An Experimental Investigation of Proportional Braking

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AN EXPERIMENTAL INVESTIGATION OF
PROPORTIONAL BRAKING

A Thesis
Presented to the
Department of Mechanical Engineering Science
Brigham Young University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Rudolf Limpert
August 1968
This thesis, by Rudolf Limpert, is accepted in its present form by the Department of Mechanical Engineering Science of Brigham Young University as satisfying the thesis requirement for the degree of Master of Science.

6 Aug 1968
Date
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NOMENCLATURE

\[ A_{WC} \] wheel cylinder area
\[ a \] deceleration ratio
\[ a/f \] utilization of friction
\[ b \] deceleration in \( \text{ft/sec}^2 \)
\[ C^+ \] brake factor
\[ F \] braking force
\[ F^+ \] non-dimensionalized braking force
\[ f \] road friction factor
\[ g \] gravitational constant
\[ h \] height of center of gravity
\[ l \] wheel base
\[ P_h \] hydraulic pressure
\[ P' \] hydraulic pressure to balance release springs of brake
\[ R \] effective tire radius
\[ r \] effective drum or disk radius
\[ S \] stopping distance
\[ V \] velocity
\[ S \] increase in stopping distance over minimum
\[ \psi \] static rear axle loading ratio
\[ \chi \] center of gravity position ratio
\[ \Phi \] actual braking force distribution
\[ \eta \] hydraulic efficiency

Subscripts
\[ F \] front axle
R  rear axle
h  hydraulic
WC wheel cylinder
id ideal
0 unloaded driving condition (driver only)
1 loaded, test weight 6812 lb
2 loaded, test weight 8602 lb


CHAPTER I

INTRODUCTION

In recent years automobile safety has become more and more important. The braking device contributes largely to the problem of safe driving.

During the braking process some load is shifted from the rear to the front axle. The frictional forces between tire and road surface vary proportionally with the normal forces. Almost all braking systems today consist of devices which distribute the front and rear axle braking forces in a constant ratio. This fixed or constant distribution is selected for a particular driving condition which occurs frequently. For dynamic situations different from the chosen one overbraking of the rear or front axle almost always occurs.

In the case of proportional braking the distribution is made variable according to the variation of the axle loads during the deceleration process to prevent overbraking.

The purpose of this study was to investigate the braking process experimentally for a prototype proportioning system and to compare it to the non-proportional device installed originally in the test vehicle. For this purpose the hydraulic pressure, deceleration, and the skid point were measured for several load conditions and the effectiveness of the braking device computed in terms of the utilization of friction. The results obtained clearly support the idea of proportional braking.
and show in particular that the stopping distance could be decreased by almost 50%.
CHAPTER II

THEORY OF PROPORTIONAL BRAKING

The forces acting on a decelerating vehicle are shown in Figure 1. In this analysis all aerodynamic forces are neglected.

For convenience, the following terms are defined:

- deceleration ratio: \( a = \frac{b}{g} \)
- static rear axle loading ratio: \( \gamma = \frac{W_R}{W} \)
- center of gravity position ratio: \( \chi = \frac{h}{\ell} \)

with deceleration \( b \), gravitational constant \( g \), static rear axle load \( W_R \), vehicle weight \( W \), height of center of gravity \( h \), and wheelbase \( \ell \).

With the above definition and the condition that the deceleration, \( a \), equals the coefficient of friction, \( f \), between tire and road surface, the ideal braking forces, (marked "id"), on the front and rear axle are derived:

\[
F_{FC(id)} = [(1-\gamma) + \chi a] a W \quad (1a)
\]
Fig. 1. -- Forces acting on a decelerating car.
Since the deceleration, \( a \), is set equal to the friction factor, \( f \), the forces \( F_F(\text{id}) \) and \( F_R(\text{id}) \) represent optimum utilization of the given road friction. For example, for \( a > f \), the demanded deceleration is greater than that allowable due to friction and in the case of small deceleration and slippery roads the front axle is overbraked, or in the case of high deceleration the rear axle begins to skid. For \( a < f \), the given road friction is not utilized completely. This results in a stopping distance greater than the minimum possible one. Both cases \( a \neq f \) represent non-ideal braking.

