A Computer Program (VEHSIM) for Vehicle Fuel Economy and Performance Simulation (Automobiles and Light Trucks)

Volume IV: Enhancements

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October 1981
Final Report

This document is available to the public through the National Technical Information Service, Springfield, Virginia 22161.

U.S. Department of Transportation
National Highway Traffic Safety Administration

Office of Research and Development
Washington DC 20590

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
DOT-HS-806-040		

4. Title and Subtitle	5. Report Date
A COMPUTER PROGRAM (VEHSIM) FOR VEHICLE FUEL ECONOMY AND PERFORMANCE SIMULATION (AUTOMOBILES AND LIGHT TRUCKS) Volume IV: Enhancements	October 1981
	6. Performing Organization Code: DTS-323
7. Author(s): Russell W. Zub	8. Performing Organization Report No.: DOT-TSC-NHTSA-81-23.IV
9. Performing Organization Name and Address: U.S. Department of Transportation, Research and Special Programs Administration, Transportation Systems Center, Cambridge MA 02142	10. Work Unit No. (TRAIS): HS273/R2410
	11. Contract or Grant No.
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: Final Report, Jan 1980-Aug 1981
	14. Sponsoring Agency Code: NRD-13

15. Supplementary Notes

16. Abstract

This report presents an updated description of a vehicle simulation program, VEHSIM, which can determine the fuel economy and performance of a specified vehicle over a defined route as it executes a given driving schedule. Vehicle input accommodated by VEHSIM include accessories, engine, rear axle, converter transmission, tires, aerodynamic drag coefficient, and shift logic. The report is comprised of four volumes. Volume I presents a description of the numerical approach and equations, Volume II is a user's manual, Volume III contains the program listings, and Volume IV describes a simulation of the Integrated Overdrive Transmission with a split-torque converter.

17. Key Words	18. Distribution Statement
Motor Vehicle, Simulation, Fuel Economy, Performance, Engine Map	DOCUMENT IS AVAILABLE TO THE PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	50	

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

PREFACE

Volume IV is a supplement to a three volume document, "A Computer Program (VEHSIM) for Vehicle Fuel Economy and Performance Simulation (Automobiles and Light Trucks," developed at the Transportation Systems Center. This volume considers implementation of a simulated split torque transmission into VEHSIM.

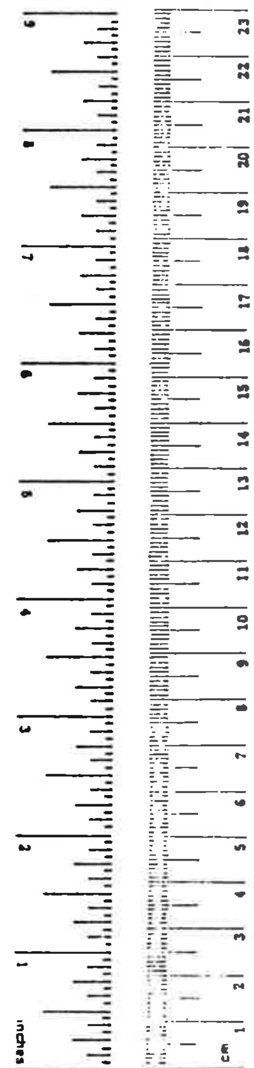
The Energy Technology Branch expresses thanks to Mr. Joseph Burshtein of SDC Integrated Services, Inc. for his invaluable help in writing the split torque algorithm; and to Mr. Jack Dolan of SDC Integrated Services, Inc., for assistance in implementing the source code into VEHSIM.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

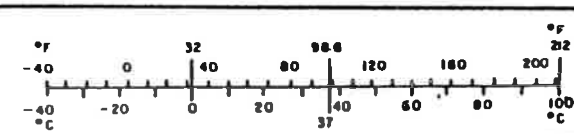
Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	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

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



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

This report considers the implementation of a simulated Split Torque Overdrive Transmission into the Transportation Systems Center computer program, VEHSIM, for automobile and light duty truck fuel economy and performance.

The presence of the split torque converter within the overdrive transmission does not allow the use of the existing "CONVTR" subroutine currently in VEHSIM. The purpose of this analysis is to modify the current model to accommodate a split torque overdrive transmission capability.

The split torque transmission covered herein is similar to Ford Motor Company's Automatic Overdrive (AOD), also referred to as the Ford Integral Overdrive (FIOD) transmission. The transmission operates with two paths of power flow: one path uses a hydrodynamic torque converter and the other path utilizes a mechanical drive. The two paths converge into a compound planetary gearset that combines the individual power flows. The transmission has four gears, with the power flow in first and second gear completely hydrodynamic. The fourth gear uses a mechanical path and the third gear uses a split torque path. The majority of the analysis is focused on the third gear power flow since this is unique to the VEHSIM program.

The approach of this report is first to examine the transmission as an object of modeling and then to look at the application of the model to the current version of VEHSIM. A kinematic analysis of the planetary gear mechanism is also presented. Finally, an algorithm for the simulation is given along with the actual software used.

The kinematic scheme of the split torque converter is presented in Figure 1. The necessity to simulate a split torque

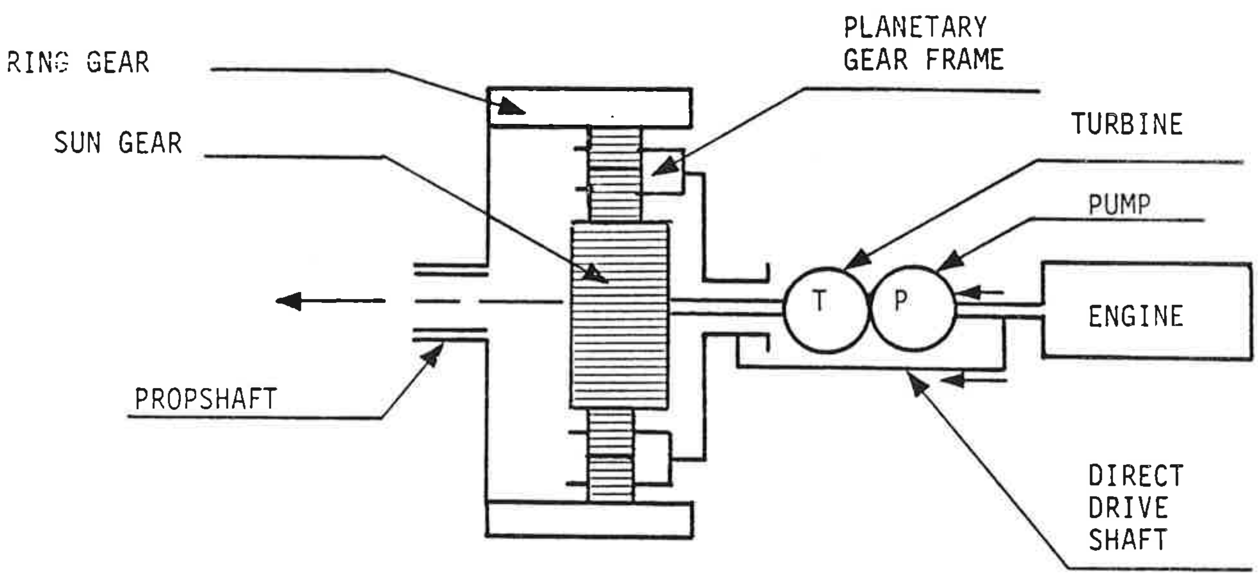


FIGURE 1. KINEMATIC SCHEME OF A SPLIT TORQUE MECHANISM

converter involves two groups of problems. The first group relates to the split torque converter proper and includes a solution of a nonlinear algebraic equation. The second group is associated with the adaptation of the split torque transmission to the VEHSIM program.

