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DEVELOPMENT OF A STABILITY MONITOR/CONTROLLER FOR MATERIAL HAND--ETC(U)
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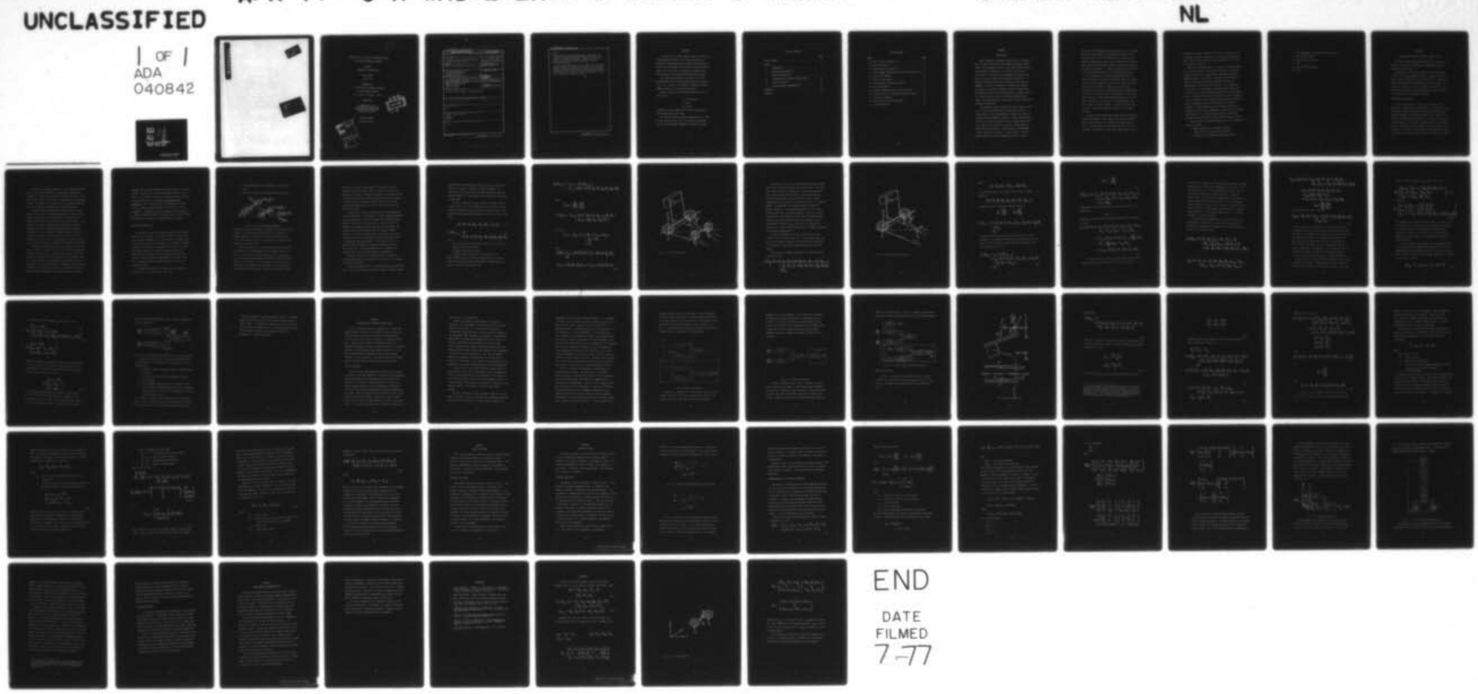

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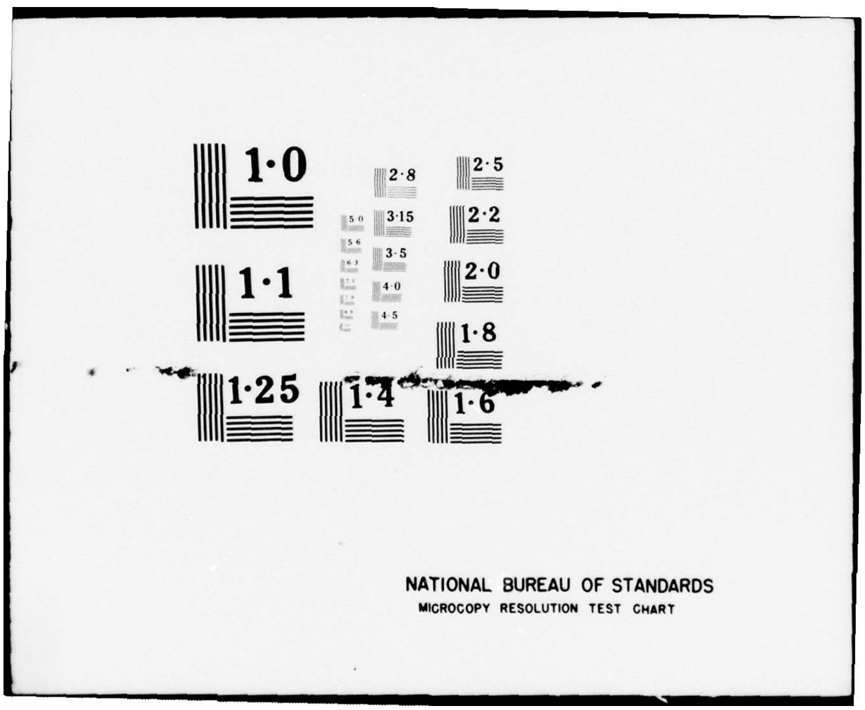
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DEVELOPMENT OF A STABILITY MONITOR/CONTROLLER

FOR MATERIAL HANDLING EQUIPMENT

PHASE I

THEORETICAL INVESTIGATION

FINAL REPORT

April, 1977

PREPARED FOR

U.S. ARMY MOBILITY EQUIPMENT RESEARCH

AND DEVELOPMENT COMMAND

Fort Belvoir, Virginia 22060

PREPARED BY

PERSONNEL OF THE
FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
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Abstract

The purpose of this two-phased study is to investigate the feasibility of developing a stability monitor or controller suitable for material handling equipment (MHE). Recommendations were to be made as to the optimal method (Phase I). Phase II required the design, fabrication, and testing of an exploratory system to demonstrate feasibility.

This report documents Phase I of the investigation. A mathematical model of a typical vehicle (warehouse lift truck) is derived. This information provides a foundation for proposing two basic approaches. Mathematical and experimental investigations reveal that flaws existed in both of these, and a third approach is proposed which is a combination of the previous two. Two types of controls are recommended - one of which is effectively a monitor-type system.

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FOREWORD

The report is the Final Technical Report for Phase I of Contract DAAG53-76-C-0144. This report discusses the theoretical investigation which provided the mathematical basis for the development of a Stability Monitor/Controller. Phase II of the contract is covered in a separate report which documents the system fabricated and test results.

This study was conducted by the staff of the Fluid Power Research Center, Oklahoma State University, directed by Dr. E. C. Fitch, Jr., Mr. Robert L. Decker was Program Manager and Mr. R. L. Brown Project Engineer.

Project Personnel contributing major efforts to the project were:

C. A. Whitehead

B. Burns

B. Talbot

In addition, various members of the Fluid Power Research staff contributed to this study.

The Contract Officers Technical Representative for this contract was Ms. Claire Orth, U.S. Army Mobile Equipment Research and Development Command, Fort Belvoir, Virginia.

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CHAPTER I

INTRODUCTION

Mobile equipment is inherently susceptible to the damage or loss of control resulting from an unstable operating configuration. This instability can be of the form of impaired vehicle controllability or complete loss of control experienced by the operator. In either case, this dangerous situation could result in injury or death to the operator or bystanders. In addition, accidents can result in financial losses caused by damages to the vehicle and payload. Although these dangers are present in all mobile equipment, material handling equipment (MHE) poses an especially acute problem. Vehicles such as lift trucks and mobile cranes often handle extremely heavy loads. The ramifications of an accident with these loads can be particularly disastrous.

The problem of vehicle stability has been studied for some time. Studies during World War II and during the post war years resulted in the establishment of design rules which were intended to provide adequate stability. These "rules-of-thumb" typically were stated in terms of ratios of moment arms which are related to the location of the vehicle counterbalancing moment arms and load moment. Various studies were made which tended to promote the concept. In recent years, lift truck manufacturers have developed mathematical models to aid in insuring a vehicle design with adequate stability [1]. Extensive effort has been expended in the development of models which accurately predict vehicle trajectories when subjected to an unstable configuration. Some of these studies have resulted in models which are described by differential equations which have no closed form solution. In addition, parameter uncertainties result in potential difficulties in utilizing these models.

At the present time, the final determining factor in stability design for lift trucks and similar vehicles is based on the successful completion of a series of static slant table tests which are intended to simulate various situations encountered in the field. The American National Standard B56.1 [2] serves as the "test-by-fire" for warehouse lift trucks and similar vehicles. ANSI Standard B56.6 delineates the test for rough terrain-type lift trucks [3]. All manufacturers test their vehicles to these standards. A series of articles over the past 20 years has attempted to justify the slant table criterion.

Although various design rules, models and test procedures have been developed, the fact remains that material handling vehicles are susceptible to instabilities and accidents do occur. A recent study of serious personal-injury lift truck accidents revealed that the second most common cause of these accidents was tipover or rollover of the truck. (This was second to passerby struck by a moving truck.) The study also found that over a third of these accidents involved operators with one year of experience or less [4].

Based on the accident reports, the control of vehicle stability is certainly needed. This control could be of the form of a hardware controller or a monitor device allowing for operator-augmented control. This study is concerned with developing the necessary mathematical basis and algorithmic processes for implementing a stability monitor/controller. The ultimate goal of this study is to investigate and demonstrate the feasibility of a device which could be installed on vehicles in field environments. A mathematical model of the vehicle and load is developed to first explain the basic phenomena encountered and, secondly, to provide a mathematical basis for specifying the controller. Pragmatic considerations dictate that the model should be of a complexity which is readily implemented by low-cost analog or digital equipment. The suitability of the model is explored by laboratory tests on an actual vehicle (refer to Phase II report).

The evaluation of the monitor/controller is to be based on the following criteria:

1. Speed of reaction to a dangerous condition.
2. Interference with normal vehicle operation.
3. Probability of malfunction and potential effects.

4. Interchangeability on several different vehicles.

5. Ease of installation.

6. Simplicity and durability.

7. Size

8. Ease of field maintenance.

9. Cost.

CHAPTER II

DEFINING AND IDENTIFYING VEHICLE INSTABILITIES

This chapter defines vehicular instability. The modes and causes of instability are discussed, and a mathematical model is developed to relate these.

When possible, the concepts developed in the following sections will be considering MHE in general. Because of the broad scope of MHE, the application of the concepts will be limited to conventional warehouse counterbalanced lift trucks (CBLT). This does not preclude the use of these concepts on other MHE. The appropriate differences in vehicle geometry and application must be incorporated in the mathematical modeling.