The data obtained from Equation (1) are shown in Figure 2. They are indicated as the curves marked (ideal). For the calculations of this example some average values for \( \psi \) and \( \chi \) have been chosen. In Figure 2 the ideal braking forces are non-dimensionalized by dividing through by the weight of the vehicle. This representation is very convenient since the braking forces on the front \( F^*_F \) and the rear \( F^*_R \) always add up to the total braking force \( (aW)/W = a \). For example, for the unloaded driving condition (driver only) and \( a = 0.6 \), \( F^*_F = 0.41 \) and \( F^*_R = 0.19 \). Considering the ratio 0.19 to 0.6, it is clear that for different decelerations different ratios are obtained for the ideal braking forces. This again shows the importance of a variable brake force distribution. For a fixed distribution only one dynamic condition corresponds to ideal braking.

In order to say something about the effectiveness of a braking
Fig. 2. -- Ideal braking forces, non-proportional, and proportional braking.

\[
F_F^* = \frac{F_{F(id)}}{W}, \quad F_R^* = \frac{F_{R(id)}}{W}
\]
device, the ideal forces are compared to the actual braking forces generated by the installed brakes.

For convenience the distribution \( \Phi \) of the actual braking forces is defined to be:

\[
\Phi = \frac{F_R}{F_F + F_R}
\]

Then for the front axle it follows that the

\[
1 - \Phi = \frac{F_F}{F_F + F_R}
\]

The actual braking forces are computed by the equation

\[
F = 2(p_h - p') \cdot A_{wc} \cdot \eta \cdot c \cdot \frac{r}{R}
\]

where

- \( p_h \) - hydraulic pressure
- \( p' \) - hydraulic pressure to balance release spring
- \( A_{wc} \) - wheel cylinder area
- \( \eta \) - hydraulic efficiency due to losses in wheel cylinder
$C^x$ - brake factor, defined as the sum of the tangential forces on the drum radius divided by the applying force in the wheel cylinder

- effective drum or disk radius
- effective tire radius

For small $p'$, non-proportional braking, and equal tire radius, the actual distribution is computed by the components usually altered between front and rear axle:

$$\Phi = \frac{[A_{wc} \cdot C^x \cdot r]_R}{[A_{wc} \cdot C^x r]_F + [A_{wc} \cdot C^x y]_R}$$

In Figure 2 the ideal braking forces are compared to the actual distribution $\Phi$. A typical value $\Phi = 0.3$ was used in the example. It can be seen that for the unloaded driving condition an overbraking of the front axle will occur for $a < 0.68$, whereas the rear axle will be overbraked for $a > 0.68$. For the loaded driving condition the front axle is always overbraked because the friction factor is always less than 1.0 for tire on pavement. The point of overbraking is determined through the intersection of $\Phi$ with the ideal braking forces.

To compare the actual braking device to the ideal one possible, i.e., the one with the minimum stopping distance, the utilization of the friction factor, $a/f$, is introduced. Here again, $a$ is the deceleration and, $f$ is the road friction factor.
The necessary friction factor $f_R$ on the rear axle which guarantees non-skidding can be computed by

$$f_R = \frac{\Phi \cdot a \cdot W}{[\gamma - \chi a]W} = \frac{\Phi a}{\gamma - \chi a}$$

The expression can be rewritten as

$$f_R \cdot \gamma - f_R \chi a = \Phi a$$

$$f_R \gamma = \Phi a + f \chi a$$

$$= a(\Phi + f \chi)$$

The rear utilization is thus

$$(a/f)_R = \frac{\gamma}{\Phi + f \chi} \quad (4a)$$

The front utilization is derived in a similar way:

$$(a/f)_F = \frac{1 - \gamma}{1 - \Phi - f \chi} \quad (4b)$$

Equation (4) is shown in Figure 3, plotted as a function of the road friction. For example, for $f = 0.42$ the utilization on the front axle $(a/f)_F = 0.9$ for the unloaded driving condition. This means that 90%
Fig. 3. -- Utilization and stopping distance for non-proportional braking.
of the given friction is utilized until the front axle begins to skid. It also means that no deceleration greater than 0.9 times 0.42 (approximately 0.38) can be achieved without overbraking of the front wheels. For the loaded condition \((a/f)_F = 0.57\) for \(f = 0.42\). This gives a possible non-skid deceleration of about 0.24.

