Analysis of the Suspension System of the M47 Tank by Means of Simulation Techniques

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Contract DA-20-018-ORD-12087
Sponsored by The Detroit Arsenal, Centerline, Michigan

2023-2-T June 1954

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Ypsilanti, Michigan

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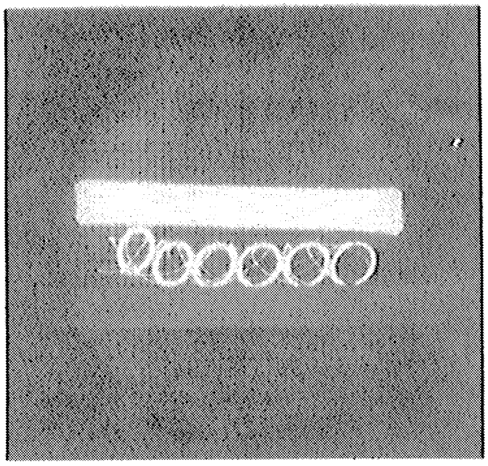
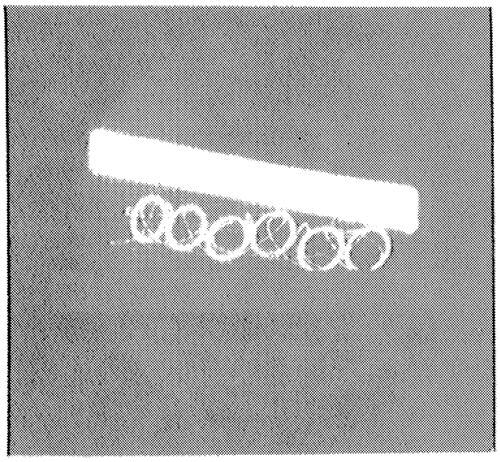
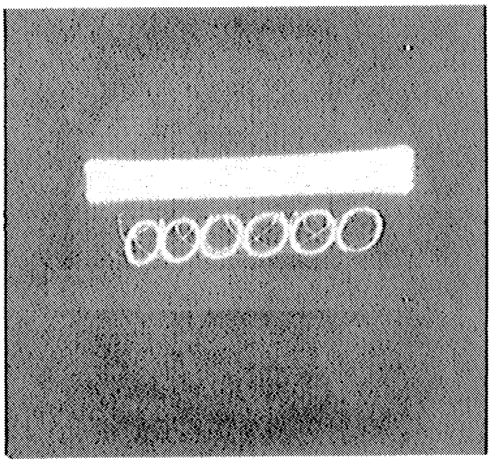


A. First wheel on bump



B. Fourth wheel on bump

← DIRECTION OF MOTION



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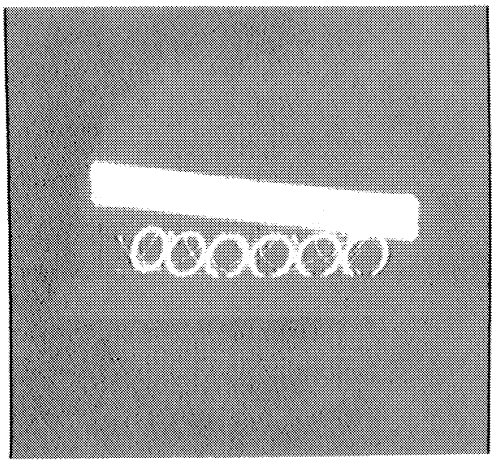

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DISPLAY OF SIMULATED TANK GOING OVER A 12-INCH-SQUARE BUMP AT 20 FEET PER SECOND

I

INTRODUCTION AND SUMMARY

In the past, engineers have designed the suspension system of a military tank by using rough estimating procedures based on the performance of earlier models, and then have tried out this design on a pilot model. Frequently the results obtained were not particularly satisfactory, and it was necessary to modify the design at considerable expense in time and money until a suitable system was obtained. The same end result can be achieved and much of the trial and error procedure can be eliminated by simulating the proposed suspension system on an electronic computer. Such a simulation allows system performance to be predicted accurately, and since the design features of the system and the values of the design parameters can easily be varied on the simulator, an optimum suspension design can be selected quickly and efficiently.

The Willow Run Research Center, under contract DA-20-018-ORD-12087 with the Detroit Arsenal, is engaged in a research program for developing the analytic basis for suspension system simulation. This report describes the work which has been done in this field thus far. Particular emphasis is placed on explaining the technique of setting up a specific problem on a computer and obtaining design data from it.

Before electronic simulation techniques can be used to study the performance of tank suspension systems, it is necessary to be sure that these techniques are capable of representing such systems accurately. For this reason, a program of comparing the response of a simulated tank with that of a real tank was the first step in the research program (Sec. II). This correlation program gave assurance that simulation techniques could provide accurate information on the performance of proposed suspension systems, thereby eliminating the necessity of obtaining these data from pilot models. It also aided in the subsequent analysis of the suspension system dynamics.

The tank for which experimental and simulated data were correlated was an M47, a medium tank weighing 48 tons. A successful comparison was achieved between the real and simulated response. The suspension systems of different types of tanks are considered sufficiently similar that the correlation program which was carried out provides the necessary data and methods required to extend the simulation to different tank designs.

Using the suspension system simulation which was developed for the correlation program, a study was made of the M47 tank to determine its performance under different conditions of tank speed and road profile and for certain variations of design features and parameter values. Specifically, the torsion bar and shock absorber constants were varied, and in one series of runs dry-friction snubbers were substituted for hydraulic shock absorbers. The results of this study are described in Section III. It was found that increasing the stiffness of the torsion bars by 50 per cent over the value of those on existing production units reduced the tendency for the suspension system to bottom when traversing a road obstacle 12 inches high. System performance was improved by reducing peak values of hull displacement and acceleration in both pitch and bounce. A comparison of the hydraulic shock absorber used at present with both weaker and stronger units indicated that the present units give approximately optimum performance. It was also learned that dry-friction snubbers rated at 3100 pounds had a performance close to that of hydraulic shock absorbers having half the strength of the production units. This study of the M47 system serves as a good illustration of the kinds of performance data which can be obtained from a suspension system simulation.

In Section IV, conclusions are drawn concerning the present status and future trends of suspension system simulation. Although good progress has already been made, much remains to be done. Necessary test data on tank components must be made available, basic analysis of suspension mechanisms must continue, and quantitative standards on such characteristics as road profiles and ride behavior must be adopted.

In this report, an understanding of analog computer simulation techniques is assumed. This subject is treated in References 1, 2, and 3. Its application to vehicle design problems is discussed in References 4, 5, 7, and 8.

The authors wish to acknowledge the contribution of Henry T. Nay in the development of the visual display of the simulation described in Appendix D, and of Roger A. Gaskill in supplying a method for simulating the effect of wheels leaving the road surface.

II

CORRELATION OF SIMULATED AND EXPERIMENTAL DATA

2.1 DESCRIPTION OF M47 SUSPENSION SYSTEM

Depending on the requirements of specific designs, the layout of a suspension system may vary considerably from one type of track-laying vehicle to another. The features peculiar to the M47 suspension system are shown in Figure 1. The weight of the tank is supported on twelve road wheels, six on each side, which are attached to the hull through road arms and torsion bars; the torsion bars have the effect of springs. To limit the maximum upward displacement of each road wheel with respect to the hull, a volute spring (or "bump stop") is installed which acts in parallel with the spring effect of the torsion bar after the road arm moves up through a certain free distance. This volute spring adds to the restoring force of the torsion bar; when fully depressed, it provides a solid stop for the road arm and prevents further appreciable deflection. Shock absorbers, attached to road wheels 1, 2, 5, and 6, damp out motion of the hull. The road wheels on each side of the tank ride on an endless track, which completes its circuit by passing over the drive sprocket at the rear of the tank, three support rollers along the side of the hull, and an adjusting idler wheel at the front of the tank. Track guides are used on each track block to prevent throwing of the track. The drive sprocket provides the means for applying power from the engine to produce tank motion. The adjusting idler wheel at the front of the tank is linked to the front road wheel in such a manner that when the front road wheel is raised, the total length of the track's path (and hence the track tension) remains relatively unchanged. A spring-loaded compensating idler wheel behind the sixth road wheel provides an additional means of maintaining track tension. Each track is installed in such a way that it is normally under 6000 pounds in tension.

2.2 FIELD TESTS

To obtain the necessary data for developing an accurate simulation of the suspension system, field tests were run on an M47 tank. These field tests consisted of running the M47 over two types of road obstacles

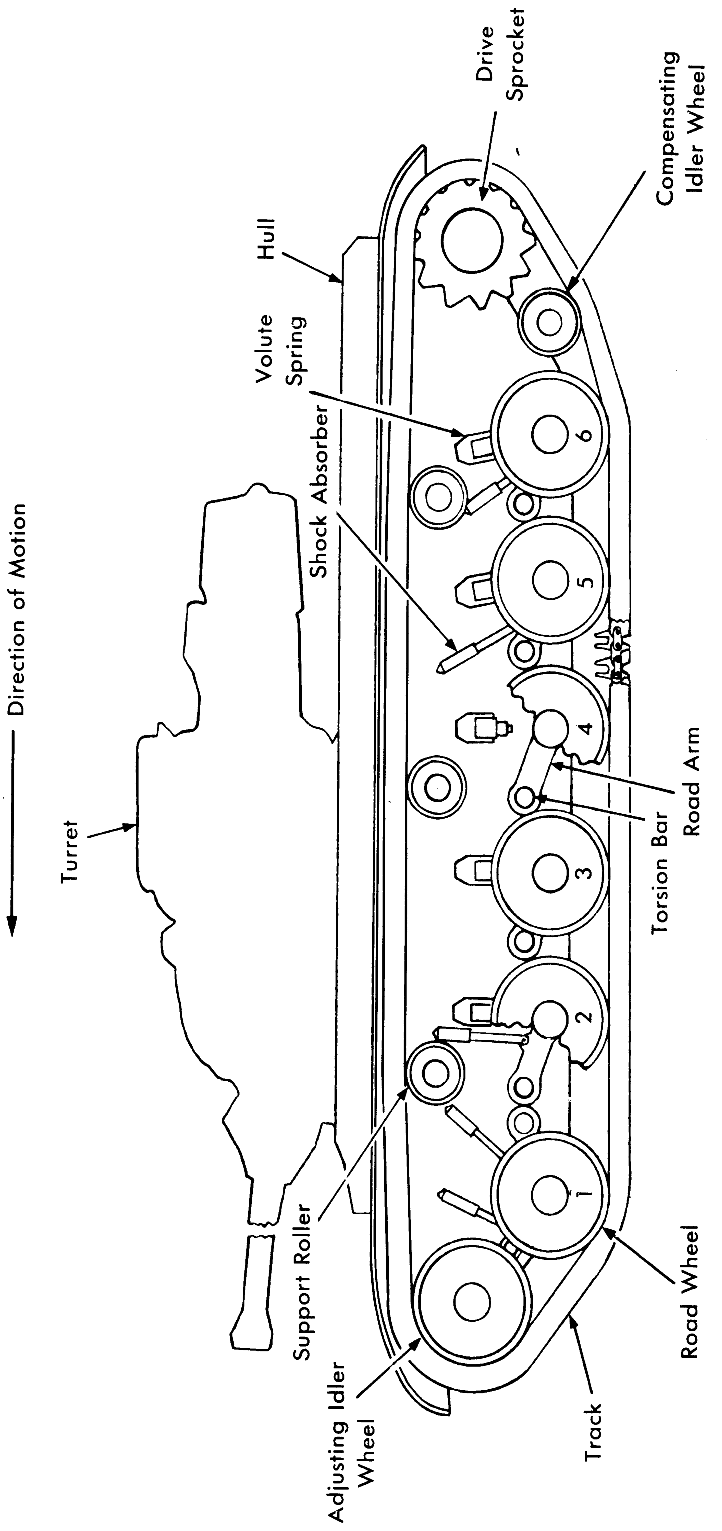


FIG. 1 FEATURES OF M47 SUSPENSION SYSTEM

and recording the pertinent data on hull and wheel motion. A 12-inch-square oak timber (referred to hereafter as a 12-inch-square) and an inverted-V wood ramp 8 inches high were used as road obstacles (Fig. 2). For each type of road obstacle, runs were made at various forward speeds of the tank, ranging from 5.3 to 27.0 feet per second. The hull pitch angle was recorded by means of a gyro and the hull vertical acceleration was recorded by means of an accelerometer. Motion pictures were taken from which the pitch angle and vertical displacement of the hull and the vertical displacement of each of the road wheels could be measured (Fig. 3).

