FEASIBILITY ANALYSIS AND EVALUATION OF AN ADAPTIVE TRACKED VEHICLE SUSPENSION AND CONTROL SYSTEM

FINAL REPORT

JUNE 1975

Contract No. DAAE07-72-C-0176

by

Robert M. Salemka
National Water Lift Company
A Division of Pneumo Corporation

and

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TACOM
MOBILITY SYSTEMS LABORATORY
U.S. ARMY TANK AUTOMOTIVE COMMAND

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2.0 ABSTRACT

This study shows that adaptive control of the jounce damping characteristics of the first and last wheel of a tracked vehicle can cause a significant improvement in performance. This improvement resulted in an overall 30 percent reduction in average pitching rate of the hull, as measured on the simulation of the MICV vehicle traversing the JEA bump course.

Verification testing of the computer model with actual performance data of the MICV vehicle showed good correlation of peak amplitudes and hull resonance. This data also confirmed that the actual dampers are working well below recommended levels.

A proposed method of mechanizing and testing the adaptive control on an actual vehicle is presented along with system schematics and preliminary performance specifications for the critical components.
3.0 **INTRODUCTION**

3.1 **Purpose**

The primary purpose of this study was to investigate the feasibility of using adaptive damping to improve the suspension characteristics of a tracked vehicle. Secondary purposes were to develop an analog computer model which could be used for more complete evaluation by TACOM and to propose a method of mechanization of the adaptive principle for actual hardware testing.

3.2 **Vehicle and Terrain**

The MICV vehicle running on the JEA bump course was selected as a candidate vehicle and terrain. The primary reason was the availability of test data of pitch and heave while traversing the bump course as well as high speed movies showing detail motion of the track, wheels, and hull. The suspension system has a dual rate mechanical spring rate which is as soft as a hydropneumatic system.

3.3 **Adaptive Control**

The adaptive control was achieved by switching the jounce damping relief valve between two different relief pressure points as a function of pitching rate of the hull. When the hull was pitching up, the damping was reduced on the front wheels and increased on the rear wheels. When the hull was pitching down the process was reversed. A modification using the heave rate of the hull was also tested by switching the damping control on the summed signal of the pitch rate times a constant plus the vertical heave velocity. The constant was determined by the
distance between the wheel and the center of gravity of the hull. Careful consideration was given to the sign of the summation so that a positive heave signal would tend to reduce the damping on both the front and rear wheel and negative heave would increase the damping on both the front and rear wheels. Other modifications investigated included the bi-level damping control in the jounce direction, and full adaptive control on the first, second and sixth wheel.

3.4 Describing Equations

Two sets of describing equations have been developed. The first set assumes the availability of quite large analog or digital computer capabilities. The second set is less complete and tailored around the limitations of a modestly sized analog computer.

The major improvement of these describing equations over previous simulations is that the ground force is considered to act on the wheel at right angles to the local slope of the ground profile. Previous studies were done with the ground force acting vertically up regardless of the slope of the bump.

The effects of the track have also been included for the first time. This effect has been found to have a major impact on the dynamic behavior of the hull primarily due to an apparent change of stiffness of the suspension system due to the track. A difference of almost 2:1 in pitching frequency of the hull has been measured with and without the track.
3.5 **Computer Model**

The analog computer model differs from previous models principally because of the use of the direction of the ground force vector acting to rotate the hull. The previous studies treated the ground force vector as a vertical force only. Thus the torque to the hull was essentially proportional to the ground force. In actual practice the torque is a function of both the magnitude and the direction of the ground force. The other major difference is the inclusion of the track and track tensioning device.

A Systron Donner Model 80H analog computer was used for the simulation. One hundred and four internal amplifiers were supplemented with an additional forty-two external amplifiers for the simulation.

3.6 **Proposed Mechanization**

An existing tracked vehicle already equipped with hydro-pneumatic suspension has been selected as ideally suited for the mechanization of adaptive damping; main reasons were the degree of improvement to be expected, the ease and cost of rework, and the applicability of the results.

The suggested design of the test hardware will result in a system with adaptive damping on the two front and two rear wheels with hull pitch, heave and roll sensors combining signals to cause switching of the damping solenoids. This system is flexible enough so that all combinations of sensors and switching solenoids can be used or deleted for system evaluation.
Hardware testing is expected to be done with emphasis placed on the quality of the ride and the life expectancy of the dampers.

**4.0 DISCUSSION**

**4.1 Computer Simulation**

**4.1.1 Candidate Terrain**

In the early phases of the study, a number of candidate terrains were considered. Availability, comparative test data, and ease of implementation were the controlling factors effecting the choice. By mutual agreement with the cognizant TACOM engineers, it was decided to use the JEA bump course for the NWL studies to be followed by a final evaluation by TACOM using the number 12 rocky Fort Knox course from RRC9.

The JEA bump course is an existing terrain simulation which NWL has used in past simulations and for which comparative performance data on the MICV vehicle has been taken. The details of the obstacles and course are shown on Figure 26. Use of this course gives a data base from which relative comparisons can be made.

**4.1.2 System Equations**

The describing equations used to simulate the system are included in Appendix "A" of this report.

The basic philosophy has been to simulate only one side of the hull, neglecting all roll characteristics. This simplifies the problem to three degrees of freedom.
A unique feature of this simulation as opposed to previous studies, was that the vector direction of the ground force with respect to the center of gravity of the hull has been considered. This causes a difference in pitching motion on the hull which is quite significant. The difference between the spring torque on the road arm and the corresponding ground reaction component due to road arm angle has also been considered.

The effects of track tension and track inertia on the performance characteristics have been included. A discussion of the simulation is included in Appendix "A".

4.1.3 One Wheel Study

The physical limitation of the size of the computer required some simplifications of the describing functions for the system. To select which parameters could be simplified without affecting final results, a single wheel of the suspension system was simulated. The simulation was initially made without regard to usage of computer components but rather to give the best mathematical model that was achievable. This model was then used as a comparison against various simplifications to determine the best model that could be made.

The major result of this study was that the centripetal force and tangential acceleration force vectors could be modeled quite closely by a single acceleration acting at a fixed radius from the center of gravity of the hull. This distance roughly corresponds to the nominal steady state distance between the center of gravity of the hull and the centerline of the wheel.
4.1.4 Computer Equations

The original system equations were simplified to accommodate the available computer. These equations along with the detailed computer simulation diagrams are included in Appendix "B" of this report.

By careful selection of variables, the horizontal motion was removed from the equations of motion. This reduced the system to only two degrees of freedom. These were rotation and vertical motion.

The ground profile was built into an electrical circuit as a separate ground profile generator. Twelve separate leads were used to generate the magnitude and slope for each of the six wheels so that the proper sequencing of each bump under each of the six wheels could be properly simulated.

4.1.5 Verification of Model

The computer model was set up to simulate the physical characteristics of the MICV vehicle. This model was then run across the JEA bump course and the results compared with actual test data from the vehicle. The results may be seen by comparing Figures 9.0, 9.1 and 8.0. It appears that the actual 15 mph test run falls between the 10 mph and 15 mph computer runs.

The damping pressure had to be reduced to a 600 psi relief pressure before the computer runs began to have the amplitude and acceleration magnitudes of the actual test data. This reduction in pressure level is substantiated by other test data
which indicates that the actual damping pressure was considerably lower than the design level or the damping level demonstrated in bench testing.

The performance of the simulation is felt to be a good match with the actual performance of the vehicle.

The base line system established by this comparison is summarized in Figures 1 and 2. The 5, 10, 15, 20, 25, 30 and 35 mph speed runs are shown in Figures 7.0, 8.0, 9.1, 10.0, 11.0, 12.0 and 13.0.

4.1.6 Performance Factor

An average pitch rate of the hull over the bump course has been selected as a comparative performance factor for use of these studies for the following reasons:

a) It is very simple to generate.

b) Pitch rate is the single most sensitive source of input disturbance to the vehicle.

c) An actual gunner ignores sharp peaks in pitch rate. Time on target is more a function of average pitch rate.

d) In general a suspension system that reduces pitching will allow a man to perform his tasks more accurately, allowing the gun stabilization system to realize its full potential.

Figures 1 and 2 show this performance factor plotted against speed for the different configurations.
4.1.7 **Improved Non-Adaptive System**

The suspension system was optimized relative to the selected performance factor prior to incorporating the adaptive damping system. The results are summarized in Figures 1 and 2. The detail computer traces are shown in Figures 7.2, 8.2, 9.3, 10.2, 11.2, 12.2 and 13.2. These studies indicate a considerable improvement in pitch rate if the damping pressure relief point is increased to 1200 psi from the apparent present value of 600 psi, and an orifice is included for increased rebound damping. The relative improvement due to increasing the jounce damping and rebound damping is shown in Figure 2.

The average velocity was reduced to .54 of the base line system with more improvement in high speed operation than low speed.

4.1.8 **Adaptive Damping**

Incorporation of adaptive damping control shows an overall average of 30% improvement over an optimized system without adaptive damping. This is summarized in Figure 1. Detail performance curves are shown in Figures 7.3, 8.3, 9.4, 10.3, 11.3, 12.3, and 13.3. Figure 1 shows a tendency for more improvement at the low speed runs than at the higher speeds.

4.1.8.1 The mechanization of the adaptive damping control was to switch levels of the jounce damping relief valve between high damping and low damping based on the sign of the pitch rate of the hull. The bi-level damping curve is shown in Figure 6.
Figure 4 shows the damping curve for the base line system and Figure 5 shows the damping curve for the optimized system without adaptive control.

4.1.8.2 An investigation showed that switching the level of rebound damping rather than jounce damping was less effective. The results are better than simple jounce damping at the same level but not as good as adaptive jounce damping. No data is included.

4.1.8.3 Pitch rate control plus heave velocity of the hull is shown in Figure 1. The data shows a very slight improvement in average pitch rate between 15 and 25 mph, and a slight loss of performance above 25 mph.

The overall effect seems to be little difference in performance between having the additional heave velocity signal and not having it. It should be pointed out however that this particular terrain does not stimulate the vertical resonance frequency of the hull and that perhaps under these admittedly special conditions, the heave signal could show a tremendous improvement.

The detail performance difference between the pitch rate adaptive and the pitch rate plus heave rate adaptive control can be seen by comparing Figures 7.3, 8.3, 9.4, 10.3, 11.3, 12.3, and 13.3 with 7.4, 8.4, 9.5, 10.4, 11.4, 12.4, and 13.4.

4.1.9 Sensitivity Study

A sensitivity study was made to determine the effects on the suspension system of variations in the road arm angle and spring rate. To some degree the two parameters are related since the
apparent vertical stiffness of the suspension is proportional to the torsional stiffness and inversely proportional to the cosine of the road arm angle. The main difference between the two parameters is that the torsional stiffness controls the total energy stored in the suspension system or the peak force at the jounce bump stop, while the road arm angle controls the shape of the energy curve, making it initially stiff, then softer as the road arm angle swings through zero degrees; then stiffer as the road arm swings up to the jounce stop.