The object of proportional braking is to bring the utilization values closer to "one" over a wide range of road conditions, that is winter and summer driving. This is done by employing a variable braking force distribution. Through a proportioning valve the actual braking forces are brought closer to the ideal ones. A two-slope distribution is shown in Figure 2. For \(a < 0.4\) the hydraulic rear pressure is not proportioned relative to the front axle. For \(a > 0.4\) the rear pressure is proportioned in such a way that an overbraking of the rear axle is prevented for all decelerations less than 0.9. In order to approximate the different load conditions between unloaded and loaded, the proportioning valve is designed in such a manner that the displacement of the body of the vehicle relative to the rear axle controls the shifting point of the valve. Since the mentioned displacement is a function of the rear axle load, the proportional system is made automatically load-dependent. This means that in Figure 2 the shifting point is moved between A and B according to the current rear axle load. The improvement over the fixed distribution is shown in Figure 4. The utilization is always greater than 0.80 for road conditions, giving \(f\) between 0.2 and 1.0.

The ideal stopping distance of a vehicle is the minimum possible. It is a function of the road conditions. Assuming the parameters \(a\) and \(f\) to be constant, the ideal stopping distance is computed by
Fig. 4. -- Utilization and stopping distance for proportional braking.
where \( V \) = velocity, \( g \) = gravitational constant, and \( f \) = friction factor.

Let the stopping distance due to the actual distribution be

\[
S = \frac{V^2}{2ga}
\]

where \( a \) = deceleration.

Then the difference between the possible and the ideal stopping distance is

\[
\Delta S = S - S_{id} = \frac{V^2}{2ga} - \frac{V^2}{2gf} = \frac{V^2}{2gf} \left( \frac{f}{a} - 1 \right)
\]

In terms of the utilization \( a/f \) the increase in stopping distance over the ideal one is

\[
\frac{\Delta S}{S_{id}} = \frac{f}{a} - 1 = \frac{1-(a/f)}{(a/f)} \times 100[\%] \quad (5)
\]

As can be seen from Equation (5), the increase in stopping distance becomes smaller with increasing utilization and is equal to zero for
a/f = 1.0. The left-hand side of Figures 3 and 4 show the graph of the functional relationship between stopping distance and utilization.
CHAPTER III

DESIGN OF PROPORTIONAL SYSTEM

The purpose of a proportional braking device is to vary the rear axle hydraulic brake fluid pressure relative to the front axle pressure. This brings the braking forces generated by the hydraulic pressure closer to the ideal ones, as outlined in Chapter II.

The system designed was not automatically load-dependent, rather it could be adjusted manually for different load conditions. The automatic load-dependence was left out in order to prevent a too great complexity of the braking system. The objective of the experiment was to investigate the braking process which could best be done by eliminating any unnecessary components.

In the design of proportional systems it is convenient to work with the hydraulic pressures directly and not with the forces generated by them. The ideal pressures are those which generate the ideal braking forces. For the test vehicle used they are shown in Figure 5 and marked as indicated (ideal). The ideal pressures are computed by Equation (2) with $F$ replaced by $F_{id}$ (for test vehicle data see Appendix A). It is noteworthy that the pressures are not non-dimensionalized and hence the deceleration parameters have a different scale for different load conditions. This form is more convenient since the hydraulic pressures can be read directly off the graph. In order to prevent an overbraking of the rear axle for all dynamic conditions, a variable distribution as
Fig. 5. -- Ideal hydraulic pressures and braking force distribution.
shown in Figure 5 was chosen. As indicated, the pressure ratio front to rear was 2.5 to 1 and the shifting point B for the loaded driving condition corresponded to a pressure $p_B = 760$ psi. This meant that up to B the front and rear axle pressures were the same whereas for conditions beyond B the front pressure increased 2.5 times that of the rear pressure for the same intervals. The proportioning valve was designed in such a manner that the shifting point could be moved between A and B according to the test load by an adjustment screw.

The hydraulic circuit of the test vehicle is shown in Figure 6. The system consisted of two independent circuits, namely the proportional and the original (non-proportioning) one. For proportional braking and decelerations below point B the hydraulic pressure was not proportioned. In this case the rear pressure was transmitted via the through-cylinder. For proportional braking the proportioning cylinder was activated by the self-disabling of the through-cylinder when the supply pressure exceeded $p_B$.