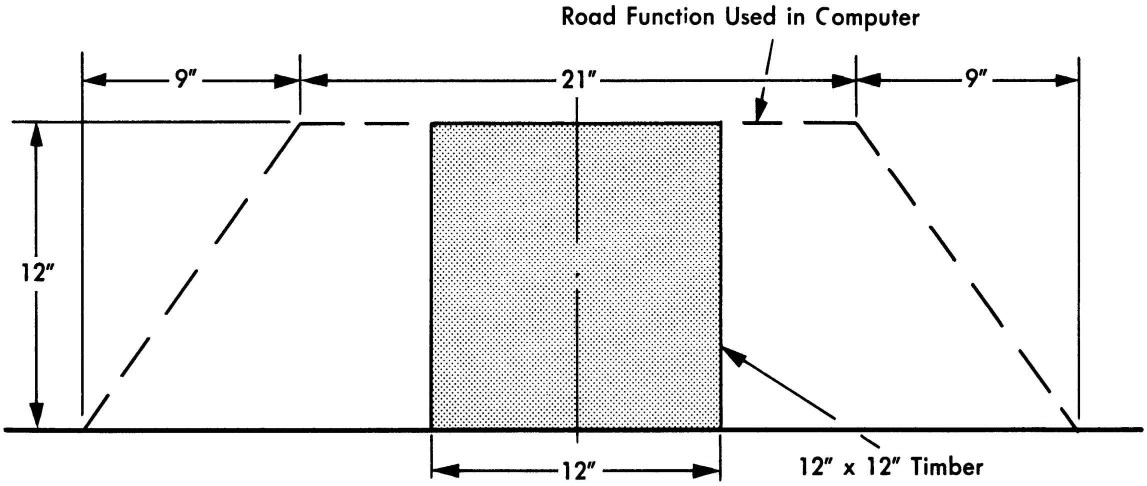
The steady-state position of the hull and each road wheel was also directly measured at various positions of the tank in relation to the road obstacles to check the computer solutions for simplified reproducible conditions.

The instrumentation techniques used are described in detail in Appendix A.

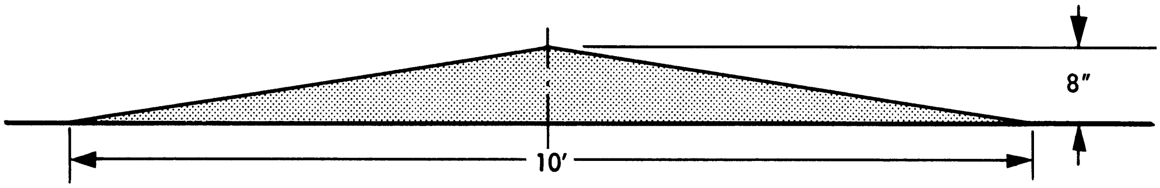
2.3 BASIC RELATIONS USED IN THE SIMULATION

Information obtained from the Detroit Arsenal on characteristics of the suspension system components and observation of the results of the field tests (Sec. 2.2) provided the basis for certain assumptions from which a simulation of the suspension system could be formulated.

Although the track produces some modification of the hull response, its total contribution is relatively small. When the tank is sitting on level ground, the track tension twists the torsion bars so that the hull of the tank sits approximately one inch lower than it otherwise would, but the track tension has no appreciable effect on the pitch angle of the hull. When torque is applied to the drive sprocket from the engine to move the tank forward, there is a slightly noticeable effect of the track in lowering the back end of the hull; however, this pitch is relatively minor, and can safely be neglected. In several tests, certain road wheels leave the ground (e.g., wheels 1 and 3 of Figure 3). Each wheel rests on the track and is thus subjected to an upward force transmitted by the track. This force, in turn, is applied to the track either from the road or from some



SQUARE TIMBER



INVERTED - V RAMP

FIG. 2 ROAD FUNCTIONS USED FOR CORRELATION TESTS

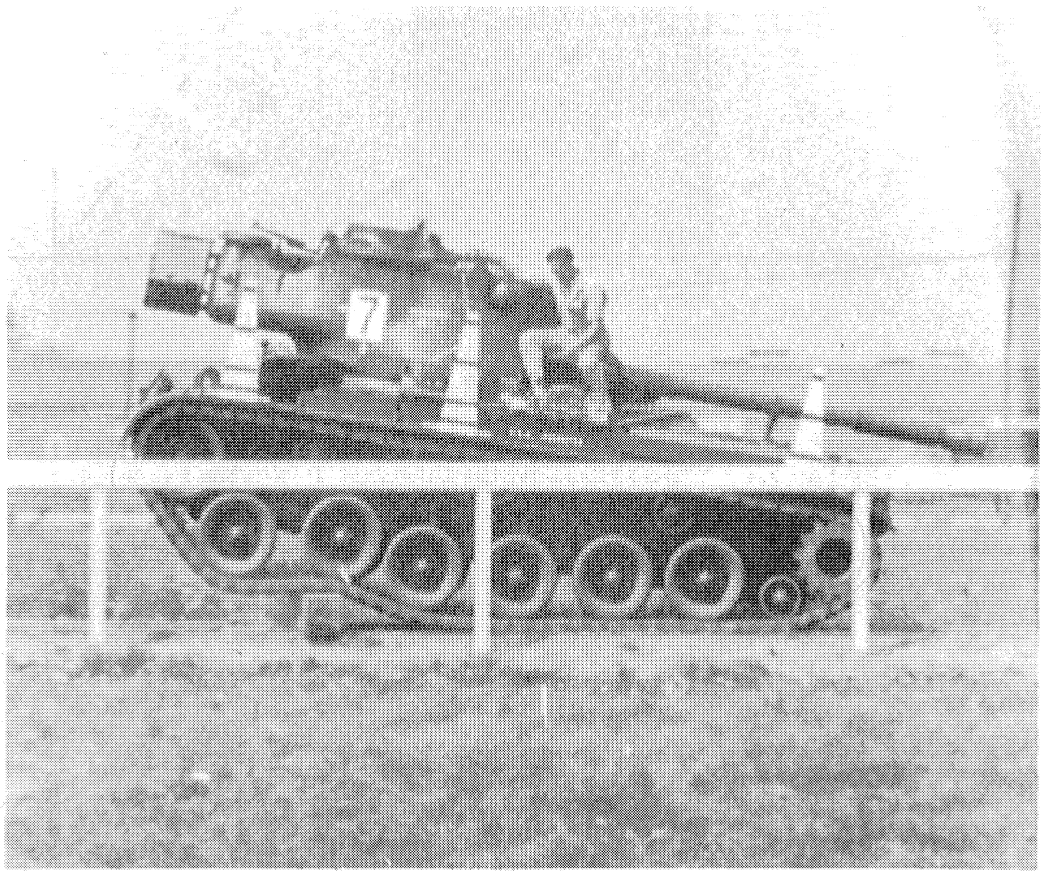


FIG. 3 M47 RUNNING OVER A 12-INCH -SQUARE

other part of the tank, such as the adjusting idler wheel. For that part of the vertical force on the roadwheel which comes from some other part of the tank, the vertical forces on the hull approximately cancel and can be disregarded. Vertical forces transmitted to the wheel from the road are always small unless the wheel is very close to the rectangular bump. To account for the effect of the track on the wheels in the vicinity of the bump, the bump can be represented in the simulation by the trapezoid shown in Figure 2 rather than as a sharp-edged rectangle.

The spring effect for each road wheel consists of the sum of terms due to the torsion bar, the volute spring, and the metal-to-metal contact which occurs when the volute spring is fully depressed. The wind-up constant of each torsion bar has been given as 7500 inch-pounds per degree of wind-up. By referring to a scaled layout of the suspension system, the torsional spring constant can be graphically converted within the allowable error to a linear spring constant of 20,000 pounds per foot. In the downward direction, the force transmitted to the hull by the torsion bar is limited to the weight of the road arm and the wheel itself. The characteristics of the volute springs, including the metal-to-metal contact, are shown in Figure 4. Combining all terms and making minor modifications to simplify the simulation circuitry produces the curves used for the simulation of the spring effect as shown in Figure 5.

The forces due to the shock absorbers are given as functions of the speed of the piston with respect to the cylinder. As in the case of the spring constant, it is necessary to convert these data graphically into functions of vertical motion of the road wheel with respect to the hull. The results (as far as data were supplied) are shown in Figure 6. Extensive extrapolation of the data was required.

The simulation computes the pitch and vertical deflection of the hull by use of equations which represent the following mathematical relations:

1. The sum of the vertical forces on the hull (due to the torsion bars, the volute springs, the shock absorbers, and gravity) is equal to the mass of the hull multiplied by its vertical acceleration.