4.1.9.1 The effects of suspension stiffness were studied for 750, 900, 1100, and 2000 in-lb/deg stiffness. The base line is 1000 in-lb/deg. Any rate less than 750 caused the wheel to toggle over to the rebound stop due to the 44° road arm angle yielding a bigger change in effective ground force than the corresponding change in force from the spring.

The results show the improvement in ride that can be achieved with a softer suspension. As may be expected, the softer spring yields a lower disturbance to the hull. The improvement however becomes less and less as the hull speed is increased until, at 30 mph there is almost no difference between a 750 in-lb/deg suspension and a 2000 in-lb/deg suspension. Detail performance curves are shown in groups of four from Figure 14.1 through Figure 19.4.

4.1.9.2 The effects of road arm angle were studied for 39°, 44°, and 49°. This is the angle with respect to the hull waterline and represents the static position of the road arms with the hull
on level ground and at rest. The 44° angle is the base line system.

The results show that the more nearly horizontal case (30°) results in less heave and pitch velocity but a greater total pitch angle up to a speed of about 20 mph. Above 20 mph the differences in ride are inconclusive.

The steeper angles had higher pitch and heave acceleration and velocity peaks, but less total pitch angle. The ride appeared to be rougher.

The performance curves for these cases are shown in groups of three from Figure 20.1 through 25.3.

4.1.10 Track Tension

The track tension equations are developed in detail as part of Appendix "A" and "B".

Track tension had the effect of quadrupling the effective stiffness of the suspension system. With the suspension damping set to a very low value, the system was excited and allowed to ring down. With the track tension activated, the pitch resonant frequency was measured at 1.4 Hz. With the track tension effects removed, this frequency dropped to .70 Hz. Because of this tremendous difference in apparent track tension, the behavior of the vehicle across the bump course was drastically different with and without the effects of track tension.

All evaluation data was taken with the track tension active. Had the data been evaluated without the track tension, the pitch
amplitudes would be greatly reduced, the pitch rates would be
down, and there would be much poorer correlation between the
actual vehicle and the simulation. Sample runs were made but
the data is not included in this report.

4.2 Proposed Mechanization

In order to properly evaluate the proposed adaptive concept
design, certain background considerations must be kept in mind,
particularly in terms of the candidate test rig and suspension
components that are chosen.

The first point that should be made is that comparison of
vehicle performances both equipped with, and without the adaptive
damping control feature must be based on optimal configurations
of each. That is, if the existing vehicle damping characteristics
are not optimum for the basic and conventionally damped system,
two situations can occur. First, the adaptive system may
exhibit performance advantages that exist only because the con-
ventional system is not optimum. Secondly, the adaptive system
may not be able to achieve the maximum performance improvement of
which it may be capable. Implicit in these two statements is
the fact that previous work has shown that best performance of
the adaptive system is achieved when it is incorporated into the
optimum conventional system.

The theoretical work also confirmed the validity of the
basic rationale for the adaptive system. When a moving vehicle
encounters ground disturbances, nonlinear periodic motions of the
suspended mass result. The suspension system must damp out these
motions by the generation of velocity-dependent forces. In a conventional system, these forces are also generated when an undisturbed hull traverses the ground disturbance. The suspension damping thus not only removes disturbances in the hull, but contributes to the source of those disturbances, when it reacts to the original ground disturbance. The adaptive concept recognized this anomaly by postulating the following control philosophy: the damping force should only exist when the local hull velocity is in a direction opposite to the damping force. As an example, consider a jounce damper on #1 wheel when it encounters a bump. The upward motion of the wheel creates an upward acting force on the hull due to the damper as well as the winding up of the spring. This increased force causes increased disturbance to the hull. After the bump has been traversed, the damping action acts to remove the periodic motion which is induced in the hull because its force is now always in the opposite direction to the hull motion. The adaptive function removes the damping force when the bump is first encountered, but applies it when the bump has been traversed.

These comments apply only when the suspension system can swallow the ground disturbance. That is, when the ground disturbance does not demand wheel travels greater than the suspension capability. With large obstructions in particular, the suspension system must generate forces large enough to move the hull away from the obstruction, so that the wheel does not bottom
out on the bump stop. This situation demonstrates the desirability of having an adaptive control system that can be switched out under certain conditions. It also points to the main advantages of an adaptive system, which are to smooth out the relatively small disturbances in order to provide a better gun firing platform, increase riding comfort, and reduce heating of the damping mechanism.

With respect to the damper heating problem it should be pointed out that reducing the damping force invariably increases the heating effect. Numerous tests, as well as simulation programs have demonstrated this phenomenon. This has to do with the resonance characteristics of the vehicle in pitch, heave and roll. With no damping at all, the hull motions become so severe that the suspension components are damaged. However, in this case, the heating is zero. Increasing the damping from zero results in a peak in the heating rate at very low damping levels. The heating effect then drops continuously, again becoming zero when infinite damping is reached. At some specific damping level, the ride motion becomes less severe, and the heat dissipation capability of the suspension is least exercised. With adaptive damping added, the damping level could presumably be increased, allowing for a rough but mobile characteristic over severe terrain when the adaptive system disconnected.

The desirability of a high damping level also is a factor when the suspension spring characteristics are optimized.
For the best ride, the spring rate must be as low as possible, since it, too, induces disturbances to the hull. This is not desirable when traversing severe terrain, however, but can be offset by having high damping levels.

**Concept Design Goals**

Previous work has indicated the feasibility and potential advantages of an adaptive system. The concept design proposed is intended to confirm these results by hardware testing and answer certain questions not answered by previous work. These goals are summarized as follows:

1) Evaluate the adaptive system as initially conceived.
2) Subject the system to terrain and operating modes not covered by previous work.
3) Evaluate the use of heave velocity sensing.
4) Evaluate the use of roll velocity sensing.
5) Determine the effect of adaptive damping on front wheels only.
6) Evaluate fail-safe feature.
7) Provide for recording of all dynamic quantities of interest.
8) Measure dynamic performance of sensors and solenoids.
9) Evaluate different levels of sensor switch bias offset.
10) Evaluate drive selection of damping mode.
11) Evaluate variable spring rate provision.
12) Evaluate damping levels.
4.2.1 System Schematic

The proposed integrated adaptive suspension and control system is shown schematically on S-2850023, and Figures 28.0 and 29.0.

It is proposed to rework the damping valve section of an existing hydropneumatic system to incorporate the solenoid valves for bi-level damping control. This rework would be done on the front and rear units of the suspension system. The addition of two rate gyros for pitch and roll, one vertical rate gyro for heave, and the associated electrical logic and switching console completes the major portion of the adaptive system.

An additional solenoid is proposed to connect the hydraulic side of the springs of the first and second units through a manual selection switch. This allows for a manual selection of two different spring rates for the loading wheels. The system is completed with the addition of electrical manual shut-off switches which will allow the vehicle to be run with various combinations of front wheel control only, front and rear wheel control, pitch control with or without heave control, with or without roll control, or no adaptive damping at all.

A complete evaluation of the adaptive damping concept can be made by electrically switching the system into the various modes of operation with no mechanical changes to the system while traversing the same terrain on the same day in the same vehicle.
The proposed system has been optimized for maximum integration, simplicity, flexibility, and ease of conversion.

The system integrates easily into existing hardware and requires only the acquisition of a few additional parts all of which, with the exception of the reworked damper valve and special electrical logic package, are standard existing hardware with proven performance characteristics.

The controlled damping is proposed to be added to both the front and rear wheels. Computer studies indicate that the system will work quite well on just the front wheels. Both front and rear wheels however can be incorporated with very little added effort and it is felt that under conditions not tested on the computer, such as undulating terrain, that both front and rear wheel control may be needed. The rear wheel adaptive damping may be switched out for evaluation of the benefit derived by this additional control.

Pitch and roll rate sensors and a vertical accelerometer make up the sensors for the system. Rate sensors have been selected on the basis of proven performance and system simplicity. A solid state accelerometer is available which has the capability of extending the life expectancy well above the 1000 hour level of the rate sensor, but the electronic circuit would have to be extended to include an integrator with proportional feedback to offset the long term drift problem, and the physical mounting of the accelerometer within the vehicle hull would become more critical.
The six solenoids used by the system are all identical. A conventional off-the-shelf type valve manufactured by UVJL for a number of aircraft and ground vehicle applications is being used. The high flow requirements of the spring rate selector solenoids are accommodated by a pilot operated valve which is driven by the solenoid valve.

Fail safe features are inherently included in this type of system. The solenoid valves are of the normally closed type so that with no electrical power, the system will automatically revert to a conventional hydropneumatic suspension system. (Reference Figure 27.0)

4.2.2 System Evaluation

Low amplitude switching of the solenoids could cause excess wear to the components, reducing their operating life. This is circumvented by the use of a small bias offset on the rate detector and summing circuit. Thus, a discrete pitching rate level must be reached before any switching of the solenoids takes place.

The rate signals from the pitch, roll and heave sensors are summed with an adjustable weighting factor given to each signal. The resulting signal will determine the solenoid position for each of the four variable damping solenoids. Because of the difference in sign of the summation and also to increase the overall reliability, each solenoid will have its own summing network.
The solenoid valve used for varying the effective spring rate for the first two wheels is triggered by a manual on-off switch. This allows the solenoid to run on normal vehicle power without any power conditioners being used.

In actual application, the valve and line restriction and inertial impedance will tend to have a dynamic effect on the modified spring rate. For slow acting disturbances such as undulating terrain, the lower spring rate will be apparent. For fast acting disturbances such as blocks or rocks, the oil transfer between the units will be delayed and the wheel will have its normal high stiffness, even when the lower rate is requested by the solenoid.

4.2.3 Sensor Trade-Off Study

4.2.3.1 Sensor Types. Standard angular rate gyros are used to indicate directly the required pitch and roll rate information. A linear accelerometer is used for the vertical (heave) direction. Linear rate sensors are not commercially available. The acceleration signal is integrated electronically to obtain the heave rate.

4.2.3.2 Rate Gyros. Angular rates of up to 60 deg/sec can occur on the hull, but the signal of interest is only in the plus and minus 5 deg/sec range. A 5 deg/sec sensor can be used, and the pickoff will be against its stops beyond this range. The characteristics of a rate gyro allow this to occur without degradation in performance.
4.2.3.3 Accelerometer. The electronic circuitry to obtain the vertical rate from the accelerometer is the critical part of this component. Very low drift requirements are necessary. Commercially available I.C. components are available, but additional circuitry is required to filter out high frequency components of the signal due to noise. Proportional feedback which has the effect of canceling out very low frequencies is also required to compensate for long term drift inherent in such a system.

4.2.3.4 Switching Logic. The local velocity of the hull at each damped wheel is required. Figure 31.0 shows the switching logic to be used.

4.2.3.5 Simplification. The complexity of the concept design is increased due to the need for examining the validity of the simplifications which the simulation study showed feasible. For instance, it is anticipated that only pitch rate will be needed ultimately to obtain most of the adaptive damping effect.

4.2.4 Performance Specifications

4.2.4.1 Appendix C gives the rate gyro specifications.

4.2.4.2 Accelerometer specification TBD.

4.2.4.3 Damper Valve. Figure 30.0 shows the damper valve design. This is an in-house design and manufacture. Additional data for design is contained in R-1649.

4.2.4.4 Electronics. TBD. This is in-house design and assembly.
4.2.4.5 Suspension Units. Existing NWL designed and manufactured units are to be used. See Section 5.0.

4.2.4.6 Solenoid Valve. This is a standard NWL Model 3785 unit.

4.2.5 Hardware Testing

4.2.5.1 Test Conditions Selection. The following conditions will be selected:

1) Jounce damping levels on the first and last wheel locations on both sides of the vehicle.
2) Removal of heave rate logic.
3) Removal of roll rate logic.
4) Spring rate modification to the #1 and #2 road wheel suspensions, by means of manual controls or, as an alternate, automatic control.
5) Damper valve damping pressure levels.

4.2.5.2 Test Instrumentation

1) A 15 channel tape recorder will be used to measure the following variables.
   a. Damper pressure levels (4).
   b. Pitch, roll and heave sensor rate signals, (3).
   c. Voice-over recording of test condition, (1).
   d. Solenoid logic signals (4).
   e. Suspension cylinder pressures (2).
   f. Vehicle speed.
5.2.5.3 Terrain Selection

1) Bump course
2) Cross-country terrain
3) Ditches

5.0 CONCLUSIONS

5.1 A pitch rate adaptive system can be expected to yield a 30% improvement in average pitch rate over a comparable non-adaptive system.

5.2 The addition of heave rate to the pitch rate signal shows only a marginal improvement between 15 and 25 mph. This control loop might become significant under the special operating conditions of undulating terrain being traversed at a speed which will excite the vertical resonance of the suspension system.

5.3 The most improvement of the pitch rate adaptive system can be expected from a vehicle with a soft suspension system.

5.4 The track tension device is so important to the characteristic behavior of the vehicle that it should be included as part of the suspension system design.

6.0 RECOMMENDATIONS

6.1 Pitch rate adaptive damping should be tried on an actual vehicle.

6.2 A vehicle with a soft suspension, preferably a hydro-pneumatic system, should be used.

6.3 Heave rate adaptive damping should be incorporated with the pitch rate damping in such a fashion that it may be switched on or off for comparative performance.
FIG. 1

VEHICLE SPEED - MPH

INCREASED CONTAMINATION

NOTE: NO. 7A HIGHT ARE MEAN-ALL
AVERAGES FOR EACH SERIES OF 3
SPEEDS.
FIG. 12.1

20 NOV 74: 10.40 - 11.40

21 NOV 74: 10.45 - 11.45

THERMOMETER

10.40/10.45

11.40/11.45
FIG. 19.2

Sensitivity Study

For this, the BPA P 7 or 800 mL/oz (1000 BPA)

Method: 2 in. 1/4 in. GPR, 100 psi, 10% chart

Break P 7 9 in. GPR, 100 psi, 10% chart
OBSOLUTE & COURSE DETAILS

SINE WAVE

6" 28'

OBSTACLES

6, 8, 10 or 12 inch obstacles

APPROACH

DEPARTURE

APPROACH

10" 12" 10" 10" 10" 10" 10" 12" 10"

30 FT 30 FT 60 FT 10 FT 10 FT 60 FT 60 FT 30 FT 30 FT

FIGURE 26
STANDARD HYdropneumatic SUSPENSION SYSTEM

CODE
1. BASIC DAMPER UNIT
2. INTERMEDIATE UNIT
3. SUSPENSION CONTROL MANIFOLD

FWD
ADAPTIVE CONTROL SYSTEM - AUSTERE

VARIABLE SPRING SOLENOID

VAR DAM

FWD

PITCH RATE GYRO

#1 VAR DAM

#2 F D

#3

#4

#5

#6 F D

SUSPENSION CONTROL MANIFOLD

ON/OFF SWITCH

ADAPTIVE CONTROL

F I G U R E 2 9 . 0
FIG. 30.0.

DAMPER VALVE-SOLENOID CONTROLLED
**SWITCH LOGIC**

**FIGURE 31.0**

**LEFT #1**

HIGH DAMPING IF:
\[ \varepsilon R + P - H > 1.0 \]

**RIGHT #1**

HIGH DAMPING IF:
\[ \varepsilon - R + P - H > 1.0 \]

**LEFT #6**

HIGH DAMPING IF:
\[ \varepsilon R - P - N > 1.0 \]

**RIGHT #6**

HIGH DAMPING IF:
\[ \varepsilon - R - P - N > 1.0 \]

NOTE: POWER OFF results in HIGH DAMPING levels on all DAMPED units.

\( R, P, N \) are ROLL, PITCH, and HEAVE RATES, RESPECTIVE TO APPROPRIATE LOGIC SWITCHING LEVELS.
APPENDIX A

SYSTEM EQUATIONS
**BASIC SYSTEM EQUATIONS**

**SYSTEM VARIABLES**

X<sub>i</sub> - ground contour height interpreted as the height of hub of undistorted i<sup>th</sup> wheel in contact with ground. See Fig. 92.

Y<sub>i</sub> - actual height of hub of i<sup>th</sup> wheel with respect to its datum plane. See Fig. 92.

Z<sub>i</sub> - horizontal position of hub of i<sup>th</sup> wheel. See Fig. 92.

θ<sub>i</sub> - angle of approach of track to i<sup>th</sup> wheel with respect to true horizontal, positive downward.

θ<sub>i</sub> - ground rise angle ahead of i<sup>th</sup> wheel relative to true horizontal. See Fig. 92.

θ<sub>i</sub> - angle of departure of track from i<sup>th</sup> wheel with respect to true horizontal, positive ascending.

θ<sub>Y</sub> - angle of i<sup>th</sup> road arm above hull horizontal.

θ<sub>F</sub> - angle of departure of track from front idler/sprocket referred to plane L to pitch plane and containing CG and front idler/sprocket axis.

θ<sub>R</sub> - angle of approach of track to rear idler/sprocket referred to plane L pitch plane and containing CG and rear idler/sprocket axis.

θ<sub>Y</sub> - angle of road arm below true horizontal (i<sup>th</sup>).

F<sub>i</sub> - ground contact normal force at i<sup>th</sup> wheel.

F<sub>f</sub> - ground contact tangential force at i<sup>th</sup> wheel.

F<sub>x</sub> - radial component of force between hull and i<sup>th</sup> road arm.

F<sub>y</sub> - perpendicular component of force between hull and i<sup>th</sup> road arm.

M<sub>i</sub> - torque between hull and i<sup>th</sup> suspension unit.

Y<sub>i</sub> - vertical position of hull C.G. with respect to datum.

Z<sub>i</sub> - horizontal position of hull C.G.

θ<sub>P</sub> - pitch angle of hull with respect to true horizontal, positive down in front.
**BASIC SYSTEM EQUATIONS**

**SYSTEM VARIABLES**

\[ D_i \] - LENGTH OF TRACK BETWEEN \( i^{th} \) WHEEL AND \((i+1)^{th}\) WHEEL

\[ D_F \] - LENGTH OF TRACK BETWEEN FRONT IDLER/SPROCKET AND FIRST WHEEL

\[ D_R \] - LENGTH OF TRACK BETWEEN REAR WHEEL AND REAR IDLER/SPROCKET

\[ h_i \] - AVERAGE HEIGHT OF GROUND CONTOUR BETWEEN \( i^{th} \) WHEEL AND THE \((i+1)^{th}\) WHEEL

\[ s_i \] - DIFFERENCE BETWEEN \( D_i \) AND NOMINAL LENGTH OF TRACK BETWEEN \( i^{th} \) WHEEL AND \((i+1)^{th}\) WHEEL

\[ T_{ai} \] - TENSION FORCE IN TRACK APPROACHING \( i^{th} \) WHEEL

\[ T_{oi} \] - TENSION FORCE IN TRACK DEPARTING \( i^{th} \) WHEEL

**NOTE:** C.G. USED IN ANY OF THE ABOVE REFERS TO THE CENTER OF GRAVITY OF THE SPRING MASS, OR THE NULL DATUM PLANES:

\[ y = 0 \] HULL LEVEL, ROAD ARMS IN NOMINAL STATIC POSITION (\( y_o \)), ALL WHEELS UNDISTORTED AND JUST IN CONTACT WITH LEVEL GROUND; ALL \( x_i \)'s AND \( y_i \)'s ARE ZERO.
BASIC SYSTEM EQUATIONS

SYSTEM CONSTANTS AND PARAMETERS

$C_0$ - suspension unit damping coefficient. This may be a function of velocity and damping torque level.

$K_3$ - ground-track-wheel interface spring rate. Not well defined. Use maximum acceptable value, possibly including some damping.

$q$ - torsional spring rate of suspension spring may vary with road arm angle.

$m$ - lumped mass of wheel and road arm at wheel hub.

$q$ - road arm length

$\theta_f$ - angle of road arm below hull horizontal at which suspension spring is fully loaded.

$\theta_0$ - nominal static angle of road arm below hull horizontal.

$A_f$ - distance, measured in pitch plane, of C.G. from front idler/sprocket axis.

$A_r$ - distance, measured in pitch plane, of C.G. from rear idler/sprocket axis.

$h$ - height of C.G. above plane of suspension unit axes.

$I$ - pitch moment of inertia of hull about its C.G. (sprung).

$L_i$ - distance measured in hull horizontal of $i^{th}$ suspension unit axis ahead of hull C.G.

$m$ - sprung mass of hull.

$\theta_i$ - angle, measured in pitch plane, between line from C.G. to $i^{th}$ suspension unit axis and hull horizontal. Sign same as for $x_i$.

$\theta_f$ - angle, measured in pitch plane, between line from C.G. to front idler/sprocket and hull horizontal.

$\theta_r$ - angle, measured in pitch plane, between line from C.G. to rear idler/sprocket and hull horizontal.
BASIC SYSTEM EQUATIONS

SYSTEM CONSTANTS AND PARAMETERS

$C_w$ - Weighting factor for determining track angles

$D_f$ - Nominal static track length between idler and first wheel

$D_b$ - Nominal static track length between front idler sprocket and first wheel

$D_p$ - Nominal static track length between rear idler/sprocket and last wheel

$K_T$ - Track effective spring rate in tension

$T_T$ - Nominal static track tension force

$h_T$ - Height of front sprocket or idler above the plane of suspension unit axes (tank-vertical dir.)