2. TRANSMISSION DESCRIPTION

The split torque transmission as an object of simulation has three modes of operation:

- o A standard combination of a gear box and a torque converter (at the first and second gear)
- o A parallel combination of a direct driveshaft and a torque converter linked to a propshaft via a planetary mechanism (at the third gear)
- o A direct mechanical overdrive (at the fourth gear).

A summary of transmission characteristics is presented in Table 1.

TABLE 1. TRANSMISSION GEAR CHARACTERISTICS

Gear	Gear load efficiency*	Gear ratio	Converter characteristics
I	a_1 = .98	2.4	Standard (as currently exists in VEHSIM)
II	a_2 = .98	1.467	Same as gear I
III	a_3 = 1.0**	1.00**	Special, to be derived
IV	a_4 = .98	0.667	No converter, direct drive

*This parameter is introduced into VEHSIM via "GEAR" data sheet.

**This coefficient is introduced artificially to use the existing data sheet.

3. PROBLEM FORMULATION

The software and data base format for the modeling of the split torque transmission involve the following problems:

- Speed equation of the split torque converter
- Torque equation of the split torque converter
- Torque loss calculation
- Drive, coast, and idle modes of operation
- Split torque converter state variables for different modes of operation
- Implementation into VEHSIM
- Overdrive transmission data base.

4. SPLIT TORQUE CONVERTER ANALYSIS

4.1 SPEED EQUATION

The kinematic scheme of a planetary mechanism of the split torque converter is displayed in Figure 2. In this scheme, radii R_1 and R_2 are proportional to the number of teeth of the sun and ring gear, respectively:

$$R_1 \equiv Z_1 \equiv 1$$

$$R_2 \equiv Z_2 \equiv 2.4$$

Designate:

RPM_1', RPM_1, RPM_2 = speed in RPM of the sun, planetary, and ring gear, respectively.

The planetary mechanism works as a summation device, and speeds of its gears satisfy an equation:

$$RPM_2 = k_1' \ RPM_1' + k_1 \ RPM_1 \qquad (4\text{-}1)$$

If $RPM_1' = RPM_1$, the planetary mechanism in Figure 2 is "locked," and from (4-1):

$$RPM_2 = RPM_1' = RPM_1$$

Then:

$$1 = k_1' + k_1 \qquad (4\text{-}2)$$

If $RPM_1 = 0$, from (4-1) obtain:

$$RPM_2 = k_1' \ RPM_1'$$

Gear ratio k_1' from the sun gear to the ring gear (planetary gear frame is fixed) is:

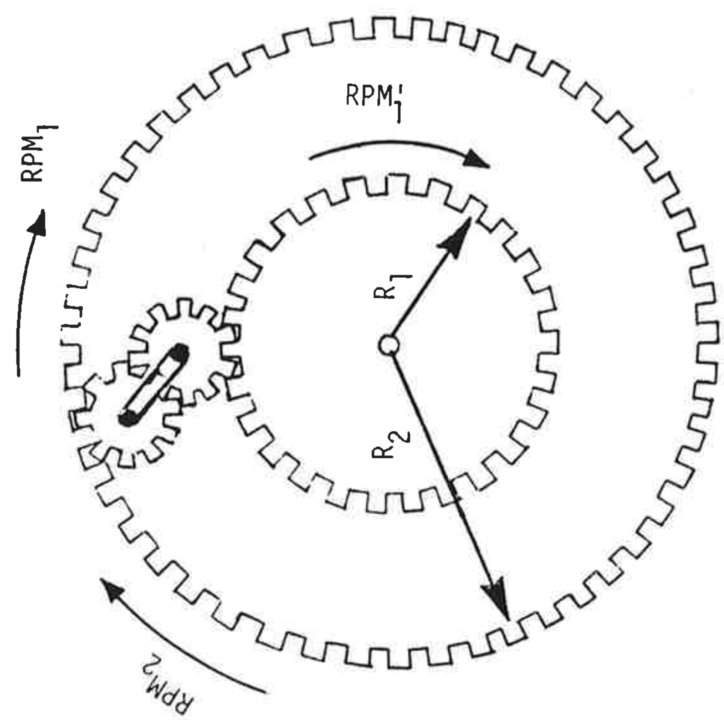


FIGURE 2. KINEMATIC SCHEME OF THE SPLIT TORQUE PLANETARY MECHANISM

$$k_1' = R_1/R_2 \qquad (4\text{-}3)$$

Substitute (4-3) into (4-2) and find k_1:

$$k_1 = (R_2 - R_1)/R_2 \qquad (4\text{-}4)$$

Substituting (4-3) and (4-4) into (4-1) and introducing parameters:

$$A_1 = R_1 = 1$$

$$A_2 = R_2 = 2.4 \qquad (4\text{-}5)$$

the desired speed equation of the split torque planetary mechanism is:

$$\boxed{RPM_2 = (A_1/A_2)RPM_1' + \big((A_2 - A_1)/A_2\big)RPM_1} \qquad (4\text{-}6)$$

where:

RPM_2 = speed at the output of the split torque converter

RPM_1' = speed of the converter's turbine

RPM_1 = speed of the direct driveshaft

A_1, A_2 = coefficients proportional to the number of teeth of the sun and ring gears, respectively.

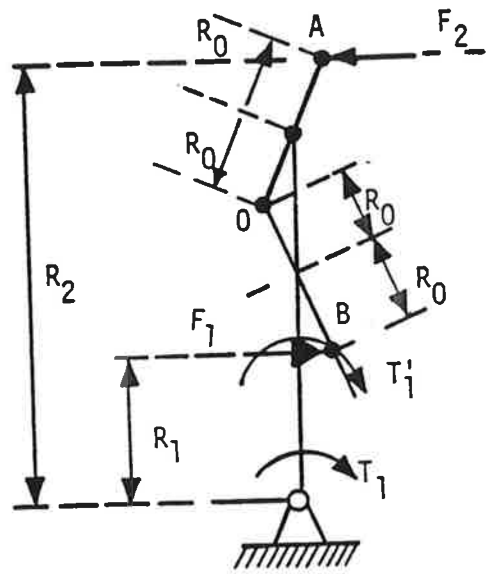
4.2 TORQUE EQUATION

The torque equation can be derived by finding the relation between T_1' and T_1:

T_1' = torque at the sun gear (converter output)

T_1 = torque at the planetary gear frame (direct drive shaft).

For this analysis, a force diagram (based on the kinematic scheme) is constructed in which levers are substituted for gears (Figure 3).



R_1, R_o, R_2 = radii of the sun, planetary, and ring gears, respectively

F_1, F_2 = force of contact between gears: sun/planetary and planetary/ring

T_1 = torque applied to the frame of the planetary gear

T_1' = torque applied to the sun gear.

FIGURE 3. FORCE DIAGRAM OF PLANETARY GEAR MECHANISM

The system of two planetary gears in Figure 2 is modeled in Figure 3 by a system of levers AO and OB. These levers are in a state of dynamic equilibrium, which means that contact forces (F_1, F_2) acting at points A and B are equal:

$$F_1 = F_2 = F \tag{4-7}$$

For the system in Figure 3 the torque is:

$$T_2 = T_1' + T_1$$

$$T_2 = F_2R_2$$

$$T_1' = F_1R_1 \tag{4-8}$$

where:

T_2 = summation torque applied to the ring gear

T_1' = torque applied to the sun gear (converter output)

T_1 = torque applied to the planetary gear frame (output of direct drive shaft).