THE CONCEPT OF INSTABILITY

The term "instability" is subject to a variety of interpretations. The motivation of this study is to eliminate or lessen the potential dangers associated with the operation of material handling equipment. Vehicular instability will be defined as a vehicle configuration and loading which will result in the loss of operator control. The concept of relative stability is important because it relates the relative danger to vehicle, operator, and bystanders. The degree of impairment of the capability to control the vehicle will be considered the relative stability of the vehicle.

Note that this defines instability. Only through appropriate choice of system and "outputs" can conventional systems theory definition of stability be equated to this above definition. This was selected in order to facilitate the development of a stability monitor. The needs of such a device dictate a departure from what can be considered a more mathematically pleasing approach.

Conventional MHE is designed for translation about a longitudinal vehicle axis. Uncontrolled longitudinal rotation (pitch), lateral rotation (roll) or spinning (yaw) of the vehicle are dangerous conditions and are therefore considered unstable. Pitch and roll are typical of instabilities encountered by counterbalanced vehicles in stacking operations and loaded transfer operations while yaw is common in unloaded transfer operations. If the vehicle is rigid, then pitch and roll will result in the removal of one or more tires from their point of support while yaw can occur with all four tires remaining in contact with the ground.

Instabilities can be separated into two classes - static and dynamic. Static, in general, refers to systems which possess the property in which the effect variable at any instant in time only depends on the value of the cause variable at any instant in time [5]. These have been called memoryless systems because there is effectively no time lag in the system. Dynamic systems are systems with memory: i.e., the time lags must be considered. Not only is the effect variable dependent on the present value of the cause variable, but it is also dependent on the past values of the cause variable. The vehicle is certainly a dynamical system. If the cause of an instability results in the system behaving as a static system, then this will be considered a statically-caused instability. This is characteristic of a CBLT which is slowly lifting a load and becomes unstable; the time lags are of no consequence. A dynamically caused instability is characterized by suddenly stopping a load as it is being lowered -- note that the system behaves as a dynamical system and memory is present.

QUANTIFYING INSTABILITY

At this point, consider the specific example of the counter-balance lift truck (CBLT). The system consists of a vehicle and load. Movable attachments to the vehicle will be considered part of the load and only the fixed portion of the system will comprise the vehicle. The vehicle is supported by four tires. The front tires (steer wheels) are attached to the vehicle via an axle assembly which is pivoted to allow rotation of the axle with respect to the vehicle about a longitudinal axis (refer to Fig. 2-1). This articulating axle allows the four tires to be on a non-planar surface.

A longitudinal instability of this vehicle is typified by rotation about either of two axes. These are:

1. Axis connecting the point-of-support of the two front tires.

2. The axis of rotation of the front tires (front axle).

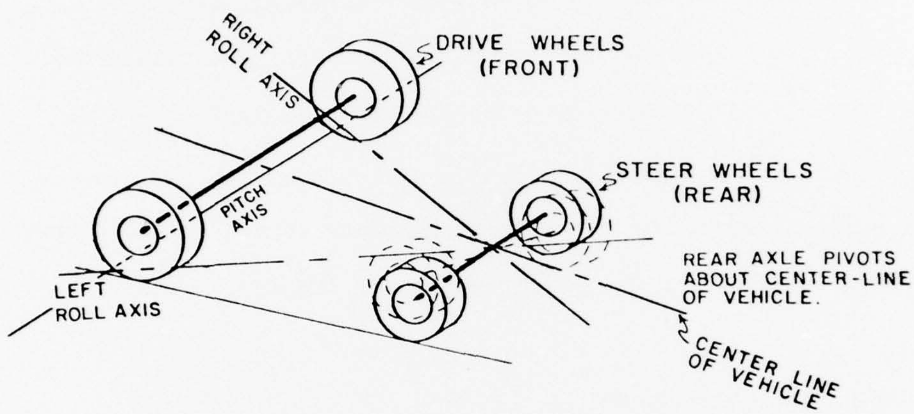

Fig. 2-1. Basic Vehicle Configuration

Rotation about both of these axes is possible if the brakes are not applied. For stacking operations, the first is considered of primary importance. This axis will be defined to be the pitch axis. Rotation about this axis will result in the removal of the reactions at the rear tires.

Lateral instabilities of the vehicle are typified by rotation about either of two axes designated left and right roll axes in Fig. 2-1. This can be more clearly understood by observing that the articulated rear axle allows for rotation of the vehicle with respect to the rear axle. In this situation, the vehicle can be considered to be supported at three points: the two front tires and the pivot point on the axle. Rotation about either of these roll axes will result in the removal of reactions at the tire opposite the axis of rotation (that is, the point of support not connected by the axis of rotation). Rotation can occur about either of these two axes until the vehicle encounters a stop on the axle. If the vehicle then continues to rotate, this will occur about an axis connecting the point of contact of the two tires on one side of the vehicle.

At this point, several observations should be made regarding the articulated axle. This results in a greatly diminished lateral stability compared to a fixed axle design. The counterbalancing moment of this vehicle is quite small as a result of this. In addition, the weight of the rear axle does not contribute to the stability because the pivot point will not support a moment. Finally, the vehicle may rotate about the roll axis, lifting the opposite front tire off the ground; but,when the stop on the axle is encountered, rotation may cease. In this case, it would require additional loading to continue rotation about the new, more stable axis. (Design techniques exist for enhancing the lateral stability [6],but these will not be considered.)

Spinning of the vehicle is defined as yaw. Although this is an important aspect of the stability of the system, it is beyond the scope of this study.

Several difficulties exist in characterizing the pitch and roll axes. First, the point of contact of the tires is actually an area and the axis passes through this area, but the exact location is difficult to specify. Secondly, the axes are dynamic. The actual position of these axes will change depending on load and vehicle orientation. This results in the necessity of specifying a nominal axis.

In order to quantify vehicle stability, consider a vehicle as shown in Fig. 2-2. The coordinate system is as shown with the origin at the intersection of the pitch and left roll axes. This system is in dynamic equilibrium; therefore, Eq. (2-1) and (2-2) are valid.

$$\Sigma \bar{F} = \bar{O} = \bar{R}_1 + \bar{R}_2 + \bar{R}_3 + \bar{R}_4 + \bar{F}_L + \bar{F}_V \qquad (2\text{-}1)$$

$$\Sigma \bar{M}_{ORIGIN} = \bar{O} \qquad (2\text{-}2)$$

$$= \bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V + \bar{P}_{R2} \times \bar{R}_2 + \bar{P}_{R3} \times \bar{R}_3 + \bar{P}_{R4} \times \bar{R}_4$$

where:

$\bar{R}_1$, $\bar{R}_2$, $\bar{R}_3$, $\bar{R}_4$ are reaction force vectors at the four tires.

$\bar{F}_{LOAD}$ is the load force vector.

$\bar{F}_V$ is the vehicle force vector.

$\bar{P} = \begin{bmatrix} x \\ y \\ z \end{bmatrix}$ are position vectors to the various forces.

Moment about the pitch axis (which coincides with the X axis) is found by performing a linear transformation on moments about the origin as shown on the next page [7].

$$\Sigma \overline{M}_{pitch} = \overline{\lambda}_{pitch} \cdot (\Sigma \overline{M}_{origin})$$
$$= \overline{\lambda}_{pitch} \cdot (\overline{P}_L \times \overline{F}_L + \overline{P}_V \times \overline{F}_V + \overline{P}_{R2} \times \overline{R}_2 + \overline{P}_{R3} \times \overline{R}_3 + \overline{P}_{R4} \times \overline{R}_4)$$

where

$$\overline{\lambda}_{pitch} = \frac{\overline{P}_{R2}}{|\overline{P}_{R2}|} = \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix}$$

$$\Sigma \overline{M}_{pitch} = \overline{\lambda}_{pitch} \cdot (\overline{P}_V \times \overline{F}_V + \overline{P}_L \times \overline{F}_L) + \overline{\lambda}_{pitch} \cdot (\overline{P}_{R2} \times \overline{R}_2)$$
$$+ \overline{\lambda}_{pitch} \cdot (\overline{P}_{R3} \times \overline{R}_3 + \overline{P}_{R4} \times \overline{R}_4)$$

Note that:

$$\overline{\lambda}_{pitch} \cdot (\overline{P}_{R2} \times \overline{R}_2) = \overline{R}_2 \cdot (\overline{\lambda}_{pitch} \times \overline{P}_{R2})$$
$$= \overline{R}_2 \cdot (\overline{0})$$
$$= \overline{0}$$

Then

$$\Sigma \overline{M}_{pitch} = \overline{\lambda}_{pitch} \cdot (\overline{P}_V \times \overline{F}_V + \overline{P}_L \times \overline{F}_L) + \overline{\lambda}_{pitch} \cdot (\overline{P}_{R3} \times \overline{R}_3 + \overline{P}_{R4} \times \overline{R}_4)$$
$$= \overline{0}$$

$$\overline{\lambda}_{pitch} \cdot (\overline{P}_{R3} \times \overline{R}_3 + \overline{P}_{R4} \times \overline{R}_4) = - \overline{\lambda}_{pitch} \cdot (\overline{P}_V \times \overline{F}_V + \overline{P}_L \times \overline{F}_L) \tag{2-3}$$

