A computer based mathematical method for predicting the directional response of trucks and tractor-trailers.

UPDATE TO THE PHASE II

TECHNICAL REPORT

UPDATE TO THE PHASE II TECHNICAL REPORT

The Phase II technical report was distributed in June, 1973. Since that time, various additions have been made to the directional response program. These are listed below.

1) The capability to simulate various antiskid mechanisms.

2) Drive torque

3) Independent front suspension option

4) Auxiliary roll stiffness

5) Split μ capability.

The first two of these additions have been reported in the Quarterly Progress Report dated June 30, 1973. The independent front suspension, auxiliary roll stiffness, and split μ capability are more recent additions. An explanation of each of these additions is included in this report.

It should be noted that the updated program cannot be run with a data stream in the form given in the Phase II report. The user must enter an auxiliary roll stiffness for all non-tandem axles at the appropriate place in the data stream as explained in this document.

It is envisioned that this document, together with the Phase II technical report, will provide a convenient reference for users of the Phase II simulation. Programming and technical comments on this subject should be directed to Mike Bodine or Jim Bernard.

A GENERAL PURPOSE MATHEMATICAL MODEL FOR SIMULATING ANTI-LOCK SYSTEMS

by

Charles MacAdam

The purpose of these notes is to document and explain an antilock simulation intended for use with the Phase II Directional Response Truck/Tractor-Trailer simulation previously developed at HSRI under contract to the Motor Vehicle Manufacturers Association.* The antilock simulation described herein attempts to offer a general framework in which the characteristics of different existing antilock systems can be modeled. The simulation concentrates on three areas common to most antilock systems: (1) wheel sensor, (2) control logic, and (3) pressure modulator. Axle-by-axle systems are allowed for in the Phase II program as well as four different side-to-side options for any axle. A description of the simulation and explanation of its use in each of these areas will follow. An example is provided in Appendix I along with the required input format for using the program.

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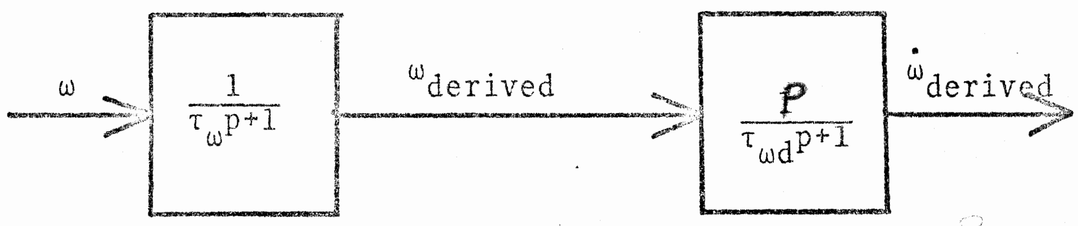
*Murphy, R.E., et al., A Computer Based Mathematical Method for Predicting the Braking Performance of Trucks and Tractor-Trailers, Phase I Report, Motor Truck Braking and Handling Performance Study, Highway Safety Research Institute, University of Michigan, September 15, 1972.

Bernard, J.E., et al., A Computer Based Mathematical Method for Predicting the Directional Response of Trucks and Tractor-Trailers, Phase II Report, Motor Truck Braking and Handling Performance Study, Highway Safety Research Institute, University of Michigan, June 1, 1973.

1. WHEEL SENSOR MODULE

At this time no attempt was made to model all the intracacies comprising a so-called typical wheel sensor. Rather, a black box or input-output modeling approach was taken. The primary effect of a wheel sensor is a phase shift and/or time delay between the actual wheel rate and the derived wheel rate. This input-output relationship can often be described adequately by transfer functions of various order and/or transport time delay expressions. The present version assumes a general first order filter of the form $\frac{1}{\tau_\omega p + 1}$ relating actual wheel rate and derived wheel rate, where τ_ω is the time constant of the filter and p is an operator denoting differentiation with respect to time.

Many antilock systems make use of wheel acceleration derived from the output of the wheel sensor. This normally involves additional delays along with a differentiation process. The assumed transfer function here was taken as $\frac{p}{\tau_{\omega d} p + 1}$ relating derived wheel rate to derived wheel acceleration. The derived wheel acceleration calculation normally takes place within the electronic control unit. However, since it, along with wheel rate, is a primary input to the control unit logic, it is included here within the wheel sensor module so that the control unit can be characterized by logical or decision-making processes only. The wheel sensor module can then be described by the following input-output relationships:



where $\omega_{derived}$ and $\dot{\omega}_{derived}$ are used as the primary inputs to the control logic module. (Other variables are provided as possible inputs to the control logic module, however, no similar operations are attempted on these other input variables.) The assumed wheel sensor and derivative circuit input-output relationships are therefore described by two input parameters τ_{ω} and $\tau_{\omega d}$ which represent the first order filter time constants of the wheel sensor and its derivative circuit.

2. CONTROL LOGIC MODULE

This portion of an antilock system, more than any other, is most responsible for distinguishing and defining one antilock system from another. It is likewise the one portion of an antilock system that varies the most between different systems and about which so little information is available. Therefore, a rather general framework was constructed in which many different control-logic schemes could be programmed by merely altering input parameters.

2.1. INEQUALITY EXPRESSIONS

The principle feature of this general framework is a set of six arithmetic inequalities of the form:

$$F_i \equiv A_i \dot{\omega} + B_i \omega + C_i + D_i \dot{x} + E_i \omega_1 \geq 0 \quad , \; i=1,6$$

where

$\dot{\omega}$ ≡ derived wheel linear acceleration (wheel sensor module) at the tire/road interface

ω ≡ derived wheel linear velocity at the tire/road interface (wheel sensor module) (Note ω is the product of the effective rolling radius and the angular velocity of the wheel.)

$\dot{x}$ ≡ vehicle velocity

ω_1 ≡ spin ~~up~~ wheel rate at time of pressure increase during an antilock cycle.

A_i, B_i, C_i, D_i, and E_i are input parameters specified by the user.

The first three inequalities are associated with the generation of an "OFF" signal which is sent to the pressure modulator—the last three with the corresponding "ON" signal.

A simple description of the sequence of operations taking place within the antilock control logic module is as follows:

During a braking maneuver, the derived wheel rate and acceleration are sent along with vehicle velocity to the control logic module for evaluation in the user-selected inequality expressions. These expressions are evaluated, and based upon their polarity, OFF signals or ON signals are sent to the pressure modulator. At the beginning of the braking maneuver, evaluation of the inequalities associated with generating the "OFF" signal takes place until an "OFF" signal is generated. Attention then is focused on the inequalities associated with generating an "ON" signal until an "ON" signal is generated. This sequence continues until either the treadle valve pressure demanded by the driver falls to near zero, or until the vehicle velocity decreases to below some cut-off velocity.

