Simulated Tractor Chassis Suspension System

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ABSTRACT

THE development of a chassis suspension system for an agricultural tractor has been described to improve the operator ride comfort. Computer simulation techniques were used to formulate the tractor models and to compute the natural frequencies and frequency response of the models, as well as the RMS acceleration response, to evaluate the effect of the suspension system and cab position on operator ride comfort.

INTRODUCTION

To increase the travel speed of an agricultural tractor for improved productivity and to reduce operator exposure to vibration, the dynamic behavior of the vehicle must be modified to improve the operator ride comfort. Up to now, the vehicle travel speed has been limited by tolerance of the operator to a rough ride rather than the capability of the vehicle. The vehicle ride quality depends upon the terrain surface roughness, the travel speed, and the vehicle geometry and suspension characteristics.

FEATURES OF THE SIMULATED SUSPENSION SYSTEM

Within the next 10 yr, farming operations may be based on tractor working speeds of 13 to 16 km/h [Reichenberger, 1980]. In all probability, an alternative suspension means, besides a seat suspension, will be required to improve the operator ride comfort. Based upon the review of the ride comfort means [Claar et al., 1980a], a chassis isolation system was selected for further investigation as a means to attenuate ground-induced motion between the axles and the chassis. Several desirable features of the concept were identified as:

1 The suspension will attenuate the bounce and pitch motion of the tractor.

2 The suspension will allow the chassis, cab, and operator platform to be included as the sprung mass. The sprung to unsprung mass ratio will thus be greater than the ratio for a cab suspension.

3 The suspension ought not to impart a sense of "insecurity" to the operator as might a cab suspension [Stayner, 1974]. This phenomenon arises because some operators do not like the large relative movement between the cab and tractor at certain times.

4 With this suspension, the seat suspension will not need to provide a large relative movement in excess of 130 mm between the seat and operator controls. Most current seat suspensions need a large relative movement [Stayner et al., 1975].

5 The suspension will reduce the loads transmitted directly to the chassis structure, cab, and operator platform.

The design of the tractor chassis suspension system has been described by Claar et al. [1980b].

OBJECTIVES

The following objectives are specified for evaluating the simulated suspension system:

1 To evaluate the ride comfort of the suspended tractor in terms of the exposure and fatigue-decreased proficiency criteria as specified in the ISO Standard 2631 on whole-body vibraiton and the SAE Recommended Practice J1013 for evaluating agricultural seat suspensions.

2 To consider only passive springing and damping elements.

3 To permit an axle displacement equal to ± 100 mm in relation to the static equilibrium position of the tractor.

4 To consider the conventional 3-point hitch mechanism for controlling the ground-engaging implement position and regulating the implement depth and attitude during operation.

5 To use a conventional 2-wheel drive agricultural tractor.

6 To provide a "constant" chassis position for varying static loads on settling down in the equilibrium of the tractor.

COMPUTER SIMULATION DESIGN APPROACH

At the design concept stage, analytical computer simulation techniques are especially useful for evaluating agricultural tractor ride characteristics. This design approach requires the quantitative description of the following ride analysis components:

1 Description of a truly representative track or terrain for exciting the vehicle.

2 Description of the vehicle as a system of components.

3 Calculation of the vehicle dynamic response.

4 Conversion of the vibrational characteristics on the operator to human-stress parameters.

To evaluate the effect of operator cab position and suspension system parameters on tractor ride characteristics, the generalized mechanical systems simulation program IMP (Integrated Mechanisms Program) was used, [Sheth and Uicker, 1972]. Nine tractor

Article was submitted for publication in March 1981; reviewed and approved for publication by the Power and Machinery Division of ASAE in December 1981. Presented as ASAE Paper No. 80-1565.