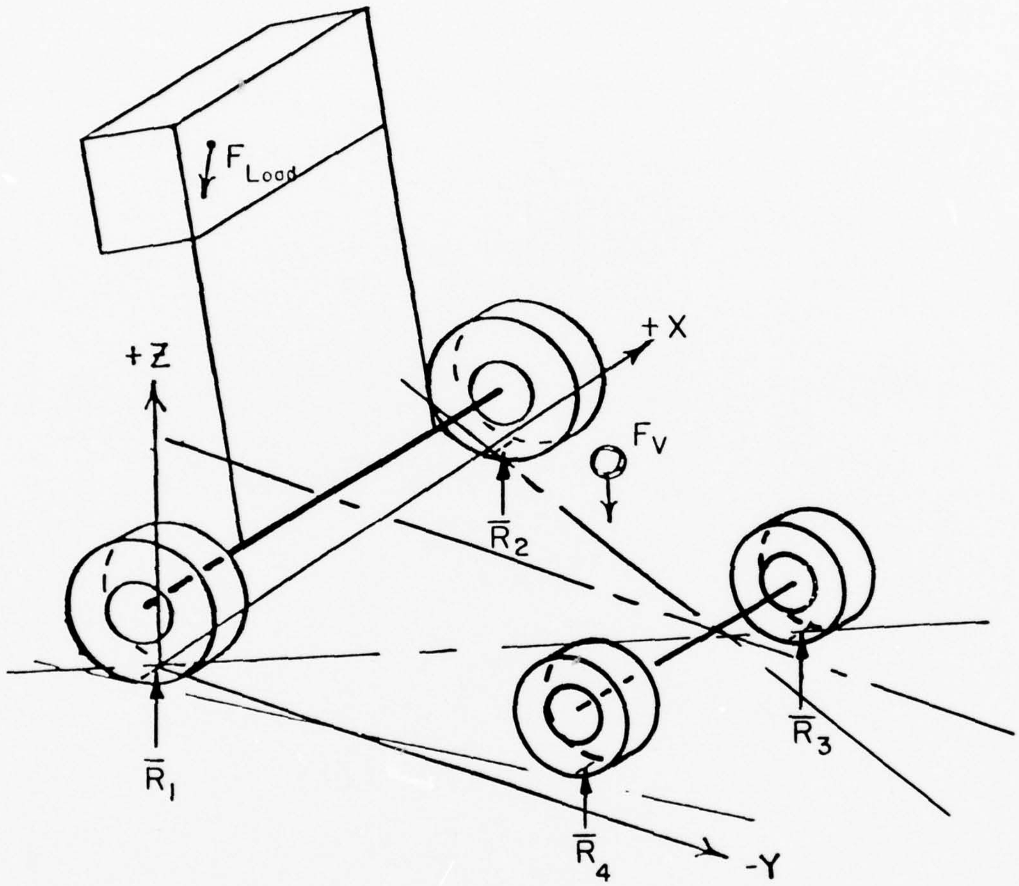

Fig. 2-2 Vehicle Reactions

The left side of Eq. (2-3) is the moment about the pitch axes which results from the reaction at the two rear tires. The right side of this equation is the negative of moment about the pitch axis caused by load and vehicle force. This reaction-action relationship quantifies the vehicle stability. For a stable configuration, the reaction at the tires must result in a negative moment. The magnitude of this moment provides a measure of relative stability. As the moment approaches zero, the vehicle stability approaches a threshold condition. Because of physical considerations, these reactions cannot support a negative moment. Similarly, the moment about the pitch axis resulting from the load and vehicle forces $[\lambda^{\cdot}_{Pitch} (\bar{P}_V \times \bar{F}_V + \bar{P}_L \times \bar{F}_L)]$ must be positive for a stable condition and approaches a threshold condition as this moment approaches zero.

In order to determine the stability about the roll axes, consider the system shown in Fig. 2-3. This system considers the pivot to be a point of support for the vehicle and is equivalent to the original system with the appropriate reaction force and moment.

This system is in dynamic equilibrium. The moment about my point $(\bar{P}_A)$ is zero.

$$\Sigma \bar{M}_{P_A} = \bar{O} = (\bar{P}_L - \bar{P}_A) \times \bar{F}_L + [(\bar{P}_V - \bar{P}_A) \times \bar{F}_V - (\bar{P}_{AXLE} - \bar{P}_A) \times \bar{F}_{AXLE}] + (\bar{P}_{R1} - \bar{P}_A) \times \bar{R}_1 + (\bar{P}_{R2} - \bar{P}_A) \times \bar{R}_2 + (\bar{P}_{R5} - \bar{P}_A) \times \bar{R}_5 + \bar{M}_{R5}$$

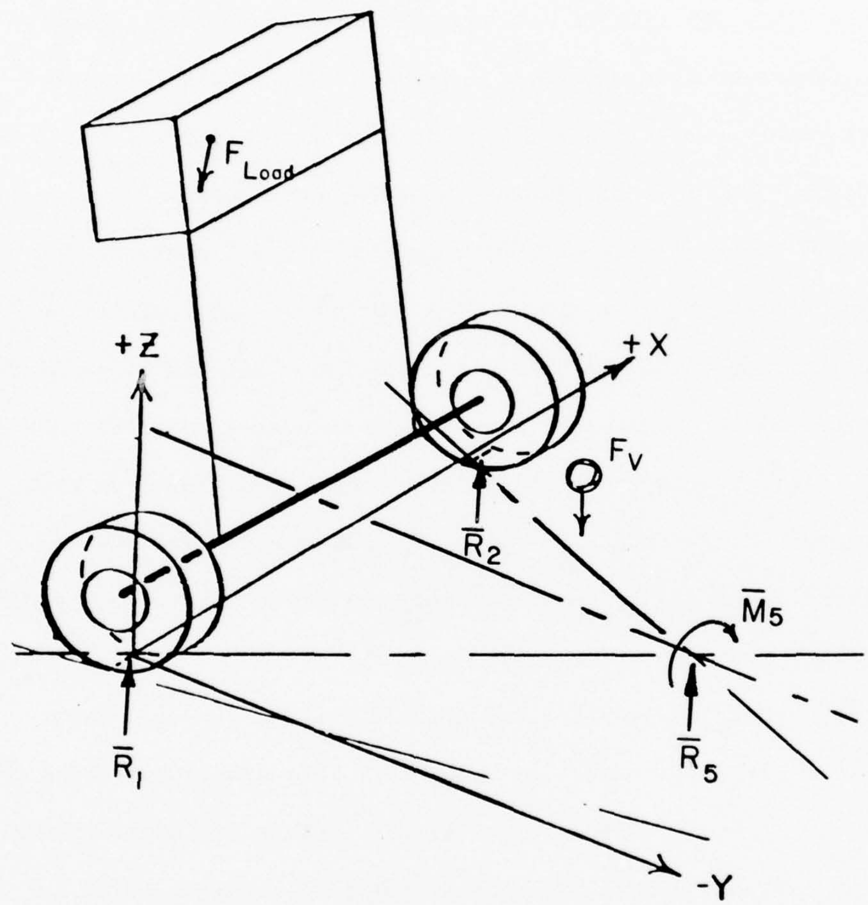

Fig. 2-3 Equivalent Vehicle Reaction

where:

$$(\overline{P}_V - \overline{P}_A) \times \overline{F}_V - (\overline{P}_{AXLE} - \overline{P}_A) \times \overline{F}_{AXLE}$$

is the moment caused by the vehicle without the axle assembly. In addition

$$\Sigma \overline{F} = \overline{0} = \overline{R}_1 + \overline{R}_2 + \overline{R}_5 + \overline{F}_L + (\overline{F}_V - \overline{F}_{AXLE})$$

The moments about the origin are found as follows:

$$\overline{P}_A = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \qquad \overline{P}_{R1} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix}$$

$$\begin{aligned} \Sigma \overline{M}_{origin} &= \overline{P}_L \times \overline{F}_L + \overline{P}_V \times \overline{F}_V - \overline{P}_{AXLE} \times \overline{F}_{AXLE} + \overline{P}_{R2} \times \overline{R}_2 + \overline{P}_{R5} \times \overline{R}_5 \\ &\quad + \overline{M}_{R5} \\ &= \overline{0} \end{aligned}$$

The moment about the left roll axis can be found by a linear combination of the moments about the origin as shown in Eq. (2-4), where $\overline{\lambda}LR$ is the direction cosine vector for the left roll axis and is defined in Eq. (2-5).

$$\begin{aligned} \Sigma \overline{M}_{LR} &= \overline{\lambda}_{LR} \cdot (\Sigma \overline{M}_{origin}) \\ &= \overline{\lambda}_{LR} \cdot (\overline{P}_L \times \overline{F}_L + \overline{P}_V \times \overline{F}_V - \overline{P}_{AXLE} \times \overline{F}_{AXLE} + \overline{P}_{R2} \times \overline{R}_2 \\ &\quad + \overline{P}_{R5} \times \overline{R}_5 + \overline{M}_{R5}) \\ &= \overline{0} \end{aligned} \qquad (2\text{-}4)$$

$$\bar{\lambda}_{LR} = -\frac{\bar{P}_{R5}}{|\bar{P}_{R5}|}$$

(2-5)

$$\begin{aligned}\Sigma \bar{M}_{LR} = \bar{O} = \bar{\lambda}_{LR} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V) - \bar{\lambda}_{LR} \cdot (\bar{P}_{AXLE} \times \bar{F}_{AXLE}) \\ + \bar{\lambda}_{LR} \cdot (\bar{P}_{R2} \times \bar{R}_2) + \bar{\lambda}_{LR} \cdot (\bar{P}_{R5} \times \bar{R}_5) \\ + \bar{\lambda}_{LR} \cdot \bar{M}_{R5}\end{aligned}$$

Assume that the center of gravity of the axle coincides with the pivot point.

$$\bar{P}_{AXLE} = \bar{P}_5$$

Using vector identity and rearranging this equation, the following is valid.

$$\begin{aligned}\bar{\lambda}_{LR} \cdot (\bar{P}_{R2} \times \bar{R}_2) &= -\bar{\lambda}_{LR} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V) + \bar{F}_{AXLE} \cdot (\bar{\lambda}_{LR} \times \bar{P}_{AXLE}) \\ &\quad - \bar{R}_5 \cdot (\bar{\lambda}_{LR} \times \bar{P}_{R5}) - \bar{\lambda}_{LR} \cdot \bar{M}_{R5} \\ &= -\bar{\lambda}_{LR} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V) + \bar{F}_{AXLE} \cdot \left(-\frac{\bar{P}_{R5}}{|\bar{P}_{R5}|} \times \bar{P}_{R5}\right) \\ &\quad - \bar{R}_5 \cdot \left(-\frac{\bar{P}_{R5}}{|\bar{P}_{R5}|} \times \bar{P}_{R5}\right) - \bar{\lambda}_{LR} \cdot \bar{M}_{R5} \\ &= -\bar{\lambda}_{LR} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V) - \bar{\lambda}_{LR} \cdot \bar{M}_{R5}\end{aligned}$$

(2-6)

The left side of Eq. (2-6) is the moment about the left roll axis caused by the right front tire. The right side of the equation is the negative of the moment resulting from the load and vehicle forces. For a stable configuration, the reaction at the tire should result in a negative moment and the moment caused by the load and vehicle forces should be positive. Unlike the previous case, the reaction at the right front tire can support a positive moment about the left roll axis. This is considered a highly undesirable situation and is therefore considered unstable. Relative stability is indicated by the magnitude of the load and vehicle moments. As these moments approach zero, the stability approaches a threshold situation.