The proportional cylinder is shown in Figure 7. Basically, it consists of the differential cylinder a, the differential piston b, the spring c, the adjustment nuts d, and e. For non-proportioned pressures, that is for pressures below $p_B$, the rear axle pressure equal to the front axle pressure, exists at location r as well as f. Due to the different size areas of the differential cylinder, the piston b moves to the left until it is balanced by the spring c. In the case of proportional braking the hydraulic pressure at f is greater than at r according to the area ratio and the spring force.

Figure 8 shows the through-cylinder with cylinder a, piston b,
Fig. 6. -- Proportional system of test vehicle.
Fig. 7. -- Proportioning cylinder.
Fig. 8. -- Through-cylinder.
Fig. 9. -- Proportional unit assembled for bench test.
adjustment screw c. During non-proportioning braking the front and rear axle pressures are approximately equal at f and r. The piston b moves to the right until it hits the stop c. From that moment on, the rear axle pressure cannot be increased via the through-cylinder. Before connecting the proportional unit to the test vehicle the system was tested for pressure ratio and shifting point as shown in Figure 9. During the bench test and pressures up to 2500 psi, no leakage was observed.
CHAPTER IV

EXPERIMENTAL PROCEDURE

The experiments were performed on a modified Chevrolet pickup truck as shown in Figure 10 (for test vehicle data see Appendix A). The proportional unit was connected to the test vehicle according to Figure 6, located at the left front end as shown in Figure 11. In order to be able to compute the utilization of friction, the hydraulic pressures front and rear, the deceleration, and the angular velocity (skid point) of a rear wheel were measured electronically, all with respect to time. This made it possible to determine the individual pressures and deceleration at the skid point.

The hydraulic pressures were also measured with two dial gauges so that the driver always had a way of checking the proper functioning of the system. The front and rear pressures were recorded on a two-channel recorder by means of two pressure transducers. Figure 12 shows the location of the pressure instrumentation. The transducers as well as the dial gauges were calibrated against a dead weight tester over a range of 0 to 2000 psi (front) and 0 to 1000 psi (rear). The transducers were excited by batteries located in the switch box shown in the center of Figure 13. The output of the pressure transducers was recorded on the smaller of the two recorders (Figures 13 and 14).

The deceleration of the test vehicle was measured with a strain gage type decelerometer which was rigidly mounted to the vehicle as
Fig. 10. -- Test vehicle loaded with water cells.

Fig. 11. -- Proportional unit mounted to the left front end of test vehicle.
shown in Figure 12. The electric output was balanced by means of a Wheatstone bridge and recorded on the recorder located in the center of Figure 13. Figure 14 shows the location of the Wheatstone bridge along with the recorder.

The angular velocity of the right rear wheel was measured by a small generator mounted to the center of the wheel by means of a frame as shown in Figure 15. The output was recorded on the recorder shown in the center of Figure 13.

The time event markers of the two recorders were connected to the stop light switch of the vehicle thus establishing a common time base.

The recorders were driven through standard electric power obtained by means of a cable towed by the test vehicle as shown in Figure 10.

For each different test the load was varied by adding more water-filled cells to the load as shown in Figure 10. The vehicle data for the different tests were determined by measuring the individual axle loads by means of a scale.
Fig. 12. -- Test vehicle instrumentation with dial gauges a and b (front and rear), pressure transducers c and d, decelerometer e.

Fig. 13. -- Dual channel recorders with pressure recorder a, deceleration and skid point recorder b, and switch box c.
Fig. 14. -- Instrumentation with Wheatstone bridge a.
Fig. 15. -- Generator assembly for measuring angular velocity of rear wheel.
CHAPTER V

RESULTS

As mentioned in Chapter IV, data were obtained for the hydraulic pressures, deceleration, and angular velocity of the rear wheel as a function of time. For the recorder outputs see Appendix C.

The road friction factor was determined to be between 0.72 and 0.75 by pulling the test vehicle with all four wheels skidding and measuring the pulling force.

Data were obtained for three different load conditions. For test vehicle data see Appendix A.

The unloaded driving condition corresponded to a test weight of 4742 lbf. The pressure variation of Run 1, Run 2, and Run 3 is shown in Figure 16. For proportional braking (Runs 1 and 2) no skidding of the rear axle was observed. It is noteworthy that according to the theory outlined in Chapter II in the case of Run 2 the rear axle should always be overbraked. The reason for non-overbraking is that the actual road friction factor (equal to 0.72 to 0.75) was greater than the deceleration $a = 0.65$ and 0.68 for Runs 1 and 2, respectively. Both decelerations were less than the road friction factor. This will be discussed later in Chapter V in terms of the utilization of friction. Run 3 represents the data obtained from testing the original braking system. The point marked, R.S., corresponded to the skid point of the...
Fig. 16. -- Pressure diagram for $W_0 = 4742$ lbf.
Fig. 17. -- Pressure diagram for $W_1 = 6812 \text{ lb}_f$. 