2.2. LOGICAL VARIABLES

Each of the six inequalities has assigned to it a logical variable that is defined as TRUE if the inequality is satisfied as shown, FALSE if not. In other words, if $F_i \geq 0$, then the logical variable L_i associated with F_i assumes the value TRUE. If $F_i < 0$, then L_i assumes the value FALSE. Since there are three inequalities for the generation of the "OFF" signal, there are three logical variables associated with the "OFF" signal. The purpose of these logical variables is to facilitate the generation of an "OFF" signal by allowing them to be "AND"-ed and "OR"-ed together by the program user. If, for example, the user had decided that F_1 and F_2 must be satisfied or else F_3 satisfied (F_1, F_2, and F_3 having been previously defined by the data set selected by the user) then the proper "OFF" signal would be defined by the following expression:

$$OFF = (L_1 \text{ AND } L_2) \text{ OR } L_3$$

The same discussion applies to the "ON" signal and logical variables L_4, L_5, and L_6.

The user specifies the relation between L_1 and L_2 and the relation between the bracketed expression and L_3 by means of two logical operator switches OP_{12} and OP_{23} that are entered as 0 or 1 input. The value 0 implies an "OR" operation, the value 1 an "AND" operation. Therefore, in the above example, OP_{12} would be specified by the user as 1 and OP_{23} as 0. The same discussion applies to logical variables L_4, L_5, L_6, and their logical operator switches OP_{45} and OP_{56}. Therefore, the general forms of these logical equations are:

$$OFF = (L_1 \; OP_{12} \; L_2) \; OP_{23} \; L_3$$

and,

$$ON = (L_4 \; OP_{45} \; L_5) \; OP_{56} \; L_6 \; .$$

If the user wishes, he may generate an "OFF" signal with only one or two of the three available inequalities, hence ignoring the remainder. This is allowed by means of three logical selection switches, LC_1, LC_2, LC_3, similar to the logical operation switches. They are specified as either 0 or 1 input by the user, 0 meaning to ignore the logical variable, 1 to include it in the final logical expression. If LC_i is 0, the corresponding A_i, B_i, C_i, D_i, and E_i should not be entered. Hence in the above example, LC_1, LC_2, and LC_3 would all have the value 1. If only $F_1 \geq 0$ was required to generate the "OFF" signal, OFF = L_1, then LC_1 should be entered as 1, and LC_2 and LC_3 should be entered as 0. This same discussion applies to the

"ON" signal and its logical selection switches LC_4, LC_5, and LC_6.

2.3. TIME DELAYS

Four programmable time delays are available in the control logic. The first time delay, τ_1, is the delay between the evaluations of F_1 and the evaluations of either F_2 or F_3. The second time delay, τ_2, is the delay between the time of generation of the "OFF" signal and the time that F_4 may be evaluated in the generation of the next "ON" signal. τ_3 is the delay between the time of evaluation of F_4 and the time of evaluation of either F_5 or F_6. τ_4 is the delay between the time of generation of the "ON" signal and the time that F_1 may be evaluated in the generation of the next "OFF" signal.

2.4. EXAMPLE

A contrived example covering the above outlined features should prove helpful. Consider an antilock system which generates an "OFF" signal subject to the following laws:

1) $\dot{\omega} \leq -50$ ft/sec^2

<u>and</u> 2) at a time .05 seconds after (1) is satisfied, $\omega \leq .9\ \dot{x}$ must also be satisfied.

Suppose the corresponding "ON" signal must satisfy the following requirements

3) $\dot{\omega} \geq -5.$ ft/sec^2

<u>and</u> 4) at a time .02 seconds after (3) is satisfied, $\omega \geq .8\ \dot{x}$ must also be satisfied.

Suppose also that once the "ON" signal is generated during any cycle, the test for the next "OFF" signal must not take place for at least 0.1 second guaranteeing a certain amount of brake on-time.

Rewriting (1) as,

$$F_1 = -\dot{\omega} - 50 \geq 0$$

$$A_1 = -1.0$$

$$B_1 = 0.0$$

$$C_1 = -50.0$$

$$D_1 = E_1 = 0.0$$

Similarly for (2),

$$F_2 = -\omega + .9\,\dot{x} \geq 0$$

$$A_2 = 0.0$$

$$B_2 = -1.0$$

$$C_2 = 0.0$$

$$D_2 = 0.9$$

$$E_2 = 0.0$$

Since there is no F_3 necessary, LC_3 should be entered as 0, while LC_1 and LC_2 are entered as 1. OP_{12} should be entered as 1 since OFF = L_1 AND L_2. OP_{23} has no meaning here and can therefore be either 0 or 1. The time delay between F_1 and F_2 implies $\tau_1 = 0.05$. Since there is no time delay specified between the generation of the "OFF" signal and the evaluation for the next "ON" signal, $\tau_2 = 0.0$.

Similarly for the "ON" criteria, (3) may be rewritten as,

$$F_4 = \dot{\omega} + 5 \geq 0$$

$$A_4 = 1.0$$

$$B_4 = 0.0$$

$$C_4 = 5.0$$

$$D_4 = 0.0$$

$$E_4 = 0.0$$

Likewise,

$$F_5 = \omega - .8\,\dot{x} \geq 0$$

$$A_5 = 0.0$$

$$B_5 = 1.0$$

$$C_5 = 0.0$$

$$D_5 = -0.8$$

$$E_5 = 0.0$$

Since there is no F_6 necessary, LC_6 should be entered as 0 while LC_4 and LC_5 are entered as 1. OP_{45} should be entered as 1 for the required "AND" operation while OP_{56} has no meaning and can be either 0 or 1. The time delay between F_4 and F_5 requires $\tau_3 = 0.05$. The time delay between the "ON" signal and the test for the next "OFF" signal requires $\tau_4 = 0.10$. Since F_3 and F_6 were not needed in the above scheme, they were ignored by setting LC_3 and LC_6 equal to 0. Their coefficients should not be entered in the input streams.