The moment about the right roll axis can be found by first determining the moments about a point on this axis ($\bar{P}_{R2}$) then performing the appropriate transformation utilizing the direction cosine for the right roll axis.

$$\begin{aligned}\Sigma \bar{M}_{R2} = \bar{0} = {} & (\bar{P}_L - \bar{P}_{R2}) \times \bar{F}_L + (\bar{P}_V - \bar{P}_{R2}) \times \bar{F}_V \\ & - (\bar{P}_{AXLE} - \bar{P}_{R2}) \times \bar{F}_{AXLE} + (\bar{P}_{R1} - \bar{P}_{R2}) \times \bar{R}_1 \\ & + (\bar{P}_{R2} - \bar{P}_{R2}) \times \bar{R}_2 + (\bar{P}_{R5} - \bar{P}_{R2}) \times \bar{R}_5 + \bar{M}_{R5}\end{aligned}$$

$$\begin{aligned}\bar{P}_{R2} \times \bar{R}_1 = {} & \bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V + \bar{P}_{R5} \times \bar{R}_5 - \bar{P}_{AXLE} \times \bar{F}_{AXLE} \\ & + \bar{M}_{R5} - \bar{P}_{R2} \times (\bar{F}_L + \bar{F}_V + \bar{R}_5 - \bar{F}_{AXLE})\end{aligned}$$

$$\bar{\lambda}_{RR} \cdot (\bar{P}_{R2} \times \bar{R}_1) = \bar{\lambda}_{RR} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V - \bar{P}_{R5} \times \bar{R}_5 - \bar{P}_{AXLE} \times \bar{F}_{AXLE} + \bar{M}_{R5} - \bar{P}_{R2} \times (\bar{F}_L + \bar{F}_V + \bar{R}_5 - \bar{F}_{AXLE}))$$

$$= \bar{\lambda}_{RR} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V - \bar{P}_{R2} \times (\bar{F}_L + \bar{F}_V)) + \bar{\lambda}_{RR} \cdot [(\bar{P}_{R5} - \bar{P}_{R5}) \times \bar{R}_2] + \bar{\lambda}_{RR} \cdot [(\bar{P}_{AXLE} - \bar{P}_{R2}) \times \bar{F}_{AXLE}] + \bar{\lambda}_{RR} \cdot \bar{M}_{R5}$$

$$\bar{\lambda}_{RR} = \frac{(\bar{P}_{R5} - \bar{P}_{R2})}{|\bar{P}_{R5} - \bar{P}_{R2}|}$$

By substitution and vector identities:

$$\bar{\lambda}_{RR} \cdot (-\bar{P}_{R2} \times \bar{R}_1) = \bar{\lambda}_{RR} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V - \bar{P}_{R2} \times (\bar{F}_L + \bar{F}_V)) + \bar{\lambda}_{RR} \cdot \bar{M}_5 \tag{2-7}$$

As in the previous two cases, the left side of this equation is the moment about the axis resulting from the reaction at the left front tire. This moment is to be negative for a stable condition. The right side is negative of the moment about the right roll axis caused by the vehicle and load forces. The moment $\bar{\lambda}_{RP} \cdot (\bar{P}_L \times \bar{F}_L + \bar{P}_V \times \bar{F}_V - \bar{P}_{R2} \times (F_L + F_V))$ should be positive for a stable configuration. The magnitude of this moment provides an indication of relative stability. As the moment approaches zero, the vehicle stability approaches a threshold condition.

Equations (2-3), (2-5), and (2-7) were derived to describe the stability of the vehicle. Each equation serves as a measure of relative stability about one of the axes of rotation. Therefore, define the Vehicle Stability Vector ($\overline{S}$) as in Eq. (2-8).

$$\overline{S} = \begin{bmatrix} S_{pitch} \\ S_{LR} \\ S_{RR} \end{bmatrix} = \begin{bmatrix} -\overline{\lambda}_{pitch} \cdot (\overline{P}_{R3} \times \overline{R}_3 + \overline{P}_{R4} \times \overline{R}_4) \\ -\overline{\lambda}_{LR} \cdot (\overline{P}_{R2} \times \overline{R}_2) \\ -\overline{\lambda}_{RR} \cdot (\overline{P}_{R2} \times \overline{R}_1) \end{bmatrix} \quad \text{(2-8A)}$$

In addition,

$$\overline{S} = \begin{bmatrix} S_{pitch} \\ S_{RR} \\ S_{LR} \end{bmatrix} = \begin{bmatrix} \overline{\lambda}_{pitch} \cdot (\overline{P}_L \times \overline{F}_L + \overline{P}_V \times \overline{F}_V) \\ \overline{\lambda}_{LR} \cdot (\overline{P}_L \times \overline{F}_L + \overline{P}_V \times \overline{F}_V) \\ \overline{\lambda}_{RR} \cdot (\overline{P}_L \times \overline{F}_L + \overline{P}_V \times \overline{F}_V - \overline{P}_{R2} \times (\overline{F}_L + \overline{F}_V)) + \overline{\lambda}_{RR} \cdot \overline{M}_5 \end{bmatrix}$$

(2-8B)

It is important to understand that the terms in the vector $\overline{S}$ are the moments about the pitch, left roll, and right roll caused by the reaction at the point of support of the vehicle which is providing the stability to the vehicle. This vector has been defined such that the sign of each term provided determines the vehicle stability.

The basic concept in the design of a counterbalanced vehicle is to provide sufficient moment resulting from vehicle forces to counterbalance the load moment which tends to cause the vehicle to become unstable. Define the Counterbalance Moment Vector ($\overline{M}$) about some point $\overline{P}_A$ as follows:

$$\overline{M}_{P_A} = (\overline{P}_L - \overline{P}_A) \times \overline{F}_L + (\overline{P}_V - \overline{P}_A) \times \overline{F}_V \quad \text{(2-9)}$$

Eq. (2-8) can then be expressed as shown in Eq. (2-10).

$$\bar{S} = \begin{bmatrix} \bar{\lambda}_{pitch} \cdot \bar{M}_o \\ \bar{\lambda}_{LR} \cdot \bar{M}_o + \bar{\lambda}_{LR} \cdot \bar{M}_5 \\ \bar{\lambda}_{RR} \cdot \bar{M}_o + \bar{\lambda}_{RR} \cdot (\bar{P}_{R2} \times (\bar{F}_L + \bar{F}_V)) + \bar{\lambda}_{RR} \cdot \bar{M}_5 \end{bmatrix} \tag{2-10}$$

$$\bar{S} = \begin{bmatrix} \bar{\lambda}_{pitch} \cdot \bar{M}_o \\ \bar{\lambda}_{LR} \cdot \bar{M}_o + \bar{\lambda}_{LR} \cdot \bar{M}_5 \\ \bar{\lambda}_{RR} \cdot \bar{M}_{R2} + \bar{\lambda}_{RR} \cdot \bar{M}_5 \end{bmatrix}$$

Appendix A analyzes the significance of $\bar{M}_5$ on the above equation. If it is found that this term adds little significance to the determination of $\bar{S}$, which is often the case, then this equation can be written as shown in Eq. (2-11).

$$\bar{S} = \begin{bmatrix} \bar{\lambda}_{pitch} \cdot \bar{M}_o \\ \bar{\lambda}_{LR} \cdot \bar{M}_o \\ \bar{\lambda}_{RR} \cdot \bar{M}_{R2} \end{bmatrix} \tag{2-11}$$

A block diagram interpretation of Eq. (2-11) is shown in Fig. 2-4. The load forces and load position are combined to create a load moment ($\overline{LM}$). The vehicle forces and position are combined for the stabilizing moment ($\overline{SM}$). The sum of these two results in the Counterbalance Moment ($\overline{M}$). This is used to determine the vehicle stability ($\overline{S}$).

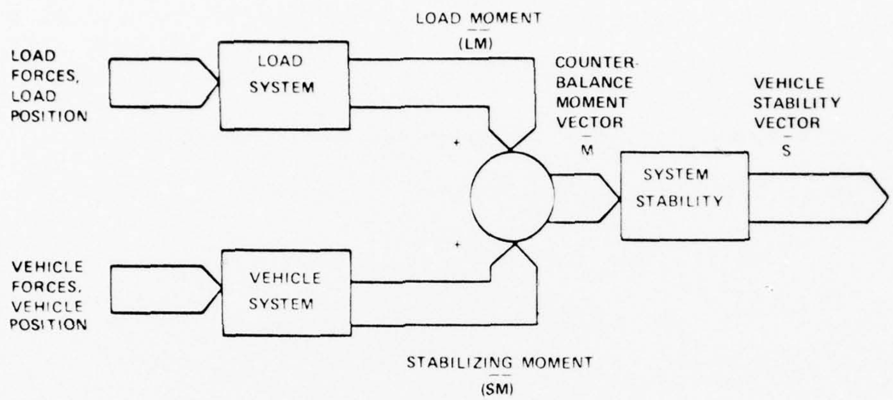

Fig. 2-4. Block Diagram Interpretation of the Vehicle Stability

Equation (2-11) will serve as a quantitative definition of vehicle stability. This definition is well suited for the following reasons:

1. Provides a measure of stability which is compatible with intuition.
2. This definition evolved from a mathematical analysis of the system.
3. The D'Almenbert approach results in a complexity which is well suited for implementation with low cost equipment.
4. This definition is well suited for integrating $\overline{S}$ into an automatic control system.

It is important to remember that the motivation of deriving this theory is not to accurately specify the trajectory of the vehicle for various inputs but to provide a theoretical basis for the for the development of a stability monitor/controller. Equipment manufacturers have developed vehicle stability models for use in design of MHE. The mathematical complexities which result would not be suited for implementation as a real-time monitor/controller. This development provides the mathematical foundation for development of a monitor/controller.

CHAPTER 3

DETERMINATION OF VEHICULAR INSTABILITIES

A device to monitor/control the stability of a vehicle must measure variables related to the vehicle's operation, operate on this information, and output the appropriate warning/control. It is necessary for the device to operate in a real-time environment. Although a wide variety of possibilities exist, the parameters measured must provide a reliable indication of stability and must be readily measured with suitable transducers. A suitable methodology must be developed to relate these measured variables to vehicle stability. This chapter utilizes the theory from the previous chapter to develop the stability monitor.

TYPES OF MONITORS

Several methods of determining the stability of the vehicle are feasible. One viable approach is to measure various parameters which will characterize the load and vehicle and calculate the stability of the vehicle. Another approach is to measure the vehicle stability directly. This is feasible since the quantitative definition was established based on physical parameters of the system. These two approaches will be defined to be the "cause method" and the "effect method," respectively. These have been named to reflect that the former measures the causes of the instability; i.e., load force, position, etc. The latter measures the actual effect of the instability.

Consider utilizing the "cause method" for a monitor/controller. A block diagram of such a monitor and the original system is shown in Fig. 3-1. The load forces and position are measured, and the load model is used to obtain an estimate of the load moment ($\overline{LM}$*). Similarly, an estimate of the Stabilizing Moment ($\overline{SM}$*) is obtained from measuring the vehicle forces and position (orientation) and inputting this into the vehicle model. Estimates of the Vehicle Counterbalance Moment ($\overline{M}$*) and Vehicle Stability Moment ($\overline{S}$*) are then obtained. This information is used for the stability evaluation. Note that this method measures the parameters affecting stability, and the vehicle stability is calculated. The accuracy of such an approach will be dependent on the accuracy and applicability of the various models.