Journal Paper No. J-10142 of the Iowa Agriculture and Home Economics Experiment Station, Iowa State University, Ames. Project No. 2343

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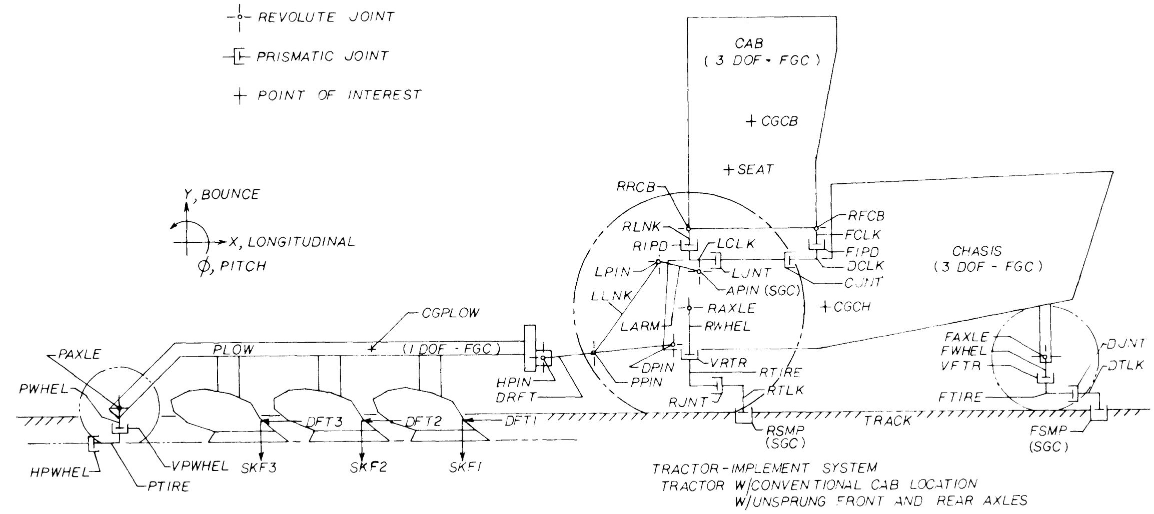

FIG. 1 Conventional cab position tractor with plow model.

configurations were formulated according to the IMP modeling procedure. Each model had one of three operator cab positions and one of three suspension systems. The three operator cabs were positioned on the tractor:

1 At 455 mm ahead of the rear wheel centerline, i.e., conventional cab position.

2 At 1270 mm ahead of the rear wheel centerline or at the midpoint of the wheelbase, i.e., mid-chassis cab position.

3 At 455 mm behind the front wheel centerline, i.e., forward cab position.

The three suspension systems were:

1 Unsprung front and rear axles.

2 Unsprung rear axle and sprung front axles.

3 Sprung front and rear axles.

In addition, these same nine tractor model configurations were formulated with a semi-mounted moldboard plow.

The fore-aft and lateral motion, as well as the vertical motion, are important in the evaluation of operator ride comfort; thus, three dimensional computer models should be used for this evaluation. For initial evaluation, however, two-dimensional models were developed to evaluate the effects of design changes only on the fore-aft and vertical operator motions.

One of nine IMP tractor-plow models is shown in Fig. 1. One may envision the remaining eight tractor-plow models and the nine tractor models. To constrain the movement between the model's links, the following two joint types were used:

1 prismatic joint to allow translation; and

2 revolute joint to allow rotation.

The formulated mathematical models have the following degrees of freedom:

1 three degrees of freedom (two traqnslation and one rotation) for the chassis;

2 three degrees of freedom (two translation and one rotation) for the cab, thus including the cab mounts effects;

3 a rotational degree of freedom for the moldboard plow;

4 a rotational degree of freedom for the implement hitch to provide implement position control;

5 a rotational degree of freedom for the front suspension system linkage as shown in Fig. 2;

6 a rotational degree of freedom for the rear suspension system linkage as shown in Fig. 3; and

7 two specified translational degrees of freedom for inputting the terrain excitation to the front and rear tractor wheels.

The tractor and plow tires are represented as a set of vertical prismatic joints and fore-aft prismatic joints.

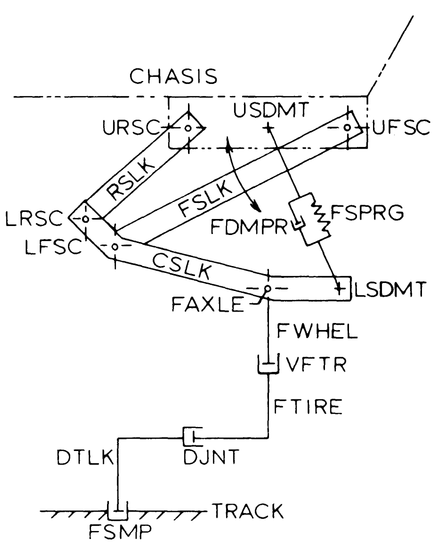

FIG. 2 Front axle suspension system with wheel model.