It is noted that wheel slip (S) is not one of the variables in the inequality expressions. However, an equivalent expression involving ω and $\dot{x}$ can be obtained by noting that

$$S = 1 - \frac{\omega}{\dot{x}} = \frac{\dot{x} - \omega}{\dot{x}} .$$

If $F_1 \equiv S \geq S_1$ is the desired expression, then,

$$\frac{\dot{x} - \omega}{\dot{x}} \geq S_1$$

or,

$$\dot{x} - \omega \geq S_1 \dot{x}$$

or,

$$-\omega + (1 - S_1)\dot{x} \geq 0$$

is its equivalent form, where,

$$A_1 = 0.0$$
$$B_1 = -1.0$$
$$C_1 = 0.0$$
$$D_1 = 1.0 - S_1$$
$$E_1 = 0.0$$

From a control theory viewpoint, any of these arithmetic inequalities are equivalent to constructing switching lines and boundaries in the $\dot{\omega}$ versus ω phase plane. Some remain stationary, such as $\dot{\omega} \leq a$, while others move continuously, such as $\omega - b\dot{x} \leq 0$, dependent on vehicle velocity.

2.5. ADAPTIVE COEFFICIENTS

A number of antilock systems possess an adaptive feature for the coefficients involved in their control laws. Usually these coefficients are adaptive to vehicle deceleration or its approximation derived from wheel spin-up time, which indirectly reflects the road surface friction condition. Such an option is available for the inequality expression coefficients in this simulation. In order to use the option, the user need only enter two additional parameters per card following the original parameter. The third parameter on the card represents the break-point value of vehicle accelerations below which the coefficient takes the first parameter value, above which the coefficient takes the second parameter value. For example, if a certain coefficient was to have the value 1.0 for vehicle accelerations less than -.4 g's (-13.0 ft/sec^2) and have the value 0.7 for vehicle acceleration greater than 0.4 g's, the values entered on that coefficient's card would be in the order:

1.0 0.7 -13.0

Any of the inequality expression coefficients have this option available. If the second and third parameter value fields are left blank, the program considers the coefficient constant and equal to the parameter value in the first field.

2.6. SIDE-TO-SIDE OPTIONS

Four different side-to-side options per axle are available. One antilock system is allowed for each axle with the same pressure being returned to each side for three of the available options while the fourth option allows for independent pressure return and wheel observation. These are summarized below:

OPTION 1 - Worst Wheel. The wheel having the lowest rotational rate for a given axle is selected by the control logic as its input. The same pressure is returned to both sides based on this input.

OPTION 2 - Best Wheel. Same as Option 1 except that the wheel with the highest rotational rate is selected as input.

OPTION 3 - Average Wheel. Both wheel rates are averaged by the control logic module and used as input. The same pressure is returned to both sides.

OPTION 4 - Independent Wheel. Each wheel per axle is selected independently of the other side as input to the control logic module and pressure is returned to each side independent of the other.

A summary of the parameter input requirements for the control logic module is as follows (one set per axle):

1) 6 logical selection switches (3 for "ON", 3 for "OFF").

2) 5 coefficients for each of the necessary inequality expressions. The adaptive feature mentioned above could result in one additional value for any or all coefficients adaptive to vehicle deceleration and its corresponding vehicle acceleration break-point (2 parameters plus their break-point value per card).

3) 4 programmable time delays; τ_1, τ_2, τ_3, τ_4.

4) 4 logical operator switches (2 for "ON", 2 for "OFF").

For vehicle velocities less than 10 ft/sec, the anti-lock simulation is inactivated and line pressures will follow the treadle pressure.

3. PRESSURE MODULATOR

3.1. TIME DELAYS

The input received by the pressure modulator is simply the "ON" and "OFF" signals generated in the control logic module. Once a control signal is received there is normally a time delay before actual pressure reduction or increase takes place. These time lags are denoted in the simulation as τ_{ON} and τ_{OFF} and are program inputs specified by the user.

3.2. RISE AND FALL RATES

The pressure fall and pressure rise are defined to be exponential in time with the pressure rise limit set by the treadle valve output pressure and the pressure fall limit as zero pressure. Two pressure fall rates and a pressure hold can be programmed by the user for the off period; similarly, two pressure rise rates and a pressure hold for the on period. The rise rates and fall rates mentioned here are defined as the inverses of the time constants associated with the exponential pressure rise and fall.

The two pressure rise rates and two fall rates are offered as input and must be specified by the user. These are denoted as PR_1, PR_2, PF_1, and PF_2. However, in order to provide a great deal of flexibility, the fall rates are defined to be a function of another variable called ε_1, and the rise rates a function of ε_2. ε_1 and ε_2 are in turn defined as follows:

$$\varepsilon_1 = H_1\dot{\omega} + H_2\omega + H_3 + H_4\dot{x} + H_5\omega_1 + H_6P + H_7P_d$$

$$\varepsilon_2 = G_1\dot{\omega} + G_2\omega + G_3 + G_4\dot{x} + G_5\omega_1 + G_6P + G_7P_d$$

where

$P \equiv$ brake pressure

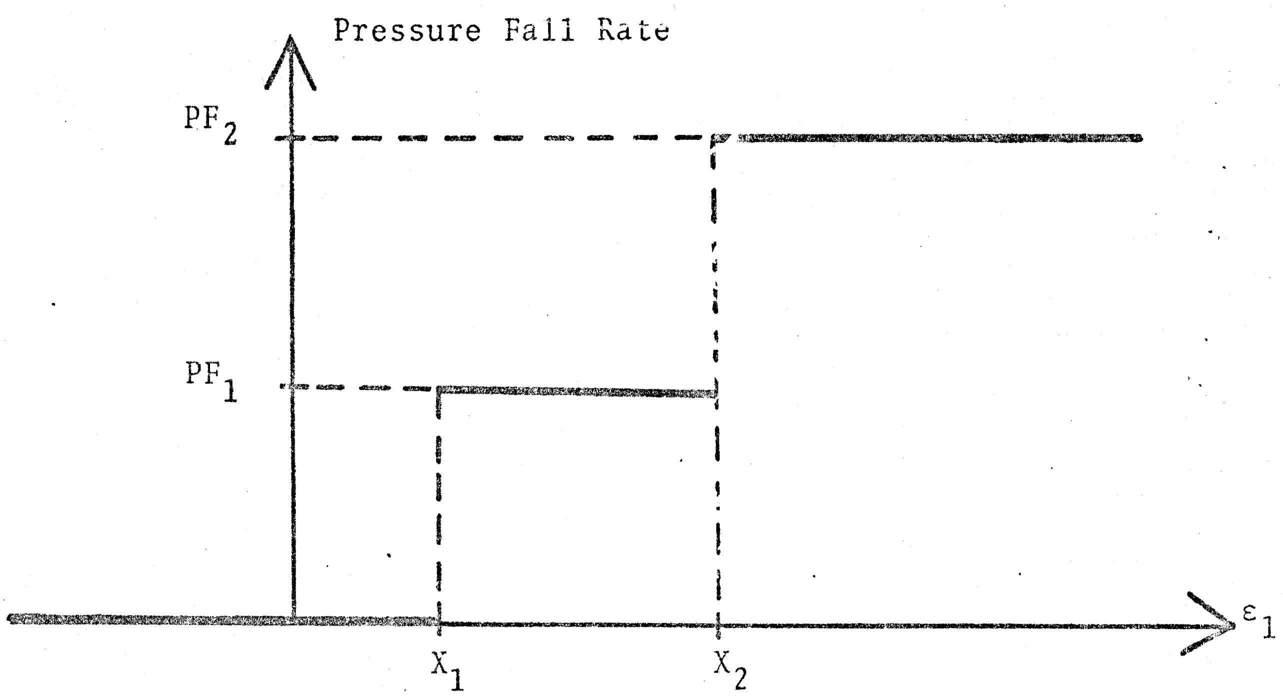
$P_d \equiv$ treadle valve output pressure

and the other variables are defined as before in the control logic module.