The utilization of the "effect" method is shown in Fig. 3-2. The stability of the vehicle is measured directly, and appropriate actions are performed based on this measurement. The accuracy of this approach is merely a function of the measurement accuracy. It is not dependent on the accuracy or suitability of models as in the previous approach. This method is certainly a more direct approach.

In order to determine an optimal approach, additional factors must be considered. First, the uncertainties of measuring required paramerers must be considered. The cause approach requires the measurement of the load forces and load position. It is extremely difficult to devise a means of uniquely characterizing the load position (load center of gravity) on a real-time basis for generalized loads. A second consideration is the suitability of the transducers to a field environment. It is not sufficient that a parameter can be measured by means of conventional measurement devices, it must be measurable in a field environment. Note that the effect type system requires the measurement of $\overline{S}$. Because of the way this parameter was defined, it can be determined by simply measuring the reactions in the four tires. This could be easily accomplished in a lab by wheel scales but is of no use for a monitor for mobile vehicle operation. Attempts to measure parameters such as axle deflections which would be directly related to the parameter of interest fail to provide adequate information.

These two considerations complicate the development of a stability monitor for MHE. The approach which is seemingly superior because of simplicity and accuracy, the effect approach, fails to be suitable when the evaluation is augmented with the considerations of measurements. In fact, it can be simply stated that the problem is one of measurement. A theoretical framework has been developed which is of manageable complexity, but the parameters required by this theory cannot be reliably measured. Unfortunately, it is a problem associated with the physical system under consideration and not the theory. If the proposition that the major obstacle which must be overcome is one of measurement, then an acceptable approach is to determine parameters which can be reliably measured and then relate these to the parameters which are required by the systems proposed in Fig. 3-1 and 3-2. In fact, it is reasonable to assume that an acceptable monitor may be a hybrid system, a combination of the two systems.

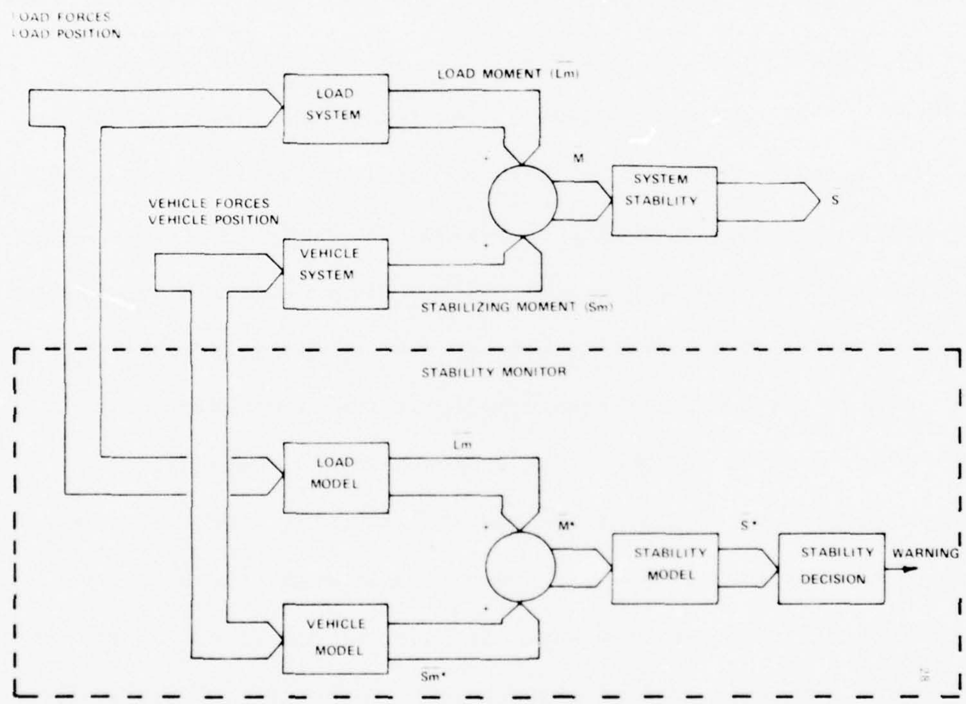

Fig. 3-1 "Cause" Type Monitor

Consider as a candidate the system shown in Fig. 3-3. This system is similar to the cause system with one difference. An estimate of the load moment $(\overline{LM*})$ is derived from various parameters of the mast assembly. If this approach is successful in obtaining the required information, an added benefit may be derived -- the model for the mast may be much simpler and more accurate than the model of the load required by the "cause" type. In particular, the load dynamics may be reflected into mast parameters such that a simple static model of the mast will be sufficient.

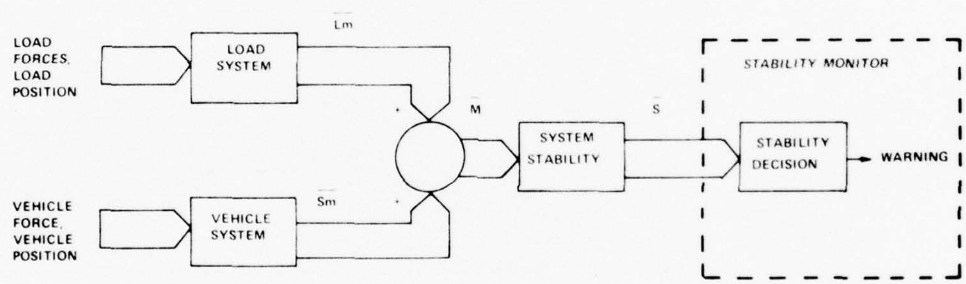

Fig. 3-2. "Effect" Type Monitor

The monitor propsed in Fig. 3-3 will be feasible if a model which relates the load moment to reaily measured mast parameters can be derived. Parameters which can reliably be measured include pressures, mast angle, and strains associated with large defro- mations. Parameters which are not readily measured are large distances (such as fork height), strains in a member which experience only small deformation, and forces internal to load bearing members.

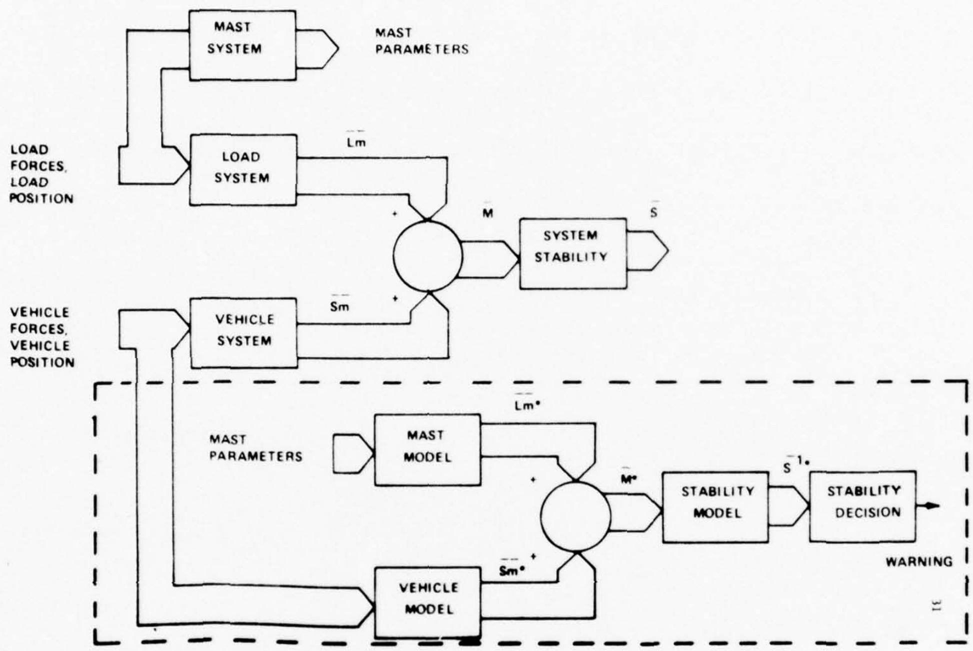

Fig. 3-3. Hybrid Stability Monitor/Controller

MONITOR DEVELOPMENT

In order to investigate the feasibility of such a system, consider the typical mast configuration shown in Fig. 3-4. The system is in dynamic equilibrium; therefore, the following is valid.

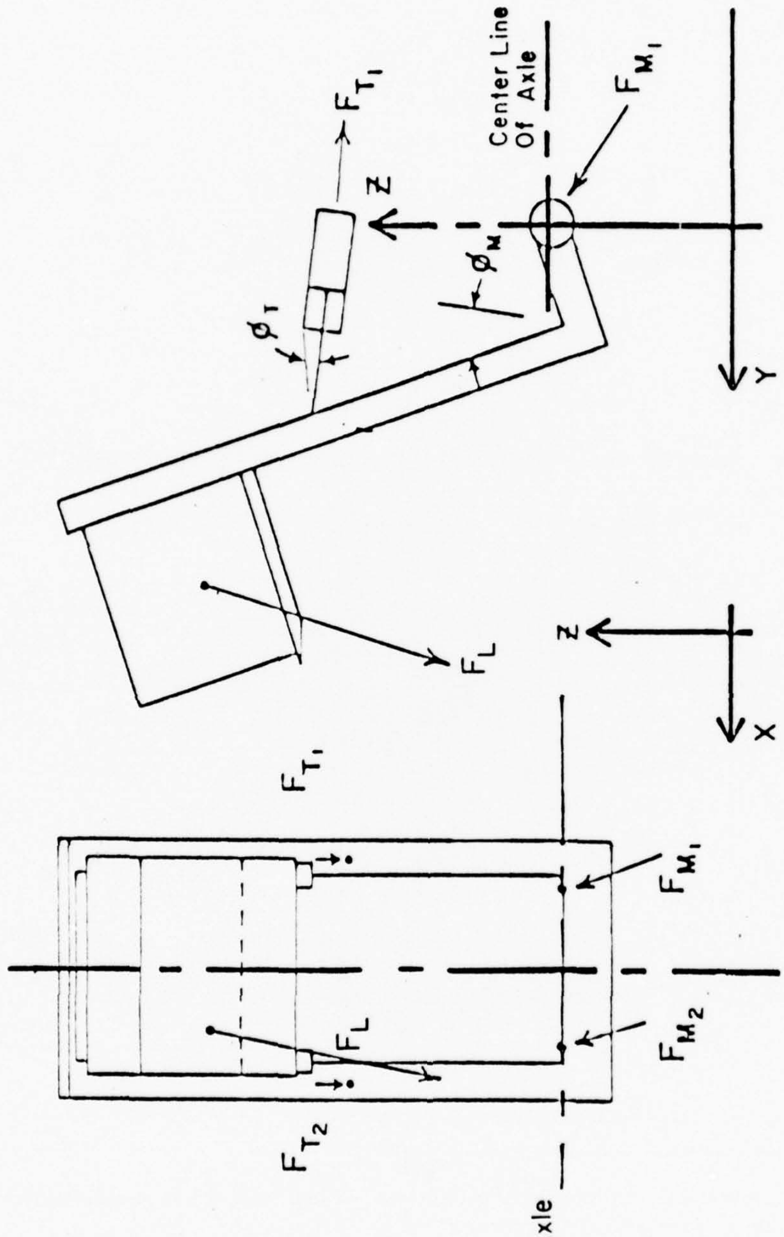

Fig. 3-4 Mast Configuration

$$\Sigma \bar{F} = \bar{0}$$

$$\Sigma \bar{M}_{P_A} = \bar{0}$$

$$= (\bar{P}_{T1} - \bar{P}_A) \times \bar{F}_{T1} + (\bar{P}_{T2} - \bar{P}_A) \times \bar{F}_{T2} + (\bar{P}_{m1} - \bar{P}_A) \times \bar{F}_{m1} + (\bar{P}_{m2} - \bar{P}_A) \times \bar{F}_{m2} + (\bar{P}_L - \bar{P}_A) \times \bar{F}_L$$