From (4-8) have:

$$F_2R_2 = F_1R_1 + T_1$$

In correspondence with (4-7), the above equation is:

$$FR_2 = FR_1 + T_1$$

Hence,

$$T_1 = F(R_2 - R_1) \tag{4-9}$$

The desired torque relation follows from (4-8) and (4-9):

$$T_1' : T_1 = R_1 : (R_2 - R_1).$$

Substituting coefficients A_1 And A_2 in place of R_1 and R_2 to get the torque equation of the split torque converter:

$$\boxed{T_1' : T_1 = A_1 : (A_2 - A_1)} \tag{4-10}$$

Where:

T_1' = torque at the sun gear (converter output),

T_1 = torque at the planetary gear frame (direct drive-shaft)

A_1, A_2 = coefficient proportional to the number of teeth of the sun and ring gear, respectively.

4.3 PROBLEM OF TORQUE LOSSES CALCULATION

Consider a problem of torque losses calculation for the Split torque Transmission.

At the first and second gear, converter/gearbox interactions within the split torque transmission corresponds to the standard scheme modeled by VEHSIM. Accordingly, the scheme of torque loss calculation within VEHSIM is appropriately modeled at the first and second gear.

A unique situation occurs for the third gear, i.e., to the split torque configuration of the transmission. If the torque losses are generated within the planetary mechanism, then the existing VEHSIM scheme for the torque losses calculation is valid. In the opposite case (affirmed by the manufacturer's data), the balance of torque losses within the split torque version of the split torque transmission must be calculated along the specially designed scheme. This will cause changes within the "GOBACK" subroutine of VEHSIM. As for the fourth gear, the converter losses are not calculated.

4.4 MODES OF OPERATION

Drive - In this mode, a direct driveshaft and converter turbine (Figure 1) rotate in the same direction. A load torque is applied to a ring gear, and this causes a slippage in the converter.

Idle - The ring gear is stemmed by a large resistance torque at the wheels. The direct driveshaft rotates the planetary gear frame with the speed of the engine. Accordingly, the outer planetary gear rolls along the inner perimeter of the ring gear. As a result, the inner planetary gear generates the rotation of a sun gear and the converter turbine into the opposite direction.

Coast - This mode takes place when a negative torque is applied by the propshaft. Such a condition may occur when a car rides down a hill or when it moves with the throttle closed. The planetary gear frame is revolved by a direct driveshaft with the speed of the engine, and ring gear speed is higher than the speed of the planetary gear frame. Correspondingly, a sun gear and converter turbine (Figure 1) have a higher speed than the engine speed.

5. ALGORITHMS OF A SPLIT TORQUE CONVERTER

5.1 DRIVING MODE

Given: RPM_2, TORQ2 = speed and torque at the output of the split torque converter

find SR and TR (speed and torque ratios) of the split torque converter

where:

$SR = RPM_2/RPM_1$

$TR = TORQ_2/(TORQ_1 + TORQ_1'/TR)$

RPM_1 = speed of engine (converter input)

$TORQ_1$ = torque at direct drive shaft

$TORQ_1'$ = torque at converter output.

The following equations are used:

$$(A_1/A_2)\ RPM_1' + \big((A_2 - A_1)/A_2\big)\ RPM_1 = RPM_2$$

$$TORQ_1'/TORQ_1 = A_1/(A_2 - A_1)$$

$$TORQ_1' + TORQ_1 = TORQ_2$$

$$RAT = RPM_1' \Big/ \sqrt{TORQ_1'}$$

$$SR = f_{SR}(RAT)$$

$$RPM_1 = RPM_1'/SR$$

where:

RPM_1, RPM_1', RPM_2 = speed of engine, converter turbine, and propshaft, respectively.

A_1, A_2 = coefficients proportional to number of teeth of a sun and ring gear

$TORQ_1'$, $TORQ_1$ = torque at converter turbine and direct drive-shaft

RAT = capacity factor of torque converter

SR = speed ratio = RPM_1'/RPM_1

SR = $f_{SR}(RAT)$ = characteristic of a given converter.

The above system of equations is solved about variable RPM_1'. The solution is organized as a process of minimization of deviation δ:

$$TORQ'_1 = (A_1/A_2)\ TORQ_2$$

$$RPM_1' = RPM_2$$

(a) $RAT = RPM_1'/\sqrt{TORQ_1'}$

(b) $SR = f_{SR}(RAT)$

(c) $RPM_1 = RPM_1'/SR$

(d) $\delta_i = (A_2/A_1)\ RPM_1' + \big((A_2 - A_1)/A_2\big)\ RPM_1 - RPM_2.$

If δ is less then tolerant deviation D_o, the output data are calculated at steps (g) through (j) below. In the opposite case, variable RPM_1' is decremented:

(e) $RPM_1' = RPM_1' - \Delta RPM$

and calculations are repeated starting from step (a) above. The preceeding step of calculations is paralleled by the following interpolation operation.

The last two values of deviation δ (positive and negative) are saved together with corresponding values of variable RPM_1': (δ_R, RPMR) and (δ_L, RPML). At the time when the deviation δ

changes its sign, the interpolated value of variable RPM_1' is then calculated from equation:

(f) $$RPM_1' = RPML + \frac{RPM_R - RPM_L}{|\delta_L| + \delta_R} |\delta_L|$$

and calculation cycle is repeated starting from step (a) above. The goal of the interpolation procedure is to obtain the solution within the limited number of computing steps.

The following final steps render the desired outputs (SR, TR):

(g) $SR = RPM_2/RPM_1$

(h) $TR = f_{TR}(RAT)$ from converter characteristic

(i) $TORQ_1 = TORQ_2 \ ((A_2 - A_1)/A_2)$

(j) $$TR = \frac{TORQ_2}{TORQ_1 + TORQ_1'/TR} = A_2/((A_2 - A_1) + A_1/TR).$$

5.2 COAST MODE

An algorithm for the coast mode differs from the one presented above in step (e) and the final step which read:

(e) $RPM_1' = RPM_1' + \Delta RPM$

(g) $SR = RPM_2/RPM_1$

(h-j) $TR = 1.0$

5.3 IDLE MODE

In this mode of operation, the values of coefficients for the split torque converter are: SR = 0, TR = 1.

5.4 TEST OF ALGORITHMS FOR SPLIT TORQUE CONVERTER

Using numerical data from Table 2 -- which is a sample VEHSIM output of a drive and coast converter table -- and interpolation technique previously specified, the offered algorithms were tested and results are presented below.

(1) <u>Driving Mode</u>

Given: $RPM_2 = 1400$ RPM

$TORQ_2 = 400$ lb. ft.

Parameters of the split torque planetary mechanism:

$$A_1 = 1, \; A_2 = 2.4.$$

The results of the computing cycles for variable RPM_1' are given in Table 3.

(2) <u>Coast Mode</u>

Initial conditions are same. Results of computations of variable RPM_1' are in Table 4.

TABLE 2. DRIVE AND COAST CONVERTER DATA (CGC117)

DRIVE CONVERTER
11.75 inch drive torque converter
diameter = 11.8
pump inertia = 0.142 FT-LB-Sec**2
turbine inertia = 0.083 FT-LB-Sec**2
constant input torque = 250.00 LB-FT

Speed Ratio	0.000	0.100	0.200	0.300	0.400	0.500	0.600	0.700
Torque Ratio	2.008	1.916	1.816	1.712	1.628	1.488	1.364	1.240
Input Speed	1640.0	1670.0	1700.0	1745.0	1810.0	1890.0	1990.0	2120.0
K-Factor	0.000	7.630	15.957	25.28	35.887	48.996	64.659	84.286
Speed Ratio	0.800	0.887	0.923	0.950	0.964	0.975	0.983	0.984
Torque Ratio	1.120	1.000	1.000	1.000	1.000	1.000	1.000	1.000
Input Speed	2300.0	2515.0	2925.0	3475.0	4150.0	4720.	5290.0	6100.0
K-Factor	109.96	141.038	170.763	208.71	252.98	290.93	328.87	379.47

COAST CONVERTER
11.75 Inch Coast Torque Converter
diameter = 11.8
pump inertia = 0.192 FT-LB-Sec**2
turbine inertia = 0.083 FT-LB-Sec**2

Speed Ratio	1.560	1.166	1.103	1.038	1.022	1.019
Torque Ratio	1.000	1.000	1.000	1.000	1.000	1.000
Input Speed	848.0	1324.0	1766.0	2626.0	3502.0	3937.0

TABLE 3. DRIVE MODE

Step #	1	2	3	4	5
RPM_1'	1400	1300	1200	1100	1135.9
RAT	108.5	100.7	93	85.3	88
SR	.8	.76	.74	.7	.714
RPM_1	1750	1710	1621.6	1571.4	1589
δ	204	139	45	-25.2	0.2

The desired results:

$$SR = \frac{1400}{1589} = \underline{\underline{0.89}}$$

$$TORQ_1 = 400\left(\frac{2.4 - 1}{2.4}\right)$$

$$TR = f_{TR}(RAT = 88) = 1.18$$

$$TR_{(Total)} = \frac{400}{400\left(\frac{1.4}{2.4}\right) + 400\left(\frac{1}{2.4}\right)\left(\frac{1}{1.18}\right)} = 1.068$$

TABLE 4. COAST MODE

Step #	1	2	3	4
RPM_1'	1400	1500	1600	1548.5
SR	1.42	1.24	1.157	1.19
RPM_1	985.5	1209.7	1382.8	1301
δ	-242	-69.2	73.4	-1.5

The searched result: SR = 1400/1301 = 1.076

6. TRANSMISSION ALGORITHMS IMPLEMENTATION INTO VEHSIM

The presence of the split torque converter at the third gear of the Overdrive transmission necessitates a construction of a special procedure within the existing VEHSIM program. In order to simplify a software effort and evade an introduction of additional microelements into the existing VEHSIM program, the following scheme of reconstruction is offered.

The existing subroutine "CONVTR" is supplemented by "IF" statement and a separate block dedicated to the overdrive transmission subroutine (Figure 4).

The structure of the Overdrive transmission subroutine is displayed in Figure 5.

At the first and second gear, the Overdrive transmission works as a standard torque converter. Accordingly, the existing subroutine CONVTR is utilized without any changes. The gear ratios (Table 1) which take place at the first and second gear are reflected as parameters of a gear box using the existing data sheet. The structure of a procedure "Split Torque" (Figure 5) is illustrated in Figure 6. The crosshatched boxes in Figure 6 are segments of a copy of CONVTR subroutine. Of these, segments "Coast" and "Drive" are used to generate values of variables SR and TR. The programs within the two clear segments in Figure 6 reproduce algorithms for the split torque converter.

A detailed flow chart for the split torque segment of the overdrive transmission subroutine is presented in Figure 7, and a glossary of symbols used in the flow charts is given in Table 5.

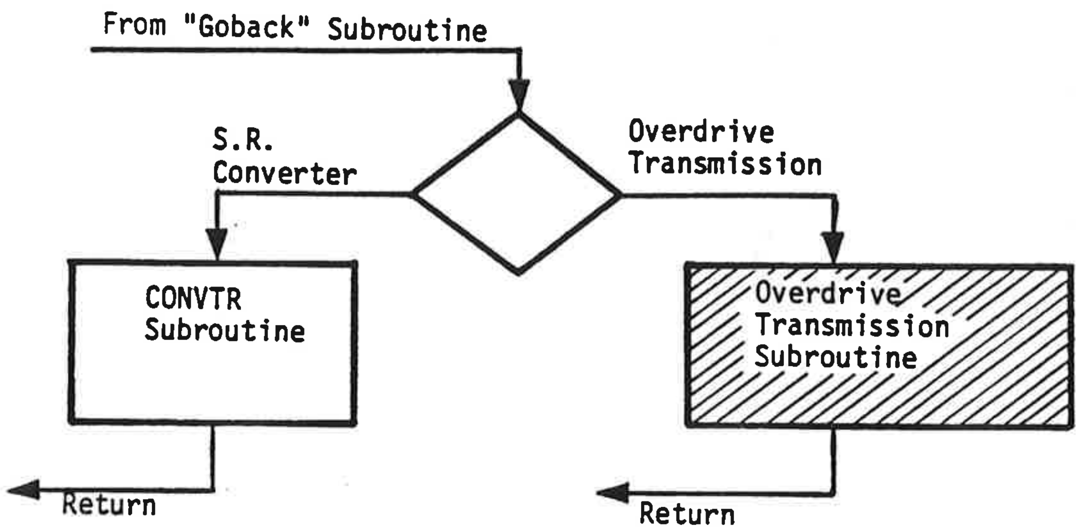


FIGURE 4. FLOW CHART OF THE OVERDRIVE TRANSMISSION SUBROUTINE IMPLEMENTATION INTO VEHSIM

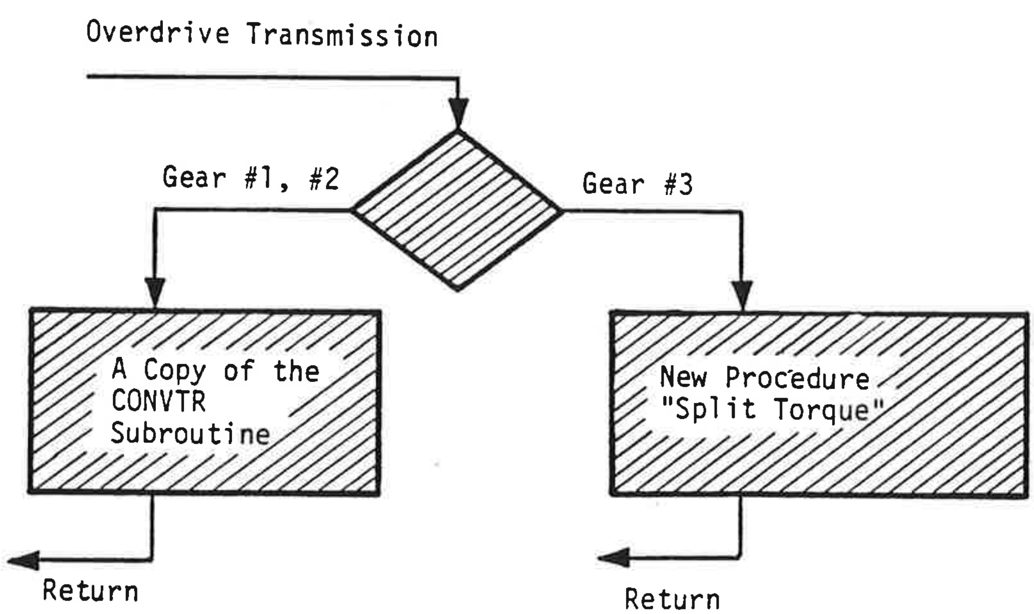


FIGURE 5. STRUCTURE OF NEW SUBROUTINE "OVERDRIVE TRANSMISSION"