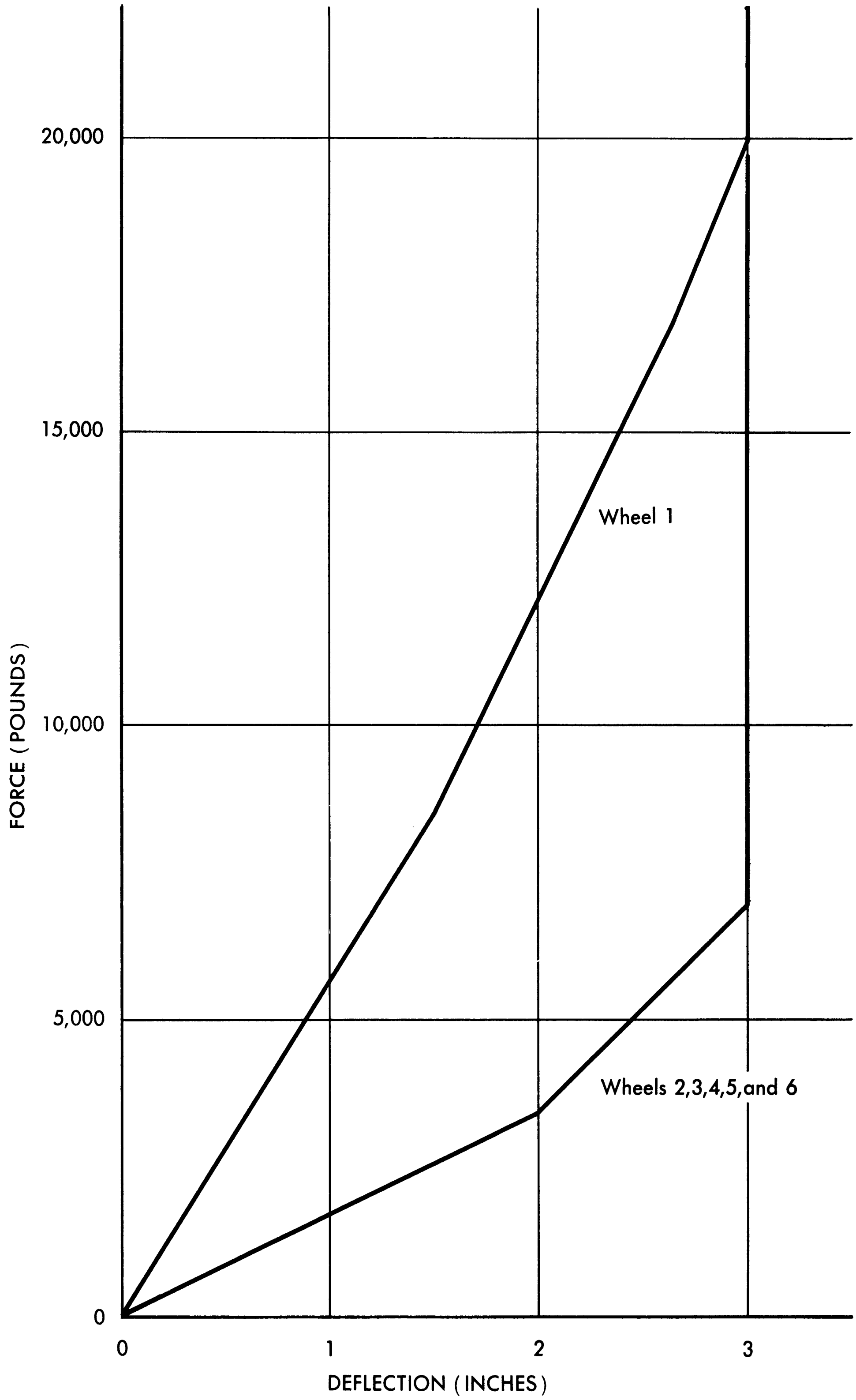


FIG. 4 VOLUTE SPRING CHARACTERISTICS

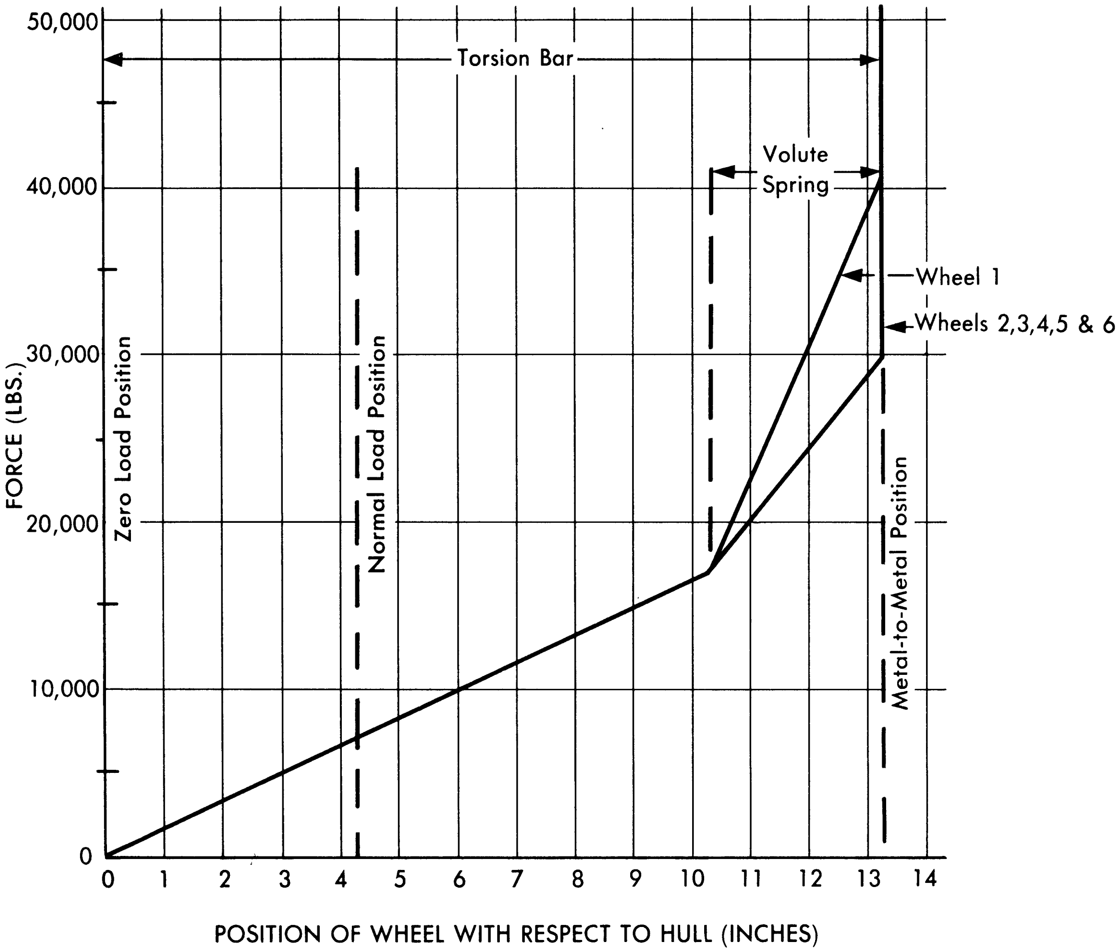


FIG. 5 COMBINED SUSPENSION SPRING CHARACTERISTICS USED FOR SIMULATION

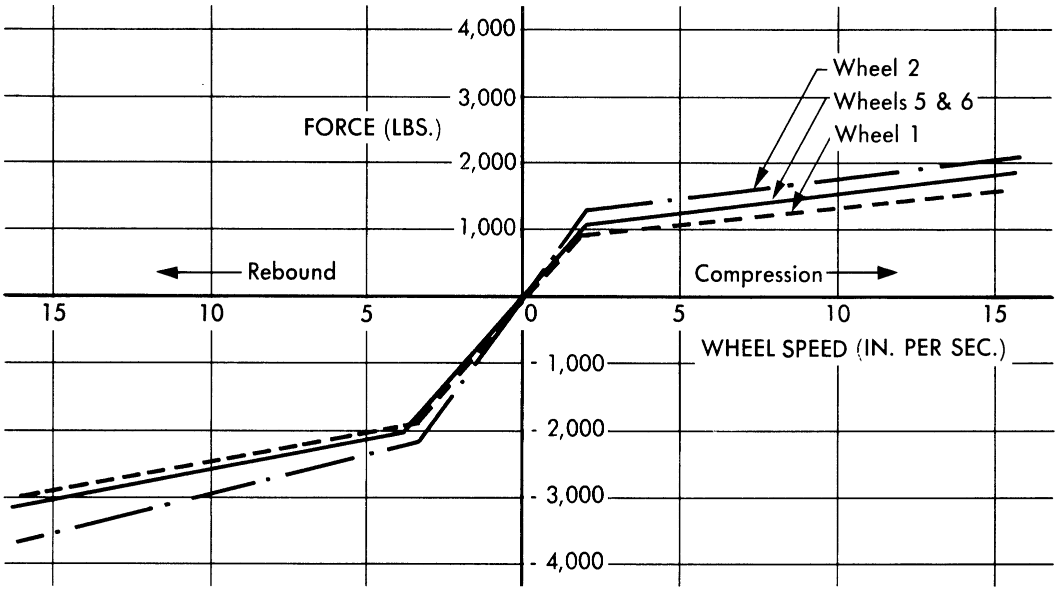


FIG. 6 SHOCK ABSORBER CHARACTERISTICS AS USED FOR SIMULATION

2. The sum of the moments applied to the hull about its center of gravity (C. G.) is equal to the polar moment of inertia about the pitch axis multiplied by the angular acceleration of the hull. All moment-producing forces are assumed to act in a vertical direction.

3. The deflection of each of the road wheels with respect to the hull is computed as a function of vertical deflection of the center of gravity of the hull, pitch angle of the hull, and vertical position of the road wheel, by use of the known geometrical arrangement of the various components of the tank.

4. The sum of vertical forces on each road wheel (due to the torsion bar, the bump stop, the shock absorber, the upward reaction of the road acting through the spring and damping effects of the tire, and gravity) is equal to the mass of the wheel multiplied by its vertical acceleration.

The equations used in the simulation and the corresponding computer set-up are given in Appendix B.

2.4 CORRELATION RESULTS

As a means of comparing the real tank with the simulated tank under the least complicated conditions, checks were made of hull pitch and vertical deflection under static conditions. These data were obtained by riding the tank onto an obstacle, allowing it to come to rest, and then measuring pertinent distances, from which vertical displacement and pitch of the hull could be determined. Comparing these measured quantities with the simulated values provides a check on the validity of the computing technique for the steady-state conditions. This comparison is shown for the case of the 12-inch square in Figure 7, where pitch and vertical displacement are plotted for different horizontal locations of the tank with respect to the bump. It can be seen that the correlation between real and simulated values is, for the most part, within the limits of the measurement errors.

An example of the correlation of the dynamic response of the tank for runs over the inverted-V ramp at 20 fps is shown in Figure 8. Again,

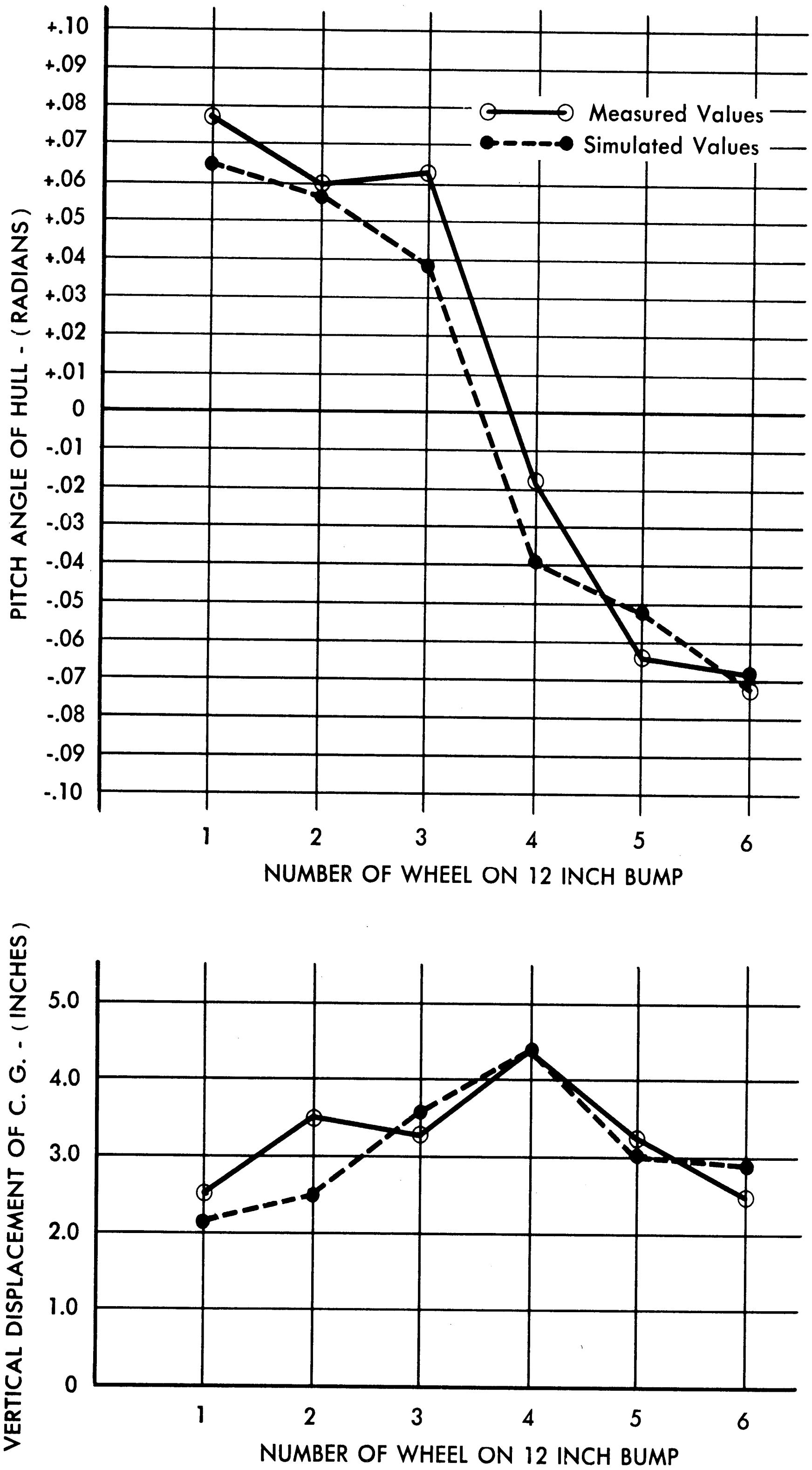


FIG. 7 COMPARISON OF MEASURED AND SIMULATED VALUES OF PITCH AND VERTICAL DISPLACEMENT OF HULL UNDER STATIC CONDITIONS

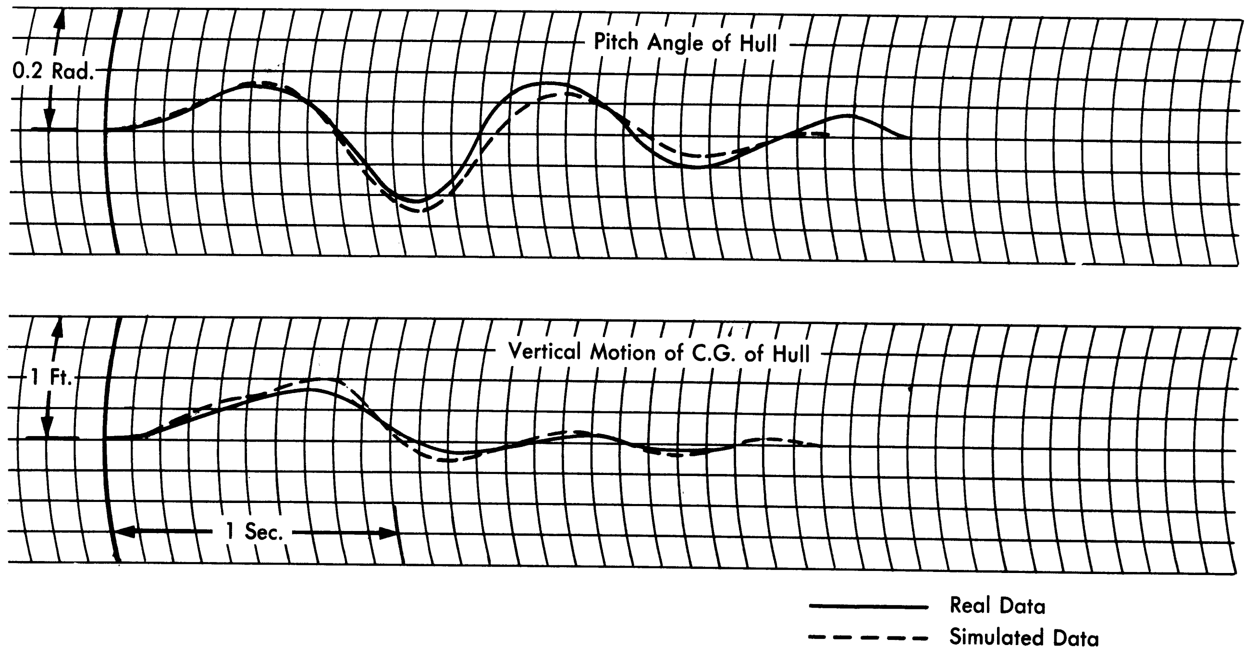


FIG. 8 REAL vs. SIMULATED RESPONSE OF M47 (8 Inch Inverted-V Ramp, Tank Speed of 20 fps.)