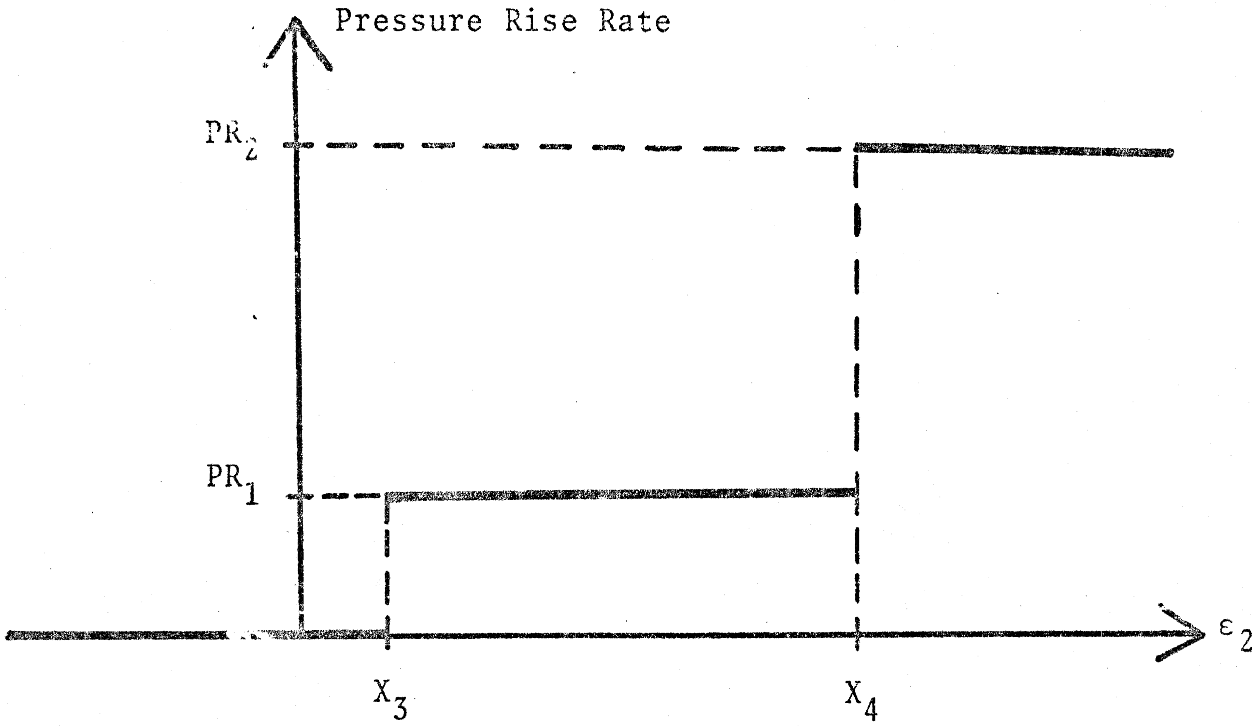
H_i and G_i , $i = 1,7$

are coefficients chosen by the user as input.

Two break-points are necessary along the ε_1 axis to distinguish the three regions of operation for the three fall rates (0, PF_1, PF_2). These are denoted as X_1 and X_2 and are specified as input by the user. The following figure illustrates this relationship:



Likewise for the pressure rise rates and their corresponding break-points X_3 and X_4:

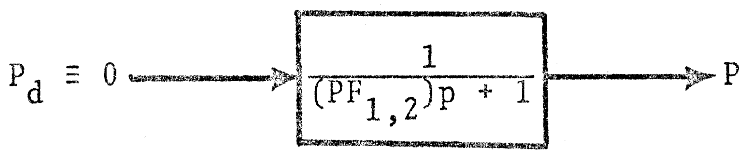


As shown, the pressure fall rate assumes the value PF_1 for, $X_1 \leq \varepsilon_1 \leq X_2$, and the value PF_2 for, $\varepsilon_1 > X_2$ and the value zero for $\varepsilon_1 < X_1$. Similarly, the pressure rise rate assumes the value PR_1 for, $X_3 \leq \varepsilon_2 \leq X_4$, and the value PR_2 for, $\varepsilon_2 > X_4$, and the value zero for $\varepsilon_2 < X_3$.

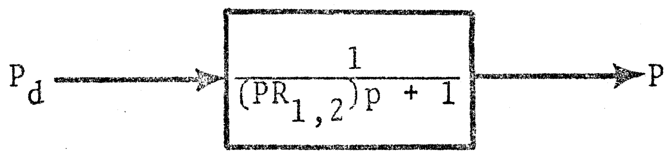
Therefore, by specifying the rise and fall rates and the associated break-points and by defining ε_1 and ε_2 as functions of the desired variables, the user has available a fairly flexible framework in which to simulate a number of different pressure modulator characteristics.

In terms of transfer function notations, the above relationships can be expressed as:

<u>Pressure Fall</u>



<u>Pressure Rise</u>



where $PF_{1,2}$ and $PR_{1,2}$, defined above, are functions of ε_1 and ε_2, respectively, and p is an operator denoting differentiation with respect to time.

3.3. EXAMPLE

In order to help clarify the above discussion, consider the following example of a certain pressure modulator's characteristics:

1) "ON" delay = "OFF" delay = 0.05 seconds.

2) The pressure rise rate assumes an approximate value of $(0.2)^{-1} = 5.0$ for differences between treadle valve output pressure and line pressure of 50 psi or more, and an approximate rise rate of $(0.33)^{-1} = 3.0$ for pressure differences of less than 50 psi.

3) The pressure fall rate is approximately constant for all line pressure values with a fall rate equal to $(0.25)^{-1} = 4.0$.