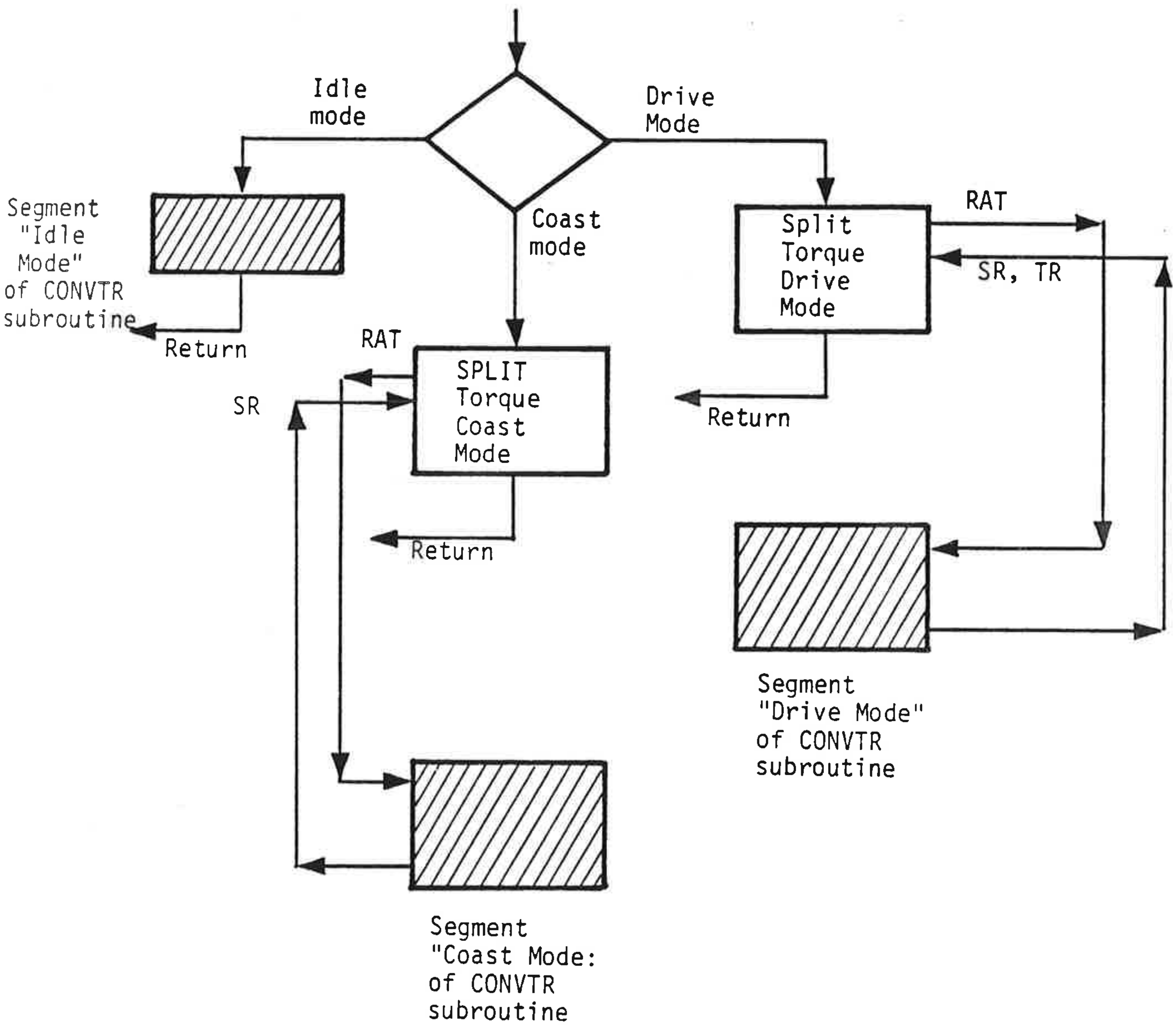


FIGURE 6. FLOW CHART FOR DRIVE, COAST AND IDLE MODES

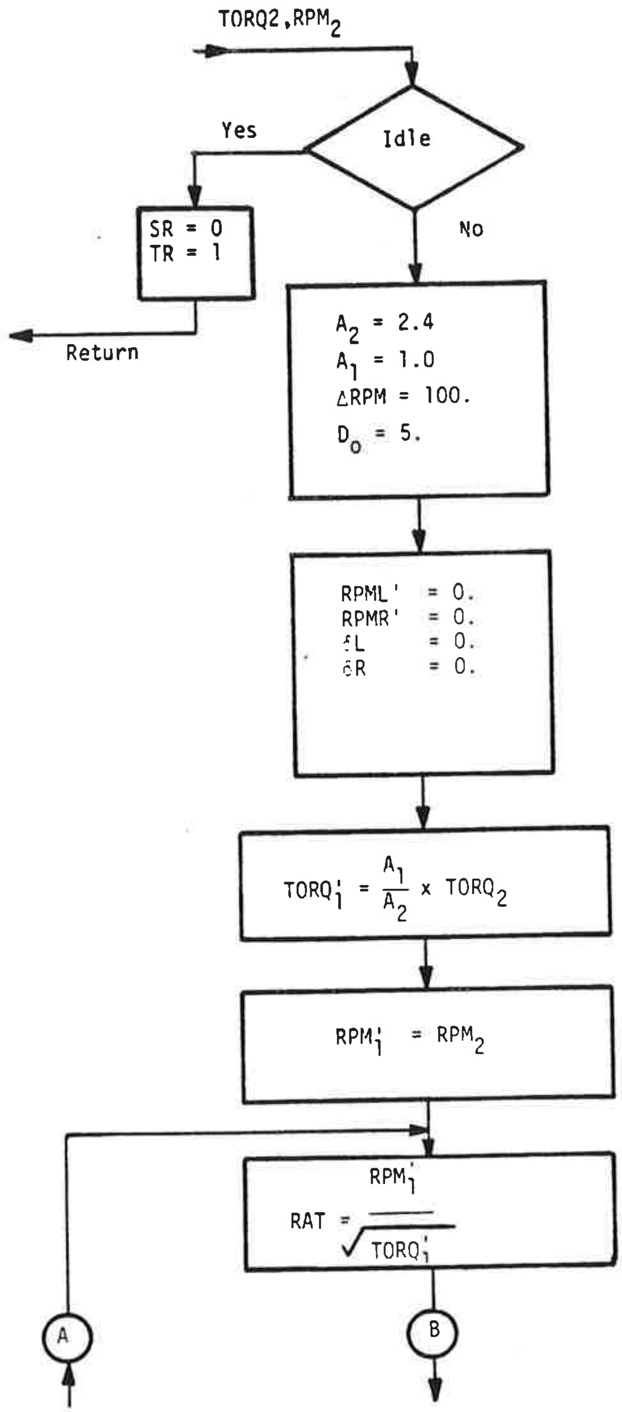


FIGURE 7. DETAILED FLOW CHART OF THE SPLIT TORQUE SEGMENT OF THE OVERDRIVE TRANSMISSION SUBROUTINE

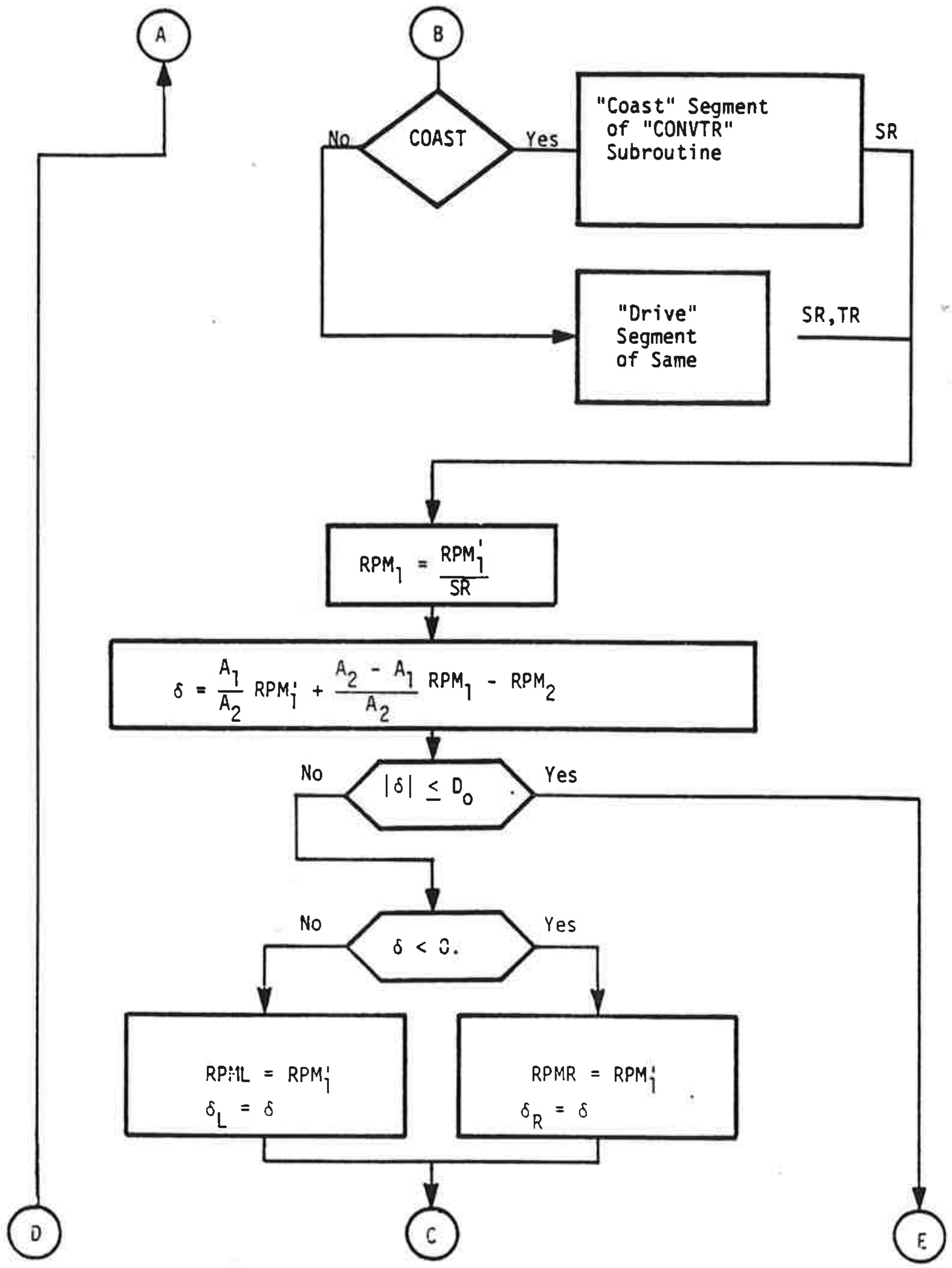


FIGURE 7. DETAILED FLOW CHART OF THE SPLIT TORQUE SEGMENT OF THE OVERDRIVE TRANSMISSION SUBROUTINE (Continued)

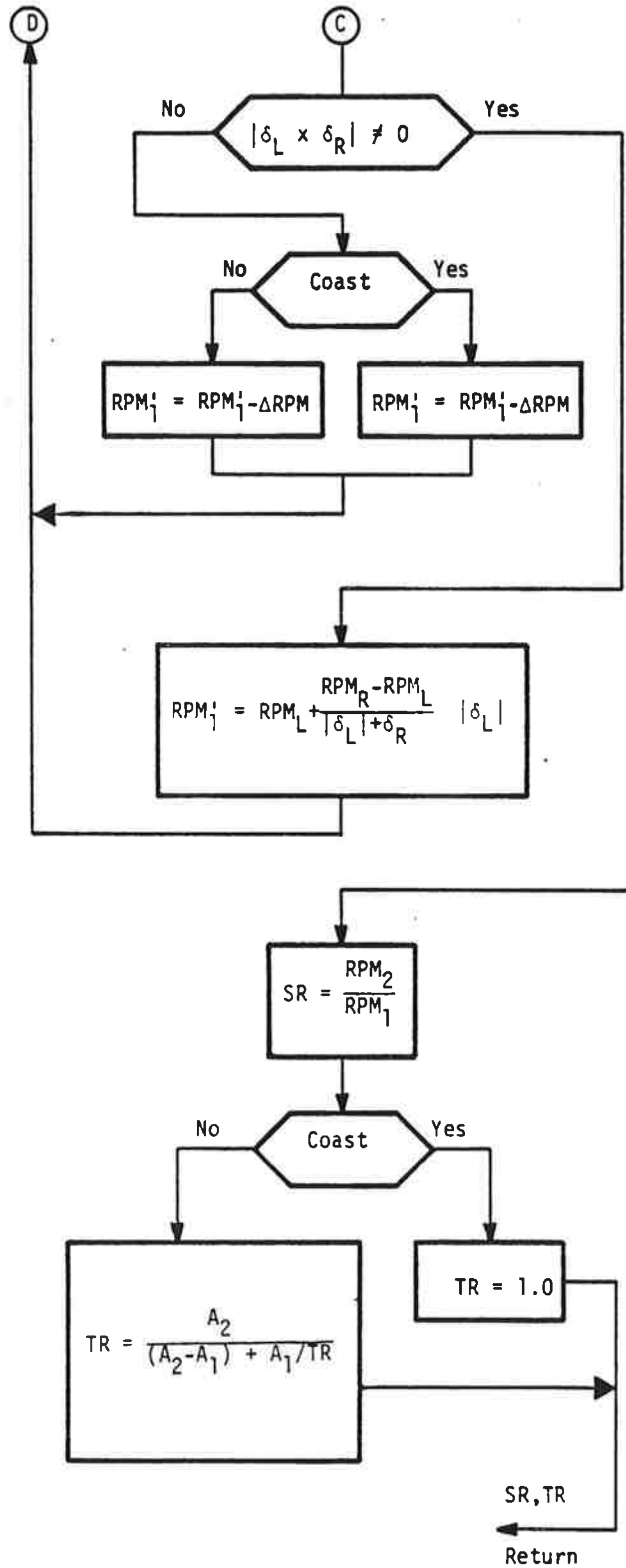


FIGURE 7. DETAILED FLOW CHART OF THE SPLIT TORQUE SEGMENT OF THE OVERDRIVE TRANSMISSION SUBROUTINE (Continued)

TABLE 5. GLOSSARY OF SYMOBLS USED IN FIGURE 7

A_1, A_2	=	coefficient proportional to number of teeth of a sun gear and ring gear of a planetary mechanism
Coast	=	Logic flag identifying the coast mode
D_o	=	Absolute value of tolerant deviation δ
Idle	=	same for the idle mode
RAT	=	capacity factor of the torque converter
RPM_1	=	engine speed
RPM_1'	=	converter output speed
RPM_2	=	split-torque converter output
RPM_L	=	the last calculated and saved value of variable RPM_1' which corresponded to the negative value of deviation δ
RPM_R	=	same for the positive value of δ
SR	=	speed ratio of the converter
$TORQ_1$	=	torque at the direct driveshaft of the split torque converter
$TORQ_1'$	=	same as the converter output
$TORQ_2$	=	same as the split torque converter output
TR	=	torque ratio of the converter
δ	=	deviation of the calculated value of RPM_1' from the given value of RPM_1'
δL, δR	=	the last calculated negative and positive values of deviation δ
ΔRPM	=	increment/decrement of searched variable RPM_1'

7. DATA SHEET FOR TRANSMISSION SIMULATION

The data base relating to the Automatic Overdrive Transmission is to be read into the computer using the existing set of data sheets:

#5. GEAR (torque loss characteristic)

#6. SPEED RATIO CONVERTER (torque loss characteristic)

#8. TRANSMISSION

#12. SHIFT LOGIC.