$h_R$ - Height of rear sprocket or idler above the plane of suspension unit axes (tank-vertical dir.)

$L_f$ - Distance of front idler or sprocket ahead of #1 suspension unit axis (tank-horizontal)

$L_R$ - Distance of rear sprocket or idler behind #6 suspension unit axis (tank-horizontal)
BASIC EQUATIONS

SUMMATION OF VERTICAL FORCES

\[ m \ddot{y}_i = F_i \cos \beta_i + T_{iN} \sin \lambda_i + T_{iD} \sin \alpha_i + F_{yi} \sin \beta_i - F_{yi} \sin \psi_i - F_{yi} \cos \psi_i - mg \]

SUMMATION OF HORIZONTAL FORCES

\[ m \ddot{x}_i = F_{wi} \sin \psi_i + T_{iN} \cos \lambda_i + F_{xi} \cos \beta_i - F_{xi} \cos \psi_i - T_{iD} \cos \alpha_i - F_i \sin \beta_i \]

SUMMATION OF MOMENTS ABOUT "O"

\[ M_i = R F_{xi} = \alpha (\beta_i + \phi) + C_0 \dot{\phi} \]
BASIC EQUATIONS
SUSPENSION UNITS

CONSTRAINTS

\[ \mu_i = \theta_i + \mu \]

\[ F_{m1} = T_{bi} - T_{ni} \]

GROUND INTERFACE

\[ F_i = \frac{K_i}{\cos \phi} [x_i \cdot \theta_i] \]
BASIC EQUATIONS

NULL

SUMMATION OF VERTICAL FORCES (1/2 NULL)

\[
\frac{\delta Y}{\delta z} = \sum F_{i z} \cos \theta_i + \sum F_{i y} \sin \theta_i - T_{ai} \sin (\theta_i + \delta_i) - \frac{M}{2}
\]

SUMMATION OF HORIZONTAL FORCES (1/2 NULL)

\[
\frac{\delta X}{\delta z} = \sum F_{i x} \cos \theta_i + \sum F_{i y} \sin \theta_i + T_{dc} \cos (\theta_i + \delta_i) - T_{ai} \cos (\theta_i + \delta_i + \theta_i)
\]

SUMMATION OF MOMENTS ABOUT C.G.

\[
\frac{\delta \theta}{\delta z} = \sum \delta_{i z} + T_{ai} R \sin \theta_i - T_{dc} R \sin \theta_i
\]

CONSTRANTS

\[
\delta_i = \tan^{-1} \left( \frac{A_i}{z_i} \right)
\]

\[
y = Y + R \left[ \sin \theta_i - \sin \theta_i \right] + L - \sqrt{z_i \sin \left( \sin (\theta_i + \delta_i) \right)}
\]

\[
\delta_i = \delta_i - R \left[ \cos \theta_i - \cos \theta_i \right] - L_i + \sqrt{z_i \sin \left( \cos (\theta_i + \delta_i) \right)}
\]

\[
T_{di} = E_{di} + \delta_i + \theta_i
\]

\[
T_{ai} = E_{ai} + \delta_i - \theta_i
\]
BASIC EQUATIONS

TRACK TENSION AND ITS APPLICATION

ASSUMPTIONS

1. TRACK IS ITSELF MASSLESS
2. TRACK IS COMPLETELY FLEXIBLE IN BENDING
3. TRACK IS INFINITELY STIFF IN TENSION
4. TRACK SPANS DEPRESSIONS BETWEEN WHEELS, AND CONFORMS TO RISES BETWEEN WHEELS
5. BECAUSE OF ITEM #3 ABOVE, TRACK TENSION WILL MODIFY THE POSITION OF THE SUSPENSION UNITS IN ORDER TO MAINTAIN AN ESSENTIALLY CONSTANT LENGTH TRACK. TRACK LENGTH BETWEEN WHEELS ONE AND SIX WILL BE APPROXIMATE BASED ON ITEM #4 ABOVE.

CHANGE IN TRACK LENGTH BETWEEN ADJACENT WHEELS:

CONDITION #1 - GROUND LEVEL BETWEEN WHEELS IS COMPLETELY BELOW THE TWO WHEELS. FOR THIS CONDITION THE DESIRED LENGTH OF TRACK WILL BE TAKEN AS THE HYPOTENUSE OF THE RIGHT TRIANGLE FORMED BY THE WHEELS HORIZONTAL AND VERTICAL SEPARATIONS. THIS CAN BE WRITTEN AS

\[ b_1 = \left\{ \left[ (x_1-x_{in})\cos\psi - R(\cos\psi; -\cos\psi_{in}) \right]^2 + [y_1-y_{in}]^2 \right\}^{1/2} \]

THE TRACK ANGLES RESULTING FROM THIS ARE

\[ \phi_1 = -\lambda_{in} = \sin^{-1} \left( \frac{2\psi_{in}-2\psi}{D_1} \right) \]

NOTE THAT THE AREA UNDER THE TANGENT LINE IS GREATER THAN THE AREA UNDER THE GROUND CONTOUR LINE.

\[
\frac{\pi}{2} \left( (L_i - L_{in}) \cos \theta - R (\cos \psi_i - \cos \psi_{in}) \right) \text{ at time } t
\]

is greater-than-or-equal-to \( \int_0^t \varepsilon(x_i - x_{in}) \, dt \)

USE CONDITION #1.

THE POSSIBILITY THAT THE ABOVE CRITERION BE SATISFIED WITH SOME OF GROUND CONTOUR ABOVE THE TANGENT LINE WILL BE IGNORED.
TRACK TENSION AND ITS APPLICATION

CONDITION 2

\[ L_i - L_{in} \]
\[ \Theta \]

\[ (L_i - L_{in}) \cos \Theta - R (\cos \phi_i - \cos \phi_{in}) \]

Note that the area under the tangent line is less than the area under the ground contour line.

If

\[ \frac{\pi}{2} \left[ (L_i - L_{in}) \cos \Theta - R (\cos \phi_i - \cos \phi_{in}) \right] \]

at time 't' is less than \( \int_{x_i}^{x_e} z(x - x_{in}) \) ac

Use condition 2

Again, conditions satisfying the above criterion but with some contour below tangent will be ignored
TRACK TENSION AND ITS APPLICATION

CONDITION #2 - GROUND LEVEL BETWEEN WHEELS IS ABOVE TANGENT BETWEEN WHEELS. FOR THIS CONDITION THE DESIRED LENGTH OF TRACK WILL BE TAKEN AS THE LENGTH OF THE GROUND CONTOUR BETWEEN WHEELS. THE VEHICLE IS ASSUMED TO HAVE STARTED FROM A NOMINAL STATIC POSITION.

\[
D_i = l_i - l_{i+1} + \int (\dot{s}_i - \dot{s}_{i+1}) dt
\]

\[
\dot{s}_i = \sqrt{\dot{x}_i^2 + \dot{z}_i^2} = \frac{2}{\tan \gamma_i + 1} \cos \beta_i
\]

\[
D_i = l_i - l_{i+1} + \int \frac{2}{\cos \beta_i - \cos \beta_{i+1}} dt
\]

THE TRACK ANGLES TO BE USED FOR THIS CONDITION DO NOT LEAD THEMSELVES TO READY CALCULATION. ASSUME THE RISE BETWEEN WHEELS TO HAVE AN EFFECTIVE HEIGHT

\[
H_i = \left[ \frac{y_i^2 + y_{i+1}^2 + \text{AREA UNDER CURVE} - \text{AREA UNDER TANGENT}}{(l_i - l_{i+1}) \cos \kappa_{i+1} - R \cos \beta_{i+1}} \right]
\]

\[
\tan^{-1} \gamma_i = \frac{y_i - y_{i+1}}{(l_i - l_{i+1}) \cos \kappa_{i+1} - R \cos \beta_{i+1}}
\]

\[
\tan \lambda_i = \frac{y_i - y_{i+1}}{(l_i - l_{i+1}) \cos \kappa_{i+1} - R \cos \beta_{i+1}}
\]

\[C_i \text{ is a factor dependant on the form of the ground contour between wheels. A value of 3 to 4 seems reasonable.}\]
TRACK TENSION AND ITS APPLICATION

GROUND CONTOUR FORM FACTOR $C_n$.

FOR CONVENIENCE, LET $\gamma_i = \gamma_{i+1} = 0$, statically.

1. TRIANGULAR CONTOUR, MIDWAY BETWEEN WHEELS

\[ H = \frac{A}{L_i - L_{i+1}} = \frac{2}{3} \]

\[ \tan^{-1} \mu_i = \tan^{-1} \mu_{i+1} = \frac{A}{L_i - L_{i+1}} = \frac{2H}{L_i - L_{i+1}} \]

\[ C_n = 2 \]

2. PARABOLIC CONTOUR, MIDWAY BETWEEN WHEELS

\[ H_i = \frac{A}{L_i - L_{i+1}} \]

\[ A = 2 \int_{-x_0}^{x_0} \left( \frac{4x^2}{a^2} \right) dx = 2 \left[ 2x - \frac{2x^3}{3a^2} \right]_0^{x_0} = \frac{4a x_0}{3} \]

\[ L_i - L_{i+1} \approx 2x_0 \]

\[ H_i \approx \frac{2a x_0}{3} \]

\[ \tan^{-1} \mu_i = \tan^{-1} \mu_{i+1} \approx \frac{x_0}{2} \text{ - shape of parabola at } x_0 \]

\[ C_n = 6 \]
TRACK TENSION AND ITS APPLICATION


\[
\text{FRONT} \quad D_F = \left[ (l_F - R \sin \theta_1)^2 + (l_F \cos \theta_1)^2 \right]^{\frac{1}{2}} = \left[ l_F^2 + l_2^2 + R^2 + 2R (l_F \cos \theta_1 - h_F \sin \theta_1) \right]^{\frac{1}{2}}
\]

\[
\text{REAR} \quad D_R = \left[ (l_R - R \sin \theta_2)^2 + (l_R - R \cos \theta_2)^2 \right]^{\frac{1}{2}} = \left[ l_R^2 + l_2^2 + R^2 - 2R (l_R \cos \theta_2 + h_R \sin \theta_2) \right]^{\frac{1}{2}}
\]

NOTICE THAT SIGN \( \theta \) IS \((-)\) FOR ROAD ARM ANGLES BELOW HORIZONTAL. \( \theta \) IS \((+)\).

THE CHANGES IN THESE LENGTHS ARE THE DIFFERENCES FROM NOMINAL STATIC VALUES:

\[
D_{D_0} = \left[ l_F^2 + l_2^2 + R^2 + 2R (l_F \cos \theta_0 + h_F \sin \theta_0) \right]^{\frac{1}{2}}
\]

\[
D_0 = \left[ l_R^2 + l_2^2 + R^2 - 2R (l_R \cos \theta_0 - h_R \sin \theta_0) \right]^{\frac{1}{2}}
\]
**Track Tension and Its Application**

**Total Change in Track Length**

\[
\Delta L = (D_n - D_0) + (D_r - D_w) + \sum_{i=1}^{n} (D_i - D_i^0)
\]

where \( D_i^0 = L_i - L_{i-1} \)

**Nominal Track Tension**

\[
T_{\text{nom}} = T_0 + K_T \Delta L
\]

where \( T_0 \) is static nominal track tension

It is possible to apply an approach similar to that used in considering the ground between wheels to begin to pick up track loading ahead of wheel \( \#1 \), and to carry it beyond wheel \( \#6 \). This would provide forces known to exist as wheel \( \#1 \) approaches sharply rising ground or as wheel \( \#6 \) leaves sharply falling ground. Equations describing this are not included.
APPENDIX
B
SYSTEM EQUATIONS AND COMPUTER DIAGRAMS
**BASIC EQUATIONS FOR 1ST WHEEL**

**Ground Wheel Interface (See Fig 1)**

\[ F_i = \frac{K_g}{\cos \alpha} (x_i - y_i) \]

- \( F_i \) = Force generated at wheel/ground contact
- \( K_g \) = Effective Ground/Wheel spring rate
- \( \alpha \) = Angle of ground with horizontal
- \( x_i \) = Position of hub of undistorted wheel above its datum
- \( y_i \) = Actual position of wheel hub above its datum

**Actual Wheel Hub Position (See Fig 2)**

\[ y_i = Y - L_i \sin (\Theta_0 + \psi_i) + R \sin \Theta_0 - R \sin \psi_i \]

- \( Y \) = Vertical Position of Vehicle C.G. above its datum, \( Y = h \) when road arms of wheels are in static position and wheels are just in ground contact but unloaded (zero pitch of wheels; \( h/2i = \sin \psi_i \))
- \( L_i \) = Distance of Road Arm pivot from Vehicle C.G. (+ for those ahead of C.G., - for those behind)
- \( \Theta_0 \) = Vehicle pitch angle measured from horizontal, positive with front end down
- \( R \) = Length of Road Arm
- \( \Theta_0 \) = Static Road Arm angle below plane of Road Arm pivots
- \( \psi_i \) = Actual angle of Road Arm below true horizontal
- \( \psi_i \) = Angle in pitch plane between line drawn from C.G. to road arm pivots and plane of road arm pivots, sign same as \( L_i \)
- \( h \) = Height of C.G. above plane of road arm pivot points.
Sketch for Equation Y

**Figure 1**
Sketch for Equation 2

Static Conditions, Wheel on Ground Reference

\[ \Theta_i = 0 \]
\[ y_i = \Theta_0 \]
\[ Y = h \]
\[ y = 0 \]

Figure 2
BASIC EQUATIONS FOR 4-WHEEL CONT'D

Road Arm Angle

\[ \psi_i = -\theta_i - \Theta_0 \]

\( \theta_i = \) Road Arm angle above plane of vehicle
\( \Theta_0 = \) Road Arm pivot

Torque Balance

\[ \eta \ddot{\psi}_i R = -mg \cos \psi_i + F_i \cos (\psi_i - \alpha) - \frac{1}{2} \left[ (M_i + M_\text{fr}) - \frac{F_i}{R} \right] \]

(Note: \( \theta_i \) is actually relative to the vehicle hull, so the angle \( \psi_i \) would appear to be preferred. Since the velocity \( \dot{\theta}_i \) is a desired quantity elsewhere in the simulation and since \( \dot{\psi}_i \) and \( \dot{\theta}_i \) are of much lower magnitude, this is felt to be a reasonable approximation.)

\( \eta \) = gravitational constant
\( m \) = lumped (effective) mass of wheel and road arm
\( R \) = Effective Radius for track tension force
\( F_i \) = Torque due to suspension springs and stops
\( M_i \) = Torque due to damping forces
\( F_{\text{fr}} \) = Track Tension Force

Spring Torque

\[ M_i = \alpha (\psi_i + \Theta_0) + \alpha_\text{fr} (\psi_i - \Theta_0) + \kappa_0 (\Theta_0 - \psi_i + \psi_i + \Theta_0) \]

\( \alpha \) = Primary torsional spring rate of suspension unit
\( \Theta_0 = \) Road Arm angle below plane of road arm pivot at which spring torque vanishes.
BASIC EQUATIONS, CONT'D, 1st WHEEL

sprinTorque, Cont'd

\[ \alpha_1 = \text{Secondary torsional spring rate of suspension unit} \]
\[ \theta_b = \text{Angle above plane of vehicle road arm pivot at which secondary spring is engaged.} \]
\[ K_a = \text{value used here to simulate stops.} \]
\[ \theta_j = \text{Angle above road arm pivot plane at which jounce stop is encountered.} \]
\[ \theta_r = \text{Angle below road arm pivot plane at which rebound stop is encountered.} \]
\[ a = \text{a for } a > 0, \text{0 for } a < 0, \text{a for } a = 0, \text{0 for } a = 0. \]

Dumper Torque

\[ M_{12} = F_s R \cos \theta_j \left( \frac{\theta_j}{\theta_j} \right) \]
\[ M_{12} \leq F_o R \cos \theta_b, C = 0 \]
\[ M_{12} \leq F_o R \cos \theta_b, C = 1 \]

\[ F_s = \text{Static Ground Force} \]
\[ F_o R \cos \theta_b = \text{Damping torque limit in jounce due to dumper valve operation (normal)} \]
\[ F_o R \cos \theta_b = \text{Switched damping torque limit} \]

(Note: This representation allows the "Damping Force" \( F_o \) or \( F_o' \) to be compared directly with the wheel nominal static load)

\[ \theta_b = \text{Jounce velocity which will develop rated damping pressure across bleed orifice} \]
\[ \theta_b' = \text{Rebound velocity required to develop rated rebound damping torque} \]
\[ C = 0 \text{ or } 1; \text{ logic signal from adaptive damping control.} \]

Effective Track Tension Radii

\[ R_i = -9.5 \left(1 - \frac{\theta_1}{90}\right) \left(1 - \frac{\theta_2 + 135}{90} + \frac{\theta_2 + 135}{90} \right), \text{wheel } i \]

\[ \theta_1 = \text{wheel } i \text{ road arm angle relative to hull} \]
\[ \theta_2 = \text{wheel } i \text{ road arm angle relative to hull} \]
Basic Equations Cont'd, 3rd Wheel

Effective Track Tension Radius

\[ R_i = \frac{1}{2} \left( \frac{y_i - 2y_i + y_i}{L} \right) \quad i = 2, 3, 5, 6, 0 \]

\[ L = \text{Distance Between Road Arm Pivots, Ave} \]

\[ R_6 = \gamma_6 \left( 2 + \frac{\alpha_5}{\eta} - \frac{\delta_5}{\eta} + \frac{\delta_6}{\eta} \right) \text{ wheel } #6 \]

\[ R_6 \leq 0 \]

Track Tension Force

\[ F_T = K_T (s_0 + s_1 + s_6) - \rho V_s^2 \quad F_T \geq 0 \]

\[ K_T = \text{Spring Rate of Track} \]

\[ s_0 = \text{initial stretch of track} \]

\[ s_1 = \text{stretch of track due to wheel } #1 \]

\[ s_6 = \text{stretch of track due to wheel } #6 \]

\[ \rho = \text{Track Density (mass/unit length)} \]

\[ V_s = \text{Vehicle Speed} \]

Track Stretch

\[ s_1 = -7.0 \left( 1 + \frac{\alpha_5}{7} \right) \left( 1 - 98 - \frac{49 + \theta_5}{625} \right) \]

\[ s_6 = -9.6 \left( 1 + \frac{\alpha_5}{7} \right) \left( 1 + 2 \frac{49 + \theta_5}{625} \right) \]

Note: Equations 35, 36, 37 based on curves shown on succeeding two pages. These curves in turn are based on geometry of the MICV-70 Test Rig. (Fig. 3, 4)
Figure 3

Wheel #1

Wheel at static position

Wheel at full bounce

Road arm angle from horizontal (degrees)
**BASIC EQUATIONS, Hull Dynamics**

**Pitch**

\[
\frac{1}{2} \dot{\theta}_i = \sum_{i=1}^{n} \left\{ F_i \left[ l_i \cos(\theta_i + \delta_i + \Theta_1) - R \cos(\psi_i - \beta_i) \right] + m R \dot{\theta}_i \left[ l_i \cos(\psi_i + \delta_i + \Theta_1) - R \right] + m R \dot{\theta}_i^2 \left[ l_i \sin(\psi_i + \delta_i + \Theta_1) \right] + mg \left[ l_i \cos(\delta_i + \Theta_1) - R \cos \psi_i \right] \right\}
\]

\( I = \text{Sprung Moment of Inertia about CG} \)

**Heave**

\[
\frac{1}{2} \dot{Y} = \sum_{i=1}^{n} \left\{ F_i \cos \beta_i - m R \dot{\theta}_i \cos \psi_i - m R \dot{\theta}_i^2 \sin \psi_i \right\} - G \frac{M}{2}
\]

\( M = \text{Vehicle Sprung Mass} \)

\( l_i \cos(\delta_i + \Theta_1) R \cos \psi_i \)
COMPUTER EQUATIONS

Cos A, in the ground force equation can be ignored in the
simulation since the value of N is small. Make the wheel
ground loop very fast, with respect to the other loops in the
problem so that the range of Cos A. will have negligible
effect at this point in the simulation, use Cos a = 1.

\[
\frac{20F_n}{4R} \left[ \frac{F_i}{2F_n} \right] = 100 \left[ \frac{K_e}{10F_n} \right] - 10 \left( \frac{\theta_i^*}{\theta_i} \right) - 10 \left( \sin \theta_0 \right) \left[ \frac{1}{10} \right]
\]

\( F_n = \) Normalizing factor, \( M_g = 9000 \) lb.

Using \( \sin (\theta_1 + \phi) = \sin \theta_1 \cos \phi + \cos \theta_1 \sin \phi \), and
noting that \( \theta_1 \) is small, \( \sin \theta_1 \approx \theta_1 \), \( \cos \theta_1 \approx 1 \),
\( \sin (\theta_1 + \phi) \approx \theta_1 \cos \phi + \sin \phi \).

Define \( y_1^* = y_i - R \sin \theta_0 \), \( y = y - l; s = y - h \)

\[
\left[ \frac{\theta_i}{\theta_n} \right] = \left[ \frac{y_i}{y_n} \right] - \sin \left( \frac{1}{10} \left( \frac{\theta_1}{\theta_n} \right) \right) - \left[ \sin y_i \right]
\]

\( \theta_0 = \) Normalizing factor = 90\(^\circ\) or 1\(^\circ\) radians

\[
10 \left[ \frac{\theta_i}{\theta_n} \right] = -10 \left[ \frac{\theta_i}{\theta_n} \right] - \left[ \frac{\theta_1}{\theta_n} \right] - \frac{100}{\theta_n} \quad \text{and}
\]

\[
\left[ \frac{\theta_i}{\theta_n} \right] = -\left[ \frac{\theta_i}{\theta_n} \right] - \frac{100}{\theta_n} - 3 \frac{1}{8} \left[ \frac{\theta_i}{\theta_n} \right]
\]

\[
\frac{\frac{m \theta_i}{\theta_n}}{2F_n} = 2 \left[ \frac{F_i \cos (y - y_i)}{2F_n} \right] + \frac{\theta_i}{\theta_n} \left[ \frac{F_i}{2F_n} \right]
\]

\[
-\left( \frac{\theta_i}{\theta_n} \right) \left[ \frac{F_i}{2F_n} \right] - \left( \frac{\theta_i}{\theta_n} \right) \left[ \frac{F_i}{2F_n} \right] - \frac{\theta_i}{\theta_n} \left[ \frac{F_i}{2F_n} \right]
\]

\( k \) = Multiplier gain factor = 3.586 = 1/3.586

\[
\left( \sqrt{\frac{m \frac{\theta_i}{\theta_n}}{2F_n}} \right) + \left( \sqrt{\frac{\theta_i}{\theta_n}} \right) \left( \frac{\theta_i}{\theta_n} \right) \left( \frac{\theta_i}{\theta_n} \right) = \frac{\theta_i}{\theta_n}, \quad \sqrt{\frac{m \theta_i}{\theta_n}} \leq \sqrt{\frac{\theta_i}{\theta_n}} \cos \phi \theta_0 \\
\left( \frac{1}{\theta_n} \right) \leq \sqrt{\frac{\theta_i}{\theta_n}} \cos \phi \theta_0 \quad \text{C = 0}
\]

\[
\left( \frac{1}{\theta_n} \right) \leq \sqrt{\frac{\theta_i}{\theta_n}} \cos \phi \theta_0 \quad \text{C = 1}
\]
COMPUTER EQUATIONS, CONT'D

6. \( \frac{d}{dt} \left[ \frac{\dot{\theta}_m}{\theta_m} \right] = 20 \left[ \frac{\dot{\theta}_m}{\theta_m} \right], \) setting \( \theta_m = \frac{\omega_1 \phi_{\text{tot}}}{2 \sqrt{R}} \)

Note \( \phi = t \) - zinc scale

7. \( \frac{d}{dt} \left[ \frac{\dot{\theta}_m}{\theta_m} \right] = 10 \left( \frac{\dot{\theta}_m}{\theta_m} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] \)

8. \( \left( \frac{F_m A}{R} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] = \left( \frac{F_m A}{R} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] + \left( \frac{1557}{A} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] - \left( \frac{C_k \dot{\theta}_m^2}{R} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] \)

\( \theta_m \) steps

9. \( \left( \frac{F_m}{k} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] = \left( \frac{9.67 + \frac{\dot{\theta}_m}{\theta_m}}{k} \right) \left[ \frac{9.67 - \frac{\dot{\theta}_m}{\theta_m}}{k} \right] \)

10. \( \left( \frac{\dot{\theta}_m}{\theta_m} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] = \left( \frac{9.67 + \frac{\dot{\theta}_m}{\theta_m}}{k} \right) \left[ \frac{9.67 - \frac{\dot{\theta}_m}{\theta_m}}{k} \right] \)

11. \( 10 \left( \frac{1}{10 R} \right) \left[ \frac{21i-129i + 150}{2} \right] = \frac{21i-129i + 150}{2} \) or

\( 10 \left( \frac{1}{10 R} \right) \left[ \frac{21i-129i + 150}{2} \right] = \frac{21i-129i + 150}{2} \) where \( i = 0 \) to 10, \( \{ 3 \} \)

12. \( k \left( \frac{\dot{\theta}_m}{\theta_m} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] = -\left( \frac{9 + \frac{\dot{\theta}_m}{\theta_m}}{k} \right) \left[ \frac{9 + \frac{\dot{\theta}_m}{\theta_m}}{k} \right] \) - 2.7 \( \frac{\dot{\theta}_m + 15}{\theta_m} \)

13. \( \left( \frac{21i-129i + 150}{2} \right) = 10 \left( \frac{126}{20} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] + \left( \frac{21i}{10} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] + \left( \frac{126}{20} \right) \left[ \frac{\dot{\theta}_m}{\theta_m} \right] \) - 10 \( \frac{21i + 15}{\theta_m} \)

*Note: In equation 11 the term involving ideal weight has been dropped. This is a minor effect.

In equation 15 the simulation approximates the truck tension radius by using the vertical component at static road-own angle.
**Computer Equations, Hull Dynamics**

The term \( \cos(\beta_i + \Theta_i) \) in equation 17 can be expanded and then simplified using the approximations \( \sin \Theta_i = \Theta_i \) and \( \cos \Theta_i = 1 \) for small values of \( \Theta_i \). This expression becomes \( \cos(\beta_i + \Theta_i) = \cos(\beta_i) - \Theta_i \sin(\beta_i) \). The angle \( \beta_i \) is \( 0^\circ \) on the level and \( \pm 30^\circ \) on an up ramp and \( \pm 60^\circ \) on a down ramp, so the angle functions can be switchable gains in the simulation. The presence of \( \Theta_i \) would normally require a multiplication, since \( F_i \) appears as a multiplier. \( |\Theta_i| \) however is less than 1, and on wheels \( 1 \) and \( 2 \) (and possibly \( 3 \)) where \( F_i \) can be large due to damping forces, the angle \( \Theta_i \) is of magnitude less than \( 45^\circ \), so that the term with the factor \( \Theta_i \) represents less than 10% of the other. We can further simplify by using \( \cos(\beta_i + \Theta_i) \approx \cos(\beta_i) - \Theta_i \sin(\beta_i) \).

\[
\frac{\text{d}^2 \cos(\beta_i + \Theta_i)}{\text{d} t^2} \approx \left( \frac{F_i}{2m} \cos(\beta_i) \right) \frac{F_i}{2m} \\
- \left( \frac{F_i}{2m} \cos(\beta_i) - \cos(\beta_i + \Theta_i) \sin(\beta_i) \right) \frac{F_i}{2m} \\
+ \left( \frac{F_i}{2m} \cos(\beta_i) - \cos(\beta_i + \Theta_i) \sin(\beta_i) \right) \frac{F_i}{2m}
\]

where \( \beta_i = \beta_0 \) when \( \text{sgn} \beta_i = (+) \)

\( \beta_i = -\beta_0 \) when \( \text{sgn} \beta_i = (-) \)

\( \beta_0 = 30^\circ \)

\( \frac{F_i}{2m} \) = Normalizing factor - use max. value of \( F_i \).
Computer Equations, Hull Dynamics, Cont'd

By the use of a simulation of the behavior of a single suspension unit it has been determined that the two terms in equation 12 which contain \( \phi_i \) derivatives and angle functions may be replaced by a single term containing \( \phi_i \) with no changes in effect due to hull or road-awn angles.

The term involving the weight of wheel and road-awn is of little significance, dynamically.

\[
\begin{align*}
&[\frac{I_{10i}}{yF_{m}a_{11}}] = \frac{6}{\pi} \left[ \frac{F_i}{2a_{10}} \frac{\cos(\phi_i + \theta_i)}{E_i} \right] \sum_{i=1}^{n} \left[ \frac{R_i}{2a_{10}} \frac{\cos(\theta_i)}{E_i} \right] \\
+ &\frac{6}{\pi} \left( \frac{E_{10i}}{4a_{10}} \right) \frac{mR_i}{E_i} \\
&[\frac{E_i}{2a_{10}} \cos(\phi_i - \theta_i)] = \left[ \frac{F_i}{E_i} \right] \cdot [\cos(\theta_i - \phi_i)] \\
&[\frac{1}{a_{10}} \frac{d}{dt} \left[ \frac{9 \dot{\theta}_i}{a_{10}} \right] = \left( \frac{16 \dot{E}_{10i}}{a_{10}^2} \right) \left[ \frac{I_{10i}}{F_{m}a_{11}} \right] \\
&\frac{d}{dt} \left[ \frac{\dot{\theta}_i}{a_{10}} \right] = 2 \left( \frac{1}{a_{10}} \right) \left[ \frac{4 \dot{E}_{10i}}{a_{10}^2} \right] \\
\end{align*}
\]

Similarly, the \( \phi_i \) and \( \phi_i^2 \) terms in the equation for vertical motion of the hull may be replaced with a single term involving \( \phi_i \) with constant coefficients in the simulations.

\[
\begin{align*}
\frac{F_i \cos \alpha_i}{2F_{m}a_{03}} &= \frac{F_i}{2a_{10}} + (\frac{1 - \cos \theta_i}{\cos \theta_i}) \frac{F_i}{2a_{10}} \left( 1 - \frac{\phi_i}{\theta_i} \right)
\end{align*}
\]
COMPUTER EQUATIONS, HULL DYNAMICS, CONT'D

20) \[ 10 \left( \frac{1}{2R_0} \right) \left[ \frac{\dot{y}}{R_0} \right] = \frac{c}{2} \left( \frac{F_i \cos \phi}{2F_i \cos \phi} \right) - \frac{c}{2} \left( \frac{A_{\text{emb}}}{2F_i} \right) \left[ \frac{m \dot{R}}{2F_i} \right] \]
   \[ - \left( \frac{c}{2F_i \cos \phi} \right) \left[ \frac{m g}{2F_i \cos \phi} \right] \left[ \frac{m g}{2F_i \cos \phi} \right] \]

21) \[ \frac{d}{dt} \left[ \frac{\dot{y}}{2R_0} \right] = 10 \left( \frac{F_{\text{net}} \tan \theta}{10R_0} \right) \left[ \frac{m \dot{y}}{4F_i} \right] \]

22) \[ \frac{d}{dt} \left[ \frac{\dot{y}}{R} \right] = 20 \left( \frac{10 \dot{\phi} \tan \theta}{20} \right) \left[ \frac{\dot{y}}{2R_0} \right] \]
The track tension force puts a load on the individual wheels which can be translated into a torque about the road-axle pivot points. Thus there can be said to be an effective radius, dependent upon wheel position and track angle which when multiplied by track tension will produce this torque as a product. Figures 3 and 4 contain plots of this radius against road-axle position (referred to full rebound) for wheels 1 and 2, respectively. Curves are shown for two positions of the adjacent wheels. The geometry used in deriving these curves was a mean between nominals for right and left hand suspension systems for the MCV-70 test rig.