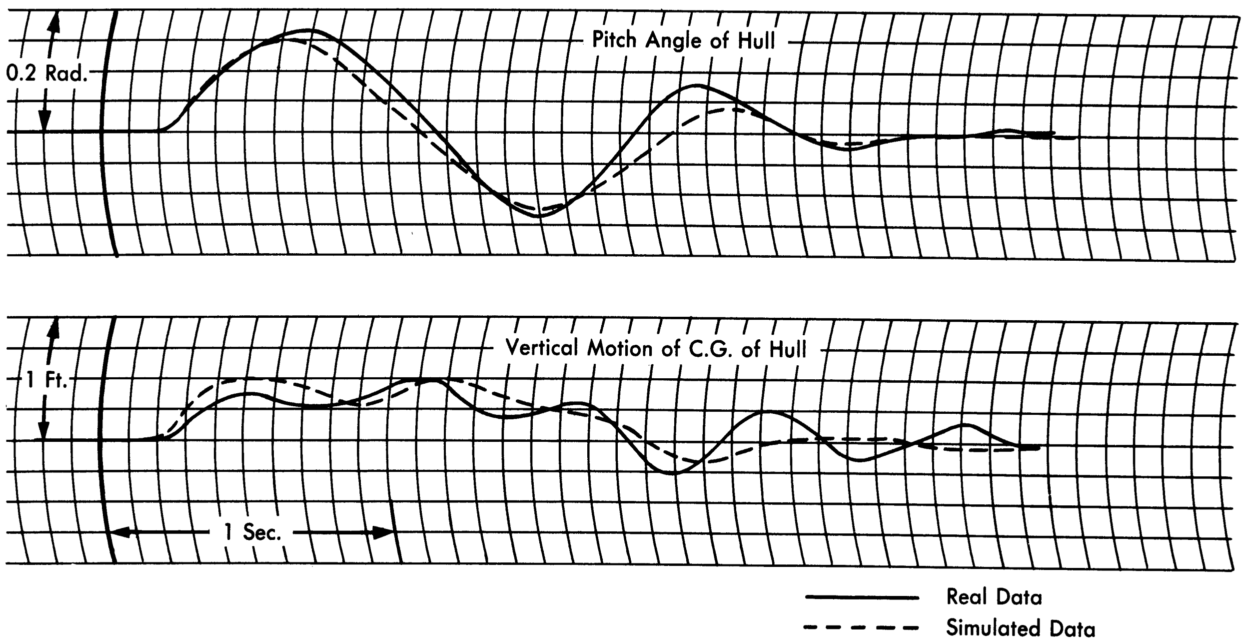


FIG. 9 REAL vs. SIMULATED RESPONSE OF M47 (12 Inch Square, Tank Speed of 11 fps.)

good agreement is obtained, although there are noticeable discrepancies. The general shapes of the curves are similar. The pitch-frequency match is necessarily good, since the response of the real tank in pitch was used as a basis for computing moment of inertia of the hull. Bounce frequencies were calculated independently of the test results but are nevertheless in fairly good agreement. Peak amplitudes of the real and simulated motion check quite closely. The simulated tank is more heavily damped than the real tank. This discrepancy may be due to uncertainties in the data for higher values of shock absorber speed.

An example of the correlation of the dynamic response for runs over the 12-inch square is shown in Figure 9. Because the tank bottoms (i.e., reaches the metal-to-metal position for one or more wheels) when traversing the 12-inch square, the shape of the response curve can be influenced markedly by small variations in the speed of the tank, the values of wheel-to-hull deflection at which bottoming occurs, and the characteristics of rubber and metal at high loadings. The correlation between real and simulated response, although somewhat less satisfactory than for the inverted-V ramp, is within the limits of accuracy required for use in engineering design. As in the comparison for the inverted-V ramp, there is close correlation of frequencies and peak amplitudes of motion, and the real tank is more lightly damped than the simulated tank.

In reviewing the results of the correlation tests, it is important to keep in mind certain limitations in the program, namely:

1. The primary field test data used for the dynamic tests were taken from the photographic records. It seems probable that individual measurements taken from these pictures contained errors of 1 or 2 inches in measured vertical deflection.

2. Knowledge of tank characteristics was limited. Measurements of some parameters could not be made directly on the components used in the M47 being tested. In some cases, the data used were made available by the Detroit Arsenal based on tests on similar components. In other cases (e.g., spring characteristics of the rubber tire and track), the type of data required was not available and was therefore estimated.

3. During the field tests, complete control of test conditions was not obtainable. The road was not absolutely hard or smooth as simulated on the computer. The 12-inch square sometimes moved ahead a few inches when the tank passed over it. The horizontal speed of the tank could not be maintained constant while passing over the bump.

III

ANALYSIS OF THE M47 SUSPENSION SYSTEM

3.1 PERFORMANCE DATA WHICH CAN BE OBTAINED BY MEANS OF SIMULATION

The performance data which can be obtained from the simulation of a suspension system include the following:

1. The ride behavior of the tank can be determined. Ride behavior is defined as the effect of the pitch and bounce of the hull on the comfort and performance of the operating personnel. Plots of pitch and bounce in terms of displacement, velocity, and acceleration at any position in the hull can be obtained, and these can be interpreted in such a way as to indicate the effect on personnel in producing fatigue over long periods of time or in interfering with the ability of the personnel to perform tasks while the tank is in motion.

2. The effect of hull motion on the performance of certain equipment can be investigated. For example, in a tank having a gun stabilization system, pitching of the hull increases the errors in gun stabilization by introducing disturbances into the system.

3. Stresses in suspension or structural components, such as road arms, volute springs, or the frame of the hull, may be computed. These will usually have maximum values at instants when the hull bottoms on the suspension system. It is also possible to infer the magnitude of stresses occurring on such items as mounting brackets from computed values of maximum acceleration occurring at the appropriate point in the hull.

The type of information described above can be used as a basis for suspension system design. The effect on ride behavior and stresses of varying the geometrical arrangement and the characteristics of the suspension components can be observed in order to arrive at the optimum system design.

Not only does the simulation of the suspension system provide useful data for suspension design, but it also provides data to aid in the design of other components of the tank, such as fire-control systems and gun shock absorber systems.

Ride behavior may be considered to include hull motions both in the range corresponding to the basic natural frequency of the suspension system, and also at high frequencies such as might be caused by the road wheels running over the irregularities of individual track blocks. The work covered in this report is concerned only with the motion at the lower range of frequencies.

3.2 COMPUTER STUDIES OF M47 HULL RESPONSE

Having established a satisfactory correlation between the real and simulated M47 tank, a study was conducted using this simulator set-up to determine the effect on the tank's performance of changing certain design parameters. The quantities recorded included:

1. Pitch angle of hull,
2. Vertical displacement of hull at C. G.,
3. Vertical displacement of hull above first wheel (i.e., at driver's position),
4. Angular acceleration of hull in pitch,
5. Vertical acceleration of hull at C. G.,
6. Vertical acceleration of hull above first wheel,
7. "Ride index" of hull above first wheel,
8. Total spring force at wheel 1,
9. Total shock absorber force at wheel 1,
10. Shear at C. G. of hull,
11. Bending moment at C. G. of hull.

The methods used in recording the shear, bending moment, and ride index are discussed in Appendices B and C.

Figure 10 shows a set of runs illustrating how the system response is recorded from the output voltages supplied by the computer. In this example the runs were made over the 12-inch square at a speed of 10 fps.

The values of the peak and average ride index provide a basis for comparing, in a general way, the ride behavior of the tank under various conditions. The ride-index curves are given in terms of an ordinate related to the recommended limit for riding comfort as given in Chart 11 of Reference 6. This recommended limit is based on riding comfort considered desirable for passenger car use; for tank operation, higher values of ride index are probably acceptable. It should also be remembered that the transients presented in this report are of relatively short duration, and unless the road obstacle were repeated at frequent intervals the over-all effect on ride behavior could be considered less serious than a steady-state behavior of the same peak or average ride index. Thus, the high peaks in the ride index curves which occur during bottoming are not particularly serious if they occur only occasionally. As noted in Appendix C, further research needs to be carried out on riding comfort measurements to establish the limits of usefulness of the ride index computation.

In Figure 10, the ride-index curve is plotted as a function of time. By comparison, the curve which would be produced by a sinusoidal vibration corresponding to the recommended limit specified in Reference 6 would appear as a sine wave having a peak amplitude of $\frac{1}{250}$ of full scale. Average ride index is obtained by computing the ratio of the average of the absolute magnitude of the ride index curve over a given period of time to the average of a sinusoidal vibration corresponding to the recommended limit.

In order to study the effect on the tank's ride behavior of variations in speed and road profile, complete series of computer runs were made, using the existing M47 design, for all combinations of 3 speeds and 3 sizes of rectangular bump. The speeds studied were 5, 10, and 15 fps. The rectangular bumps were 12 inches wide and 4, 8, or 12 inches high. Runs were also made over a bump 16 inches high at speeds of 5 and 10 fps.

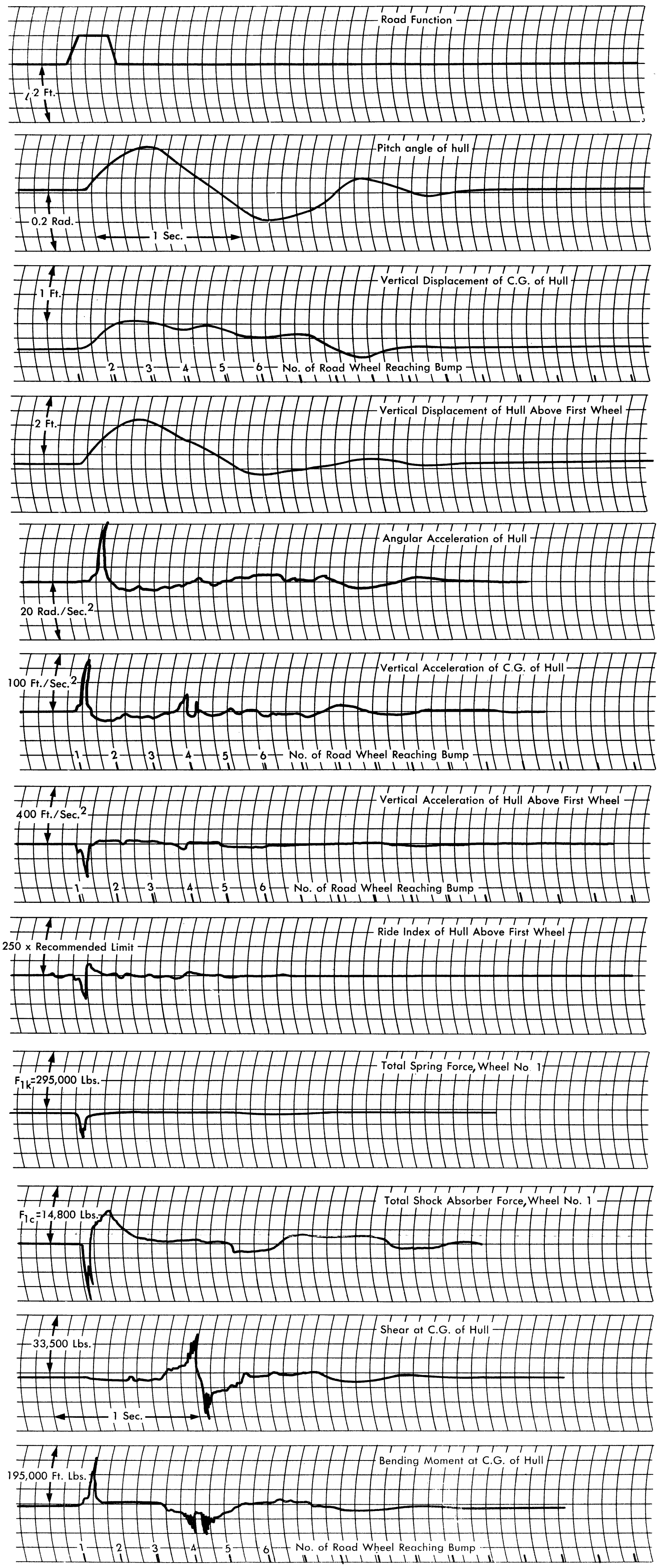


FIG. 10 SIMULATED RESPONSE OF M47 (12 Inch Square Bump, Tank Speed of 10 fps.)

For a number of reasons, the results (Sec. 3.4) of the runs over the 16-inch bump are not considered to provide accurate data and should be used only to indicate general trends.

Further studies were made to determine the effect of varying design parameters, and were confined to a tank speed of 10 fps over a 12-inch square. During one series of these runs, the hydraulic shock absorber characteristics were varied, while the torsion bars and volute springs were retained at their original values. The shock absorbers in one case were changed to produce only half the force of the standard units at a given velocity and in another case to produce twice the force. Another variation was to replace the hydraulic shock absorbers with snubbers having a constant damping force in both directions of 3100 pounds. Another set of runs was made to determine the effect of varying suspension spring characteristics. In these runs, the stiffness of all torsion bars and volute springs was increased by 50 per cent.

3.3 SUMMARY OF SIMULATION RESULTS

Runs similar to Figure 10 were made for all the conditions listed in Section 3.2 except the total spring force and total shock absorber force of wheel 1. Table 1 presents a summary of the data obtained on the hull response by the computer for the various runs.

As the bump height is increased from 4 inches to 16 inches, it can be seen that generally, at a given speed, all the response variables of the hull increase with the height of the bump. For example, the approximately exponential variation of pitch angle with bump height is shown graphically in Figure 11. Similar curves could be drawn for the vertical displacement at the C.G. and at the driver's seat and for the maximum shear and bending moment at the C. G. The values of maximum pitch and vertical accelerations as well as ride index remain fairly constant for the 4- and 8-inch bumps and then rise sharply for the 12- and 16-inch bumps. An example of this is shown in Figure 12 where maximum vertical accelerations at the C. G. and the driver's seat are plotted as a function of the height of the bump.