(3-1)

In order to simplify this discussion, assume that the forces in the two tilt cylinders are equal*. In addition, define two new forces as shown in Eq. (3-2).

$$\bar{F}_m = \frac{\bar{F}_{m2} + \bar{F}_{m1}}{2}$$

$$\overline{\Delta F_m} = \frac{\bar{F}_{m2} - \bar{F}_{m1}}{2}$$

(3-2)

The forces $\bar{F}_{M1}$ and $\bar{F}_{M2}$ can be expressed as shown in Eq. (3-3).

---

* The two hydraulic cylinders which comprise the tilt mechanism typically are hydraulically connected in parallel. In a static configuration this assumption is correct. Pressure drops in the connecting lines can result in this assumption being invalid in the dynamic case. This assumption does not reduce the generality of this deviation and is used only for simplification.

$$\overline{F}_{m1} = \overline{F}_m - \overline{\Delta F}_m$$
$$\overline{F}_{m2} = \overline{F}_m + \overline{\Delta F}_m$$

(3-3)

These equations and the equality of the tilt cylinder forces is used to obtain (3-4) from (3-1).

$$\overline{F}_T \triangleq \overline{F}_{T1} = \overline{F}_{T2}$$

$$\Sigma \overline{M}_{P_A} = (\overline{P}_{T1} + \overline{P}_{T2} - 2\overline{P}_A) \times \overline{F}_T + (\overline{P}_{m1} - \overline{P}_A) \times (\overline{F}_m - \overline{\Delta F}_m) + (\overline{P}_{m2} - \overline{P}_A) \times (\overline{F}_m + \overline{\Delta F}_m) + (\overline{P}_L - \overline{P}_A) \times \overline{F}_L = \overline{0}$$

$$(\overline{P}_L - \overline{P}_A) \times \overline{F}_L = -[(\overline{P}_{T1} + \overline{P}_{T2} - 2\overline{P}_A) \times \overline{F}_T + (\overline{P}_{m1} + \overline{P}_{m2} - 2\overline{P}_A) \times \overline{F}_m + (\overline{P}_{m2} - \overline{P}_{m1}) \times \overline{\Delta F}_m]$$

(3-4)

$$\Sigma \overline{F} = \overline{0} = \overline{F}_L + \overline{F}_{m1} + \overline{F}_{m2} + \overline{F}_{T1} + \overline{F}_{T2}$$
$$\overline{0} = \overline{F}_L + (\overline{F}_m - \overline{\Delta F}_m) + (\overline{F}_m + \overline{\Delta F}) + 2\overline{F}_T$$
$$\overline{F}_m = -\frac{1}{2}\overline{F}_L - \overline{F}_T$$

(3-5)

Substitute (3-5) into (3-4).

$$(\bar{P}_L - \bar{P}_A) \times \bar{F}_L = -[(\bar{P}_{T1} + \bar{P}_{T2} - 2\bar{P}_A) \times \bar{F}_T + (\bar{P}_{m1} + \bar{P}_{m2} - 2\bar{P}_A) \times (-\tfrac{1}{2}\bar{F}_L - \bar{F}_T) + (\bar{P}_{m2} - \bar{P}_{m1}) \times \Delta\bar{F}_m]$$

$$= -[(\bar{P}_{T1} + \bar{P}_{T2} - \bar{P}_{m1} - \bar{P}_{m2}) \times \bar{F}_T + \tfrac{1}{2}(\bar{P}_{m1} + \bar{P}_{m2} + 2\bar{P}_A) \times \bar{F}_L + (\bar{P}_{m2} - \bar{P}_{m1}) \times \overline{\Delta F_m}$$

$$\bar{P}_T \triangleq \bar{P}_{T1} + \bar{P}_{T2}$$

$$\bar{P}_m \triangleq \bar{P}_{m1} + \bar{P}_{m2}$$

$$P_D \triangleq \bar{P}_{m2} - \bar{P}_{m1}$$

Then

$$(\bar{P}_L - \bar{P}_A) \times \bar{F}_L = (\bar{P}_m - \bar{P}_T) \times \bar{F}_T + \tfrac{1}{2}(\bar{P}_m - 2\bar{P}_A) \times \bar{F}_L - \bar{P}_D \times \Delta\bar{F}_m \qquad (3\text{-}6)$$

Let

$$\bar{P}_A = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix}$$

Then

$$\bar{P}_L \times \bar{F}_L = (\bar{P}_m - \bar{P}_T) \times \bar{F}_T + \tfrac{1}{2}\bar{P}_m \times \bar{F}_L - \bar{P}_D \times \overline{\Delta F}_m \qquad (3\text{-}7)$$

Equation (3-7) can be used to determine the load moment ($\bar{P}_L \times \bar{F}_L$) if all of the variables in this equation are readily measured. The force in the tilt cylinder is easily determined by measuring the pressure in the cylinder. Unless $\overline{F}_L$ and $\Delta\overline{F}_M$ (or $\overline{P} \times \Delta\overline{F}_M$) can be readily determined, this method offers no improvement over the previously considered method. In further investigation of these, consider the following.

The load force, $\overline{F}_L$, must include the force resulting from weight of the load (i.e., gravitational acceleration of the mass) and external accelerations. Eq. (3-8) provides this basic relationship.

$$\overline{F}_L = w_L \overline{\lambda}_L + m_L \overline{a}_L$$

where:

W_L = weight of load

M_L = mass of the load

g = gravitational acceleration

λ_L = direction cosine vector mapping Newtonian referenced weight force into vehicle coordinates

$\overline{a}_L$ = load accelerations

Note that the direction cosine is required to map the weight force into the vehicle coordinate system. This allows the vehicle to be on non-horizontal surfaces. If the weight of the load is to be measured via a vehicle mounted transducer, then the weight will not be measured, but a component of the load force in a vehicle referenced direction will be measured. Accelerations of the load, whatever the source, can only be sustained for short periods of time. Filtering this measurement, by whatever means available, will remove the influences of the load accelerations. Therefore, consider only the component in $\bar{u}$ direction:

$$F_{Lu} = \lambda_{Lu} W_L + m_L a_{Lu}$$

where:

a_{Lu} is the component of the load acceleration in the $\bar{u}$ direction.

λ_{Lu} direction cosine relation weight force direction in a Newtonian reference to the $\bar{u}$ direction (scalar)

$$(a_{Lu})_{FILTER} = 0$$
$$(F_{Lu})_{FILTER} = \lambda_{Lu} W_L$$
$$W_L = (F_{Lu})_{Filter} / \lambda_{Lu} \tag{3-9}$$

Substituting this result into Eq. (3-8), the load forces can be characterized by measuring a component of the load force and measuring the load acceleration. The force in the mast lift cylinder is the component of the load force in the direction parallel to the axis of this cylinder. Measuring the lift cylinder pressure will provide the needed information.

In order to determine $\overline{P}_D \times \Delta\overline{F}_M$, assume:

$F_{TX} = 0$	i.e., no side load on tilt cylinders
$Y_D = Z_D \approx 0$	result of vehicle geometry
$Y_M = 0$	mast rotates about front axle
$X_M = X_T$	result of vehicle geometry

Then, in Eq. (3-6), let

$$\overline{P}_A = \frac{1}{2}\overline{P}_m$$

$$(\overline{P}_L - \frac{1}{2}\overline{P}_m) \times \overline{F}_L = (\overline{P}_m - \overline{P}_T) \times \overline{F}_T + \frac{1}{2}(\overline{P}_m - \overline{P}_m) \times \overline{F}_L - \overline{P}_D \times \overline{\Delta F}_m$$

$$(\overline{P}_L - \frac{1}{2}\overline{P}_m) \times \overline{F}_L = \begin{bmatrix} (Y_m - Y_T)F_{Tz} - (Z_m - Z_T)F_{Ty} \\ 0 \\ 0 \end{bmatrix} - \begin{bmatrix} 0 \\ -X_D \Delta F_{mz} \\ X_D \Delta F_{my} \end{bmatrix} \quad (3\text{-}10)$$

Let

$$\overline{m}_{cpy} = \begin{bmatrix} 0 \\ -X_D \Delta F_{mz} \\ X_D \Delta F_{my} \end{bmatrix} = \overline{P}_D \times \overline{\Delta F}_m \quad (3\text{-}11)$$

The left side of Eq. (3-10) is the moment caused by load about the point 1/2 $\overline{P}_M$. This point is on the axis of rotation of the mast (the front axle) midway between the two attachment points $\overline{P}_{M1}$ and $\overline{P}_{M2}$. This equation describes the tendency of the mast assembly to rotate about the point 1/2 $\overline{P}_M$. The term $\overline{P}_D \times \Delta\overline{F}_M$ then describes the tendency of the mast assembly to rotate about axes which pass through 1/2 $\overline{P}_M$ and are parallel to the Y and Z coordinate axes. $\overline{P}_D \times \Delta\overline{F}_M$ can be determined by measuring the tendency of the mast assembly to rotate about these two axes. This can be accomplished by measuring the strain in the mast uprights which results from the bending moment on these uprights.