Consider Figure 3. The effective radius shown here includes the effect of the track tension on the front idler, acting through the linkage to the front suspension unit. The appearance of the two curves shown suggests that the variation in track angle between wheels 1 and 2 due to the position of wheel 2 may be simulated by using wheel 2 position in a factor which decreases in size with increase in the wheel 2 road-axle angle.

The basic curve form on Figure 3 was taken in the approximation for simulation to be triangular. The apex occurs at approximately 65° above the rebound stop, which corresponds to 19° in the simulation. A value of 9.5 was used as an apex value for #2 wheel in static position. The factor involving #2 wheel position was given a value of 4.0 at static position and approximately 6.2 at full jounce. The slopes were chosen to approximate those of the curve for #2 wheel in static position. We have

\[ R_i = 9.5 \left(1.45 - \frac{0.1}{10}\right) \left(1 - \frac{0.23}{45^\circ} + \frac{0.047^\circ}{\theta}\right) \]
Now consider figure 4. These curves suggest that the effect of the track angle between wheels 5 and 6 is to shift the vertical location of the curve as a whole. For the simulation this shift was assumed to be linear with the position of wheel 5 or 6. As with wheel 4, the approximation substitutes a triangular form for the curves. The vertical location of the triangle is determined by the position of wheel 4 or 6. The apex of these triangles was set at 90° above the road or -30° in the simulation. A value of 4.5 for R₀ was taken at this location with wheel 4 at static position, and a value of 6.11 with wheel 6 at full jounce. Approximating the slopes, we have

\[ R_0 = 4.5 \left( 2 + \frac{\theta_{25}}{32} - \frac{\theta_{45} + 32}{32} + \frac{\theta_{65} + 32}{32} \right) \]

For the simulation a sign must be associated with these radii to give the correct sense to the torque.
Track Stretch due to Motion of Wheels 1 and 6

The same curves used to determine the effective radius for the track tension at wheels 1 and 6 may be used to develop a relationship between track stretch and the wheel positions. At any point \((\theta, R)\) on the curves, with \(\theta\) now expressed in radians, the amount the track has stretched from the \(\theta_0\) position may be expressed as

\[
\Delta s = \frac{1}{\theta_0} \int_{\theta_0}^{\theta} \sqrt{R^2 \cos^2 \theta - \frac{1}{k^2}} \, d\theta
\]

This gives the average ordinate over the interval.

The equations developed to linearize this relation for the simulation are:

\[
s_1 = 20 \left(1 + \frac{\theta}{75}\right) \left(1.75 - \frac{0.44 + \theta^2}{67.5}\right)
\]

\[
s_6 = -2.6 \left(1 + \frac{\theta}{75}\right) \left(1 + 2 \frac{0.44 + 0.5}{67}\right)
\]

where \(s_1\) and \(s_6\) changes in length from static conditions.
INERTIA FORCES ON A TRACK FREE OF GROUND CONTACT

Consider for this derivation an interior (that is neither a front nor a back) wheel of a tracked vehicle. Assume a vehicle forward speed \( v_3 \). Assume also a no-slip condition between ground and track, so that \( v_3 \) is also the track speed relative to the hull.

The downward velocity of points on the track ahead of the \( i \)th wheel and out of wheel contact is \( v_3 \sin \beta_i \). The upward velocity of points just behind the \( i \)th wheel and out of wheel contact is \( v_3 \sin \beta_i \). The change in vertical velocity is then \( v_3 (\sin \beta_i - \sin \beta_i) \). The rate of momentum change going around the \( i \)th wheel is therefore \( \rho v_3 (\sin \beta_i + \sin \beta_i) \) where \( \rho \) is equal to \( \rho v_3 \) and \( \rho \) is the mass density/unit length of track. This gives \( \rho v_3 (\sin \beta_i + \sin \beta_i) \) as the force required to effect this change of momentum.

\[
\sin \beta_i = \frac{v_3}{(L - q_i - q_j)^2}, \quad \sin \beta_i = \sqrt{\frac{L^2 - (q_i - q_j)^2}{L^2 (q_i - q_j)^2}}
\]

If \( q_i, q_j, q_k \) are small with respect to \( L \), we can approximate this by substituting the tangent and write for the momentum force:

\[
F_i = \rho v_3 \left[ \frac{q_i - q_j}{L} \right]
\]
Track Tension Forces on Interior Wheels

For the configuration shown above, the vertical component of the track tension force \( T_f \) acting on the wheel can be seen to be

\[
F_v = T_f (\sin \beta + \sin \gamma)
\]

This can be seen to be of the same form as that for the track momentum (inertia) force, so that the momentum force can be interpreted as a modification of the effect of the track tension on any wheel.

For the interior wheels for \( y_i - y_j \) and \( y_j - y_i \) small with respect to \( L \) (ref. previous page), we can substitute the tangent for the sine.

\[
F_v = T_f \left( \frac{y_i - y_j}{L} \right)
\]
Refracture Capacity for Application of Track Tension at Wheel #4, for Various Positions of #5 Track-Arm.
Effective Track Tension Radius
**Circuit Variations Used**

**Adaptive Damping (Pitch Rate)**

**Wheel #1**

For Pitch Down Rates Greater than $\frac{\text{L}}{400}$, the Light Damping Limit is Shut Off.

Note: Pitch-Up $\rightarrow$ $(\text{IN})$ has a Negative Sign. Pitch-Down $\rightarrow$ $(\text{IN})$ has a Positive Sign.

**Wheel #6**

The arrangement is similar to the above, except:

For Pitch Up Rates Greater than $\frac{\text{L}}{400}$, the Light Damping Limit is Shut Off.
CIRCUIT VARIATIONS USED

Adaptive Damping (Pitch Rate + Heave Rate)

The circuits used to provide adaptive damping using heave rate in addition to pitch rate were the same as for pitch rate alone, except that heave rate was summed with pitch rate. In the case of wheel #1 the normal sign of heave rate was used (heave rate = \( \dot{y} \)), and is positive upward, so that a rising hull tended to keep the damping at a low level. The same requirement exists for wheel #6, but because of the difference in the use of the comparator, the inverse (-) sign of \( \dot{y} \) had to be used.

Heave rate was summed with pitch rate at various levels and thresholds (\( \text{V}_{\text{nap}} \) and \( \text{V}_{\text{nap}}^{*} \)). Recorded data was run with

\[
\frac{30}{20} \leq \frac{\dot{y}}{\dot{\phi}} \leq \frac{120}{100} \quad \text{\text{V}_{\text{nap}}} \quad \text{switching off low limit damping on wheel #1 and}
\]

\[
-\left[\frac{30}{20} \leq \frac{\dot{y}}{\dot{\phi}} + \frac{120}{100}\right] \quad \text{\text{V}_{\text{nap}}^{*}} \quad \text{switching off low limit damping on wheel #6}
\]

\( \text{cw} = \text{wheel base} = 164.5 \text{ in.} \)
\( \text{V}_{\text{nap}} = \text{V}_{\text{nap}}^{*} = 0 \) seemed to be the best choice.
CIRCUIT VARIATIONS USED

Ground/Track/Road Wheel Interface Damping

\[
\begin{align*}
10 \frac{\dot{\theta}_i}{\theta_i} - 100 \frac{\dot{\theta}_i}{\theta_i} + 10 \sin \theta_i &= \left[ 3.5 \left( 1 + \frac{0.0185}{\frac{1}{\omega}} \right) \right] \frac{F_i}{2F_m} \\
\text{Equation:} \\
\end{align*}
\]

Ignoring the constant part of the equation and defining \( \Delta \chi = x_i - y_i \), we have

\[
\Delta F_i = \frac{20F_m}{3.5} \left[ \frac{1 + 0.0185}{1 + 0.0015} \right] \Delta \chi
\]

The form desired for damping is

\[
\Delta F_i = K_g \Delta x_i + C_g \Delta y_i
\]

Considering only the lead term:

\[
K_g = \frac{20F_m}{3.5} = \frac{20(40000)}{3.5} = \frac{800000}{3.5} = 96571.43 \text{ N/m}
\]

\[
C_g = \frac{30(40000)(0.025)}{3.5} = \frac{1200000}{3.5} = 128571.43 \text{ N/m}
\]

Damping Ratio (ignoring torsion bar)

\[
\zeta = \frac{C_g}{2K_g} = \frac{128571.43}{2 \times 96571.43} = 0.735
\]
PARAMETER AND NORMALIZING FACTOR VALUES

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>1000 in.-lb/deg, nom. 75, 300, 1100, 2000 used</td>
</tr>
<tr>
<td>$\beta_0$</td>
<td>90 deg.</td>
</tr>
<tr>
<td>$\beta_1$</td>
<td>6.65 deg.</td>
</tr>
<tr>
<td>$\beta_2$</td>
<td>10.91 deg.</td>
</tr>
<tr>
<td>$\beta_3$</td>
<td>26.39 deg.</td>
</tr>
<tr>
<td>$\beta_4$</td>
<td>-32.97 deg.</td>
</tr>
<tr>
<td>$\beta_5$</td>
<td>-12.91 deg.</td>
</tr>
<tr>
<td>$\beta_6$</td>
<td>-7.14 deg.</td>
</tr>
<tr>
<td>$\varphi_0$</td>
<td>Various values</td>
</tr>
<tr>
<td>$\varphi_1$</td>
<td>Various values</td>
</tr>
<tr>
<td>$\varphi_2$</td>
<td>4000 lb.</td>
</tr>
<tr>
<td>$\varphi_3$</td>
<td>3533.5 lb.</td>
</tr>
<tr>
<td>$\varphi_4$</td>
<td>886 in./sec²</td>
</tr>
<tr>
<td>$\varphi_5$</td>
<td>10.1 in.</td>
</tr>
<tr>
<td>$\varphi_6$</td>
<td>58100 in. lb. sec²</td>
</tr>
<tr>
<td>$\varphi_7$</td>
<td>25.56</td>
</tr>
<tr>
<td>$K_g$</td>
<td>9697 lb/in. Also used 5000, 17777 lb/in.</td>
</tr>
<tr>
<td>$K_T$</td>
<td>3000 lb/in.</td>
</tr>
<tr>
<td>$L_1$</td>
<td>87.1879 in.</td>
</tr>
<tr>
<td>$L_2$</td>
<td>54.294 in.</td>
</tr>
<tr>
<td>$L_3$</td>
<td>22.725 in.</td>
</tr>
<tr>
<td>$L_4$</td>
<td>-16.375 in.</td>
</tr>
<tr>
<td>$L_5$</td>
<td>96.377 in.</td>
</tr>
<tr>
<td>$L_6$</td>
<td>-82.795 in.</td>
</tr>
<tr>
<td>$L_7$</td>
<td>83 in.</td>
</tr>
<tr>
<td>$L_8$</td>
<td>82.1979 in.</td>
</tr>
<tr>
<td>$E$</td>
<td>0.1 lb sec²/in.</td>
</tr>
<tr>
<td>$M$</td>
<td>103.6 lb sec²/in.</td>
</tr>
<tr>
<td>$\rho$</td>
<td>70 lb/ft³</td>
</tr>
<tr>
<td>$K$</td>
<td>15 in.</td>
</tr>
<tr>
<td>$S_0$</td>
<td>Adjusted to level tank at reference speed</td>
</tr>
<tr>
<td>$78°$</td>
<td>Nom. depends on $\alpha, \beta, M, R$</td>
</tr>
<tr>
<td>$\beta_0$</td>
<td>21 deg.</td>
</tr>
<tr>
<td>$\beta_1$</td>
<td>78 deg.</td>
</tr>
</tbody>
</table>
PARAMETER AND NORMALIZING FACTOR VALUES