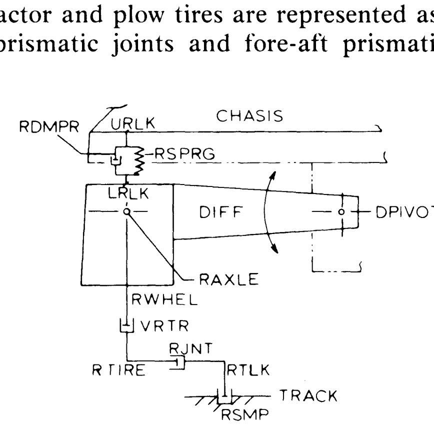

FIG. 3 Rear axle suspension system with wheel model.

The orientation of these joints allows the tractor chassis to have bounce and fore-aft motions. Each of these joints allows a translational spring and damper to act within it. The revolute joints allow the tractor chassis and the plow to have a rotational or pitching motion.

A set of specified vertical prismatic joints are used to input the time-varying track excitation to the tractor wheels. At a constant velocity of 12.5 km/h, the ISO 5007 right smooth track is passed under each model, and the joints are displaced at an amplitude equal to the track elevation at that particular instant in time. The track input is phased according to each model's wheel spacing.

The cab isomount pads are represented as a set of vertical prismatic joints and fore-aft prismatic joints. The orientation of these joints allows the cab to have bounce and fore-aft motions. Each prismatic joint allows a translational spring and damper to act within it. The fore-aft and vertical springs and dampers have equal spring and damping rates. The revolute joints allow the tractor cab to have a rotational or pitching motion.

The mathematical tractor and tractor-implement models implemented for analysis require data, such as geometric dimensions, inertial properties, etc., to describe the tractor chassis and cab, the implement hitch system, the tires, the cab isomount pads, the front and rear suspension systems, the plow, and the trailers. These data have been detailed by Claar et al. [1980b].

The suspension design parameters were calculated for the three tractor configurations with the front and rear axle suspension systems using automotive suspension design theory. The three tractor configurations, with only a front axle suspension system, have the same front suspension parameters as the fully suspended tractor configurations. No implement drawbar loads were considered as part of the unsprung rear axle mass.

A wide variety of suspension parameters may be evaluated to arrive at an optimum set of suspension parameters to improve the operator ride comfort. Because of computer bedget constraints, five suspension parameter changes were considered:

1 A rear suspension ride rate 20 percent greater than the front suspension ride rate with the front and rear suspension damping rates one-fourth of the sprung mass critical damping rates, respectively. These parameters are used as a starting point for suspension design [D. E. Cole. 1972. Elementary vehicle dynamics. Class notes. ME 498. The University of Michigan, Ann Arbor, and S. Mola. 1974. Fundamentals of vehicle dynamics. Class notes. General Motors Institute, Flint].

2 A rear suspension ride rate 20 percent greater than the front suspension ride rate with front and rear suspension damping rates one-eighth of the sprung mass critical damping rates, respectively.

3 A rear suspension ride rate 40 percent greater than the front suspension ride rate with front and rear suspension damping rates one-fourth of the sprung mass critical damping rates, respectively.

4 A rear suspension rate determined from the tire stiffness rate for the implement load acting on the tractor, a rear suspension ride rate 20 percent greater than the front suspension ride rate, and front and rear suspension damping rates one-fourth of the sprung mass critical damping rates, respectively.

5 A rear suspension ride rate 20 percent greater than the front suspension ride rate with a front suspension

TABLE 1. MODEL DESIGNATION CODE

T:	TRACTOR
S:	TRACTOR-PLOW SYSTEM
C:	CONVENTIONAL CAB POSITION
M:	MIDCHASSIS CAB POSITION
F:	FORWARD CAB POSITION
U:	UNSPRUNG FRONT AND REAR AXLES
F:	SPRUNG FRONT AXLE ONLY
B:	SPRUNG FRONT AND REAR AXLES
—:	$K_{FS} = 1/8\ K_{RT}$; $K'_{RS} = 1.2\ K'_{FS}$; $C = 1/4\ C_C$
D:	$K_{FS} = 1/8\ K_{RT}$; $K'_{RS} = 1.2\ K'_{FS}$: $C = 1/8\ C_C$
R:	$K_{FS} = 1/8\ K_{RT}$; $K'_{RS} = 1.4\ K_{FS}$; $C = 1/4\ C_C$
S:	$K_{FS} = 1/8\ K_{RT}$; $K'_{RS} = 1.2\ K'_{FS}$; $C = 1/4\ C_C$; $K_{RS} = f(K_{RTP})$
X =	$K_{FS} = 1/8\ K_{RT}$; $K'_{RS} = 1.2\ K'_{FS}$; $C = 1/4\ C_C$; $m = m_{UR}$

where:

K_{FS} = front suspension spring rate
K_{RS} = rear suspension spring rate
K_{RT} = rear tire spring rate
K_{RTP} = rear tire spring rate/include implement load
K'_{FS} = front suspension ride rate
K'_{RS} = rear suspension ride rate
C = damping rate
C_C = critical damping rate
m = unsprung mass
m_{UR} = unsprung rear axle mass

damping rate one-fourth of the sprung mass critical damping rate and a rear suspension damping rate one-fourth of the unsprung mass critical damping rate.

A designation code has been developed to describe the model configuration according to its cab position, suspension type, and variation of suspension parameters. This designation code is presented in Table 1. For example, the code TCBD represents a tractor with a conventional cab position, suspended front and rear axles, and the following suspension parameters: A front suspension spring rate one-eighth of the rear tire spring rate, a rear suspension ride rate 20 percent greater than the front suspension ride rate, and a front and rear suspension damping rate one-eighth of the sprung mass critical damping rate.

The IMP system was used for computing the system transfer functions for each of the 24 tractor and 24 tractor-plow models. A post-processor program was used for computing the dynamic response by combining the transfer functions with the digitized ISO 5007 smooth track input excitation. With the dynamic response of the tractor system, the weighted RMS acceleration ride number was computed in accordance to SAE J1013. The post-processor program has been described by Claar et al. [1908b].

RESULTS AND DISCUSSION

Forty-eight models were evaluated: three unsprung tractor models, three unsprung tractor-plow models; six suspended front axle tractor models; six suspended front axle tractor-plow models; 15 fully suspended tractor models; and 15 fully suspended tractor-plow models. The three unsprung tractor and three unsprung tractor-plow models were used as a base to evaluate the ride comfort of the 21 suspended tractor and suspended tractor-plow models.

In turn, the post-processor program used these system transfer function results from IMP to calculate the frequency response plots, the RMS acceleration ride numbers, and the RMS acceleration response plots for each model. Fig. 4 shows the acceleration frequency-domain