Equation (3-7) can be used to obtain an estimate of the load moment $\overline{LM}^*$ using the above information. This can be incorporated in the determination of $\overline{M}^*$, but it is also necessary to consider the stabilizing moment. The vehicle force ($\overline{F}_V$) is expanded into the components in Eq. (3-12).

$$\overline{F}_V = W_V \overline{\lambda}_V + m_V \overline{a}_V \tag{3-12}$$

where:

W_V = vehicle weight

$\overline{\lambda}_V$ = direction cosine vector to resolve weight force vector into vehicle reference components

M_V = vehicle mass

$\overline{a}_v$ = vehicle accelerations

Therefore, Eqs. (3-7), (3-8), (3-9), (3-11) and (3-12) can be used to obtain M*.

$$\overline{M}_o^* = \overline{P}_v \times (\omega_v \overline{\lambda}_v + m_v \overline{a}_v) + (\overline{P}_m - \overline{P}_T) \times \overline{F}_T + \frac{1}{2} \overline{P}_m \times (\omega_L \overline{\lambda}_L + m_L \overline{a}_L) - \overline{m_{cp}} \tag{3-13}$$

where

$$\omega_L = (F_{Lu})_{Filter} / \lambda_{Lu} = mg$$

The use of Eq. (3-12) will require the measurement of the following variables: vehicle orientation, vehicle acclerations, tilt cylinder force, (pressure), load orientation ($\overline{\lambda}v$), load accelerations, mast bending moment, and lift cylinder force (pressure). All of these parameters are readily measured using conventional transducers. Several of the parameters are related. The load orientation and vehicle orientation are certainly related, as are the load and vehicle accelerations, which could reduce the number of input variables. Restricting the application of this equation may further reduce the number of measured variables. In addition, it is feasible to eliminate the acceleration measurements by determining maximum expected accelerations and then perform a "worst-case" analysis to determine the loci of operating conditions resulting in a safe operation.

CHAPTER 4

STABILITY CONTROL

This chapter proposes two possible stability controllers which utilize the methodology developed in the previous chapter. A version of one of these controllers is considered in detail. This system was implemented in order to evaluate the feasibility of the concept. The results of this are discussed. The restriction of the concept is discussed.

PROPOSED CONTROLLERS

One possible stability controller is shown in Fig. 4-1. This system is an operator-augmented system. An estimate of the relative stability is calculated from measured parameters. This information is communicated to the operator via an audible or visual indication, for the operator to evaluate and alter his inputs to the system based on the relative stability. This type of control is particularly valuable to the inexperienced operator, who, as previously mentioned, is involved in more than a third of the accidents. This system takes advantage of the operator's ability to evaluate the situation and act accordingly. This system is by no means fool-proof. It is highly dependent on the operator's proper use of the information.

Another possible stability control system is shown in Fig. 4-2. This system is not as dependent on the operator's

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Another possible stability control system is shown in Fig. 4-2. This system is not as dependent on the operator's abilities. This system diminishes the operator's capabilities based on the vehicle's relative stability. The operator would be prevented from entering unstable configurations. This could be

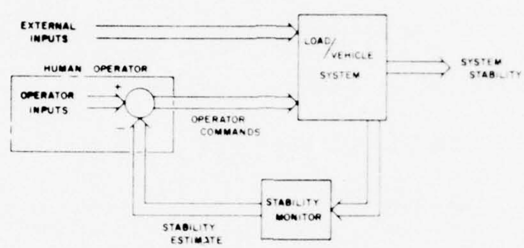

Fig. 4-1. Operator-Augmented Stability Control System

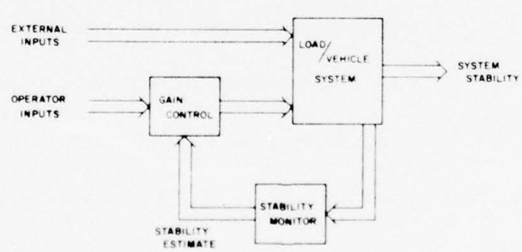

Fig. 4-2. Stability Control Via Diminished Operator Control

implemented by placing valves in the hydraulic circuit for the lift cylinder and the tilt cylinder. As the vehicle approached an unstable configuration, the power capabilities in these circuits would be gradually reduced and finally inhibited. The operator would be prevented from continuing to operate the vehicle in the unstable condition. The operator would be allowed to operate the vehicle to lessen the instability (for example, lower the mast if appropriate).

Neither of these systems provides any corrective control. The ramifications of the correcting actions are difficult to evaluate for all possible situations. The corrective control is deemed undesirable.

IMPELEMENTATION OF A STABILITY CONTROLLER

Consider the implementation of the human-augmented controller in order to explore the feasibility of the methodology developed. For this investigation, only operation on horizontal surfaces will be considered. Vehicle and load accelerations will not be considered. Therefore, this system will be considering only statically caused instabilities. (Although dynamically caused instabilities are not considered, their effect could be incorporated into the system using the worst-case analysis discussed in Chapter 3. This is beyond the scope of this study.)

Equation (3-13) was derived to provide an estimate of $\overline{M}$ and is repeated in Eq. (4-1).

$$\overline{M}_o^* = \overline{P}_v \times (\omega_v \overline{\lambda}_v + m_v \overline{a}_v) + (\overline{P}_m - \overline{P}_T) \times \overline{F}_T + \frac{1}{2} \overline{P}_m \times (\omega_L \overline{\lambda}_L + m_L \overline{a}_L) - \overline{m}_{cp} \tag{4-1}$$

From the above restrictions,

$$\bar{a}_V = \bar{a}_L = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \qquad \lambda_V = \lambda_L = \begin{bmatrix} 0 \\ 0 \\ -1 \end{bmatrix}$$

then,

$$M_o^* = \bar{P}_V \times \left(\omega_V \begin{bmatrix} 0 \\ 0 \\ -1 \end{bmatrix} \right) + (\bar{P}_m - \bar{P}_T) \times \bar{F}_T + \tfrac{1}{2} \bar{P}_m \times \left(\omega_L \begin{bmatrix} 0 \\ 0 \\ -1 \end{bmatrix} \right) - \bar{m}_{CP}$$

$$\bar{F}_T = (P_1 A_1 - P_2 A_2) \begin{bmatrix} 0 \\ \cos \theta_T \\ \sin \theta_T \end{bmatrix}$$

where

P_1 = Pressure on rod side of tilt cylinder

P_2 = Pressure on head side of tilt cylinder

A_1 = Area of rod size

A_2 = Area of head side

θ_T = Tilt cylinder angle (referenced to horizontal)

The tilt cylinder angle can be found by determining a relationship between it and the mast angle and then measuring the mast angle.

$$\theta_T = f(\theta_m)$$

θ_m = mast angle

$$W_L = (F_{L\,filter}) / \cos\theta_m = P_{Lift} A_{Lift} / \cos\theta_m$$

where

P_{LIFT} = Lift cylinder pressure

A_{LIFT} = Effective lift cylinder area

It was initially assumed that the load included the mast assembly, but this fact has not been previously considered. The mast assembly is typically comprised of two sets of uprights -- one set is elevated by the lift cylinder and the other is "stationary" to serve as guides. The weight of the elevating uprights is reflected in the lift cylinder pressure measurement. In order to account for the weight of the *stationary uprights* (W_{MS}), *the weight* of the load will be altered as follows:

$$W_L = P_{Lift} A_{Lift} / \cos\theta_m + W_{ms}$$

$$W_L = W_{mm} + W_{ms}$$

where

$$W_{mm} = P_{Lift} A_{Lift} / \cos\theta_m$$

For a typical vehicle

Y_M = 0

Y_D = 0

Z_D = 0

$X_M - X_T = 0$

It can be shown that

$$\Delta F_{MX} = 0$$

$$\Delta F_{MY} = 0$$

Then:

$$\overline{M}_o^* = \begin{bmatrix} X_V \\ Y_V \\ Z_V \end{bmatrix} \times \begin{bmatrix} 0 \\ 0 \\ -W_V \end{bmatrix} + \begin{bmatrix} X_m - X_T \\ Y_m - Y_T \\ Z_m - Z_T \end{bmatrix} \times \begin{bmatrix} 0 \\ F_{TY} \\ F_{TZ} \end{bmatrix} + \frac{1}{2} \begin{bmatrix} X_m \\ Y_m \\ Z_m \end{bmatrix} \times \begin{bmatrix} 0 \\ 0 \\ -W_L \end{bmatrix}$$

$$- \begin{bmatrix} X_D \\ Y_D \\ Z_D \end{bmatrix} \times \begin{bmatrix} \Delta F_{mx} \\ \Delta F_{mY} \\ \Delta F_{mZ} \end{bmatrix}$$

$$\overline{M}_o^* = \begin{bmatrix} X_V \\ Y_V \\ Z_V \end{bmatrix} \times \begin{bmatrix} 0 \\ 0 \\ -W_V \end{bmatrix} + \begin{bmatrix} 0 \\ -Y_T \\ Z_m - Y_T \end{bmatrix} \times \begin{bmatrix} 0 \\ F_{TY} \\ F_{TZ} \end{bmatrix}$$

$$+ \frac{1}{2} \begin{bmatrix} X_m \\ 0 \\ Z_m \end{bmatrix} \times \begin{bmatrix} 0 \\ 0 \\ -W_m - W_{sm} \end{bmatrix} + \begin{bmatrix} X_D \\ 0 \\ 0 \end{bmatrix} \times \begin{bmatrix} 0 \\ 0 \\ \Delta F_{mz} \end{bmatrix}$$

$$\overline{M}^*_o = \begin{bmatrix} -Y_v W_v \\ X_v W_v \\ 0 \end{bmatrix} + \begin{bmatrix} -Y_T F_{TZ} - (Z_m - Z_T) F_{TY} \\ 0 \\ 0 \end{bmatrix} + \frac{1}{2} \begin{bmatrix} 0 \\ X_m (W_{mm} + W_{sm}) \\ 0 \end{bmatrix} - \begin{bmatrix} 0 \\ -X_D \Delta F_{mz} \\ 0 \end{bmatrix}$$

$$\overline{M}^*_o = \begin{bmatrix} -Y_v W_v \\ X_v W_v + \frac{1}{2} X_m W_{sm} \\ 0 \end{bmatrix} + \begin{bmatrix} -Y_T F_{TZ} - (Z_m - Z_T) F_{TY} \\ 0 \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{2} X_m W_m \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ X_D \Delta F_{mz} \\ 0 \end{bmatrix}$$

This equation, in conjunction with equations previously derived, defines calculations which must be performed using either analog or digital techniques. The analog approach is a continuous control system. The digital approach results in a sample data control system. Each of these has advantages and disadvantages.

Consider implementing the stability controller with a microcomputer-based hardware system. Fig. 4-3 shows the relationship of the major components of the system. Pressures are measured via conventional pressure transducers. The mast angle is measured via a pendulum-type transducer. The mast bending moment is measured via a strain gauge bridge. An active gauge is mounted on each side of the mast stationary upright orientated such that bending moment from a side load is measured. These gauges are connected in one leg of the bridge. (This is commonly known as a cantilever configuration). Temperature compensating gauges are connected in the other leg.