The coefficients A_1, A_2 (proportional to teeth number of the sun and ring gear of the planetary mechanism) must be placed as an additional information of the data sheet. For this purpose, the data sheet "S. R. Converter" could be used. The projected positions for coefficients A_1, A_2 are designated at the existing "S. R. Converter" data sheet turned into "Automatic Overdrive Transmission" (Figure 8).

In addition to that, for this data sheet a special code word must be introduced. This word (OVDRV) must be used within the CONVTR subroutine to direct the flow of information to the Overdrive Transmission segment of the named subroutine (see Vol. II).

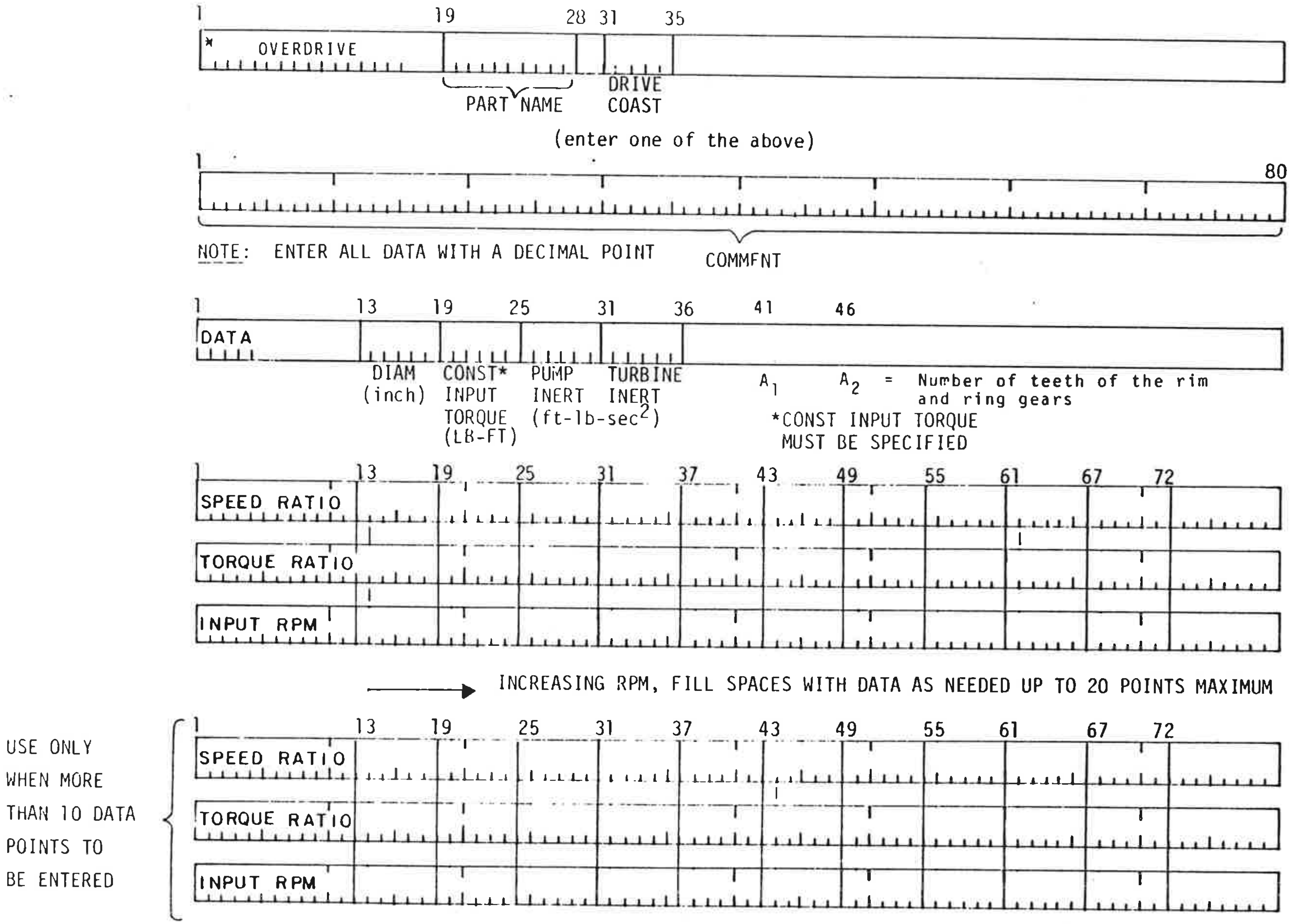


FIGURE 8. DATA SHEET: AUTOMATIC OVERDRIVE TRANSMISSION

APPENDIX A

SHIFT LOGIC LINES DEFINED IN DEGREES OF THROTTLE

The Integrated Overdrive Transmission uses throttle degrees as a control signal for its shift logic. The value of this parameter is calculated within the existing version of subroutine ENGINE. Accordingly, the shift lines determined within the Data Sheet - SHIFT LOGIC must be defined in degrees of throttle rotation. The capability to read the named segment of the SHIFT LOGIC Data Sheet expressed in throttle degrees is accomplished by the changes within the respective subroutines of VEHSIM program. These changes are summarized in Table A-1.

TABLE A-1. IMPLEMENTATION OF CAPABILITY TO READ SHIFT LOGIC LINES IN THROTTLE DEGREES

List of Changes Within VEHSIM				
#	Subroutine/ Section	Line	New Edition	Comment
1	Data sheet 12 shift logic	The 5th from the top	Vacuum unit-vacuum THROTTLE Throttle (%WOT) DTHRO Throttle (degrees)	The degree range of shift logic lines must correspond to the throttle degree range of the engine map.
2	Data sheet 12 shift logic	The 7th line from the top columns 19-26	Same	DTHRO - a new word

3	Subroutine SHIFTS	In place of two lines from #7 down	X = [Torqe-TMIN/(TWOT-TMIN)] *100
			IF (LVAC) X = Vac
			IF (LTHR) H = THR
				LTMR - new logic case
				TMR = throttle positions in degrees from engine map

4	SHIFTS	Overdrive shift criteria	IF (LVAC.OR.LTHR) PCTMR = [(TORQE-TMIN)/(TWOT-TMIN)] *100	
5	INPBAT	Shift logic The 20th line from #2780	IF(Word.NE.HVACUU, and Word.NE.HTHROT) N002, N001 N001 IF (Word.NE.HDTHRo) go to 300 N002 (configuration of the program)	N - appropriate number for the given statement

6	INPBAT	From the 22nd line down after #2780	LVAC =.FALSE.
			LTHR =.FALSE
			IF (Word.EQ.HVACUU) LVAC = True
			IF (Word.EQ.HDTHRo) LTHR =.True
				In place of the 22nd and 23rd lines

TABLE A-1. IMPLEMENTATION OF CAPABILITY TO READ SHIFT LOGIC LINES IN THROTTLE DEGREES (CONTINUED)

#	Subroutine/ Section	Line	New Edition	Comment
7	INPBAT	From the 7th line down	IF (Word 1.NE.HVACUU, AND. Word.NE.HTHROT) N'002, N'001 N'001 (Word 1.NE.HbTHRo) go to 300 N'002 (configuration of the program)	N' - appropriate number for the given statement
8	INPBAT	"DATA"	DATA HDTHRO/SHDTMRO/	Add to "Data"
9	VEHSIM COMMON	Common (Eng. map)	LTHR	Add to "Common" (Eng. map)
10	VEHSIM Common	Logical	LTHR	Add to "Logical"
11.	DSKRD	Load shift logic data	LTHR	Add to "Load Shift Logic"
12	DSKWR	Store Shift Logic Data	LTHR	Add to store shift logic page

APPENDIX B - SOURCE CODE LINES

```
      SUBROUTINE OVRDRV
C
C     ENTRY POINTS: OVRDRV
C
C     CALLED BY: GOBACK
C
C**********************************************************************
C
C     USED FOR THE SPLIT TORQUE CONVERTER IN RESPONSE TO THE
C     OVERDRIVE TRANSMISSION
C
C     DICTIONARY OF VARIABLES USED IN THIS ROUTINE
C
C     RPM2 - INPUT VARIABLE INTO SPLIT TORQUE SEGMENT
C     TORQ2 - INPUT VARIABLE INTO SPLIT TORQUE SEGMENT
C     ATSIN-CONTRACTION COEFFICIENT OF INCREMENT FOR
C            CONVERTER SPEED
C     A1 - COEFFICIENT PROPORTIONAL TO THE NUMBER OF
C             TEETH ON RING GEAR OF SPLIT TORQUE PLANETARY
C             MECHANISM
C     A2 - COEFFICIENT PROPORTIONAL TO THE NUMBER OF
C             TEETH ON RING GEAR OF SPLIT TORQUE PLANETARY
C             MECHANISM
C     DVV - TOLERANCE OF DEVIATION OF CALCULATED RPM2
C            FROM INPUT VARIABLE
C     DLL - THEORETICAL INFINITY, TAKEN FOR COMPARISON
C     DL - CURRENT DEVIATION OF CALCULATED RPM2 FROM THE
C           INPUT VARIABLE
C
      INCLUDE 'COMMS/NOLIST'
C
C
C     INITIALIZE VARIABLES
      A1=1.0
      A2=2.4
      DVV=5.0
      DLLRPM=100.0
      RPML=0.0
      RPMR=0.0
      DLL=0.0
      DLR=0.0
      IF(RPM2.GT.1.0) GO TO 9
C
C     IF IDLE SET TO LOWEST SPEED RATIO
C     IDLE MODE
      SR=SRD(1)
      TR=TRD(1)
      GO TO 90
9     CONTINUE
      TR=1.0
      TORQ1=(A1/A2)*TORQ2
      RPM11=RPM2
110   RAT=RPM11/SQRT(TORQ11)
      IF(COAST) GO TO 100
C
C     FOR DRIVE MODE
C
      IF(TORQ2.GE.0.000001) GO TO 8
      SR=1.0
      RETURN
8     CONTINUE
      RAT=RPM2/SQRT(TORQ2)
      IF(RAT.LT.TORRPK) GO TO 20
      IF(RAT.LT.ARD(NTD)) GO TO 10
      SR=SRD(NTD)
```

```
      GO TO 140
   10 J=NTD-1
   11 IF(RAT.GT.AKD(J)) GO TO 12
      J=J-1
      GO TO 11
   12 JP=J+1
      SR=(SRD(J)-SRD(JP))/(AKD(J)-AKD(JP))*(RAT-AKD(JP))+SRD(JP)
      IF(SR.GT.1.0) SR=1.0
      GO TO 140
   20 IF(RAT.LT.AKD(1)) GO TO 30
      J=NTDP-1
   22 IF(J.LT.1) GO TO 30
      IF(RAT.GT.AKD(J)) GO TO 25
      J=J-1
      GO TO 22
   25 JP=J+1
      SR=(SRD(J)-SRD(JP))/(AKD(J)-AKD(JP))*(RAT-AKD(JP))+SRD(JP)
   26 TR=(TRD(J)-TRD(JP))/(AKD(J)-AKD(JP))*(RAT-AKD(JP))+TRD(JP)
      IF(SR.GT.1.0) SR=1.0
   27 IF(SR.GE.0.0) GO TO 140
      SR=0.0
      RAT=-(AKD(J)-AKD(JP))/(SRD(J)-SRD(JP))*SRD(JP)+AKD(JP)
      GO TO 26
   30 J=1
      GO TO 25
C
C     FOR COAST MODE
C
  100 CONTINUE
      IF(RPM2.LT.SRC(1)*AKC(1)) GO TO 55
      IF(RPM2.GT.SRC(NTC)*AKC(NTC)) GO TO 60
      GO TO 65
C
C     IF BELOW LOWEST SR GIVEN AS INPUT
C
   55 J=1
      JP=2
      GO TO 75
C
C     IF ABOVE HIGHEST SR GIVEN AS INPUT
C
   60 JP=NTC
      J=JP-1
      GO TO 75
C
C     FIND CORRECT SEGMENT FOR CURRENT POINT
C
   65 DO 70 J=2,NTC
      IF(RPM2.GE.SRC(J-1)*AKC(J-1).AND.RPM2.LE.SRC(J)*AKC(J))
     1 GO TO 73
   70 CONTINUE
   73 JP=J
      J=JP-1
C
C     COMPUTE SPEED RATIO BY INTERPOLATION
C
   75 SR=(SRC(J)-SRC(JP))/(AKC(J)*SRC(J)-AKC(JP)*SRC(JP)) *
     1 (RPM2-AKC(J)*SRC(J))+SRC(J)
      IF(SR.LT.1.0) SR=1.0
      GO TO 140
C
```

```
C        IF SR GREATER THAN MAX INPUT , SET TO MAX
C
90       CONTINUE
         IF(SR.GT.SRD(NTD)) SR=SRD(NTD)
         RETURN
140      CONTINUE
         RPM1=RPM11/SR
150      DL=((A1/A2)*RPM11+((A2-A1)/A2)*RPM1)-RPM2
         IPASS=IPASS+1
         IF(IPASS.GE.100) STOP
C        WITHIN TOLERANCE? IF YES,END OF ITERATIVE PROCESS
160      IF(ABS(DL).LE.DVV) GO TO 310
C        CONTINUE ITERATION ALONG SAME DIRECTION
170      IF(DL.LT.0.0) GO TO 210
180      RPMR=RPM11
190      DLR=DL
         GO TO 230
210      RPML=RPM11
220      DLL=DL
230      IF(ABS(DLR*DLL).NE.0.0) GO TO 290
         IF(COAST) GO TO 270
C        DRIVE MODE
         RPM11=RPM11-DLLRPM
         GO TO 110
270      RPM11=RPM11+DLLRPM
         GO TO 110
290      RPM11=RPML+(((RPMR-RPML)/(ABS(DL)+DLR))*ABS(DL))
         GO TO 110
310      SR=RPM2/RPM1
         IF (COAST) GO TO 360
         TORQ1=TORQ2*((A2-A1)/A1)
         TR=TORQ2/(TORQ1+TORQ11/TR)
         RETURN
360      CONTINUE
C        COAST MODE
         TR=1.0
         RETURN
         END
```

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