This could be simulated by the following choice of input parameters:

$$\tau_{ON} = 0.05$$

$$\tau_{OFF} = 0.05$$

$$H_1 = H_2 = H_4 = H_5 + H_6 = H_7 = 0.0$$

$$G_1 = G_2 = G_3 = G_4 = G_5 = 0.0$$

$$H_3 = 1.0 \qquad \} \rightarrow \varepsilon_1 = 1.0$$

$$G_6 = -1.0 \ , \quad G_7 = 1.0 \quad \} \rightarrow \varepsilon_2 = P_d - P$$

$$X_1 = X_2 = 0.0$$

$$PF_1 = PF_2 = 4.0$$

$$X_3 = 0.0 \ , \quad X_4 = 50.0$$

$$PR_1 = 3.0 \ , \quad PR_2 = 5.0$$

APPENDIX I

INPUT DATA

If antilock is not to be used, nothing need be done to the data stream given in the Phase II Technical Report. If an antiskid system is to be entered, a -1 should be entered before the variable IWIND in the input data stream. The remaining antilock keys and data are then entered after the variable IWIND.

$IALOPT_i$ is a key which must be entered for each axle on the vehicle. A value of less than zero implies that a new antilock system follows. All of the parameters described below must be entered for this axle. A value of zero implies there will be no antilock for axle i. A value greater than zero implies that axle i will have the same system as a previous axle, namely, the axle corresponding to the value entered for $IALOPT_i$. Thus, $IALOPT_i$ must be less than i. $IALOPT_i$ is entered in I2 format.

The following list defines all the input parameters required for each antilock system used (one or none per axle). The parameters should be entered in the order given below.

OPTION , Side-to-Side Option Key

01 => Worst Wheel

02 => Best Wheel

03 => Average Wheel

04 => Independent Wheel

(I2 Format)

LC_1, LC_2, LC_3, LC_4, LC_5, LC_6	logical selection switches to include or ignore each of the inequalities (3I1 format, 3 values per card)
(A_i, B_i, C_i, D_i, E_i ,	$i = 1,6$) up to 6 sets of 5 coefficients each; one set for each of the necessary inequality expressions; an additional coefficient and the vehicle acceleration break-point are required per card for any coefficients adaptive to vehicle acceleration (3F15.5 format, one coefficient per card)
τ_1, τ_2, τ_3, τ_4	programmable time delays in the control logic (F15.5 format, one value per card)
OP_{12}, OP_{23}, OP_{45}, OP_{56}	logical operator switches indicative of "OR" or "AND" operations (2I1 format, 2 values per card)
H_1, H_2,...,H_7	seven coefficients in the ε_1 expression (F15.5 format, one value per card)
G_1, G_2,...,G_7	seven coefficients in the ε_2 expression (F15.5 format, one value per card)
τ_{ON}	time delay between time "ON" signal is received and time that actual pressure increase begins (F15.5 format)

τ_{OFF}	time delay between time "OFF" signal is received and time that actual pressure decrease begins (F15.5 format)
X_1, X_2	break-points along the ε_1 axis which define the three different pressure fall rate regions (F15.5 format, one value per card)
X_3, X_4	break-points along the ε_2 axis which define the three different pressure rise rate regions (F15.5 format, one value per card)
PF_1, PF_2	two pressure fall rates (F15.5 format, one value per card)
PR_1, PR_2	two pressure rise rates (F15.5 format, one value per card)
τ_ω	time constant of first order filter relating wheel rate to derived wheel rate (F15.5 format)
$\tau_{\omega d}$	time constant of first order filter and differentiator expression relating derived wheel rate to desired wheel acceleration (F15.5 format)

Note: Only those sets of inequality expression coefficients (A_i, B_i,..., E_i) should be entered for which the corresponding logical selection switches (LC_1, LC_2,...,LC_6) are selected as 1. That is if $LC_1 = LC_2 = LC_4 = 1$, and $LC_3 = LC_5 = LC_6 = 0$, enter only in order the inequality expression coefficients (A_1, B_1, ..., E_1), (A_2, B_2,..., E_2), and (A_4, B_4,..., E_4).

The following is an example input listing. The integer -1 has been placed before IWIND to call for the antiskid algorithm. The second variable is IWIND. Then the antiskid data follows.

	-1
IWIND	-1
$IALOPT_1$	-1
OPTION	01
LC_1, LC_2, LC_3	100
LC_4, LC_5, LC_6	100
A_1	-1.0
B_1	0.0
C_1	-5.0
D_1	0.0
E_1	0.0
A_4	1.0
B_4	0.0
C_4	10.
D_4	0.
E_4	0.
τ_1	0.0
τ_2	0.2
τ_3	0.0
τ_4	0.0
OP_{12}, OP_{23}	00
OP_{45}. OP_{56}	00
H_1	1.0
H_2	0.0
H_3	100.
H_4	0.0
H_5	0.0
H_6	0.0
H_7	0.0

G_1	1.0
G_2	0.0
G_3	-5.0
G_4	0.0
G_5	0.0
G_6	0.0
G_7	0.0
τ_{ON}	0.06
τ_{OFF}	0.04
X_1	-10000.
X_2	0.0
X_3	-10000.
X_4	0.0
PF_1	10.0
PF_2	5.0
PR_1	5.0
PR_2	10.0
τ_{ω}	0.01
$\tau_{\omega d}$	0.02
$IALOPT_2$	01
$IALOPT_3$	01
$IALOPT_4$	01
$IALOPT_5$	01

TINC

TRUCK

Note that the last 4 parameters in this list ($IALOPT_2$,..., $IALOPT_5$) are set to 01, indicating that the systems for axles 2 through 5 are identical to the system for axle 1. If a different system is desired for one of these axles, a -1 should be entered instead, followed by its appropriate input data similar to the above example for axle 1.

EXAMPLE

Suppose an antilock system possesses the following features: (1) a wheel sensor time delay effect of 10 ms. and another 20 ms. in the derivation of wheel acceleration; (2) control logic which generates an "OFF" signal once the wheel acceleration falls below -50.0 ft/sec^2 and an "ON" signal for wheel accelerations greater than -10.0 ft/sec^2; (3) pressure modulator time delays of 40 ms. for "OFF" signals and 60 ms. for "ON" signals. The supposed pressure rates are functions of wheel deceleration defined as follows:

$$\text{Pressure Fall Rate} \equiv \begin{cases} (0.1)^{-1} = 10.0 \text{ for } \dot{\omega} \leq -100 \text{ ft/sec}^2 \\ (0.2)^{-1} = 5.0 \text{ for } \dot{\omega} > -100 \text{ ft/sec}^2 \end{cases}$$

$$\text{Pressure Rise Rate} \equiv \begin{cases} (0.2)^{-1} = 5.0 \text{ for } \dot{\omega} \leq 50 \text{ ft/sec}^2 \\ (0.1)^{-1} = 10.0 \text{ for } \dot{\omega} > 50 \text{ ft/sec}^2 \end{cases}$$

The following choice of input parameters would satisfy the above antilock system:

$\tau_\omega = 0.01$

$\tau_{\omega d} = 0.02$

$A_1 = -1.0$

$C_1 = -50.0$ $\quad \rightarrow F_1 = -\dot{\omega} - 50.0 \geq 0$

$A_4 = 1.0$

$C_4 = 10.0$ $\quad \rightarrow F_4 = \dot{\omega} + 10.0 \geq 0$

All other arithmetic inequality coefficients set to zero.