**EXPERIMENT:**

- **RUN 4**
- **RUN 5**
- **RUN 6 (ORIGINAL)**

**IDEAL**

$a = 0.8$

$a = 0.7$

$a = 1.1$

$a = 0.56$

$a = 0.5$

$a = 0.64$

$a = 0.6$

$a = 0.67$

$a = 0.63$

$a = 0.6$
rear axle. The decelerations measured and computed from the hydraulic pressures recorded agree very well. For Run 1 the results were 0.65 and 0.66, for Run 2, 0.68 and 0.74, and for Run 3, 0.68 and 0.78, respectively. The skid point of the original system was measured and computed to be 0.48. It was felt that the agreement was very good because of the fact that in the case of the theoretical analysis the brake factor $C^+$ of the brakes was determined experimentally and checked against the theory (for determination of the brake factor $C^+$ see Appendix A). The $C^+$ value obtained was $4.33 \times 10^2$. This explains the differences of the experimental results to a certain degree.

The second test weight was 6812 lb$_f$. The pressure variations of Runs 4, 5, and 6 are shown in Figure 17. The shifting points for Runs 4 and 5 were 320 and 400 psi, respectively. The decelerations measured and computed were 0.54 and 0.56, 0.60 and 0.64, for Runs 4 and 5, respectively. For the original system the measured and computed decelerations were equal to 0.64.

In the case of the third load condition with 8602 lb$_f$, only one run with proportional braking was performed. The shifting point was moved up to 500 psi. The measured and computed decelerations were 0.44 and 0.50, respectively. The data for the original system were 0.56 and 0.58 for the measured and computed decelerations, respectively.

A more revealing discussion of the results can be done in terms of the utilization of the friction factor.

For the unloaded driving condition the data obtained for the utilization for Runs 1, 2, and 3 are shown in Figure 19. The theoretical
EXPERIMENT

O - RUN 7
D - RUN 8
(CORRECTED)

Fig. 18. -- Pressure diagram for \( W_2 = 8602 \text{ lb} \).
utilizations for the three runs are marked (Theory). The data points computed from the measurements agree very well with the analytical results. As is evident from Figure 19, more road conditions should have been investigated in order to obtain utilization points for winter driving as well. Unfortunately, this was impossible because the climatic conditions were not correct.

For the second load condition with 6812 lbf the results are shown in Figure 20. Here again, the experimental results agree very well with the theory. In the case of Run 5 the theoretical data point is approximately 11% larger than the experimental result. It is felt that this is reasonably close in the light of the factors influencing the analysis.

The utilization points for the loaded driving condition with 8602 lbf are shown in Figure 21. The results for the original system deviated by approximately 13% which is still agreeable. The brake factor $C^+$ alone might vary by 10% which could cause a variation of the braking force distribution $f$ by 20%. In the case of the original system this would alter $f$ from 50% to 40% or 60%. This $f$ variation could have changed the utilization curve obtained for the theory by more than 12%.

From the $\Delta S$-curve in Figure 19, it can be seen that the stopping distance $\Delta S$ was decreased theoretically from 56% over the minimum possible for the original braking system to approximately 6% for proportional braking. These values were taken at a road friction factor of 0.72 and for the unloaded driving condition. Figures 20 and 21 reveal similar results.

For future research it is recommended that the proportional unit
Fig. 19. -- Utilization diagram for $W_0 = 4742$ lb$_x$. 
Fig. 20. -- Utilization diagram for $W_\perp = 6812$ lb.$^\circ$. 
Fig. 21. -- Utilization diagram for $W_2 = 8602$ lb.
be installed in a faster vehicle thus making it possible to obtain longer stopping distances. This would yield an accurate comparison of theoretical and experimental results.
CHAPTER VI

CONCLUSION AND RECOMMENDATIONS

The results obtained support the theory as outlined in Chapter II. This suggests that for designing proportional systems the ideas of ideal hydraulic pressures and utilization of friction are very helpful in determining the components of the braking device.