Examination of the computer recordings from which the above data

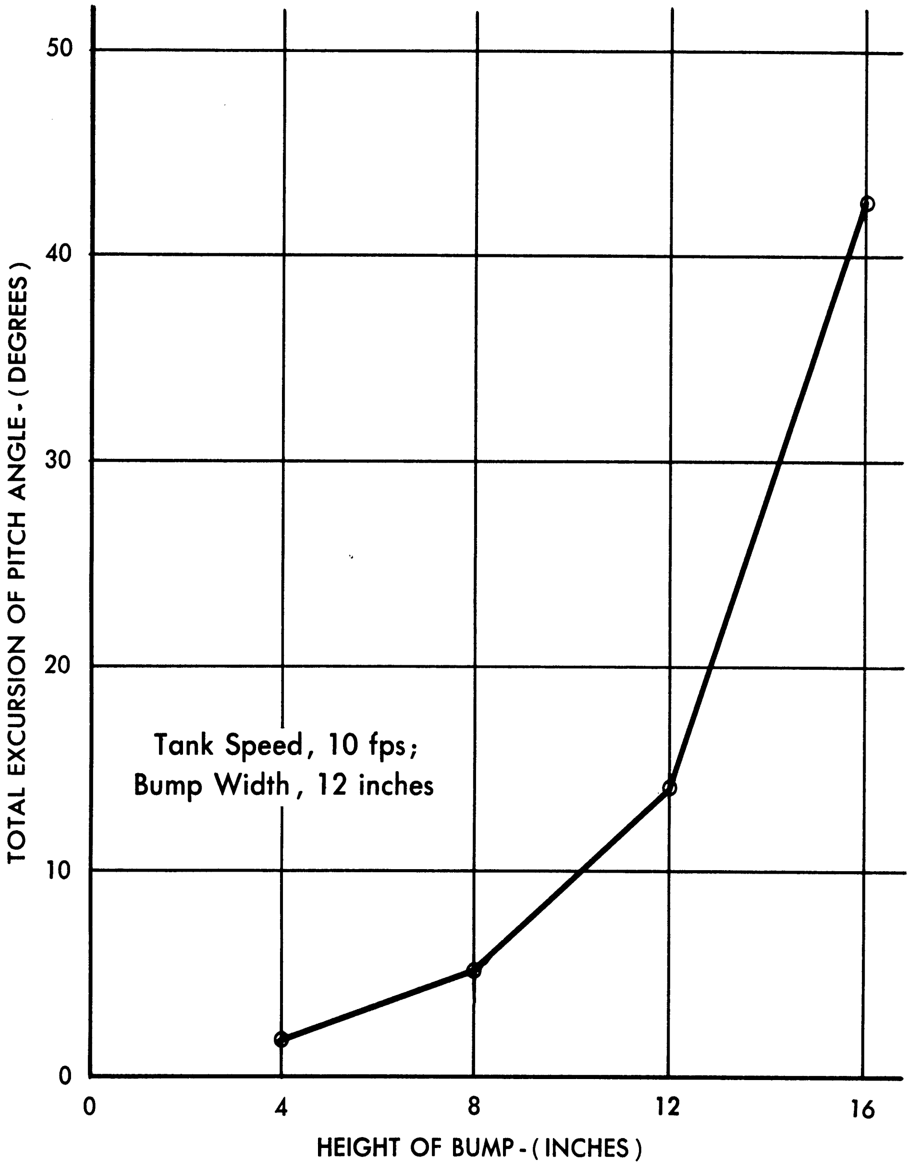


FIG. 11 PITCH ANGLE OF HULL OF M47 VS. HEIGHT OF 12 INCH WIDE BUMP

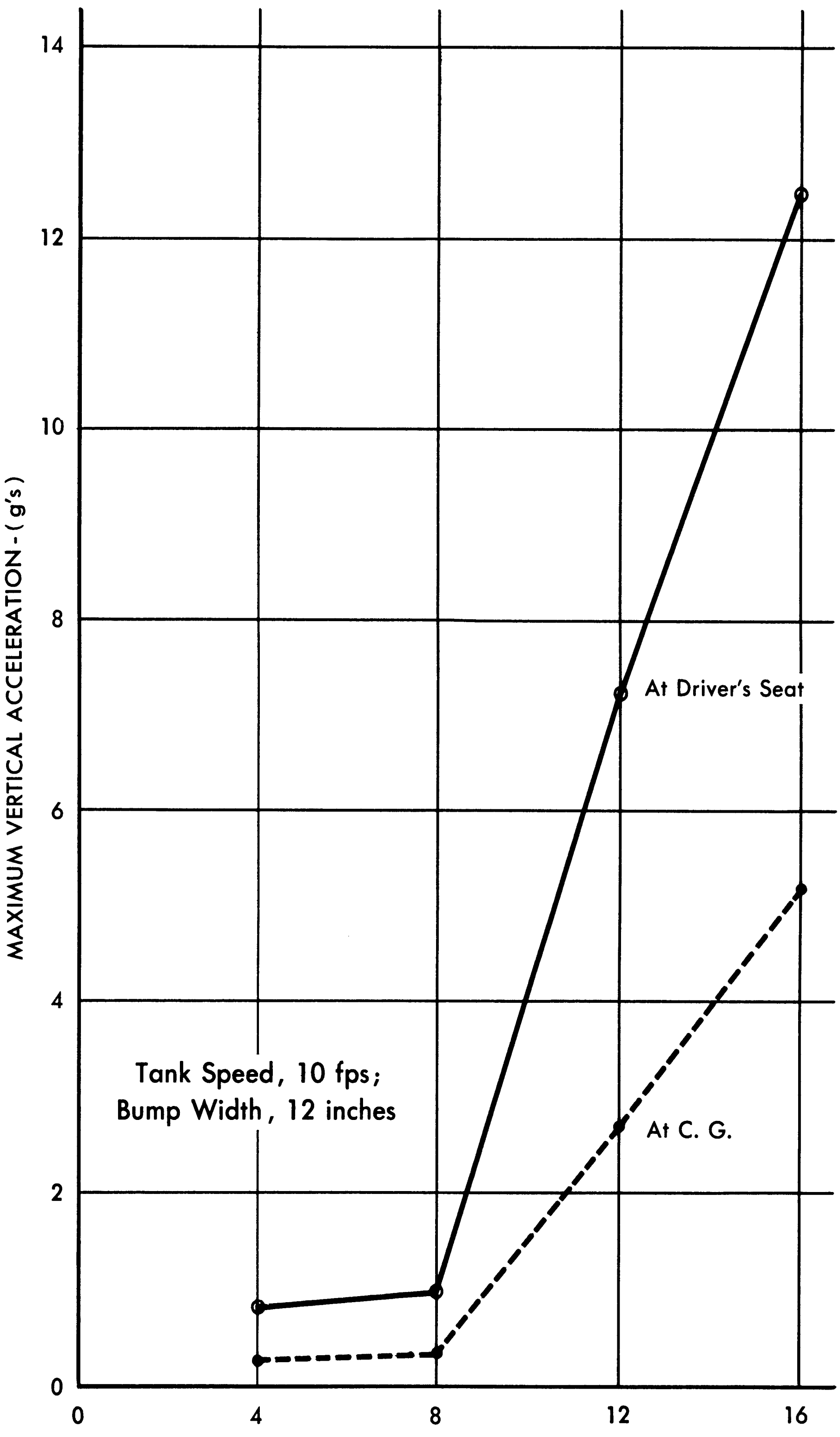


FIG. 12 MAXIMUM VERTICAL ACCELERATION AT C. G. AND DRIVER'S SEAT VS. HEIGHT OF 12 INCH WIDE BUMP

were obtained indicates that the large increases in the variables for the higher bumps are due to the bottoming which occurs when the bump height is 12 inches or more. For these heights, bottoming occurs consistently when the first road wheel rides up the bump. Additional indications of bottoming occur for the back wheels, but the particular wheels affected depend on the height of the bump and the speed of the tank. Higher bumps tend to increase the total number of wheels which bottom, while higher tank speeds tend to shift the secondary bottoming farther to the rear.

For constant bump height, the values of pitch, vertical displacement of the hull at the C. G. and driver's seat, and shear and bending moment at the C. G. seem to remain fairly constant with changes in speed. However, there is a rather uniform rise in the values of ride index, maximum pitch acceleration, and maximum vertical acceleration at the driver's seat and C. G. as the speed increases from 5 to 15 fps.

The response of the M47 hull with suspension systems having shock absorbers one-half standard, standard, and twice the standard strength used in the other computer runs are also shown in Table 1. For these suspension systems at a constant bump height of 12 inches and speed of 10 fps, it can be seen that the values of nearly all of the variables show an appreciable decrease as the shock absorber parameters are raised from one-half standard to standard value. Raising the parameters to twice standard value produces a further decrease in most of the variables, but by a relatively small amount. The oscillation of the hull, in response to the bump, is noticeably under-damped for the lowest value of damping used. This is not enough to be disturbing as far as ride behavior is concerned, but might interfere to some extent with the effectiveness of a gun stabilization system. On the basis of this study it can be concluded that the present shock absorber settings give approximately optimum performance.

Some of the later production versions of the M47 use friction snubbers instead of hydraulic shock absorbers; therefore, it is of interest to compare the response of the tank with this type of damping against the response using the hydraulic shock absorbers. The snubbers were assumed to react with 3100 pounds of force in a direction parallel to their length in both compression and rebound. They were also assumed to be positioned in the same physical location as the hydraulic shock absorbers are on the M47. The simulated runs with the snubbers were made over the 12-inch square at 10 fps. It is seen from Table 1 that the values of all the variables of response for the snubbers is always slightly higher than the values for the half-strength shock absorbers. A comparison of the curves on which Table 1 is based also brings out the fact that the damping for the two systems is about the same. It can be concluded that the variations in the type of damping device which were tried did not have any major effect on the tank response and that the 3100-pound snubbers are almost equivalent to the half-strength shock absorber.

The effect of using torsion bar spring constants 50 per cent greater than the standard values can also be determined from Table 1. At a speed of 10 fps over a 12-inch square, this system shows a decided decrease in all of the variables of the hull response compared to the standard suspension system. The values of pitch angle and vertical displacement are decreased slightly by increasing the torsion bar spring constants while the values of ride index, pitch acceleration, vertical acceleration, and shear and bending moment are reduced to a larger degree. The reduction in peak values is due to the fact that the increased cushioning effect of the stronger torsion bars materially reduces the impact which occurs when wheel 1 bottoms and completely eliminates the bottoming of wheels 3 and 4.

From the curves of Figure 10, numerical values can be obtained to indicate the magnitude of stresses in certain of the suspension and structural components of the tank. It can be determined, for example, that the following maximum values exist for the condition studied at some instant during the run:

Vertical acceleration of hull at C. G. = 2.64 g.

Vertical acceleration of hull at driver's seat = 7.15 g.

Shear at section through C. G. of tank = 27,600 lbs.

Bending moment at section through C. G. of tank = 156,000 ft. -lbs. (in direction to produce compression at top of hull).

Velocity of front shock absorber on wheel 1 = 156 in./sec.

Velocity of rear shock absorber on wheel 1 = 35 in./sec.

Force transmitted through bump stop of wheel 1 = 103,000 lbs.

By setting up appropriate computer circuits and recorder connections, data similar to the above can be found for other positions in the hull or other road wheels.

IV

CONCLUSIONS

Based on the work described in this report, it is possible to draw the following general conclusions concerning the present state and future trends of suspension system simulation.

4.1 PRESENT STATE OF THE ART

The suspension system studies undertaken thus far have been confined to motion of the tank in the pitch plane and have been limited to problems not requiring extensive consideration of the effects of the track or of longitudinal forces. At present, results of engineering accuracy can be expected from suspension system simulations only if computer studies are confined to situations which do not result in bottoming of the suspension system. When bottoming does occur, computer results should be accepted as having qualitative significance only; the general nature of the effects produced by changing a given parameter can be observed and insight can thus be gained into the operation of the mechansim, but values of the quantities recorded by the computer should be used on a comparative basis only.

4.2 FUTURE TRENDS

The development of a more comprehensive and more accurate suspension system simulation should include the following phases:

1. Collection of test data on vehicle characteristics and performance, personnel capabilities, and terrain characteristics.
2. Mathematical analysis of the mechanics of the suspension system.
3. Preparation of the computer set-up for more complex simulations.

4.2.1 Collection of Test Data

Basic knowledge of vehicle characteristics and performance, personnel capabilities, and terrain characteristics is still quite limited and much of it will have to be obtained by conducting tests under carefully controlled conditions. In particular,data are needed on rubber characteristics, deflection and coefficient of restitution of metal in nominally rigid members under heavy loading, and the relationship of the wheels to the track in order to allow accurate computation of maximum forces and stresses. Also, generally accepted quantitative criteria of environment and performance need to be established, particularly characteristics of road profiles and standards of acceptable vehicle ride behavior.