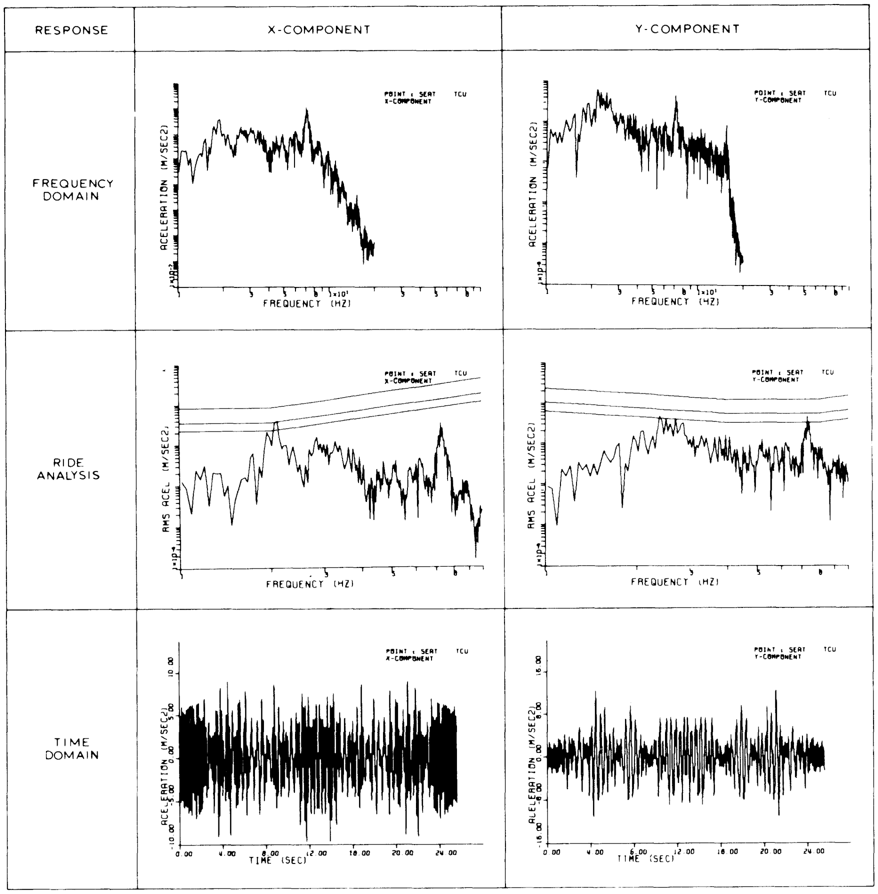

FIG. 4 Acceleration response of the tractor seat point in the frequency and time domains.

response for the variable point SEAT for the conventional tractor model.

On the basis of the RMS ride numbers, as shown in Figs. 5 and 6, and the RMS acceleration response plots for the point SEAT, the following conclusions and observations can be made:

1 The chassis suspension system with the appropriate suspension parameters does reduce vibration levels and improve operator ride comfort.

2 When the results for the three unsuspended tractor configurations alone are compared, the tractor with the conventional cab position provided the best overall ride comfort.

3 When the results for the three suspended front axle tractor configurations alone are compared, all three tractors with their suspension system parameters provided approximately the same ride comfort.

4 When the results for the three fully suspended tractor configurations alone are compared, the tractor with the conventional cab position and its suspension system parameters provided the best ride quality.

5 When the results for the three unsuspended tractor configurations with attached plows are compared, the tractor with the conventional cab position provided the best ride comfort.

6 When the results for the suspended front axle tractor configurations with attached plows are compared, the tractor with the forward cab position and its suspension

	Tractor, T			Tractor-plow, S			Suspension Parameters
	Conventional cab, C	Mid-Chassis cab, M	Forward cab, F	Conventional cab, C	Mid-chassis cab, M	Forward Cab, F	
Unsprung	0.33 (TCU)	0.31 (TMU)	0.31 (TFU)	0.27 (SCU)	0.27 (SMU)	0.30 (SFU)	U
Sprung Front Axle Only	0.24 (TCF)	0.25 (TMF)	0.24 (TFF)	0.17 (SCF)	0.19 (SMF)	0.18 (SFF)	F
	0.47 (TCFD)	0.45 (TMFD)	0.47 (TFFD)	0.43 (SCFD)	0.44 (SMFD)	0.58 (SFFD)	FD
Both Axles Sprung	0.15 (TCB)	0.16 (TMB)	0.17 (TFB)	0.16 (SCB)	0.12 (SMB)	0.18 (SFB)	
	0.22 (TCBX)	0.24 (TMBX)	0.22 (TFBX)	0.16 (SCBX)	0.15 (SMBX)	0.16 (SFBX)	BX
	0.14 (TCBD)	0.16 (TMBD)	0.17 (TFBD)	0.45 (SCBD)	0.40 (SMBD)	0.67 (SFBD)	BD
	0.14 (TCBR)	0.17 (TMBR)	0.16 (TFBR)	0.44 (SCBR)	0.40 (SMBR)	0.66 (SFBR)	BR
	0.14 (TCBS)	0.16 (TMBS)	0.16 (TFBS)	0.44 (SCBS)	0.40 (SMBS)	0.66 (SFBS)	BS

RMS Acceleration Number (m/sec^2)

FIG. 5 Longitudinal RMS acceleration ride numbers for the point SEAT.