$\tau_1 = \tau_3 = \tau_4 = 0.0$

$\tau_2 = 0.2$

$H_1 = 1.0$

$\rightarrow \varepsilon_1 = \dot{\omega} + 100.0$

$H_3 = 100.0$

$X_1 = -10000.0$

$X_2 = 0.0$

$PF_1 = 10.0$

$PF_2 = 5.0$

$G_1 = 1.0$

$\rightarrow \varepsilon_2 = \dot{\omega} - 50.0$

$G_3 = -50.0$

$X_3 = -10000.0$

$X_4 = 0.0$

$PR_1 = 5.0$

$PR_2 = 10.0$

$LC_1 = 1$

$LC_2 = LC_3 = 0$

$LC_4 = 1$

$LC_5 = LC_6 = 0$

$OP_{12} = OP_{23} = OP_{45} = OP_{56} =$ either 0 or 1

$\tau_{ON} = 0.06$

$\tau_{OFF} = 0.04$

Note that these parameters define the same axle system as specified in the previous example listing for axle 1.

ACCELERATION CAPABILITY FOR THE DIRECTIONAL RESPONSE PROGRAM

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1. INTRODUCTION

In all the runs documented in the Phase II Technical Report, the simulated vehicle was assumed to be in a free-rolling condition. Thus the longitudinal force resulting from front wheel steer angles caused the simulated vehicle to slow down during the course of all the handling maneuvers. This problem has now been remedied since the capability to apply drive torque to the vehicle has been added to the simulation. Thus the user may now elect to try to hold the vehicle speed constant, or, in fact, to increase the vehicle speed during a handling maneuver.

2. THE EQUATIONS OF MOTION

Since the wheels are already set up to receive brake torque, we simply enter a negative brake torque vector value in the event drive torque is to be used. However, the "roll" torque applied to the drive axles from the drive line must now be considered in the axle equations. This was done in a straightforward fashion. Consider Figure 3-12 on page 29 of the Phase II Technical Report, which has been reproduced here as Figure 1.

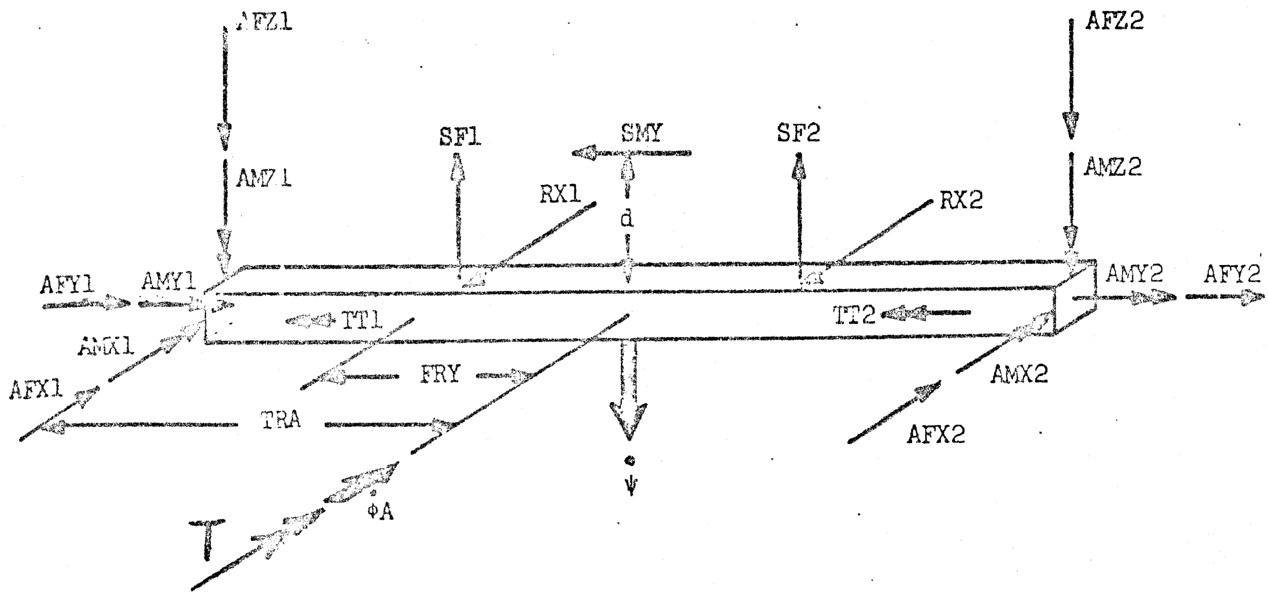

Figure 1. Free body diagram: single axle.

Note that we have added to Figure 3-12 the drive torque T being applied from the drive line to the differential. The differential is modeled by the following equations:

$$TT1 = TT2 \qquad (1)$$

$$-(TT1 + TT2) = T \cdot ARATIO \qquad (2)$$

where ARATIO is an input variable indicating the ratio of torque applied at the differential to the drive torque. In the case of tandem axles, it is assumed that half the drive torque is applied to each axle, and that the differential on each axle splits the torque as in the case of the single axle. Thus in the case of a single drive axle or of tandem drive axles, only an algebraic manipulation is necessary to find the drive torque applied to the wheels.

It should be noted, however, that the torque T applied from the drive line to the axle may result in sizeable side-to-side load transfer. The additional term in the equations of motion is not complicated—Equation (3-38a) in the Phase II Technical Report must be modified to read:

$$(SF1-SF2)FRY + (AFZ2-AFZ1)TRA$$

$$- SMY(d) + AMX1 + AMX2 + T = J_a \cdot \ddot{\phi}A \qquad (3)$$

The results of this side-to-side load transfer may be quite important. Note that in the four spring suspension, one must expect the rear axle to unload due to drive torque. This, in addition to the side-to-side load transfer, may result in quite a low normal load for one side of the rear tandem axle.