4.2.2 Mathematical Analysis

More mathematical analysis of suspension systems must be carried out for the pitch plane simulation as well as for more advanced simulations. It would be desirable to increase the scope of the present simulation to make it more accurate and comprehensive:

1. The effect of longitudinal forces on the motion of the tank can be introduced. This would include the forces transmitted by the engine acting through the drive sprocket, forces produced by the friction between track and road or by road obstacles, gun recoil forces, and the effect of gravity when the tank is on a slope.
2. Track effects can be introduced. Support of road wheels by the track as well as downward forces exerted on the drive sprocket and adjusting idler wheel would be allowed for. Since the magnitude of these effects depends on track tension, this variable would also be brought into the computation.
3. The simulation can be extended to three dimensions by including roll motion as well as pitch motion.

4.2.3 Simulation Techniques

The preparation of the computer set-up for more complex simulation does not appear to present any difficult problems. The suspension system of a tank can be treated as an assemblage of masses, non-linear springs and non-linear damping devices for simulation purposes. Simulation of the dynamics of such a system makes use of basic computer circuits and techniques already well established in the analog computer field. It is believed that the work discussed in this report has successfully dealt with most of the problems likely to be met in setting up and operating the computer. Not only have the Newtonian equations of motion of this complex system been set up and solved on the computer, but solutions have been obtained for special equations of a type capable of indicating motions, forces, or stresses at virtually any point of interest in the system. The one special item of computer equipment that is required for advanced methods of simulation is a device for generating more complicated road functions than the ramp and the square.

Thus, the use of the analog computer for suspension system simulation consists largely of applying known methods to a specific type of problem. The principal requirement for developing an improved suspension simulation is, therefore, continued basic research on tank design.

APPENDIX A

INSTRUMENTATION METHODS USED IN OBTAINING DATA ON TANK RESPONSE FROM FIELD TESTS

To obtain the necessary data on the motion of the M47 tank as it rode over obstacles in the road, a measuring scale, electronic measuring and recording instruments, and motion pictures were used. Data taking was confined to motions of the tank in the plane perpendicular to its pitch axis.

A.1 STEADY-STATE MEASUREMENTS

As a first step in justifying the assumptions used in simulating the tank, a series of steady-state measurements was made with a ruler while the tank rested in various positions with respect to the square timber or the inverted-V ramp. From dimensions h_f and h_r (Fig. A-1), it is possible to calculate both pitch angle, θ, and vertical displacement, y_o, of the hull. The vertical position of each road wheel was also measured.

A.2 ELECTRONIC INSTRUMENTATION

For measuring the pitch angle of the hull, a gyro was mounted on the hull of the tank so that the spin axis of the gyro was vertical. The electrical pickoff on the gimbal whose axis is parallel to the pitch axis of the hull provided a direct measurement of the pitch angle. The a-c output of the pickoff was amplified, demodulated, and filtered, and the resulting d-c signal was applied to a recording system. This recorder thus furnishes a continuous record of pitch angle as a function of time.

The physical quantity measured by the accelerometer is the vertical acceleration of the tank at the point at which the accelerometer is located. To obtain the acceleration data, the output of the accelerometer was amplified, demodulated, filtered, and applied to a recorder.

Power required for the above instrumentation consisted of 28 volts d-c supplied from the tank battery and 115 volts 60 cps a-c supplied by a gasoline-engine-driven generator mounted on one fender of the tank.

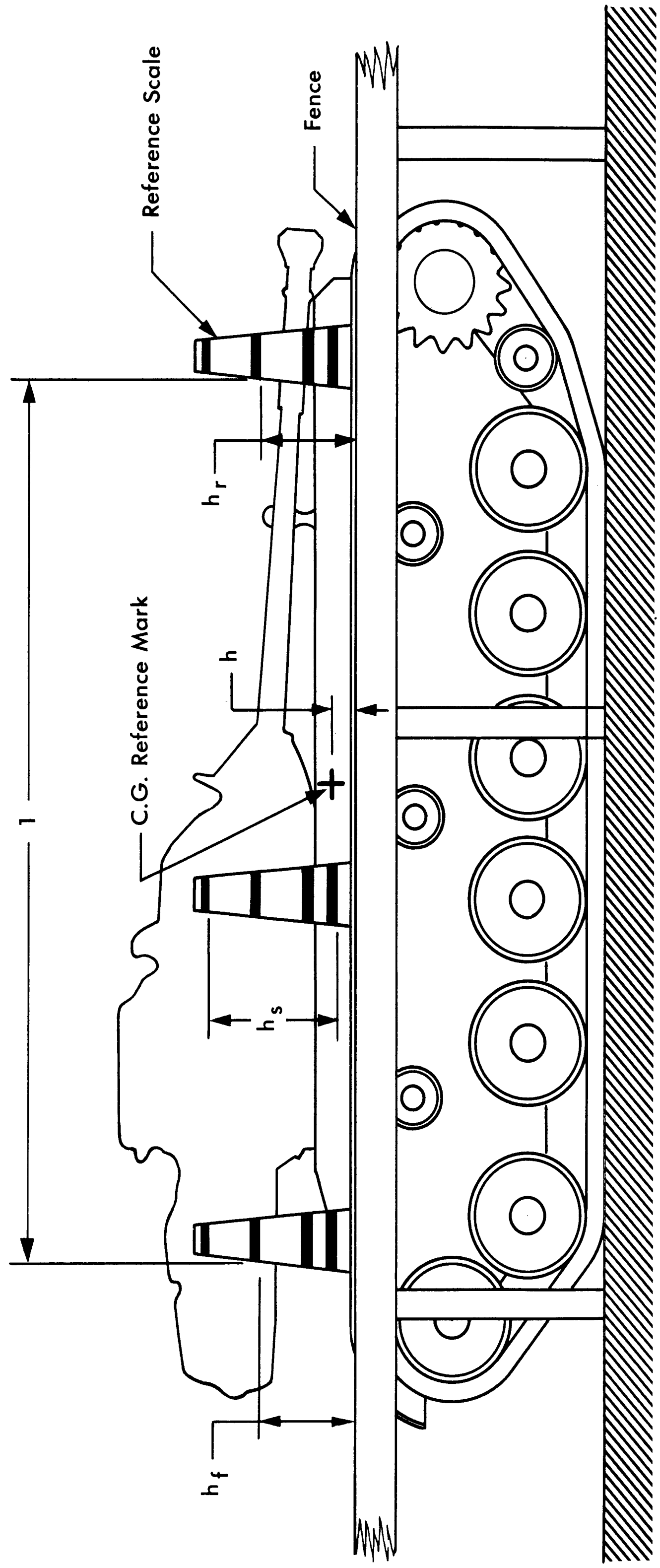


FIG. A-1 MEASUREMENTS TAKEN TO OBTAIN DATA ON PITCH AND BOUNCE

A.3 USE OF MOTION PICTURE CAMERA

Another method of measuring tank motion is the use of motion picture camera techniques. While this approach does not give data as directly as electronic instrumentation, it provides records not only of the hull, but also of the movement of the road wheels and track as they operate over obstacles in the road. A single frame from one of the motion picture records is shown in Figure 3 for the tank traveling over a 12-inch square.

In preparation for recording a run by motion pictures, a straight course was mapped out over which the tank was to be driven and an obstacle such as a square timber or ramp was placed in the road. The camera was set up at a distance of about 100 feet from the road and was equipped with a telephoto lens so that the tank would take up as much of the picture frame as possible. At the same time, the camera was far enough back from the road so that problems due to photographing the tank at oblique angles were minimized. On the side of the road nearest the camera a fence was set up with a continuous horizontal bar on it to provide a reference line from which vertical measurements could be made. The posts were spaced at regular intervals so as to provide an indication of the location of the tank for each frame of the picture, and could therefore be used to determine the tank speed throughout the runs. As shown in Figures 3 and A-1 vertical bars were placed on the tank at the front, middle, and back so that measurements could be taken from these marks to the horizontal line established by the fence. In a typical run, the tank started at one end of the road, got up to speed, passed over the obstruction, and maintained its speed until the transient disturbance died out, after which it came to a stop. The camera followed the tank throughout the run.

Data reduction was accomplished by projecting the completed picture on a vertical surface, one frame at a time, and measuring appropriate dimensions. The vertical deflection of the hull at the center of gravity, y_o, was obtained by substituting in the formula:

$$y_o = \frac{\bar{h}_s}{h_s} (h - h_i)$$

where a barred quantity indicates a true distance, an unbarred quantity indicates a distance measured off the projected picture, and h_i is the value of h when the tank is sitting on level ground.

The pitch angle, θ, was obtained from the formula:

$$\theta = \frac{\bar{h}_s}{h_s} \cdot \frac{(h_f - h_r)}{\bar{l}}$$

APPENDIX B

EQUATIONS USED IN THE SIMULATION

Although a solid body in space has six degrees of freedom (three of translation and three of rotation), only two degrees of freedom were considered in the simulation of the M47 tank on the analog computer, namely:

1. Vertical motion of the center of gravity (usually called bounce).
2. Rotational motion (usually called pitch) about a transverse axis running horizontally through the center of gravity of the tank.

In the simulation, the hull of the tank was considered to be a rigid bar coupled to six masses (wheels) by six springs (torsion bars plus volute springs) and four dampers (shock absorbers). These six masses were, in turn, supported above the road by six springs and dampers representing the spring and damping effect of the rubber tires and track. Only six wheels and half the mass of the tank were used in the simulation since it was assumed that the tank is symmetrical about a longitudinal vertical plane. This schematic representation of the tank suspension system is shown in Figure B-1.

The equations of motion are given below. A quantity of the form $k_i(y_a - y_b)$ represents the force exerted by a spring as a function of $(y_a - y_b)$, the change in its length from the reference condition. This is a non-analytic function consisting of a series of straight-line segments (see Figure 5). A quantity of the form $c_i(\dot{y}_a - \dot{y}_b)$ represents the force exerted by a shock absorber as a function of $(\dot{y}_a - \dot{y}_b)$, the time rate of change of its length. This is also a non-analytic function consisting of a series of straight-line segments (see Figure 6). Constants used in the simulation are given in Table B-1.

1. Vertical motion of the hull.

$$
\begin{aligned}
M\ddot{y}_o = & -k_1(y_1 - y_{w1}) - k_2(y_2 - y_{w2}) - k_3(y_3 - y_{w3}) - k_4(y_4 - y_{w4}) \\
& - k_5(y_5 - y_{w5}) - k_6(y_6 - y_{w6}) - c_1(\dot{y}_1 - \dot{y}_{w1}) - c_2(\dot{y}_2 - \dot{y}_{w2}) \\
& - c_5(\dot{y}_5 - \dot{y}_{w5}) - c_6(\dot{y}_6 - \dot{y}_{w6}) + Mg
\end{aligned}
$$

NOTES:

1. Positive direction of force and displacement is assumed upward. Positive direction of moment and angle is assumed clockwise.
2. Reference position of hull and each wheel is vertical position of each item if tank were resting on level ground and had no weight.

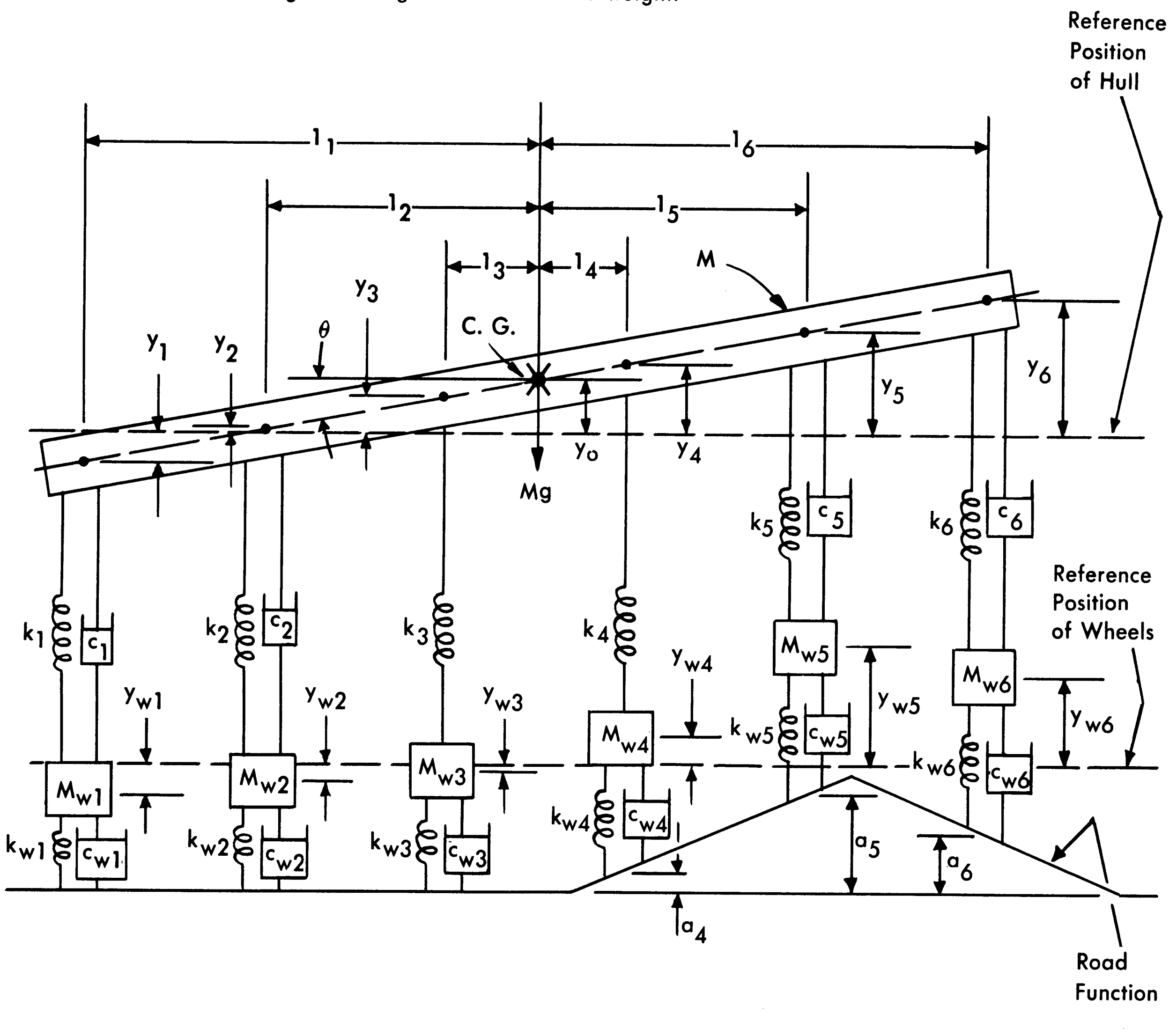


FIG. B-1 SCHEMATIC REPRESENTATION OF THE TANK SUSPENSION SYSTEM

2. Vertical motion of wheel 1.

$$M_{w1}\ddot{y}_{w1} = k_1(y_1 - y_{w1}) - k_{w1}(y_{w1} - a_1) + c_1(\dot{y}_1 - \dot{y}_{w1})$$
$$- c_{w1}(\dot{y}_{w1} - \dot{a}_1) + M_{w1}g$$

3. Vertical motion of wheel 2.

$$M_{w2}\ddot{y}_{w2} = k_2(y_2 - y_{w2}) - k_{w2}(y_{w2} - a_2) + c_2(\dot{y}_2 - \dot{y}_{w2})$$
$$- c_{w2}(\dot{y}_{w2} - \dot{a}_2) + M_{w2}g$$

4. Vertical motion of wheel 3.

$$M_{w3}\ddot{y}_{w3} = k_3(y_3 - y_{w3}) - k_{w3}(y_{w3} - a_3) - c_{w3}(\dot{y}_{w3} - \dot{a}_3) + M_{w3}g$$

5. Vertical motion of wheel 4.

$$M_{w4}\ddot{y}_{w4} = k_4(y_4 - y_{w4}) - k_{w4}(y_{w4} - a_4) - c_{w4}(\dot{y}_{w4} - \dot{a}_4) + M_{w4}g$$

6. Vertical motion of wheel 5.

$$M_{w5}\ddot{y}_{w5} = k_5(y_5 - y_{w5}) - k_{w5}(y_{w5} - a_5) + c_5(\dot{y}_5 - \dot{y}_{w5})$$
$$- c_{w5}(\dot{y}_{w5} - \dot{a}_5) + M_{w5}g$$

7. Vertical motion of wheel 6.

$$M_{w6}\ddot{y}_{w6} = k_6(y_6 - y_{w6}) - k_{w6}(y_{w6} - a_6) + c_6(\dot{y}_6 - \dot{y}_{w6})$$
$$- c_{w6}(\dot{y}_{w6} - \dot{a}_6) + M_{w6}g$$

8. Pitch of the hull.

$$J\ddot{\theta} = - l_1k_1(y_1 - y_{w1}) - l_2k_2(y_2 - y_{w2}) - l_3k_3(y_3 - y_{w3})$$
$$+ l_4k_4(y_4 - y_{w4}) + l_5k_5(y_5 - y_{w5}) + l_6k_6(y_6 - y_{w6})$$
$$- l_1c_1(\dot{y}_1 - \dot{y}_{w1}) - l_2c_2(\dot{y}_2 - \dot{y}_{w2}) + l_5c_5(\dot{y}_5 - \dot{y}_{w5})$$
$$+ l_6c_6(\dot{y}_6 - \dot{y}_{w6})$$

9. $J = Mh^2$

10. $y_1 = y_o + l_1\theta$

11. $y_2 = y_o + l_2\theta$

12. $y_3 = y_o + l_3\theta$

13. $y_4 = y_o - l_4\theta$

14. $y_5 = y_o - l_5\theta$

15. $y_6 = y_o - l_6\theta$

16. $\dot{y}_1 = \dot{y}_o + l_1\dot{\theta}$

17. $\dot{y}_2 = \dot{y}_o + l_2\dot{\theta}$

18. $\dot{y}_3 = \dot{y}_o + l_3\dot{\theta}$

19. $\dot{y}_4 = \dot{y}_o - l_4\dot{\theta}$

20. $\dot{y}_5 = \dot{y}_o - l_5\dot{\theta}$

21. $\dot{y}_6 = \dot{y}_o - l_6\dot{\theta}$

The values of shear and bending moment which were computed in the simulation are for a section (designated Section z - z) of one side of the hull through the C.G. of the tank.

22. $$V_{z-z} = k_1(y_1 - y_{w1}) + k_2(y_2 - y_{w2}) + k_3(y_3 - y_{w3}) + c_1(\dot{y}_1 - \dot{y}_{w1}) + c_2(\dot{y}_2 - \dot{y}_{w2}) + \frac{M}{2}\left((\ddot{y}_o + l_o\ddot{\theta}) - g\right)$$

where: V_{z-z} = Shear exerted on front section of tank at Section z-z in pounds.

l_o = distance from C.G. of tank to C.G. of front half of hull = 4.17 feet.

23. $$M_{z-z} = l_1k_1(y_1 - y_{w1}) + l_2k_2(y_2 - y_{w2}) + l_3k_3(y_3 - y_{w3}) + l_1c_1(\dot{y}_1 - \dot{y}_{w1}) + l_2c_2(\dot{y}_2 - \dot{y}_{w2}) + \frac{M}{2}\left((\ddot{y}_o + l_o\ddot{\theta}) - g\right)$$

where: M_{z-z} = Bending moment exerted on front section of tank at Section z-z in foot-pounds.

The schematic diagram of the analog computer set-up is shown in Figure B-2. The circuits for computation of bending moment, shear and ride index are not included in this figure.

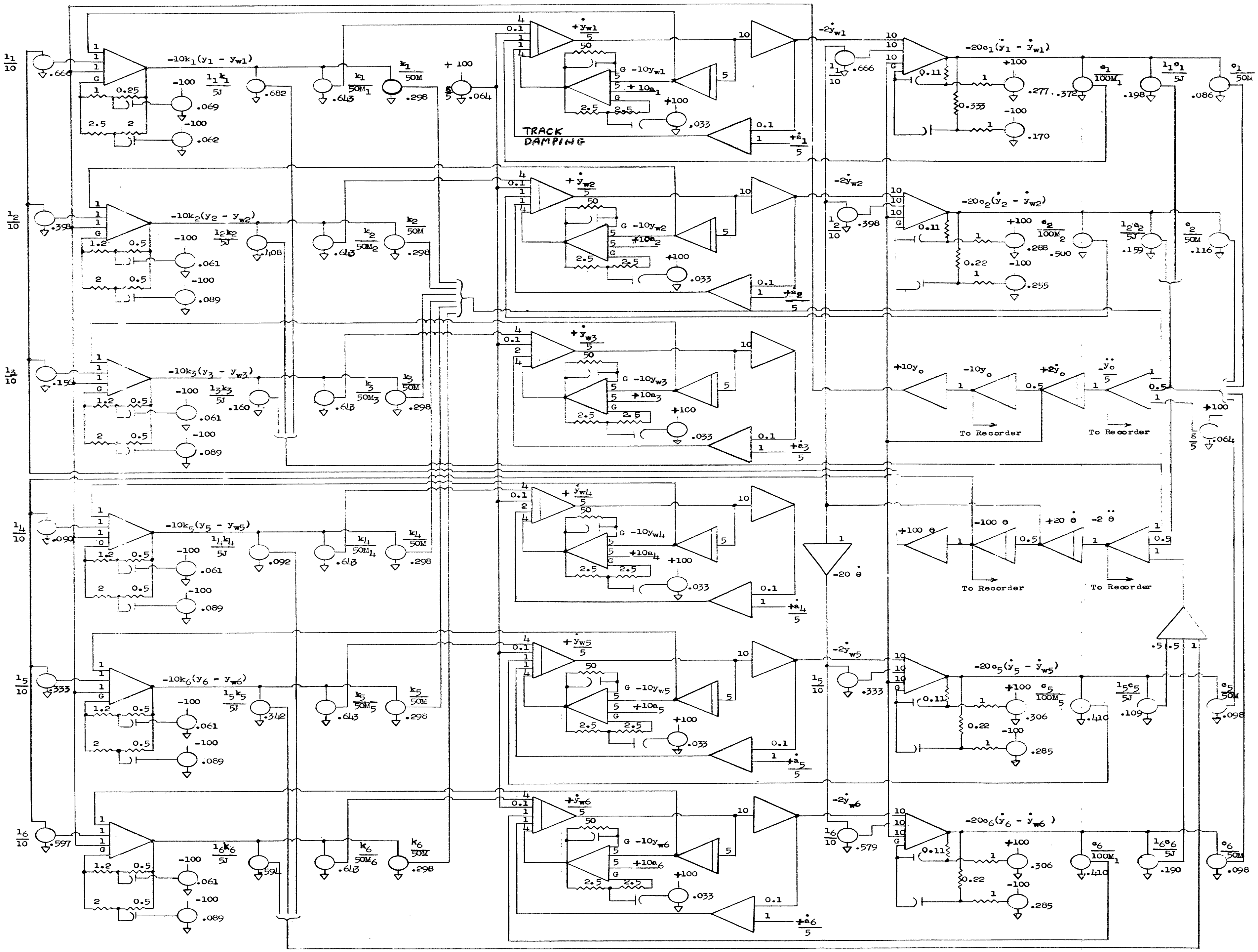


FIG. B-2 SCHEMATIC DIAGRAM OF SUSPENSION SYSTEM SIMULATION

NOTE: NO PROVISION FOR WHEELS LEAVING THE GROUND

TABLE B-1

CONSTANTS USED IN SIMULATION

M = Mass of half the tank = 1340 slugs

h = radius of gyration = 5.4 ft.

J = moment of inertia = 39,000 slug-ft.^{2}

$l_1 = 6.66$ ft.

$l_2 = 3.98$ ft.

$l_3 = 1.56$ ft.

$l_4 = 0.90$ ft.

$l_5 = 3.33$ ft.

$l_6 = 5.79$ ft.

$M_{w1} = M_{w2} = M_{w3} = M_{w4} = M_{w5} = M_{w6} = 15.6$ slugs

$k_1 = 20{,}000$ lbs /ft when $0 < (y_1 - y_{w1}) < 0.86$ ft.
$= 87{,}000$ lbs /ft when 0.86 ft. $< (y_1 - y_{w1}) < 1.11$ ft.
$= \infty$ when 1.11 ft. $< (y_1 - y_{w1})$

$k_2 = k_3 = k_4 = k_5 = k_6 = 20{,}000$ lbs /ft when $0 < (y_i - y_{wi}) < 0.86$ ft.
$= 48{,}000$ lbs /ft when 0.86 ft. $< (y_i - y_{wi}) < 1.11$ ft.
$= \infty$ when 1.11 ft. $< (y_i - y_{wi})$

$c_1 = 5800$ lbs/fps when $0 < (\dot{y}_1 - \dot{y}_{w1}) < 0.154$ fps	Compression
$= 580$ lbs/fps when 0.154 fps $< (\dot{y}_1 - \dot{y}_{w1})$	Compression
$= 5800$ lbs/fps when $0 < (\dot{y}_1 - \dot{y}_{w1}) < 0.283$ fps	Rebound
$= 1400$ lbs/fps when 0.283 fps $< (\dot{y}_1 - \dot{y}_{w1})$	Rebound
$c_2 = 7800$ lbs/fps when $0 < (\dot{y}_2 - \dot{y}_{w2}) < 0.16$ fps	Compression

TABLE B-1 (Continued)

	$= 780$ lbs/fps when $(\dot{y}_2 - \dot{y}_{w2}) > 0.16$ fps	Compression
	$= 7800$ lbs/fps when $0 < (\dot{y}_2 - \dot{y}_{w2}) < 0.28$ fps	Rebound
	$= 1400$ lbs/fps when $(\dot{y}_2 - \dot{y}_{w2}) > 0.28$ fps	Rebound
C_5	$= C_6 = 6400$ lbs/fps when $0 < (\dot{y}_{5-6} - \dot{y}_{w5-6}) < 0.17$ fps	Compression
	$= 640$ lbs/fps when $(\dot{y}_{5-6} - \dot{y}_{w5-6}) > 0.17$ fps	Compression
	$= 6400$ lbs/fps when $0 < (\dot{y}_{5-6} - \dot{y}_{w5-6}) < 0.31$ fps	Rebound
	$= 1150$ lbs/fps when $(\dot{y}_{5-6} - \dot{y}_{w5-6}) > 0.31$ fps	Rebound

$k_{w1} = k_{w2} = k_{w3} = k_{w4} = k_{w5} = k_{w6} = 156{,}000$ lbs/ft when $0 < (y_{wi} - a_i) < 0.133$ ft.