	Tractor, T			Tractor-plow, S			
	Conventional cab, C	Mid-Chassis cab, M	Forward cab, F	Conventional cab, C	Mid-Chassis cab, M	Forward cab, F	Suspension Parameters
Unsprung	0.72 (TCU)	0.78 (TMU)	1.00 (TFU)	0.58 (SCU)	0.62 (SMU)	0.82 (SFU)	U
Sprung Front Axle Only	0.61 (TCF)	0.66 (TMF)	0.63 (TFF)	0.53 (SCF)	0.53 (SMF)	0.44 (SFF)	F
	4.41 (TCFD)	3.68 (TMFD)	2.96 (TFFD)	4.34 (SCFD)	3.59 (SMFD)	2.92 (SFFD)	FD
Both Axles Sprung	0.33 (TCB)	0.41 (TMB)	0.61 (TFB)	0.26 (SCB)	0.27 (SMB)	0.45 (SFB)	
	0.79 (TCBX)	0.69 (TMBX)	0.81 (TFBX)	0.53 (SCBX)	0.37 (SMBX)	0.47 (SFBX)	BX
	4.39 (TCBD)	4.31 (TMBD)	4.44 (TFBD)	1.68 (SCBD)	1.19 (SMBD)	1.99 (SFBD)	BD
	4.18 (TCBR)	4.31 (TMBR)	4.22 (TFBR)	1.69 (SCBR)	1.19 (SMBR)	2.00 (SFBR)	BR
	4.18 (TCBS)	4.30 (TMBS)	4.38 (TFBS)	1.67 (SCBS)	1.19 (SMBS)	2.00 (SFBS)	BS

RMS Acceleration Number (m/sec^2)

FIG. 6 Vertical RMS acceleration ride numbers for the tractor and tractor-plow models.

parameters provided the best ride comfort.

7 When the results for the fully suspended tractor configurations with attached plows are compared, the tractor with the midchassis cab position and its suspension system parameters provided the best ride comfort.

8 The moldboard plow provided an effective damping means to reduce the vertical only vibration levels imposed on the operator, however, the vibration excitations due to the plow-soil interaction were not considered in this study.

For an agricultural tractor, the bounce and pitch modes of vibration are nearly always coupled. The degree of coupling is variable and is dependent on the tractor wheelbase, the tire stiffness, the suspension system parameters, the centers of gravity and inertia of the tractor and the attached implement. The location of the pitch center is an important consideration for ride comfort. If the operator is located at the pitch center, the operator would experience little vibration despite the fact that the tractor is pitching around him. On the other hand, the further the operator is located away from the pitch center, the worse will be the vibration levels experienced by him. On most tractors, the operator will be located always above and usually away from the pitch center. Therefore, the pitching motion will cause the operator to move in both the longitudinal and vertical directions.

Crolla [1976] has also investigated the effect of a tractor pulling a moldboard plow. It was shown that the operator ride vibration in the pitch mode was heavily damped due to the action of the soil forces acting on the plow. The vibration levels were lower when plowing than when the tractor was operating alone.

From the computer simulation analyses, the ride vibration levels were generally lower when the tractors were attached to plows than when operated alone. The phenomena are observed in the RMS acceleration ride numbers that were computed for the tractor alone and tractor-plow system configurations.

Based on the RMS acceleration ride numbers computed for the tractor and tractor-plow configurations, and the results presented by Crolla [1976], the following conclusions and observations can be made:

1 For the tractor alone configurations, the tractor with the conventional cab position provided the best ride comfort because the operator is located approximately at the pitch center of the tractor. The chassis suspension system further improved the operator ride comfort.

2 For the tractor-plow configurations, the tractor with the forward cab position and a suspended front axle, and the tractor with a midchassis cab position and the fully suspended chassis provided the best ride comfort.

3 For the tractor-plow configurations, the pitch centers of the tractors were toward the front of the tractors. Hence, the ride comfort for the tractors with midchassis and forward cab positions was better because the operator was positioned nearer to the pitch centers of these tractors.

SUMMARY

The development of a chassis suspension system for an agricultural tractor has been described to improve the operator ride comfort. Twenty-four tractor and 24 tractor-plow models were formulated with the IMP analysis program to evaluate the effect of a suspension system and its parameters and the effect of the position of the operator cab. RMS ride numbers and RMS acceleration plots provided the basis to evaluate the effect of the suspension system and cab position on operator ride comfort. The IMP simulation and post-processor programs provided the natural frequencies, the frequency response magnitude and phase angle plots, and the mode shapes of the vibration modes to gain further insight to evaluate the effect of ride comfort. The generalized mechanical systems simulation program, IMP, has proven to be a valuable tool for evaluation of the suspension parameters affecting operator ride comfort.

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