3. PROGRAMMING DETAILS

The addition of the acceleration capability to the Phase II simulation requires no change in the data stream if the acceleration capability is not to be used. If drive torque is required, the floating point variable +2. should be entered preceding the G1 data entry.

If the drive torque capability is to be used, the axle ratio, ARATIO, should be entered following the +2. entry. Next, a time a versus total drive torque table must be entered. The first table entry is a data card in I2 format giving the number of time drive shaft torque pairs in the table. Following this entry, up to 25 coordinate pairs may be entered in 2F10.2 format. The first of the two numbers is time. The second is the corresponding total drive torque value in inch/pounds. All the above data should be entered immediately before the variable G1.

An example drive torque data list is given below. The first data entry is the axle ratio. The second entry gives the number of pairs in the time versus drive torque table, in this case five. This entry is followed by the table itself. A plot of drive torque versus time called for by the table is given in Figure 2.

DATA LIST
TOTAL DRIVE TORQUE

3.1	
05	
0.	0.
1.	0.
1.05	1500.
2.0	1500.
2.05	0.

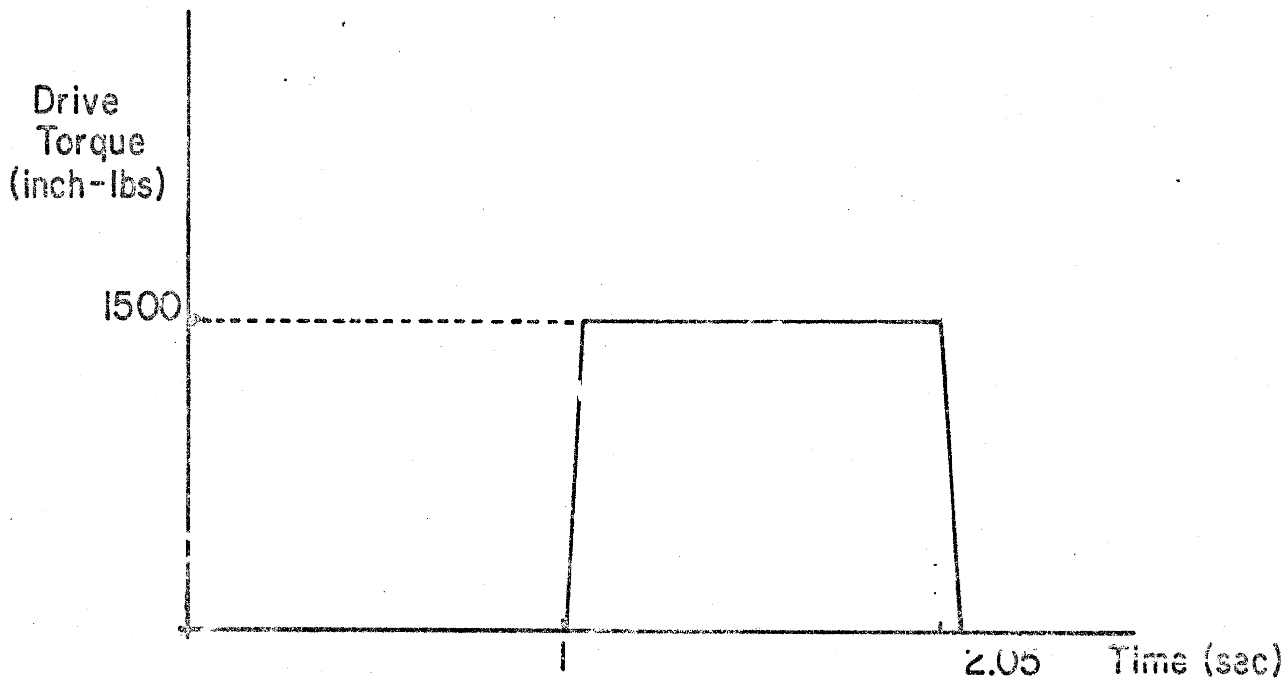

Figure 2: Total Driving Torque vs. Time

INDEPENDENT FRONT SUSPENSION

Technical Details

A schematic diagram of the independent front suspension is given in Figure 1. The wheels are assumed to be restrained laterally by an imaginary link from the tire-road interface to the roll center as is shown in the free-body diagram of the left front wheel given in Figure 2. (Note that θR is assumed to be a constant.)

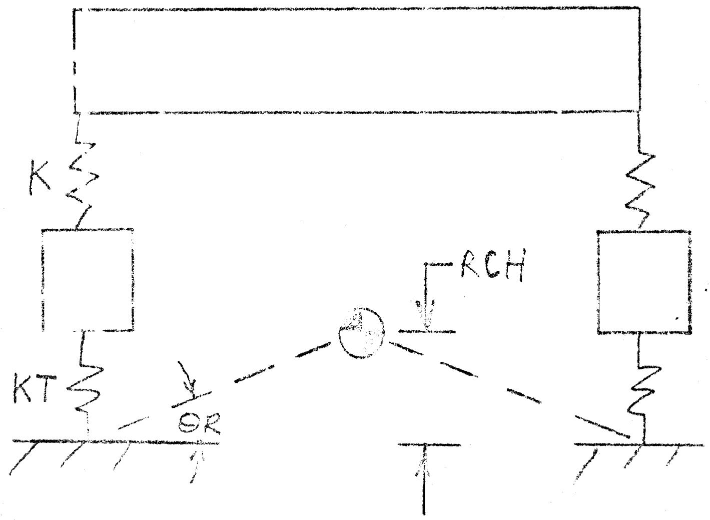

Figure 1. Independent front suspension, rear view

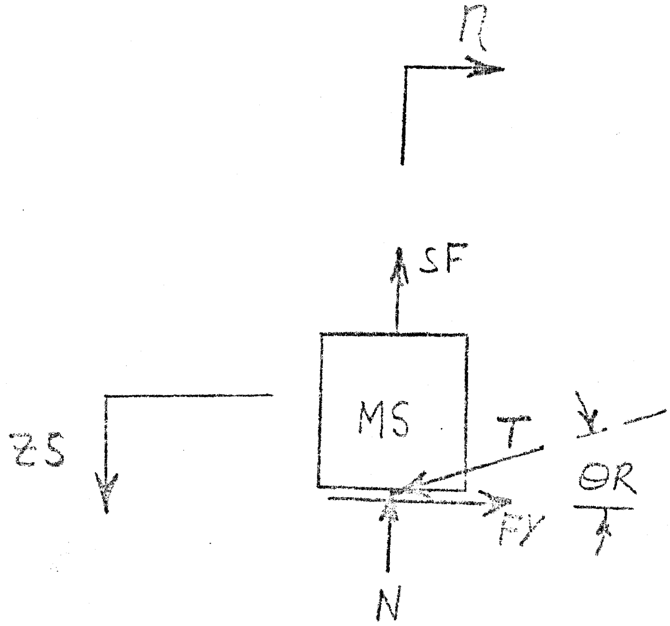

Figure 2. Free-body diagram, left front wheel

The summation of forces laterally yields

$$MS\ \ddot{\eta} = FY - T \cdot \cos\theta R \qquad (1)$$

The lateral acceleration of the mass is estimated by procedures explained in Section 3.3 of the Phase II report. Since the lateral component of the shear force is known, T may be calculated using Equation (1).