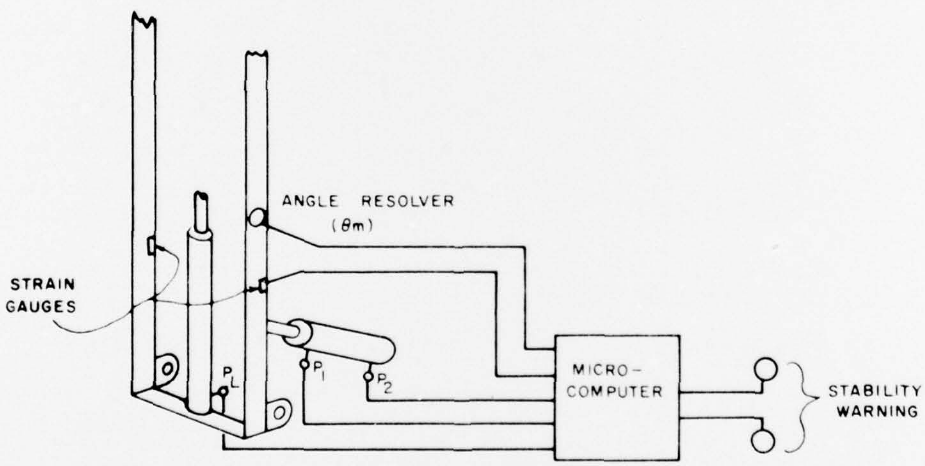

Fig. 4-3. Controller Components

The microcomputer would measure the input variables and perform calculations based on the previously derived equations. A flow chart for this, along with the equations, is shown in Fig.

4-4. The stability decision requires the microcomputer to compare S_{pitch}, S_{LR}, and S_{RR} to predetermined limits. If any of these is less than the limit, then the operator is warned.

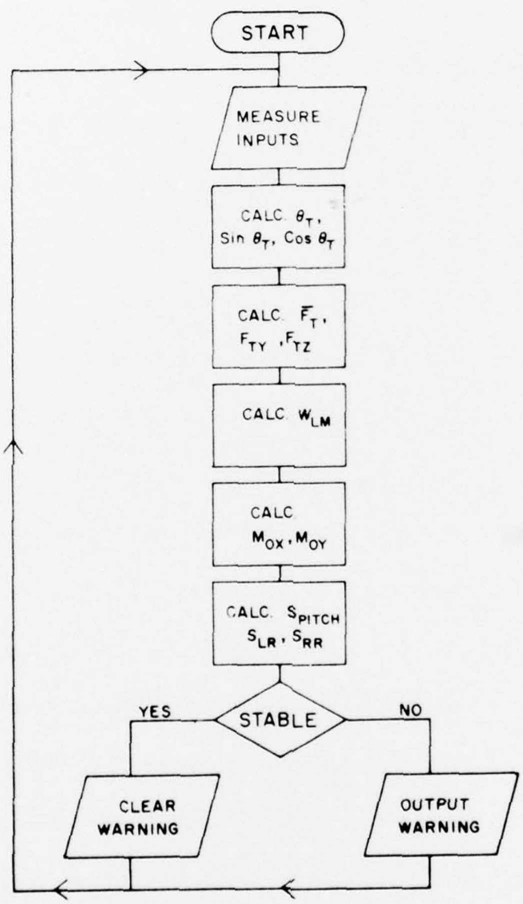

Fig. 4-4. Flow chart for Monitor/Controller

An exploratory system was designed and built using a four-bit microcomputer. The system was installed on a 5,000 pound warehouse forklift. Tests were performed on the system which simulated typical stacking operations. The test results showed that device could accurately detect potentially dangerous conditions. The limits were set such that the warning sounded prior to encountering the instability. As the vehicle approaches an unstable configuration, the alarm sounds to warn the operator of potential danger. It is not sufficient to simply warn when the vehicle is unstable.

During the testing, various parameters which were calculated by the microcomputer were monitored. These included S_{pitch}, S_{LR}, S_{RR}. Several of the tests involved applying a load to the vehicle causing an unstable condition and allowing the system to remain in an equilibrium condition. A value of zero should be calculated for S_{pitch}, S_{LR}, and S_{RR}. Although it is not important that these values are actually calculated to be zero,* the repeatability of measurements and relative difference between tests are extremely important. The repeatability of readings was found to be good. Comparisons between tests were found to be good in all cases except one. When a dead load exceeding the capacity of the vehicle was applied to just one fork and the vehicle attempted a lift against this load until the vehicle became unstable,

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* The values which represent various vehicle parameters affect the actual calculated results. No attempt was made to establish these parameters other than to obtain approximately correct results.

extreme bending of both mast and forks occurred. Although the device determined the correct indication (relative stability decreased and warning sounded), comparison of this test to others revealed substantial differences in the value calculated for S_{pitch}. This is certainly a direct result of assumptions made in modeling the mast assembly.

SYSTEM RESTRICTIONS

It is important to understand the limitations of the stability monitor/controller. The exploratory system assumed a horizontal operating plane and only statically caused instabilities. These assumptions were made only to limit the scope of the exploratory system and are not a general restriction on the system. Additional measurements and analysis are required to add these capabilities.

The theory utilized pressure measurements to ascertain forces. Difficulties are encountered when cylinders with mechanical stops are used. When the cylinder is against its stop, the pressure measurement results in inferring an erroneous force measurement. The cylinders should have hydraulic stops, or the system should incorporate the logic to take this into account.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

This study investigated methods of monitoring and/or controlling the stability of material handling equipment and demonstrated the feasibility via an exploratory system. It was found that such a device is feasible. Two methods have been proposed to control the vehicle stability. Although external influences prevent the system from being completely controllable, the use of either of these two systems certainly could improve the safety of MHE.

The exploratory system was designed and built to demonstrate the feasibility of the concept (see Phase II report). It is realistic to optimize the system to satisfy the evaluation criteria previously stated. Although the exploratory system is quite slow, it is well within the state-of-the art to design a similar device with an update rate of 0.1 seconds. This, combined with sufficiently conservative stability decisions, could result in a device which responds to dangerous situations with ample time for the operator to correct the situation. Neither system interferes with normal, safe operation of the vehicle.

The microcomputer-based system allows for a low-cost, small system which could be installed on a variety of vehicles. Utilization of a device known as a strain link [8] could result in a system which is easily installed and maintained.

Further investigation in the effect of dynamically caused instabilities is recommended. Evaluation of the worst-case analysis proposed should be performed. This should be tested both by computer simulations and by lab testing. In addition, augmentation of the stability decision based on parameters such as vehicle velocity may be required for dynamically caused instabilities.

It is also recommended that a sensitivity analysis be performed on all equations. The assumption of nominal vehicle parameters could result in potential inaccuracy. Changes in axes of rotation for various loadings should be investigated.

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APPENDIX A

In order to determine $\bar{R}_5$ and $\bar{M}_5$, consider the free body diagram in Fig. A-1. This sytem is in dynamic equilibrium. Then,

$$\Sigma \bar{F} = \bar{R}_3 + \bar{R}_4 - \bar{R}_5 = \bar{O}$$

$$\bar{R}_5 = \bar{R}_3 + \bar{R}_4 \qquad \text{(A-1)}$$

$$\Sigma \bar{M}_{P5} = \bar{O} = (\bar{P}_{R4} - \bar{P}_{R5}) \times \bar{R}_4 + (\bar{P}_{R3} - \bar{P}_{R5}) \times \bar{R}_3 + (\bar{P}_{R5} - \bar{P}_{R5}) \times \bar{R}_5 + \bar{m}_{R5}$$

$$\bar{m}_{R5} = -(\bar{P}_{R4} - \bar{P}_{R5}) \times \bar{R}_4 - (\bar{P}_{R3} - \bar{P}_{R5}) \times \bar{R}_3 \qquad \text{(A-2)}$$

Equations (A-1) and (A-2) serve to define $\bar{R}_5$ and $\bar{M}_{R5}$. The articulating axle results in M_{R5Y} equal to zero. Expanding (A-2),

$$Y_{R3} = Y_{R4} = Y_{R5} \qquad Z_{R3} - Z_{R5} = Z_{R4} - Z_{R5}$$

$$Z_{R3} = Z_{R4}$$

$$\bar{m}_5 = -\begin{bmatrix} X_{R3} - X_{R5} \\ O \\ Z_{R3} - Z_{R5} \end{bmatrix} \times \begin{bmatrix} R_{3X} \\ R_{3Y} \\ R_{3Z} \end{bmatrix} - \begin{bmatrix} X_{R4} - X_{R5} \\ O \\ Z_{R4} - Z_{R5} \end{bmatrix} \times \begin{bmatrix} R_{4X} \\ R_{4Y} \\ R_{4Z} \end{bmatrix}$$

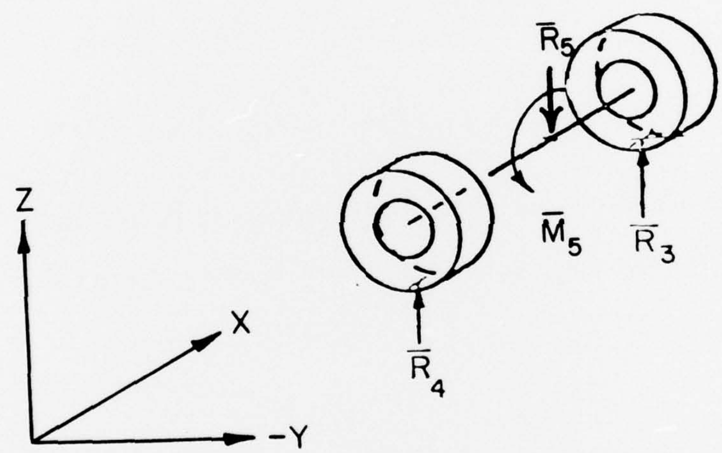

Fig. A-1. Axle Configuration

$$\overline{m}_5 = -\begin{bmatrix} -(Z_{R3} - Z_{R5})\, R_{3y} \\ 0 \\ (X_{R3} - X_{R5})\, R_{3Y} \end{bmatrix} - \begin{bmatrix} -(Z_{R4} - Z_{R5})\, R_{4Y} \\ 0 \\ (X_{R4} - X_{R5})\, R_{4Y} \end{bmatrix}$$

$$\overline{m}_5 = \begin{bmatrix} (Z_{R3} - Z_{R5})(R_{3Y} + R_{4Y}) \\ 0 \\ -(X_{R3} - X_{R5})(R_{3Y} + R_{4Y}) \end{bmatrix}$$

Therefore, $\overline{M}_{R5}$ is only a function of the y components of $\overline{R}_3$ and $\overline{R}_4$. These components only exist during braking. M_{R5X} is a function of the total braking force while M_{R5Z} is a function of the braking imbalance.

In evaluating the significance of $\overline{M}_{R5}$ in the equation for $\overline{S}$, it should be remembered that $\overline{M}_{R5}$ is combined with a direction cosine vector whose dominant component is the y component.