$= 1{,}560{,}000$ lbs/ft when $(y_{wi} - a_i) > 0.133$ ft.

$c_{w1} = c_{w2} = c_{w5} = c_{w6} = 156$ lbs/fps

$c_{w3} = c_{w4} = 312$ lbs/fps

APPENDIX C

RIDE INDEX CIRCUIT

The purpose of recording the ride index of the simulated tank was to provide a quantitative record of its motion in terms of a recognized standard of performance related to riding comfort. A suitable standard is that based on work done by the SAE Riding Comfort Research Committee; this standard has been reported in Reference 6 and is summarized in Chart 11 of that report. Essentially, this standard consists of a curve giving the upper limits of related values of amplitude and frequency of a sinusoidal vertical vibration for a satisfactory ride. The ride index circuit is designed in such a way as to modify a vertical acceleration signal to provide an indication whose amplitude corresponds directly to the riding comfort as defined by the curve previously mentioned.

The ride index circuit is shown in Figure C-1. The vertical acceleration signal received from the computer is fed to an equalizing network. The equalizing network produces an output voltage whose amplitude is such a function of input amplitude and frequency as to relate it directly to the riding comfort curve. For example, any sinusoidal voltage whose amplitude and frequency place it on the recommended limit curve would produce a sinusoidal output from the equalizing network having a constant amplitude independent of frequency. This output can be recorded on a direct inking recorder, in which case riding comfort can be read off by noting the amplitude of the recorded signal.

The circuit just described gives valid results for sinusoidal motion; however, there is as yet insufficient justification for assuming that it gives a correct measure of the ride for most practical cases in which the ride consists of miscellaneous motions of a non-periodic nature. To establish the validity of the ride index under these conditions, an experimental program would have to be carried out to correlate the ride indicator results with the subjective responses of human beings subjected to the same ride.

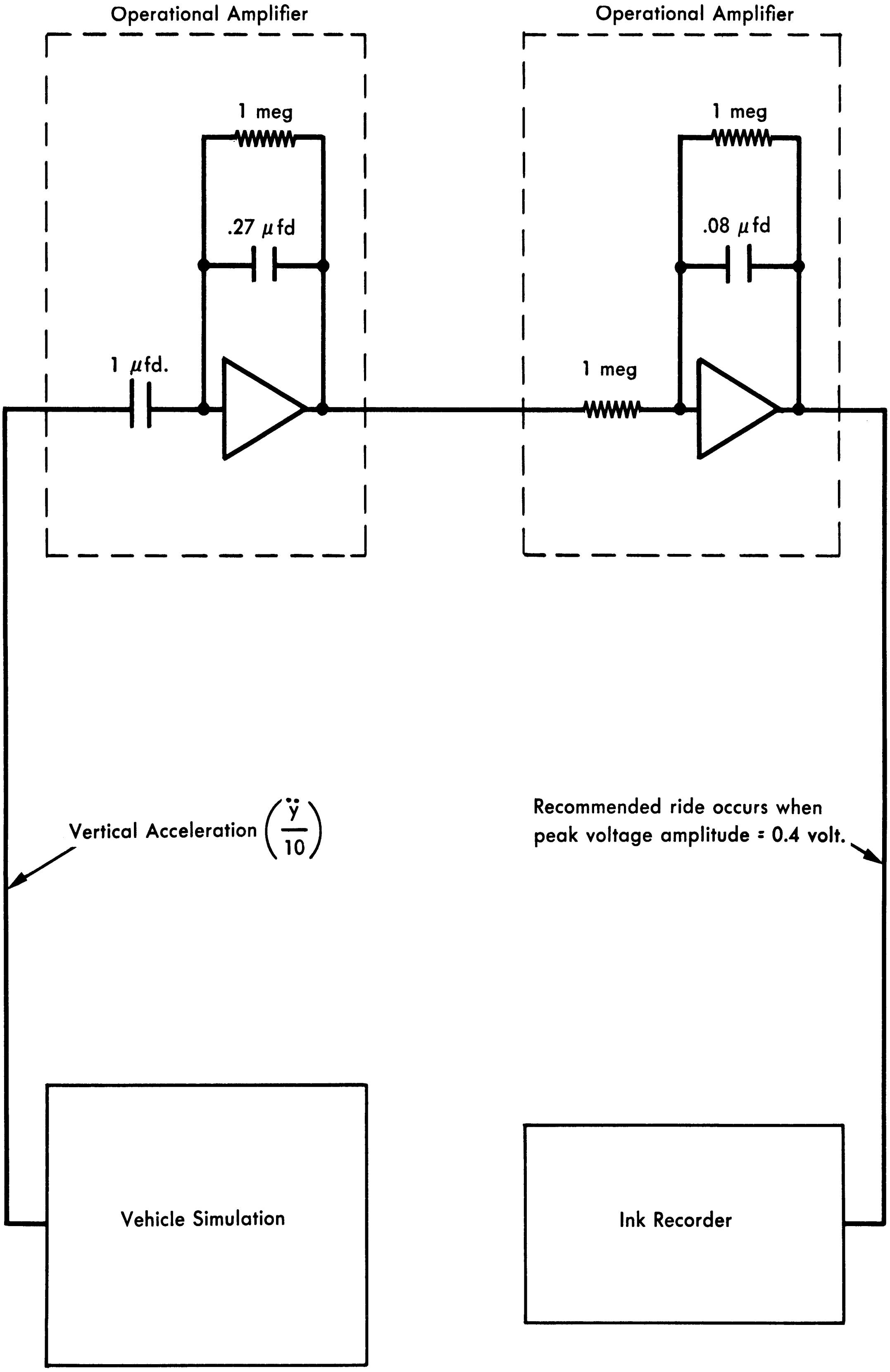


FIG. C-1 RIDE INDEX CIRCUIT

APPENDIX D

VISUAL DISPLAY OF TANK SIMULATION

Graphic records obtained from a simulation by such devices as ink recorders or plotting tables give accurate data on the values of simulated variables as a function of time. However, for complicated systems such as the suspension system of a tank, it is frequently difficult for the engineer to visualize the performance of the simulated system in physical terms. A visual display which presents the physical action of the system as it is simulated on the computer can be of great usefulness in this respect. Such a display would also be very useful in demonstrating the essential nature of the simulation to non-technical personnel.

To provide such a device, an electronic system was developed which presented on a cathode-ray tube a picture showing a representation of the elevation view of the hull and each of the road wheels of a tank as it goes over a bump. The frontispiece shows a series of photographs of this cathode-ray tube display as the simulated tank is passing over a bump.

A block diagram of the components involved in producing the display is shown in Figure D-1. The simulated M47 tank was run over a simulated 12-inch square at varying speeds. Voltages representing the vertical position of each wheel and the vertical position and pitch angle of the hull as functions of time were available as outputs from the computer. As shown in the block diagram, a dual-beam oscilloscope was used in the display. One gun was used to paint the wheels and the other gun was used to paint the hull.

The rectangle representing the hull was painted on the scope in the following manner: the simulated voltage representing the pitch angle, θ, was converted by means of a vibrator and filter to a 60-cps a-c voltage whose amplitude corresponded to the absolute magnitude of θ and whose phase corresponded to the sign of θ. This a-c voltage was routed to a summing amplifier where it was added to the d-c voltage from the suspension simulation representing y_0, vertical displacement of the C. G. If the sum of these two voltages were fed to the vertical deflection plates

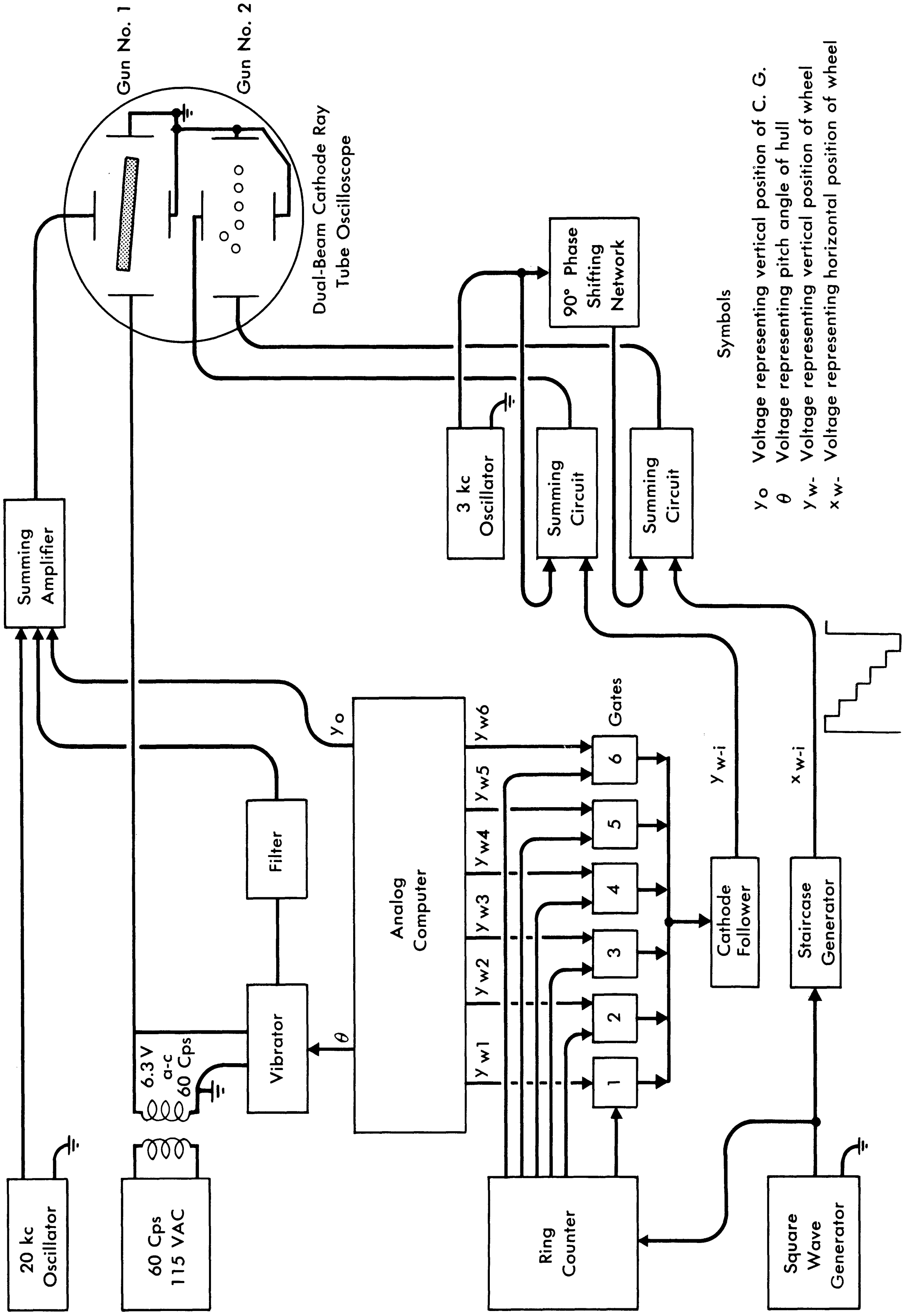


FIG. D-1 BLOCK DIAGRAM OF TANK SIMULATION DISPLAY of the cathode-ray tube and the 60-cps signal from the transformer were fed to the horizontal deflection plates, the result would be a straight line on the scope whose pitch and vertical displacement would match that of the simulated hull. In the display this line has been expanded vertically to give the appearance of appreciable height of the hull by adding to the y_o and θ signals an additional 20-kc signal.

The six wheels of the tank were produced by the second gun of the oscilloscope. Pairs of time-varying voltages were fed into the horizontal and vertical deflection plates, respectively, of the cathode-ray tube in time sequence at a frequency high enough to avoid flicker. Each wheel was represented by a circle whose center was fixed in the x direction and variable in the y direction, with the y position being made to follow one of the outputs of the computer. To produce a circle, the horizontal deflection plate received a fixed voltage corresponding to the x position of the wheel, x_{w-i}, plus a modulation voltage consisting of a 3-kc sine wave, while the vertical deflection plate received a voltage corresponding to the vertical displacement of the road wheel, y_{w-i}, as a function of time plus a modulation voltage consisting of a 3-kc sine wave voltage 90 degrees out of phase with that sent to the horizontal deflection plate. The sine-wave modulation produced a circle of appropriate diameter whose center moved in a vertical direction only.

Since all six wheels must be presented simultaneously on the tube, it is necessary to gate appropriate voltages corresponding to each of the six road wheels into the deflection plates of the cathode-ray tube in rapid succession. Because the size of all road wheels is identical, the gating process could be confined to the voltages representing the coordinates of the centers of the wheels. The ring counter sequentially opened a series of gates such that voltages from the suspension simulation representing the vertical positions of the individual wheels were applied to the y axis of the cathode-ray tube. The oscillator which drove the ring counter also drove a staircase generator, and the staircase voltage was applied to the x axis of the cathode-ray tube. This voltage fixed six spots equally spaced in the x direction.

The hull and the wheels represented in this way on the oscilloscope made it possible to observe the behavior of eight time-varying quantities simultaneously in a manner exactly corresponding to their relationship in the real physical system.

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