The vertical equation now may be written

$$MS \cdot \ddot{ZS} = -N - SF + T \cdot \sin\theta R \qquad (2)$$

Similar equations have been written to compute the vertical accelerations of the right front wheel. Thus the free-body diagram of the front end of the vehicle, given in Figure 3, yields the following information.

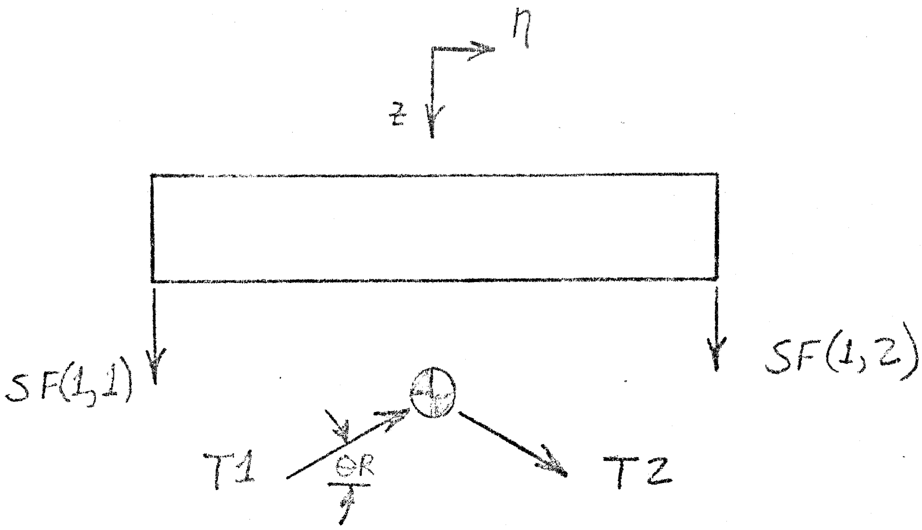

Figure 3. Free-body diagram, sprung mass front end, rear view

$$\Sigma F_z = S(1,1) + S(1,2) + (T2 - T1) \sin \theta R \qquad (3)$$

$$\Sigma F_\eta = (T1 + T2) \cos \theta R \qquad (4)$$

Equations (3) and (4) are used in the computations of the sprung mass accelerations.

Programming Details

If the user wishes to simulate a solid front axle suspension, the data stream is ~~unchanged~~ from that given in the Phase II Technical Report. For an independent front suspension, the first entry in the data stream should be the integer -1. This card follows the header card and precedes the axle key.

See next page.

Camber is an important consideration in the prediction of side forces of the tires of an independent front suspension. To include the effect of the camber angle, γ, in the tire model, the slip angle, α, is modified in the following way.

$$\alpha' = \alpha + (-1)^J \frac{C_\gamma}{C_\alpha} \cdot \gamma$$

where C_γ is the camber stiffness in pounds/degree and J is 1 for the left side, 2 for the right side. Note that for $\alpha = 0$ and longitudinal slip S = 0 we have

$$F_{y\gamma} = (-1)^J C_\alpha \cdot \alpha' = (-1)^J C_\gamma \cdot \gamma$$

Programming Details

Camber data must be entered where the independent front suspension is used. The camber stiffness, C_γ (pounds/degree), is inserted in front of the longitudinal stiffness C_S. A table of camber angle versus suspension deflection is inserted after the steer tables in the usual way, i.e., the first entry is an integer in 02 format giving the number of pairs in the table, then the table entries follow in F10.2 (deflection first, angle second).

AUXILIARY ROLL STIFFNESS

Technical Details

The auxiliary roll stiffness model takes a slightly different form for the solid axle and the independent front suspension. We have assumed the auxiliary roll stiffness is zero for all tandem axles. (Note that air suspensions are not a Phase II option.) In the solid axle configuration the auxiliary roll stiffness is assumed to apply a roll moment to the sprung mass and the axle that is proportional to their relative roll angles. Thus if the axle roll angle is THETA and the body roll angle is ϕ, the applied moment is

$$\text{AUXROL} = \text{KRS} * (\phi - \text{THETA}) \qquad (5)$$

where KRS is the auxiliary roll stiffness.

This term then appears as a negative moment in the sprung mass roll equations, and a positive roll moment in the axle roll equations.

In the case of an independent front suspension, the equations become slightly more complicated since there is no axle roll equation. To facilitate the computations, a hypothetical axle roll angle is computed for the independent front suspension.

$$\text{THETA} = \frac{\text{ZS}(1,2) - \text{ZS}(1,1)}{\text{TRA}} \qquad (6)$$

where the ZS are the wheel positions, and TRA is the track.

The roll moment is again assumed to be

$$\text{AUXROL} = \text{KRS} * (\phi - \text{THETA}) \qquad (7)$$

In this case, however, the moment is applied by adding a couple through the suspension forces. Thus

$$SF(1,1) = SF(1,1) + \frac{AUXROL}{TRACK} \tag{8a}$$

$$SF(1,2) = SF(1,2) - \frac{AUXROL}{TRACK} \tag{8b}$$

These suspension forces are used in both the sprung mass and the unsprung mass equations, thus, the appropriate moment is transferred to the sprung mass and the appropriate forces are transmitted to the unsprung masses.

Programming Details

The auxiliary roll stiffness must be entered for each axle which is not a tandem axle. Thus KRS, in units of inch pounds/degree, should be input after the suspension spring constant K and before the tire spring constant KT. Nothing should be done to the tandem axle data stream.

SPLIT μ SURFACE

The original version of the Phase II program required side-to-side symmetry in tire parameters. The program has now been altered to allow different μ_o values for each tire (or set of dual tires). To alter the input data stream to accept these variations, the integer -1 must be entered after the last KT entry, and then a μ_o value for each tire (or set of dual tires). Note that if side-to-side symmetry is desired, the user should follow the procedure outlined in the Phase II Technical Report.