The experiments have shown that a proportional system, designed in such a way that the actual braking forces are always equal to or a little less than the ideal ones, will bring a considerable improvement of the braking device.

It is recommended that the proportional system designed be used in obtaining data points for road friction factors as low as 0.15. This will allow the determination of the utilization of friction for conditions similar to winter driving.
APPENDIX A

DATA OF TEST VEHICLE

The test vehicle used was a modified 1962 3/4-ton Chevrolet pickup truck. The following table gives the data of the truck.

\[
\begin{align*}
W_0 &= 4724 \text{ lbf}, \gamma_0 = 0.416, x_0 = 0.22 \\
W_1 &= 6812 \text{ lbf}, \gamma_1 = 0.655, x_1 = 0.29 \\
W_2 &= 8602 \text{ lbf}, \gamma_2 = 0.690, x_2 = 0.35
\end{align*}
\]

Wheel base 1 = 127.5 inches.

The data of the original braking system is given below.

- Brakes: 11" x 2-3/4" (type duo-servo)
- Wheel Cylinder: 1-1/8", area = 0.995 square inches
- Brake Factor: \( C^+ = 4.33 \), \( p' = 20 \text{ psi} \)
- Tire Radius: \( R = 15 \text{ inches} \)

Before designing the proportional system, the brake factor \( C^+ \) was determined experimentally by measuring the deceleration and the hydraulic pressure on the front axle for the original braking system and computing the information by use of the equation given below.

\[
C = \frac{W_0 \left[ 1 - (\gamma - a \chi^2) \right]}{2 (A \nu \gamma/R) \rho n} = \frac{4995(1.5)\left[ 1 - (46 - (0.5)(0.235) \right]}{2 (0.995) \cdot 5.5/15 \cdot 600} = 4.33
\]
Fig. 22. -- Brake factor $C^*$. 
The brake factor was also determined analytically. The relationship between the brake factor $C^+$ and the friction factor $\mu$ of lining and drum is shown in Figure 22. The value for $C^+$ obtained experimentally corresponds to a friction factor $\mu = 0.39$ which is a typical value for brake lining.
APPENDIX B

INSTRUMENTATION DATA

The following table lists the instruments used during the experiment.

Oscillograph, Model BL-262, Brush Electronics Company

Dual Channel D.C. Amplifier, Model RD 562101, Brush Electronics Company

Mark 220 Recorder, Brush Electronics Company

Ellis Wheatstone bridge

Pressure transducer, Type 1743, range 0 to 3500 psi, Trans-Sonics

Pressure transducer, Type 46155-AC-A-100-75, range 0 to 1000 psi, General Dynamics

Decelerometer (strain gage type), Type C-5-225, range ± 5 g, Statham Laboratories

7 volt D.C. generator, Electro-Tek Products Co., Inc.
APPENDIX C

RECORER OUTPUT

Appendix C contains the outputs for hydraulic pressures front and rear, skid point, and deceleration as recorded on the dual recorders.

The measurements are shown in the following order from the top of the page to the bottom.

Angular velocity: 1 line = 1 mph
Deceleration: 25 lines = 1 g
Hydraulic pressure: 25 lines = 1000 psi
(on the front axle)
Hydraulic pressure: 25 lines = 1000 psi
(on the rear axle)

The paper speed was 25 mm per second for both recorders.
Fig. 23. -- Recorder output, Run 1.
Fig. 24. — Recorder output, Run 2.
Fig. 25. -- Recorder output, Run 3.
Fig. 26. — Recorder output, Run 4.
Fig. 27. -- Recorder output, Run 5.
Fig. 28. -- Recorder output, Run 6.
Fig. 29. -- Recorder output, Run 7.
Fig. 30. -- Recorder output, Run 8.
LIST OF REFERENCES


AN EXPERIMENTAL INVESTIGATION OF
PROPORTIONAL BRAKING

An Abstract of a Thesis
Presented to
The Department of Mechanical Engineering Science
Brigham Young University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Rudolf Limpert
August 1968
ABSTRACT

The purpose of this thesis was to investigate the braking process of a vehicle experimentally for proportional braking and to compare it to the theory. The results obtained are in good agreement with the analytical predictions. From the investigation it is clear that proportional braking devices have an advantage over systems with fixed brake force distribution since a utilization of road friction of about 90% can be achieved over a wide range of road conditions, depending on the type of proportioning valve used.

APPROVED: