{S_1}{A_1}$	$\frac{S_2 A_2}{P_2 (A_2+S_2)}$	$-\frac{S_2}{A_2+S_2}$
R	CL_1, B_2	1.0	−2.3909	−0.4516	−1.1072	−1.6471	0
		1.0	$-\frac{S_1(A_2+S_2)}{P_1 A_2}$	$-\frac{S_2}{A_2}$	$-\frac{S_1 (A_2+S_2) + S_2 A_1}{A_1 A_2}$	$-\frac{S_2}{P_2}$	0
Direct	CL_1, CL_2	1.0	1.0	1.0	1.0	1.0	1.0
		1.0	1.0	1.0	1.0	1.0	1.0

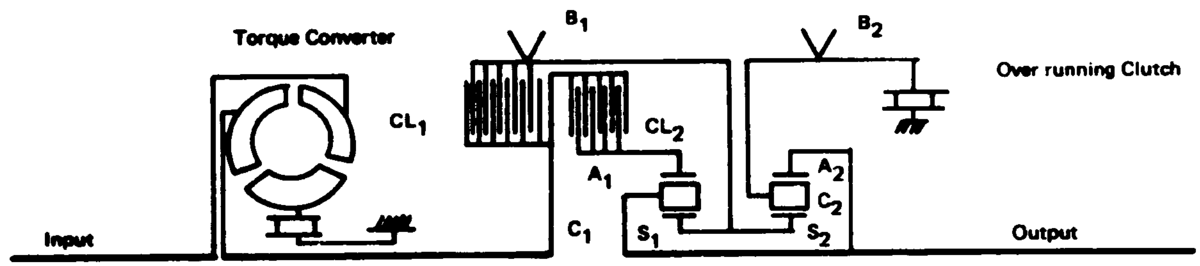


FIGURE C–1 GEAR RATIO DETERMINATIONS FOR SIMPSON GEAR SET AS FOUND IN CHRYSLER A–904 TRANSMISSION

APPENDIX D

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APPENDIX F

REPORT OF NEW TECHNOLOGY

After a diligent review of the work performed under this contract, no new innovation, discovery, improvement, or invention was made. However, the work of this contract resulted in the development and documentation of a simple method for estimating the characteristics of a particular torque converter when actual performance data is unknown. (Appendix B).