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \theta_1 )</td>
<td>90 deg. (% rad.)</td>
</tr>
<tr>
<td>( \theta_2 )</td>
<td>44 deg. nom., also 39, 99 deg.</td>
</tr>
<tr>
<td>( \theta_3 )</td>
<td>0.322 rad/sec (.05 in. orifice)</td>
</tr>
<tr>
<td>( \theta_4 )</td>
<td>17.65 rad/sec (.368 in. orifice)</td>
</tr>
<tr>
<td>( \theta_5 )</td>
<td>68.904 rad/sec.</td>
</tr>
<tr>
<td>( \theta_0 )</td>
<td>1</td>
</tr>
<tr>
<td>( V_0 )</td>
<td>5, 10, 15, 20, 25, 30, 35 MPH, etc.</td>
</tr>
</tbody>
</table>

**DAMPERS:**

\[
\tau_{da} = F \cos \theta_0 \\
A = 7.95 \text{ in.}^2 \text{ VANE AREA} \\
R = 3.173 \text{ IN. EFFECTIVE RADIUS} \\
\dot{\theta}_{ar} = 100 A_0 \sqrt{\frac{F}{R}} \text{ PE LIM.} \\
A_0 = \frac{\pi}{4} (D_0)^2 \text{ ORIFICE AREA} \\
D_0 = 0.05 \text{ in. (also 0.057, 0.099 in. BLEED 0.068 in. (also 0.077 in.) REDUNDANT ORIFICE}}
\]

HULL INPUT APPROXIMATION FOR \( \ddot{\theta}_1 \) AND \( \ddot{\theta}_f \):

| \( C_1 \) | 1.0 |
| \( C_2 \) | 0.944 |
| \( C_3 \) | 0.788 |
| \( C_4 \) | 0.742 |
| \( C_5 \) | 0.776 |
| \( C_6 \) | 0.726 |
| \( D_1 \) | 75.081 in. |
| \( D_2 \) | 22.571 in. |
| \( D_3 \) | 42.571 in. |
| \( D_4 \) | 23.679 in. |
| \( D_5 \) | 56.679 in. |
| \( D_6 \) | 92.679 in. |
APPENDIX C

RATEGYRO SPECIFICATIONS
NWL MODEL 925064
## RATE GYRO SPECIFICATIONS
**NWL Model No. 925064**

Prepared by:  
Checked by:  
Approved by:  

**Released:** Feb 5 73

### REVISIONS

<table>
<thead>
<tr>
<th>LTR</th>
<th>BY - DATE</th>
<th>PARA. OR PAGES</th>
<th>DESCRIPTION OF CHANGE</th>
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<tbody>
<tr>
<td>A</td>
<td>Taylor</td>
<td>8.3</td>
<td>86 ma was 75 ma; 17 ma was 15 ma.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>9.7</td>
<td>JVDC was 1:VDC</td>
</tr>
<tr>
<td></td>
<td></td>
<td>12.2</td>
<td>0.075% max was 0.05%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>15.0</td>
<td>Was &quot;Alignment of gyro in mount&quot;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>18.1</td>
<td>±70 deg/sec was ±10 deg/sec.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8.5 &amp; 8.6</td>
<td>Add winding</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8.10</td>
<td>Add (NWL to supply capacitor)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Per ECO 62039</td>
</tr>
<tr>
<td>B</td>
<td>Taylor</td>
<td>1.0</td>
<td>Was Type: GR-G5-AH7 IS: GR-G5A-1.74N</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8.6</td>
<td>Was 1.375 + j 6780 IS: 625 + j 3550</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8.10</td>
<td>Was TBD IS: 0.22 ± 10% capacitor</td>
</tr>
</tbody>
</table>

**Per ECO 63148**

**Revision Letter of this Specification Must Appear on Its Applicable Source Control Drawing**
1.0 This specification covers Northrop Type GR-G5A-1.74N gyro.

2.0 TEMPERATURE RANGE
2.1 Operating: -25°F to +165°F
2.2 Storage: -70°F to +165°F

3.0 NOMINAL RATE RANGE: ±100°/sec

4.0 STOPS SET AT: 100 to 120°/sec

5.0 OVER-RANGE: 500°/sec with no change in characteristics

6.0 NATURAL FREQUENCY: 55Hz nominal

7.0 SCALE FACTOR
7.1 At room temperature, measured at ±10°/sec: 54 to 60 mV/°/sec
7.2 Scale factor change with temperature: 0.02%/°F

8.0 PICKOFF
8.1 Excitation Voltage: 16 VRMS
8.2 Excitation Frequency: 5000 Hz
8.3 Excitation Current: 86 mA nominal untuned, 17 mA nominal after tuning
8.4 Series Choke: None required
8.5 Primary winding impedance at room temperature: 41 + j 189 nominal
8.6 Secondary winding impedance at room temperature: 625 + j 3550 nominal

8.7 Phase Angle
8.7.1 At room temperature: ±5°
8.7.2 Over operating temperature range: ±8°
8.8 Padding: As required
8.9 Load: 10,000 ohms in parallel with 1000 pf capacitor
8.10 Power Factor: 0.9 minimum with a 0.22 ±10% MFD capacitor connected across the primary (NNL to supply capacitor).

9.0 MOTOR

9.1 Excitation Voltage: 26 V RMS, single phase

9.2 Excitation Frequency: 400 Hz

9.3 Excitation Waveform: Non-sinusoidal; QUASI square Wave

9.4 Power Plan

9.4.1 Starting: 3.5 watts maximum

9.4.2 Running: 3.0 watts maximum

9.5 Excitation Current

9.5.1 Starting: 125 ma maximum

9.5.2 Running: 110 ma maximum

9.6 Power Factor: Not less than 0.9

9.7 Phase splitting capacitor (to be supplied by NNL): 0.75 MFD at 100 WVDC

9.8 Synchronization Time

9.8.1 At room temperature: 30 seconds maximum

9.8.2 Over the operating temperature range: 30 seconds maximum.

10.0 Damping Ratio over the operating temperature range: 0.5 to 1.0

11.0 MASS UNBALANCE: 0.05 °/sec/g maximum

12.0 HYSTERESIS

12.1 Definition: Hysteresis shall be calculated as the total width of the hysteresis loop at its widest point divided by total rate input used in generating the complete loop. For example, if a hysteresis loop is generated by operating the gyro first at 100°/sec CW then at 100°/sec CCW and it is found that the widest width of the loop is 4 mv while the outputs at 100°/sec are 5.98 and 5.92 volts the hysteresis shall then be
calculated as:

\[ \text{HYS} = \frac{(4)(10^{-2})}{5.98 + 5.92} (100) = 0.034\% \]

12.2 Value: 0.075% maximum

13.0 Zero Offset with Output axis Up

13.1 At room temperature: ±0.2°/sec maximum

13.2 Shift over the operating temperature range: ±0.3°/sec maximum

14.0 AC NULL VOLTAGE

14.1 Total Null Voltage: 100 MVRMS maximum

14.2 Quadrature Null: TBD maximum

15.0 ALIGNMENT OF GYRO INPUT AXIS IN MOUNT

15.1 Error: ±0.1 degrees maximum

15.2 Orientation: As shown in NUL drawing 925064

16.0 THRESHOLD: 0.01 deg/sec maximum

17.0 RESOLUTION: 0.01 deg/sec maximum

18.0 LINEARITY

18.1 Definition: The linearity error is defined as the difference between the measured output at any rate and the output as indicated by a straight line through the ±70 deg/sec points.

18.2 Value: 0.5% of full scale plus 0.5% of the reading.

19.0 SELF TEST CHARACTERISTICS: No self test capabilities are required

20.0 DIELECTRIC STRENGTH

20.1 Once only: 250 VRMS, 60 Hz

20.2 Repeated: 150 VRMS, 60 Hz
21.0 **OUTLINE CONFIGURATION**: As shown in NWL drawing 925064

22.0 **VIBRATION**

22.1 Type: Random

22.2 Bandwidth: 1200 Hz (20 to 1200 Hz)

22.3 Density: 0.006 g²/Hz

22.4 Amplitude: 3.8 g's peak (2.68 g's RMS) nominal

22.5 **Gyro Error**

22.5.1 During Vibration: Gyro to operate within spec.

22.5.2 After Vibration: No damage

23.0 **ACCELERATION**: These requirements are TBD

24.0 **SHOCK**

24.1 One-half sine, 40 g's peak, 45 MS: No damage after repeated exposure

24.2 Triangular, 10.5 g's peak, 100 MS: No damage after repeated exposure

25.0 **EMI**: MIL-STD-461A, Notice 4

26.0 **MODULATION NOISE**

26.1 0 to 1.4 Hz: Maximum allowable modulation noise increases linearly with frequency from zero at zero frequency to 0.05 deg/sec at 1.4 Hz.

26.2 1.4 to 20 Hz: 0.05 deg/sec maximum

26.3 20 to 100 Hz: 0.15 deg/sec maximum

27.0 **LIFE**: 1000 hours of operation minimum
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<th>Address</th>
</tr>
</thead>
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Feasibility Analysis and Evaluation of an Adaptive
Trackad Vehicle Suspension and Control System

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Abstract
This study shows that adaptive control of the bounce damping characteristics of
the first and last wheel of a tracked vehicle can cause a significant improve-
ment in performance. This improvement resulted in an overall 30 percent reduc-
tion in average pitching rate of the hull, as measured on the simulation of the
MVC vehicle traversing the JHA bump course.

Verification testing of the computer model with actual performance data of the
HCNV vehicle showed good correlation of peak amplitudes and hull resonance. This data also confirmed that the actual dampers are working well below recommended levels.

A proposed method of mechanizing and testing the adaptive control on an actual vehicle is presented along with system schematics and preliminary performance specifications for the critical components.